Dynamic load analysis of centrifugal machine foundations

A Dissertation Report
Submitted in the Partial fulfillment of the
Requirement for Award of the Degree
of
MASTER OF THECHNOLOGY
In STRUCTURAL ENGINEERING
By

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Under the guidance of
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DECLARATION

I hereby declare that the dissertation entitled, "**Dynamic load analysis ofcentrifugal machine foundations"** " is an authentic record of my own research work carried out as a requirement for the preparation of M-Tech dissertation for the award of Masters of Technology Degree in Structure Engineering from Lovely Professional University, Phagwara, Punjab, under the guidance of Mr. Manoharan Rajalingam, during the period between January 2017 and March 2017. All the information furnished in this review is based upon my intensive work and is completely genuine to the best of my knowledge.

Date:

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ACKNOWLEDGEMENT

Any accomplishment requires the efforts of people and, this is not different. So it has bees also complete with cooperation of many persons. I am thankful to guide **“Mr. ManoharanRajalingam”** gave me the golden opportunity to do this wonderful research on the topic **“Study on Sustainable construction building materials”**, which also helped us in doing a lot of Research and I came to know about so many new things.

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Table of Contents

ABSTRACT

In this project which is basically based on the study to determine the response of vibratory load on the foundation of structure. The plan of a machine foundation is generally made by idealizing the foundation- soil system as spring-mass dashpot model having one or two degrees of freedom. Surface footing and the soil spring are the 2 functions which are treated by machine foundation and restraining values are determined by using the elastic-half space analog. Novak's work is the concept of elastic half space concept which also determined the spring and restraining value for the action of Foundation. The soil spring and damping values can also be obtained following the impedance compliance function approach. This project also involves the brief study of the assumed and detected action for this type of foundation.

Introduction

Machine foundation is provided under the super structure of vibrating and rotating machine. Foundation are subject to the various force like dynamic and static caused by the machine. The dynamic forces that is produce by the machine is transferred to the foundation. Moving parts of the machine are balanced but there is always some unbalance in field due to eccentricity of rotating part occur. Foundations are subjected to combination of both static and dynamic loads. In addition to static loads, in some cases underlying soils are subjected to dynamic loads. Static loads produce no vibration and loads are imposed slowly. Dynamic loads produce vibration to foundation soil system. Foundations supporting machines such as reciprocating engines, radar tower, punch presses, turbines, large electric motors and generators etc are subjected to vibration caused by unbalanced machine forces as well as static weight of machine.

Objective of study

To develop an analytical model for the design of equipment foundation with varying cross section to support centrifugal type machines which is subjected to both static and dynamic loadings. The objective is to focuses on the first two steps of the analysis and accordingly details the various aspects involved in the development of a realistic finite element model required for dynamic analysis. The response of the foundation is then obtained through free vibration analysis (Eigen analysis) and harmonic forced vibration analysis.

Literature Review

Jayarajan P.,Kouzer K.M 2014 Turbo-generators are power generation apparatus utilized as a part of the power plants. It is the most key and costly gear of a power plant complex and is by and large put inside a power house. The turbo-generator establishment comprises of turbine, generator and its assistants mounted on a fortified bond concrete (RCC) table best structure comprising of the best deck, sections and base pontoon. Considering the troublesome common parameters, hugeness of the machines and hazard engaged with terms of open clamor, examination and outline of turbo-generator establishments still stay a standout amongst the most troublesome and testing assignment in structural building calling. A key fixing to the fruitful establishment outline for a turbo-generator is the watchful building examination of the establishment reaction to the dynamic loads from the expected operation Turbo-generators are power generation apparatus utilized as a part of the power plants. It is the most key and costly gear of a power plant complex and is by and large put inside a power house. The turbo-generator establishment comprises of turbine, generator and its assistants mounted on a fortified bond concrete (RCC) table best structure comprising of the best deck, sections and base pontoon. Considering the troublesome common parameters, hugeness of the machines and hazard engaged with terms of open clamor, examination and outline of turbo-generator establishments still stay a standout amongst the most troublesome and testing assignment in structural building calling.

Manoj sao,Gulab chand sahu 2017 The paper presented the concept of analytic study on equipment foundation subjected to dynamic loads. The foundation of cold rolling mills and other equipment are subjected to static load due to dynamic loads. The term foundation load are used here relative to single force, force couple , load per unit area or pure torques. These torque and forces have great effect between machine or components of machine via foundation. This project work covers various types of loads acting on the equipment and there effect on foundation

Porto B.thiago,Mendoca 2012 The nearness of present day adapt, for instance, crushers, processors, compressors, sifter shaker and dryer, make vibrations in structures. In this way, evacuations, speeds and expanding speeds show up and cause trouble in the midst of customary human activities, operational or potentially operational issues in mechanical rigging. It is, thusly, a most remote point state, with the significance of allowable limits in the structure to evacuating, speed and expanding speed. These qualities are endorsed by outfit makers or potentially worldwide measures. This work sets up the key necessities that must be known in the arrangement of strong structures subjected to dynamic exercises got from present day adapt. In this paper, a ball procedure and a dryer are discussed as logical investigations. For the numerical showing of the proposed issue, was used the Finite Element Method (FEM)

CALCULATION

A.Machine Parameters

Compressor Weight (W_C) = 3600 lbs

Rotor Weight (W_R) = 2200 lbs

Operating Speed (f) = 7000 rpm

(ω) = $2\pi f/60$

= 2 ×3.14 ×7000/60 =732.67 rad/sec

Critcal Speed (f_C) = 1st ∽3500 rpm

2nd∽ 9850 rpm

e = 0.0015 (provided by manufacturer)

Dynamic eccentricity at operating speed

$e = e/[1-(f/f_C)^2]$

$= 0.0015/[1-(7000/3500)^2] = 0.0005$ inch

Centrifugal force F_O

$F_O = (W_R/g)\times e\times\omega^2$

$=(2200/9.81\times3.28)\times0.0005/12\times732.67^2=1530$ lbs

Turbine Weight (W_T) = 17000 lbs

Rotor Weight (W_R) =570 lbs

Operating Speed (f) = 7000 rpm

(ω) = $2\pi f/60$

= 2×3.14×7000/60 =732.67 rad/sec

Critcal Speed (f_C) = 1st ∼ 2100 rpm

2nd ∼ 9870 rpm

e = 0.0015 (provided by manufacturer)

$e = e/[1-(f/f_C)^2]$

$= 0.0015/[1-(7000/2100)^2] = 0.0001483$ inch

Centrifugal force F_O

$F_O = (W_R/g) \times e \times \omega^2$

$=(570/9.81\times3.28) \times 0.0001483/12\times732.67^2=118$ lbs

Total Centrifugal force F_O = 1530 + 118 =1648 lbs

Base Plate weight (W_B) = 6000 lbs

Total Machine weight (W_M) = $W_C+W_T+W_B$

=36000 + 17000 + 6000= 59000 lbs

Soil Parameter (from geotechnical field)

Soil Density (γ) = 125Pcf

Shear Modulus (G) = 6500 Psi

Poisson's Ratio = 0.45

Concrete Footing Trial outline

Weight of the footing (W_F) = 118000 lbs

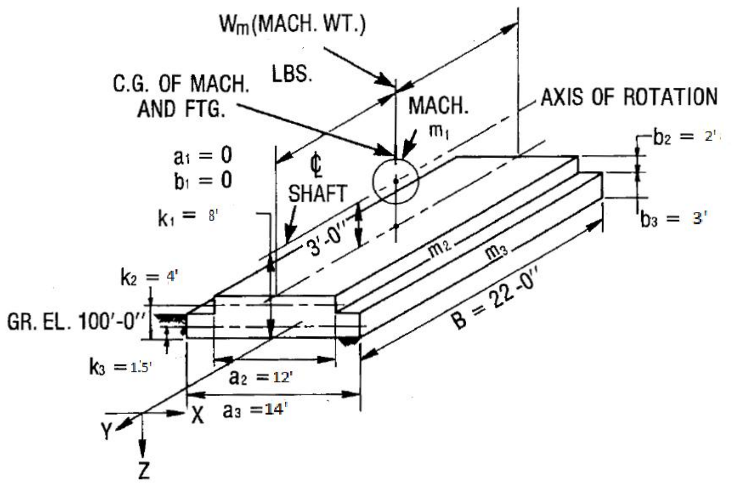
Weight of footing / weight of machine = 118000/59000 = 2

Total static load (W) = Machine weight + weight of footing

= 59000 + 118000 =177000 lbs

Actual Soil Pressure = 177000/14(22) = 632 Psf

Dynamic Analysis



1. Equivalent radius r_o

(a) Vertical excitation = $\sqrt{BL/\pi}$

$=\sqrt{14 \times 22/3.14}$ = 9.9 ft

(b) Horizontal excitation = $\sqrt{BL/\pi}$

$= \sqrt{14 \times 22/3.14}$ = 9.9 ft

(c) Rocking excitation = $\sqrt[4]{BL^3/3\pi}$

$= \sqrt[4]{14^3 \times 22/3 \times 3.14}$ = 8.9 ft

2. Mass and Mass Moment of inertia

(a) Vertical excitation

M = W/g

= 177000/9.81×3.28 = 5500 lbs .sec^2/ft

(b) Horizontal excitation

M = W/g

= 177000/9.81×3.28 = 5500 lbs .sec^2/ft

(c) Rocking excitation

(1) Machine (I_ψ) $=\sum_i^n [mi/12(ai^2 + bi^2) + mi \times k^2]$ = (I_ψ) = 117267

(2) Footing (I_ψ) $= \sum_i^n [mi/12(ai^2 + bi^2) + mi \times k^2]$ = (I_ψ) =82773.3

= (I_ψ) = 200040.3 lbs-sec^2-ft

3. Mass Ratio

(a) Vertical excitation

B_z= (1-v)/4×W/γr_o^3

= (1-0.45)/4×177000/125×9.9^3 = 0.20

(b) Horizontal excitation

B_x= (7-8v)/31824×W/γr_o^3

= (7- 8×0.45/32(1-045) ×177000/125×9.9^3) = 0.28

(c) Rocking excitation

B_ψ = $3(1-V)\times I_\psi/8p r_o^5$

= $3\times(1-0.45)/8\times200040.3/3.85\times8.9^5$=0.19

4. Geometric Damping ratio

(a) Vertical excitation

$D_z = 0.425/\sqrt{\mathrm{Bz}}\times\alpha$

$D_Z = 0.425/\sqrt{0.20}\times1 = 0.95$

(b) Horizontal excitation

$D_x = 0.288/\sqrt{B_\chi}\times\alpha$

$D_x = 0.288/\sqrt{0.28}\times1 = 0.54$

(c) Rocking excitation

$D_\psi= 0.15\alpha_\psi/(1+\eta_\psi B_\psi)\sqrt{\eta^\varphi B^\varphi}$

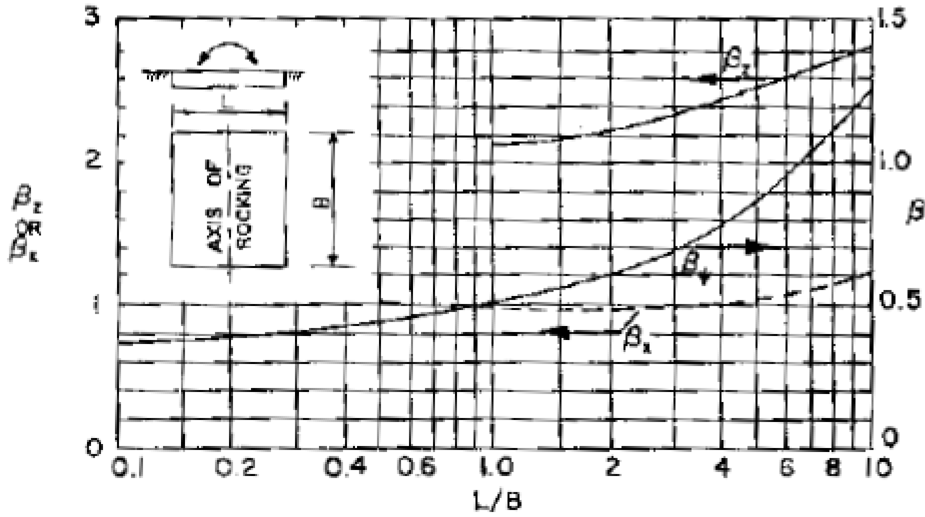
$= 0.15\times1/(1+1\times0.19)\times\sqrt{1\times0.19}$

= 0.28

5.Spring Coefficient

(a) Vertical excitation

β_z = 2.16



(b) Horizontal excitation

β_x = 0.96

(c) Rocking excitation

β_ψ = 0.47

6.Equivalent Spring constants

(a) Vertical excitation

$K_z = G/1\text{-}v \times \beta_x \sqrt{BL}\eta_z$

$= 6500 \times 144/1\text{-}0.45 \times 2.16 \times \sqrt{14 \times 22 \times 1} = 64 \times 10^6$ lbs/ft

(b) Horizontal excitation

$K_x = 2(1+V)G\ \beta_x \sqrt{BL\eta x}$

$= 2(1+0.45) \times 6500 \times 144 \times 0.96 \times \sqrt{14 \times 22 \times 1} = 45 \times 10^6$lbs/ft

(c) Rocking excitation

$K_\psi = G/1\text{-}v \times \beta_\psi\ BL^2 \eta_\psi$

$= 6500 \times 144/1\text{-}0.45 \times 0.47 \times 22 \times 14^2 \times 1 = 3448 \times 10^6$ lbs-ft/rad

7. Natural Frequency

(a) Vertical excitation

$F_{nz} = 60/2\pi \times \sqrt{K/M}$

$= 60/2 \times 3.14 \times \sqrt{64000000/5500} = 1030.6$ rpm

(b) Horizontal excitation

F_{nx} = 60/2π× $\sqrt{\frac{K}{M}}$

= 60/2×3.14× $\sqrt{45000000/5500}$ = 864.2 rpm

(c) Rocking excitation

F_n = 60/2π $\sqrt{K/M}$

= 60/2×3.14× $\sqrt{\frac{3448000000}{200040.3}}$ = 1254 rpm

8. Magnification Factor

(a) Vertical excitation

M = 1/ $\sqrt{(1 - r^2)^2 + (2Dr)^2}$

=1/ $\sqrt{(1 - 6.672^2)^2 + (2 \times 0.95 \times 6.67)^2}$ = 0.092

(b) Horizontal excitation

M = 1/ $\sqrt{(1 - 8.10^2)^2 + (2 \times 0.54 \times 8.10)^2}$ = 0.015

(c) Rocking excitation

M = $\sqrt{(1 - 5.58^2)^2 + (2 \times 0.28 \times 5.58)^2}$ = 0.032

9. Dynamic Force

(a) Vertical excitation

V_o = 1648 lbs

(b) Horizontal excitation

H_o = 1648 lbs

(c) Rocking excitation

$T_o = H_o \times K_o$ = 1648 × 8 = 13184 lbs ft

11(a).Vibration Amplitude

(a) Vertical excitation

$Z = M(F_o/K)$

$= 0.025\times(1648/65\times10^{6}) = 0.6\times10^{-6}$ ft

(b) Horizontal excitation

$X = M(F_o/K)$

$= 0.1712\times(1648/43\times10^{6}) = 0.65 \times 10^{-6}$ ft

(c) Rocking excitation

$\psi = M(F_o/K)$

$= 0.032 \times (1648/3448\times10^{6}) = 0.2711\times10^{-6}$ rad

11. (b)Component of rocking oscillation

(a) Vertical excitation

At edge of ftg

$\psi \times R_h = 0.2712 \times 10^{-6}(7) = 1.89\times10^{-6}$ ft

(b) Horizontal excitation

At center of bearing

$= 0.2712\times10^{-6}\times(8) = 2.16\times10^{-6}$ ft

11. (c) Resultant Vibration Amlitude

(a) Vertical excitation

= 11(a) + 11 (b)

$Z_t = 0.2711\times10^{-6}+1.89\times10^{-6} =0.000025$ inch

(b) Horizontal excitation

= 11(a) + 11 (b)

$X_t = 0.2711\times10^{-6}+2.16\times10^{-6} = 0.000029$ inch

12.Transmissibility Factor

(a) Vertical excitation

$T_r = \sqrt{1 + (2 \times \mathrm{Dr})^2}/\sqrt{(1 - \mathrm{r}^2)^2 + (2 \times \mathrm{Dr})^2}$

$= \sqrt{1 + (2 \times 0.95 \times 6.67)^\wedge 2}/\sqrt{(1 - 6.67^\wedge 2)^\wedge 2 + (2 \times 0.95 \times 6.67)^\wedge 2}$

= 0.28

(b) Horizontal excitation

$T_r = \sqrt{1 + (2 \times \mathrm{Dr})^2} / \sqrt{(1 - \mathrm{r}^2)^2 + (2 \times \mathrm{Dr})^2}$

$= \sqrt{1 + (2 \times 0.54 \times 8.10)^\wedge 2} / \sqrt{(1 - 8.10^\wedge 2)^\wedge 2 + (2 \times 0.54 \times 8.10)^\wedge 2}$

=0.13

(C) = Rocking excitation

$T_r = \sqrt{1 + (2 \times \mathrm{Dr})^2} / \sqrt{(1 - \mathrm{r}^2)^2 + (2 \times \mathrm{Dr})^2}$

$= \sqrt{1 + (2 \times 0.28 \times 5.58)^\wedge 2} / \sqrt{(1 - 5.58^\wedge 2)^\wedge 2 + (2 \times 0.28 \times 5.58)^\wedge 2}$

= 0.10

Conclusion

This work is proposed to make available the analytical model for the analysis and design of foundation for centrifugal machines that can withstand both dynamic and static loadings.

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