الجمهورية الجزائرية الديموقراطية الشعبية

وزارة التعليم العالي والبحث العلمي

SAAD DAHLEB BLIDA’S UNIVERSITY

Faculty of Technology

Department of Civil Engineering

MASTER DISSERTATION IN CIVIL ENGINEERING

Specialty: Structural engineering

THE TITLE :

THE STEADY STATE ANALYSIS FOR INDUSTRIAL APPLICATION

CASE STUDY: BLOWER FAN FOR POLLUTION CONTROL

Submitted by:

BELLALA Islem

Supervisor:

Mr.BELHOUCHET Fouzie

Co-Supervisor:

Dr. SI AHMED Mohammed

Blida,November 2021

MASTER DISSERTATION IN CIVIL ENGINEERING

Specialty: Structural engineering

THE STEADY STATE ANALYSIS: INDUSTRIAL APPLICATION

CASE STUDY: BLOWER FAN FOR POLLUTION CONTROL

Submitted by:

BELLALA Islem

Supervisor:

Mr.BELHOUCHET Fouzie

Co-Supervisor:

Dr. SI AHMED Mohammed

Blida, November 2021

ملخص.

يهدف هذا العمل إلى فهم ودراسة سلوك الهياكل عند تعرضها لمعدات هزازة، وتستمر هذه الاهتزازات لفترة طويلة من الزمن (ربما لعقود) وهذا ما يسمى حالة الاهتزاز المستقر وفقًا لأدبيات الديناميكا. خضع تحليل الحالة المستقرة لعدة فحوصات مثل السعة والسرعة والتسارع، والأهم من ذلك فحص ظاهرة الرنين وفقًا للكود ACI 351 "تقرير عن أسس المعدات الديناميكية" و ISO 1940 "متطلبات جودة التوازن للدوارات الصلبة" كما تم إجراء تحليل مقارن بين النماذج العامة (نموذج كامل) والنماذج المجردة لتبسيط وتقليل وقت التحليل وتبين أن استخدام نموذج تقديري أكثر كفاءة فيما يتعلق باستهلاك الوقت وكذلك تم التحقق من أهمية موقع الجهاز بالنسبة للهيكل الحامل له في ضوء النتائج التي تم العثور عليها والتي اتضح أنها مهمة جدًا وتؤدي دائمًا إلى تقليل تكلفة حامل الماكينات، تم إجراء جميع هذه التحليلات باستخدام برنامج SAP2000.

Abstract.

This work aims to understand and study how structures behave when being subjected to vibrating equipments, these vibrations last for a long period of time (maybe decades) and this is what is called a steady-state vibration according to dynamics literature. The steady-state analysis underwent several checks such as the amplitude, velocity, acceleration, and most importantly the check of resonance phenomenon in accordance with the codes ACI 351 "Report on foundations for dynamic equipment" and ISO 1940 "Balance quality requirement of rigid rotors». Comparative analysis also was done between Global and discretized models to simplify and reduce analyzing time and found that using a discretized model is more efficient in matter of time consumption, and also, checked the importance of locations of the machine on the found results which turned out to be very important and always lead to reduce the cost of the support of the machines, All these analyses were done using the software SAP2000.

Keywords: Vibrations, Dynamic, Steady state analysis, resonance, SAP2000.

Defining the location of the machinery on the support is very important, and always lead to reduce the cost of the support.

Résumé

Ce travail a pour but de comprendre et d'étudier le comportement des structures lorsqu'elles sont soumises à des équipements vibrants, ces vibrations restent à long terme (peut-être des décennies) et c'est ce qu'on appelle une vibration en régime permanent selon la littérature de la dynamique. L'analyse en régime permanent subi plusieurs vérifications telles que l'amplitude, la vitesse, l'accélération, et surtout la vérification du phénomène de résonance conformément aux codes ACI 351 "Rapport sur les fondations des équipements dynamiques" et ISO 1940 "Exigence de qualité d'équilibre des rotors rigides » Une analyse comparative a également été effectuée entre le modèle global et discrétisé pour simplifier et réduire le temps d'analyse et trouvé que l'utilisation d'un modèle discrétisé est plus efficace en termes de consommation de temps, également vérifié l'importance des emplacements de la machine sur les résultats trouvés ce qui s'est avéré très important et conduit toujours à réduire le coût du support de la machine ,Toutes ces analyses ont été effectuées à l'aide du logiciel SAP2000.

Mots clés : Vibrations, Dynamique, l'analyse en régime permanent, résonance, SAP2000.

Acknowledgment

First, I am grateful towards almighty Allah, our creator for granting me the strength and well to prepare and finish this work

I would like to thank my supervisor ***Mr. BELHOUCHET Fouzie*** *for his support, he always listened and was available throughout the production of this dissertation, as well as for the inspiration, the help and the time he was kind enough to devote to me, and without whom this dissertation would never have seen the light of day.*

am also thankful to my co-supervisor ***Dr. SI AHMED Mohammed*** *for his help and guidance*

I also thank the committee members for taking the time to read and evaluate this thesis.

At the end, I would like to express my gratitude towards the teachers and professors of the department of civil engineering of the University of SAAD DAHLEB who have contributed to my academic journey.

Dedications

To my dear parents and brother.

To my dear family.

To my dear friends

I cannot find the right and sincere words to express my affection and my thoughts to you, to me you are brothers, sisters and friends on whom I can count.

As a testament to the friendship, that unites us and to the memories of all the times we have spent together, I dedicate this work to you and I wish you a life full of health and happiness.

Thank you all.

List of illustrations

List of tables

Acronyms

Latin upper-case letters:

A : Zone acceleration factor
Ce : Exposure Coefficient
Cr: Roughness coefficient
Cpe: External pressure coefficient
Cpi: Internal pressure coefficient
D: Dynamic amplification factor
F_0: Dynamic force amplitude (zero-to-peak), (N);
DL: Dead load
LL: Live load
SDL: Super imposed dead load
Ex: The seismic load on direction X
Ey: The seismic load on direction Y
Iv : Turbulence intensity
Q : Quality factor
R: Structural behavior factor
S_n The snow load on the roof
S_k : The characteristic value of snow load on the ground
S_f: Service factor, used to account for increased unbalance during the service life of the machine,
T_n : The natural period (undamped)
T_D The natural period (damped)
W: Aerodynamic pressure
W1: The wind load for the first direction
W2 : The wind load for the first direction

Latin lower-case letters:

c_{cr} The critical damping.
f : The natural frequency.
m_r: Rotating mass, (kg);
e_m: Mass eccentricity, (mm)
qp Peak velocity pressure

Greek lower-case letter:

μ : The snow load shape coefficient
ξ : The damping ratio.
ω_0: Circular operating frequency of the machine, (rad/s)
ω_n : The circular natural frequency.
ω : The circular forcing frequency
ω_d : The damped frequency.

Table of content

General introduction

The industry is a fundamental activity in the economy of any country, and it’s responsible for the processing and the transformation of natural products (raw materials) into other finished and semi-finished products.

Every industrial warehouse contains several equipment and machines, those machines are multipurpose and can be used to (perform, operating like cutting, sanding, drilling...etc.), all those machines fall into three major types of machines: rotating machines, reciprocating machines, impact machines, each one of them has its own system and own motion, therefore, they have different effects on the structure or the support element that carries on the equipment of machines.

Heavy machinery with reciprocating, impacting, or rotating masses requires a support system that can resist dynamic forces and the resulting vibrations. When excessive, such vibrations may be detrimental to the machinery, its support system, and operating personal subjected to them.

The steady state analysis is a very common type of analysis used to analyze and calculate the different parameters of vibrating machine that can be generated, such as amplitude, velocity, and accelerations.

This dissertation topic was chosen to show the effect of a blower (special fan) which belong to the rotating machine types, on the structural support and try to design its appropriate supporting system to avoid several problems like unallowed amplitudes or even resonance.

The first chapter gives a theoretical background about the basics of structural dynamics, passing by explaining what vibrations are, and giving some terminology for better understanding of the next chapters. Afterwards, it talks about single and multi-degree of freedoms, then shows the difference between transient and steady state response, the end of the chapter goes into more detail and focuses on the undamped steady state response.

The second chapter includes the warehouse and mezzanine design, where the Blower fan is placed. The design of the warehouse with all its parts (action design / members design / connections design) is not the main interest, but it’s necessary to check that the structure is well connected, the load path is respected and make sure that results in the next chapter are correctly calculated, especially when it comes to amplitudes and velocities.

In the 3rd chapter a case study is treated, starting with an introduction to steady state analysis, and then the most used codes and references in the field and present a small definition of these codes such as (the ACI 351.3R/ ISO 1940/DIN 4024.1/ IS 2974.3) among this code the ACI 351 R3-2018 is used to accomplish this work, the next section presents the machine characteristics and calculates the unbalanced loads generated by the blower fan. At the end of the chapter a two types of finite element models are done (global model and discretized model) and check the resonance phenomenon with all cases that could appear in the case study, at the end of the section a comparative analysis of the best location of the blower is done to see how the choice of the location influences the amplitude and the velocity calculations.

At the end, this dissertation is concluded with a general conclusion and recommendations for future students who want to carry on this work and develop it.

I. Literature review / Theoretical framework

I.1 The concept of vibration:

I.1.1 Definition:

Vibration, periodic back-and-forth motion of the particles of an elastic body, commonly resulting when almost any physical system is displaced from its equilibrium condition and allowed to respond to the forces that tend to restore equilibrium, in another word, it is a time dependent displacement of a particle (system) with respect to its equilibrium position.

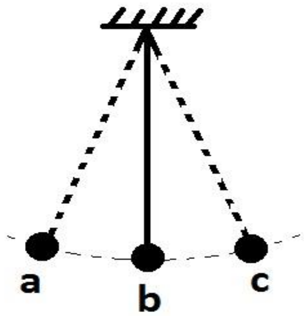


Figure I-1 Representation of a simple pendulum

I.1.2 Vibration categories:

Vibrations fall into two categories: **free** and **forced.**

a) Free vibration:

A structural system, when disturbed from its position of equilibrium and released, oscillates about its mean position of equilibrium. This state of vibration of the structure without any external excitation force is termed as **free vibration**.

b) Forced vibration:

A structural system, when subjected to time dependent excitation force, is set to motion. This state of vibration of the structure is termed as **forced vibration**.

I.1.3 Terminology:

❖ The Amplitude:

The amplitude is the maximum displacement from the equilibrium position of an object oscillating around the equilibrium position due to its dynamic mass.

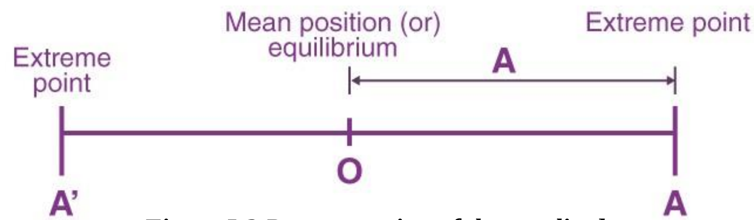


Figure I-2 Representation of the amplitude

- **Simple Harmonic Motion:**

Motion of particle with time that moves around a circle with uniform angular velocity. Trigonometric functions can be used to represent such motion like a sinusoidal function.

- **The period T :**

The required time for a motion to be repeated or to complete a full cycle.

- **Circular frequency of a system ω:**

The circular frequency is the angular displacement of a particle (system) in a time interval.

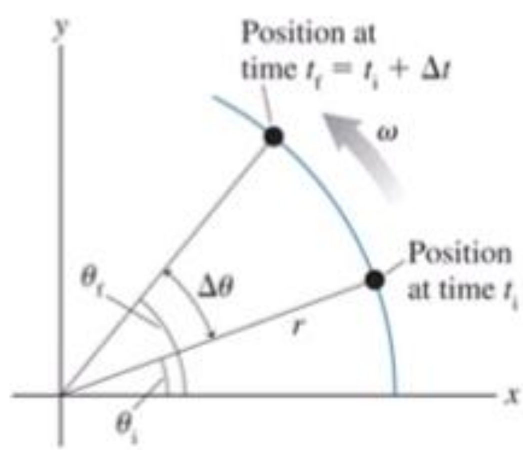


Figure I-3 Graph of the circuler frequency

- **Frequency:**

It is the number of cycles per unit time. Frequency and time period are inversely proportional to each other. A vibratory motion can have either a very high frequency or a very low frequency. Frequency can be expressed either as angular (circular) frequency (ω) or as oscillatory frequency (f). ω is expressed in radians per second and f is expressed in cycles per second or Hertz

❖ Resonance:

This phenomenon occurs mostly for forced vibration systems, and it’s a vibration of a system when the frequency of external force is equal to the natural frequency of the system. The amplitude of vibration at resonance becomes excessive. During resonance, with minimum input, there will be a maximum output. Hence, both displacement and the stresses in the vibrating body become very high.

❖ Damping:

Property which makes the vibration diminishes in amplitude “energy dissipation mechanism’’, for most cases, the viscous damping (linear) is usually used to catch the governing responses of an excitation.

Damping sources: Repeated straining, friction at joints, temperature due to strain,

Sound energy, damping devices.

❖ Degrees of Freedom:

The number of independent displacements required to define the displaced positions of all the masses relative to their original position is called the number of degrees of freedom (DOFs).

❖ Mode of Vibration:

In a system undergoing vibration, a mode of vibration is a characteristic pattern assumed by the system in which the motion of every particle is simple harmonic with the same frequency. Two or more modes may exist concurrently in a multi-degree freedom system.

I.2 Single degree of freedom (free vibration):

A structure is said to be undergoing free vibration when it is disturbed from its static equilibrium position and then allowed to vibrate without any external dynamic excitation.

I.2.1 Undamped free vibration:

The differential equation of the undamped simple oscillator in free motion is:

$$m\ddot{u} + ku = 0$$

this homogenous differential equation has mathematical solution and its general solution is:

$$u(t) = A\cos\omega t + B\sin\omega t$$

where A and B are constants of integration determined from initial conditions of the displacement u_0 and of the velocity $\dot{u}_0$.

where:

$A = u_0$

$B = \dot{u}_0/\omega$

$\omega = \sqrt{k/m}$ is the natural frequency in rad / sec

$f = \omega/2\pi$ is the natural frequency in Hz

$T = 1/f$ is the natural period in seconds

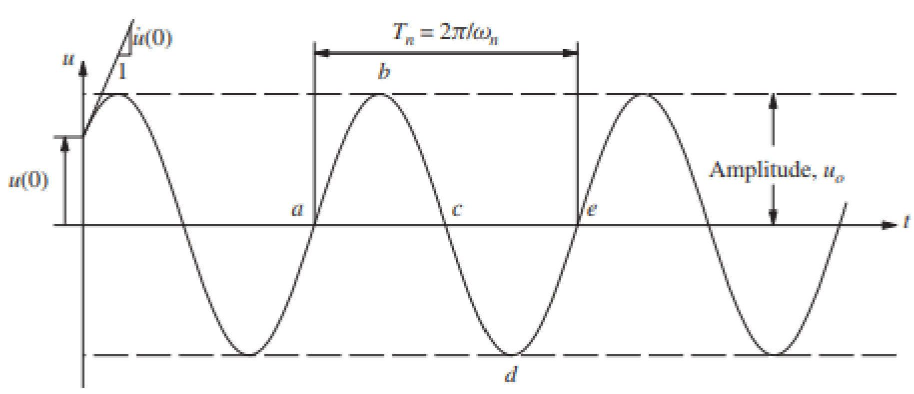


Figure I-4 Free vibration of a system without damping

I.2.2 <u>Damped free vibration:</u>

Real structures dissipate energy while undergoing vibratory motion. The most common and practical method for considering this dissipation of energy is to assume that it is due to viscous damping forces.

These forces are assumed to be proportional to the magnitude of the velocity but acting in the direction opposite to the motion. The factor of proportionality is called <u>the viscous damping coefficient.</u> It is expedient to express this coefficient as a fraction of the <u>critical damping</u> in the system ($\xi = c/c_{cr}$). The critical damping may be defined as the least value of the damping coefficient for which the system will not oscillate when disturbed initially, but it will simply return to the equilibrium position.

The homogenous differential equation (with constant coefficient type) of motion for the free vibration of a damped single degree-of-freedom system is given by:

$$m\ddot{u} + C\dot{u} + ku = 0$$

The analytical expression for the solution of this equation depends on the magnitude of the damping ratio. Three cases are possible:

1. Critically damped system ($\xi = 1$).

2. Overdamped system ($\xi > 1$).

3. Underdamped system ($\xi < 1$).

For the underdamped system ($\xi < 1$), the solution of the differential equation of motion may be written as:

$$\mathbf{u(t)} = e^{-\xi\omega t}\left(u_0 \cos \omega_D t + \frac{\mathring{u}_0 + u_0 \xi \omega}{\omega_D} \sin \omega_D t\right)$$

in which

$\omega = \sqrt{k/m}$ is the undamped frequency

$\omega_d = \omega\sqrt{1-\xi^2}$ is the damped frequency

$\xi = c / c_{cr}$ is the damping ratio

$c_{cr} = 2\sqrt{km}$ is the critical damping

and u_0 and $\dot{u}_0$ are, respectively, the initial displacement and velocity.

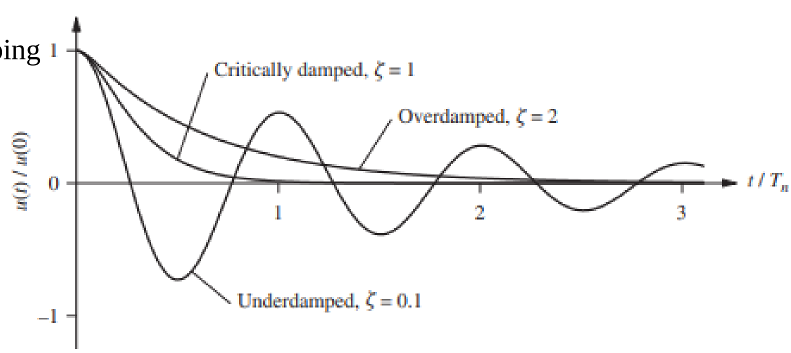


Figure I-5 Free vibration of underdamped, critically damped, and overdamped systems.

I.3 Multi-degree of freedom (M.D.O.F):

I.3.1 Equations of motion for undamped linear systems with M.D.O.F:

We always express the equations of motion for a system with many degrees of freedom in a standard form. The two degree of freedom system shown in the picture can be used as an example. We won't go through the calculation in detail here, but here is the final answer:

$$m_1 \frac{d^2u_1}{dt^2} + (k_1 + k_2)u_1 - k_2u_2 = 0$$
$$m_2 \frac{d^2u_2}{dt^2} - k_2u_1 + (k_2 + k_3)u_2 = 0$$

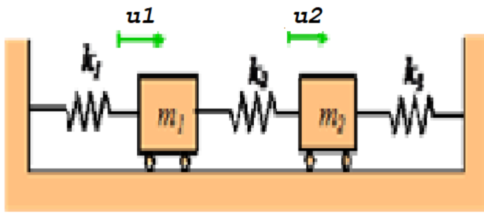


Figure I-6 The two degree of freedom system

To solve vibration problems, we always write the equations of motion in matrix form. For an undamped system, the matrix equation of motion always looks like this

$$\mathrm{M}\frac{d^2\mathrm{u}}{dt^2} + \mathrm{Ku} = 0$$

where **u** is a vector of the variables describing the motion, **M** is called the 'mass matrix' and **K** is called the 'Stiffness matrix' for the system. For the two spring-mass example, the equation of motion can be written in matrix form as:

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \frac{d^2}{dt^2} \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 + k_3 \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

For a system with two masses (or more generally, two degrees of freedom), **M** and **K** are 2x2 matrices. For a system with *n* degrees of freedom, they are *nxn* matrices.

The spring-mass system is linear. A nonlinear system has more complicated equations of motion, but these can always be arranged into the standard matrix form by assuming that the displacement of the system is small, and linearizing the equation of motion. For example, the full nonlinear equations of motion for the double pendulum shown in the figure are:

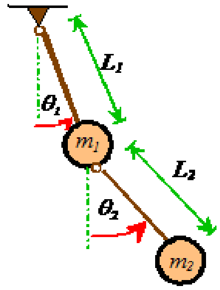


Figure I-7 Double pendulum system

$$(m_1+m_2)L_1\ddot{\theta}_1+m_2L_2\dot{\theta}_2^{\ 2}\sin(\theta_1-\theta_2)+m_2L_2\ddot{\theta}_2\cos(\theta_1-\theta_2)+(m_1-m_2)g\sin\theta_1=0$$
$$m_2L_2\ddot{\theta}_2+m_2L_1\ddot{\theta}_1\cos(\theta_1-\theta_2)-m_2L_1\dot{\theta}_1^{\ 2}\sin(\theta_1-\theta_2)+m_2g\sin\theta_2=0$$

Here, a single dot over a variable represents a time derivative, and a double dot represents a second time derivative (i.e., acceleration). These equations look horrible, but if we assume that if θ_1 , θ_2 , and their time derivatives are all small, so that terms involving squares, or products, of these variables can all be neglected, that, and recall that $\cos x \approx 1$ and $\sin x \approx x$ for small x, the equations simplify to:

$$(m_1+m_2)L_1\ddot{\theta}_1+m_2L_2\ddot{\theta}_2+(m_1+m_2)g\theta_1=0$$

$$m_2L_2\ddot{\theta}_2+m_2L_1\ddot{\theta}_1+m_2g\theta_2=0$$

Or, in matrix form:

$$\begin{bmatrix}(m_1+m_2)L_1 & m_2L_2\\ m_2L_1 & m_2L_2\end{bmatrix}\frac{d^2}{dt^2}\begin{bmatrix}\theta_1\\ \theta_2\end{bmatrix}+\begin{bmatrix}(m_1+m_2)g & 0\\ 0 & m_2g\end{bmatrix}\begin{bmatrix}\theta_1\\ \theta_2\end{bmatrix}=\begin{bmatrix}0\\ 0\end{bmatrix}$$

I.3.2 <u>Natural frequencies and mode shapes for undamped linear systems with M.D.O.F:</u>

First, let's review the definition of natural frequencies and mode shapes. Recall that we can set a system vibrating by displacing it slightly from its static equilibrium position, and then releasing it. In general, the resulting motion will not be harmonic. However, there are certain special initial displacements that will cause harmonic vibrations. These special initial deflections are called mode shapes, and the corresponding frequencies of vibration are called natural frequencies.

M

The natural frequencies of a vibrating system are its most important property. It is helpful to have a simple way to calculate them.

Fortunately, calculating natural frequencies turns out to be quite easy (at least on a computer). Recall that the general form of the equation of motion for a vibrating system is

$$M\frac{d^2u}{dt^2}+Ku=0$$

where **u** is a time dependent vector that describes the motion, and **M** and **K** are mass and stiffness matrices. Since we are interested in finding harmonic solutions for **u**, we can simply assume that the solution has the form $U\sin\omega t$, and substitute into the equation of motion:

$$-MX\omega^2\sin\omega t+KX\sin\omega t=0\ \Rightarrow\ KX=\omega^2MX$$

The vectors **u** and scalars $\boldsymbol{\lambda}$ that satisfy a matrix equation of the form **KX** = $\boldsymbol{\lambda}$**MX** are called 'generalized eigenvectors' and 'generalized eigenvalues' of the equation. It is impossible to find exact formulas for $\boldsymbol{\lambda}$ and **u** for a large matrix (formulas exist for up to 5x5 matrices, but they are so messy), but MATLAB or other numerical programs has built-in functions that will compute generalized eigenvectors and eigenvalues given numerical values for **M** and **K**.

The special values of λ satisfying $\boldsymbol{KX} = \boldsymbol{\lambda MX}$ are related to the natural frequencies by $\boldsymbol{\omega_i} = \sqrt{\boldsymbol{\lambda_i}}$ The special vectors **X** are the 'Mode shapes' of the system. These are the special initial displacements that will cause the mass to vibrate harmonically.

I.3.3 Free vibration of undamped linear systems with many degrees of freedom:

As an example, consider a system with n identical masses with mass m, connected by springs with stiffness k, as shown in the picture. Suppose that at time t=0 the masses are displaced from their static equilibrium position by distances u_1, $\mathbf{u_2}$…. $\mathbf{u_n}$ and have initial speeds $\mathbf{v_1}$, $\mathbf{v_2}$…$\mathbf{v_n}$ We would like to calculate the motion of each mass $\mathbf{u_1(t)}$, $\mathbf{u_2(t)}$…$\mathbf{u_n(t)}$ as a function of time.

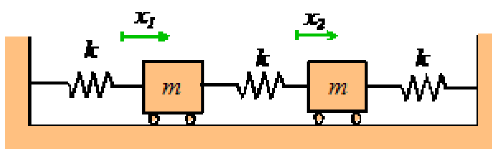


Figure I-8 Two degree of freedom system

It is convenient to represent the initial displacement and velocity as *n* dimensional vectors **u** and **v**, as $\mathbf{u} = [\mathbf{u_1}, \mathbf{u_2} \ldots \mathbf{u_n}]$ and $\mathbf{v} = [\mathbf{v_1}, \mathbf{v_2} \ldots \mathbf{v_n}]$ In addition, we must calculate the natural frequencies ω_i and mode shapes X_i, i=1..*n* for the system.
The motion can then be calculated using the following formula:

$$u(t) = \sum_{i=1}^{n} A_i \mathrm{X}_i \cos \omega_i t + B_i \mathrm{X}_i \sin \omega_i t$$

where:

$$A_i = \frac{\mathrm{u} \cdot \mathrm{X_i}}{\mathrm{X_i} \cdot \mathrm{X_i}} \qquad B_i = \frac{\mathrm{v} \cdot \mathrm{X_i}}{\omega_i \mathrm{X_i} \cdot \mathrm{X_i}}$$

Here, the dot represents an *n* dimensional dot product
This expression tells us that the general vibration of the system consists of a sum of all the vibration modes, (which all vibrate at their own discrete frequencies). You can control how big the contribution is from each mode by starting the system with different initial conditions. The mode shapes X_i have the curious property that the dot product of two different mode shapes is always zero ($\mathrm{X}_1.\mathrm{X}_2 = 0$, $\mathrm{X}_1.\mathrm{X}_3 = 0$, etc.) – so you can see that initial displacements **u** happen to be the same as a mode shape, the vibration will be harmonic.

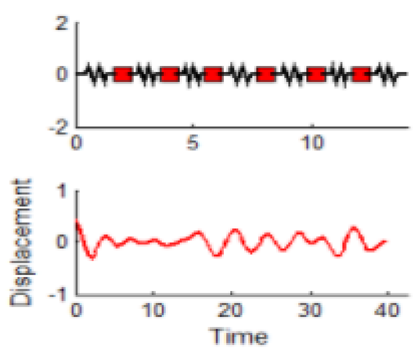


Figure I-9 The motion of a system with 6 masses

The figure on the right animates the motion of a system with 6 masses, which is set in motion by displacing the leftmost mass and releasing it. The graph shows the displacement of the leftmost mass as a function of time.

I.4 Transient and steady state response to harmonic excitation:

I.4.1 Harmonic vibration of undamped systems:

P(t) = $P_0\cos(\omega t)$

The harmonic force is or

P(t) = $P_0\sin(\omega t)$

where: P_0 = the amplitude

ω = forcing frequency

T = forcing period = $\frac{2\pi}{\omega}$

The equation of motion of this undamped system due to harmonic vibrations is:

$$m\ddot{u} + ku = P_0\sin\omega t$$

for this equation we have two solutions (a particular solution and a complementary solution):

- the particular solution is: $u_p(t)$
- and the complementary solution is: $u_c(t) = A\cos\omega_n t + B\sin\omega_n t$

after imposing the initial conditions, we can determine the constants A and B and the final solution will be:

$$u(t) = \underbrace{u_0\cos\omega_n t + \left[\frac{\dot{u}_0}{\omega_n} - \frac{p_0}{k}\frac{\omega/\omega_n}{1-(\omega/\omega_n)^2}\right]\sin\omega_n t}_{\text{transient}}$$

$$+\underbrace{\frac{p_0}{k}\frac{1}{1-(\omega/\omega_n)^2}\sin\omega t}_{\text{steady state}}$$

look at detail:

As we find the steady state response is equal to: $x(t) = \frac{p_0}{k}\frac{1}{1-\left(\frac{\omega}{\omega_n}\right)^2}\sin \omega t$

And the steady state amplitude is $x0 \ = \frac{p_0}{k}\frac{1}{1-\left(\frac{\omega}{\omega_n}\right)^2}$

in this formulation : $\frac{p_0}{k}$ represent the static response $\left(xst \ = \frac{p_0}{k}\right)$

so, the static response is multiplied by another factor $\left(\frac{1}{1-\left(\frac{\omega}{\omega_n}\right)^2}\right)$

here we can find **3 cases:**

1/ if that factor is **greater** than 1 that means ⟶ the dynamic response will be **greater than** the static response.

2/ and if that factor is **less** than 1 that means → the dynamic response will have an amplitude even **less** than the static response.

3/ and if that factor is **equal** to 1 that means → the dynamic or the steady state response will be **equal** to the static response.

In the Figure I-10 we can see in detail how this factor will vary with the value of ratio frequency $\left(\frac{\omega}{\omega_n}\right)$

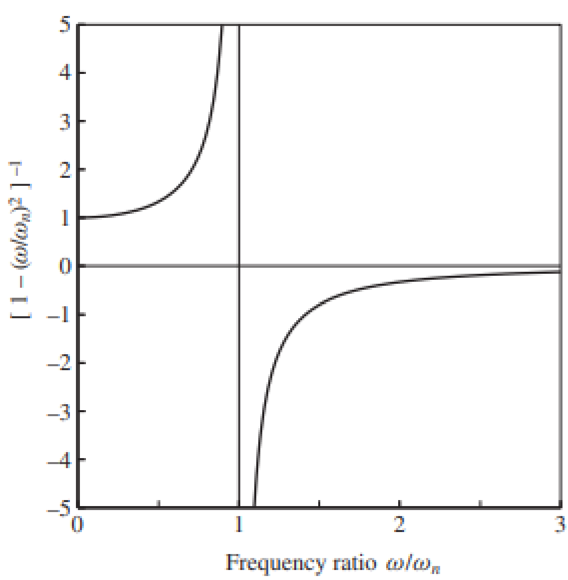


Figure I-10 The Frequency ratio diagram for the undamped system

I.4.2 Harmonic vibration of damped systems:

The equation of motion of this damped system due to harmonic vibrations is:

$$m\ddot{u} + C\dot{u} + ku = P_0 \sin\omega t$$

as we already solved in the undamped system, we will use the initial conditions to solve this equation.

and also, we will find that we have two solutions (a particular solution and a complementary solution):

- the particular solution is: $u_p(t) = C\cos\omega_n t + D\sin\omega_n t$

where:

$$C = \frac{p_0}{k}\frac{1-\left(\frac{\omega}{\omega_n}\right)^2}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left[2\xi\left(\frac{\omega}{\omega_2}\right)\right]^2}$$

$$D = \frac{p_0}{k}\frac{-2\xi\omega/\omega_n}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left[2\xi\left(\frac{\omega}{\omega_2}\right)\right]^2}$$

- and the complementary solution is:

$$u_c(t) = e^{-\xi\omega_n t}(A\cos\omega_D t + B\sin\omega_D t)$$

with $A = u_0$

$$B = \frac{\dot{u}_0}{\omega_n} - \frac{p_0}{k}\frac{\omega/\omega_n}{1-(\omega/\omega_n)^2}$$

$$\omega_D = \omega_n\sqrt{1-\xi^2}$$

The complete solution is: $u(t) = e^{-\xi\omega_n t}(A\cos\omega_D t + B\sin\omega_D t) + C\cos\omega_n t + D\sin\omega_n t$

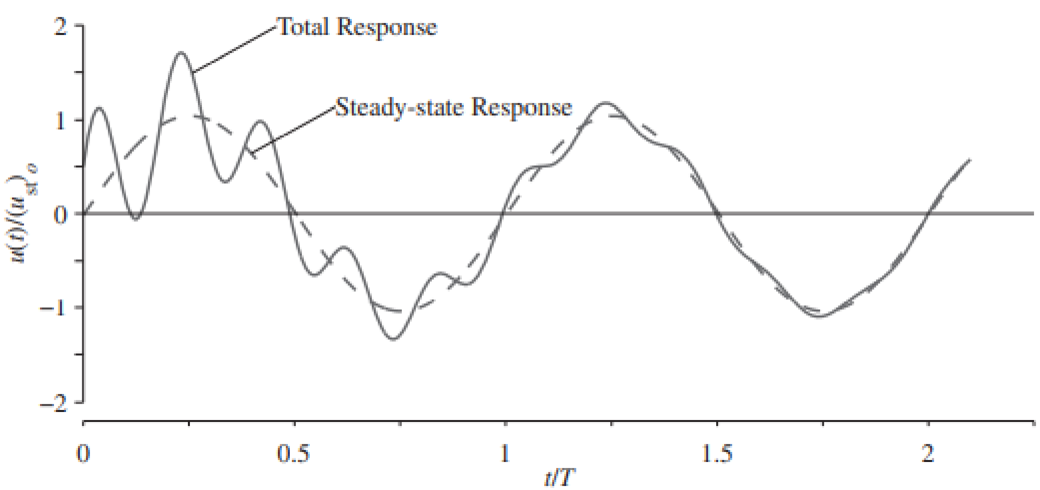


Figure I-11 Response of damped system to harmonic force

I.5 undamped steady state response and resonance:

As we already seen in the previous chapter the equation of motion of undamped system due to a harmonic vibration is:

$$m\ddot{u} + ku = P_0 \sin\omega t$$

and their solution is:

$$u(t) = \underbrace{u(0)\cos\omega_n t + \left[\frac{\dot{u}(0)}{\omega_n} - \frac{p_o}{k}\frac{\omega/\omega_n}{1-(\omega/\omega_n)^2}\right]\sin\omega_n t}_{\text{transient}} + \underbrace{\frac{p_o}{k}\frac{1}{1-(\omega/\omega_n)^2}\sin\omega t}_{\text{steady state}}$$

but we will focus just on the steady state solution which is: $x(t) = \frac{p_0}{k}\frac{1}{1-\left(\frac{\omega}{\omega_n}\right)^2}\sin\omega t$

The steady-state dynamic response, a sinusoidal oscillation at the forcing frequency, may be expressed as:

$$x(t) = (u_{st})_0 \frac{1}{1-\left(\frac{\omega}{\omega_n}\right)^2}\sin\omega t$$

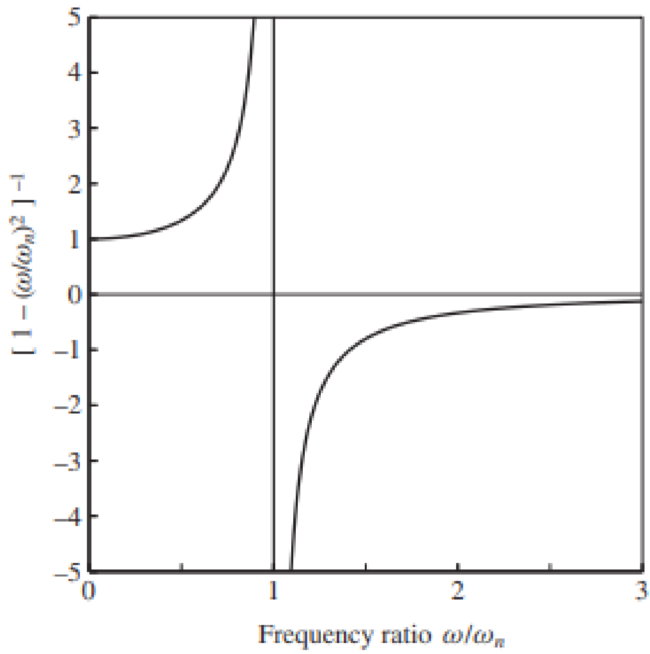


Figure I-12 The Frequency ratio diagram for the undamped system

Ignoring the dynamic effects signified by the acceleration term gives the static deformation (indicated by the subscript "st") at each instant:

$$u_{st}(t) = \frac{p_0}{k}\sin\omega t$$

The maximum value of the static deformation is:

$$(u_{st})_0 = \frac{p_0}{k}$$

which may be interpreted as the static deformation due to the amplitude P_0 of the force; for brevity we will refer to $(u_{st})_0$ as **the static deformation**. The factor in brackets $\frac{1}{1-\left(\frac{\omega}{\omega_n}\right)^2}$ as been plotted in Figure I-13 against ω/ω_n, the ratio of the forcing frequency to the natural frequency. For ω/ω_n < 1 or ω 1 or $\omega > \omega_n$ this factor is negative, indicating that u(t) and p(t) have opposing algebraic signs (i.e., when the force acts to the right, the system would be displaced to the left). The displacement is said to be out of phase relative to the applied force.

To describe this notion of phase mathematically, the equation is rewritten in terms of the amplitude u_o of the vibratory displacement *u(t)* and phase angle ϕ:

$$u(t) = u_o \sin(\omega t - \phi) = (u_{st})_o\, R_d \sin(\omega t - \phi)$$

Where

$$R_d = \frac{u_o}{(u_{st})_o} = \frac{1}{\left|1-\left(\frac{\omega}{\omega_n}\right)^2\right|} \; and \;\; \phi = \begin{cases} 0^\circ & \omega < \omega_n \\ 180^\circ & \omega > \omega_n \end{cases}$$

For $\omega < \omega_n$, $\phi = 0^\circ$, implying that the displacement varies as $\sin(\omega t)$, in phase with the applied force.
For $\omega > \omega_n$, $\phi = 180^\circ$, indicating that the displacement varies as $\sin(\omega t)$, out of phase relative to the force.

This phase angle is shown in Figure I-13 as a function of the frequency ratio ω/ω_n.
The deformation (or displacement) response factor R_d is the ratio of the amplitude $u_\circ$ of the dynamic (or vibratory) deformation to the static deformation $(u_{st})_\circ$.Figure I-13, which shows R_d plotted as a function of the frequency ratio ω/ω_n , permits several observations: If ω/ω_n is small (i.e., the force is “slowly varying”), R_d is only slightly larger than 1 and the amplitude of the dynamic deformation is essentially the same as the static deformation. If $\omega/\omega_n > \sqrt{2}$ (i.e., ω is higher than $\omega_n\sqrt{2}$), R_d <1 and the dynamic deformation amplitude is less than the static deformation.

As ω/ω_n increases beyond $\sqrt{2}$, R_d becomes smaller and approaches zero as $\omega/\omega_n \rightarrow\infty$, implying that the vibratory deformation due to a “rapidly varying” force is very small. If ω/ω_n is close to 1 (i.e., ω is close to ω_n), R_d is many times larger than 1, implying that the deformation amplitude is much larger than the static deformation.

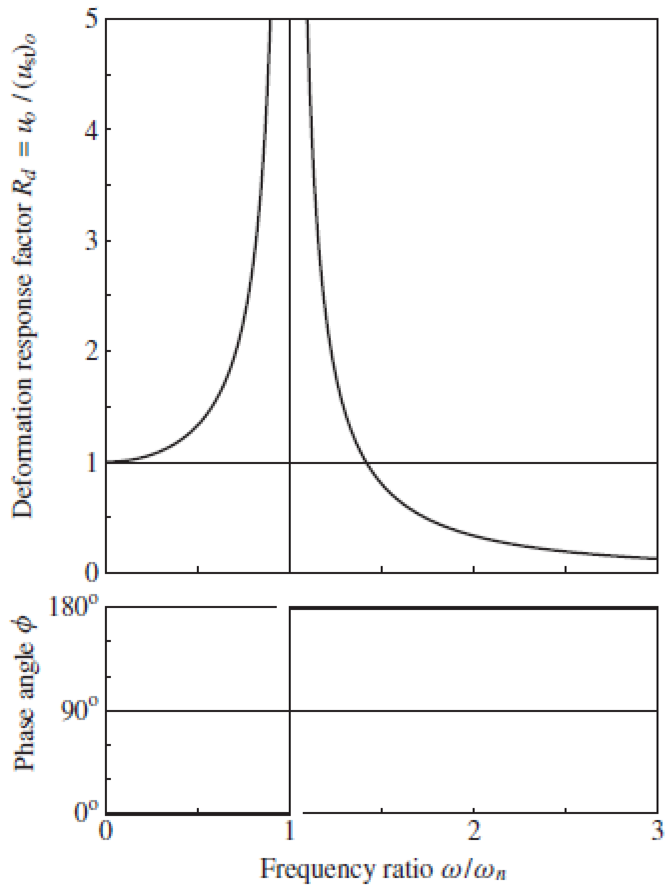


Figure I-13 Deformation response factor and phase

The resonant frequency is defined as the forcing frequency at which R_d is maximum. For an undamped system the resonant frequency is ω_n and R_d is unbounded at this frequency. The vibratory deformation does not become unbounded immediately, however, but gradually, as we demonstrate next.

If $\omega = \omega_n$, the solution given is no longer valid. In this case the choice of
the function $C\sin(\omega t)$for a particular solution fails because it is also a part of the complementary solution. The particular solution now is:

$u_p(t) = -\frac{p_o}{2k}\omega_n t\cos\omega_n t \qquad \omega = \omega_n$

and the complete solution for at-rest initial conditions, $u(0)= \dot{u}(0)=0$, is :

$$u(t) = -\frac{1}{2}\frac{p_o}{k}(\omega_n t\cos\omega_n t - \sin\omega_n t)$$

Or

$$\frac{u(t)}{(u_{st})_0} = -\frac{1}{2}\left(\frac{2\pi t}{T_n}\cos\frac{2\pi t}{T_n} - \sin\frac{2\pi t}{T_n}\right)$$

This result is plotted in Figure I-14, which shows that the time taken to complete one cycle of vibration is $\boldsymbol{T_n}$. The local maxima of $u(t)$, which occur at $t=(j-1/2)T_n$, are $\pi(j-1/2)\ (u_{st})_o$—j=1, 2, 3, . . .— and the local minima, which occur at $t = jT_n$, are $-\pi j(u_{st})_o$—j = 1, 2, 3, ... In each cycle the deformation amplitude increases by :

$$|u_{j+1}| - |u_j| = (u_{st})_o\ [\pi(j+1)-\pi j] = \frac{\pi p_o}{k}$$

The deformation amplitude grows indefinitely, but it becomes infinite only after an infinitely long time.

This is an academic result and should be interpreted appropriately for real structures. As the deformation continues to increase, at some point in time the system would fail if it is brittle. On the other hand, the system would yield if it is ductile, its stiffness would decrease, and its “natural frequency” would no longer be equal to the forcing frequency, and Figure I-14 would no longer be valid.

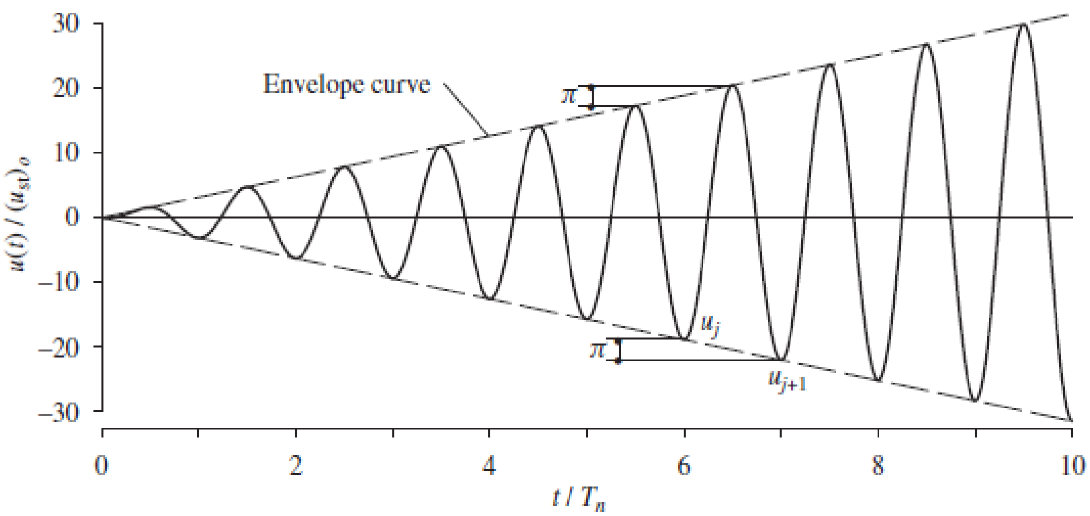


Figure I-14 Response of undamped system to sinusoidal force of frequency ω = ωn;

I.6 Conclusion:

In this chapter we started from giving a theoretical background about the basics of structural dynamics, passing by explaining what vibrations are, and giving some terminology for better understanding of the next chapters. Afterwards, it talks about single and multi-degree of freedoms, then shows the difference between transient and steady state response, the end of the chapter goes into more detail and focuses on the undamped steady state response.

II. Steel Frame Design

II.1 Introduction:

The work study herein is industrial steel warehouse with a mezzanine floor, Located at Benchaben-Boufarik, Blida, a region considered as a high seismic activity area (III) according to Algerian seismic code (RPA 99 modify 2003).

Figure II-1 warehouse's location using Google MAPS.

Table 2.1 Basic information

Information	Description
Structural system	Steel warehouse with a mezzanine floor
Number of storeys	one story + mezzanine floor
Floor heights	Eave height : 9 m Ridge height (clear height) : 10,75 m Mezzanine floor height : 4,9 m
Building Length	54 m
Building Width	27,2 m
Bay spacing	6 m
Pitch angel	7,25 degree (in the left side) / 7,40 degree (in the right side)
Wind Zone	I (26 m/s)
Seismic zone	III
Snow zone	B
altitud (above sea leve	68 m

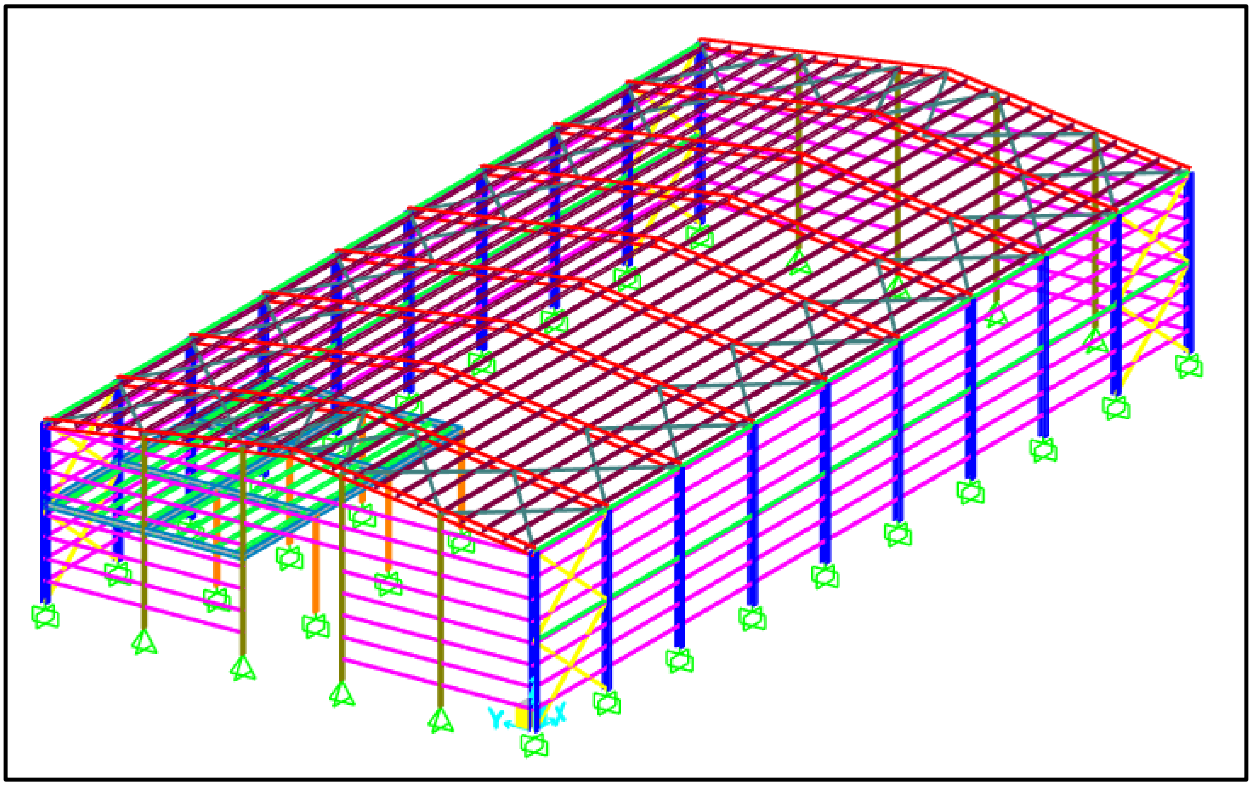


Figure II-2 3D view of the structure.

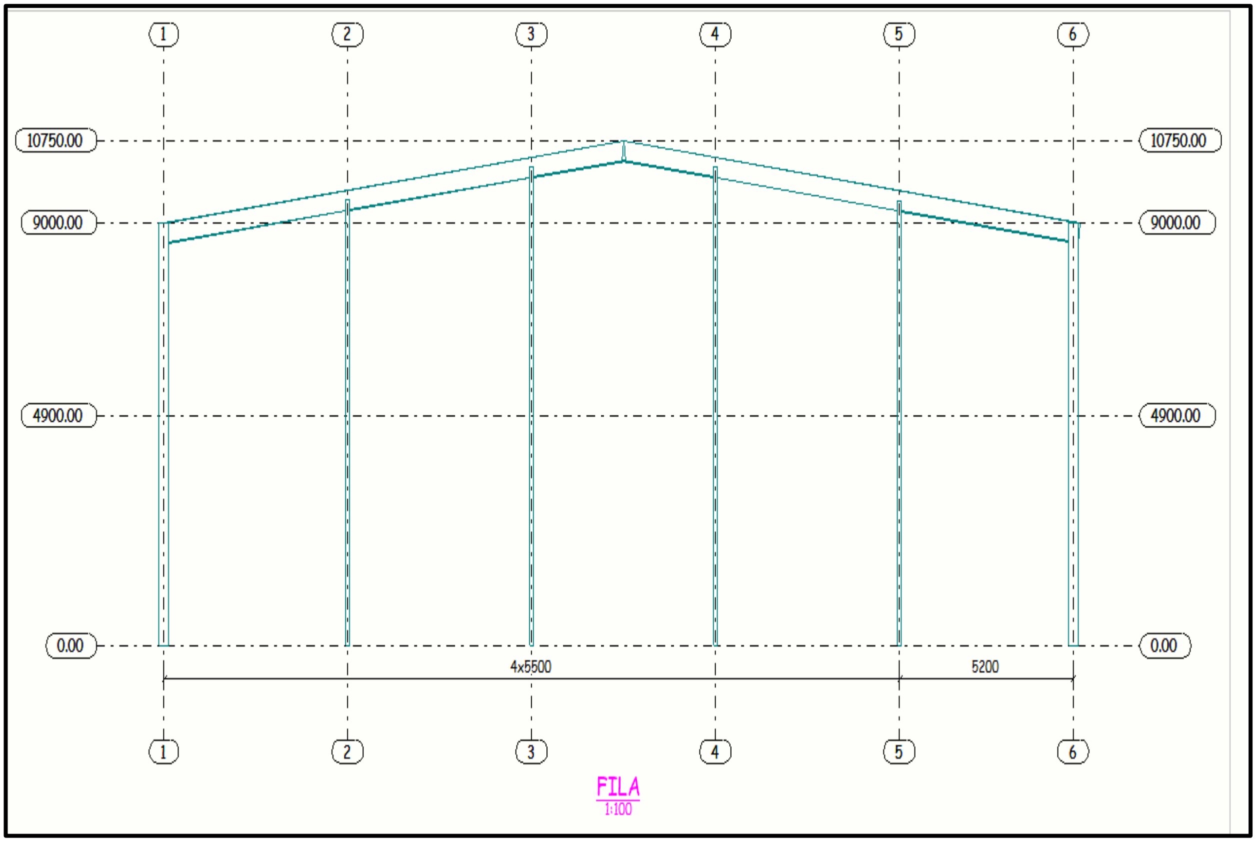


Figure II-3 2D view of the gable wall by TEKLA

II.3.1 Wind load (W1/W2):

Table II-1 Wind zone.

Zone	I
q_{ref} (N/m^2)	375

Table II-2 Terrain category

Category	Kt	Z_0(m)	Z_{min}(m)	ε
III	0.215	0.3	5	0.61

Table II-3 Topography coefficient

Site	Ct
Flat site	1

Table II-4 Roughness coefficient Cr

For : Zmin ≤ Z ≤ 200 m		For : Z < Zmin
	Cr	
Gable wall	0,77	0,60
Side wall	0,73	0,60
Roof	0,77	0,60

Table II-5 Turbulence intensity Iv

For : Z > Zmin		For : Z ≤ Zmin
	Iv	
Gable wall	0,28	0,36
Side wall	0,29	0,36
Roof	0,28	0,36

Table II-6 Exposure Coefficient Ce

	Ce
Gable wall	1,75
Paroi vertical longpan	1,64
Roof	1,75

Table II-7 Peak velocity pressure qp (N/m2)

	qp (N/m^2)
Gable wall	656,29
Side wall	613,23
Roof	656,29

For The Direction V1:

1. External pressure coefficient Cpe :

For the side wall:

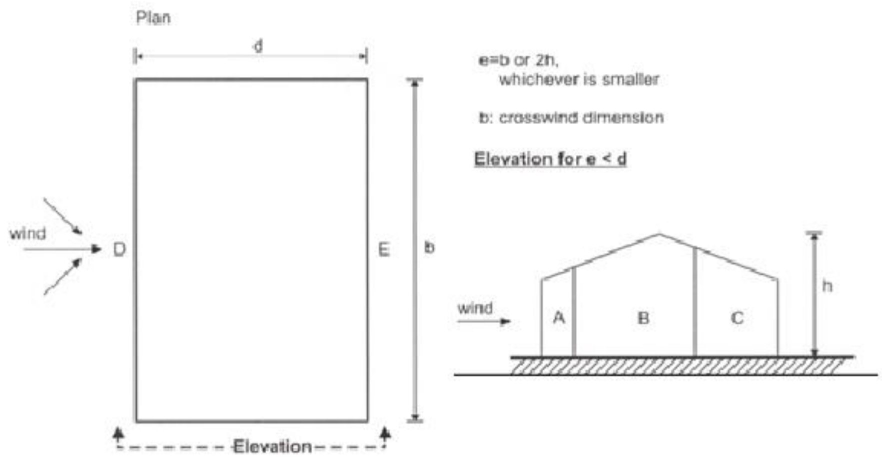


Figure II-4 Key for vertical walls

b	54	m
d	27,1	m
h	9	m

e = min[b,2h]

e	18	m
d > e	first cas	
d ≤ e	second cas	

Table II-8 Cpe coefficient and areas for the side wall zone(Direction V1)

e/5	3,6	Zones	A	B	C	D	E
(4/5)e	14,4	Area m²	32,4	129,6	82,8	486	486
d-e	9,2	Cpe	-1	-0,8	-0,5	0,8	-0,3
			Cpe=Cpe10				

and for the roof :

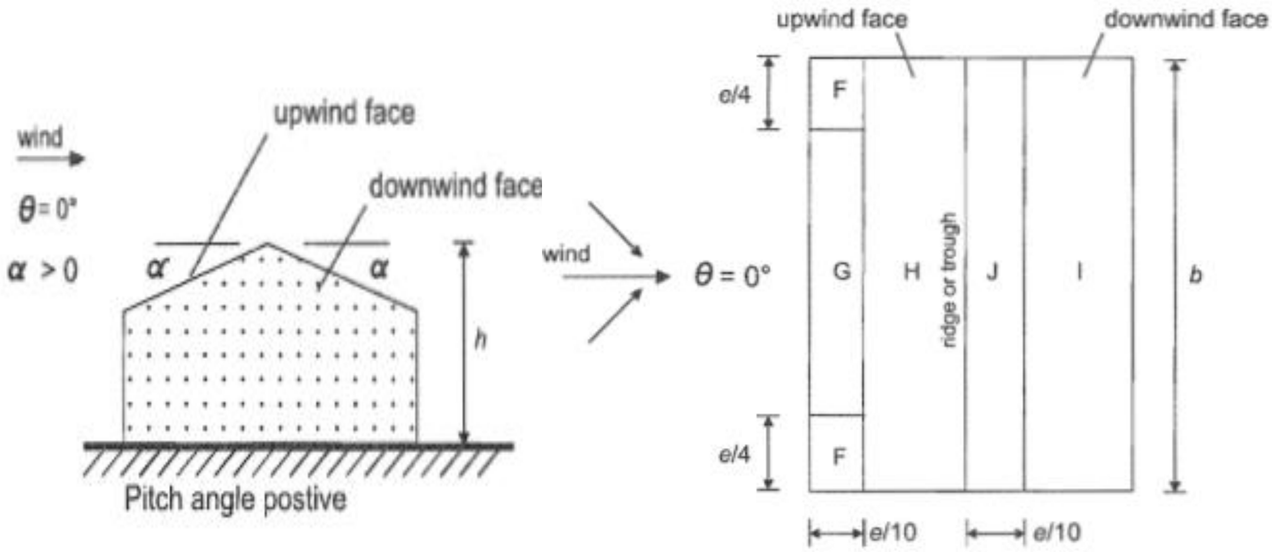


Figure II-5 Key for flat roofs

b	54	m
d	27,2	m
h	10,75	m

e = min[b,2h]

e	21,5	m

Table II-9 Cpe coefficient and areas for the roof zone(Direction V1)

e/5	4,3		Zones	F		G	H	I	J
(4/5)e	17,2		Area (m^2)	11,556		92,988	618,300	618,300	116,100
e/4	5,375	α	5,00	Cpe10	Cpe1	Cpe=Cpe10			
e/10	2,15			-1,70	-2,50	-1,20	-0,60	-0,60	0,20
α	7,41			0,00		0,00	0,00		-0,60
			Cpe(-)	-1,65					
			Cpe(+)	0,00					
		α	15,00	-0,90	-2,00	-0,80	-0,30	-0,40	-1,00
				0,20		0,20	0,20	0,00	0,00
			Cpe(-)	-0,83					
			Cpe(+)	0,20					
		α	7,41	-1,452		-1,104	-0,528	-0,552	-0,696
				0,048		0,048	0,048	-0,552	0,152

2. Internal pressure coefficient Cpi:

Because we don't have enough information about the **openings** in our structure, we can use this note from the Eurocod 1 part 1.4 (wind actions) ≪ Where it is not possible, or not considered justified, to estimate μ for a particular case then C_{pi} should be taken as the more oneros of +0,2 and -0,3 ≫.

3. Aerodynamic pressure W:

Side wall e = 1,1 m (the spacing between the Girts)

Table II-10 Aerodynamic pressure value for the side wall in the direction V1

for Cpi= 0,2

Zones	Area(m²)	qp(N/m²)	Cpe	Cpi	W(N/m²)	W*e (KN/ m)
A	32,4	613,230	-1,000	0,200	-735,876	-0,81
B	129,6	613,230	-0,800	0,200	-613,230	-0,67
C	82,8	613,230	-0,500	0,200	-429,261	-0,47
D	486	613,230	0,800	0,200	367,938	0,40
E	486	613,230	-0,300	0,200	-306,615	-0,34

for Cpi= -0,3

Zones	Arae(m²)	qp(N/m²)	Cpe	Cpi	W(N/m²)	W*e (KN/ m)
A	32,4	613,230	-1,000	-0,300	-429,261	-0,47
B	129,6	613,230	-0,800	-0,300	-306,615	-0,34
C	82,8	613,230	-0,500	-0,300	-122,646	-0,13
D	486	613,230	0,800	-0,300	674,553	0,74
E	486	613,230	-0,300	-0,300	0,000	0,00

Roof e = 1,2 m (the spacing between the Purlins)

Table II-11 Aerodynamic pressure value for the roof in the direction V1

for Cpi= 0,2

Zones	qp(N/m²)	Surf(m2)	Cpe(-)	Cpe(+)	Cpi	W(N/m²)	W(N/m²)	W*e'(kN/ m)
F	656,288	11,556	-1,452	0,048	0,2000	-1 084,454	-99,625	**-1,30**
G	656,288	92,988	-1,104	0,048	0,2000	-855,538	-99,625	**-1,03**
H	656,288	498,15	-0,528	0,048	0,2000	-477,581	-99,625	**-0,57**
I	656,288	498,15	-0,552	/	0,2000	-493,398	/	**-0,59**
J	656,288	116,1	-0,696	0,152	0,2000	-588,297	-31,633	**-0,71**

for Cpi= -0,3

Zones	qp(N/m²)	Surf(m2)	Cpe(-)	Cpe(+)	Cpi	W(N/m²)	W(N/m²)	W*e'(kN/ m)
F	656,288	11,556	-1,452	0,048	-0,3000	-756,310	228,520	**-0,91**
G	656,288	92,988	-1,104	0,048	-0,3000	-527,393	228,520	**-0,63**
H	656,288	498,15	-0,528	0,048	-0,3000	-149,437	228,520	**-0,18**
I	656,288	498,15	-0,552	/	-0,3000	-165,253	/	**-0,20**
J	656,288	116,1	-0,696	0,152	-0,3000	-260,153	296,511	**-0,31**

For The Direction V2:

1. External pressure coefficient Cpe :

For the side wall:

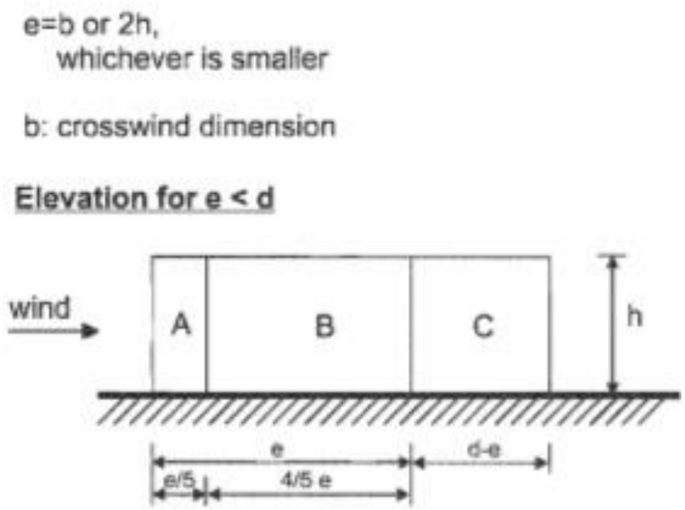


Figure II-6 Key for vertical walls

b	27,2	m
d	54	m
h	10,75	m

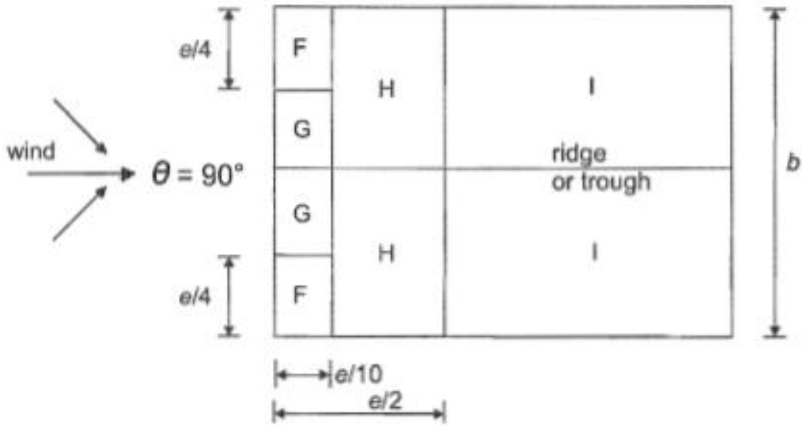
e = min[b,2h]

e	21,5	m
d > e	First cas	
d ≤ e	second cas	

Table II-12 Cpe coefficient and areas for the side wall zone(Direction V2)

e/5	4,3	**Zones**	**A**	**B**	**C**	**D**	**E**
(4/5)e	17,2	**Area m²**	46,225	184,9	349,375	292,4	292,4
d-e	32,5	**Cpe**	-1	-0,8	-0,5	0,8	-0,3
			Cpe=Cpe10				

for the roof:



(c) wind direction θ= 90°

Figure II-7 Key for flat roofs

b	27,2	m
d	54	m
h	10,75	m

e = min[b,2h]

e	21,5	m

Table II-13 Cpe coefficient and areas for the roof zone(Direction V2)

<table>
<tr><td>e/5</td><td>4,3</td><td>Zones</td><td colspan="2">F</td><td>G</td><td>H</td><td>I</td></tr>
<tr><td>(4/5)e</td><td>17,2</td><td>Area (m²)</td><td colspan="2">11,556</td><td>36,98</td><td>292,40</td><td>1176,40</td></tr>
<tr><td>e/4</td><td>5,375</td><td>/</td><td>CPE10</td><td>Cpe1</td><td>-1,30</td><td>-0,70</td><td>-0,60</td></tr>
<tr><td>e/10</td><td>2,15</td><td rowspan="2">5,00</td><td>-1,60</td><td>-2,20</td><td>-1,30</td><td>-0,70</td><td>-0,60</td></tr>
<tr><td>α</td><td>7,41</td><td colspan="2">-1,562</td><td colspan="3">Cpe=Cpe10</td></tr>
<tr><td colspan="2" rowspan="3"></td><td rowspan="2">15,00</td><td>-1,30</td><td>-2,00</td><td>-1,30</td><td>-0,60</td><td>-0,50</td></tr>
<tr><td colspan="2">-1,256</td><td colspan="3">Cpe=Cpe10</td></tr>
<tr><td>7,41</td><td colspan="2">-1,330</td><td>-1,30</td><td>-0,62</td><td>-0,52</td></tr>
</table>

2. Internal pressure coefficient Cpi :

For the Cpi we will take the same values as the first direction (Cpi = 0,2 / Cpi = -0,3)

3. Aerodynamic pressure W:

Side wall e = 1,1 m (the spacing between the Girts)

Table II-14 Aerodynamic pressure value for the side wall in the direction V2

Zones	Area(m²)	qp(N/m2)	Cpe	Cpi	W(N/m²)	W*e (KN/m)
A	46,225	656,288	-1,000	0,200	-787,546	-0,87
B	184,9	656,288	-0,800	0,200	-656,288	-0,72
C	349,375	656,288	-0,500	0,200	-459,402	-0,51
D	292,4	656,288	0,800	0,200	393,773	0,43
E	292,4	656,288	-0,300	0,200	-328,144	-0,36

Zones	Area(m^2)	qp(N/m2)	Cpe	Cpi	W(N/m²)	W*e (KN/m)
A	46,225	656,288	-1,000	-0,300	-459,402	-0,51
B	184,9	656,288	-0,800	-0,300	-328,144	-0,36
C	349,375	656,288	-0,500	-0,300	-131,258	-0,14
D	292,4	656,288	0,800	-0,300	721,917	0,79
E	292,4	656,288	-0,300	-0,300	0,000	0,00

Roof e = 1,2 m (the spacing between the Purlins)

Table II-15 Aerodynamic pressure value for the roof in the direction V2

for Cpi= 0,2

Zones	Area(m²)	qp(N/m²)	Cpe	Cpi	W(N/m²)	W*e'(kN/m)
F	11,556	656,288	-1,330	0,200	-1 004,018	**-1,20**
G	36,980	656,288	-1,300	0,200	-984,433	**-1,18**
H	292,400	656,288	-0,624	0,200	-540,847	**-0,65**
I	1 176,400	656,288	-0,524	0,200	-475,218	**-0,57**

for Cpi= -0,3

Zones	Area(m²)	qp(N/m2)	Cpe	Cpi	W(N/m²)	W*e'(kN/m)
F	11,556	656,288	-1,330	-0,300	-675,873	**-0,81**
G	36,980	656,288	-1,300	-0,300	-656,288	**-0,79**
H	292,400	656,288	-0,624	-0,300	-212,703	**-0,26**
I	1 176,400	656,288	-0,524	-0,300	-147,074	**-0,18**

- **The results for the wind loading:**

For the first direction:

Side wall

Table II-16 The wind loading on the side wall for V1

Zones	W1(KN/m)
A	-0,47
B	-0,34
C	-0,13
D	0,74
E	0,00

Roof

Table II-17 The wind loading on the roof for V1

Zones	W1(KN/m)
F	-1,30
G	-1,03
H	-0,57
I	-0,59
J	-0,71

And for the second direction:

Side wall

Table II-18 The wind loading on the side wall for V2

Zones	W2(KN/m)
A	-0,51
B	-0,36
C	-0,14
D	0,79
E	0,00

Roof

Table II-19 The wind loading on the roof for V2

Zones	W2(KN/m)
F	-1,20
G	-1,18
H	-0,65
I	-0,57

II.3.2 Snow Load (Sn):

Snow loads n roofs shall be determined as follows:

$$\mathrm{Sn} = (\mu \times S_k) \times e$$

where:

μ : is the snow load shape coefficient. $\mu = 0.8$ (for $0° < \alpha < 30°$)

S_k : is the characteristic value of snow load on the ground

$$S_k = \frac{0{,}04.H+10}{100} \quad \text{for the zone B}$$

For H= 68 m (altitude above the sea level) $\Longrightarrow S_k = 0{,}127 \cong 0{,}13$ Kn / m^2

e= 1,2 m (the spacing between the Purlins)

$$\boldsymbol{Sn = 0{,}13 \times 0{,}8 \times 1{,}2 = 0{,}125 \text{ Kn/m}}$$

II.3.3 Dead Load (DL):

Steel self-weight is taken by SAP2000.

We will increase the Dead load by 20% because there's several elements they're not considered in the self-weight of structure.

Dead Load = self-weight (SAP2000)×1,2

II.3.4 Super Dead Load (SDL):

A super dead load is the weight of the cladding (for the roof and the wall)

For the roof cladding we are using **TL 75** which he has 0,14 Kn/m^2

And for the wall cladding we are using **LL40** with 0,11 Kn/m^2

After we multiplied the panels weight with the spacing (e = 1,2 for the roof and e = 1,1 for the wall) we will find the following results:

SDL(Roof) = 0,202 Kn/m

SDL(wall) = 0.121 Kn/m

II.3.5 Live Loads (LL):

$$\mathbf{LL} = \boldsymbol{q_k} \times \boldsymbol{e}$$

q_k : is the recommanded value for roofs that are not accessible except for maintenance and repair, (its equal to 0,4 kn/m^2).

$\mathrm{e} = 1{,}2$ m (the spacing between the Purlins)

$\mathrm{LL} = 0{,}4 \times 1{,}2$

LL = 0,48 Kn/m

II.3.6 Seismic Loads (Ex/Ey):

In structural engineering it's a common knowledge that earthquakes have a negligible effect on warehouses due to its light weight, but this isn't the case in all structures, that's why we need to check this in our case study.

To do this we will follow 4 steps:

- Calculate the base shear force with the equivalent static method.
- Calculate the base shear force with the modal response spectrum method.

- Comparing the calculate force of the static equivalent method multiply by 0,8 with the base shear force of the modal response spectrum method (from the SAP2000), and choose the max between them.
- Comparing the seismic load with the wind load and check who has the major effect on our structure.

II.3.6.1 The base shear force of static method (V_{stat}):

The height of our structure is 10,75m with single story and we are in a Seismic zone **III** and a use group **2,** According to The RPA the conditions of application of the static equivalent method are verified and we can apply it on our case.

$$V_{stat} = \frac{A \times D \times Q \times W}{R}$$

where:

- $A = 0{,}25$ (Zone acceleration factor)
- $R = 4$ (Structural behavior factor, depends on the structure's bracing system)
- $Q = 1{,}2$ (Quality factor)
- $W \approx 3004$ KN (The weight of the structure obtained from the SAP2000)
- $Dx = Dy = 2{,}5\eta$ for $0 \leq T \leq T2$ where: $Tx = 0{,}297$ /$Ty = 0{,}186$ / $T_2 = 0{,}5$ / $\eta = 1{,}08$

  (D: Dynamic amplification factor. This factor is a function of the site category, the damping correction factor "η", and the fundamental period of the structure "T".)

$$\boldsymbol{Vx_{stat} = Vy_{stat} = 608{,}3\ KN}$$

II.3.6.2 The base shear force of response spectrum (V_{dym}):

The most common procedure to define the seismic force is by using a response spectrum. Every structure can be represented as a multi-oscillator system, the response of the structure to a dynamic acceleration is a function of the damping factor (ξ) and the natural frequency (ω). If we evaluate the maximum response in function of the period "T" of a given accelerogram, we obtain a graph named "response spectrum". The seismic force is represented by the following response spectrum:

$$\frac{S_a}{g}=\begin{cases} 1,25A\left(1+\frac{T}{T_1}\left(2,5\eta\frac{Q}{R}-1\right)\right) & 0\le T\le T_1 \\ 2,5\eta\left(1,25A\right)\frac{Q}{R} & T_1\le T\le T_2 \\ 2,5\eta\left(1,25A\right)\frac{Q}{R}\left(\frac{T_2}{T}\right)^{2/3} & T_2\le T\le 3,0s \\ 2,5\eta\left(1,25A\right)\frac{Q}{R}\left(\frac{T_2}{3}\right)^{2/3}\left(\frac{3}{T}\right)^{5/3} & T\ge 3,0s \end{cases}$$

Graphical Representation of the response spectrum:

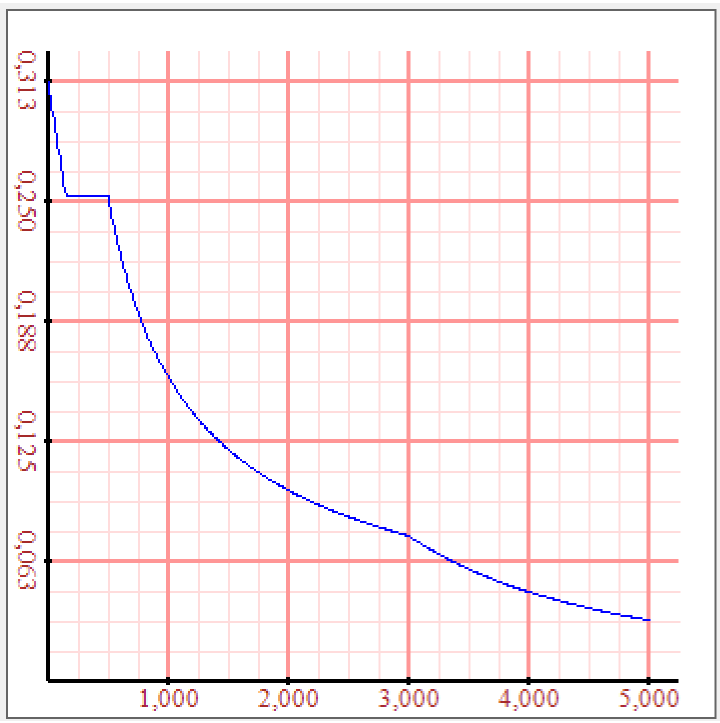


Figure II-8 Graphical Representation of the response spectrum

After using an application to obtain the spectrum responses and add these resultants the SAP2000 program we can acquire the shear forces resultant at the base of the structure due to the seismic jolts:

	Fx	Fy
Ex max	87.893	1.345
Ey max	1.345	265.022

$$V_{x,dyn} = \sqrt{F_x^2 + F_y^2} = 87.90\ KN$$

$$V_{y,dyn} = \sqrt{F_x^2 + F_y^2} = 265.03\ KN$$

II.3.6.3 Comparing V_{stat} and V_{dym}:

$$V_{stat} = 608{,}3 \text{ KN} \Rightarrow 0{,}8 * \text{V}_{\text{stat}} = 486{,}65 \text{ KN}$$

$$0.8V_{x.stat} = 486{,}65\ KN \quad > \quad V_{x.dym} = 87.90\ KN$$

$$0.8V_{y.stat} = 486{,}65\ KN \quad > \quad V_{y.dym} = 265.03\ KN$$

As we can see the base shear force of the static equivalent method is bigger than the base shear force of the response spectrum method in both directions, therefore this base shear force of the static equivalent method will represent the seismic loads.

II.3.6.4 Comparing the seismic loads with wind loads:

The wind loads on the base of the structure are:

	Fx	Fy	Fz
W1	0.0001	312.787	-782.511
W2	-150.179	-1.801	-750.491

$$W_x = \sqrt{F_y^2 + F_z^2} = 842.71\ KN$$

$$W_y = \sqrt{F_X^2 + F_z^2} = 765.37\ KN$$

$$W_x = 842.71\ KN > \quad V_{x.stat} = 486{,}65\ KN$$

$$W_y = 765.37\ KN > \quad V_{x.stat} = 486{,}65\ KN$$

As we can see the wind is the major load in the both directions (x, y), that means it’s the governing load who will affect on our warehouse.

Therefore, the seismic loads will have a negligible effect of the structure.

II.4 Load Combination:

Depending on the load type, the following combinations can be distinguished:

Table II-20 The Combinations for ULS and SLS(The new combination from the CNEREP)

ComboType	ComboName	CaseName	ScaleFactor
ULS			
	1.35DL+1.35SDL+1.5LL	DEAD	1.35
		LL	1.5
		SDL	1.35
	1.35DL+1.35SDL+1.5Sn	DEAD	1.35
		Sn	1.5
		SDL	1.35
	1.35G+1.35SDL+1.5W1	DEAD	1.35
		W1	1.5
		SDL	1.35
	1.35DL+1.35SDL+1.5W2	DEAD	1.35
		W2	1.5
		SDL	1.35
	DL+SDL+1.5W1	DEAD	1
		W1	1.5
		SDL	1
	DL+SDL+1.5W2	DEAD	1
		W2	1.5
		SDL	1
	1.35DL+1.35SDL+1.5LL+0.9Sn+0.9W1	DEAD	1.35
		SDL	1.35
		LL	1.5
		Sn	0.9
		W1	0.9
	1.35DL+1.35SDL+1.5LL+0.9Sn	DEAD	1.35
		SDL	1.35
		LL	1.5
		Sn	0.9
	1.35DL+1.35SDL+1.5LL+0.9Sn+0.9W2	DEAD	1.35
		SDL	1.35
		LL	1.5
		Sn	0.9
		W2	0.9
	1.35DL+1.35SDL+1.5Sn+0.9W1+1.05LL	DEAD	1.35
		SDL	1.35
		Sn	1.5
		W1	0.9

		LL	1.05
	1.35DL+1.35SDL+1.5Sn+0.9W2+1.05LL	DEAD	1.35
		SDL	1.35
		Sn	1.5
		W2	0.9
		LL	1.05
	1.35DL+1.35SDL+1.5Sn+1.05LL	DEAD	1.35
		SDL	1.35
		Sn	1.5
		LL	1.05
	1.35DL+1.35SDL+1.5W1+1.05LL	DEAD	1.35
		SDL	1.35
		W1	1.5
		LL	1.05
	1.35DL+1.35SDL+1.5W2+1.05LL	DEAD	1.35
		SDL	1.35
		W2	1.5
		LL	1.05
	1.35DL+1.35SDL+1.5W1+0.9Sn+1.05LL	DEAD	1.35
		SDL	1.35
		W1	1.5
		Sn	0.9
		LL	1.05
	1.35DL+1.35SDL+1.5W2+0.9Sn+1.05LL	DEAD	1.35
		SDL	1.35
		W2	1.5
		Sn	0.9
		LL	1.05
SLS			
	DL+SDL+LL	DEAD	1
		LL	1
		SDL	1
	DL+SDL+Sn	DEAD	1
		Sn	1
		SDL	1
	DL+SDL+W1	DEAD	1
		W1	1
		SDL	1
	DL+SDL+W2	DEAD	1
		W2	1
		SDL	1

II.5 Members design check:

Herein in this table you can find the final section for all the members after all deisgn checks was done using the software SAP2000

Table II-21 the design sections

Members	Final sections
Colums	HEA 340
Mein beam (Rafter)	IPE 450
Wind colums	IPE 220
Girts	UPN 140
Purlins	IPE 160
Roof bracing	2L 90*9/10/
Wall bracing	2UPN 100/10/
Eave purlins	HEA 100
Mezzanine Colums	IPE 270
Mezzanine Beams	IPE 300
Mezzanine Joist	IPE 180

in

Figure II-9 a 3D model show us that all that structural members check have passed:

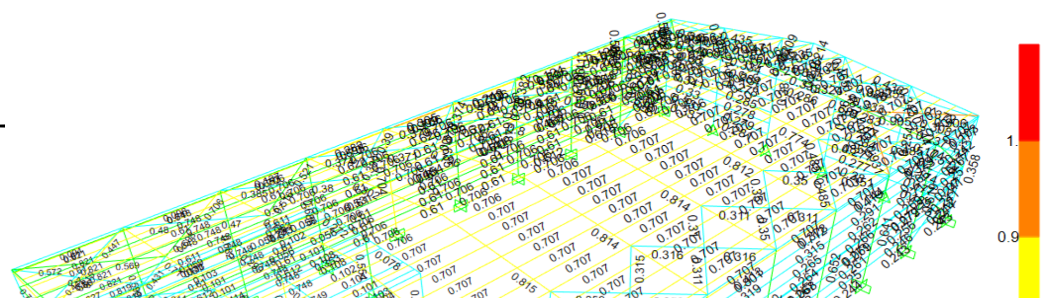


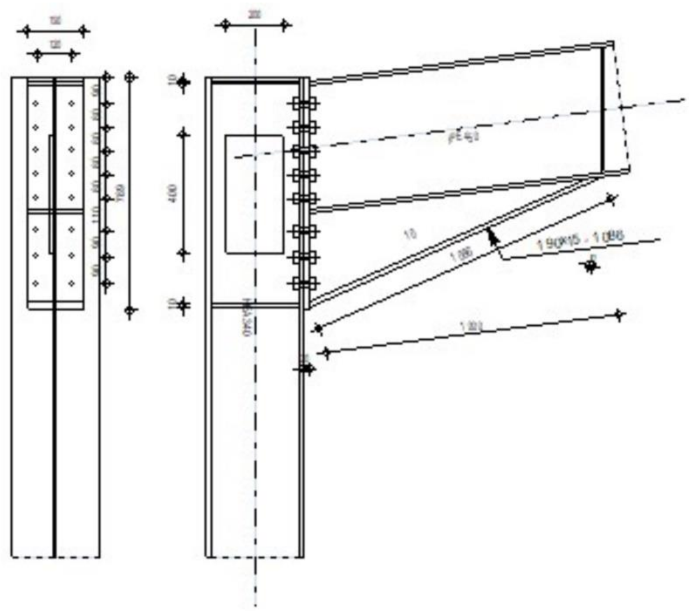
Figure II-9 Design of structural in 3D view (DCR Ration < 1.0 , All Passed).

II.6 Connection’s design:

The design of all the connections was done by the software Robot Structural Analysis Professional 2019.

II.6.1 Design of fixed beam-to-column connection:

	Autodesk Robot Structural Analysis Professional 2019 **Design of fixed beam-to-column connection** NF EN 1993-1-8:2005/NA:2007/AC:2009	OK Ratio **0,94**



General

Connection no°: 1
Connection name: Frame knee

Geometry

Column

Section HEA 340

α=	-90 [Deg]	Inclination angle	t_{fc} = 17 [mm]		Thickness of the flange of column section
h_c =	330 [mm]	Height of column section	r_c = 27 [mm]		Radius of column section fillet
b_{fc} =	300 [mm]	Width of column section	A_c = 133.47[cm^2]		Cross-sectional area of a column
t_{wc} =	10[mm]	Thickness of the web of column section	I_{xc} = 27693.10 [cm^2]		Moment of inertia of the column section

Material:: ACIER E28
f_{yc} = 275,00 [MPa] Resistance

Beam

Section: IPE 450

a =	7.2	[Deg]	Inclination angle
h_b =	450	[mm]	Height of beam section
b_f =	190	[mm]	Width of beam section
t_{wb} =	9	[mm]	Thickness of the web of beam section
t_{fb} =	15	[mm]	Thickness of the flange of beam section
r_b =	21	[mm]	Radius of beam section fillet
r_b =	21	[mm]	Radius of beam section fillet
A_b =	98.82	[cm^2]	Cross-sectional area of a beam
I_{xb} =	33742.90	[cm^4]	Moment of inertia of the beam section

Material: ACIER E28
f_{yb} = 275.00 [MPa] Resistance

Bolts

The shear plane passes through the UNTHREADED portion of the bolt.

d =	20	[mm]	Bolt diameter
Class =	HR 10.9		Bolt class
F_{tRd} =	176.40	[kN]	Tensile resistance of a bolt
n_h =	2		Number of bolt columns
n_v =	8		Number of bolt rows
h_1 =	90	[mm]	Distance between first bolt and upper edge of front plate

Horizontal spacing e_i = 120 [mm]
Vertical spacing p_i = 80;80;80;80;110;90;90 [mm]

Plate

h_p =	789	[mm]	Plate height
b_p =	190	[mm]	Plate width
t_p =	20	[mm]	Plate thickness
w_d =	190	mm	Plate width

Lower stiffener

t_{fd} =	15	mm	Plate width
h_d =	315	mm	Flange thickness
t_{wd} =	10	mm	Plate height
l_d =	1000	mm	Web thickness
α =	24,0		Plate length

Material : ACIER E24
f_{yp} = 235,00 [MPa] Resistance

Column stiffener

Upper		Lower	
h_{su} = 297 [mm]	Stiffener height	h_{sd} =297 [mm]	Stiffener height
b_{su} =145 [mm]	Stiffener width	b_{sd} =145 [mm]	Stiffener width
t_{hu} =10 [mm]	Stiffener thickness	t_{hd} =10 [mm]	Stiffener thickness

Material: ACIER E24
f_{ysu} = 235.00 [MPa] Resistance

Plate strengthening column web

Typ: unilateral

h_a =	400	[mm]	Plate length
w_a =	200	[mm]	Plate width
t_a =	10	[mm]	Plate thickness

Material: ACIER E24
f_{ya} = 235.00 [MPa] Resistance

Fillet welds

a_w =	7	[mm]	Web weld	a_{fd} =	7	[mm]	Horizontal weld
a_f =	8	[mm]	Flange weld	a_{p1} =	1	[mm]	Horizontal weld
a_s =	7	[mm]	Stiffener weld	a_{p2} =	1	[mm]	Vertical weld

Loads

Serviceability limit state

Case: Manual calculations.

$M_{b1,Ed,ser}$ =	-423.07	[kN*m]	Bending moment in the right beam
$V_{b1,Ed,ser}$ =	-110.15	[kN]	Shear force in the right beam
$N_{b1,Ed,ser}$ =	-94.20	[kN]	Axial force in the right beam

Ultimate limit state

Case: Manual calculations.

$M_{b1,Ed}$ =	-423.07	[kN*m]	Bending moment in the right beam
$V_{b1,Ed}$ =	-110.15	[kN]	Shear force in the right beam
$N_{b1,Ed}$ =	-94.20	[kN]	Axial force in the right beam

Fillet welds

a_w = 7 [mm] Web weld | a_{fd} = 7 [mm] Horizontal weld

$M_{c1,Ed}$ = 423.06 [kN*m] Bending moment in the lower column
$V_{c1,Ed}$ = -78.42 [kN] Shear force in the lower column
$N_{c1,Ed}$ = -119.61 [kN] Axial force in the lower column

Results

Beam resistances

$V_{b1,Ed} / V_{cb,Rd} \leq 1,0$ 0,08 < 1,00 verified (0,08)

Column resistances

$V_{wp,Ed} / V_{wp,Rd} \leq 1,0$ 0,70 < 1,00 verified (0,70)

Geometrical parameters of a connection

EFFECTIVE LENGTHS AND PARAMETERS - COLUMN FLANGE

Nr	m	m_x	e	e_x	p	$l_{eff,cp}$	$l_{eff,nc}$	$l_{eff,1}$	$l_{eff,2}$	$l_{eff,cp,g}$	$l_{eff,nc,g}$	$l_{eff,1,g}$	$l_{eff,2,g}$
1	34	-	90	-	90	211	258	211	258	196	179	179	179
2	34	-	90	-	90	211	247	211	247	180	90	90	90
3	34	-	90	-	100	211	247	211	247	200	100	100	100
4	34	-	90	-	95	211	247	211	247	190	95	95	95
5	34	-	90	-	80	211	247	211	247	160	80	80	80
6	34	-	90	-	80	211	247	211	247	160	80	80	80
7	34	-	90	-	80	211	247	211	247	160	80	80	80
8	34	-	90	-	80	211	255	211	255	186	171	171	171

EFFECTIVE LENGTHS AND PARAMETERS - FRONT PLATE

Nr	m	m_x	e	e_x	p	$l_{eff,cp}$	$l_{eff,nc}$	$l_{eff,1}$	$l_{eff,2}$	$l_{eff,cp,g}$	$l_{eff,nc,g}$	$l_{eff,1,g}$	$l_{eff,2,g}$
1	47	-	35	-	90	298	244	244	244	239	173	173	173
2	47	-	35	-	90	298	233	233	233	180	90	90	90
3	47	-	35	-	100	298	233	233	233	200	100	100	100
4	47	-	35	-	95	298	233	233	233	190	95	95	95

Connection resistance for compression

$N_{b1,Ed} / N_{j,Rd} \leq 1,0$ 0,05 < 1,00 verified (0,05)

Connection resistance for bending

$M_{b1,Ed} / M_{j,Rd} \leq 1,0$ 0,94 < 1,00 verified (0,94)

Connection resistance for shear

$V_{b1,Ed} / V_{j,Rd} \le 1,0$	0,06 < 1,00	verified	(0,06)

Bolt arrangement check due to slip in prestressed connection

$k_s =$	1,00		Coefficient for calculation of $F_{s,Rd}$	[3.9.1]
$\mu =$	0,50		Friction coefficient	[3.9.1]
$F_{p,C} =$	171,50	[kN]	Slippage resistance of a bolt	[3.9.1]

$V_{sj,Rd,ser} = k_s\, n_h\, n_v\, \mu\, F_{p,C} / \gamma_{M3}$ [3.9.1]

$V_{sj,Rd,ser} =$ 1247,27 [kN] Slippage resistance of a connection [3.9]

$V_{b1,Ed,ser} / V_{sj,Rd} \le 1,0$	0,09 < 1,00	verified	(0,09)

Weld resistance

$\sqrt{[\sigma_{\perp max}^2 + 3*(\tau_{\perp max}^2)]} \le f_u/(\beta_w*\gamma_{M2})$	199,55 < 365,00	verified	(0,55)
$\sqrt{[\sigma_{\perp}^2 + 3*(\tau_{\perp}^2+\tau_{II}^2)]} \le f_u/(\beta_w*\gamma_{M2})$	181,79 < 365,00	Verified	(0,50)
$\sigma_{\perp} \le 0.9*f_u/\gamma_{M2}$	99,77 < 262,80	verified	(0,38)

Connection stiffness

STIFFNESSES OF BOLT ROWS

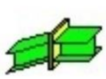
Nr	hj	k_3	k_4	k_5	$k_{eff,j}$	$k_{eff,j}\ h_j$	$k_{eff,j}\ h_j^2$
					Sum	37.50	1891.74
1	683	4	19	12	2	12.54	855.70
2	593	2	10	6	1	6.43	380.94
3	503	2	11	7	1	5.95	298.82
4	393	2	10	6	1	4.45	174.85
5	313	2	8	5	1	3.07	96.05
6	233	2	8	5	1	2.29	53.18
7	153	2	8	5	1	1.50	22.90
8	73	4	18	11	2	1.28	9.30

Weakest component:

COLUMN WEB PANEL - SHEAR

Connection conforms to the code Ratio 0.94

II.6.2 Design of fixed beam-to-beam connection:



Autodesk Robot Structural Analysis Professional 2019

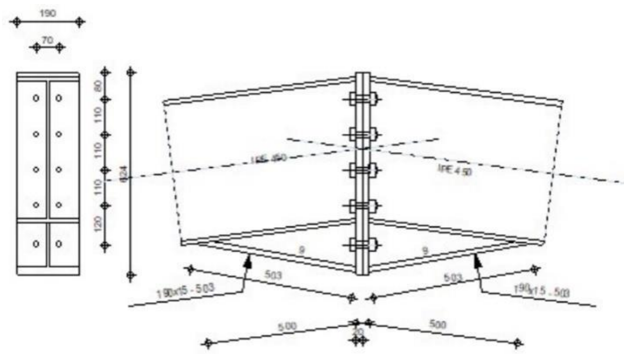
Design of fixed beam-to-beam connection

NF EN 1993-1-8:2005/NA:2007/AC:2009

OK

Ratio **0,67**

General



Connection no.: 2

Connection name:: Beam-Beam

Geometry

Left side Beam

Section:: IPE 450

α =	-172,8	[Deg]	Inclination angle
h_{bl} =	450	[mm]	Height of beam section
b_{fbl} =	190	[mm]	Width of beam section
t_{wbl} =	9	[mm]	Thickness of the web of beam section
t_{fbl} =	15	[mm]	Thickness of the flange of beam section
r_{bl} =	21	[mm]	Radius of beam section fillet
A_{bl} =	98,82	[cm^2]	Cross-sectional area of a beam
I_{xbl} =	33742,90	[cm^4]	Moment of inertia of the beam section

Material: ACIER E28

f_{yb} = 275,00 [MPa] Resistance

Right side Beam

Section: IPE 450

α =	-7,4	[Deg]	Inclination angle
h_{br} =	450	[mm]	Height of beam section
b_{fbr} =	190	[mm]	Width of beam section
t_{wbr} =	9	[mm]	Thickness of the web of beam section
t_{fbr} =	15	[mm]	Thickness of the flange of beam section
r_{br} =	21	[mm]	Radius of beam section fillet
A_{br} =	98,82	[cm^2]	Cross-sectional area of a beam
I_{xbr} =	33742,90	[cm^4]	Moment of inertia of the beam section

Material: ACIER E28

f_{yb} = 275,00 [MPa]

Bolts

The shear plane passes through the UNTHREADED portion of the bolt.

d =	20	[mm]	Bolt diameter
Class =	HR 10.9		Bolt class
F_{tRd} =	176.40	[kN]	Tensile resistance of a bolt
n_h =	2		Number of bolt columns
n_v =	5		Number of bolt rows
h_1 =	80	[mm]	Distance between first bolt and upper edge of front plate

Horizontal spacing e_i = 70 [mm]
Vertical spacing p_i = 110;110;110;120 [mm]

Plate

h_{pr} =	624	[mm]	Plate height
b_{pr} =	190	[mm]	Plate width
t_{pr} =	20	[mm]	Plate thickness

Material: ACIER E24
f_{ypr} = 235.00 [MPa] Resistance

Lower stiffener

w_{rd} =	190	[mm]	Plate width
t_{frd} =	15	[mm]	Flange thickness
h_{rd} =	150	[mm]	Plate height
t_{wrd} =	9	[mm]	Web thickness
l_{rd} =	500	[mm]	Plate length
a_d =	9.9	[Deg]	Inclination angle

Material: ACIER E28
f_{ybu} = 275.00 [MPa] Resistance

Fillet welds

a_w =	7	[mm]	Web weld
a_f =	8	[mm]	Flange weld
a_{fd} =	7	[mm]	Horizontal weld

Loads

Serviceability limit state
Case: `Manual calculations.`

$M_{b1,Ed,ser}$ =	257.30	[kN*m]	Bending moment in the right beam
$V_{b1,Ed,ser}$ =	6.21	[kN]	Shear force in the right beam
$N_{b1,Ed,ser}$ =	86.82	[kN]	Axial force in the right beam

Ultimate limit state
Case: `Manual calculations.`

$M_{b1,Ed}$ =	257.30	[kN*m]	Bending moment in the right beam
$V_{b1,Ed}$ =	6.21	[kN]	Shear force in the right beam
$N_{b1,Ed}$ =	86.82	[kN]	Axial force in the right beam

Results

Beam resistances

TENSION

$V_{b1,Ed} / V_{cb,Rd} \leq 1,0$	0,01 < 1,00	verified	(0,01)

Geometrical parameters of a connection

EFFECTIVE LENGTHS AND PARAMETERS - FRONT PLATE

Nr	m	m_x	e	e_x	p	$l_{eff,cp}$	$l_{eff,nc}$	$l_{eff,1}$	$l_{eff,2}$	$l_{eff,cp,g}$	$l_{eff,nc,g}$	$l_{eff,1,g}$	$l_{eff,2,g}$
1	22	-	60	-	110	141	164	141	164	180	137	137	137
2	22	-	60	-	110	141	165	141	165	220	110	110	110
3	22	-	60	-	110	141	165	141	165	220	110	110	110
4	22	-	60	-	115	141	165	141	165	230	115	115	115
5	22	-	60	-	120	141	165	141	165	190	142	142	142

Connection resistance for tension

$N_{b1,Ed} / N_{j,Rd} \leq 1,0$ 0,05 < 1,00 verified (0,05)

Connection resistance for bending

SUMMARY TABLE OF FORCES

Nr	h_j	$F_{tj,Rd}$	$F_{t,fc,Rd}$	$F_{t,wc,Rd}$	$F_{t,ep,Rd}$	$F_{t,wb,Rd}$	$F_{t,Rd}$	$B_{p,Rd}$
1	526	349,46	-	-	349,46	363,50	352,80	660,49
2	416	273,23	-	-	349,56	363,50	352,80	660,49
3	306	200,99	-	-	349,56	363,50	352,80	660,49
4	196	128,75	-	-	349,56	363,50	352,80	660,49
5	76	27,26	-	-	349,56	363,50	352,80	660,49

$M_{b1,Ed} / M_{j,Rd} \leq 1,0$ 0,67 < 1,00 verified (0,67)

Connection resistance for shear

SUMMARY TABLE OF FORCES

Nr	$F_{tj,Rd,N}$	$F_{tj,Ed,N}$	$F_{tj,Rd,M}$	$F_{tj,Ed,M}$	$F_{tj,Ed}$	$F_{vj,Rd}$
1	352,80	17,36	349,46	232,73	250,10	143,30
2	352,80	17,36	273,23	181,96	199,32	173,14
3	352,80	17,36	200,99	133,85	151,22	201,41
4	352,80	17,36	128,75	85,75	103,11	229,68
5	352,80	17,36	27,26	18,15	35,52	269,41

$V_{b1,Ed} / V_{j,Rd} \leq 1,0$ 0,01 < 1,00 verified (0,01)

Bolt arrangement check due to slip in prestressed connection

k_s =	1,00		Coefficient for calculation of $F_{s,Rd}$	[3.9.1]
μ =	0,50		Friction coefficient	[3.9.1]
$F_{p,C}$ =	171,50	[kN]	Slippage resistance of a bolt	[3.9.1]

$V_{b1,Ed,ser} / V_{sj,Rd} \leq 1,0$ 0,01 < 1,00 verified (0,01)

Weld resistance

$\sqrt{[\sigma_{\perp max}^2 + 3*(\tau_{\perp max}^2)]} \leq f_u/(\beta_w*\gamma_{M2})$ 173,85 < 365,00 verified (0,48)

$\sqrt{[\sigma_{\perp}^2 + 3*(\tau_{\perp}^2+\tau_{II}^2)]} \leq f_u/(\beta_w*\gamma_{M2})$ 154,08 < 365,00 verified (0,42)

$\sigma_{\perp} \leq 0.9*f_u/\gamma_{M2}$ 86,92 < 262,80 verified (0,33)

Connection stiffness

STIFFNESSES OF BOLT ROWS

Nr	hj	k_3	k_4	k_5	$k_{eff,j}$	$k_{eff,j}\ h_j$	$k_{eff,j}\ h_j^2$
					Sum	85,46	3310,81
1	526	∞	∞	88	6	30,10	1583,71
2	416	∞	∞	71	6	23,07	959,86
3	306	∞	∞	71	6	16,97	519,41
4	196	∞	∞	74	6	10,95	214,61
5	76	∞	∞	90	6	4,37	33,22

Weakest component:

BEAM WEB OR BRACKET FLANGE - COMPRESSION

$N_{j,Rd}$ = Min ($N_{tb,Rd}$, n_v n_h $F_{t,Rd}$, n_v n_h $B_{p,Rd}$)

$N_{j,Rd}$ = 1764.00 [kN] Connection resistance for tension [6.2]

$N_{b1,Ed}$ / $N_{j,Rd} \le 1,0$ 0,05 < 1,00 verified (0,05)

Remarks

The thickness of bracket web is less than the thickness of beam web 9 [mm] < 9 [mm]

Connection conforms to the code Ratio 0.67

II.6.3 Pinned column base design:

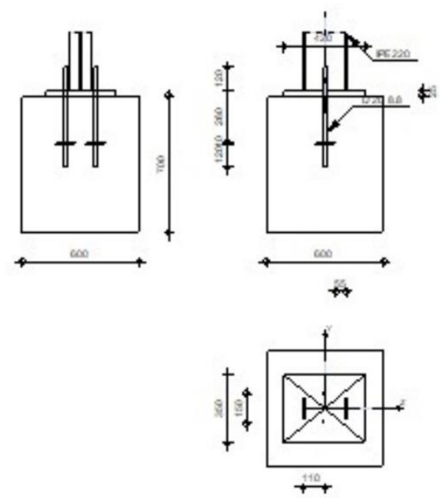
Autodesk Robot Structural Analysis Professional 2019

Pinned column base design

Eurocode 3: NF EN 1993-1-8:2005/NA:2007/AC:2009 + CEB Design Guide: Design of fastenings in concrete

OK

Ratio **0,12**



General

Connection no.: 3

Connection name: Pinned column base

Geometry

Column

Section: IPE 220

L_c =	5.00	[m]	Column length
α =	0.0	[Deg]	Inclination angle
h_c =	220	[mm]	Height of column section
b_{fc} =	110	[mm]	Width of column section
t_{wc} =	6	[mm]	Thickness of the web of column section
t_{fc} =	9	[mm]	Thickness of the flange of column section
r_c =	12	[mm]	Radius of column section fillet
A_c =	33.37	[cm^2]	Cross-sectional area of a column
I_{yc} =	2771.84	[cm^4]	Moment of inertia of the column section

Material: ACIER E28

f_{yc} =	275.00	[MPa]	Resistance
f_{uc} =	405.00	[MPa]	Yield strength of a material

Column base

l_{pd} =	420	[mm]	Length
b_{pd} =	350	[mm]	Width
t_{pd} =	25	[mm]	Thickness

Material:	ACIER E24	
f_{ypd} =	235.00 [MPa]	Resistance
f_{upd} =	365.00 [MPa]	Yield strength of a material

Anchorage

The shear plane passes through the UNTHREADED portion of the bolt.

Class =	8.8		Anchor class
f_{yb} =	550.00	[MPa]	Yield strength of the anchor material
f_{ub} =	800.00	[MPa]	Tensile strength of the anchor material
d =	20	[mm]	Bolt diameter
A_s =	2.45	[cm^2]	Effective section area of a bolt
A_v =	3.14	[cm^2]	Area of bolt section
n =	2		Number of bolt rows
e_V =	150	[mm]	Vertical spacing

Anchor dimensions

L_1 =	120	[mm]
L_2 =	260	[mm]
L_3 =	120	[mm]

Anchor plate

l_p =	100	[mm]	Length
b_p =	100	[mm]	Width
t_p =	10	[mm]	Thickness
Material:	ACIER E24		
f_y =	235.00	[MPa]	Resistance

Washer

l_{wd} =	0	[mm]	Length
b_{wd} =	0	[mm]	Width
t_{wd} =	1	[mm]	Thickness

Material factors

γ_{M0} =	1.00	Partial safety factor
γ_{M2} =	1.25	Partial safety factor
γ_C =	1.50	Partial safety factor

Spread footing

L =	600	[mm]	Spread footing length
B =	600	[mm]	Spread footing width
H =	700	[mm]	Spread footing height

Concrete

Class			BETON25
f_{ck} =	25.00	[MPa]	Characteristic resistance for compression

Grout layer

t_g =	0	[mm]	Thickness of leveling layer (grout)
$f_{ck,g}$ =	12.00	[MPa]	Characteristic resistance for compression
$C_{f,d}$ =	0.30		Coeff. of friction between the base plate and concrete

Welds

a_p =	7	[mm]	Footing plate of the column base

Loads

Case: Manual calculations.

$N_{j,Ed}$ =	-30.79	[kN]	Axial force
$V_{j,Ed,y}$ =	-9.92	[kN]	Shear force

$N_{j,Ed}$ =	-30.79	[kN]	Axial force
$V_{j,Ed,z}$ =	0.34	[kN]	Shear force

Resultats

Compression zone

COMPRESSION OF CONCRETE

Connection capacity check

$N_{j,Ed} / N_{j,Rd} \le 1,0$ (6.24)	0.02 < 1.00	verified	

Cisaillement

SHEAR CHECK

$V_{j,Rd,y} = n_b * \min(F_{1,vb,Rd,y}, F_{2,vb,Rd}, F_{v,Rd,cp}, F_{v,Rd,c,y}) + F_{f,Rd}$

$V_{j,Rd,y}$ =	32.23	[kN]	Connection resistance for shear	CEB [9.3.1]
$V_{j,Ed,y} / V_{j,Rd,y} \le 1,0$	0.31 < 1.00	verified		(0.31)

$V_{j,Rd,z} = n_b * \min(F_{1,vb,Rd,z}, F_{2,vb,Rd}, F_{v,Rd,cp}, F_{v,Rd,c,z}) + F_{f,Rd}$

$V_{j,Rd,z}$ =	41.81	[kN]	Connection resistance for shear	CEB [9.3.1]
$V_{j,Ed,z} / V_{j,Rd,z} \le 1,0$	0.01 < 1.00	verified		(0.01)
$V_{j,Ed,y} / V_{j,Rd,y} + V_{j,Ed,z} / V_{j,Rd,z} \le 1,0$	0.32 < 1.00	verified		(0.32)

Welds between the column and the base plate

$\sigma_\perp / (0.9 * f_u/\gamma_{M2}) \le 1.0$ (4.1)	0,01 < 1,00	vérifié	(0,01)
$\sqrt{(\sigma_\perp^2 + 3.0 (\tau_{yII}^2 + \tau_\perp^2))} / (f_u/(\beta w * \gamma_{M2})) \le 1.0$ (4.1)	0,02 < 1,00	vérifié	(0,02)
$\sqrt{(\sigma_\perp^2 + 3.0 (\tau_{zII}^2 + \tau_\perp^2))} / (f_u/(\beta w * \gamma_{M2})) \le 1.0$ (4.1)	0,02 < 1,00	vérifié	(0,02)

Weakest component:

FONDATION – Edge failure

Connection conforms to the code Ratio 0,12

II.6.4 Fixed column base design :

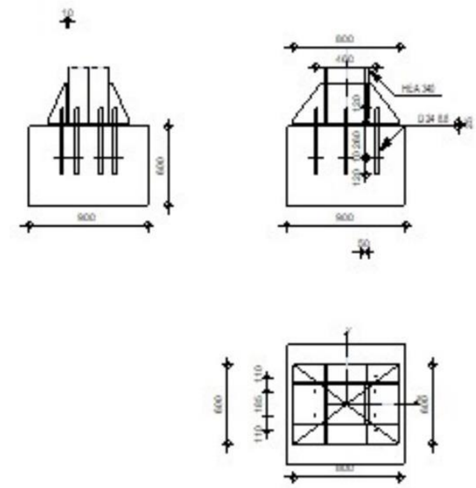
Autodesk Robot Structural Analysis Professional 2019

Fixed column base design

Eurocode 3: NF EN 1993-1-8:2005/NA:2007/AC:2009

OK

Ratio **0,97**



General

Connection no.: 4
Connection name: Fixed column base

Geometry

Column

Section: HEA 340

L_c =	9.00	[m]	Column length
α =	0.0	[Deg]	Inclination angle
h_c =	330	[mm]	Height of column section
b_{fc} =	300	[mm]	Width of column section
t_{wc} =	10	[mm]	Thickness of the web of column section
t_{fc} =	16	[mm]	Thickness of the flange of column section
r_c =	27	[mm]	Radius of column section fillet
A_c =	133.47	[cm^2]	Cross-sectional area of a column
I_{yc} =	27693.10	[cm^4]	Moment of inertia of the column section

Material: ACIER E28

f_{yc} =	275.00	[MPa]	Resistance
f_{uc} =	405.00	[MPa]	Yield strength of a material

Column base

l_{pd} =	800	[mm]	Length
b_{pd} =	600	[mm]	Width
t_{pd} =	25	[mm]	Thickness

Material: ACIER E24

f_{ypd} = 235.00 [MPa] Resistance

f_{upd} = 365.00 [MPa] Yield strength of a material

Anchorage

The shear plane passes through the UNTHREADED portion of the bolt.

Classe =	8.8		Anchor class
f_{yb} =	550,00	[MPa]	Yield strength of the anchor material
f_{ub} =	800,00	[MPa]	Tensile strength of the anchor material
d =	24	[mm]	Bolt diameter
A_s =	3.53	[cm^2]	Effective section area of a bolt
A_v =	4.52	[cm^2]	Area of bolt section
n_H =	3		Number of bolt columns
n_V =	4		Number of bolt rows

Horizontal spacing e_{Hi} = 230 [mm]

Vertical spacing e_{Vi} = 185;110 [mm]

Anchor dimensions

L_1 =	120	[mm]
L_2 =	260	[mm]
L_3 =	120	[mm]

Anchor plate

l_p =	100	[mm]
b_p =	100	[mm]
t_p =	10	[mm]

Washer

l_p =	100	[mm]
b_p =	100	[mm]
t_p =	10	[mm]

Material: ACIER E24

f_y = 235.00 [MPa] Resistance

Stiffener

l_s =	800	[mm]	Length
w_s =	600	[mm]	Width
h_s =	300	[mm]	Height
t_s =	10	[mm]	Thickness
d_1 =	20	[mm]	Cut
d_2 =	20	[mm]	Cut

Material factors

γ_{M0} =	1.00	Partial safety factor
γ_{M2} =	1.25	Partial safety factor
γ_C =	1.50	Partial safety factor

Spread footing

L =	900	[mm]	Spread footing length
B =	900	[mm]	Spread footing width
H =	600	[mm]	Spread footing height

Concrete

Class BETON25

f_{ck} = 25.00 [MPa] Characteristic resistance for compression

Grout layer

t_g =	0	[mm]	Thickness of leveling layer (grout)
$f_{ck,g}$ =	12.00	[MPa]	Characteristic resistance for compression
$C_{f,d}$ =	0.30		Coeff. of friction between the base plate and concrete

Welds

a_p =	7	[mm]	Footing plate of the column base
a_s =	4	[mm]	Stiffeners

Loads

Case: Manual calculations.

$N_{j,Ed}$ =	-163.32	[kN]	Axial force
$V_{j,Ed,y}$ =	-78.00	[kN]	Shear force
$V_{j,Ed,z}$ =	0.01	[kN]	Shear force
$M_{j,Ed,y}$ =	-280.83	[kN*m]	Bending moment
$M_{j,Ed,z}$ =	0.09	[kN*m]	Bending moment

Results

Connection capacity check

$N_{j,Ed} / N_{j,Rd} \leq 1,0$ (6.24)	0,02 < 1,00	verified	(0,02)
$M_{j,Ed,y} / M_{j,Rd,y} \leq 1,0$ (6.23)	0,94 < 1,00	verified	(0,94)
$M_{j,Ed,z} / M_{j,Rd,z} \leq 1,0$ (6.23)	0,02 < 1,00	verified	(0,02)
$M_{j,Ed,y} / M_{j,Rd,y} + M_{j,Ed,z} / M_{j,Rd,z} \leq 1,0$	0,97 < 1,00	verified	(0,97)

Shear

$V_{j,Ed,y} / V_{j,Rd,y} \leq 1,0$	0,09 < 1,00	verified	(0,09)
$V_{j,Ed,z} / V_{j,Rd,z} \leq 1,0$	0,00 < 1,00	verified	(0,00)
$V_{j,Ed,y} / V_{j,Rd,y} + V_{j,Ed,z} / V_{j,Rd,z} \leq 1,0$	0,09 < 1,00	verified	(0,09)

Stiffener check

Trapezoid plate parallel to the column web

max (σ_g, τ / (0.58), σ_z) / (f_{yp}/γ_{M0}) ≤ 1.0 (6.1)	0,44 < 1,00	verified	(0,44)

Stiffener perpendicular to the web (along the extension of the column flanges)

max (σ_g, τ / (0.58), σ_z) / (f_{yp}/γ_{M0}) ≤ 1.0 (6.1)	0,26 < 1,00	verified	(0,26)

Welds between the column and the base plate

$\sigma_\perp / (0.9 f_u/\gamma_{M2}) \leq 1.0$ (4.1)	0,14 < 1,00	verified	(0,14)
$\sqrt{\sigma_\perp^2 + 3.0 (\tau_{yII}^2 + \tau_\perp^2)} / (f_u/(\beta w * \gamma_{M2})) \leq 1.0$ (4.1)	0,21 < 1,00	verified	(0,21)
$\sqrt{\sigma_\perp^2 + 3.0 (\tau_{zII}^2 + \tau_\perp^2)} / (f_u/(\beta w * \gamma_{M2})) \leq 1.0$ (4.1)	0,18 < 1,00	verified	(0,18)

Vertical welds of stiffeners

Trapezoid plate parallel to the column web

max ($\sigma_\perp$, $\tau_{II} * \sqrt{3}$, σ_z) / ($f_u/(\beta w * \gamma_{M2})$) ≤ 1.0 (4.1)	0,64 < 1,00	verified	(0,64)

Stiffener perpendicular to the web (along the extension of the column flanges)

max ($\sigma_\perp$, $\tau_{II} * \sqrt{3}$, σ_z) / ($f_u/(\beta w * \gamma_{M2})$) ≤ 1.0 (4.1)	0,34 < 1,00	verified	(0,34)

Transversal welds of stiffeners

Trapezoid plate parallel to the column web

max ($\sigma_\perp$, $\tau_{II} * \sqrt{3}$, σ_z) / ($f_u/(\beta w * \gamma_{M2})$) ≤ 1.0 (4.1)	0,60 < 1,00	verified	(0,60)

Stiffener perpendicular to the web (along the extension of the column flanges)

max ($\sigma_\perp$, τ_{II} * $\sqrt{3}$, σ_z) / (f_u/(β_W*γ_{M2})) ≤ 1.0 (4.1)	0,49 < 1,00	verified	(0,49)

Connection stiffness

Bending moment $M_{j,Ed,y}$

$S_{j,ini,y}$ =	36172.45	[kN*m]	Initial rotational stiffness	[Table 6.12]
$S_{j,rig,y}$ =	193851.70	[kN*m]	Stiffness of a rigid connection	[5.2.2.5]
$S_{j,ini,y}$ < $S_{j,rig,y}$ SEMI-RIGID				[5.2.2.5.(2)]

Bending moment $M_{j,Ed,z}$

$S_{j,ini,z}$ =	522452.98	[kN*m]	Initial rotational stiffness	[6.3.1.(4)]
$S_{j,rig,z}$ =	52052.00	[kN*m]	Stiffness of a rigid connection	[5.2.2.5]
$S_{j,ini,z}$ ≥ $S_{j,rig,z}$ RIGID				[5.2.2.5.(2)]

Weakest Componant:

BASE PLATE - BENDING

BASE PLATE - BENDING	Ratio 0,97

II.7 Conclusion:

In this chapter, we have provided the general principles and design consideration for determining the loads acting on the studied structure (Dead load, live loads, snow load, wind load….ect).

The results found have been used for the dimensioning of the elements of the structure (purlin, column, …), And after the necessary checks of all the elements using SAP2000 we were able to determine all the final sections for all the members that will withstand the acting forces.

The design of all the connections also was done and checked.

III. Steady state analysis

III.1 Introduction:

Heavy machinery with reciprocating, impacting, or rotating masses requires a support system that can resist dynamic forces and the resulting vibrations. When excessive, such vibrations may be detrimental to the machinery, its support system, and operating personal subjected to them.

Therefore, it's very important to study and analyze this type of vibrations and check the strength of the system under this kind of excitation, that's why in this chapter we will go through some verifications and see how our structure will behave.

To do so, the steady state analysis is the fitting analysis that will help us to achieve that, fortunately the SAP2000 will make it easy for us to analyze these steady state vibrations

In the next sections we will start by presenting all the codes that are related to our analysis and what is the use of each one, then we are going to see how the vibrations of the blower will affect on the structure and what are the characteristics of our blower, next will jump into the finite element model using the SAP2000 and we will try many models with changing different factors (like the material of the slab or the location of the blower…etc.) and see how this will affect on the results

At the end we will interpret all these results and make a conclusion.

III.2 Codes and References:

Codes suggest guidelines and limitations in the design of a structure, to design our support system that will carry out the dynamic responses will work with some industrial codes.
In this section we will present those codes and references:

III.2.1 ACI 351.3R:

Full name: ACI 351.3R Report on Foundations for Dynamic Equipment.

Number of pages: 60 pages in v04 and 80 pages in v18

Number of chapters: 6 chapters in v04 and 10 chapters in v18.

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

Why we chose this code?

This code gives us all the dynamic loads that are occur during the operation of the machine and allow us to calculate them and check them.

III.2.1.1 Contents:

III.2.1.1.1 Types of Vibrating Machines: (chapter 2 in the code)

Based on type of motion, the machines are broadly classified as:

- Rotating Machines.
- Reciprocating Machines.
- Impact Type Machines.

a) Rotating Machines (centrifugal machines):

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

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

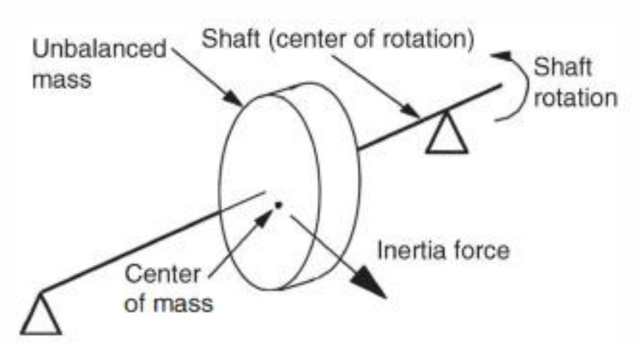


Figure III-1 Rotating machine diagram.

b) Reciprocating Machines:

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

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

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

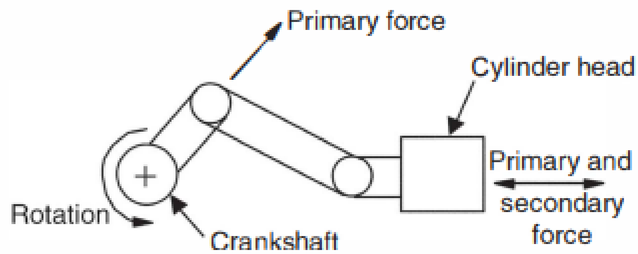


Figure III-0-2 Reciprocating machine diagram.

c) Impact Type Machines:

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

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

Forging presses perform a similar manufacturing function as forging hammers but are commonly mechanically or hydraulically driven. These presses form the material low velocities but with greater forces. The mechanical driven system generates horizontal dynamic forces that the engineer should consider in the design of the support system. Rocking stability of this construction is important.

Mechanical metal forming presses operate by squeezing and shearing metal between two dies. Because this equipment can vary greatly in size, weight, speed, and operation, forces and design criteria used for the foundation design can vary greatly. Speeds can vary from 30 to 1 800 strokes per minute. Dynamic forces from the press develop from two sources: the mechanical imbalance of the moving parts in the equipment and the response of the press frame as the material is sheared (snap-through forces). Imbalances in the mechanics of the equipment can occur both horizontally and vertically. Generally, high-speed equipment is well balanced. Low-speed equipment is often not balanced because the inertia forces at low speeds are small. The dynamic forces generated by all of these presses can be significant as they are transmitted into the foundation and propagate into the subgrade.

As we have seen in this section there is 3 types of vibrating machines, but in this case study our **Blower Fan** is classify as a rotating machine that’s why in the next sections we will focus just on the rotor machines and their effect.

III.2.1.1.2 **Design loads (chapter 4 in the code):**

The loads on the machines may be both static and dynamic.

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

- **Unbalanced load:**

as we already seen in the definition of the rotating machines unbalanced forces in rotating machines are created when the mass centroid of the rotating component does not coincide with the center of rotation. This dynamic force is a function of the mass of the rotating component, speed of rotation, and the magnitude of the eccentricity of offset.

When the mass unbalance (eccentricity) is known or stated by the manufacturer, the resulting dynamic force amplitude is:

$$F_0 = m_r e_m \omega_0{}^2 S_f / 1000 \text{ N} \qquad \ldots\ldots\ldots \quad [\text{ ACI351\_art 3.2.2.1a }]$$

Where:

F_0 : Dynamic force amplitude (zero-to-peak), (N);

m_r : Rotating mass, (kg);

e_m : Mass eccentricity, (mm)

ω_0 : Circular operating frequency of the machine, (rad/s)

S_f : Service factor, used to account for increased unbalance during the service life of the machine, generally greater than or equal to 2.

Many rotating machines are balanced to an initial balance quality Q either in accordance with the manufacturer's procedures or as specified by the purchaser. ISO 1940-1 (which we are going to talk about in next section) define balance quality in terms of a constant $e_m\omega_0$. Typical balance quality grade examples are shown in Table III-0-1.

To meet these criteria, a rotor intended for faster speeds should be better balanced than one operating at a slower speed. The dynamic force amplitude can be rewritten as:

$$F_0 = m_r Q S_f / 1000 \quad \text{N}$$

Table III-0-1 Balance quality grades for selected groups of representative rigid rotors

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

For resonance check the code give some hints about the interval of safety the engineer must take in consideration to avoid resonance, for the ACI 351 R3-2018, and according to article 6.5.1, it is recommended to take 20% to 30% of safety margin, which means that the engineer has to keep the eigenvalues (natural frequencies) far from the forcing frequencies by a factor of 0.8 to 1.2 more or less.

III.2.2 ISO 1940:

Full name: The International Standards Organization, ISO, published Standard 1940/1 "Balance Quality Requirements of Rigid Rotors». Which has been adopted by the American National Standards Institute, ANSI, as S2.19-1975, "Balance Quality Requirements of Rotating Rigid Bodies." It has also been adopted by BRITISH Standards as BS 6861: Part 1 and by GERMAN Standards as VDI 2060.

Description: International Standard ISO 1940/1 is a widely accepted reference for selecting rigid rotor balance quality. This paper is presented as a tutorial and user's reference of the standard and its practical applications.

A simplified method is shown for determining permissible residual unbalance for various rotor classifications. Emphasis is given to allocating permissible residual unbalance to appropriate correction planes for rotor configurations.

The importance of the code is to help engineers to check the amplitudes determined by the steady state analysis of any equipment, and compare it to the required limits of the code regarding to the needed balance quality.

Use the **Figure III-0-3** graph to determine the permissible residual specific unbalance value e_{per}

(The maximum displacement limit) for the rotor's maximum operating speed and the selected Q (the balance quality guide)

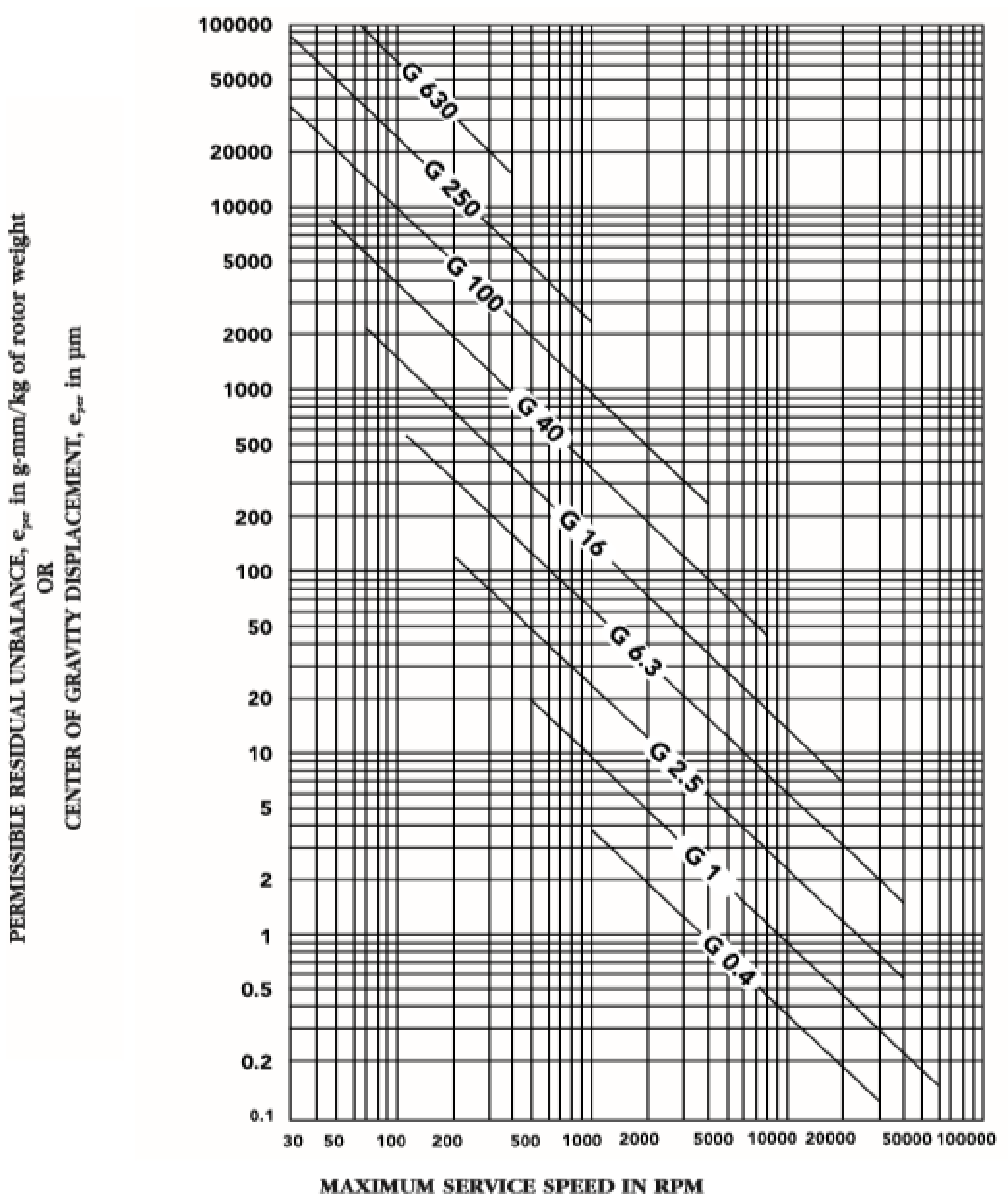


Figure III-0-3 Maximum permissible residual unbalance, e_{per} (From ISO 1940/1)

III.2.3 Other codes:

Besides the ACI 351.3R and ISO 1940.1 there are many other codes that can be used as well for the steady state vibrations analysis.

Even though we are not going to work with those codes in our case study, but we would also present them briefly for their importance in this domain.

III.2.3.1 DIN 4024.1:

Full name: DIN 4024 Part 1 Machine foundations; flexible structures that support machines with rotating elements.

Description: This standard specifies requirements for steel or reinforced concrete structures that support mechanical system. Such mechanical systems are understood to be machinery with mainly rotating elements, the foundations of which are capable of generating flexural vibration in at least one plane.

The requirements specified in this code are intended to prevent the static and dynamic loads from transmitting unacceptable vibration to the environment or causing damage to the machinery and its foundation. This standard establishes criteria for determining vibration behavior, deals with design action-effects, and covers principles of construction based on experience to date with machine foundations.

.

Contents

Figure III-0-4 The table of content shows most treated topics in this German code

III.2.3.2 IS 2974.3.1992(reaffirmed 2006):

Full name: Indian Standard; Design and Construction of Machine Foundations - Code of Practice Part 3 Foundations for Rotary Type Machines (Medium and High Frequency).

Description: This code is primarily meant for designing framed type foundations for turbo-generators machinery. However, the provisions of this code may be used suitably for other machine foundations of similar types, for example, foundations of turbo-compressors, boiler feed pumps, etc.

III.3 Machine Parameters for our case:

After describing the meaning of the steady state analysis and giving some codes that govern this kind of vibrating behavior, and also speaking about most renown types of vibrating machines, we have chosen the type of rotating machines for our study, and most precisely a blower fan, next the equipment is well presented in order to use its vibrating parameters in our analysis

III.3.1 Presentation of the Blower fan:

Also called: **centrifugal fans**, the type of this Blower fan is **rotating machine**

A blower fan or centrifugal fan is a type of fan commonly used to power ventilation systems in environments that would otherwise have a low standing air quality either due to limited oxygen or the presence of harmful gases or particulates. These fans are sometimes called “squirrel cage fans” because they are composed of a series of overlapping blades grouped in a manner that make the fans look somewhat like hamster wheels.

Centrifugal fans work by drawing air toward the center of the fan and then discharging it at a predictable 90-degree angle from the direction of air intake. They have a relatively low cost to operate, can effectively move air within a variety of airflow conditions, and operate quietly. Because of these properties, they are commonly used to move air throughout ventilation systems, or they are oriented on top of chimneys for the purpose of driving hot, stale, or dirty air out of a workspace and ejecting it into the surrounding environment.

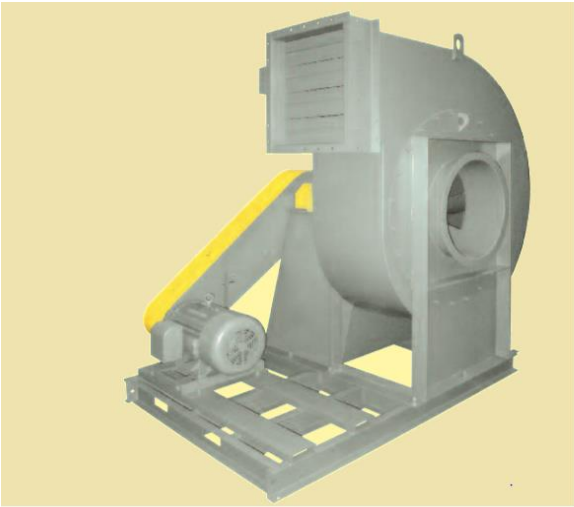


Figure III-0-5 Blower fan from the new-york company technical file

III.3.2 Characteristics of the blower:

From the technical file of our Blower fan there is many size for the machine, to study the behavior of the structure we will analyze for two different sizes of the machine, for the size 144 and for Size 364.

Table III-0-2 The circural forcing frequncy for different size of machines

Size	Wheel Max Safe Speed		AH wheel
	Max Speed (RPM)	Service Speed (RPM)	Weight (ibs)
144	4605	4605	25
174	3930	3745	33
194	3425	3115	55
224	2900	2635	72
264	2510	2280	91
294	2195	1995	123
334	2035	1790	189
364	1840	1620	229

III.3.3 Loads calculation:

As we already mentioned in design loads of the ACI 351.3R there are two types of loads:

III.3.3.1 Static loads:

The static load of our machine represents the weight of the Blower Fan and all its auxiliary equipment.

From the technical file of the Blower the weight of the hole machine is:

Table III-0-3 The static loads of the machine for differenet sizes

Size	Fan Weights(ibs)	The static load of the machine(Kn)
144	130	0,6
364	1170	5,2

III.3.3.2 Dynamic loads:

During the operation of the Blower Fan (which is consider as a rotating machine) an unbalanced load is produced.

- **The unbalanced load:**

$$F_0 = \frac{m_r Q S_f}{1000} \quad N$$

The calculation of the unbalanced load will be for the Blower Fan Type AH size 144 and for the type AH size 364, and each size of them will have two values: one for the maximum frequency and the other one for the service frequency (for the life of the machine).

a. <u>Fan type AH size 144:</u>

- For the max value of ω_0

$\boldsymbol{m_r} = 25\ ibs = 111{,}206$ N

$\boldsymbol{Q} = 6{,}3\ \frac{mm}{s} = 6{,}3 \times 10^{-3}\ \frac{m}{s}$

$\boldsymbol{\omega_0} = 4605\ RPM$ = 482,234 rad/s

$$F_0 = \frac{111{,}206 \times 6{,}3 \times 10^{-3} \times 482{,}234 \times 2}{1000}$$

$\boldsymbol{F_0 = 0{,}676\ KN}$

- For the service value of ω_0

$\boldsymbol{\omega_0} = 3980\ RPM$ = 416,785 rad/s

$\boldsymbol{F_0 = 0{,}584\ KN}$

b. <u>Fan type AH size 364:</u>

- For max value of ω_0

$\boldsymbol{m_r} = 229\ ibs = 1018{,}643\ N$

$\boldsymbol{\omega_0} = 1840\ RPM = 192{,}684\ rad/s$

$$F_0 = \frac{1018{,}643 \times 6{,}3 \times 192{,}684 \times 2}{1000}$$

$\boldsymbol{F_0 = 2{,}47\ KN}$

- For service value of ω_0

$\boldsymbol{\omega_0} = 1620\ RPM = 169{,}645\ rad/s$

$$F_0 = \frac{1018{,}643 \times 6{,}3 \times 169{,}645 \times 2}{1000}$$

$\boldsymbol{F_0 = 2{,}18\ KN}$

Table III-0-4 The unbalance load for both types of machine

Size	The Unbalanced load (Kn)	
	for the max	for the service
144	0,676	0,584
364	2,47	2,18

III.4 The finite Element model (using SAP2000):

SAP2000 provides an option that allows us to easily assign steady-state harmonic forcing function over a range of frequency to elements or structures, this allows vibration analysis to be performed for structures subjected loads resulting from oscillating equipment.

Our machine is placed on the top of the mezzanine (the choice of the exact placement of the machine on the mezzanine will be discussed in detail in the next sections).

Modeling the equipment on the mezzanine will lead to some dynamic misinterpretation, especially when it comes to mass participation ratios, the more the participation of the modes of the vibrating equipment are considerable, the more important these modes are, which means, we need to check with the participating modes of the equipment in the global behavior of the structure, if the mass participation ratio of these modes is negligible (less than 5%), then, we can discretize the structure supporting the equipment and check its governing modes with their real mass participation ratio and compare them to the forcing frequencies, so next we'll analyze two kind of models, global model and discretized model.

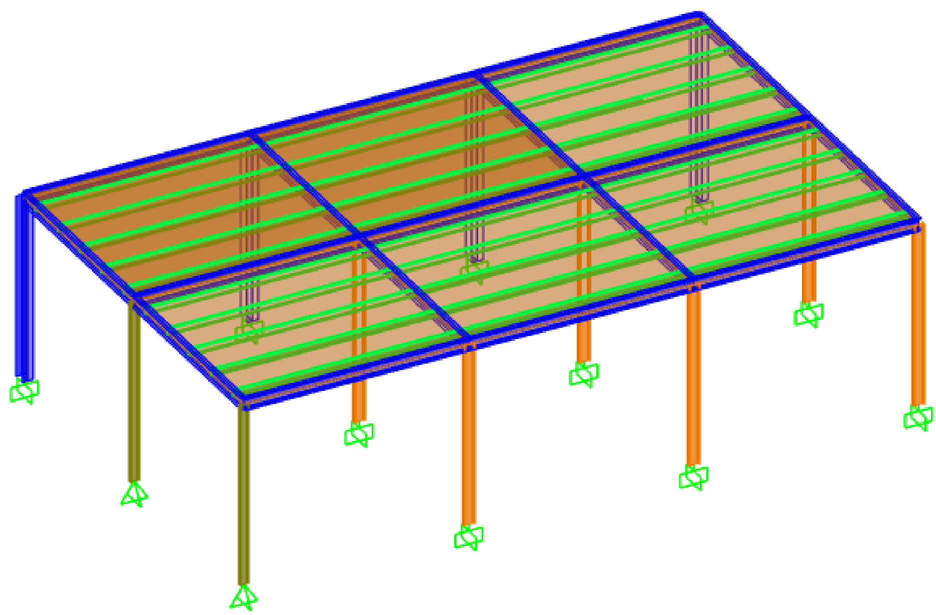


Figure III-0-6 3D view of the mezzanine

III.4.1 Global Model:

After we input all the data of the steady state analysis in the software (steady state functions ⟶ load patterns ⟶ load cases ⟶ Assign joint forces) and then run the analysis, we could finally extract the results of our analysis.

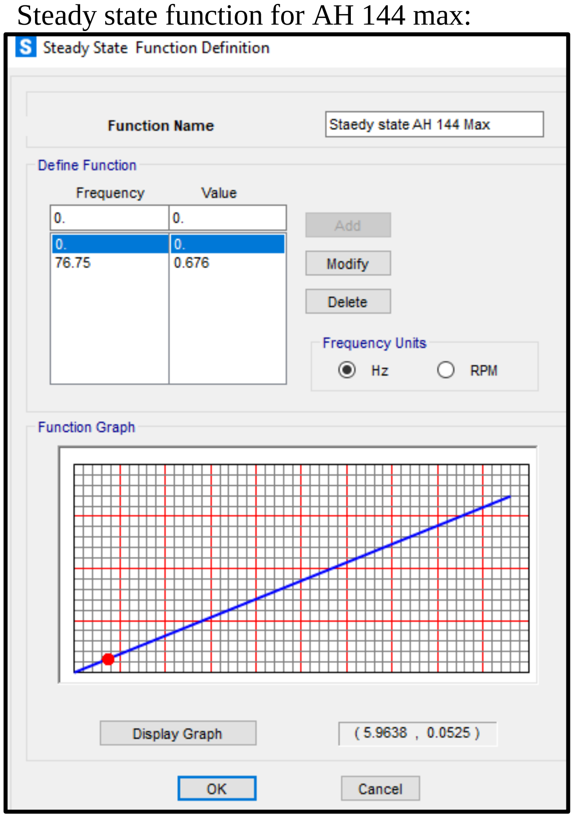
But first, the mass participating ratio of the governing modes must be analyzed, and then we can reach for the check of the resonance

To do so, we need to check if there's a considerable participation mass ratio for the equipment supporting structure, then we need to check the interval safety (this interval is presenting frequencies which will may generate the risk of resonance as it's specified in the ACI 351.3R (Art.6.5.1) ; "±20% of the operating frequency at minimum."

Type of the machine	Operating ferquency (Hz)	Operating ferquency (Hz) * 0,8	Operating ferquency (Hz) * 1,2
AH 144 (Max)	76.75		92.10
AH 144 (Service)	66.33	53.07	
AH 364 (Max)	30.67		36.80
AH 364 (service)	27.00	21.60	

- For the machine type AH 144: **53.07 < risk of resonance < 92.10**
- For the machine type AH 364: **21.60 < risk of resonance < 36.80**

Steady state function for AH 144 max:



Steady state function for AH 144 service:

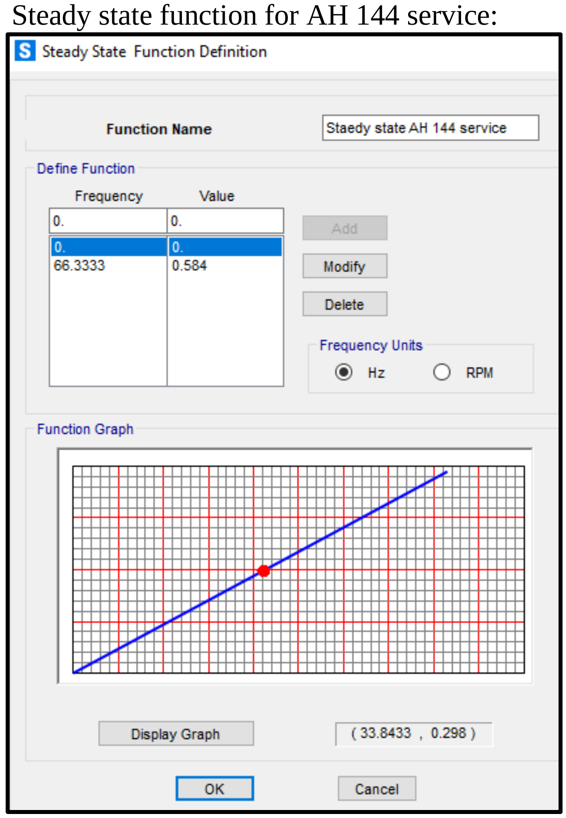


Figure III-7 Steady state function for AH 144 max service

Figure III-8 Steady state function for AH 144

Steady state function for AH 364 max:

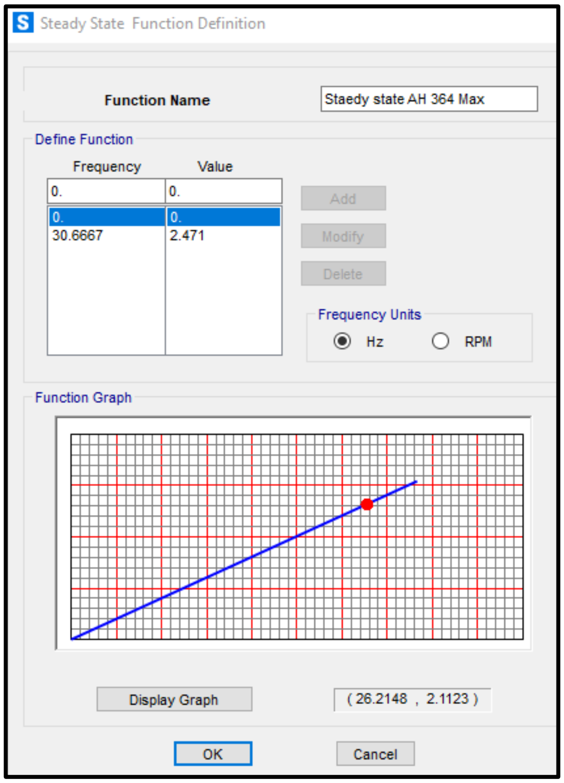


Figure III-9 Steady state function for AH 364 max

Steady state function for AH 364 service:

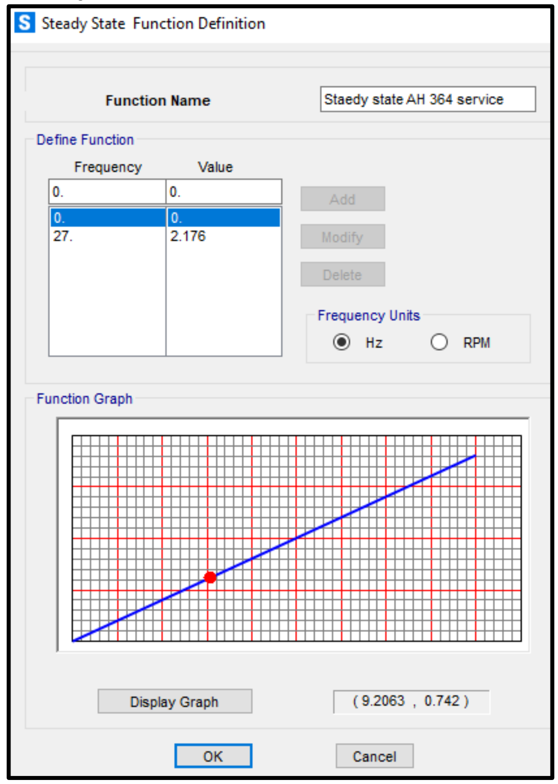


Figure III-10 Steady state function for AH 364 service

III.4.1.1 <u>Mass participating ratios in the global model:</u>

- **For the machine type AH 144:**

Table III-0-5 Mass participating ratios in the global model for the machine type AH 144

number of modes	Frequency Cyc/sec	Participating Mass Ratios UZ	number of modes	Frequency Cyc/sec	Participating Mass Ratios UZ	number of modes	Frequency Cyc/sec	Participating Mass Ratios UZ
292	53.468	0.00139	347	65.902	0.0004	402	79.158	0.00003753
293	53.545	0.00114	348	66.548	0.00013	403	79.318	0.00005938
294	53.697	0.00691	349	66.822	0.00001919	404	79.355	0.000004874
295	53.853	0.00018	350	66.935	0.00001052	405	79.397	0.00001775
296	54.136	0.00014	351	67.015	0.00016	406	79.717	0.00265
297	54.424	0.00015	352	67.215	0.00012	407	79.850	0.00009036
298	54.854	0.00079	353	67.351	0.00031	408	80.069	0.00005427
299	55.075	0.00887	354	67.668	0.00026	409	80.929	0.000001677
300	55.106	0.0007	355	67.749	0.00002769	410	81.310	0.0041
301	55.326	0.00088	356	67.933	0.00056	411	81.816	0.00072
302	55.922	0.00334	357	68.575	0.00015	412	81.998	0.00128
303	56.386	0.0000922	358	68.823	0.00452	413	82.460	0.00005072

304	56.496	0.00012	359	69.021	0.00006864	414	82.604	0.00089
305	56.531	0.00075	360	69.284	0.00532	415	83.030	0.000007764
306	56.592	0.00004265	361	69.516	0.00055	416	83.516	0.00038
307	56.661	0.000003664	362	69.567	0.00718	417	83.786	0.0148
308	56.749	0.00004668	363	69.567	0.00167	418	84.073	0.00077
309	56.906	0.00367	364	69.890	0.00675	419	84.943	0.00056
310	57.053	0.00063	365	70.294	0.00001258	420	85.093	0.0003
311	57.216	0.00579	366	70.497	3.91E-08	421	85.280	0.00033
312	57.288	0.00005832	367	71.031	0.0001	422	85.712	0.00213
313	57.420	0.00022	368	71.387	0.00009624	423	85.732	0.000005051
314	57.874	0.00013	369	71.629	0.00007649	424	86.114	0.0096
315	58.045	0.00031	370	72.508	0.00025	425	86.213	0.0483
316	58.384	0.00059	371	72.849	0.00016	426	86.222	0.00001595
317	58.428	0.0002	372	73.102	0.00353	427	86.237	0.00757
318	58.502	0.00011	373	73.111	0.00004896	428	86.252	0.00001491
319	58.555	0.00004478	374	73.111	0.00702	429	86.520	0.00019
320	59.008	0.00019	375	73.614	0.00006824	430	86.618	0.0005
321	59.126	0.00003068	376	73.887	6.283E-10	431	86.944	0.00083
322	59.307	0.00016	377	74.413	0.00003501	432	87.836	0.00019
323	59.387	0.0002	378	74.788	0.0000455	433	88.010	0.00007552
324	59.695	0.000007782	379	75.621	0.00006221	434	88.216	0.00015
325	59.745	0.00002488	380	75.753	0.00001194	435	88.343	0.00016
326	59.880	0.00003253	381	75.927	0.0013	436	88.482	0.00002723
327	60.230	0.00009075	382	76.069	0.00014	437	89.111	4.378E-07
328	60.338	0.00018	383	76.156	0.00025	438	89.118	1.066E-07
329	60.450	0.00002061	384	76.243	0.000004666	439	89.161	0.0000135
330	60.914	0.00002471	385	76.360	0.00106	440	89.179	0.00015
331	61.005	0.000007957	386	76.448	0.00073	441	89.241	0.00024
332	61.070	0.00264	387	76.501	0.0000092	442	89.408	0.005
333	61.586	0.00018	388	76.561	0.000006774	443	89.710	0.0023
334	61.957	0.00003711	389	76.609	0.00007958	444	89.941	0.00019
335	62.082	0.00168	390	76.641	0.00163	445	90.159	0.00004381
336	62.427	0.00014	391	77.070	0.00037	446	90.582	0.00032
337	63.270	7.417E-07	392	77.300	0.00004897	447	90.751	0.00037
338	63.571	0.00007508	393	77.307	0.00002357	448	90.986	0.00005874
339	63.642	0.000002434	394	77.315	0.000006554	449	91.397	0.00008881
340	63.709	0.00001002	395	77.327	0.000008284	450	91.565	0.00042
341	63.956	0.00001824	396	77.592	0.00202	451	91.646	0.00004435
342	64.259	0.00053	397	78.126	0.00069	452	91.764	0.00006612
343	64.538	0.00005406	398	78.268	0.00008101	453	91.818	0.00007496
344	64.807	0.00039	399	78.729	0.00018	454	91.921	0.00024
345	64.887	0.00018	400	79.003	0.00029	455	92.029	0.000003449
346	65.465	0.00002114	401	79.136	0.00059			

- **For the machine type AH 364:**

Table III-0-6Mass participating ratios in the global model for the machine type AH 364

Number of modes	Frequency Cyc/sec	Participating Mass Ratios UZ	Number of modes	Frequency Cyc/sec	Participating Mass Ratios UZ	Number of modes	Frequency Cyc/sec	Participating Mass Ratios UZ
143	21.816	0.04302	171	28.805	0.00006868	199	34.080	5.14E-08
144	22.074	0.0003	172	28.925	0.00998	200	34.247	7.54E-10
145	22.746	0.000007549	173	29.239	0.00034	201	34.588	0.000002312
146	22.848	0.00023	174	29.955	1.05E-07	202	34.743	0.00002439
147	23.118	0.00014	175	30.302	0.00001667	203	34.798	0.00029
148	23.321	0.000001039	176	30.706	4.38E-07	204	34.869	0.0000155
149	23.696	0.00001485	177	30.881	0.00006041	205	34.987	0.00000181
150	23.851	0.00168	178	30.961	0.000003833	206	35.040	0.000001878
151	23.984	0.00004	179	30.978	4.06E-08	207	35.087	0.00021
152	24.275	0.00003393	180	30.996	2.35E-07	208	35.194	0.00131
153	24.304	0.00004715	181	31.038	0.000002778	209	35.256	0.00612
154	24.403	0.00001929	182	31.094	0.000002171	210	35.304	0.00353
155	24.757	2.85E-07	183	31.472	0.00077	211	35.371	0.00252
156	24.891	0.000004839	184	31.619	0.00028	212	35.411	0.00264
157	24.923	0.000005087	185	31.819	0.00012	213	35.494	0.00185
158	25.235	0.000005604	186	31.998	0.01265	214	35.589	0.00083
159	25.445	0.00002592	187	32.101	0.00189	215	35.602	0.00073
160	25.562	0.000009278	188	32.465	0.00005821	216	35.778	0.00323
161	25.736	0.0013	189	32.520	0.00025	217	35.902	0.00043
162	25.759	0.0451	190	32.602	0.000001171	218	35.923	0.0004
163	25.856	8.24E-07	191	33.016	0.00000112	219	35.971	0.0017
164	26.614	0.000000047	192	33.133	0.00001779	220	36.100	0.00013
165	26.811	0.00003036	193	33.208	0.000001385	221	36.137	0.00004197
166	27.207	0.01186	194	33.282	0.00002439	222	36.324	0.000001611
167	27.419	0.00714	195	33.529	0.00002411	223	36.393	0.00002419
168	27.554	0.04561	196	33.598	7.72E-07	224	36.437	0.00016
169	28.279	0.00268	197	33.642	0.00038	225	36.655	0.00001945
170	28.368	0.00003509	198	33.958	0.000001502			

- **Results interpretation (for the two types of machines):**

As we can notice from our results, the mass participation ratio for the global modes governing the vibrating equipment behavior, are very small (less than 5%) for the ranging modes in the interval of the resonance.

This means that our analysis problem it's not consider as global problem, in another word we are dealing with a local problem and a discretized model will be better for describing the real behavior of the vibrating equipment (the blower).

III.4.2 <u>Discretized Model:</u>

After we realized that the global model it's not the fitting model for our analysis, we will move on to a discretized model (local model).

A discretized model represents a model of the support elements that will carry on our machine equipment, in our case (and before studying the exact placement of the machine) we will suppose that our machine equipment will be on the midspan of a joist.

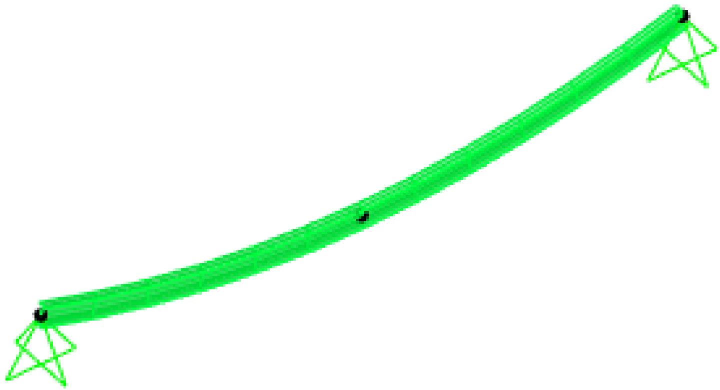


Figure III-0-7 3D view of the descritzed model of a joist

The interval of safety will be the same as specified in the global model:

- For the machine type AH 144: **53.07 < risk of resonance < 92.10**
- For the machine type AH 364: **21.60 < risk of resonance < 36.80**

III.4.2.1 Mass participating ratios in the discretized model:

- **For the machine type AH 144:**

Table III -0-7 Mass participating ratios for AH 144

Number of modes	Frequency Cyc/sec	Participating Mass Ratios X	Participating Mass Ratios Y	Participating Mass Ratios Z
1	2.864	0	1	0
2	10.286	0	0	100%
3	241.435	1	0	0

- **For the machine type AH 364:**

Table III-0-8 Mass participating ratios for AH 364

Number of modes	Frequency Cyc/sec	Participating Mass Ratios X	Participating Mass Ratios Y	Participating Mass Ratios Z
1	1.393	0	1	0
2	5.004	0	0	100%
3	117.445	1	0	0

- **Results interpretation:**

In this discretized model we have just 3 modes, and from our results we can notice that there's a large participating mass in the second mode, but fortunately the frequency of this mass is out of the range of resonance's risk, that means that there's no risk of resonance in our case but this does not mean that the amplitudes resulting from the machine vibration are acceptable too.

Note:

These results and conclusions are just for the given location of the machine equipment, but in the next section after choosing the perfect location of the machine equipment we will re-check the resonance risk as well as the acceptable amplitudes.

III.4.3 <u>Best Location for the Blower:</u>

According to the ISO-1940, the calculation of the amplitudes, velocities and accelerations must be check with the allowable values that we can determine from the figure ..(for the amplitude) and from the table…(for the velocity).

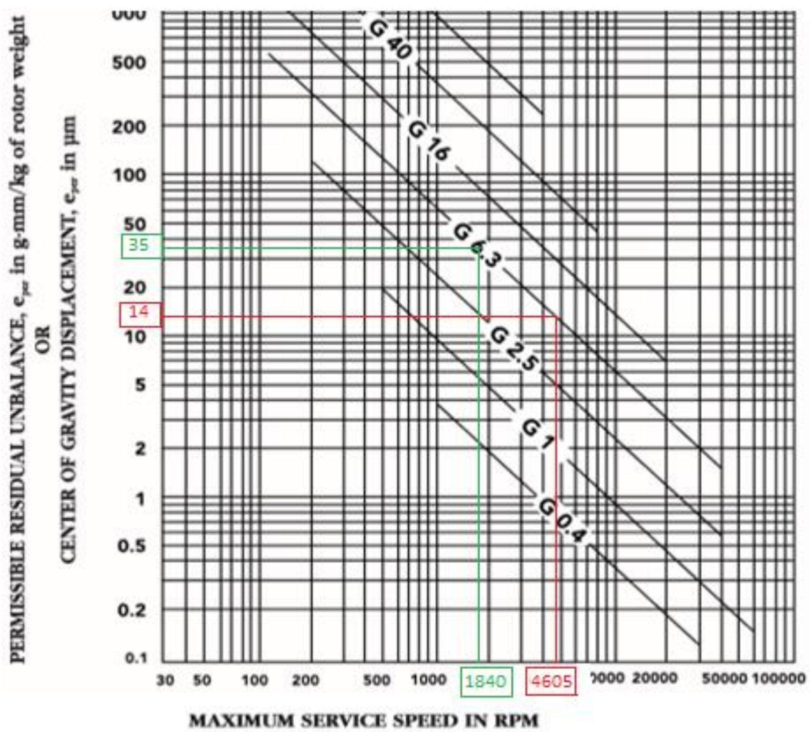


Figure III-0-3 Maximum permissible residual unbalance, e_{per} (From ISO 1940/1)

Using the graph in figure **Figure III-0-3** and the table **Table III-**0-1 allowed us to extract the following limitations:

Table III-0-9 The permissible amplitude and velocity

	Circular operating frequency Of the machine in RPM	The permissible amplitude in µm	The permissible velocity in mm/s
For AH 144	4605	14	0.25
For AH 364	1840	35	0.25

the steady state analysis it's very powerful analytic tool to check the calculated amplitude and the velocities of the machinery and compare it with those given by the code.

This amplitude and velocity are related directly to stiffness and the mass of the support system, to change and increase the stiffness we have two methods:

1) Increase the section of the supporting element.
2) Chose the right and the prefect placement of machine equipment.

the 2nd method is both more economical and practical, therefore in this section we will try to put our blower on different locations and see how the amplitude and velocity will change, and finally choosing the safest and the most economical placement.

thus, we will place our blower on a number of joints as follow:

1. On the joint 9 (in the midspan of closest joist to the wall)
2. On the joint 7 (in the midspan of the middle joist of the panel)
3. On the joint 421 (the first beam's joint closes the wall)
4. On the joint 419 (the 2nd beam's joint closes the wall)
5. On the joint 417 (the 3rd beam's joint closes the wall)
6. On the joint 415 (the 4th beam's joint closes the wall)
7. On the joint 321 (above the central column)

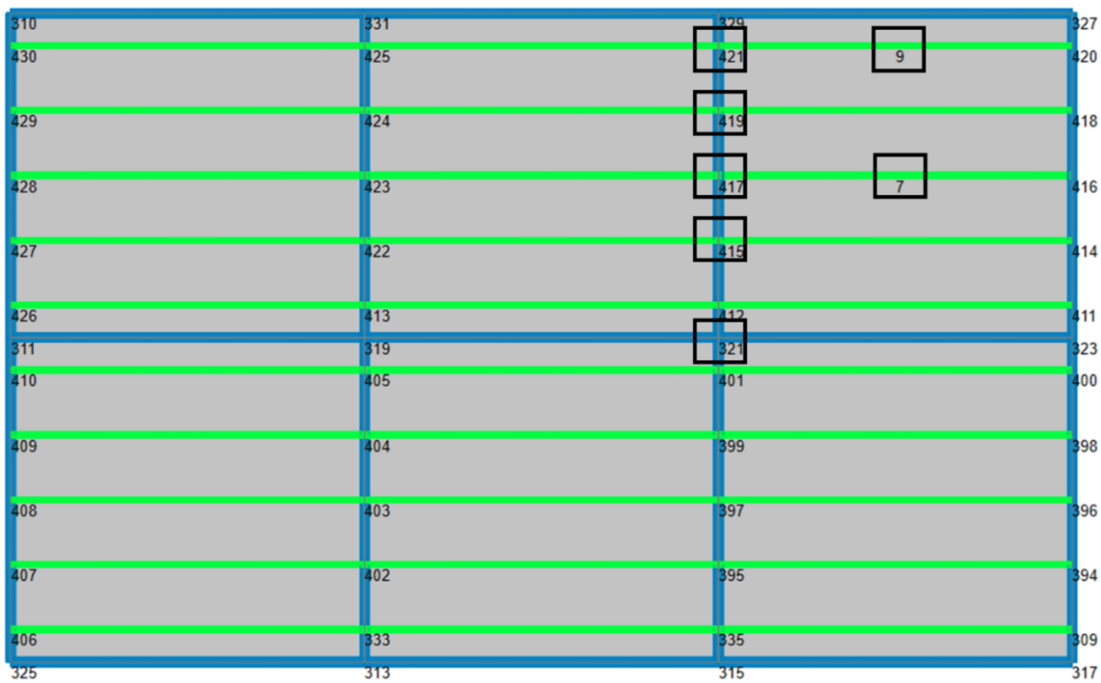


Figure III-0-8 2D of the mezzanine shows the selected joints

Note:

All the comparative results and graphs are for the machine type AH 144.

For the machine type 364 we will suffice with a table contains all the results.

III.4.3.1 On the Joint 9 (in the midspan of closest joist to the wall):

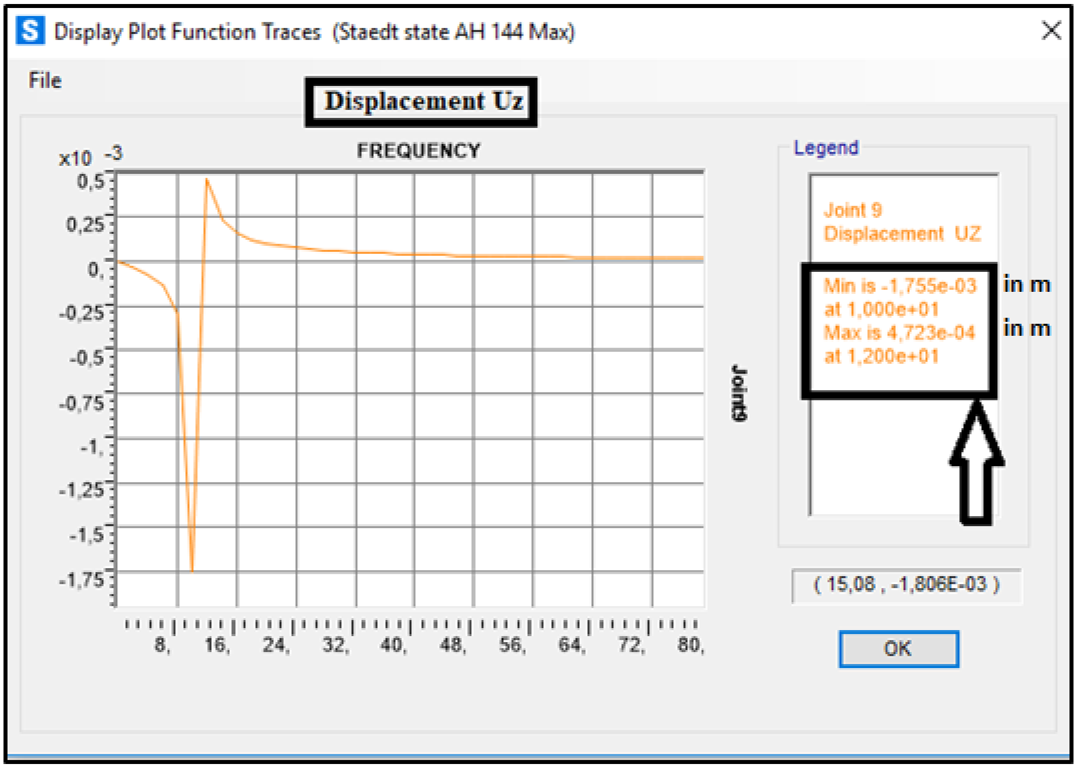


Figure III-9 The displacement diagram on the joint 9

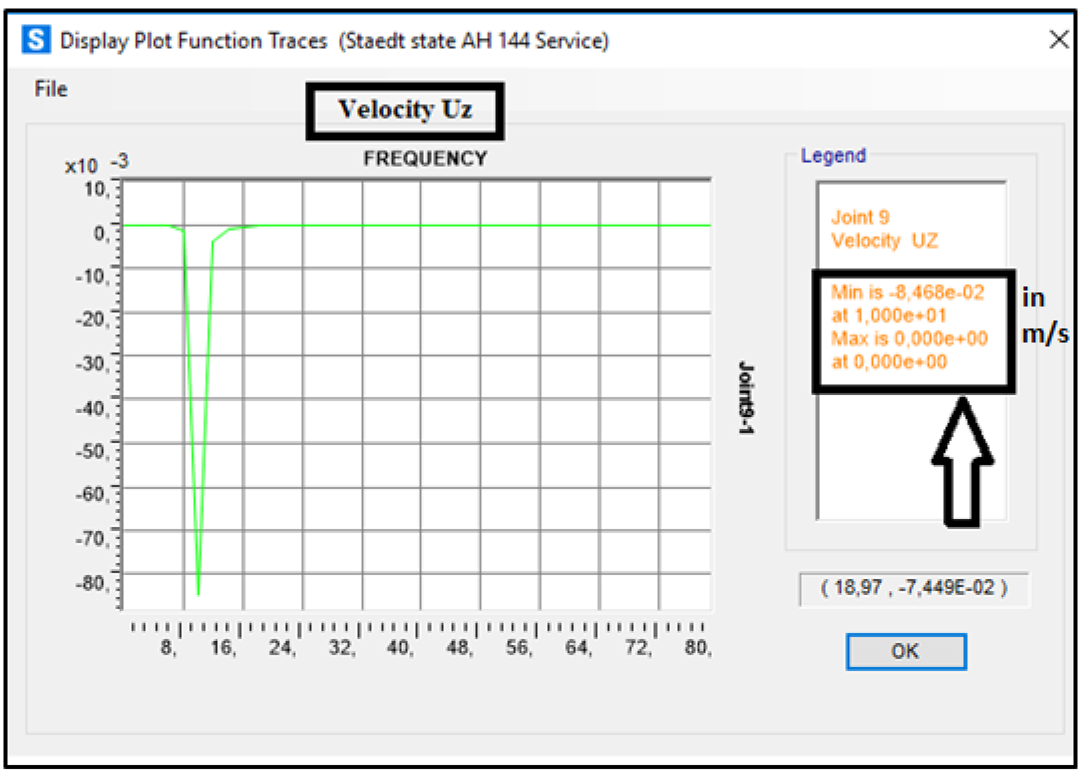


Figure III-10 The velocity diagram on the joint 9

III.4.3.2 On the Joint 7 (in the midspan of the middle joist of the panel):

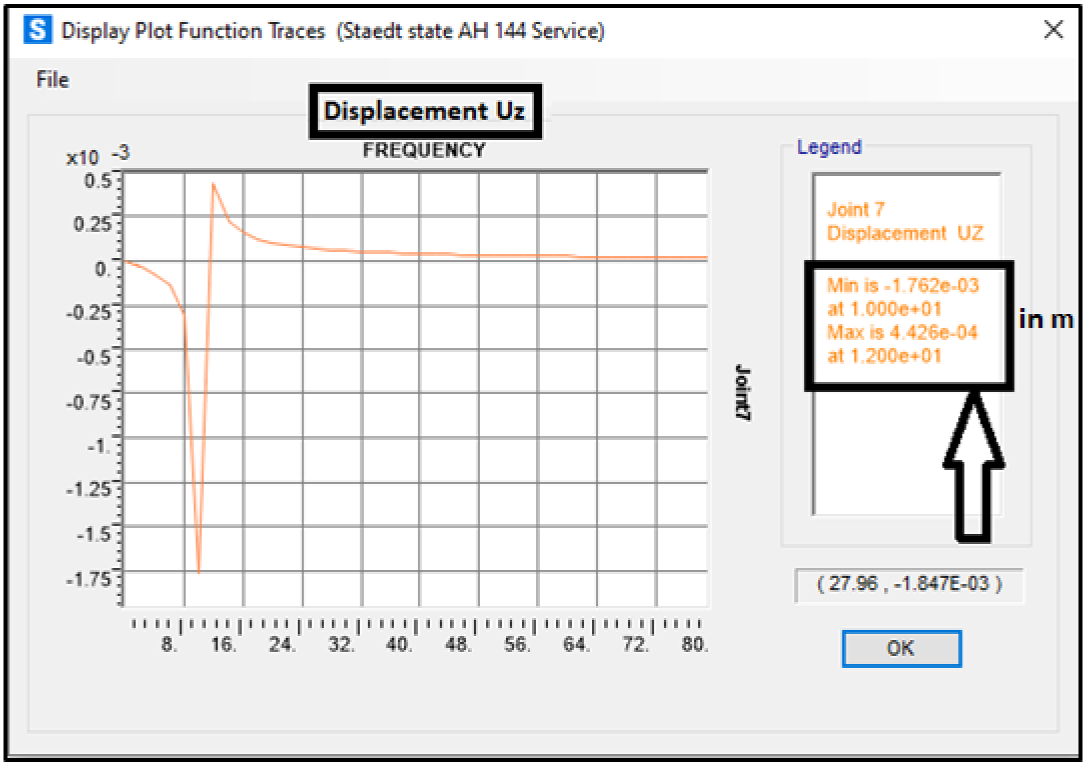


Figure III-11 The displacement diagram on the joint 7

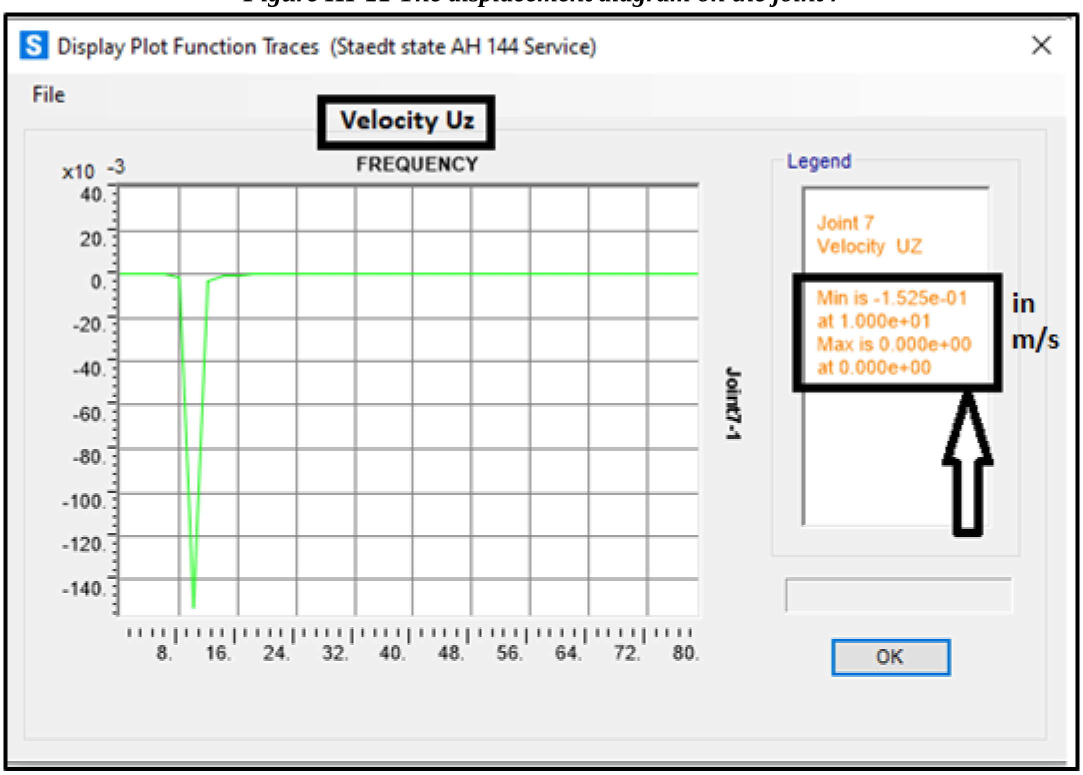


Figure III-12 The velocity diagram on the joint 7

III.4.3.3 On the Joint 421 (the first beam's joint closes the wall):

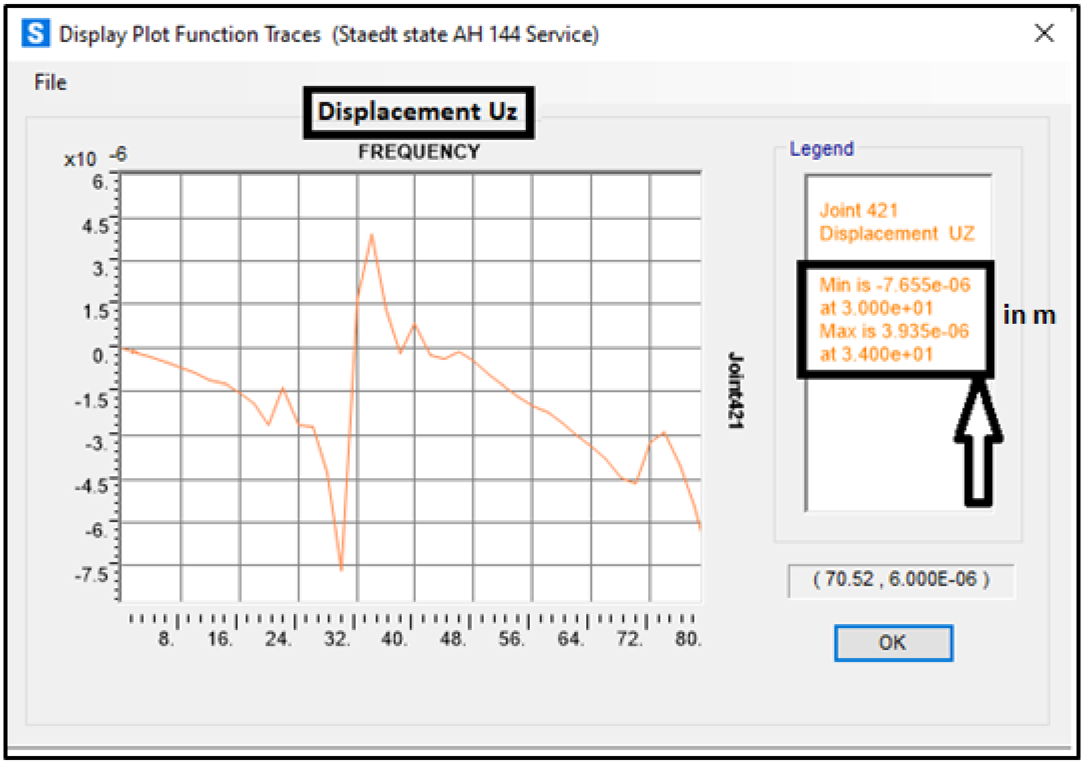


Figure III-13 The displacement diagram on the joint 421

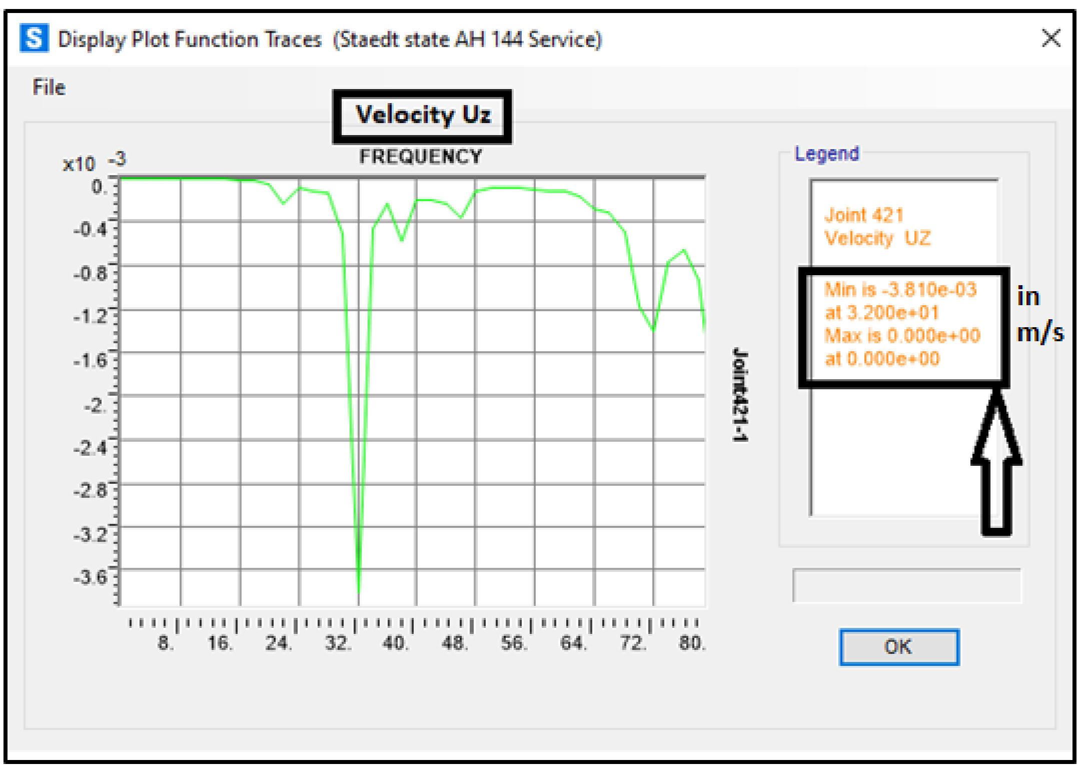


Figure III-14 The velocity diagram on the joint 421

III.4.3.4 On the joint 419 (the 2nd beam's joint closes the wall):

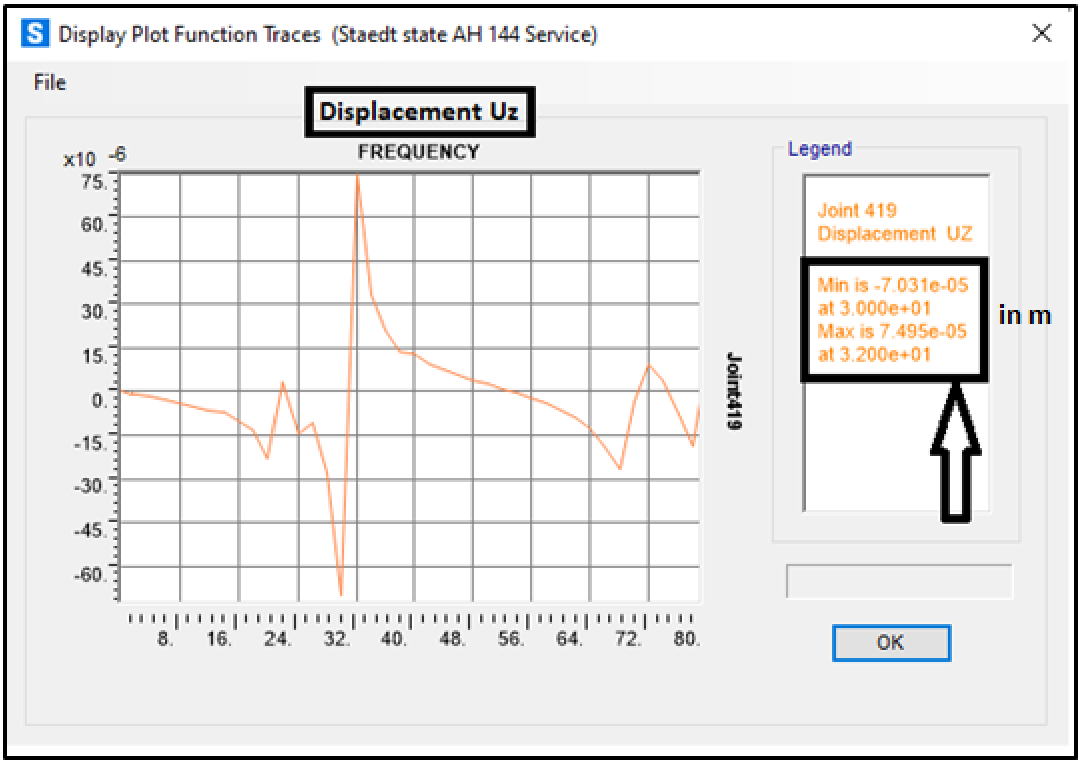


Figure III-15 The displacement diagram on the joint 419

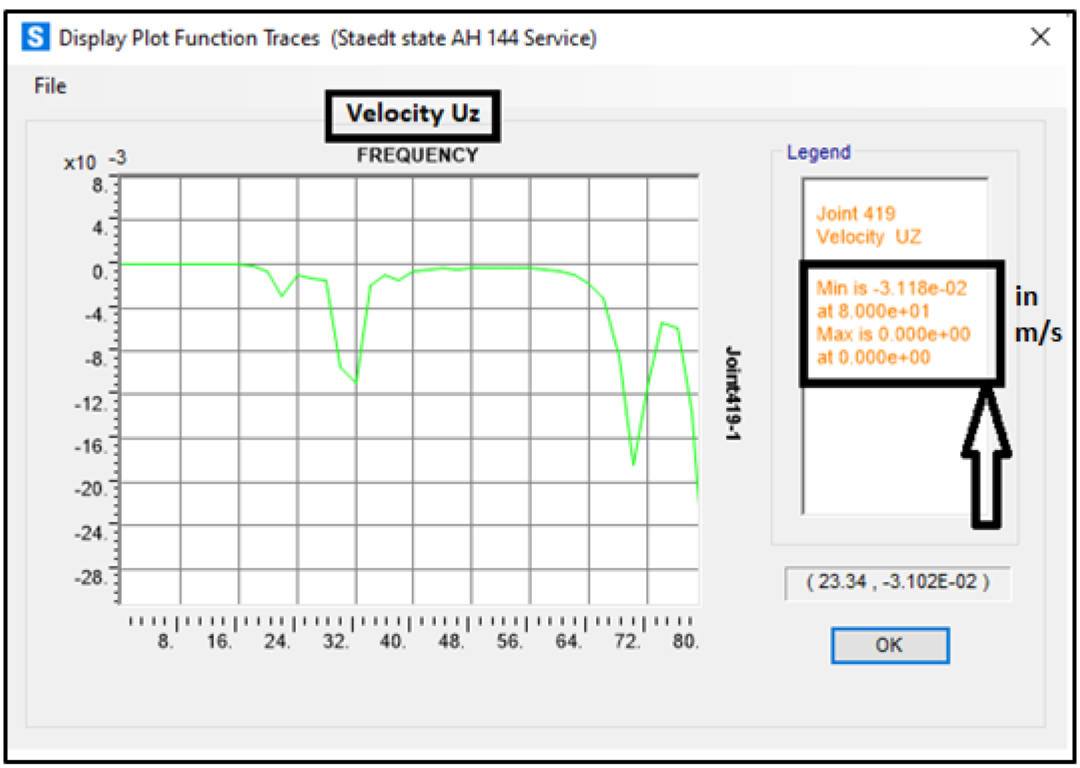


Figure III-16 The velocity diagram on the joint 419

III.4.3.5 On the joint 417 (the 3rd beam's joint closes the wall):

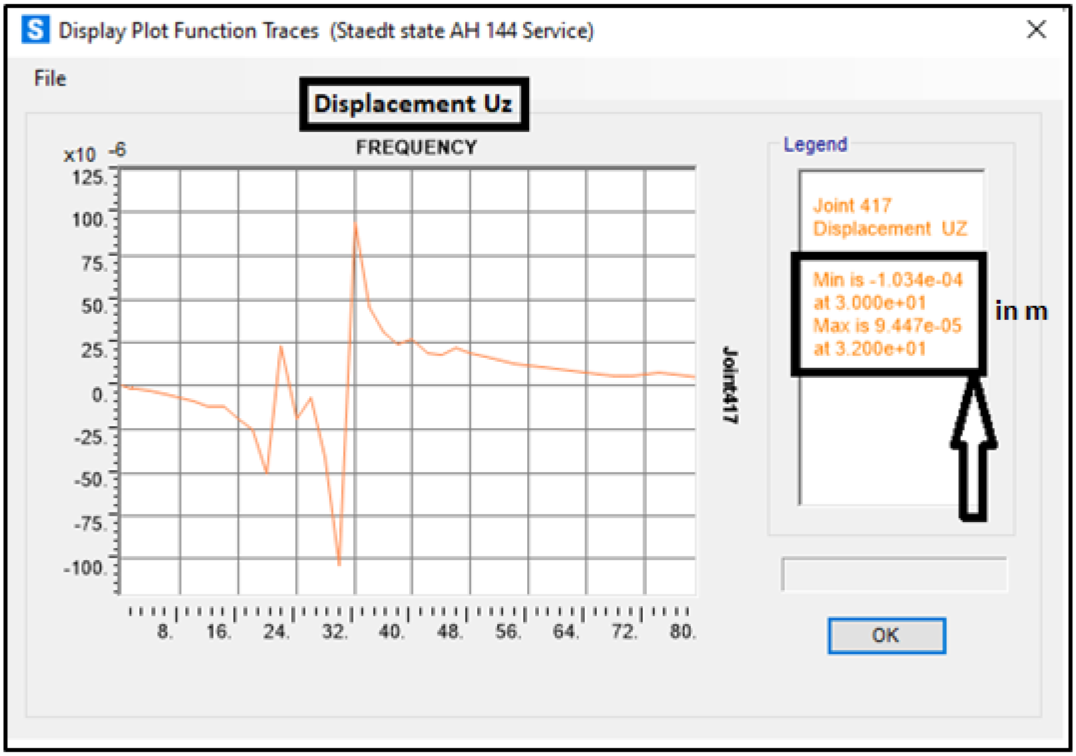


Figure III-17 The displacement diagram on the joint 417

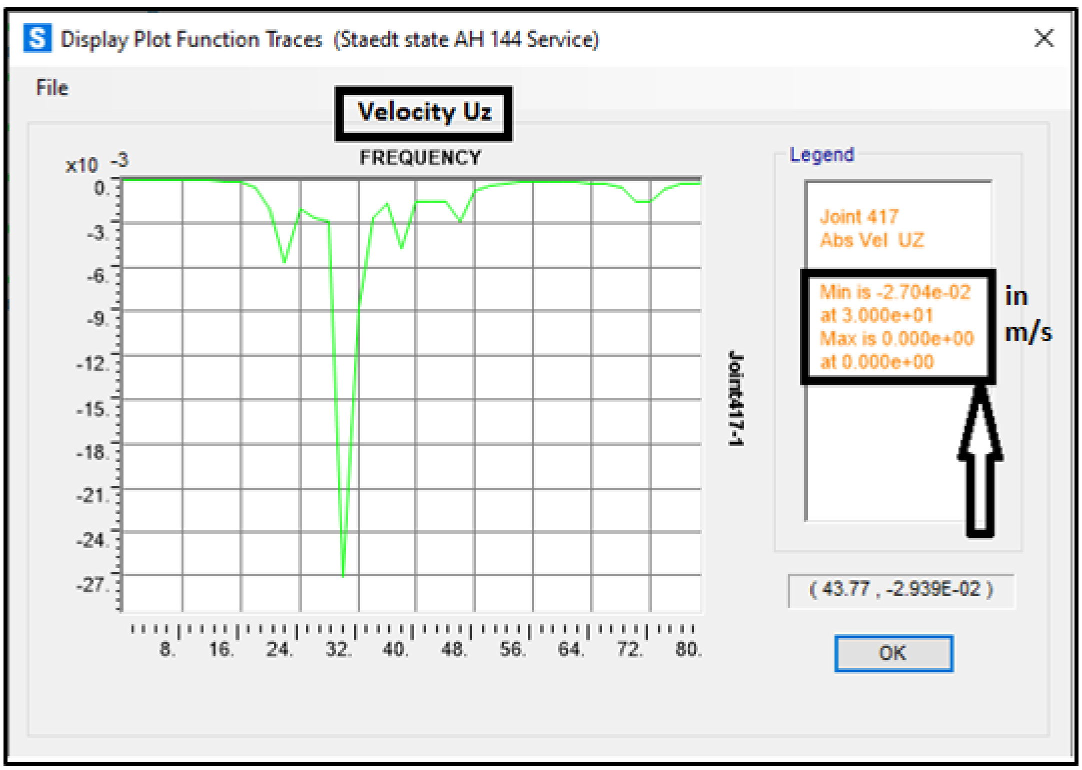


Figure III-0-18 The velocity diagram on the joint 417

III.4.3.6 On the joint 415 (the 4th beam's joint closes the wall):

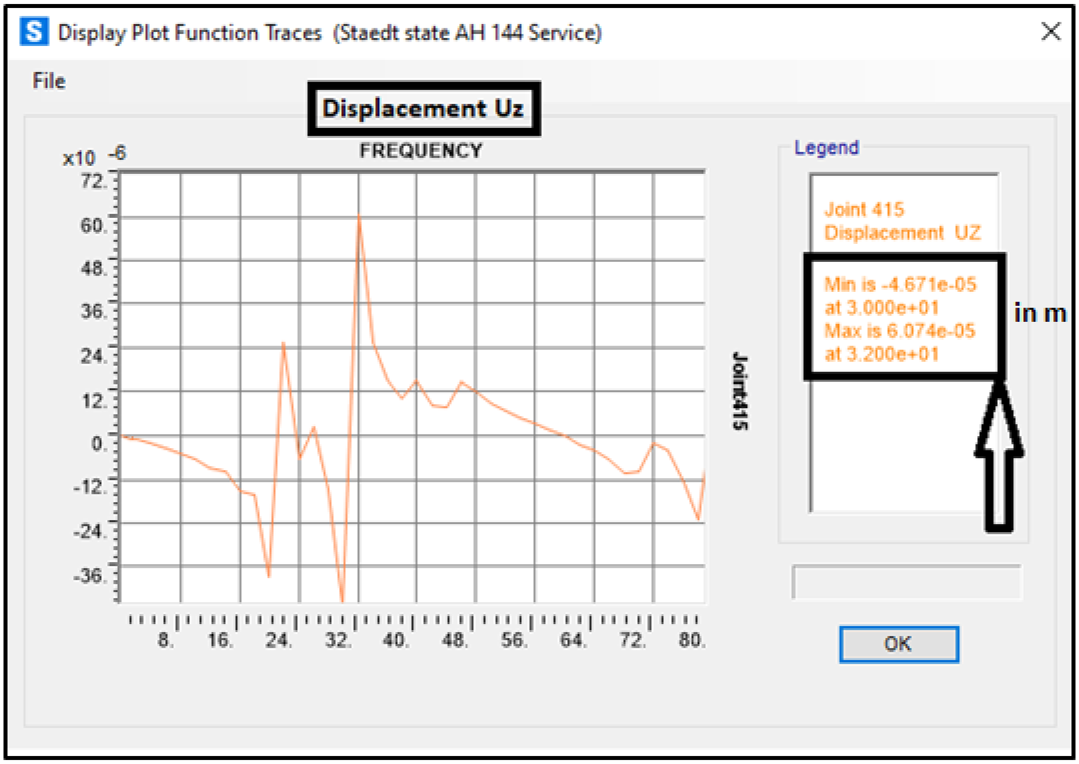


Figure III-19 The displacement diagram on the joint 415

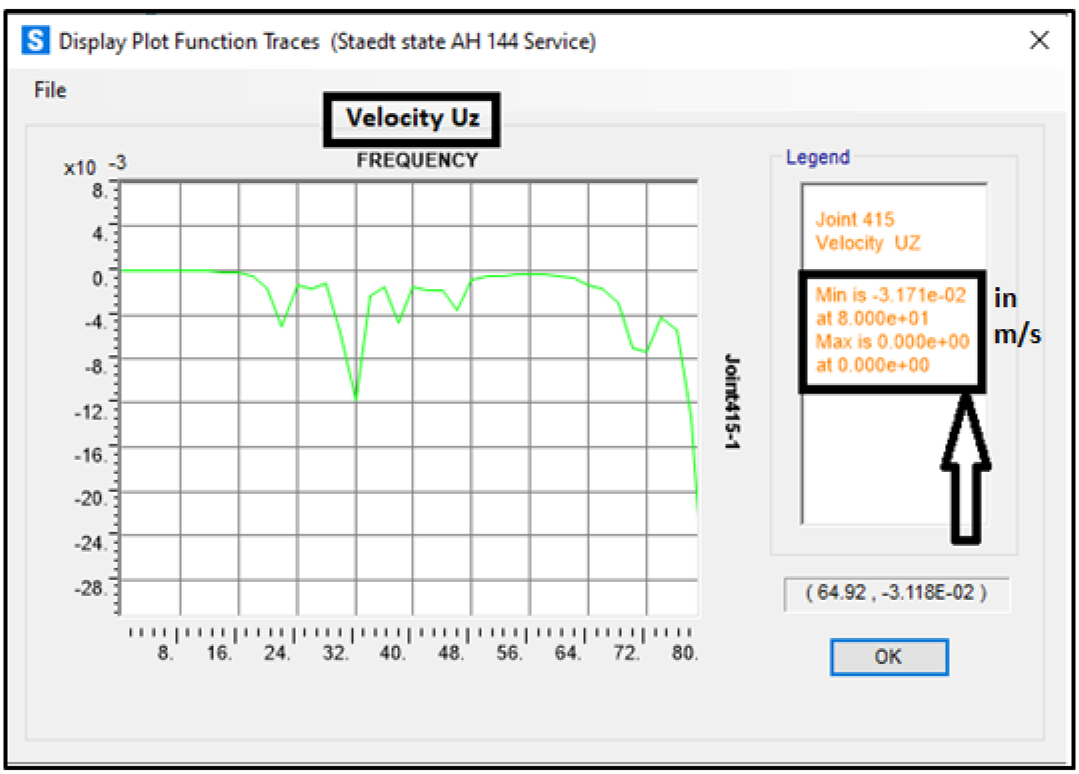


Figure III-20 The velocity diagram on the joint 421

III.4.3.7 On the joint 321 (above the central column):

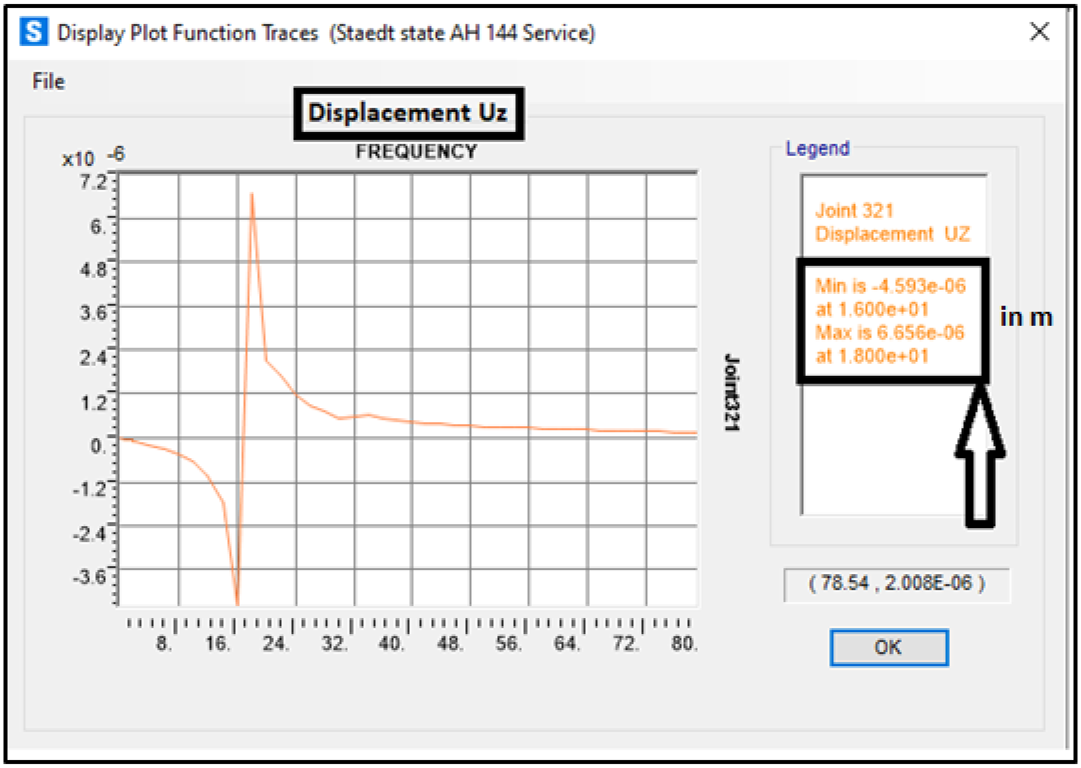


Figure III-21 The displacement diagram on the joint 321

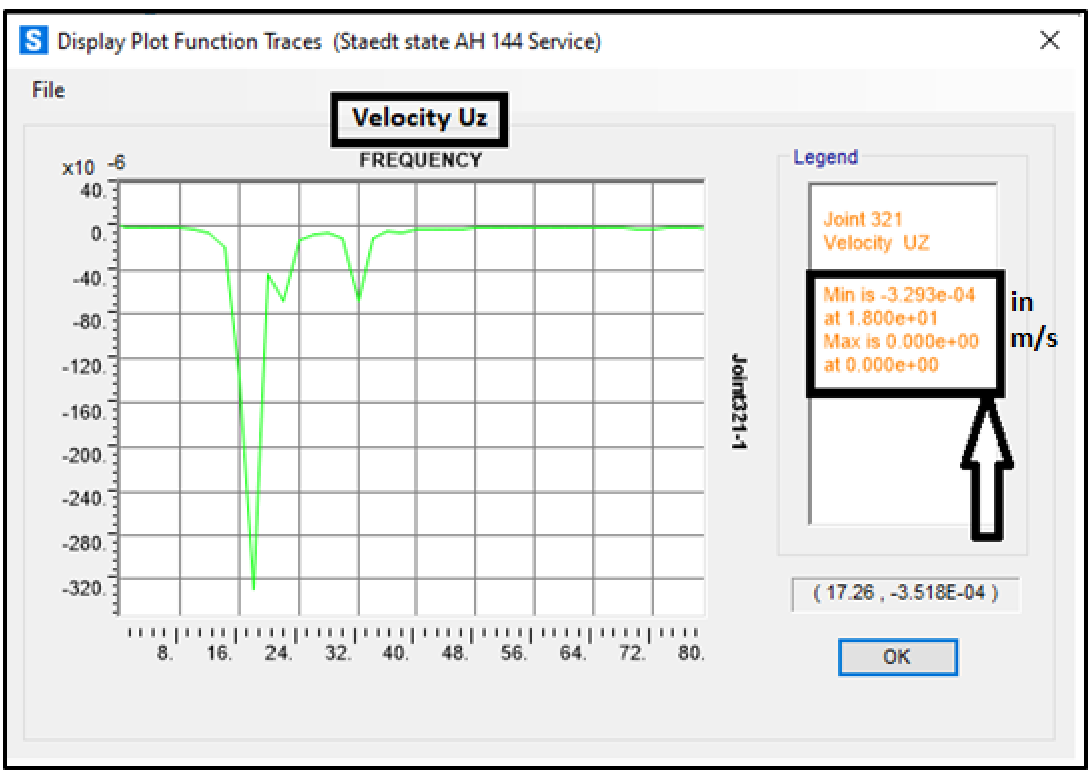


Figure III-22 The velocity diagram on the joint 321

III.4.3.8 Final results for both machines:

	Loaction	The amplitude in µm	The permissible amplitude in µm	The amplitude check	The velocity in mm/s	The permissible velocity in mm/s	The velocity check
Machine type AH 144	joint 9	1755	14	☒	84.68	25	☒
	joint 7	1762		☒	152.5		☒
	joint 421	7.66		☑	3.81		☑
	joint 419	74.95		☒	31.18		☒
	joint 417	103.4		☒	27.04		☒
	joint 415	60.74		☒	31.71		☒
	joint 321	6.66		☑	0.33		☑
Machine type AH 364	joint 9	1792	35	☒	6.13	25	☑
	joint 7	1709		☒	5.48		☑
	joint 421	98.15		☒	25.31		☒
	joint 419	296.8		☒	122.9		☒
	joint 417	441.8		☒	151.7		☒
	joint 415	419.5		☒	60.91		☒
	joint 321	50.15		☒	1.91		☑

Table III-10 The final result for both machines

III.4.3.9 Results interpretation:

- For the machine type AH 144 :

As seen in Table 5.1 we have tried to put the Blower fan loads on many joints (locations) to check the amplitude and velocity on each one of them.

For the joints 9 and 7 (the midspan of the joist) the value of the amplitude was very large comparing with the permissible amplitude, and this is due to their low stiffness.

For the joints 419,417 and 415 (the three middle joints of a beam) we had a better result but unfortunately, it’s still larger than allowable amplitude.

For the joints 421 and 321 (the closest beam’s joint to the wall and the joint that is above the central column of the mezzanine in order) the results were perfect for both the amplitude and velocity, actually the joint 321 give us a better result but for process and execution purposes it’s not recommended to put the Blower fan far from the cladding wall.

therefore, for the machine type AH 144 the best location choice will fall on the joint 421.

➢ For the machine type AH 364:

As seen in the same table 5.1 we had a different result, and for all the joints the amplitude was larger than admissible amplitude and this is due to the important unbalanced load generated by the blower type AH 364.

For this specific problem there are many solutions we can apply to solve this problem such as:

- Increasing the section of the support element(the beam in our case study)
- Put a damping system under the machine to dissipate the energy of the vibrations.
- Reinforcing the support element by a stiffener or a haunch.

For our case we tried to increase the section of the beam (for the joint 421 which consider the best location from the previous results) and after trying many sections we get the permissible amplitude and velocity at IPE 750.

Conclusion

As a conclusion, it could be summarized under these points:

- High speed machines must be checked and analyzed according to current codes such, as ACI 351 or others.
- The steady state analysis represents a powerful tool to analyze and study machine vibrations to determine the allowable amplitudes as well as velocities.
- Sometimes working with a global model for steady state analysis will lead to time consuming analyzing task, that’s why the discretized model is more efficient in matter of time consumption and both methods gave same results (with approximate accuracy).
- Defining the location of the machinery on the support is very important, and always lead to reduce the cost of the support.
- Big machinery always needs an appropriate support to avoid collapse, stiffness is an important parameter when it comes to under-tuned vibrating system.

Recommendations

At the end we recommend for the future student to go further in the study and try to touch more complicated cases and try to apply the steady state analysis in other fields:

- As we have already seen there's other types of machines with another type of motion, that means they will have different effect on the structure, therefore, we recommend students to study different machine types like reciprocating machines or impact machines.

- In our case study all the interactions were between the support structural elements and the machine, and that's because our Blower fan was placed on the mezzanine floor but in other cases the machine's equipment can be placed on the foundations and this type is called machine foundation and in this case there will be an interactions between the machine, the foundation and the soil, and this is a very recommended case to study.

References

American Concrete Institute (2018*) ACI 351.3R Report on Foundations for Dynamic Equipment.*

Anil K. Chopra (1988) *Dynamics of Structures Theory and Applications to Earthquake Engineering Fifth Edition (2020) In Si Unit.*

Deutsche Norm (1988) *DIN 4024 Machine Foundations Flexible Structures That Support Machines with Rotating Elements*

IRD Balancing *Balance Quality Requirements of Rigid Rotors the Practical Application of ISO 1940/1.*

Indian Standard (1982) *IS: 2974 Code of Practice for Design and Construction of Machine Foundations Part 3 Foundation for Rotary Type Machines. (Reaffirmed 2008).*

K.G. Bhatia (2008) *Foundations for Industrial Machines Handbook for Practicing Engineers.*

Mario Paz. Young Hoon Kim (1997) *Structural Dynamics Theory and Computation. Sixth Edition (2019).*

Suresh C.Arya (1973) *Design of Structures and Foundations for Vibrating Machines, Fourth Edition 1984.*

CNERIB (1999) *R.N.V 99 version 2013 (D.T.R.C 2-4.7) Snow and wind Regulations.*

CNERIB (2001) *Algerian equipements: calculates structures (the new action combinations)*

ECS (1993) *Eurocode 3: Design of Steel Structures version 2005.*

Md. Humayun Kabir Jony (2016) *Design Report of Steel Shed with Mezzanine Floor.*

Dahmani Lahlou (2012) *Calcul des Eléments de Constructions Métalliques.*

Mechehed Zakaria (2020) *Design of A Basement+16 Storys Office Tower Using a Response Spectrum Modal Analysis and A Static Nonlinear Pushover Analysis, University of Saad Dahleb, Blida.*

Manseur Sara (2021) *Étude D'un Parking En Charpente Métallique R+4, Universite Saad Dahleb Blida 1.*

Steel Warehouse Design Report

Www.brown.edu