THE DESIGN OF MACHINERY FOUNDATIONS

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David Owen Van Strien

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THESIS

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INTRODUCTION

All too often the design of machine foundations is left to chance instead of to science.

It is the job of the structural engineer and architect to design a building with walls, floors, and roof capable of supporting the loads that will be imposed upon it and also will house the machinery which is required. The mechanical engineer in turn very precisely designs these machines to as efficiently as possible carry out their intended job. The plant lay-out man arranges the machines in the completed building, taking great care that the flow of material is as efficient as possible as it passes from machine to machine. No one however, gives much scientific thought to the design of the foundation which must support the machine. This is often true because the person in authority does not realize the extremely important part that an adequate foundation may play in the operation of the entire plant.

It is astounding to note the lack of written information on the subject of machine foundation design. Unfortunately, it has too often been the practice to place a block of concrete in the ground, set a machine on it, and, because, as luck would have it, the concrete was massive enough, the soil was good, and nothing of a delicate character was around to shake, the foundation was a success. When the installation of another foundation, for a similar machine, was required, the original drawings were used again. This time, perhaps, the soil was not so good and the result was excessive vibration. A third foundation was required, so remembering the unhappy experience of number two, the foundation size was increased to be "safe". It seems evident that the evolution of foundation design so often has followed this process and that seldom has an evaluation of the items envolved been made.

It might be observed that the manufacturers of the machinery should be the ones to give advise as to the foundation requirements for their machine. In many cases they may give valuable aid, however, more often they will evade the responsibility and give such advise as one maker of engines gave: "make the foundation large enough to encompass the anchor bolts and deep enough to carry down to supporting soil, and the mass of concrete will be sufficient". You can see that this leaves much to be desired, especially when it is considered that the engine is one having unbalanced inertia forces. As often as not the machine for which you are designing a foundation is an old one for which no design data exists.

It is to familiarize myself with the items involved in the complete and successful design of a machine foundation that I have undertaken this investigation.

Each different type of machine offers an individual problem to the foundation designer. Some machines require only provisions for the distribution of their static weight upon the supporting medium. Other types of machinery require provisions to handle forces which arise from the motion of the moving parts or from gear thrust or belt pull. In some types of machinery a provision for the damping or isolation of vibration is of paramount importance. Thus, a treatise of this kind cannot possibly cover adequately the complete field of machinery foundation design. As a result only a few of what I consider the most important features of design have been included with some accompanying examples, only two of which represent complete designs.

It is admitted that the dynamic forces in some machines are so complicated that they cannot be determined mathematically. In these cases the foundations must be designed by empirical methods based on past successes. These things are pointed out in the following pages.

As a preliminary to the problem at hand it might be well to list the various purposes for which a machine foundation is required:

1. A foundation is required to maintain the machinery fast in position and level with all parts in true alignment.
2. A foundation is required to transmit the dead weight of the machine to the ground or other supporting medium in such a manner that the safe bearing pressure of the ground or other supporting medium is not exceeded, thereby, preventing settlement or deformation of the supports.

3. A foundation may be required to transmit the live load of the machine to the ground or supporting medium.

4. A foundation may be required to overcome dynamically the effects of free inertia forces and couples by balancing such kinetic reactions within the foundation itself so as to cause them to wholly or partially cancel.

5. A foundation may be required to prevent sychronism with adjoining machines or structures, by proper location or distribution of loads, by correct proportioning of structural members, or by isolation.

6. A foundation is required to absorb as far as possible the residual vibrations set up in the machinery so as to transmit as little as possible to the surrounding ground.

According to the type of machine and the nature of its supporting medium, one or the other of the above requirements will assume primary importance. Thus in the case of an electric motor generating set, leveling and alignment are of more importance than the absorption of vibration. While in considering a suitable foundation for a diesel engine, the problem would certainly involve a study of the dynamic forces involved and a consideration of the vibrations if any. Certainly a heavier foundation would result than for the electric motor.

II. THE DISTRIBUTION OF LOADS TO THE SUPPORTING MEDIUM

Various requirements of a machine foundation have been stated in the previous article. A discussion of these requirements follows in detail, beginning with the problems incurring in the distribution of the loads to the supporting soil.

It seems in order, before we discuss just how the loads are transmitted to the ground, that some time be spent on the capability of the soil to support the imposed loads.

A-1 Safe soil pressures

It is obvious that different soils and soil conditions will have different safe bearing pressures, and that any empirical rules regarding these have to be followed with caution. It is with this word of warning that Table I is offered. This table gives the safe bearing powers of various soils as advised by two different authorities. These values have been established by experimentation and observation. Note that Mr. Crofts' figures are similar to those given by the New York City Building code. For machine foundations having vibration it appears that he has cut his figures in half. In regard to the table, Mr. Croft points out that by "non vibrating"machinery he refers to practically no vibration, such as motor generators, sychronous condensers, smoothly running motors and nicely running steam turbines.**

Most machines with excessive vibrations are set on foundations which rest directly on the soil. It is common knowledge that a portion of the soil will vibrate with the foundation. The soil must, of course, support the machine and foundation with a certain minimum deflection, but another extremely important factor to consider is that the vibrations occuring must not be transmitted thru the soil to the detriment of other machines or structures.

As one can see, it is imperative that the designer know something about the soil upon which the machine is to be supported. Part of the answer is to run static soil tests on the proposed soil. Thus, bearing pressure values might be obtained which multiplied by a safety factor, will at least give us an allowable which if not exceeded will provide for the first aforementioned requirement.

Something more, however, must be known about the soil before we can be sure that it will not transmit vibrations from or to the machine. A trial and error method is sometimes rather an expensive way to arrive at this information.

**Terazagi, an eminent soils engineer has this to say about soil loading when they are not static: "For vibrating machines and machines with noticable dynamic loadings a factor of safety must be included when using soil pressure tables, to take care of the uncertainity of the reactions of various soils to dynamic loads." He does not, however, suggest what safety factors to use.

It must be understood that in dealing with these things we are treading on a more or less unexplored ground (on vibration in soils). Studies have been made, most of them in Europe, and are now proceeding but much information must still be left to chance.

It is well known that water-logged or wet soil is prone to transmit vibrations readily. There are many instances reported in engineering literature where excessive ground water has increased the transmission of vibrations thru wet sand and to a lesser extent thru well saturated clay. I have noticed this fact to be true when using a surveying instrument along a railroad track. When a train passes, the vibrations are quickly damped out by a dry well drained ballast or sub-grade, but if the ground is swampy and badly drained,the instrument will vibrate excessively, and settle from its original position.

The fundamental principle which should be deduced from such instances is that to help prevent the transmission of vibrations thru the soils from or to the foundation the ground water should be lowered below that of the foundation walls and footings, by providing adequate drainage.

Of course, the above provision will not keep all soils from vibrating excessively, under all frequences of vibrations. In many instances, however, it will make a sufficient difference so that additional provisions need not be considered.

If it doesn't the next step must be either to change the operating frequency of the machine or to isolate the foundation from the soil in some way.

TABLE I

$\#/ft.^2$ by Terrel Croft "Machinery Foundations and Erection"

Kind of Soil		For Vibrating Machines	For Non-Vibrating Machines
Rock -	The hardest in thick layers in natural bed	40,000	80,000
	Equal to best ashlar masonary	25,000	50,000
	Equal to best brick masonary	15,000	30,000
	Equal to poor brick masonary	5,000	10,000
Clay -	In thick beds, always dry	4,000	8,000
	In thick beds moderately dry	2,000	4,000
	Soft	1,000	2,000
Hardpan		8,000	16,000
Gravel		6,000	12,000
Sand -	Dry, compact and well cemented	4,000	8,000
	Clean and dry	2,000	4,000
	Wet Sand	2,000	4,000
Sand and Clay mixed or in layers		2,000	4,000
Quicksand, alluvial soils		500	1,000

New York City Building Code - 1936

Kind of Soil	$\#/ft.^2$
Hard Rock	80,000
Medium Rock	30,000
Soft Rock	16,000
Hard dry clay	8,000
Firm clay	4,000
Wet clay	2,000
Hardpan	20,000
Gravel	12,000
Coarse Sand	8,000
Fine and Dry Sand	6,000
Wet Sand	4,000
Sand Clay Mix	4,000

A-2. Foundations in which the axis of loads passes through the center of gravity of the area of the base.

The first type of machine foundation which we will take up will be one of the simplest. The axis of loads passes through the center of the area of the base and the loads which must be considered are static, and the problem is not complicated by belt pull or gear thrust as would be the case with an electric motor which usually drives something. A good example of this type of foundation would be a syncronous converter. Illustrated in figure 1.

Requirements of this type of foundation:

1. The size must be sufficient to include the anchor bolts.

2. Area of the base must be great enough to transmit the dead load weight of the machine plus the weight of foundation to the ground without exceeding its safe bearing pressure.

3. The foundation must support the machine in its desired position. If in an exposed position it must extend below the frost line. Very often the foundation can be made hollow to provide space inside for appertenances to the machine.

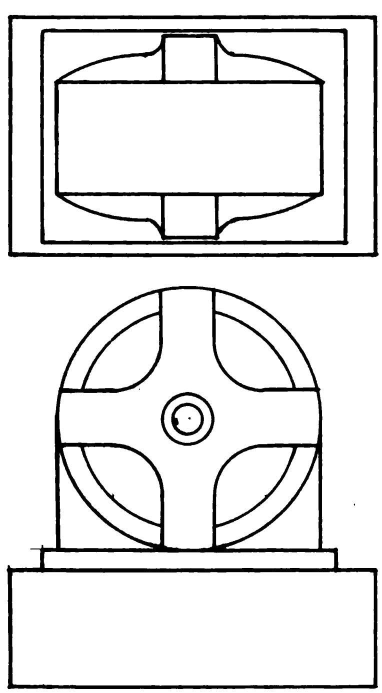


Figure 1

In a foundation of this type the design is usually one of expediency, as the soil is usually able to support the load involved. The problem sometimes becomes one of decreasing the bearing area.

Figure 2 shows a diagram of the loading of such a foundation, a study of which indicates that the cross sectional area must be symetrical with c. g. of the machine and the unit pressure "p" must not exceed the soil pressure specified for static loads on the type of soil on which the foundation is to be constructed.

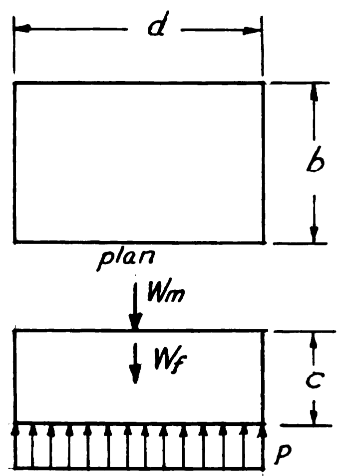


Figure 2.

In this design we make use of the simple formula:

$$f = \frac{P}{A} = \frac{Wm + Wf}{b \times d} \qquad (1)$$

Where "f" is the allowable soil pressure, W_m is the weight of the machine, W_f is the weight of the foundation, b and d the plan dimensions of the foundation.

We can safely assume that the pressures are the same at any point under the foundation.

<u>Numerical example:</u> (Illustrating above formula)

Design a concrete foundation for a syncronous converter, whose weight is 4000#, in an exposed location, where the frost depth is 4'. The base of the machine must be 28" above the ground. The soil is a soft clay. Figure 3 shows the location of the anchor bolts.

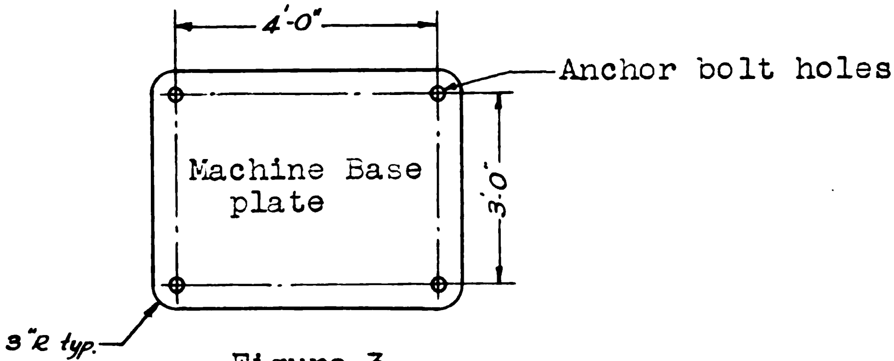


Figure 3

Solution:

This machine would require a base of at least 54" x 42" Usual practice being to allow at least a couple of inches around the edge of the base plate. The concrete foundation would then be 60" x 48" x 72".

Then applying formula (1) with:

$$W_f = 150^{\#}/_{cu'} \times 5' \times 4' \times 6' = 18{,}000\#$$

$$f = \frac{4000 + 18000}{5 \times 4} = 1100^{\#}/_{ft.^2}$$

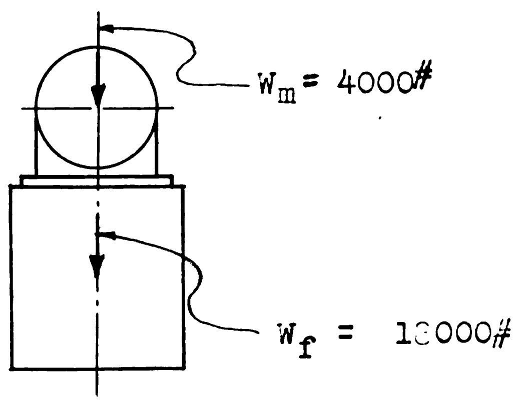


Figure 4

This pressure is probably well within the allowable even for a soft clay. (For a vibrating machine it would be close to the allowable.)

Note: the details of a foundation of this type, such as reinforcing - anchor bolts, etc. will be taken up later, (Part 7) as they apply to nearly every type of foundation.

A-3(a) <u>Foundations in which the loads do not pass through the center of foundation</u>.

A good example of this type of foundation would be one designed to carry the loads from an electric motor driving a machine by means of a belt.

In figure 5 T = load due to belt tension (strictly speaking, the resultant of pulls from the tight and slack sides), W_m = the weight of the motor, W_f = the weight of the foundation.

The resultant "R" of all the forces acting on the soil is shown below. It is obvious that R does not act through the center of the foundation and that the pressures on the soil would not be constant but would vary in direct proportion to their distance from the axis.

With this type of loading it is imperative that R does not fall outside the base line and it is usually suggested by foundation experts that the resultant fall within the middle one-third of the base. Of course the three requirements mentioned in the previous article hold for this one too.

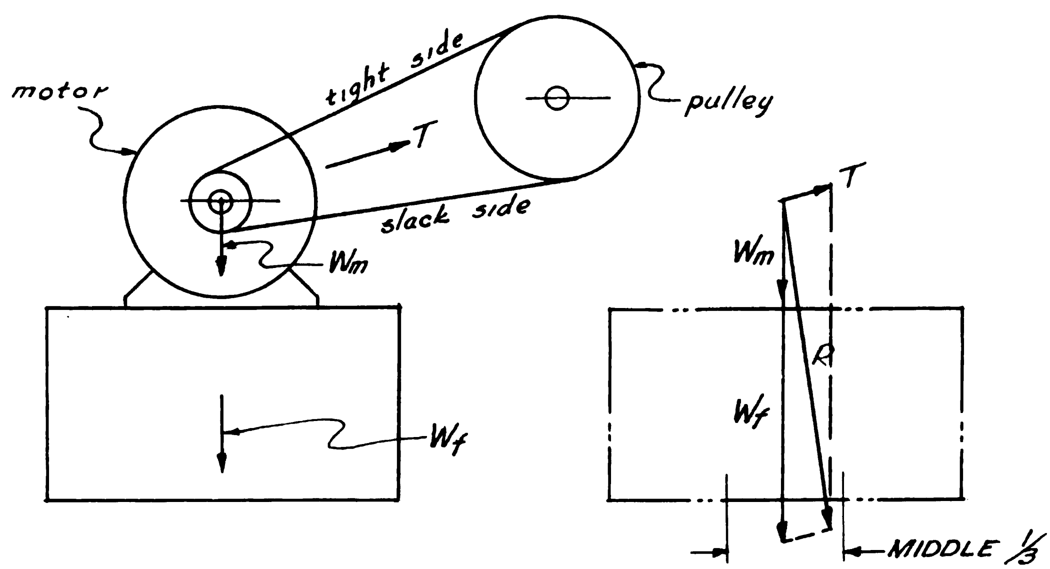


Figure 5

After finding the resultant of the loads acting on the soil and making the weight of the foundation sufficient to keep R within the middle one-third of the base, the next step is to calculate the resulting pressures on the ground.

If we resolve the resultant force, R into its vertical and horizontal components at the point where it intersects the base line, (figure 6) we see that the downward pressure on the soil is $R_v \pm$ a force due to a couple $R_v xe$. The horizontal component of R causes a horizontal load R_h which tends to slide the foundation on the soil.

The downward unit pressure on the soil therefore could be considered as a uniformly distributed pressure $R_v/bc \pm$ the varying pressure Mc/I.

$$p = R_v/bc \pm R_v ex/I \qquad (2)$$

<u>Where</u>: $R_v =$ vertical component of the resultant of loads, R

b,c = the dimensions of the foundation base.

e = the distance from the center of the base to the point where the vector R intersects the base line

x = the distance from the center of the base to the point where we are concerned with the pressure.

I = moment of inertia of the base about an axis through the center of the base, $1/12\ cb^3$ for a rectangular foundation.

The maximum pressure will be at "n" and of a magnitude:

$$P(n) = R_v/bc \pm 6R_v e/cb^2 \qquad (3)$$

It is obvious that the greatest unit pressure occuring, should be below that which the supporting soil can safely carry. There is, however, another important consideration. The foundation is bound to settle a small amount, depending on the pressure and the type of soil; practically speaking

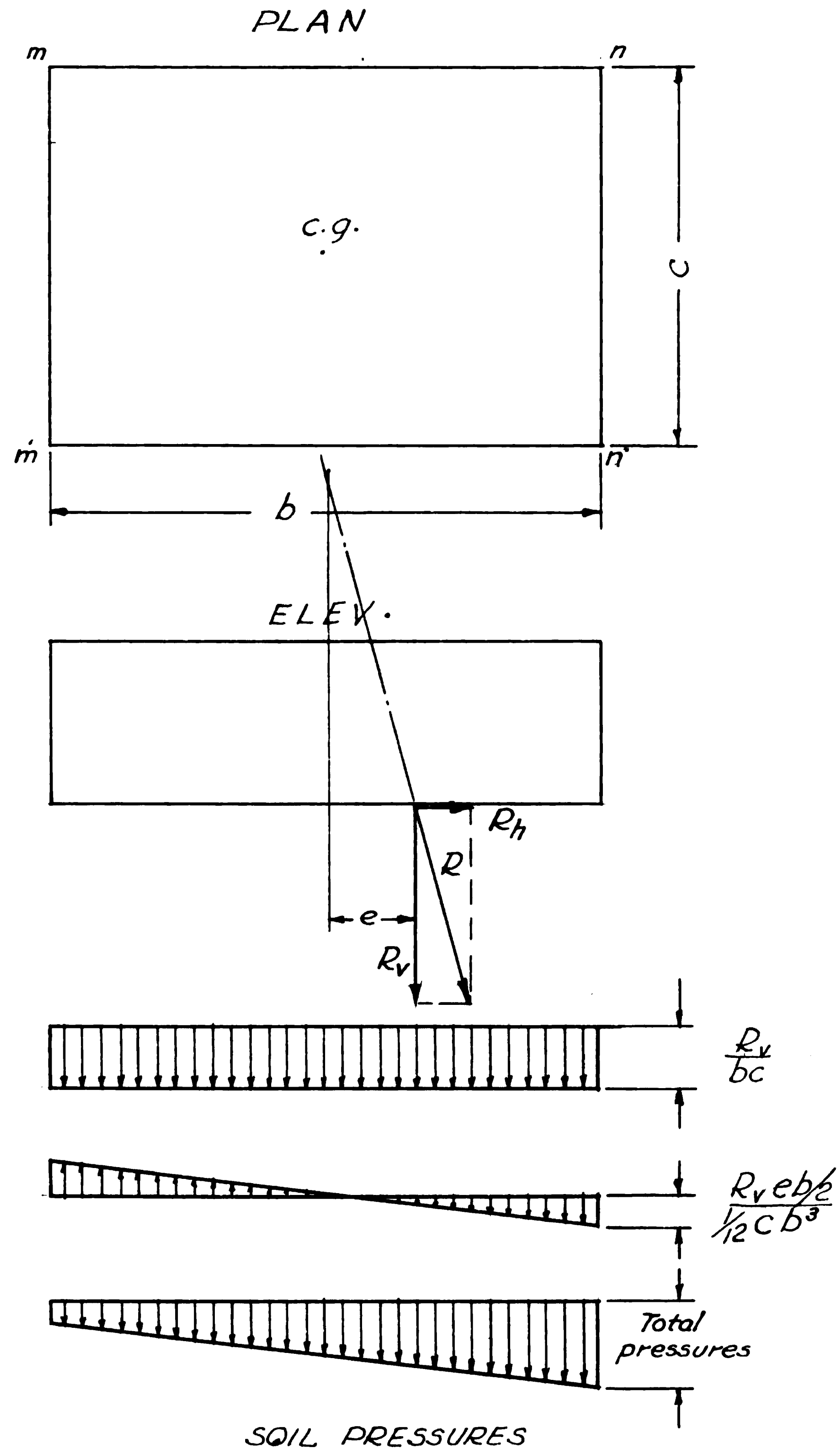

PLAN
m
n
c.g.
c
m'
n'
b
ELEV.
R_h
R
e
R_v
$\frac{R_v}{bc}$
$\frac{R_v e b/2}{1/12\, c b^3}$
Total pressures
SOIL PRESSURES

Figure 6

the settlement will be proportional to the pressures. It will be seen, therefore, that if the settlement of every portion of the foundation is to be about equal the pressures must also be about equal. Very little trouble will be experienced if the maximum pressure is kept below twice the minimum pressure.

The problem just illustrated assumes that the force T acted centrally to the foundation block. Actually the belt pull, T, acts eccentrically since it is usual for the pulley of the motor to be overhung. The foundation has therefore to resist an additional overturning moment equal to the product of the distance from the pully center to the center of the armature, and the force T.

This problem is also simplified to the extent of our assuming a rectangular foundation base, which in a great many cases is not possible.

The next article deals with a foundation in which the base is not a rectangle.

A-3 (b) Foundations in which the base of the foundation is not rectangular

For the foundation problem in which the base is not rectangular, we can use the same theory developed for a short strut subjected to an eccentric load. The problem is to determine the pressure at various points due to the eccentric load R_V. From the sketch (figure 7) it is seen that X-X and Y-Y are the centroidal axis. If it is assumed that a plane before bending is a plane after bending, Hookes Law holds and pressures will vary over the cross section as the ordinates of the plane (efgh).

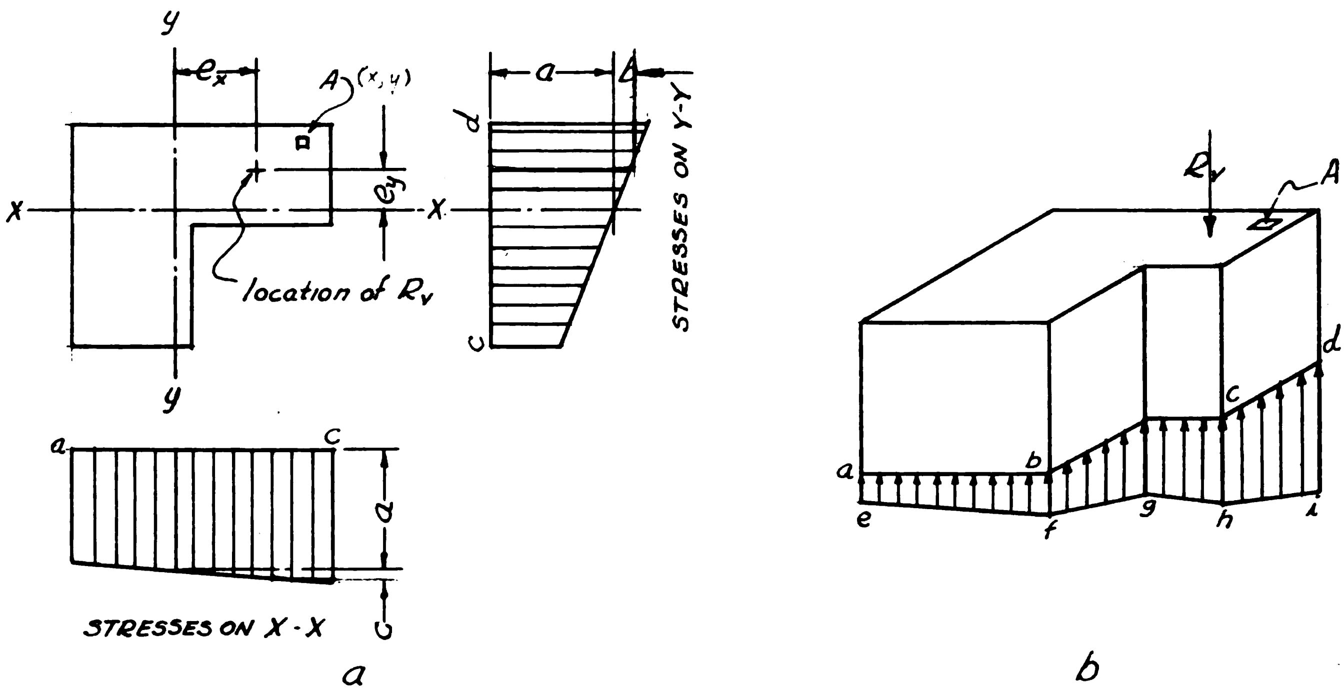


Figure 7.

These pressures are represented in figure 7 as vectors drawn to the plane abcd. The stress at A is : $p = a + by + cx$ (1) depending on the values of the constants a,b, and c, which in turn are dependent upon the loading and dimensions of the cross section. The three equations of equilibrium will be used to find these constants;

1. Sumation $M_x x = 0$
2. Sumation $M_y y = 0$
3. Sumation $F_z = 0$

By condition 3, $R_v = \int pdA$, where dA is the differential area at any point A. Substituting the value for p noted above (1).

$$R_V = a\int dA + b\int xdA + c\int ydA$$

since X-X and Y-Y are centroidal axis $\int xdA = 0$ and $\int ydA = 0$

therefore:

$$R_V = a\int dA \quad \text{or} \quad a = P/A$$

Using the second equilibrium condition $\sum M_y = 0$ and calling the moment of the force R_V about the Y axis M_y.

$$M_y = \int px\, dA$$

$$M_y = a\int xdA + b\int x^2 dA + c\int xydA$$

The first term on the right hand side of the equation = 0 and therefore:

$$M_y = bI_y + cI_{xy} \qquad (5)$$

where $I_y = \int x^2\, dA$ = the moment of inertia about the Y axis.

and $I_{xy} = \int xydA$ = the product of inertia.

By the third condition of equilibrium $M_x = 0$ and

$$M_x = \int pydA$$

$$M_x = a\int ydA + b\int xydA + c\int y^2 dA$$

$$M_x = bI_{xy} + cI_x \qquad (7)$$

Equations 5 and 7 can be solved simultaneously for b and c:

(Multiply 5 by I_x and 7 by I_{xy} then subtract 7 from 5)

$$M_y I_x = bI_y I_x + cI_{xy} I_x$$

$$M_x I_{xy} = bI_{xy}^2 + cI_{xy} I_x$$

$$M_y I_x - M_x I_{xy} = b(I_x I_y - I_{xy}^2)$$

$$b = \frac{M_y I_x - M_x I_{xy}}{I_x I_y - I_{xy}^2}$$

In a similar way:

$$c = \frac{M_x I_y - M_y I_{xy}}{I_x I_y - I_{xy}^2}$$

Placing the values found above for a,b, and c in equation (1):

$$s = \frac{P}{A} + \frac{M_y I_x - M_x I_{xy}}{I_x I_y - I_{xy}^2} x + \frac{M_x I_y - M_y I_{xy}}{I_x I_y - I_{xy}^2} y \qquad (4)$$

Where: $M_y = Pe_y$, $M_x = Pe_x$, and e_y and e_x are the eccentricities measured from the x and y axis respectively.

The stress s is now completely defined at any point A on the cross-section x (see figure 7) in terms of the dimensions and the load.

Some convention for signs must be used when applying the above equation (4): A tensile force P is positive, and the values of x and y in the upper right hand quadrant are positive; therefore, M_y is positive if e_y is negative.

In a particular problem, the problem of calculating the stress s at some point A is as follows:

(1) Determine the centroidal axis of the cross-section.

(2) Determine the moments of inertia about the centroidal axis, I_x and I_y.

(3) Determine the product of inertia of the cross-section I_{xy}.

(4) Calculate $M_y = Pe_y$ and $M_x = Pe_x$

(5) Determine the stress s at the selected point A, using formula (4)

In special cases of loading and cross sections, the formula just developed is somewhat simplified:

(1) Where base area has two axis of symmetry (rectangles, circles, I sections, etc.)

For these, the product of inertia I_{xy} becomes zero and formula (4) becomes:

$$s = \frac{P}{A} + \frac{M_x y}{I_x} + \frac{M_y x}{I_y} \qquad (5)$$

(2) Where the cross-section has two axis of symmetry and the load is eccentric on the x-x axis:

In this case, equation (5) may be used, but M_x becomes zero because e_x is zero and $M_x = Pe_x = P.0 = 0$. Thus, equation (5) becomes:

$$s = \frac{P}{A} + \frac{M_y x}{I_y} \qquad (6)$$

In most practical problems the designer is usually interested only in the maximum and minimum soil pressures. Then x becomes c, the distance from the centroidal axis to the edge of the foundation.

An example follows where it is desired to find the maximum and minimum soil pressures under a certain foundation with the resultant load on the x - x axis.

Numerical Example:

With the loadings as shown on (figure 8) find the maximum and minimum unit pressures which will be imposed on the soil by the foundation.

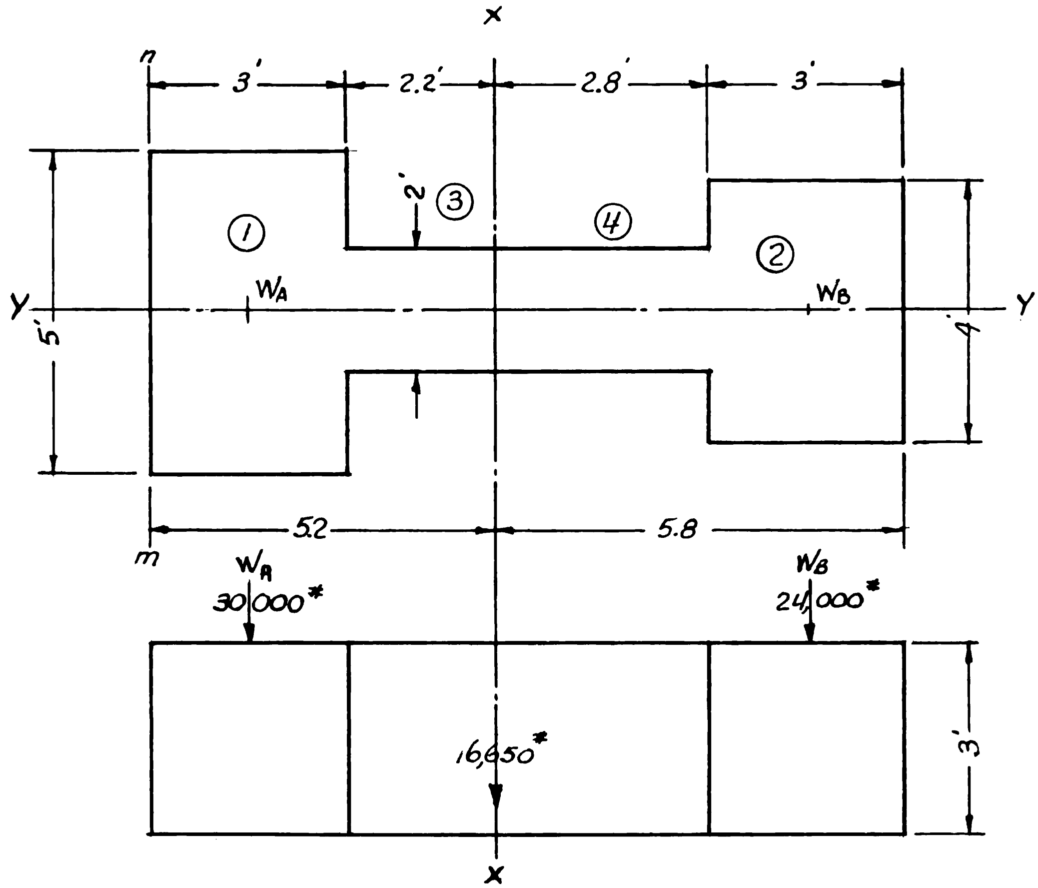


Figure 8

Solution:

1. Find the weight of the foundation.

$$150\# \times 3(15+12+10) = 16{,}650\#$$

2. Find the centroidal axis.

Use method of moments taking moments about "mn".

$$A = 37 \text{ sq.}'$$

$$37\bar{y} = 12 \times 9.5 + 10 \times 5.5 + 15 \times 1.5$$

$$37\bar{y} = 114 + 55 + 22.5$$

$$\bar{y} = \frac{191.5}{37} = 5.2$$

3. Find I_x (The moment of inertia of a rectangle about its base is : $1/3\ bh^3$)

$$\frac{1}{3} \times 5 \times 5.\bar{2}^3 = 234$$
$$\frac{1}{3} \times 4 \times 5.\bar{8}^3 = \underline{260}$$
$$+\ 494$$
$$-\ \underline{\ 25\ }$$
$$+\ \underline{469}\ ft.^4$$

$$\frac{1}{3} \times 3 \times 2.\bar{2}^3 = 10.65$$
$$\frac{1}{3} \times 2 \times 2.\bar{8}^3 = \underline{14.65}$$
$$-\ 25.30$$

4. Find e_x (the distance from the center of area to the point where the resultant load W_t acts.)

Take moments about "mn".

$$70,650 \times y_1 = 30,000 \times 1.5 + 16,650 \times 5.2 + 24,000 \times 9.5$$
$$y_1 = \frac{359,580}{70,650} = 5.08$$
$$e_x = 5.2 - 5.08 = .12'$$

5. $M_x = e_x \cdot W_t = .12 \times 70,650 = 8479'\#$

6. Determine the stress at the maximum and minimum points by using formula (4).

$$s_{max.} = \frac{P}{A} + \frac{M_x y}{I_x}$$

$$s_{min.} = \frac{P}{A} + \frac{M_x y}{I_x}$$

$$s_{max.} = \frac{70,650}{37} + \frac{8478 \times 5.2}{469} = 1912 + 94 = \underline{2006}\#/ft.^2$$

$$s_{min.} = \frac{70,650}{37} - \frac{8478 \times 5.8}{469} = 1912 - 105 = \underline{1807}\#/ft.^2$$

B. <u>When the horizontal component of the resultant load, acting on the foundation is significant.</u>

In the previous articles, it was suggested that the resultant of all the forces acting on the foundation should fall within the middle third of the foundation base, and that the vertical pressures on the soil should not be too great or of too great variation. A consideration now arises as to the horizontal component of the resultant load, if there be one. Where a horizontal component does exist, the only thing which keeps the foundation from slipping on the soil or its supporting medium is their frictional resistance, unless, of course, it is supported also horizontally as by a floor slab.

The weight of the foundation must be sufficient to prevent slipping. If the horizontal component of the load acts through the c.g. the required weight is found quite simply. The assumption is made that the load is distributed uniformily over the base area. Therefore, the horizontal force P is resisted by the product of the coefficient of friction and the sum of the weights of the machine and the foundation, which can be assumed to act at the c.g. or:

$$P = \mu (W_f + W_m) \qquad (7)$$

From which the required weight of foundation plus machine becomes:

$$W_f + W_m = P/\mu \qquad (8)$$

It must be noted, however, that in many cases the horizontal component does not act through the c.g. If this be the case, an expression must be derived for the required weight to resist this turning tendency and added to the above, in order to find the necessary weight of the foundation.

Derivation of expression for W_f required to resist turning.

Figure 9 shows a cross section of a foundation with a horizontal load P acting at a distance e from the c. g. The weight of the machine and foundation will be assumed to be uniformily supported over the entire base, therefore, the frictional resistance of each infinitesimal area dA will be the same. If the coefficient of friction between the soil and the concrete is μ, then the resisting force is,

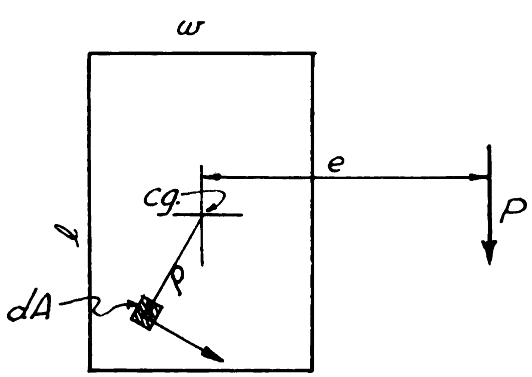


Figure 9

Where A is the area of base:

$$\mu (W_f + W_m) \frac{dA}{A}$$

The resisting moment of the force acting on the increment area dA is:

$$\rho \left[\mu (W_f + W_m) \frac{dA}{A} \right] \quad (A = \text{area of base})$$

If the resisting moments of all these forces are summed up, we have an expression for the resistance to pure turning which the friction between the foundation and soil offers:

$$\text{Resisting } M = \sum \mu \rho (W_f + W_m) \frac{dA}{A}.$$
$$= \frac{\mu}{A} (W_f + W_m) \sum \rho \, dA.$$

Equating this to the moment causing the turning and substituting K for $\sum \rho \, dA$:

$$Pe = \frac{\mu}{A} (W_f + W_m) K .$$

From this expression the required weight to prevent pure turning becomes:

$$W_f + W_m = \frac{PeA}{\mu \cdot K} \qquad (9)$$

To this must be added the weight required to prevent slipping in the direction of P (formula 8) and the total weight required to resist Pe is:

$$W_f + W_m = \frac{PeA}{\mu K} + \frac{P}{\mu} \qquad (10)$$

The value for "K" in the above formula must be found from calculus. First the area is divided into 8 sections as shown in figure 10. Then K is found for the "a" sections and called K_a, and for the "b" sections and called K_b . The sum of K_a and K_b is the K in formula (10).

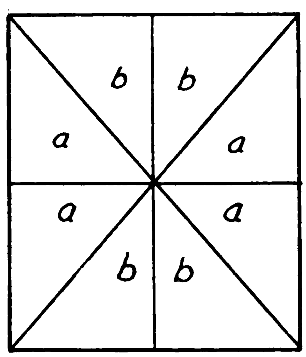


Figure 10

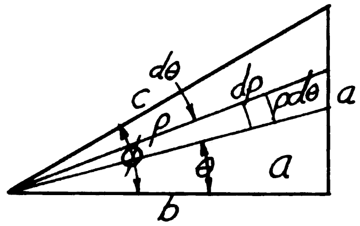


Fig. 11a

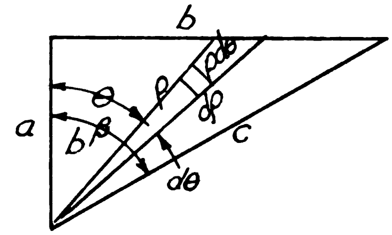


Fig 11 b

$$K_a \text{ (see fig. 11a)} = 4\int_0^{\phi}\int_0^{\frac{b}{\cos\theta}} \rho^2 d\rho\, d\theta = \frac{4}{3} b^3 \int_0^{\phi} \frac{d\theta}{\cos^3\theta} =$$

$$= \frac{4}{3} b^3 \left[\frac{1}{2}\frac{\sin\theta}{\cos^2\theta} + \frac{1}{2}\cdot\frac{1}{2}\cdot \log\left(\frac{1+\sin\theta}{1-\sin\theta}\right)\right]_0^{\phi}$$

$$= \frac{4}{3} b^3 \left(\frac{1}{2}\frac{\sin\phi}{\cos^2\phi} + \frac{1}{4}\log\left(\frac{1+\sin\phi}{1-\sin\phi}\right)\right) \quad \text{But } \sin\phi = \frac{a}{c} \text{ and } \cos\phi = \frac{b}{c}$$

$$\therefore K_a = \frac{4}{3} b^3 \left(\frac{1}{2}\cdot\frac{a}{c}\cdot\frac{c}{b}\cdot\frac{c}{b} + \frac{1}{4}\log\left(\frac{1+\frac{a}{c}}{1-\frac{a}{c}}\right)\right) = \frac{4}{3} b^3 \left(\frac{1}{2}\frac{ac}{b^2} + \frac{1}{4}\log\left(\frac{c+a}{c-a}\right)\right)$$

$$K_b \text{ (see fig. 11b)} = 4\int_0^{\beta}\int_0^{\frac{a}{\cos\theta}} \rho^2 d\rho\, d\theta = \frac{4}{3} a^3 \left(\frac{1}{2}\frac{\sin\beta}{\cos^2\beta} + \frac{1}{4}\log\left(\frac{1+\sin\beta}{1-\sin\beta}\right)\right)$$

but $\sin\beta = \frac{b}{c}$ and $\cos\beta = \frac{a}{c}$

$$\therefore K_b = \frac{4}{3} a^3 \left(\frac{1}{2}\cdot\frac{b}{c}\cdot\frac{c}{a}\cdot\frac{c}{a} + \frac{1}{4}\log\left(\frac{1+\frac{b}{c}}{1-\frac{b}{c}}\right)\right) =$$

$$K_b = \frac{4}{3} a^3 \left(\frac{1}{2}\frac{bc}{a^2} + \frac{1}{4}\log\left(\frac{c+b}{c-b}\right)\right)$$

$$K = K_a + K_b = \frac{4}{3}\left[\frac{bac}{2} + \frac{b^3}{2}\log\left(\frac{c-a}{c-a}\right) + abc + \frac{a^3}{2}\log\frac{(c+b)}{(c-b)}\right]$$

$$K = \frac{2}{3}\left[2abc + \frac{b^3}{2}\log\left(\frac{c+a}{c-a}\right) + \frac{a^3}{2}\log\left(\frac{c+b}{c-b}\right)\right] \qquad (11)$$

Where the foundation base is square or side "l" equals side "w", a equals b and K now becomes:

$$K = \frac{2}{3}\left[2a^3\sqrt{2} + 2\,\frac{a^3}{2}\log\left(\frac{a\,(\sqrt{2}-1)}{a\,(\sqrt{2}-1)}\right)\right]$$

$$K = \frac{2}{3}a^3\,(2.828 + \log 5.82) = \frac{2}{3}a^3(2828 - 1.761)$$

$$K = 3.06a^3 \quad \text{but } a = \frac{W}{2} \quad \text{so} \quad K = \frac{3.06}{8}W^3 = .38W^3$$

Putting this into the expression for the required weight to resist the turning moment we have:

$$W_f = \frac{P}{\mu}\left(1 + \frac{e}{.38w}\right) - W_m \qquad (12)$$

This of course is only true when the foundation cross-section is square. For other relationships between the side dimensions the general formula (11) must be used for K.

<u>The coefficient of friction between the concrete and the soil:</u>

There is a great deal to be said in regard to a coefficient of friction between a concrete foundation and soils. Many things enter into its determination for which there is not time here to discuss. Certainly the type of soil and its water content are of utmost importance. Table II gives values for the coefficient (μ) which have been found by experimentation and represent probably the most reliable general values obtainable.

TABLE II

Values of (μ) for Earth Foundations*

Material		μ	safety factor	(μ) with safety factor
Concrete or Masonry	on gravel	0.5	2.5	0.20
	on sand	0.4	2.5	0.16
	on clay	0.3	2.5	0.12

* From "Low Dams" Water Resources Committee - National Resources Board.

C. When there exists an upward component of force on the foundations.

It is possible with a gear or belt driven machine or other type of similar mechanism, that an upward thrust may exist which would cause the foundation weight to be so unevenly distributed, as to be reflected in settlement on one side. The possibility may also occasionally arise where the weight of the foundation block must be made sufficiently large so as to keep the machine from being lifted.

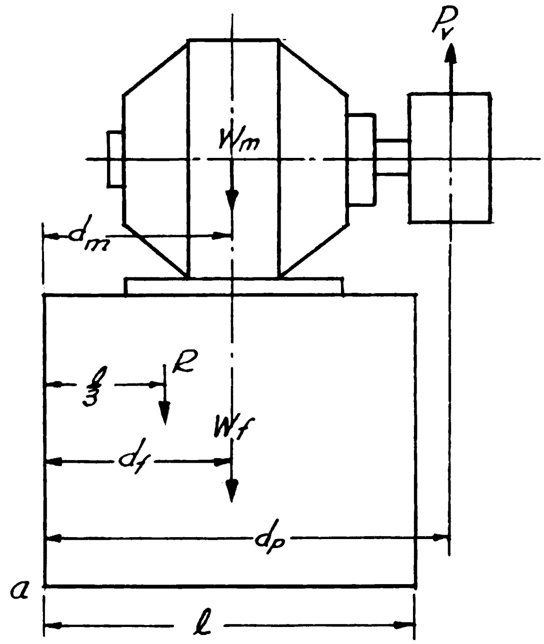


Figure 12

An expression for the required weight of foundation to keep the machine from lifting may be obtained by applying the principle of equilibrium; namely, that the summation of moments of all the forces acting on a body about any point are zero.

Figure 12 shows a machine and foundation with an upward component of force P_V. The moments of the downward forces W_f and W_m, about a, (the point about which the foundation would pivot) must at least equal the moment of the force P_V about the same point same point if equilibrium is going to result.

$$W_f \times d_f + W_m \times d_m = P_V \times d_p$$

and transposing, the required foundation weight:

$$W_f = \frac{P_V \times d_p - W_m \times d_m}{d_f}$$

In regard to the first possibility mentioned above, it has been cited in previous articles, that to insure against the possibility of unequal settlement it is usually assumed that the resultant of forces acting on the ground should pass through the middle 1/3 of the foundation. If this is to be provided the resultant of all the forces acting on the soil "R" must be at least $\frac{\ell}{3}$ from the point "a" (figure 12).

As the moment of the resultant R about a, must be equal to the sum of the moments of all of the other forces about a,

$$R\frac{\ell}{3} = W_m \times d_m + W_f \times d_f - P_v \times d_p$$

the resultant

$$R = W_m + W_f - P_v$$

$$(W_m + W_f - P_v)\frac{\ell}{3} = W_m.d_m + W_f \times d_f - P_v.d_p$$

and transposing the required foundation weight.

$$W_f = \frac{P_v(3d_p - \ell) - W_m(3d_m - \ell)}{3d_f - \ell} \qquad (13)$$

III - THE DETERMINATION OF FORCES SUCH AS GEAR THRUST AND BELT TENSION WHICH MAY ACT ON A MACHINE FOUNDATION

The easily derived formulas given below can be used in the determination of gear thrust or belt pull on a foundation:

$$F_g = \frac{10500}{N.D_f} \cdot P \qquad \text{(gear drives)} \quad (14)$$

$$F_b = \frac{30,000}{N.D_f} . P \qquad \text{(belt drives)} \quad (15)$$

<u>Wherein:</u> F_g = tangential force on the gear teeth in pounds.

P = the power which is being transmitted in horse-power

N = the speed of the gear or pulley in rpm.

D_f = the diameter of the gear or pulley in feet.

F_b = the pull of the machine due to the belt, in pounds.

For derivation of these formulas see any machine design hand-book.

IV - LOADINGS DUE TO TORQUE OF A MOTOR WHICH MAY CHANGE THE LOADING OF A FOUNDATION.

In the design of a foundation to support a rotating motor, we have in addition to the static weight of the machine, the forces transmitted to the machine supports arising from the torque of the motor itself.

An investigation of just what this torque is and how it may be calculated from known data, follows.

When a force "P" drives a body through a distance "s" against an equal and opposite resistance "Q" work is performed.

$$wk = Ps = Qs$$

(In the case of an electric motor the force which drives the armature is "P" and the equal and opposite force Q will cause a torque, which must be resisted by the machine supports.)

The work noted above will be in foot pounds if P is in pounds and s is in feet. If this work is done in time t, the average velocity will equal the displacement s over t.

$$V_{av.} = {}^{s}/_{t}$$

The rate of doing work would be P_v or Q_v. In our case the velocity is that of a pt on a circular path or:

$$V_{av.} = 2\pi r N$$

where: r is radius at which P acts and N is revolutions per minute.

work done per minute, $P_v = P 2\pi r N$

One horse-power = 33,000'#/minute $\therefore HP = \frac{P_v}{33,000} =$

$$HP = \frac{P_v}{33,000} = \frac{P.2\pi rN}{33,000} = \frac{PrN}{5252.1} \quad \text{and} \quad P = \frac{HP.5252}{Nr} \qquad (16)$$

The above relationship will allow us to solve for the force P as the horse-power, the revolutions per minute and the value "r" are usually known or easily obtainable from the manufacturer or by tests.

A problem follows which will illustrate the use of the above to find the additional loadings on the foundation due to torque. (see problem 9A)

V - THE THEORY BEHIND INTERNAL UNBALANCED MECHANICAL FORCES, ORGINATINATING IN THE MACHINE ITSELF, WHICH MAY ACT ON A MACHINE FOUNDATION.

In the preceding articles the loads acting on the foundation were assumed to be gradually applied, or static like the load on a building foundation. In many machine foundations however, resistance to loads caused by moving bodies must be provided. A study of these loads and the design of foundations to provide for them follows:

A study of the internal or unbalanced, mechanical forces which may act on a machine shows that they may be due to either rotating or reciprocating masses.

A - <u>In purely rotative machines</u> the internal forces, (usually small) which do act, are due to a lack of perfect balance in the rotating parts. If the center of gravity of the rotating mass is at the exact center of rotation there can be no internal forces.

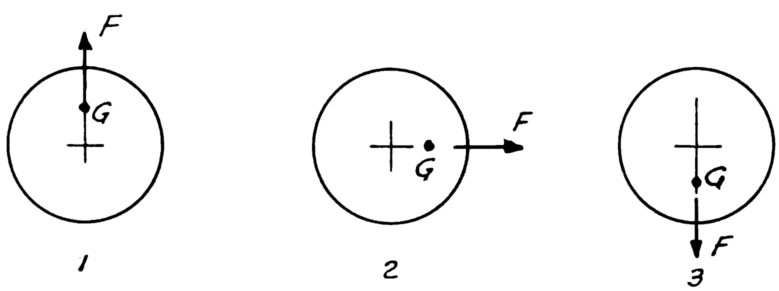


Figure 13

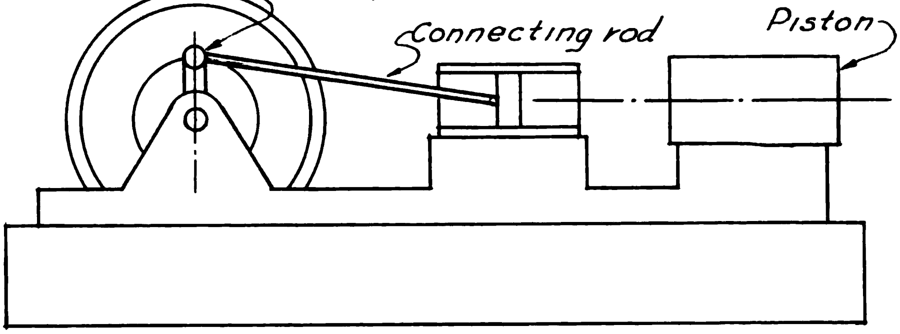
Figure 13 shows the rotative parts of a machine in three positions. It is rotating about the longitudinal axis of the shaft and the center of gravity is atG. The machine is continually acted upon by a centrifugal force F caused by the unbalanced mass. This force, varying in direction, must be resisted by the bearings, and ultimately by the foundation.

If weight is added to the rotative part on the side opposite the point G, the centrifugal force would be decreased. It is thus possible to balance, to a certain extent the rotative parts of many machines. The extent of balance in various parts however, is usually a question mark in the mind of the foundation designer.

If there exists such an unbalance,the load which is normally delivered to the foundation from the machine will be increased by the amount of this centrifugal force. This force in itself might not be too critical if it were not for the fact that it occurs periodically thus giving rise to vibrations which may or may not be harmful. (vibrations and their treatment will be discussed later.)

B - <u>In reciprocating machinery</u> there are usually forces of the type occuring in rotating machine, as well as forces due to the acceleration and deceleration of reciprocating parts.

In order to study these forces, consider a single cylinder horizontal engine of the corliss type. (see figure 14).



Diag. of corliss engine

Figure 14

The moving parts of the engine possess mass and therefore inertia and when they are accelerated or retarded give rise to inertia forces in accordance with Newton's fundamental law:

Force = Mass x Acceleration

The summation of these forces at a given instant is therefore the force transmitted to the foundation at that particular instant.

These forces are in general approximately harmonic and therefore periodic. The resultant must of necessity also be periodic.

In order to express the relationships involved in a mathematical manner let:

W_1 = wt of eccentric rotating parts.

W_2 = wt of reciprocating parts

l = length of connecting rod between centers of pins.

r = crank radius

$q = l/r$

ω = angular velocity of crank

In regard to the rotating parts:

$r\omega^2$ is the normal acceleration for uniform speed.

Then F_1 (inertia force) = $\frac{W_1}{g} r\omega^2$

and this force is directed radially outward from the center of rotation.

In regard to the reciprocating parts:

The maximum acceleration occurs at the ends of the strokes where the direction of motion is reversed. For a connecting rod of finite length, acceleration depends on a ratio of connecting rod length to the crank radius.

In this case it is easily shown that the acceleration at the out end of stroke is $r\omega^2 (1-\frac{1}{q})$ and at the in end of stroke is $r\omega^2 (1+\frac{1}{q})$

The inertia force due to this acceleration or retardation is $F_2 = \frac{W_2}{g} r\omega^2 (1 \pm \frac{1}{q})$

Therefore, the maximum kinetic reaction exerted by the engine would be:

$$F_1 + F_2 = \frac{W_1}{g} r\omega^2 + \frac{W_2}{g} r\omega^2 (1 \pm \frac{1}{q}) \qquad (17)$$

The magnitude of the internal forces then depends (1) In purely rotative machines on the weight, speed and distance of the center of gravity of the rotating mass from the axis of rotation. (2) In reciprocating machines, on speed of the machine, the weight of the rotating masses, the weight of the connecting rod, the ratio of crank length to connecting-rod length, the radius of the center of gravity of the rotative weights, and on the degree of effectiveness of the counter balancing.

Now to incorporate provisions in the design of a foundation to take care of these internal forces is a rather complicated and partially unsolvable problem. The advice given by most authorities is to make allowance for them by a sufficient factor of safety.

The designer can usually do a little better than this. He can in many cases calculate these forces approximately and make allowances for this approximation with a sufficient factor of safety.

C - THEORY BEHIND VIBRATIONS IN MACHINE FOUNDATIONS

A Vibration may be defined as a periodic motion which changes direction twice during a complete cycle, and repeats itself after a certain interval of time (its period). The simplest kind of periodic motion is simple harmonic motion.

Simple harmonic motion may be defined as the motion of a point in a straight line such that the acceleration of the point is proportional to the distance, x, of the point from some fixed origin, O, in the line, and is directed toward O. (It is a special case of rectilinear motion).

Examples of simple harmonic motion are:

1. The motion of a weight attached to the lower end of a helical spring.
2. The motion of a cross-head of a steam engine closely approximates a harmonic motion if the ratio of length of connecting-rod to that of the crank is large.
3. The motion of an oscillating pendulum if the arc is small.

In fact many vibrational motions so common to engineering problems may be assumed without serious error to be simple harmonic motions.

The harmonic motion of one vibrating mechanism may differ considerably from the harmonic motion of another vibrating mechanism. The differences are described by such terms as:

Amplitude: A: The magnitude of the vibratory motion from the equilibrium point to the extreme position.

Frequency: The number of cycles of motion per unit time (measured in cycles per second (f)) (or radians per second (p)).

Period P: The time it takes to complete a cycle (measured in seconds).

For our purpose vibrations may be considered to consist of two types: free damped vibrations and forced vibrations.

Free damped vibrations are the same as those described above except that they are affected by forces such as air resistance, internal friction of the vibrating material,

friction between sliding surfaces, etc. These forces are always present and act as sources of damping forces.

Forced vibrations are the result of a periodic disturbing force acting on a vibrating body and exist wherever there are moving parts in machines.

The frequency of vibration of the body subjected to a continuously acting period force will be that of the disturbing force. Therefore, if vibrations are to be minimized, the forced vibrations must be brought under control. This and the problem of making sure that the condition of resonance, (when the natural frequency of the foundation, supporting soil, or supporting structure is nearly the same as the frequency of the periodic force) does not exist, are the two problems facing the foundation designer in regards to vibrations.

From The Theory of Vibrations (Forced vibrations with viscous damping.)

$$A = \Delta \cdot \frac{1}{\sqrt{(1+\frac{\omega^2}{\rho^2})^2 + (2c \cdot \omega/\rho)^2}} \qquad (18a)$$

Where:

A = amplitude

Δ = deflection due to static load of the same magnitude as the unbalanced periodic force of engine. (W)

ω = frequency of unbalanced periodic force of engine.

ρ = undamped natural frequency of elastic system.

C = coefficient of damping; $C = 1$, is point of critical damping where damping is so great as to prevent vibration.

The right hand term above is called the magnification factor. Figure 15 shows curves with various values of c. It should be noted that they all have a sharp peak where

$$\frac{\omega}{p} = 1.$$

The values for the natural frequency of various supporting structures, at least those of a relatively simple nature, can be obtained rather easily as we will demonstrate later.

The above is also approximately true for a machine on elastic subsoil, and it can be seen from the curve that the amplitude of the vibration is greatly increased if its natural frequency is close to the forced frequency of the unbalanced forces of the machine, and of necessity this condition must be avoided.

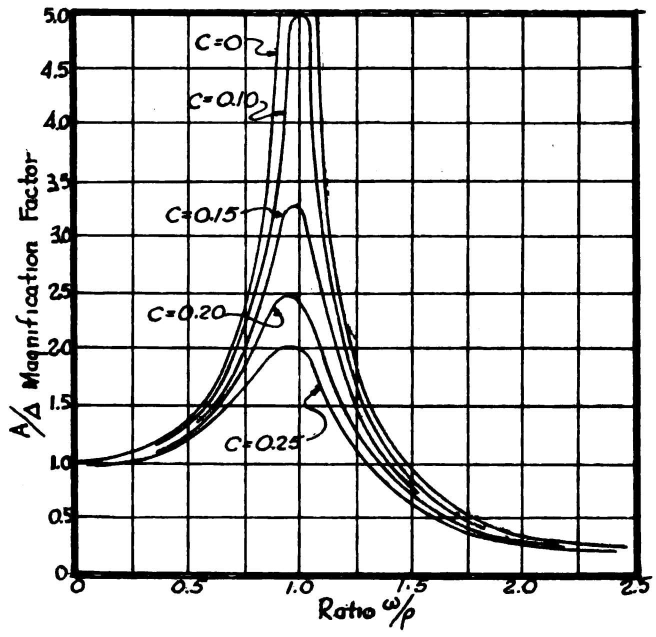
The value of c will vary for different soils or supporting media. If we assume c = 0.25 as an average soils, we can see that if we avoid ω/p values between 0.5 and 1.25 the amplitude will not be greatly increased.

From "Mechanical Vibrations" by DenHartog it is seen that the undamped natural frequency of an elastic system:

$$\left(p = \sqrt{\frac{g}{\Delta}}\right) \text{ and } f_n = \frac{1}{2\pi}\sqrt{g/\Delta} \qquad (19)$$

$$f_n = 3.14\sqrt{\frac{1}{\Delta}} \text{ cycles per second} \qquad (19a)$$

$$f_n = 188\sqrt{\frac{1}{\Delta}} \text{ cycles per minute} \qquad (19b)$$



-COEFICIENT OF DAMPING CURVES-

Figure 15

Note: This formula (19b) may be used for computing the horizontal translational natural frequency as well, if for the deflection an apparent deflection is substituted,

The torsional natural frequency is not unlike the above.

$$F_t = 9.54\sqrt{\frac{K_t}{I}} \qquad (20)$$

Where:

K_t = torsional spring rate

I = moment of inertia of mounted equipment (in slug feet sqd.)

K_t is figured by multiplying the translational spring rate (load reqd. per foot of linear deflection) x the distance from the elastic center (in feet) squared for each individual mounting and summing the products, giving the torsional spring rate in (pounds feet/radians) for the entire installation. The elastic center is a point about which a couple would cause rotary movement of the mounted equipment and is sometimes known as the center of elastic resistance.

Having arrived at this natural frequency the transmissability formula given applies fully. (recognized that figures on moment of inertia are not always available, but such data can commonly be arrived at by observing torsional resonance in a trial installation.)

Thus, it is seen that if a value for Δ is known we can predict the undamped natural frequency from formula (19b), and the magnification factor from figure 15. Δ , the static deflection for soils should be determined by static load tests on the soil at the bottom of the foundation in the field.

The undamped natural frequency for some typical soils are given in Table III, which was taken from an article by Larkin.[20]

TABLE III

THE UNDAMPED NATURAL FREQUENCY OF SOME TYPICAL SOILS[20]

Material	Loading $\#/ft^2$	Undamped Natural Frequency - CPM.
Clay (wet)	2000	540
Clay (wet)	4000	380
Clay (dry hard)	2000	660
Clay (dry hard)	8000	330
Gravel	2000	1330
Gravel	8000	660
Sandstone	2000	4200
Sandstone	8000	2100

Causes of Vibration: The causes of vibration spring from "free inertia forces" due to the unbalance of the moving components of a machine. A not-perfectly balanced motor shaft is a cause of simple vertical translation vibration of the first order (f = 1 x shaft rpm).

The eccentricity in bearings can cause vibration. A motor driven by common 60 cycle A.C. current will deliver a constantly changing motor torque, with peaks and valleys causing a torsional oscillation or vibration of 7,200 c.p.m.

See the discussion of unbalanced internal mechanical forces (page 35).

VI - MAKING ALLOWANCES FOR INTERNAL MECHANICAL FORCES IN THE DESIGN OF FOUNDATIONS FOR MACHINERY

A - Designing of an Elastic Framework for the Support of Machinery.

It is sometimes necessary in the design of machinery foundations to support a machine on an elastic framework. The dimensions of the component parts of this framework cannot be determined alone from the strength required to support the static weight of the machine.

The framework must be designed by taking into account these static loads, the kinetic forces due to the moving parts of the machine, and just as important, the possibility of vibrations.

When any machine is supported on an elastic framework, its motion causes vibrations. These will become excessive when the speed of the machine becomes nearly equal to the natural frequency of the parts of the supporting structure.

It is, therefore, necessary in the design of the framework, to be certain that the proportions are such that the natural frequencies are not near the operating speed of the machine.

A-1. Design of Beam Supports. To explain the design of beams, suppose a machine of weight W is located at the center of a simple beam of span L. (figure 16) If for a first approximation,

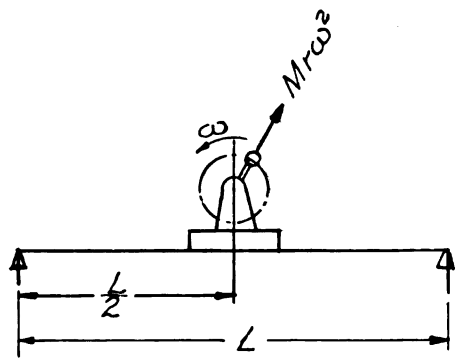


Figure 16

the weight of the beam is neglected, the deflection at the center of the span is:

$$\Delta = \frac{WL^3}{48\ EI}$$

the natural period P of free vibration of the beam:

$$P = 2\pi\sqrt{\frac{\Delta}{g}} \qquad (19)$$

where Δ is in inches and $g = 386$ "/sec^2

Inserting into this the expression for deflection, P becomes:

$$P = 2\pi\sqrt{\frac{WL^3}{48\ EIg}}$$

If N represents the revolutions per minute of the machine, then as the frequency in cycles per second = the revolutions per second;

$$N = 60f \quad \text{or } N = \frac{60}{P}$$

The speed at which the amplitude of vibration of the beam will be a maximum or the critical speed is:

$$N_{critical} = \frac{30}{\pi}\sqrt{\frac{48EI}{WL^3}\,g} \quad (21)$$

where E, I, and L are in inches, W is in pounds and N is in revolutions per minute.

The critical value of N must be calculated by the above formula and if the operating speeds are anywhere close to the resulting N critical, the design must be changed.

An approximate method for including the weight of the beam in a formula for N critical follows:

The deflection now consists of two parts. The part due to the weight of the machine W.

$$\Delta_1 = \frac{WL^3}{48EI}$$

and the part due to the uniformly distributed weight of the beam wL,

$$\Delta_2 = \frac{5wL^4}{384EI}$$

the total deflection at the center of the span is therefore:

$$\Delta = \Delta_1 + \Delta_2 = \frac{L^3}{48EI}\left(W + \frac{5}{8}wL\right)$$

which is the same as for a single concentrated load of amount $(W + \frac{5}{8}wL)$. Assuming the natural frequency of vibration of the beam is the same for a single concentrated load as when the load is partly uniformly distributed the formula for critical speed becomes:

$$N_{critical} = \frac{30}{\pi}\sqrt{\frac{48EI_g}{(W + \frac{5}{8}wL)\,L^3}} \quad (21a)$$

(The correct solution, given by S. Timoshenko shows that 5/8 should be replaced by 17/35)

In the design of a beam to avoid synchronism a working condition must be assumed, (say N critical = 10 N), and the size of the beam determined accordingly. The procedure could be to first, neglect the weight of the beam and solve for I. Second, using this I, calculate the dimensions and weight of the beam. Then as a check, the critical speed could be redetermined using (W + 17/35 wL) for the concentrated load at the center.

A-2. <u>Design of Column Supports.</u> To explain the procedure in the design of columns to prevent the synchronism discussed above, consider a weight W supported on a column of length L. (figure 17)

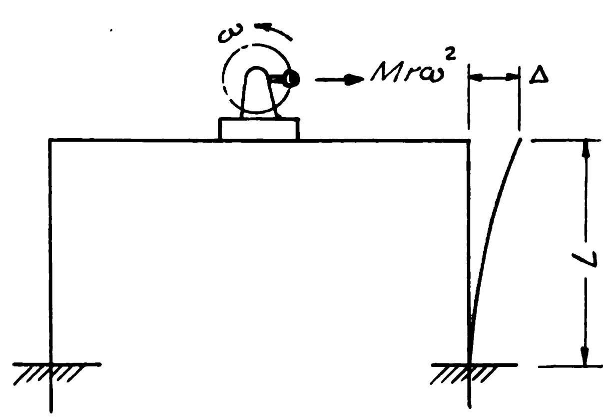


Figure 17

For a first assumption neglect its weight and consider the column as deflecting as a cantilever beam under the action of the components of kinetic forces of the machine acting horizontally. If F_x represents the horizontal components acting on one column, the deflection laterally of its upper end will be:

$$\Delta = \frac{F.L^3}{3\,EI}$$

Because F_x is proportional to Δ we know the motion is simple harmonic and the fundamental equation may be used.

$$m\,\frac{d^2x}{dt^2} = -\,K_x$$

(K is the disturbing force at unit distance from upright position). In this case K is the value of H for $\Delta = 1$. Therefore:

$$K = \frac{3EI}{L^3}$$

The period of harmonic motion is: (m being the mass of the body)

$$P = 2\pi\sqrt{\frac{m}{K}}$$

and the frequency is

$$f = \frac{1}{P} = \frac{1}{2\pi}\sqrt{\frac{K}{m}} = \frac{1}{2\pi}\sqrt{\frac{3EIg}{WL^3}}$$

then the critical speed for the column becomes

$$N_{critical} = \frac{30}{\pi}\sqrt{\frac{3EIg}{WL^3}} = 325\sqrt{\frac{EI}{WL^3}} \qquad (22)$$

where E, I, and L are expressed in inch units and W in pounds.

If the weight of the column is included in the calculations, an approximate formula is found as follows: The deflection of a cantilever beam under uniform load is:

$$\triangle = \frac{wL^4}{8EI}$$

and the total deflection under both a concentrated load W and a uniform load wL will be:

$$\triangle = \frac{WL^3}{3EI} + \frac{wL^4}{8EI} = \frac{L^3}{3EI} \quad (W + 3/8\ wL)$$

which is the same as a concentrated load of (W + 3/8 wL). Assuming that the frequency of the column is the same for a single concentrated load as when part of the load is uniformly distributed, we have for N critical approximately:

$$N_{critical} = \frac{30}{\pi}\sqrt{\frac{3EIg}{L^3(W + 3/8\ wL)}} \qquad (22\ a)$$

(By Timoshenkos' rigorous analysis the 3/8 wL becomes 33/140 wL)

In the design of a column the procedure for determining that it will not synchronize will be similar to that described above for the beam.

B - Designing Machine Foundations to Prevent or Minimize the Transmission of Vibrations.

The prevention of vibration in the parts of a machine, its structural members, or in the structure of the building is important; in eliminating excessive wear, in reducing repeated stresses that are likely to cause the failure of a member by fatigue, and in reducing objectionable noise.

One of the problems which therefore faces the machine foundation designer is to find a way of reducing the vibrations which may be present. There are several possibilities to consider in attacking this problem:

1. By Balancing. An attempt might be made to balance the machine and thus remove the exciting force. This, however, is the job of the machine designer and usually has been done to the best of his ability, although it must be understood as mentioned on a previous page that it is not always practicable or possible to completely balance a machine. In any event this is a condition over which the foundation designer has no direct control.

2. By Tuning. In certain instances excessive vibrations in a machine could be prevented by so changing the design of the machine, that it will not operate near its critical or resonant speed. An example of how this worked in one instance is given in Mr. DenHartogs' book - "Mechanical Vibrations". Excessive vibrations were eliminated in this case by replacing the existing 17 bucket runner of a hydraulic turbine by a 16 bucket runner, thus changing the interval of time between the impulses of two adjacent guide vanes. As in number 1, the foundation designer seldom may resort to expediencies of this sort.

3. By Damping. It is sometimes possible to control vibrations by damping or by the introduction of frictional forces in order to reduce their amplitudes. Damping has

little effect except in the neighborhood of the resonant frequency. An example of damping is the use of shock absorbers as friction dampers on an automobile, to limit the resonant vibrations of the body of the car induced by road irregularities. In a machine anchored rigidly to a massive fixed foundation, the mass of the foundation, acts as an inertia damper to limit the amplitude of the vibrations.

4. By Isolation. A method may be applied by which the vibration is so isolated that the periodic force reaction on the soil or supporting medium is reduced. The usual method of isolation is to use some form of elastic suspension of the vibrating body.

We shall limit this discussion to methods 3 and 4 above.

B-1. Massive Foundations as Inertia Dampers.

The use of heavy foundations is the simplest and most primitive means of providing resistance to the kinetic reactions arising from the moving parts of a machine. Obviously this type of design cannot be used in many cases, but where they can be used they serve effectively to decrease the amplitudes of the foundations forced vibrations. They are never efficient, however, from the standpoint of power losses and strain on the machine and foundation, often causing failure of the metal and disintegration of concrete foundations, by fatigue failure.

It must be noted that in this type of design the theory will not be to attain an amplitude of zero, because such an amplitude would require an infinitely large mass, and unless a small amount of vibration is allowed strains and resulting damage are likely to occur.

It is unscientific as well as uneconomical to mount a precisely designed machined machine on a foundation whose design has been mostly guess work. Although this seems to be the prevalent method of attack, a more scientific method of approach to the problem is now given, illustrated by the same single cylinder engine discussed in article 5-B.

The periodic inertia force representing the sumation of all the forces caused by rotating and reciprocating parts, is resisted jointly by the inertia of the masses to which it is transmitted including the mass of the machine, the mass of the foundation, and the mass of that part of the soil or sub-foundation, which may be assumed to act as a unit with it.

Thus it is seen that the practice of anchoring a machine to a massive foundation is based on the principle of using the relatively small accelerations set up by a large mass, namely foundation and underpining, to balance the large accelerations of relatively small masses, namely the moving parts of the machine. For this reason the soil on which a foundation rests usually adds greatly to its effectiveness, for the mass of soil is also accelerated just as far as the disturbance transmitted to it by the foundation extends.

Let K denote the ratio of the soil accelerated to the mass of the foundation proper, and let (a) denote the average acceleration for the entire mass set in motion. Then the condition for equilibrium against horizontal translation in the present case is: (from for. 17)

$$\frac{W_m + W_f + KW_f}{g} . a = \frac{W_1}{g} r\omega^2 + \frac{W_2}{g} r\omega^2 \left(1 + \frac{1}{q}\right) \qquad (23)$$

Since the motion of the foundation is periodic with the same period as the engine speed, it is a sufficiently close approximation to assume that it is harmonic. If the amplitude of this harmonic motion is 2b then $a = b\omega^2$. Substituting this value, canceling out the common term ω^2/g and solving for the required weight of the foundation, W_f we have:

$$(W_m + W_f + KW_f) \cdot b \cdot \frac{\omega^2}{g} = W_1 r \frac{\omega^2}{g} + W_2 r \frac{\omega^2}{g} \cdot \left(1 + \frac{1}{q}\right)$$

$$W_f (b + Kb) = W_1 r + W_2 r \left(1 + \frac{1}{q}\right) - W_m b$$

$$W_f = \frac{r}{b(1 + K)} \left[W_1 + W_2\left(1 + \frac{1}{q}\right)\right] - \frac{W_m}{1 + K} \qquad (24)$$

The horizontal reaction applied to the bedplate of the engine is accompanied by a vertical overturning couple acting on the foundation in the plane of the motion. In this case however the effect of this couple is unimportant in comparison with the lateral motion due to the horizontal reaction.

Example: In an engine of the Corliss type, having the approximate dimensions given below, the lateral motion must be limited to 0.005" in either direction from rest position. Find the required foundation weight.

Solution: Substitute in formula 24 above, (assume K = 10)

Weight of rotating parts = 150#

Weight of reciprocating parts = 400#

Total weight of engine = 12 tons

Speed = 120 rpm

Length of connecting rod = 5'

Length of stroke = 20"

$$W_f = \frac{10"}{.005"(1 + 10)} \left[150\# + 400\# \left(1 + {}^{1/60}/_{10}\right)\right] - \frac{24000}{1 + 10}$$

$= 181.82\ (150 + 466.67) - 2181.82$

$= 112,171 - 2182 = 109,989\#$ say 55 tons.

By the method just described we can determine the mass of block required for a certain allowable amplitude. The uncertainty involved is in the amount of subsoil that should be included in the mass subject to vibration. This is variable due to the type of soil, the unit loading, whether the foundation sets on or in the subsoil, and other factors. Certainly the choice of K would require some experience.

The theory does show us that the amplitude of vibration varies directly as the unbalanced inertia force, and invercely as the mass subject to vibration. Therefore, it is clear that any means of increasing the mass subject to vibration will reduce the amplitude of vibration, such as tamping, the use of piles, etc.

B-2. Isolation by Suspension of the Machine.

A more modern method of dealing with the kinetic forces involved in moving machinery is to mount the machine on some type of resilient mounting, or to mount the machine plus some massive foundation on a resilient mounting.

It is now necessary to get a little background before moving into the actual design procedure.

In studying this type of mounting, we shall confine ourselves to dealing with forces which originate within the machine itself. There are two types of problems with which we are concerned.

First the reduction of transmitted vibrations in the form of a wave motion, and second, the consideration of impact shock, which may occur at such infrequent intervals that it can hardly be classed as a wave motion.

A resiliently mounted piece of equipment has six degrees of freedom, that is, motion of a greater or less amplitude is possible in as many directions. Three of these directions are of translational form in three separate planes; three are of a rotational nature about three separate axis.

These two types of vibration can be exemplified by a motor mounted on resilient mounts. The translational type motion could take place in a plane thru the shaft of the motor. The most obvious evidence of the rotational type of vibration is about the shaft axis of the motor. The word freedom used in this sense should be understood to be purely relative and accordingly, motion might be brought about with much less effort in one direction than in another. Moreover, the fact that a degree of freedom exists in a given direction does not in any sense imply that forces or moments actually prevail to cause movement of this character.

Naturally, the support will sustain a steady deflection due to the weight of the machine supported; then as vibratory impulses are set up, oscillating motion occurs on both sides of this staticly deflected position. The effect of a vibratory force exerted from within is to accelerate the machine in the instantaneous direction of the force and it is resisted but little by the comparatively soft resilient mountings. But, before the distance thru which the machine moves becomes very great, the vibrating force has changed direction; accordingly the only part of the disturbing force transmitted to the frame is that small amount involved in deflecting the resilient mountings a distance corresponding to the motion of the machine.

Vibratory forces are transmitted directly from the body generating the forces to the supporting foundation, if the two are rigidly attached. However, if the same vibratory force or forces could be made to appear in a body completely free in space, the force would be resisted solely by the inertia of the body and the body would vibrate only slightly. In practice this free condition is approximated by placing the body on resilient mountings.

The mountings must be sufficiently flexible so that the vibratory forces are resisted by the inertia of the body and thus reduce the oscillatory motion of the body.

Although a large portion of the vibration may be absorbed by the mountings, small forces will act on the supporting structure. This portion is a function of the ratio between the disturbing frequency and the natural frequency of the isolated system. The effectiveness of the resilient mounting can be obtained from this relationship.

$$T = \text{Transmissibility factor} = \frac{1}{\left(\frac{\text{disturbing frequency}}{\text{natural frequency}}\right)^2 - 1}$$

$$T = \frac{1}{\left(\frac{F}{f_n}\right)^2 - 1} \qquad (24)$$

Solution of the above will give the portion of the vibrating force transmitted as compared to that if a solid support was used. This percentage subtracted from 100 will give the efficiency of the mounting.

For effective isolation of vibration the ratio between the disturbing frequency and the natural frequency must be greater than $\sqrt{2}$. Isolation becomes more efficient as the ratio becomes greater. As the ratio of F/f_n becomes smaller than $\sqrt{2}$, a magnification of forces will occur.

This condition, spoken of as resonance, would be infinitely worse than if no insolators were used, so it is important to keep out of this dangerous range.

The natural frequency should be less than 1/2 preferably 1/3 of the forced frequency.

At ratio of 2:1 it is possible to obtain 66% eff.

At ratio of 4:1 it is possible to obtain 93% eff.

We are now prepared to pick an isolator and complete the formula. The disturbing frequency (discussed) of the equipment is usually the operating frequency of greatest amplitude. This can be found from the number of revolutions per minute or the number of impacts per minute.

The natural frequency of the isolator depends on its deflection under static load, and this can be found by formula (19b)

$$f_{cmp} = 188\sqrt{\frac{1}{\Delta}}$$

For example: a certain isolator defects 1/16"

$$\therefore \quad f = 188 \frac{1}{.0625} = \frac{188}{.250} = 750 \text{ cpm}$$

if the operating frequency is 1500 cpm, the frequency ratio would be $^{1500}/_{750}$ or 2.

The transmissibility (formula 24) would be 0.33. Therefore, the vibration absorption would be 67%.

TABLE IV

RELATIONS BETWEEN DISTURBING FREQUENCY AND DEFLECTION

Disturbing frequency cpm	Reqd. Deflection of Isolation Medium	
	For 65% eff.	For 93% eff.
4240	1/100 in.	1/32 in.
1220	1/10 in.	3/8 in.
752	1/4 in.	1 in.
432	3/4 in.	3 in.

The next step is to select the proper mounting. The merits of each type should be studied, and the best one for your problem chosen.

Some of those in greatest use today are: Cork, Balsa wood, Felt, Rubber, Timber, and Steel Springs.

Various considerations limit the choice. From a study of Table IV, it is evident that low frequency disturbances and high vibration require large deflections of the elastic medium. Deflections of ½" are difficult to obtain with organic materials unless one goes to complicated constructions. Therefore, we see that rubber, cork, felt and like materials have a definitely limited region in which they can work*

* (Rubber shear loaded mountings are good to get large deflections.)

Steel springs particularly of the coil spring type also readily provide the large deflections necessary for low frequency disturbances. They are easily adaptable to all purposes because their elastic properties can be accurately predetermined and controlled thru a wide range of dimensions and combinations. It should be kept in mind that it is necessary to design a spring suspension in such a manner that all six degrees of freedom are provided for.

Certain of the materials mentioned above are not too well suited for machinery foundation isolation. Rubber is adversely affected by oil; it is expensive; it may take a permanent set; is not permanently elastic and is likely to become hardened due to exposure; it has therefore not found much use on foundation isolation for permanently placed machinery.

Felt may in time take a permanent set; it is not waterproof; it is absorbent; it may be attacked by insects.

Untreated granulated cork may in time pack down and unless confined tends to flow.

Timber is not waterproof, when used alone in small quantities; it does not have sufficient deformation to make it useful under most services; it may be affected adversely by oil. However, in the past, treated timber has been widely used for cushions under forging hammer foundations.

It is apparent from the above that choosing an isolating material calls for a certain amount of experience and wisdom.

A simple example in which a choice of materials has to be made, follows:

<u>Example</u>: A motor generator set operates at 1750 rpm. We have a choice between three mountings, that undergo static deflections in all mounting points of 1/32 in., 1/16 in. and 3/32 in. respectively. Which is the best choice?

By the use of formulas (19b) and (24) it is apparent that the first is critical, the second provides 65% isolation and the third provides 85% isolation. Obviously the third will be the best.

It is here necessary to point out a very important fact, oftentimes forgotten in the design of isolations for a foundation, namely that the isolating layer may not be inserted at any arbitrary place, for, as already mentioned the amplitude of vibration depends on the ratio of the mass of the moving parts to that of the foundation and fixed parts. Since inserting a resilient layer diminishes the effective mass of the foundation, these layers should, therefore, be placed at such a depth that a machine will still be attached to a sufficiently heavy foundation mass. Moreover, inserting a resilient layer has the effect of raising the center of gravity of the machine and therefore affects its stability, which must also be taken into account in determining the position of such a layer.

<u>Impact Shock Isolation</u>

In dealing with the reduction of transmitted vibrations most discussions are concerned mostly with the general principle of vibration isolation which deals with vibration in the form of wave motion, and do not consider separately the effect of an impact shock. A heavy impact blow may occur only at distant or infrequent intervals, and in itself may be damaging to the mechanical equipment and foundations, although its frequency is so low or irregular, that it cannot be treated as a wave motion.

In shock protection, the fundamental principle is to increase the period over which the impact forces are applied. Rubber in compression is recommended where shock protection is paramount, because of the continuously smooth deflection curve which presents no opportunity for shock to be re-created.

The Fabreek Products Company of Boston, manufacturers of a pad made of layers of rubber and impregnated cotton duck, that have been vulcanized together, offers the following Table V, and I quote their example of the design for pads under a typical shock mounting.

TABLE V

Stiffness Coefficients

Fabreek Products Company

	#per sq. inch, per inch of deflection				
Loading in psi	50	100	150	200	400
14 ply 1/4" thick	45,000	50,000	54,000	58,000	71,000
17 ply 9/32" thick	36,000	40,000	43,000	46,000	57,000
21 ply 11/32" thick	28,000	32,000	34,000	37,000	44,000
31 ply 1/2" thick	18,000	20,000	22,000	25,000	29,000
39 ply 5/8" thick	14,000	16,000	18,000	20,000	23,000
Double 39 1-1/4"	7,000	8,000	9,000	10,000	11,000

From the Fabreeka Company Bulletin:

"The force of an impact is measured by the work done; that is if a 2000# weight drops 2' the work done is 2000 x 2 = 4000'#. To find the force this blow exerts it is necessary to know the distance in which it is brought to rest. Assuming 1/16" (or .0052') then $4000/.0052$ = 770,000# is the dynamic load.

If the 2000# wt. dropping 2" is a board hammer and the deflection (1/16") caused by the hammer, is the deflection of the work being done, then the dynamic load on the base of the anvil is 7 70,000#. If area is 2000 sq." the unit loading would be 400 psi. If two Fabreeka pads each 5/8" thick are placed under this base the deflection of the pads is .060 or .050 feet which added to .0052 = .0102'. The dynamic load is now $\frac{4000}{.0102}$ = 392,000#, decreasing it 49%

The above assumes a rigid foundation. On soft marshy soil the necessity of pads would be reduced, because the ground would be absorbing some of impact.

Also pads increase efficiency due to damping effect."

VII - GENERAL FOUNDATION DESIGN DETAILS

After the dimensions of a foundation base have been determined by some means or other depending upon the loadings and allowable pressures on the soil, or supporting medium, there still remain some questions which must be cleared up before the foundation design may be termed complete. Such details as foundation material, reinforcing steel, anchor bolts and plates, grouting, etc. must be considered. In a rather brief treatise of this kind none of these topics can be covered in more than a fragmentary fashion, but they are important and must be mentioned if but briefly.

A - Materials

Machinery foundations, at least those where some mass is required, are made almost universally of concrete. It is important to specify a concrete with a low volume change and high unit strength. It is also important to get a good strong uniform concrete, as the constant vibration of the attached machine will subject the concrete to a rather severe life.

In regard to the shape of the foundation, it will often be found that a simple forming job will be more economical than a complicated forming job which may require a lot less concrete. To provide additional bearing area than is available when the block is made about the size of the base plate, it will be found most economical to "step" the foundation, as in any wall or column foundation.

B - Reinforcing Bars

With the exception of a cantilever footing slab or on foundations which are hollow, there are no calculated stresses which must be carried by reinforcing steel, but it is my belief that its use in nominal quantities is necessary to prevent cracking. Cracks which would cause no concern in ordinary concrete construction are often serious in machinery foundations, especially where there exist vibrations, which may cause progressive cracking. Larkin[20] advises using deformed intermediate grade steel reinforcing bars $\frac{1}{2}$ or 5/8 inches in diameter spaced on 12" centers, both horizontally and vertically near all faces of the foundation block. Other authors try to prove the steel is not necessary, but I think that the general practice is to put in steel bars.

C - Anchor Bolts

Anchor bolts are usually made up locally and therefore do not receive the constant improvement and development which a manufactured product does. The calculation of stresses and the design of bolts is a relatively simple procedure as they are usually only required to take a tension load. A minimum diameter bolt would be 5/8" as a smaller one could be twisted in tightening the nut. The size of bolt required for a certain machine will usually be specified by the manufacturer. A few important things about anchor bolt installation will now be mentioned.

For heavy machines the anchor bolts are usually made removable so that it is not necessary to lower the machine over the bolts. Of course, this necessitates some kind of pockets. For most light machines the bolts are not made removable, but are installed in such a way as to allow for a certain amount of springing to align with the holes in the machine bases, because a carpenter cannot set bolts to the tolerances usually held on drilled holes or castings.

A good design has some provision such as a pipe at the top with the bolt inside to provide for a small amount of springing. There should be some means of holding the pipe as well as the bolt in place while the concrete is being placed.

The only other recommendation is that the bolts should be relatively long so as to offer a certain amount of resiliency against breaking.

D - Grouting

For obvious reasons the foundation is not built quite as high as the elevation to which the base of the machine must set. The difference, (3/4" to 1½") is made up by leveling the machine and pouring or forcing a 1:1 mixture of portland cement grout between the foundation and the machine base. The grout is required to take the horizontal loads into the foundation, as well as to level up the machine. Other materials are sometimes used for grouting, but portland cement is by far the most widely used.

E - Design of Concrete Mixtures

Concrete is a mixture of cement, fine aggregate, coarse aggregate and water. It is a generally accepted fact that the strength depends entirely upon the ratio of water to cement, provided (1) that the cement and aggregates used are suitable, (2) that the amounts of aggregates are correct to give a workable mix, (3) that proper conditions are maintained during the hardening period.

Proportioning by weight is admittedly the most accurate because of bulking of the aggregate, however, proportioning by volume is by far the most popular method due to the fact that expensive weighing equipment is not required, and it can be done with reasonable accuracy if allowance is made for bulking. In view of this fact, in the following discussion when proportions are mentioned they will be given by volume, so that a 1:2:4 concrete implies: one cubic foot of cement to 2 of fine, and 4 of coarse aggregate.

The first and most important ingredient of concrete is portland cement. All of the manufacturers of cement in this country turn out a good product so all that need be said is that a cement be used which is not more than a year old and has been kept dry. In certain instances it may be wise to build the foundation using High-early strength portland cement which would allow its early use. At 72 hours it would have the strength ordinarily obtainable at 28 days. The water must be clean and free from oil, and alkali, organic matter or other deleterious substances.

The fine aggregate, usually sand, should be well graded and free from organic matter, clay and silt. (See ASTM specification.)

The coarse aggregate, washed gravel or crushed stone, should be clean, hard, and durable; free from alkali organic or other deleterious matter. Coarse aggregate includes all particles larger than 1/4 inch. It is suggested that for foundations the maximum size be one inch. As with fine aggregate it is important that they be well graded. (See ASTM specification.)

Concrete Mixtures. In general the design of a concrete mix consists in choosing a water-cement ratio, and then selecting the proportions of fine and coarse aggregate so that the combination will be workable. The choice of a water-cement ratio in general requires the consideration of strength, exposure, and class of structure.

The selection of a proper water-cement ratio for any desired strength under average job conditions may safely be made from the specification for strength of the American Concrete Institute, which follows the straight line empirical formula $S_c = 1000(7 - \frac{2}{3} R)$. Where S_c is the compressive strength of the concrete in 28 days (in pounds per square inch), and R is the water-cement ratio (U. S. gal. water per sack of cement (94#)). Thus, an ultimate compressive strength of 3000 psi would require a water-cement ratio at 6.

To select the necessary water-cement ratio, however, is more a matter of judgement, as the actual compressive strength required in a machine foundation is probably a rather small figure. It is noted, however, that as the compressive strength of the concrete increases so does its density and its ability to withstand vibrations without disintegration. As a result if one chooses a strength requirement somewhere between 2500 and 3000 psi, (the lower for machines relatively free from vibration or impact forces and the higher for machines with excessive vibrations or impact), the result should be satisfactory.

A workable concrete mix is one which can be worked into all the corners of the forms without excessive puddling, without segregation of the ingredients, and without water collecting on the top surface. Workability is influenced by the amount of each ingredient and by the nature of the aggregates.

Some measure of the workability of a concrete mixture is obtained by making a slump test. (See ASTM specification). For most concrete machine foundations a slump of from 3 to 5 inches is satisfactory.

Having selected a water-cement ratio and knowing the approximate slump desired, the next step is to determine the proper amounts and proportions of the aggregates. This can best be done by making several trial mixes, recording the proportions and choosing the most workable. A balance must be maintained between the coarse aggregate which will up to a certain point reduce the cement factor and beyond that will produce an under-sanded harsh concrete,(difficult to place,) and the fine aggregate which produces a smoother mix but which used in excess, makes the concrete too expensive.

For these small trial mixes 1/10 of a bag of cement (9.4#) may be conveniently used and the aggregates surface dried so that corrections for moisture will not be required.

All aggregates under job conditions contain more or less moisture which must be calculated in with the mixing water. The amount of moisture may be found by drying a sample of the aggregate to a constant weight. If w = weight of the damp sample, w' = weight after drying and p = the percentage of total moisture:

$$p = 100 \frac{w - w'}{w}$$

The moisture which becomes a part of the mixing water is this percentage p (above) minus the percentage which will be absorbed by the aggregates, about 1% for average sand pebbles and crushed limestone, o.5% for traprock and granite.

Moisture in sand and even in coarse aggregate has the effect of bulking the aggregate, therefore calculations must be made to determine the "bulking factor" or relation between the damp volume and the dry volume of a given quantity of dry aggregate. Essential computations are given in the example below:

Item	Sand	Coarse Aggregate
A. Weight of damp samples, oz.	35.0	32.5
B. Weight of ovendry sample, oz.	33.0	31.7
C. Weight of water in damp sample, oz. (A - B)	2.0	0.8
D. Per cent of total moisture in terms of dry aggregate (C/B)X100)	6.0	2.5
E. Per cent of absorption (assumed)	1.0	1.0
F. Per cent of surface moisture (D - E)	5.0	1.5
G. Weight per cu. ft. damp, loose, lb.	97.2	94.4
H. Weight of surface-dry aggregate in 1 cu. ft. of damp loose material, lb. (G X(100/100+F)	92.6	93.0
I. Weight of water in 1 cu. ft. of damp loose material, lb. (G-H)	4.6	1.4
J. Weight per cu. ft. of surface-dry compact aggregate (by test)	112.0	99.0
K. Bulking factor (J/H)	1.21	1.06

The correction for bulking is made by adding proportionately larger amounts of the bulked aggregate to secure any desired actual volume of dry, compact aggregate. Thus, if a dry, compact mix of 1:2¼:3½ is to be used with the above aggregates, the proportion of sand, based on loose volume, will be 2¼ x 1.21 = 2.7, and of coarse aggregate 3½ x 1.06 = 3.7, giving a field mix of 1:2.7:3.7.

If the water-cement ratio for the above mix is to be 7 gal. per bag, the amount of water to be added at the mixer is determined as follows: Since 1 gal. of water weighs 8-1/3 lb., the amount of surface moisture in 1 cu. ft. of damp, loose sand is 4.6/8.33 = 0.55 gal., and in the coarse aggregate 1.4/8.33 = 0.17 gal. In each one-bag batch the amount of free or surface water in the sand is 0.55 X 2.7 = 1.48 gal., and in the coarse aggregate 0.17 X 3.7 = 0.63 gal. The amount of water to be added to each 1-bag batch is, therefore, 7 - 1.48 - 0.63 = 4.89 gal., or approximately 5 gal.

When the proportions have been decided upon the determination of yield and quantities required is the next task.

The yield is the sum of the absolute volumes of the cement, sand, and coarse aggregate, and the volume of water. For the mix computed in the above example, the yield for a one-bag mix would be computed as follows:
There is 0.487 cu. ft. of solids in 1 cu. ft. of cement. The specific gravity of the common aggregates usually averages 2.65. One cu. foot of dry sand contains 112.0/2.65 x 62.4 = 0.68 cu. ft. of solids, and one cu. ft. of dry coarse aggregate contains 99.0/2.65 x 62.5 = 0.60 cu. ft. of solids. For a 1-bag batch, which contains 7 gal. (or 7x0.134 = 0.938 cu. ft.) of water, including the surface water in the aggregates, the yield will be $0.487+2\frac{1}{4}x0.68+3\frac{1}{2}x0.60+0.938 = 5.06$ cu.'

If 200 cu. feet of concrete were required for the job the quantities required would be determined as follows:

No. of 1-bag batches required = (200x27/5.06) = 1068	
Cement	= 1068 bags
Sand = (1068 x 2.7) / 27	= 108 cu. yds.
Coarse aggregate = (1068 x 3.7) / 27	= 148 cu. yds.
Water - at mixer = (1068 x 5)	= 5340 gal.

To provide proper curing conditions the concrete should be protected against premature drying out for at least one week.

VIII - VARIOUS DESIGN CONSIDERATIONS DEPENDING ON THE TYPE OF MACHINE.

In a study of this kind, one can well generalize up to a certain point, but then he finds that each machine is an individual problem. With this in mind the following rather incomplete remarks on the specific requirements of individual machines have been included, together with the references for a more complete discussion.

A - Machine Tools (12)(7)

1. The principle requirement is rigidity, in order to maintain accuracy of operation. (requires massive foundation)

   (a) Planers : Planer tables are very sensitive and may easily be warped enough to make close machining impossible. It is, therefore, important that the foundation be rigid and that the planer bed bear uniformly upon the foundation. No anchor bolts should be used. On large foundations, leveling blocks are usually used. Longitudinal steel is a necessity both in top and bottom because of the length of the foundation.

B - Hammers (7)(15)(32)

1. Besides the weight of the foundation and the hammer, the forces caused by the dropping of the hammer (all of these vertically) must be

distributed to the soil.

2. Excessive noise and shock likely to endanger the building, and produce nervous fatigue, must be guarded against. Usually by insulation. See article B-2

3. Most hammers being used today are of what is known as a high ratio construction. That is the weight of the anvil to the weight of the hammer is high. (15:1 or greater). These require only a light simple foundation, with usually some isolation material to minimize the transmission of jar and noise.

C - <u>Electrical Machinery</u> [7] [33]

1. The elimination of vibration is an important aspect in the design of any high speed machine. (see article B-2)

2. It should be noted that in many cases the electrical machine is only a small part of complex machine. The nature of the other part may greatly alter the foundation required.

3. Some provision must often be made in the foundation for ventilation, particularly in large capacity machines.

4. Table VI gives some typical figures on successful foundations for rotary electrical machinery foundations (taken from "Factory Installation Work" - A. J. Coker.)

TABLE VI

Machine Description	Speed r.p.m.	Ratio of Foundation Wt. to Machine wt.	Total Pressure on ground Tons/sg.′
75 KW belt driven generator	850	2.1	0.34
100 KW belt driven generator	625	3.6	0.51
20 hp belt-drive motor	775	5.1	0.32
45 hp belt-drive motor	950	4.7	0.37
500 KW Synchronous convertor	500	0.51	0.29
900 KW Synchronous convertor	250	0.67	0.35

D - Textile Looms

1. The reduction of shock and noise is of paramount importance. Often a whole bank of looms is insulated together.

E - Steam and Other Reciprocating Engines

1. Besides the static loads, it is necessary to design the foundation to take care of periodic inertia forces occasioned by the rotating and reciprocating parts of the engine. (see article 5 B). The eccentric rotating parts would include the crank pin, crank cheeks, and the crank end of the connecting rod. The reciprocating parts include the piston head, wrist pin, piston rod, crosshead, and the reciprocating end

of the connecting rod.

The general requirements for the foundations for a reciprocating engine are somewhat more exacting than those for rotative machines. Most engine foundations are designed using one of the three following empirical methods:

(a) The weight of the foundation is based on the weight of the machine. (Its weight should be 3 to 5 times the weight of the machine.)

(b) The foundation depth is based on cylinder diameter (the other dimensions determined by the size of the bed plate). This method is used only for simple Corliss engines.

$$L_{hf} = \frac{Kd_1}{4} + 3$$

A formula given by Wm. E. Ninds where:

L_{hf} = depth of foundation in feet

K = constant depending on steam pressures (see Table VII)

di = diameter of cylinder in inches.

TABLE VII

Steam Pressure	K
100	0.895
110	0.938
120	0.98
130	1.02
140	1.06
150	1.10
160	1.13
170	1.17
180	1.20
190	1.24
200	1.27

(c) The foundation weight is based on both the weight and speed of the supporting machine.

A formula is given by E. W. Roberts:

$$W_f = KW_E \times N$$

where:

W_f = weight of foundation in pounds

W_E = weight of the engine in pounds

N = speed of engine in rev. per min.

K = a constant as given in Table VIII

TABLE VIII

Type of Machine	K
4-cylinder vertical gas engine	0.130
3-cylinder vertical gas engine	0.150
2-cylinder vertical gas engine	0.175
Single-crank double-acting tandem	0.320
Double-crank double-acting tandem	0.190
Single cylinder, horizontal semi diesel	0.300
2 - cylinder horizontal semi diesel	0.240
3 - cylinder horizontal semi diesel	0.230
4 - cylinder horizontal semi diesel	0.225
2 cycle horizontal semi diesel	0.230
4 cylinder vertical diesel engine	0.177

2. Steam engine foundations can be designed with safety using one of the above rules. It is usually suggested that they be made monolithically.

3. Compressor foundations can be designed according to rule (a) above.

4. Reciprocating Pump foundation design depends entirely on the type of pump. Generally speaking much less foundation is required for pumps than for steam engines occupying the same space.

Direct acting duplex pumps require the smallest foundation as the plunger motion is almost balanced within the machine. Single-cylinder pumps should be supplied with deeper foundations their depth may be computed from formula given in (b) above. Crank - and - fly wheel pumps require foundations fully as heavy as a steam engine of comparable size, because of the lack of balance in their reciprocating parts.

F - Motor Generator Sets

A small motor generator set, with a bed plate, often requires no more foundation than the ordinary floor.

G - Electric Motors Driving Fans for Ventilation.

Because a ventilating fan must of necessity be located directly in a ventilating duct where noise and vibration may be easily carried throughout the building, the set must be carefully isolated to prevent the transmission of said vibration.

H.- Steam Turbine Units (26) (27) (28) (29) (30) (34)

Steam turbine units are purely rotative, self contained machines, and are not normally subject to unbalanced forces. In the design of a foundation for a steam turbine, however, one should anticipate certain abnormal operating conditions such as short circuits, bad synchronizing, broken blades, water or mud slugs, etc.

It is usually wise to make the foundation structurally independent of its building to prevent vibration transmission.

An excellent answer to the problem of design of steam turbine foundations is an idea originated by N. W. Akimoff of the General Machinery Foundations Company, Philadelphia.

The result of this type of mounting is to free the platform from its support allowing the machine and platform to vibrate as a unit with small amplitude about an axis thru a fixed point of support, without straining either the machine or its support as would be the case with a rigid anchorage. An equally important result is that the possibility of forced vibrations being set up in adjacent structures by means of synchronism is likewise prevented.

Explanation: The usual practice in designing machinery foundations consists in anchoring the machine rigidly to the foundation which is as massive as can conveniently be obtained. The foundation design suggested by Akimhoff is based on the principle that it is desirable to have a certain flexibility in the foundation rather than extreme rigidity. In this design the machine itself and the cap slab which supports it are permitted a limited amplitude of vibration within definitely assigned limits. These limits may be made as small as desired and without in any way decreasing the strength of the structure under static loading. The machine frame and the foundation may be lightened by increasing the allowable working stresses to the values ordinarily used for dead loads. The arrangement consists in mounting the turbo-unit on a structural steel platform which resembles a deck-plate girder. The turbo generator is bolted down rigidly to the platform, but the platform itself is supported at only three points, on the superstructure. Of these three points of support one is rigid and two are resilient. The rigid support is placed at the turbine end. It consists of a rocker plate which is so designed as to allow the platform freedom to pivot slightly about this point. However, the rigid support effectively anchors the platform against any tendency to slide transversely or longitudinally.

The two resilient supports consist of vertical and transverse springs of large capacity, designed so that there is no possibility of their motion being violent.

The design includes vertical and transverse stops so in case of a short circuit or other accident the platform will come to a solid bearing.

With this three point design the level of the platform may be maintained by shimming up one or at most two of the supports.

Mr. Rathbone in an article "Turbine Foundations" in the A.S.M.E. transactions has this to say, among other things, about turbine foundations:

> "Much may be gained as an aide to future designs by vibration studies on existing structures, now made possible thru the use of improved seismic instruments."

NOTE: A list of manufacturers of seismic instruments appears on page 107.

I - Diesel Engines (20)(22)

The design of a foundation and a suitable isolation mounting for a diesel engine, or for that matter any multicylinder engine, presents a rather formidable problem to one unexperienced in its design. There may be unbalanced lateral forces due to unbalanced revolving masses, there may be torques acting to rock the engine in a transverse direction and couples acting to rock the engine longitudinally. All of these may act together giving to the system six different modes of vibration each with its own frequency. For a description of the design procedure for diesel engines see "Design of Diesel Engine Foundations" by Kenneth H. Larkin in A.S.M.E. transactions, March, 1942.

In this same article Mr. Larkin gives some average values for foundation yardage required for diesel engines. His values, given in Table IX below were obtained from recommendations of several engine manufacturers. The values are all within 20% of those recommended and most are within 10%.

TABLE IX

Average Values for Reqd. Foundation Yardage for Diesel Engines

No. of Cylinders	cu. yds./hp	No. of Engines Tabulated
3	0.141	7
4	0.120	10
5	0.108	20
6	0.100	20
7	0.096	14
8	0.091	14

These yardages are the result of years of experience with damping engine vibration by the use of a mass of concrete. They are based on a hard firm subsoil.

It is interesting to note that the above table gives no relationship between foundation size, speed or cylinder size, which items are bound to influence the yardage required.

I also take the liberty of quoting Mr. Larkin in regard to empirical dimensions.

"The depth of foundation should be not less than 4 to 5 times the stroke of the engine, with a minimum of 5 feet. The width of the base should be at least equal to the vertical height from the bottom of the foundation to the center of the shaft, and should be increased for engines having unbalanced horizontal forces". --- "The length of the base is usually so large that no consideration is required in this direction, except that the length should be so adjusted that the center of gravity of the total weight on the subsoil coincides with the c.g. of the area of contact with the subsoil. All of the rules are strictly empirical and when necessary should give way to more important considerations".

IX - DETAIL DESIGN OF FOUNDATIONS FOR CERTAIN MACHINES

There follows two complete foundation designs in which some of the principles heretofore discussed will be put to use. It has already been emphasized that each type of machine has its individual problems, although they have much in common. There is space here only to single out one or two of the many problems for a more complete analysis.

A - Design

Design a reinforced concrete foundation for a large constant speed, non-reversing snycronous motor (shaft drive.). The motor has a rated horse-power of 1200 at 200 rpm. It can operate at a 33-1/3 overload. Figure 18 shows dimensions of the base plate and also the points of application of the loads.

Weights of the component parts are listed below:

Bedplate	27,000#
Rotor	35,000#
Frame	30,000#
Details	2,500#
Pedestals and Shaft	6,400#
	6,400#
Total Weight	107,300#

First step - To choose dimensions of concrete foundation.

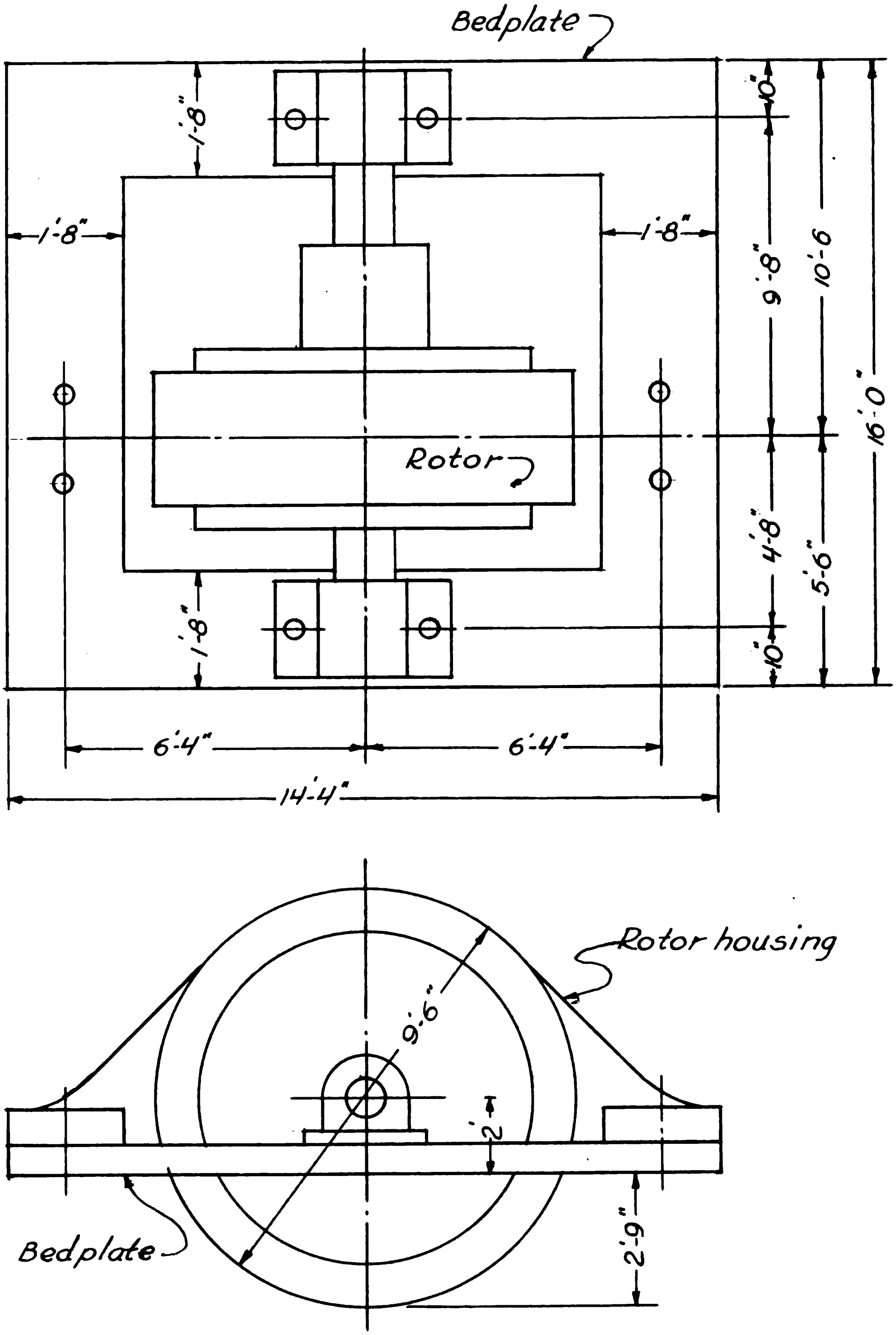
It is usually desirable to allow for the possibility of errors and thus the dimensions on the design should be slightly larger all around than the bedplate. It is also a good idea to bevel the exposed corners so that there will be less chance of their being damaged and thus marring the appearance of the foundation. Corners with 90° angles are much easier to form than obtuse or acute angles so sides if possible should be vertical. The plan dimensions of the foundation must completely encompass the bed plate and two inches will be left all around. The vertical dimension must be great enough to set the foundation on solid soil. In this case a minimum depth will be assumed satisfactory. The soil in this case is a thick bed of clay, it will be drained and kept moderately dry but allowance should be made for the possibility of a small amount of moisture in it.

4000 #/sq. ft. is a good allowable on this soil (see Table I).

The dimensions choosen are shown in figure 19. These will be checked to see if the resulting soil pressures are satisfactory. The requirements are that the maximum pressure must not be too high, that the variation in soil pressures must not be too great, and that the resultant of loads on the soil must pass through the middle 1/3 of the foundation.

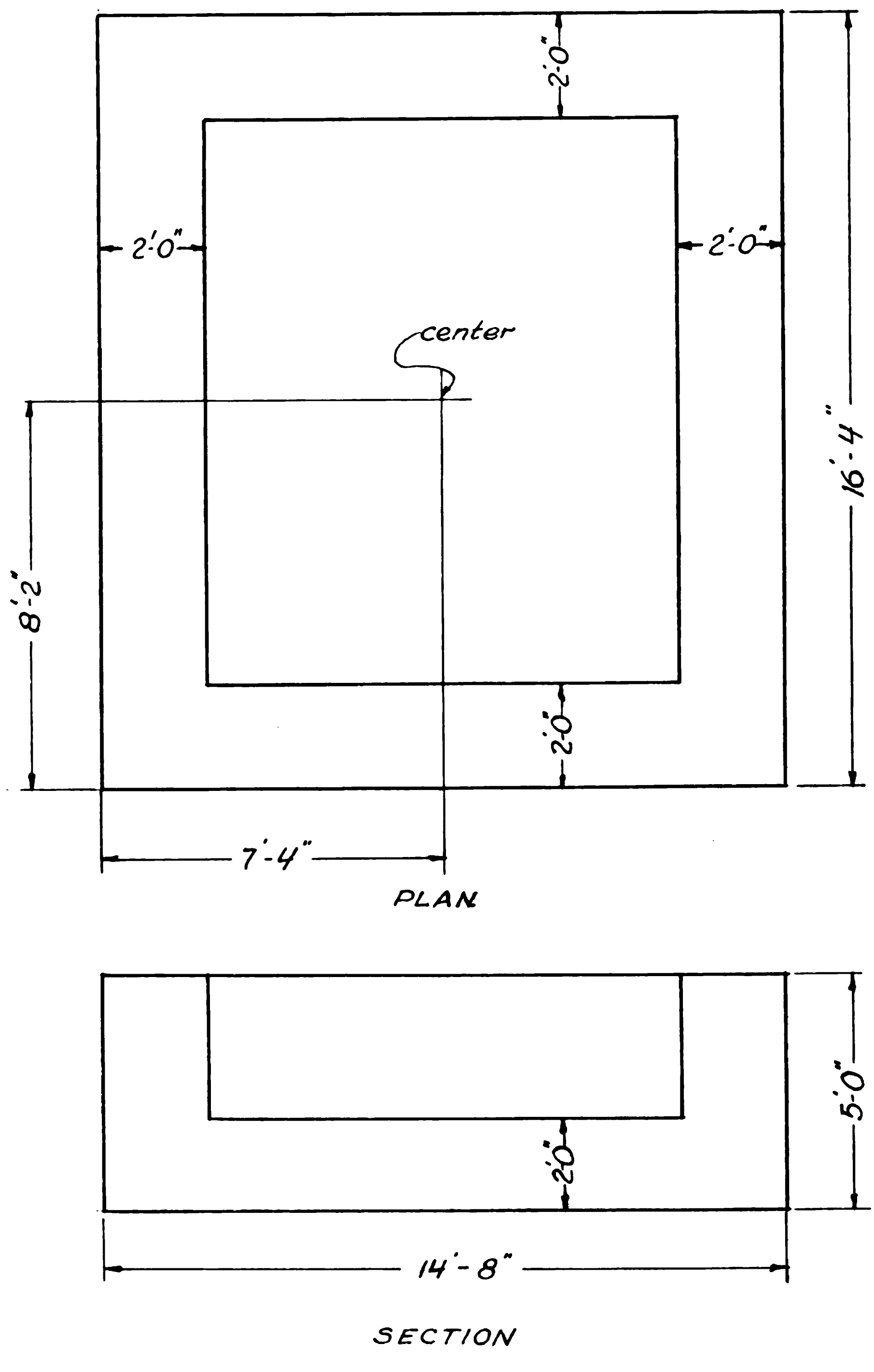
Second step: <u>To calculate the loads at various points on the foundation.</u>

The static loads due to the weights of the component parts of the machine are given on page 91. The distribution of these along with the load caused by the torque of the motor must now be determined.



Sketch showing 1200 HP Syncronous Motor

Figure 18



Chosen dimensions for foundation

Figure 19

(A) To find the torque load to be applied at points Band C (see figure 20)

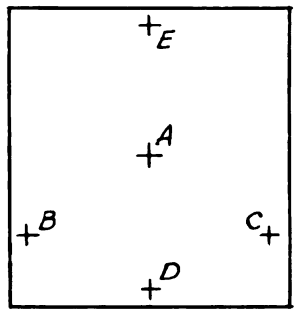


Figure 20

The motor is rated at 1200 hp. but there is a possibility of a 33-1/3 % overload, so in calculating the torque load the motor will be considered as developing 1600 hp.

Applying formula 16, page 33; P, the load caused by the torque is:

$$P = \frac{H.P. \times 5252.1}{N \times r} = \frac{1600 \times 5252.1}{200 \times 6.33} = 6640\#$$

One half of this amount or 3320# is applied at both B and C one up and one down, along with the loads caused by the weights of the motor's parts.

<u>Load at A:</u>

W_f	weight of concrete foundation	120,750#
	weight of bedplate	27,000#
	weight of details	2,500#
	Total at A	150,250#

Load at B:

½ weight of frame	15,000#
½ torque load	6,640#
Total at B	21,640#

Load at C:

½ weight of frame	15,000
½ torque load	-6,640#
Total at C	8,360#

From figure 21 it is seen that the weight of the rotor must be distributed properly to D and E.

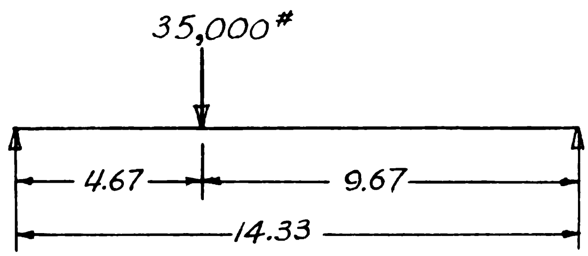


Figure 21

$$R_d = 9.67/14.35 \cdot 35,000 = 23,600$$

$$R_e = 4.67/14.33 \cdot 35,000 = 11,400$$

Load at D:

Proportional weight of rotor	23,600#
Weight of pedestal	6,400#
Total at D	30,000#

Load at E:

Proportional weight of rotor	11,400#
Weight of pedestal	6,400#
Total at E	17,800#

Third step - To calculate the stress at the maximum and minimum points.

(A) The centroidal axis are the geometrical axis of the foundation.

(B) Find: I_x and I_y.

$I_x = 1/12\ bh^3 = 1/12 \cdot 14.67 \cdot 16.33^3 = 5323.6$

$I_y = 1/12\ bh^3 = 1/12 \cdot 16.33 \cdot 14.67^3 = 4296.3$

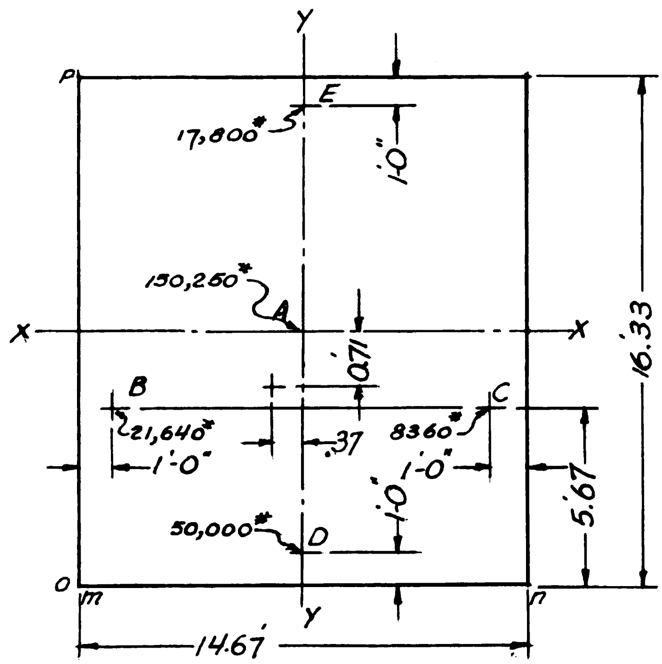


Figure 22

(C) The product of inertia = 0.

(D) Find the resultant of all the loads W_t and the point through which it acts.

The resultant is the sum of all the loads:

$$
\begin{array}{rr}
 & 150,250 \\
 & 8,360 \\
 & 21,640 \\
 & 30,000 \\
 & \underline{17,000} \\
W_t = & 228,050\#
\end{array}
$$

Take moments about m-n:

$$228,050 \cdot y = 21,640 \cdot 5.67 + 8,360 \cdot 567 + 150,250 \cdot 8.17$$
$$+ 30,000 \cdot 1 - 17,800 \cdot 15.33 = 1,700,517$$

$$228,050 \cdot y = 1,700,517$$
$$y = 7.457$$
$$e_y = 0.71$$

Take moments about O-p:

$$228,050\ x = 150,250 \cdot 7.35 + 21,640 \cdot 1 + 8,360 \cdot 13.67$$
$$+ 30,000 \cdot 7.33 + 17,800 \cdot 7.33$$

$$228,050\ x = 1,587,628$$
$$x = 6.962$$
$$e_x = 0.37$$

(E) Calculate M_y and M_x :

$$M_x = P\ e_y = -288,050 \cdot (-.37) = 106,578$$
$$M_x = P\ e_x = -288,050 \cdot (-.71) = 204,516$$

(F) Calculate max. and min. soil pressures:

$p_{max} = P/A + M_x y/I + M_y x/I$

$A = 16.33 \, . \, 14.67 = 239.56$

$p = -228,050/239.56 + 204,516 \, . \, (-8.17)/5,324 +$

$106,578 \, . \, (-7.33)/4,296$

$= -952 - 314 - 182 = -1448$ psf.

$p_{min} = -952 + 204,516 \, . \, 8.17/5324 + 106,578 \, .$

$7.33/4296 = -952 + 314 + 182 = -456$ psf.

(G) Conclusions:

1. The calculated soil pressures are well below the allowable.
2. The maximum soil pressure does not exceed twice the average. (a good indication of a satisfactory variation).
3. The resultant is obvious within the middle 1/3 of the foundation.

(H) For details of foundations in general see article 7.

B - Design

Design a reinforced concrete foundation block to be set directly on hard clay to support the electric motor shown in figure 23. The motor and bedplate weigh 1120#. The motor develops 30 hp at 1500 rpm. The dimensions are shown in figure 23.

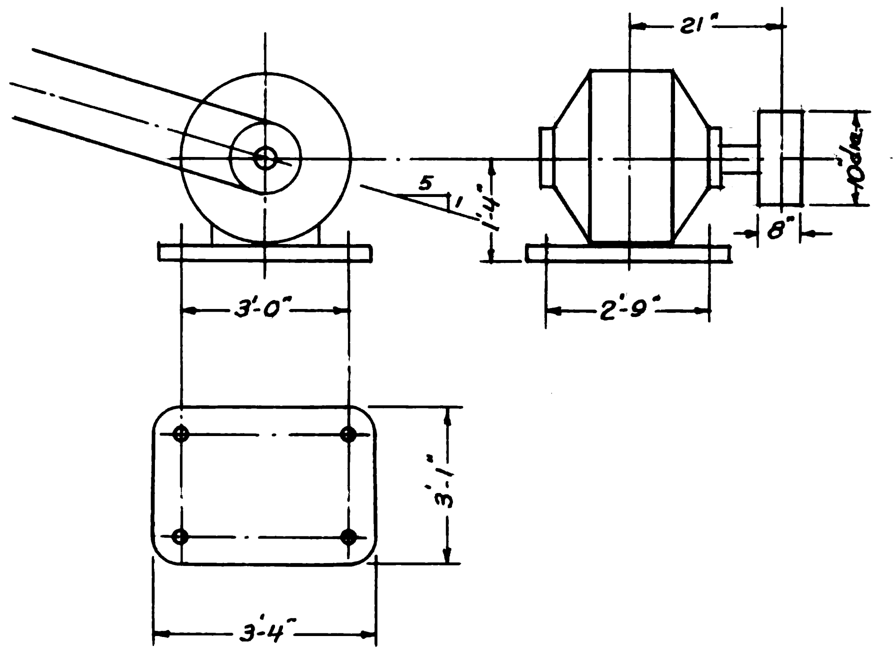


Figure 23

<u>First step</u> - Find the loading due to the belt pull; (assume that the belt takes off at an angle as shown in figure 23) applying formula 15, page 32.

$$F_b = \frac{30,000 \times 30}{1500 \times \frac{10}{12}} = 720\#$$

The components of the belt pull in x and y directions are:

$$F_{bx} = \frac{5}{5.1} F_b = \frac{.5 \times 720}{5.1} = 705\#$$

$$F_{by} = \frac{F_b}{5.1} = \frac{720}{5.1} = 141\#$$

Second step. Assume horizontal dimensions as in figure 24.

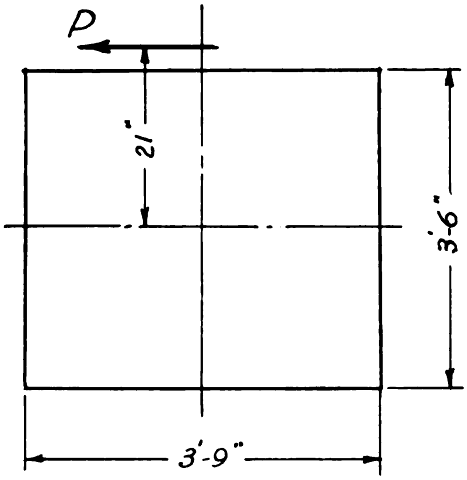


Figure 24

Third step. Find the weight of foundation necessary to prevent sliding on the soil:

Apply formula 10, page 26 which expresses the required weight to prevent slipping.

$$W_f = \frac{P_e A}{\mu K} + \frac{P}{\mu} - W_m$$

$P = 705\#$

$e = 21/12 \quad 1.75'$

$A = 3.75 \cdot 3.5 = 13.1$ sq'

μ = (from Table II, page 29) 0.12

$W_m = 1220\#$

$$K = \frac{2}{3}\left(2abc + \frac{b^3}{2} \log \frac{e+a}{e-a} + \frac{a^3}{2} \log \frac{c+b}{c-b}\right)$$

(formula 11, page 28)

where: a is 1/2 of the short dimension of the foundation = 1.75 and b is 1/2 the long dimension of the foundation = 1.875.

$$c = \sqrt{a^2 + b^2} = \sqrt{1.75^2 + 1.875^2} = 2.54$$

$$K_{regd.} = \frac{2}{3}\left(2x1.75x1.875x2.54 + \frac{1.875^3}{2}\log\left(\frac{2.54 + 1.75}{2.54 - 1.75}\right) + \frac{1.75^3}{2}\log\left(\frac{2.54 + 1.875}{2.54 - 1.875}\right)\right)$$

$$K = \frac{2}{3}\left(16.678 + 3.29 \log\left(\frac{4.29}{.79}\right) + 2.65 \log\left(\frac{4.415}{.665}\right)\right)$$

$$K = \frac{2}{3}(16.678 + 329 \log 5.43 + 2.68 \log 6.64)$$

$$K = \frac{2}{3}(16.678 + 3.29 \times 1.69 + 2.68 \times 1.89)$$

$$K = \frac{2}{3}(16.678 + 5.556 + 5.073) = \frac{2}{3}(27.317)$$

$$K = 18.21$$

Inserting these values in the formula above:

$$W_f = \frac{705x1.75}{0.12x18.21} + \frac{705}{0.12} - 1220$$

$$= 565 + 5875 - 1220$$

W_f = 5230# (Necessary to prevent slipping)

<u>Third step</u> - Find the weight of foundation necessary to prevent unequal settlement.

Referring to article 2-C, page 29, we shall now make use of formula 13 to solve for the foundation weight which will insure that the resultant of loads will be within the middle 1/3 of the foundation base.

$$W_f = \frac{P_v\ (3d_p - 1) - W_m(3d_m - 1)}{3d_f - 1}$$

Where P_v (as previously computed) = 141#

$d_p = 3.5'$

$l = 3.5'$

$W_m = 1220\#$

$d_m = 1.75'$

$d_f = 1.75'$

$$W_f = \frac{141(3 \times 3.5 - 3.5) - 1220(3 \times 1.75 - 35)}{(3 \times 1.75 - 3.5)}$$

$$= \frac{141 \times 7 - 1220 \times 1.75}{1.75} = \frac{987 - 2140}{1.75}$$

$$W_f = -660\#$$

Which means that the weight of the motor is sufficient to prevent unequal settlement.

Fourth step - Design the foundation block:

The weight of the foundation, having chosen the cross-sectional dimensions, must be 5230#. Or in cu. ft. of concrete required:

5230/150 = 34.8 cu. ft.

The cross-sectional area = 13.1 sq. ft. so the required foundation height is:

34.8/13.1 = 2.66'

Detailed design: see article 7-A,B,C, and D.

APPENDIX A

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APPENDIX B

Vibration problems related to operating machinery and mechanisms can often be best solved by the use of vibration recording analyzing systems or measuring instruments.

Manufacturers of these instruments include:

Aeroquip Corporation -- Jackson, Michigan

Brush Development Company -- Cleveland, Ohio

Consolidated Engineering Corporation -- Pasadina, Calif.

General Electric Company -- Cambridge, Mass.

MB Manufacturing Company -- New Haven, Conn.

Western Electric Company -- New York, N. Y.

Westinghouse Electric Corporation -- Pittsburg, Pa.

APPENDIX C

Manufacturers of isolating materials, most of which welcome requests for information and are prompt to offer their engineering services if required have been listed below with their addresses:

American Felt Company -- Glenville, Conn.

L. N. Barry Co. Inc., (Rubber Products), Cambridge, Mass.

Electron Corp., (confined air column), Freeport, New York.

Fabreeka Products Company (rubber and cotton duck pads), Boston, Mass.

Felter Company, Inc., (felt), Boston, Mass.

Firestone Industrial Product Co., (rubber), Akron, Ohio.

International Balsa Corp., (Balsa) Jersey City, N. J.

B. F. Goodrich Company, (isolators, rubber), Akron, Ohio.

Korfund Company, Inc., (isolation), Long Island, N. Y.

Lord Manufacturing Company, Erie, Pa.

MB Manufacturing Company, New Haven, Conn.

U. S. Rubber Co., Meck Goods Division, New York, N. Y.

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