VIBRATION ISOLATION: DESIGN APPROACH, THEORETICAL MODELING AND PRACTICAL APPLICATIONS

L. ElHifnawy, M. ElMenoufy and M. Gad ElHak
Structural Engineerring Department, Faculty of Engineering,
Alexandria University, Alexandria, Egypt.

ABSTRACT

There are two tecnniques for vibration isolation, namely mechanical isolation and foundation isolation. Mechanical isolation means the isolation of machines or structures from their foundations or from larger structural elements using localized isolators. Foundation isolation may provide the required protection for vibration problems in which mechanical isolation is inadequate. This research presents the selection and design procedures of localized isolators. Also, vibration standards developed by several international organizations are provided. If the vibration is found to be out of the standard limits, then a suitable isolator is to be used. Finally, the dynamic analysis of different structural elements of a paper manufacturing factory is presented. The analysis is performed by means of the complex eigen value approach. The study indicates that mechanical balancing and vibration isolation substantially reduces the transmitted force and motions.

Keywords: Vibration isolation, Mechanical balancing, Vibration standards, Machine foundation, Isolating Materials

1. INTRODUCTION

Isolators insure a smooth running of the machine to avoid harmful vibrations to both the workers and or the structure. The machine foundation should be designed carefully considering both static and dynamic loads. The importance of machine foundation design was not recognized in the past. It was believed that the larger the mass of the machine foundation, the smaller the amplitude of its vibration. Therefore massive foundations were designed using an estimated dynamic factor. Also, it is not enough to base the design on vertical loads only, multiplied by a dynamic factor because the operation of any machine would generate also forces acting in other directions. Fatigue must be considered through the design procedures too. Also, the foundation natural frequency should be different from the machine speed to avoid resonance. After designing the foundation, the whole system should be analysed to find out whether its vibration are acceptable or not. If not, the machine should be isolated. If the vibrations appear after a while in a working machine, then this machine must be mechanically rebalanced, first, to minimize these vibrations and to get smaller dimensions of isolators.

This research presents the design procedures of machine foundations under several types of forces, and the vibration standards for some types of machines, structures, and for the human organism. Also presents the isolator design procedures for different types and shapes.

Practical examples are studied at a paper manufacturing factory in one of its new buildings. All machines in this building are divided into four sections. The first section has all small machines which do not cause any harm. The second section has four fans which produce dangerous vibrations. The third section has two drums which produce high vibrations and shocks. The fourth section has four washing straw vessels which cause dangerous vibrations to the whole building.

2. MACHINES AND VIBRATIONS

An ideal machine would produce no vibration at all because all energy would be channelled into the job of work to be done. In practice, vibrations occur as a by-product of the normal transmission of cyclic forces through the mechanism. Machine elements react against each other and energy is dissipated through the structure in the form vibrations.

A fundamental requirement in all vibration work is the ability to obtain an accurate description of the vibration by measurements and analysis. Plot of the measured vibration amplitudes against frequencies is called vibration signal. Frequency analysis of vibration signals of the machine will make it possible to locate the source of many of the frequency components present in the frequency spectrum and to indicate which machine parts are deteriorating. Unbalance of rotating members, misalignment, bent shaft, bearing erosion,gear tooth damage,... etc. will all have their characteristic frequencies which can be revealed with the help of the frequency analysis.

The frequency spectrum of a machine in a normal running condition can therefore be used as a reference "Signature" for that machine. Subsequent analysis can be compared to this reference so that not only the need for action is indicated but also the source of the fault is diagnosed. The diagnostic charts (trouble shooting) given in Tables (1) and (2) will help determining the common faults causing excess vibrations [1].

3. VIBRATION STANDARDS

Standards should establish and control quality, performance and safety of equipments, structures and personal. The two recognized international organizations for vibration standards are the International Standards Organization (ISO) and the International Electrical Commission (IEC) [2]. The International Standards Organization has adopted a special quantity-vibration severity [3] for this purpose. The RMS value of vibration velocity of a machine is measured (at prescribed measuring points and in prescribed directions); the largest such measured value is said to characterize the vibration severity.

Table (3) shows a classification scheme [3] which applies to rotating machine in the operating speed range of 600 to 1200 rpm. It may be used to compare similar machines or to compare the vibration of normal machines with respect to their reliability, safety and effects on the environment. Also, Standards for acceptable levels of vibrations for compressors, shafts, induction motors, ... etc. are available in Ref. [4].

On the other hand, structures such as public buildings, offices, factories, bridges and power plants are subjected to vibration as a result of forces generated by ground motion, wind traffic and machinery. Vibration limits for structures and persons are shown in Figure (1) [5]. The figure shows five curves limit the zones for different sensitivities of responses by persons ranging from "not noticeable" to "severe". The envelope described by the shaded line as "limit for machines and machine foundations" indicates a limit for safety and not a limit for satisfactory operation of machines. Two curves are also included in Figure (1) to indicate limiting dynamic conditions for motions of structures caused by blasting.

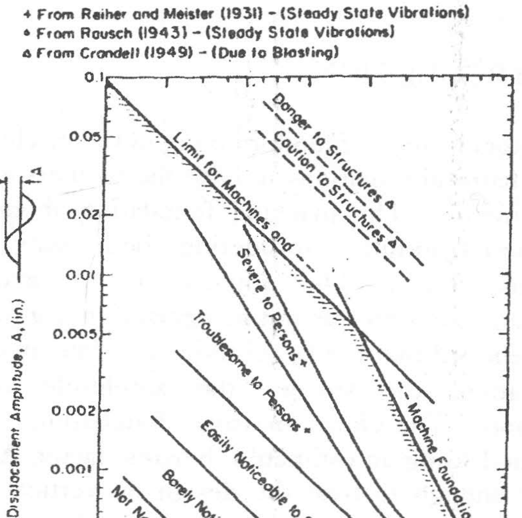


Figure 1. General limits of displacement amplitude for a particular frequency of vibration. Ref. [5].

Table 1. Vibration Trouble-shooting.

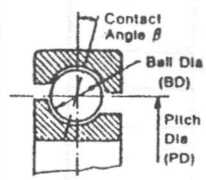
Nature of Fault	Frequency of Dominant Vibration (Hz=rpm/60)	Direction	Remarks
Rotating Members out of Balance	1 × rpm	Radial	A common cause of excess vibration in machinery
Misalignment & Bent Shaft	Usually 1 × rpm Often 2 × rpm Sometimes 3&4 × rpm	Radial & Axial	A common fault
Damaged Rolling Element Bearings (Ball, Roller, etc.)	Impact rates for the individual bearing component* ,o vibrations at ıigh frequencies ,2 to 60 kHz) often related to radial resonances in bearings	Radial & Axial	Uneven vibration levels, often with shocks. * Impact-Rates:  Impact Rates f (Hz) For Outer Race Defect $f(Hz) = \frac{n}{2} f_r \left(1 - \frac{BD}{PD} \cos\beta\right)$ For Inner Race Defect $f(Hz) = \frac{n}{2} f_r \left(1 + \frac{BD}{PD} \cos\beta\right)$ For Ball Defect $f(Hz) = \frac{PD}{BD} f_r \left[1 - \left(\frac{BD}{PD} \cos\beta\right)^2\right]$ n = number of balls or rollers f_r = relative rev./s between inner & outer races
Journal Bearings Loose in Housing	Sub-harmonics of shaft rpm, exactly 1/2 or 1/3 × rpm	Primarily Radial	Looseness may only develop at operating speed and temperature (e.g. turbomachines).
Oil Film Whirl or Whip in Journal Bearings	Slightly less than half shaft speed (42% to 48%)	Primarily Radial	Applicable to high-speed (e.g. turbo) machines.

Table 2. Vibration Trouble-shooting

Nature of Fault	Frequency of Dominant Vibration (Hz=rpm/60)	Direction	Remarks
Hysteresis Whirl	Shaft critical speed	Primarily Radial	Vibrations excited when passing through critical shaft speed are maintained at higher shaft speeds. Can sometimes be cured by checking tightness of rotor components.
Damaged or Worn gears	Tooth meshing frequencies (shaft rpm × number of teeth) and harmonics	Radial & Axial	Sidebands around tooth meshing frequencies indicate modulation (e.g. eccentricity) at frequency corresponding to sideband spacings. Normally only detectable with very narrow-band analysis and cepstrum
Mechanical Looseness	2 × rpm		Also sub- and interharmonics, as for loose Journal bearings
Faulty Belt Drive	1, 2, 3 & 4 × rpm of belt	Radial	The precise problem can usually be identified visually with the help of a stroboscope
Unbalanced Reciprocating Forces and Couples	1 × rpm and/or multiples for higher order unbalance	Primarily Radial	
Increased Turbulence	Blade & Vane passing frequencies and harmonics	Radial & Axial	Increasing levels indicate increasing turbulence
Electrically Induced Vibrations	1 × rpm or 1 or 2 times synchronous frequency	Radial & Axial	Should disappear when turning off the power

Table 3. Vibration severity ranges and examples of their applications Ref. [3].

Range of vibration severity	Examples of quality judgment for separate classes of machines			
Limits of range, mm/sec (RMS)	Small machines, class I	Medium machines, class II	Large machines, class III	Turbo-machines, class IV
0.28	A			
0.45		A	A	
0.71				A
1.12	B			
1.80		B		
2.80	C		B	
4.50		C		B
7.10	D		C	
11.20				C
18.00				
28.00		D	D	
45.00				D

The latters A, B, C and D represent machine vibration quality grades, ranging form good (A) to unacceptable (D).
(Rotating machines for speeds from 600 to 1200 rpm)

4. VIBRATION ISOLATION

Two different classes of problem may be identified in which vibration isolation may be necessary: (1) operating equipment may generate oscillatory forces which could produce harmful vibrations in the supporting structure, then isolation of forces is important, or (2) sensitive instruments may be supported by a structure which is vibrating appreciably, then isolation of motion becomes important [6].

The ratio of the maximum base force to the applied force amplitude, which is known as the transmissibility T_r of the support system, thus is given by (for the first class)

$$T_r = \varepsilon \sqrt{1+(2D\frac{\omega}{\omega_o})^2} \qquad (1)$$

where ω= force frequency, ω_o= natural frequency, D = damping ratio, and ε= dynamic magnification factor. The transmissibility in the second situation is defined as the ratio of the amplitude of motion of the mass to the base-motion amplitude. The expression for transmissibility is the same as that given by Eq. (1). A plot of the transmissibility for both basic SDOF isolation classes, as a function of the frequency ratio and damping ratio, is shown in Figure (2). It is noted that an isolation system is effective only for frequency ratio $\omega/\omega_o > \sqrt{2}$ and that the damping is undesirable in this range.It is more convenient to express the behavior of the system in terms of its isolation effectiveness R rather than the transmissibility, where the effectiveness is defined as $1\text{-}T_r$.

In fact there are three factors controlling isolation, namely the mass of the vibrating system, m, the stiffness, k, and the damping, c, of the system [7], as shown by the equation of motion of a single DOF system subjected to a dynamic force p(t),

$$m\ddot{z} + c\dot{z} + kz = p(t) \qquad (2)$$

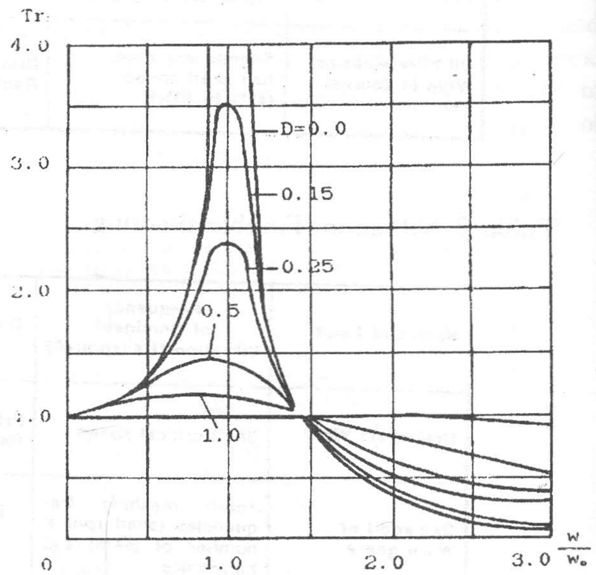


Figure 2. Transmissibility for damped systems [Ref. [6]].

The heavier the mass, the smaller is the movement for a given disturbing force. Also the heavier mass requires stronger springs for its support or thicker absorber to bear that load. The stiffer the springs, the less effective is the vibration isolation.

In general, the first step to isolate any vibrating machine is to study its mechanical balance to get the lowest vibrations transmitted to the supporting structure and then a suitable isolator may be used.

5. DESIGN PROCEDURES FOR MACHINE FOUNDATION

The design of a dynamically loaded structure requires that certain site and loading parameters be known even before preliminary sizing of the structure can be completed [8]. These design conditions and requirements may be generally classified into three groups: machine properties and requirements, soil parameters, and environmental requirements.

Once the proposed structure is modeled, trial sizes are selected, and an analysis is performed, the predicted behavior of the proposed structure is checked or compared against certain design requirements. These design requirements include: (1) the usual static strength check against soil or structural failures and excessive deformation; (2) comparison to limiting dynamic behavior including possible resonance conditions and maximum transmissibility factor; (3) consideration of possible fatigue failures in the machine or structure; and (4) consideration of environmental demands such as physiological and psychological effects on people or effect on adjoining sensitive equipment [8].

6. DYNAMIC ANALYSIS OF MACHINE FOUNDATION

The machine foundation should be designed carefully considering both the static and dynamic loads. Novak and Elhifnawy [9,10] were the first to present a complete dynamic analysis on machine foundation. With the aid of the computer their approach is suitable for any number of degrees of freedom; this is an advantage particularly for multi mass systems and eccentric forces on foundations. The solution is based on the notion of the complex eigenvalues. The foundation is considered as a rigid body resting on an elastic bed. The elastic bed may be the subsoil, the piles of a pile foundation or an artificially formed elastic pad consisting of steel springs. The various types of machine foundations can be modeled by lumped mass systems as shown in Figure (3). The one mass model (a) can be used for a foundation with no elastic pad under the machine. In the two mass model (b), the mass m_1 represents the elastically mounted machine and m_2 the foundation block supported by soil or piles. Model (c) comprises the mass of the machine, m_1, mass of the block, m_2, and mass m_3 of a protective through if it is needed to protect the isolation elements from the environment.

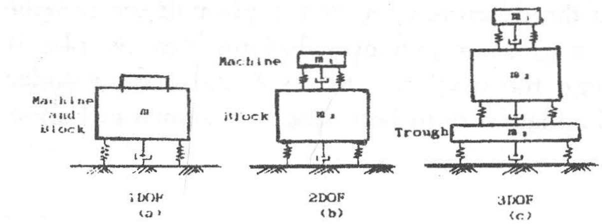


Figure 3. Basic mathematical models for machine foundations.

If the machine forces are centric and the foundation arrangement is symmetrical, only vertical vibrations occur and the foundations have one, two or three degrees of freedom as indicated. With eccentric force and/or asymmetrical arrangement of the system, horizontal translations and rotations of all masses occur and the number of degrees of freedom grows to three, six or nine for the three models shown in Figure (3).

7. EQUATION OF MOTIONS

By the aid of the schematic of a typical machine foundation shown in Figure (3), the governing equation of motion can be written as

$$[m]\ [\ddot{u}] + [c]\ [\dot{u}] + [k]\ [u] = [p(t)] \qquad (2)$$

in which [m], [c] and [k] are mass, damping and stiffness matrices respectively of the machine foundation system and [u] and [p(t)] are the displacement and load vector respectively.

8. DESIGN OF ISOLATION

For designing the vibration isolators and selecting the used material, many basic requirements should be considered. These requirements are-

- The isolator must support the vibration system and have high vertical stiffness
- The horizontal stiffness of the isolator must be sufficient.
- The isolator must contain sufficient damping to limit transmitted vibrations.
- The isolator material and shape must be carefully chosen to suit the environmental conditions.

Also the determination of the place of the isolator is of a great importance. Isolator can be placed between the machine and its foundation or under the foundation or in both places as shown in Figure (4).

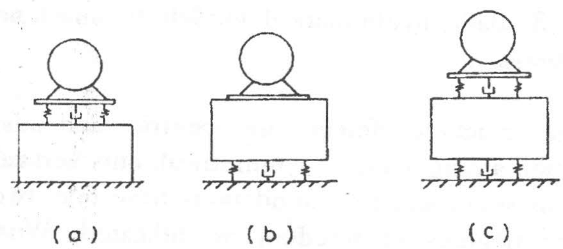


Figure 4. Possible places for the isolators: a- Between the machine and its foundation b- Under the foundation c- Both Places.

Isolating materials

As previously mentioned, a great care must be given to the choice of the isolating material. The used materials for isolato s may be grouped as:-
a) Natural and synthet rubber, b) Metal springs, c) Crok, felt, and various ompounded materials.

The materials listed in the third group are available only in sheet form and are used principally to mount machinery for easy and small applications. So, the first and second groups only will be discussed. All of these isolating materials posses damping as well as elastic properties. In general, the good damping properties of cork, rubber and felt make these materials very useful in isolating high frequency, small amplitude vibrations.

On the other hand, the metalic springs have very little damping capacity, so that high frequency vibration is transmitted through them. To eliminate this inconvenience, a layer of rubber or felt is often placed between the metalic springs and the foundation.

Rubber isolators

Natural and synthetic rubber find wide use because they may be molded to many shapes and can be easyly bonded to the isolated surfaces [11,12]. Natural rubber is made from the milky juice of the Hevea Brasiliensis tree, and synthetic rubber is made of polymers and copolymers of various organic compounds. Table (4) shows a qualitative comparison of the properties of the common kinds of rubber to select the suitable one for the environmental conditions [12].

The main step for the isolator design is to determine its static deflection (z_{st}). It depends on the allowable percentage of the transmitted vibrating force. These percentages are presented in Tables (5) and (6) for various kinds of machines. The isolator deflection is obtained through the working frequency of the system, f, [7] using the following equations

$$f = 15.8 \sqrt{\frac{1}{z_{st}} \frac{2-R}{1-R}} \text{ Hz} \tag{3}$$

where

z_{st} = the deflection in millimeters.

The dynamic stiffness of the isolator is defined by

$$k_d = W/z_{st} \tag{4}$$

where (W) is the machine weight.

Due to the rubber damping, the calculated stiffness (k_d) should be slightly reduced to obtain the stiffness value to be used in the design (k_s).

$$k_s = k_d/d_f \tag{5}$$

where (d_f) is a design factor always greater than one depending on the rubber type (d_f=1.4 for synthetic rubber) [13].

Table 4. Relative properties of various polymers Ref. [12].

Polymer designation	Common name	Shore A hardness range	Max. tensile strength lb/in.²	Max. tensile strength kg/cm²	Compression set	Tear resistance	Resilience Room temp.	Resilience High temp.	Heat resistance	Outdoor aging resistance	Low temperature flexibility	Specific gravity	Abrasion resistance	Adhesion strength*	Flex life
NR	Natural	30-100	4,000	280	Good	Good	High	High	Fair	Fair	Excellent	0.93	Excellent	Excellent	Excellent
SBR	SBR	40-100	3,000	210	Good	Fair	Fairly high	Fairly high	Fair	Fair	Good	0.94	Excellent	Excellent	Good
CR	Neoprene	40-95	3,000	210	Poor (ON) Good (W)	Good	Fairly high	Fairly high	Good	Excellent	Fair	1.23	Good	Good	Good
IIR	Butyl	40-75	2,000	140	Fair	Good	Low	Fairly high	Good	Good	Fair	0.92	Fair to good	Fair	Fair to good
EPDM	EPDM	45-100	2,000	140	Fair	Fair	Fairly high	Fairly high	Excellent	Excellent	Excellent	0.90	Fair to good	Fair	Fair
NBR	Nitrile	20-100	2,500	176	Good	Fair	Medium	Medium	Good to excellent	Poor	Fair	1.00	Good	Good	Fair
PO	Propylene Oxide	45-50	2,000	140	Fair	Fair	Fairly high	Fairly high	Good to excellent	Excellent	Excellent	1.01	Fair to good	Fair	Fair
[illegible]	Thiokol	20-80	1,300	91	Poor	Good	Medium	Fairly high	Fair	Excellent	Good	1.31	Fair	Poor	
S.	[illegible]	20-90	1,000	70	Excellent	Poor	Fairly high	Medium	Excellent	Excellent	Excellent	0.95	Poor	Poor	Good
CSM	[illegible]	45-[illegible]	2,800	197	Fair	Fair to good	Fairly high	High	Excellent	Excellent	Fair	1.10	Excellent	Poor	Good
ACM	[illegible]late	[illegible]-90	1,800	127	Good	Fair to poor	Low	High	Excellent	Excellent	Poor	1.09	Good	Poor	Good
FKM	Fluoro-[illegible]	[illegible]0-90	3,000	210	Excellent	Fair to poor	Medium	Medium	Excellent	Excellent	Fair	1.85	Good	Fair	Good

Table 5. Recommended isolation efficiencies for concrete floor slabs (Critical Areas) Ref. [7].

Kind of Machines		Transmissibility	Isolation Efficiency
Centrifugal Compressors		0.5%	99.5%
Centrifugal Fans	greater than 25 HP	2%	98%
Reciprocating Compressors	greater than 50 HP		
Pumps	greater than 5 HP		
Axial Flow Fans	greater than 50 HP	4%	96%
Centrifugal Fans	5 to 25 HP		
Reciprocating Compressors	10 to 50 HP		
Pumps	3 to 5 HP		
Unit Air Conditioners	Supported		
Fan Coil Units	Supported		
Axial Flow Fans	10 to 50 HP	6%	94%
Centrifugal Fans	up to 5 HP		
Reciprocating Compressors	up to 10 HP		
Pumps	up to 3 HP		
Air Handling Units			
Axial Flow Fans	up to 10 HP	10%	90%
Unit Air Conditioners	Hung		
Fan Coil Units	Hung		
Pipes	Hung		
Gas Fired Boilers (more than 100,000 BThU, 25 KW)			7 to 12 hz
Oil Fired Boilers (more than 60,000 BThU, 15 KW)			4 to 7 hz

Table 6. Recommended isolation efficiencies for concrete floor slabs (Less critical Areas) Ref. [7].

Kind of Machines		Transmissibility	Isolation Efficiency
Centrifugal Compressors		6%	94%
Centrifugal Fans	greater than 25 HP	10%	90%
Reciprocating Compressors	greater than 50 HP		
Pumps	greater than 5 HP		
Unit Air Conditioners	Supported		
Fan Coil Units	Supported		
Axial Flow Fans	greater than 50 HP	20%	80%
Centrifugal Fans	5 to 25 HP		
Reciprocating Compressors	10 to 50 HP		
Pumps	3 to 5 HP		
Air Handling Units			
Unit Air Conditioners	Hung		
Fan Coil Units	Hung		
Axial Flow Fans	10 to 50 HP	25%	75%
Axial Flow Fans	up to 10 HP	30%	70%
Centrifugal Fans	up to 10 HP		
Reciprocating Compressors	up to 10 HP		
Pumps	up to 3 HP		
Pipes	Hung		
Gas Fired Boilers (more than 100,000 BThU, 25 KW)			1[illegible] to 12 hz
Oil Fired Boilers (more than 60,000 BThU, 15 KW)			12 to 20 hz

To get the isolator thickness (t) and the cross section area A, the following equation is used

$$k_s = E.A/t \tag{6}$$

The value of the thickness should not be less than 100/15 Z_{st} [14].

Metal spring isolators

Springs are commonly used when the static deflection under the working system is great, or when temperature and other environmental conditions are unsuitable for rubber. The final deformation of steel springs occurs at the moment of loading, but for rubber the final deformation occurs after a certain time under loading and when the load is relieved, rubber does not go back to its original shape for a long period of time, while steel springs returns to its original shape at once.

There are three factors controlling the spring stiffness constant; the diameter of the spring (d), the diameter of the spring coil (D_c) and the number of turns in the spring (i). Because of the third factor, springs considered more economic than rubber since all sizes of springs are available and its stiffness can be controlled easyly by cutting some turns of the spring coil.

The stiffness factors for steel springs are [13]:

- The vertical stiffness factor

$$k_v = n\, k_{1sp} \tag{7}$$

and

$$k_{1sp} = \frac{1}{8i}\frac{d^4}{D_c^3}G \tag{8}$$

where

n = the number of springs, G = shear modules.

- The horizontal stiffness factor

$$k_u = k_v \frac{1}{\alpha\,(0.385)\,[1 + 0.77(h/D_C)^2]} \qquad (9)$$

where:

h = the height of the spring, h/D_C = the slenderness of the spring, and the coefficient α can be obtained from Figure (5).

- The stiffness factor for rotation in the vertical plane

$$k_\Psi = k_{1sp}\, I \qquad (10)$$

in which I= second moment of base area.

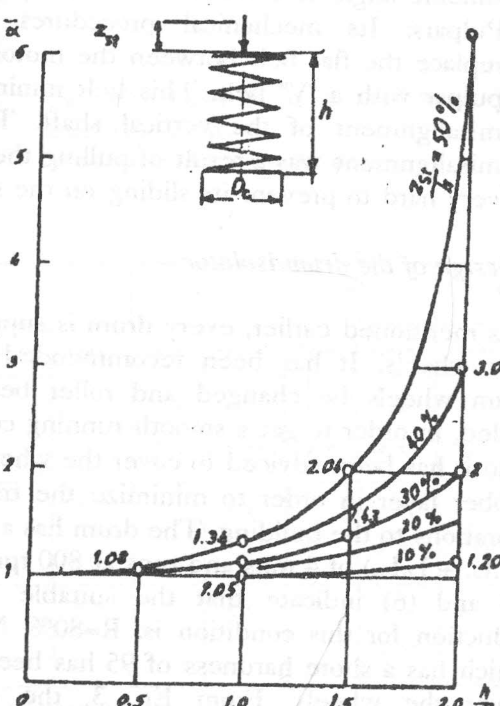


Figure 5. Factor (α) as a function of slenderness (h/D_c) and of (z_{sf}/h) Ref. [11].

9. PRACTICAL APPLICATIONS

The applications presented in this paper were chosen at one of the biggest paper manufacturing factory in the middle east (THE GENERAL INDUSTRIAL PAPER COMPANY-RAKTA). The washing straw building, which is relatively a new building, is annalyzed.

The building height is 22.7 m and consists of 8 different levels. The straw is introduced into the building to be washed and dewatered and then comes out to another building through two large spiral screws. The industrial process can be outlined as: first, conveying the straw to the first pulpar to be washed with water and when it floats, it is transferred to the second pulpar to be rewashed. When the straw floats again, then it is carried out to the dewatering drum to get rid of water. Finally, the dry straw will be moved out of the building through a spiral screw. This building has two producing lines. Every line consists of two pulpars, a drum, and a spiral screw. The pulpar is a big cylinder 36 m^3 capacity made of steel sheets and it has a fan that rotates around the vertical axis by a horizontal belt using a vertically mounted motor. The motor runs at 1000 rpm with a reduction ratio of 20:69 to run the pulpar at 290 rpm. The drum is 2.5 m diameter, 6m length, and rotates around its horizontal axis at a speed of 800 rpm. Also, there are four suction fans at the roof of the building running at 1500 rpm. In addition, there are a lot of small motors inside the building which may have small vibration effects and they may be neglected in the successive vibration analyses.

The four pulpars are established on two separated slabs at 8.4 m level. The Separation of slabs and columns from the rest of the building (2 cm) is to avoid the vibrations due to the rotation of the pulpars. The separation is not complete as all columns rest on combined foundations. The entire building is supported by pile foundations. The company has started using the building since September-1980, and according to the available reports, in February-1981, some walls cracked. The slabs vibrated highly causing fracture to the fixing anchor bolts of the pulpar and cracking the reinforced concrete pulpar knees. The pulpar was repaired, but a cycle of failures and repairs have been repeated every two months approximately.

On the other hand, drums cause another problem. Every drum is supported over four wheels, as shown in Figure (6). It has two steel rings to bear the drum over the wheels. Due to irregular loading, the rings lost its circular shape and the wheels weared off. In order to avoid this problem the wheels were brought closer together to maintain the drum level as shown in Figure (7). Consequently the angle between the drum and the wheels became smaller causing the drum to vibrate vigorously with very strong shock. This shock is due to the drum falling at the wheels. The shock can be felt clearly every where in the building and became worst with overloading.

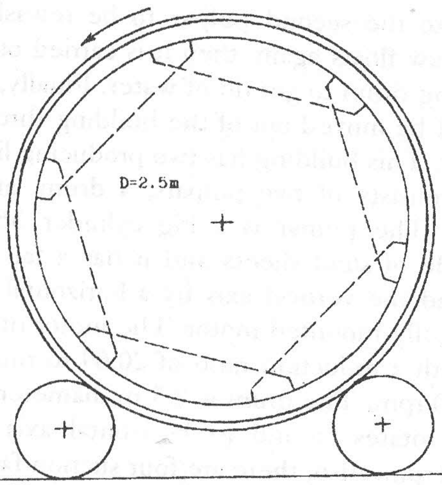


Figure 6. Details of the drum.

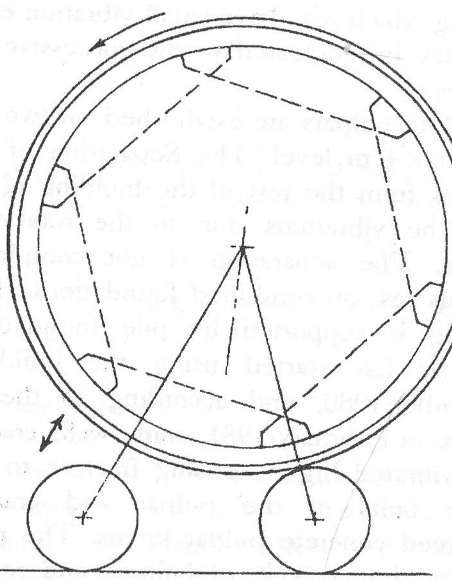


Figure 7. The drum problem.

Measurements and diagnosis

The vibration measurements of the building are presented in Table (7). All the measurements were found to be off standard limits. The diagnosis and the mechanical balancing procedures concluded from the vibration measurements were as follows:

- Fans: were unbalanced and one of these fans had a broken blade. Then, fans were balanced and vibration was measured again in December-1988. Considerable reduction of the peak velocity of vibration is accomplished as shown in Figure (8) and also in Table (8).
- Drums: were unbalanced because of its mechanical faults. The repairing procedures included the supporting wheels and its places. The wheels were replaced with other ones with roller bearings covered with a rubber layer as shown in Figure (9). Also they were returned back to its original places in order to get a suitable angle to avoid the drum jumping.
- Pulpars: Its mechanical procedures were to replace the flat belt between the motor and the pulpar with a "V" belt. This belt minimizes the misalignment of the vertical shaft. The shaft misalignment was a result of pulling the flat belt very hard to prevent its sliding on the shaft.

Design of the drum isolator

As mentioned earlier, every drum is supported by four wheels. It has been recommended that the drum wheels be changed and roller bearings be added, in order to get a smooth running conditions. Also it has been adviced to cover the wheels with a rubber layer in order to minimize the transmitted vibrations to the building. The drum has a total load (D.L. + L.L.) of 4 tons and runs at 800 rpm. Tables (5) and (6) indicate that the suitable vibrations reduction for this condition is, R=80%. Neoprene, which has a shore hardness of 95 has been used to cover the wheels. From Eq. 3, the deflection Z_{st}=8.425 mm, and from Eqs. 4,5; $k_d = W/Z_{st}$ =118.689 kg/mm, and $k_s = k_d/1.4$ = 84.778 kg/mm.

For a rubber width = 10 cm and a contact length between the drum and the wheel = 7 cm (Figure (9)), and if Neoprene has modules of elasticity E = 180 k_g/cm^2, then Eq. 6 yields the rubber thickness t = 14.86 cm.

Table 7 Specimen of The Vibration Measurements

	Direction of Measurements	Displacement µm	Velocity mm/sec	Accelera.
Pulpar Leg	Horiz.	133	3.09	0.060g
Pulpar Leg	Vert.	100.1	1.62	0.025g
Pulpar Leg	Vert.	58.5	0.71	0.009g
Pulpar Leg	Vert.	30.4	2.20	0.060g
Pulpar Leg	Vert.	30.3	4.00	0.080g
Pulpar Leg	Vert.	51.6	1.28	0.168g
Pulpar Leg	Vert.	15.7	1.60	0.028g
Pulpar Leg	Horiz.	83.5	1.20	0.044g
Pulpar Leg	Horiz.	29.9	0.81	0.068g
Pulpar Leg	Vert.	44.0	1.23	0.023g
Beam (8-9)	Vert.	26.5	1.20	0.034g
Beam (8-9)	Horiz.	23.3	1.30	0.060g
Beam (14-13)	Vert.	72.0	5.00	0.012g
Beam (11'-11)	Vert.	25.1	0.80	0.069g
Col. (14)	Horiz.	29.5	1.00	0.033g
Col. (13)	Horiz.	53.0	7.00	0.200g
Col. (11)	Horiz.	51.6	0.41	0.067g
Col. (11')	Horiz.	51.6	0.67	0.066g

Table 8 Overall Levels of Vibrations for Suction Fans
- peak velocity (mm/sec) -

Fan No. / Bearing No.	F1		F2		F3		F4	
	Axial	Radial	Axial	Radial	Axial	Radial	Axial	Radial
1	2.6	2.8	7.8 (3.5)	7.2 (2.6)	2.5 (2.6)	2.4 (2.4)	4.5 (2.3)	4.0 (2.1)
2		5.3		6.2 (1.3)		3.2 (2.3)		2.4 (1.6)
3		5.5		4.0		3.1 (2.6)		2.7

0.0 Before Repairing
(0.0) After Repairing

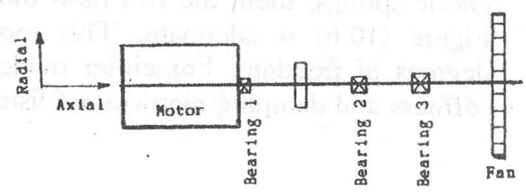


Table 9 Comparison of The Output Results for The Pulpar Slab System

	No Isolation	Isolation Kind		
		Springs	Springs + SBR	Springs + Butyl
w1	27.68	0.92	1.262	1.1
D1	0.66%	6.45%	6.50%	9.00%
w2	66.275	23.96	22.824	20.594
D2	1.5%	13.96%	14.50%	20.80%
U1max	0.0783mm	0.1544mm	0.1490mm	0.1560mm
U2max		0.0188mm	0.0161mm	0.0145mm
ψ1max	9.21E-6	136.4E-6	133.8E-6	141.2E-6
ψ2max		4.8E-6	4.1E-6	3.7E-6

Table 10 The output Results for Different Spring Isolators Having Same Vertical Stiffness and Different Dimensions

$$k_v = \frac{1}{8i}\frac{d^4}{D_c^3} G \quad , \quad h > d * i + z_{st}$$

D_c =	20.0	20.0	20.0	17.0	17.0cm
d =	4.1	4.3	4.85	3.8	3.8cm
i =	4	5	8	4.9	7.1
h/D_c =	1.25	1.50	2.35	1.65	2.06
w1	0.925	0.920	0.915	0.917	0.920
D1	6.7%	6.45%	6.5%	6.5%	6.5%
w2	27.04	23.96	17.95	23.06	16.26
D2	13.13%	13.96%	15.03%	14.17%	15.24%
U1max	0.2397	0.1544	0.0758	0.1401	0.0624mm
U2max	0.0397	0.0188	0.0049	0.0157	0.0033mm
ψ1max	211.3E-6	136.4E-6	68.0E-6	124.2E-6	56.49E-6
ψ2max	10.04E-6	4.8E-6	1.25E-6	3.95E-6	0.839E-6

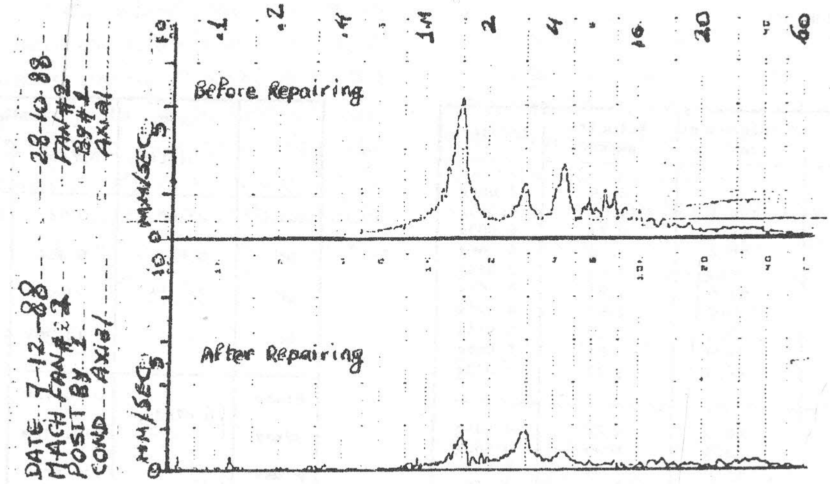


Figure 8. Vibration analysis for suction fan no. (2) before and after repairing (axial direction-bearing 1).

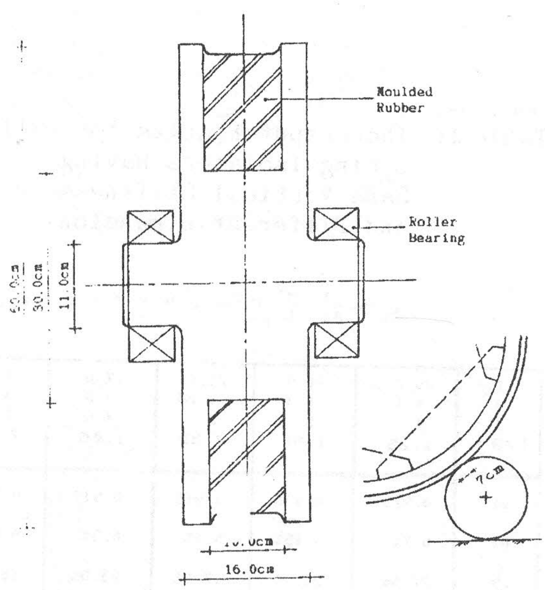


Figure 9. The isolated drum wheel.

Design of the pulpar isolator

The pulpar capacity is 36 m^2 and is made of steel sheets weighing 2.2 tons with total load (D.L. + Cargo L.) of 40 tons. The pulpar runs at 290 rpm. For a vibrations reduction of R = 80%, then the static deflection would be z_{st} = 65.46 mm.

Because of this high value of deflection, rubber may be unsuitable to be used as an isolator and springs may be more suitable. The stiffness coefficient needed for this deflection is

$$k_{1leg} = F/z_{st} = 347.376 \quad \text{kg/mm}$$

here F is the centrifugal force on the pulpar = 22.74 ton. With four springs for each leg

$$k_{1sp} = 346.376/4 = 86.844 \quad \text{kg/mm}$$

The spring dimensions can be determined using Eq. 8.

According to this equation, one can choose the suitable dimensions of the springs available at local markets and its stiffness can be controlled easyly by cutting some turns of the spring coil.

Theoretical modeling of pulpar

The pulpar and its structure are modeled as lumped mass system. The number of degrees of freedom depends on whether there is isolation or not. All the required equations for base isolated structures have been provided in Refs. [10].

For the case of no isolation (Figure (10-a)) a one mass system with two degrees of freedom; the horizontal translation, u(t); and rotation in the vertical plan, $\Psi(t)$ is used. When the pulpar rests on elastic springs, then, the two mass model shown in Figure (10-b) is adequate. This model has four degrees of freedom. For either models, the mass, stiffness and damping matrices are listed in Ref. [4].

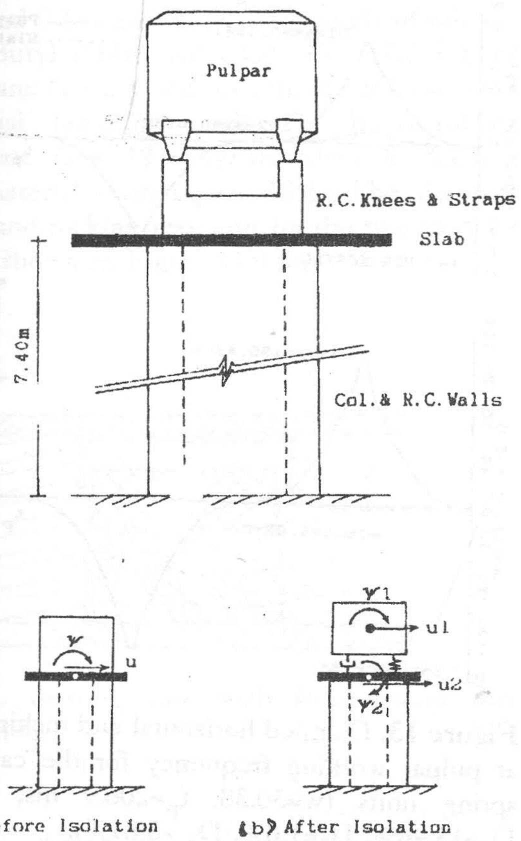


Figure 10. Modeling the pulpar and its structure.

The Complete Solution and Result Analysis

The response of the pulpar and its structure due to the unbalanced (centrifugal) force is treated as one due to a sine load which has a frequency of $\Omega = 2\pi/t_p$, where t_p=60/290=0.2069 sec. Then the unbalanced force can be defined as $P = P_o \sin \Omega t$, and the resulting moment can be defined as $M = P_o e \sin \Omega t$, where P_o= 101.16 ton, e = 1.3 m, and M=131.51 t.m.

Using the complex eigenvalue analysis, the equation of motion, Eq. 2 (of order n), is transformed into 2n reduced differential equations of the first order:

$$A \dot{z} + B z = F(t) \tag{11}$$

in which

$$A = \begin{bmatrix} 0 & m \\ m & c \end{bmatrix} \quad B = \begin{bmatrix} -m & 0 \\ o & k \end{bmatrix} \tag{12-a}$$

$$z = \begin{Bmatrix} \dot{u} \\ u \end{Bmatrix} \quad F(t) = \begin{Bmatrix} 0 \\ p(t) \end{Bmatrix} \tag{12-b}$$

For the solution of the response to pulse loading, free vibration has to be analyzed first. For free vibration F(t)= 0 and the solution of the complex eigenvalue problem yields 2n eigenvalues, μ_i and corresponding eigenvectors Z_j for j=1,2,... 2n.

The response to a given pulse can be obtained from Eq. (11) by means of the complex eigenvectors. Using the linear transformation:

$$z(t) = Z\ q(t) \tag{13}$$

In which, the components of the vector q(t) are the generalized coordinates. Then Eq. 11 becomes

$$A Z \dot{q} + B Z q = F(t) \tag{14}$$

The complete solution to Eq. 14 is established as a sum of the homogeneous solution, given by the initial condition $q_j(0)$, and a particular integral of the nonhomogeneous equation. For a sine pulse and zero initial conditions, the solution is [15]

$$q_j(t) = \frac{1}{A_j} \left(\sum_{i=1}^{2n} F_i * Z_{j,i} \right) * \frac{-\mu_j \sin \Omega t + \Omega \left[e^{\mu_j t} - \cos \Omega t \right]}{\mu_j^2 + \Omega^2} \tag{15}$$

where F is the load vector defined as $\{F(t)\} = <0\ 0\ P_o\ P_o\ e>^T$ for before isolation case and $\{F(t)\} = <0\ 0\ 0\ 0\ P_o\ P_o e\ 0\ 0>^T$ for after isolation case and $[A_j] = [Z]^T [A]\ [Z]$. The complete solution of the problem have been obtained using a suitable computer program. The following remarks are prescribed:

Figure (11) shows the damped vibration modes for the two cases, before and after isolation. Figure (11-b) shows absolute values of the complex damped modal displacements but does not indicate the phase shift between the pulpar and its supporting structure.

The damped horizontal and rocking response calculated for the sine load are shown in Figures (12) and (13) for both cases. The importance of vibration isolation is clearly obvious.

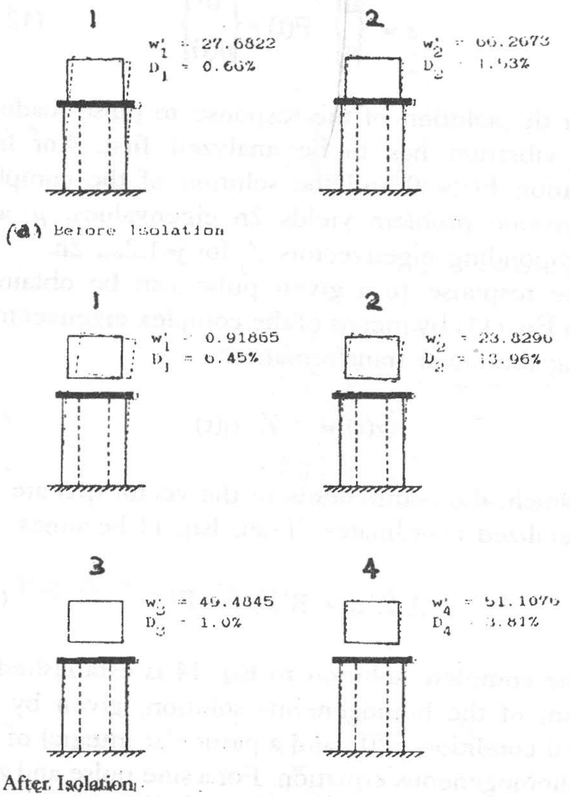


Figure 11. Damped vibration modes for the pulpar and its supporting structure.

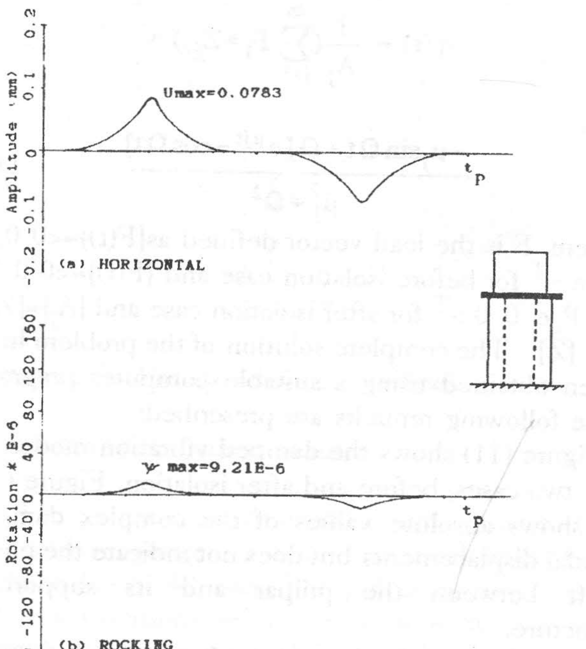


Figure 12. Damped horizontal and rocking desponse at the pulpar working frequency for the case of before isoltion (w=30.38, t_p=206 ms, D_1=0.66%, D_2=1.53.

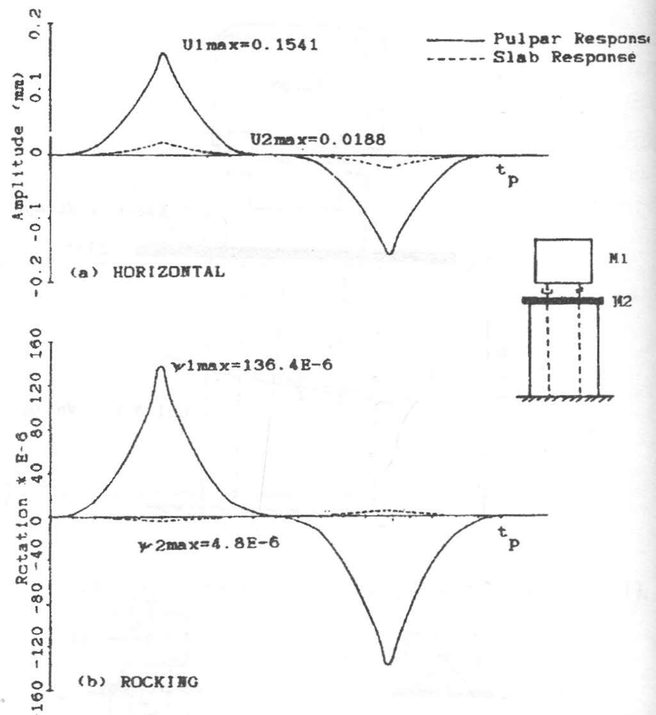


Figure 13. Damped horizontal and rocking response at pulpar working frequency for the cases of four spring units (w=30.38, t_p=206.9 ms, D_1=6.45%, D_2=13.96%, D_3=1.0%, D_4 =3.81%).

For the case of before isolation, the maximum horizontal and rocking responses are 0.0783 mm and 9.21E-6 rad respectively. After isolation, the maximum horizontal response for the supporting slab is tremendously decreased (24% of its previous value). The rocking response dropped down to one half of its value before isolation. Naturally the pulpar body responses increased because the pulpar body is mounted on springs which produce high response for low frequencies. This increase is within the machine standard limits [Ref. 5 and Figure 1].

The importance of mechanical balancing is clearly obvious by comparing the pulpar measurements from Table (6) and the response calculated for before isolation case, Figure (8), and how these values have been reduced considerably after balancing.

The same pulpar slab system is isolated using rubber in addition to springs. Special rubber cylinder is arranged in parallel to the spring system to increase the horizontal damping. The springs and rubber cylinder are combined in one unit as shown in Figure (14). The rubber cylinder is designed not to carry any vertical load. The rubber cylinder is 0.3 diameter and 0.1m thickness. Two kinds of rubber, SBR and Butyl rubber are used. The SBR rubber properties are: G = 20.5 kg/cm^2, shore hardness = 90, and material damping = 5%. The Butyl rubber properties are: G = 13.9 kg/cm^2, shore hardness = 70, and material damping = 10%. The damped horizontal and rocking response for the two kinds of rubber are shown in Figure (15) and (16).

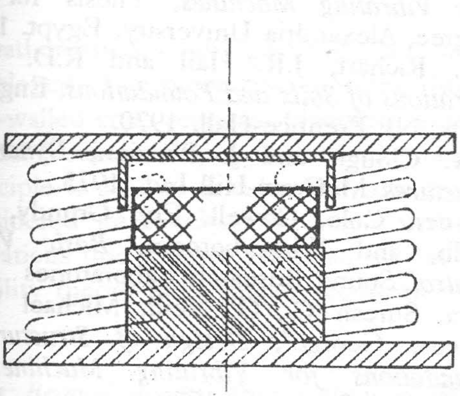


Figure 14. Bearing case with four springs and rubber cylinder.

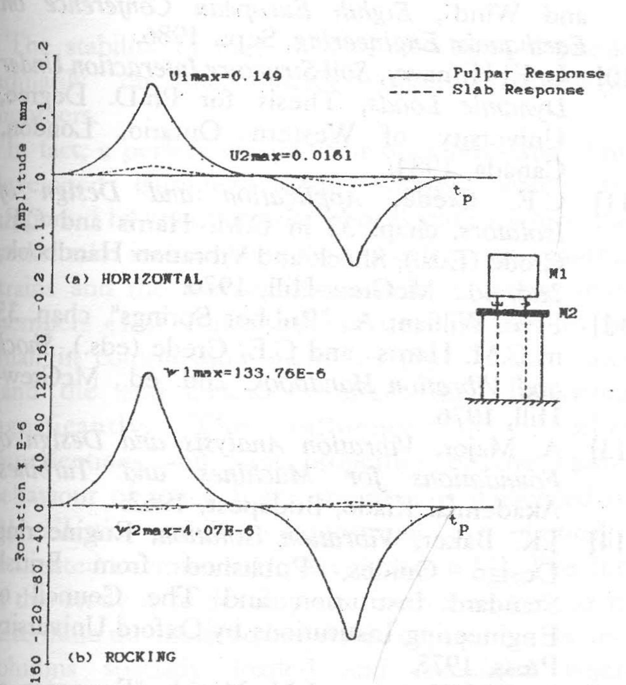


Figure 15. Damped horizontal and rocking response at the pulpar working frequency for the case of four spring units and four SBR rubber cylinders (w=30.38, t_p=206.9 ms, D_1=6.5%, D_2=14.5%, D_3=0.01%. D_4=2.6%.

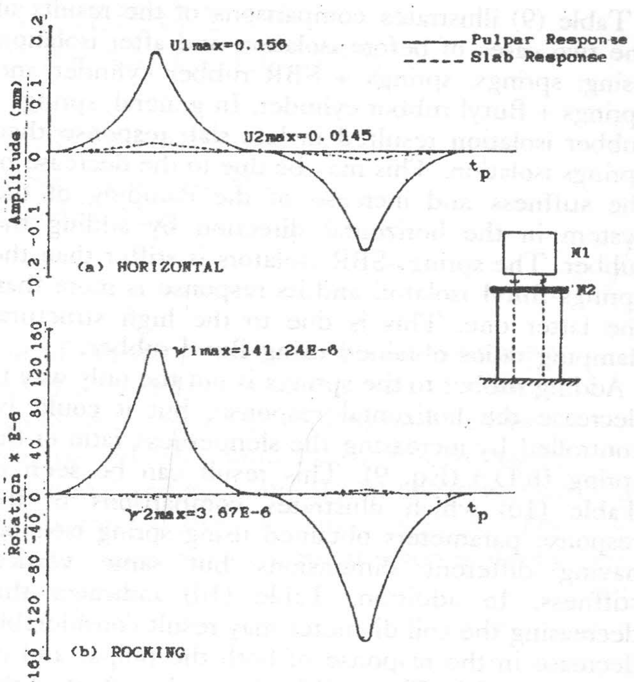


Figure 16. Damped horizontal and rocking response at the pulpar working frequency for the case of four spring units and four butyl rubber cylinders (w=30.38, t_p=206.9 ms, D_1=9.0%, D_2=20.8%, D_3=0.01%. D_4=3.0%).

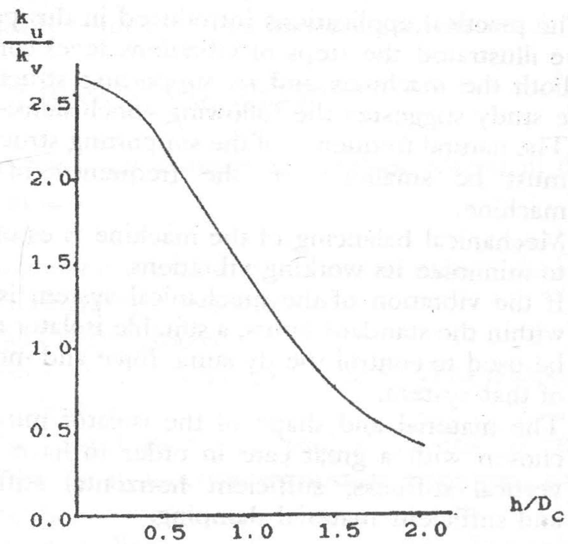


Figure 17. The effect of spring slenderness (h/D_c) its horizontal stiffness for z_{st}/h=20.

Table (9) illustrates comparisons of the results of the two cases of before isolation and after isolation using: springs, springs + SBR rubber cylinder and springs + Butyl rubber cylinder. In general, springs - rubber isolation resulted in less slab response than springs isolation. This may be due to the decrease of the stiffness and increase of the damping of the system in the horizontal direction by adding the rubber. The springs-SBR isolators is stiffer than the springs-Butyl isolator, and its response is more than the latter one. This is due to the high structural damping ratios obtained using Butyl rubber.

Adding rubber to the springs is not the only way to decrease the horizontal response, but it could be controlled by increasing the slenderness ratio of the spring (h/D_c) (Eq. 9). This result can be seen in Table (10) which illustrates comparisons of the response parameters obtained using spring isolators having different dimensions but same vertical stiffness. In addition, Table (10) indicates that decreasing the coil diameter may result considerable decrease in the response of both the pulpar and its supporting slab. Figure (17), shows the effect of the slenderness ratio on the horizontal stiffness of the spring. It can be seen that the horizontal stiffness decreases with the increase of the slenderness ratio. Consequently, the second natural frequency and response displacements decrease considerably.

10. CONCLUSIONS

The practical applications introduced in this paper have illustrated the steps of vibrations level control of both the machines and its supporting structure. The study suggestes the following conclusions:-

- The natural frequen[illegible] of the supporting structure must be smaller [illegible]n the frequency of the machine.
- Mechanical balancing of the machine is essential to minimize its working vibrations.
- If the vibration of the mechanical system is not within the standard limits, a suitable isolator must be used to control the dynamic force and motion of that system.
- The material and shape of the isolator must be chosen with a great care in order to have high vertical stiffness, sufficient horizontal stiffness and sufficient material damping.

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