

THE USE OF TUNED MASS DAMPERS TO CONTROL ANNOYING FLOOR VIBRATIONS

by

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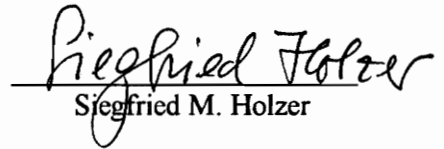
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(ABSTRACT)

Floor vibrations due to occupancy activities on a floor are sometimes annoying to the occupants. Correcting floor vibrations is difficult and can be expensive. The use of tuned mass dampers to control annoying floor vibrations is sometimes a viable solution.

Tuned mass dampers (TMDs) have been used primarily to control only one or two modes of vibration of a floor. Experimental research was performed using prototype TMDs to control one, two, and three modes of vibration of various floors. Results from this research are presented in this thesis. Analytical research, performed to obtain information about floor vibration characteristics, is presented and used for the initial design of TMDs and placement of TMDs on a floor. Also, computer models of the floors with TMDs to control one, two, and three modes of vibration were analyzed to obtain further information about changes in floor response and vibration characteristics. This research was performed to provide further insight on the effectiveness of TMDs to control one, two, and three modes of floor vibration and the effects of TMDs on floor vibration characteristics.

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CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

1.1 Introduction

With our increasing understanding of material behavior, and the increasing strength of materials, the optimization of building design continues. One product of the optimization process is lightweight floor systems. The strength of these floor systems has not been compromised; however, in some instances, floor serviceability has been compromised. Over the past few decades, as floors span greater distances and floor slabs decrease in thickness and weight, annoying floor vibrations have gained increasing attention as a serviceability concern.

The Load and Resistance Factor Design Specification for Structural Steel Buildings (1993) states in Chapter L,

vibration shall be considered in designing beams and girders supporting large areas free of partitions or other sources of damping where excessive vibration due to pedestrian traffic or other sources within the building is not acceptable.

It is up to the designer to decide if floor vibrations should be considered in the design of floor slabs and their supporting members. In the design of office buildings, floor vibrations usually are not considered. Office environments usually have fixed and/or movable partitions and other furniture which are considered to provide adequate damping. However, this furniture does not always provide enough damping to keep floor vibrations at an acceptable level. If an in-use floor has annoying levels of floor vibration, several methods can be used to control the vibrations. One particular method of vibration control, which has been used to control floor vibrations and other

structural vibrations, is the use of tuned mass dampers. The use of tuned mass dampers to control annoying floor vibrations is the focus of this research.

1.2 Literature Review

1.2.1 Floor Vibrations - background information

Every floor has certain frequencies at which it vibrates. These frequencies are the floor's natural frequencies of vibration, and are dependent on the mass, stiffness, and damping of the floor system. The first frequency of vibration is the fundamental mode of vibration, and is referred to as the dominant mode of vibration. Many who have studied floor vibrations have focused on the fundamental mode of vibration (e.g. Allen and Murray 1993; Murray 1991); however, there have been instances in which two or more modes of vibration have annoyed the occupants of a building and have had to be dealt with (e.g. Setareh and Hanson 1992a, b, and c; Shope and Murray 1994).

Floor vibrations are usually not annoying to humans unless they fall within a specific range of frequencies. From automobile and aircraft studies it has been determined that frequencies in the range of 5-8 Hz are most annoying to humans (Hanes 1970); however, it should be noted that frequencies outside this range can also be annoying to humans. Within this range, most of the internal organs resonate. If a floor has one or more natural frequencies within or near this range, and the frequencies are a harmonic of a forcing frequency, floor vibration can be very annoying to humans.

Annoying floor vibrations can result from various sources. The vibrations can be steady-state or transient. Steady-state vibrations are, as their name implies, steady or constant over a period of time. Steady-state vibrations are usually caused by in-use equipment or machinery. The

most effective method for controlling steady-state vibrations is the isolation of the equipment or machinery from the main structure, so that it no longer excites floor motion (Lenzen 1966).

Transient vibrations are caused by an impact and die out after a period of time. The amount of time it takes for a vibration to diminish depends on the amount of damping in the system. In 1966 Lenzen reported that damping plays a very important role in floor vibration. The greater the damping in the floor system, the sooner the vibration oscillations are diminished, and the less perceptible the floor vibration to the occupant. Lenzen found that vibrations are definitely perceptible if the amplitude of the vibration after 5 cycles is greater than 0.4 of the initial amplitude. If after 5 cycles of vibration, the amplitude is between 0.2 and 0.4 of the initial amplitude, the vibration is between perceptible and barely perceptible. After 5 cycles if the amplitude is less than 0.2 of the initial amplitude, the vibration is considered barely or not perceptible. He found that vibrations which lasted for less than five cycles are generally not annoying to building occupants.

The most common sources of annoying, transient vibrations on floors are the occupants of a building. The human footfall creates the impact needed to initiate a vibration in a floor. Most human motions across or on a floor (e.g. walking, running, jumping) fall within a frequency range of 2-4 Hz (Ellingwood and Tallin 1984). These forcing frequencies can excite a floor's natural frequencies, and as a result, floor vibrations occur. The human impacting the floor can not be isolated from the floor, so some other method of controlling the floor's reaction to a human footfall needs to be employed.

1.2.2 Methods for Controlling Annoying Vibrations

Methods used to control floor vibrations can be divided into two groups - active and passive controls. Active controls rely on an outside energy source to respond to and control a floor vibration. Hanagan (1994) reported success in controlling floor vibrations with an active control using a proof mass actuator. The largest drawbacks to this control are cost and the need for periodic maintenance. Passive controls rely on a device's reaction to the floor vibration in order to control the floor vibration. Several types of passive control options include installation of full height partitions, installation of damping posts, structural stiffening of framing members, and installation of tuned mass dampers. The use of tuned mass dampers (TMDs) to control annoying floor vibrations has been implemented on several floors, and of all the passive control devices has met with the most success.

1.2.2.1 The Use of Tuned Mass Dampers to Control Annoying Floor Vibrations

A tuned mass damper is a single degree of freedom system. It consists of a mass supported by a spring and a damping element in parallel with one another. The frequency of this single degree of freedom system relies primarily upon the stiffness of the spring and the mass. If the damping element provides considerable mass or stiffness, it will contribute to the frequency; otherwise, it's primary function is to provide damping to the system. As the damping is increased, the number of oscillations that a tuned mass damper will experience after initiation of motion is decreased.

To control annoying floor vibrations, a TMD is tuned to an "optimal" frequency, which is usually 95% - 99% of the frequency of the floor (Bachmann, et al. 1995). When placed at the

point of largest amplitude of the floor vibration, the TMD counteracts the floor motion and diminishes the floor vibration.

Thorton et al. (1990) used TMDs to control annoying floor vibrations on the second and third floors of a school. When the school gym was in use, vibrations were felt in the adjacent second and third floor classrooms and offices. Initial field measurements indicated that the third floor had a fundamental frequency of 4.5 Hz, while the second floor had a fundamental frequency of 5.5 Hz. Later, through detailed dynamic measurements, they found the first four modes of vibration of the third floor to occur at frequencies of 4, 4.6, 5.3, and 6.2 Hz, and determined that the damping ratios ranged from 1.7% for the first mode to 0.9% for the second mode of vibration.

The typical forcing frequencies for synchronous movement in the gym ranged from 2 to 2.5 Hz. Because the forcing frequency range was about one half of the structure's lowest natural frequencies, an "almost-resonant" condition occurred on the second and third floors of the school. Tuned mass dampers were chosen to control the floor vibration problem. Eight TMDs were designed to treat these four modes of vibration, and the TMD units were suspended from the concrete T-beams of the gym floor. No detailed information was given about the TMD design.

Dynamic measurements were recorded with the TMD units in place. The accelerations on the gym floor and second floor were reduced by a factor of two or more. The occupants of the building said the gym floor felt stiffer and there was a dramatic improvement in the control of floor vibrations. The floor vibrations were still perceived when an aerobics class was in session; however, the vibrations are now felt as transient rather than continuous.

Setareh and Hanson (1992a) used TMDs to control the floor vibrations of a balcony in an auditorium. The auditorium is used for live performances and concerts. The structural framing members consist of cantilever members, girder members, and a truss, which is the main cross-

balcony supporting member. The horizontal portion of the concrete balcony varies in thickness from 2.5 in. to 3.5 in., while the vertical risers are 4 in. deep.

At the time of the initial dynamic tests, the balcony floor was bare with no carpeting, seats, or people. From the dynamic tests, they determined the first two modes of vibration to occur at frequencies of 2.9 Hz and 4.2 Hz. The damping ratio for the first mode of vibration was found to range from 1.4% to 1.8%.

Setareh and Hanson then modeled the balcony using the structural analysis program, SAP80 (Wilson and Habibullah 1986). The model had a dead load which included the floor framing, concrete floor, suspended ceiling, and heating and ventilation duct weights. In performing the dynamic analysis, the displacements in the transverse balcony direction were restrained. The dynamic analysis resulted in the first three natural frequencies being 3.17, 4.54, and 5.33 Hz. To predict the behavior of the system under full occupancy, the weight of the seats and people were added to the model. With this modification the dynamic analysis resulted in the first three natural frequencies being 2.55, 3.68, and 4.14 Hz. The first mode frequency of 2.55 Hz falls in the center of the audience participation rock-music-beat frequency.

Since the first two modes of vibration had the highest tendency of being excited by human movement, and only experimental results for the first two modes of vibration were available, tuned mass dampers were used to control only the first two modes of vibration. Before the tuned mass dampers were manufactured, a final measurement of the balcony natural frequencies were taken during a concert. The frequencies were 2.8 and 3.9 Hz for the first two modes of vibration, respectively. These frequencies were relatively close to the predicted frequencies of 2.55 and 3.68 Hz.

Three TMDs weighing 4000 pounds each and two TMDs weighing 2000 pounds each were designed for the first and second modes of vibration, respectively. The corresponding mass ratios for modes one and two were determined to be 0.0307 and 0.01234, respectively. In general, the higher the mass ratio, the more effective the TMDs (Bachmann, et al. 1995). Bachmann et al. (1995) suggest a mass ratio of 0.02 to 0.0667 for the design of TMDs. All five units were placed as close to the balcony edge as possible and at locations of highest vibration amplitudes for the respective modes of vibration.

With the TMDs in place, the recorded acceleration varied from 0.02g to 0.10g, depending on the type of auditorium activity taking place. According to Setareh and Hanson (1992a), 0.10g is considered to be an acceptable acceleration for concert halls. The displacements ranged from 0.1 in. to 0.2 in. The largest accelerations and displacements occurred when a rock concert was in progress. With the TMDs in place, the amplitudes of vibration were reduced to about 22% of the amplitudes that occurred without TMDs in place.

Webster and Vaicaitis (1992) used tuned mass dampers to control the floor vibrations of a longspan, cantilevered, composite floor system. This particular floor has a unique layout. Four columns support a cross-shaped pattern of floor girders and an elliptical ring girder. The floorbeams span between the cross-shaped girders and ring girder, and are cantilevered from the elliptical ring girder out to the rectangular face of the building. The floor is a “reinforced concrete deck-formed slab resting on top of and periodically welded to the floor-beams.”

This floor was used as a dining and dancing area, with the dining taking place near the center of the floor area and the dancing on the cantilevered portion of the structure. Guests complained about the floor vibrations. Accelerations reached 0.07g and displacements were measured to be as high as 0.13 inches. An acceleration of 0.025g to 0.03g is considered to be an

acceptable acceleration for a dining/dancing environment according to Allen (1990). Preliminary vibration tests found the first natural frequency to be 2.3 Hz, which corresponds to the forcing frequency of many dances (NBCC 1985). Using the half power method (Meirovitch 1986), the damping ratio for the first mode of vibration was determined to range from 2.8% to 3.6%.

Due to the large amplitudes of floor motion, light structural damping, and the fact that the floor was primarily excited in its first mode of vibration, they decided to use tuned mass dampers to control the floor vibration. The tuned mass dampers consist of springs and a viscous dashpot in parallel and supporting the mass. Above the mass and connecting the mass to the upper portion of the frame is another viscous dashpot. The TMDs were located at the four corners of the building, as close to the cantilevered ends as possible. The TMD used in one quadrant of the building weighs 18,400 pounds. This results in a mass ratio of about 0.046. With the TMDs in place, the floor accelerations during dance events were reduced by at least 60%. Since the installation of tuned mass dampers, no complaints of floor vibration have been reported.

Shope and Murray (1994) used tuned mass dampers to control several modes of vibration in an office floor. In this case, the damping device was a multi-celled liquid damping device that was not in parallel with the spring portion of the TMD.

Three bays on the second floor of an office building were reported to have annoying floor vibrations. The steel joist-concrete slab floor vibrated at frequencies of 5.13 and 6.5 Hz at the center of a bay and at a frequency of 5.25 Hz at the edge of a bay.

TMDs were installed to control the vibration of the two most significant modes. The TMDs for the two modes of vibration were designed in the same fashion. These tuned mass dampers do not have the springs and dampers in parallel. Instead the spring portion of these TMD units consists of a 12 in. wide steel plate which rests on angle sections. The plate thickness could

be varied between 5/8 in. and 1 in. in order to vary the stiffness. The span of the plate between supports was adjusted in order to change the stiffness in the field. The weights (i.e. mass) and multi-celled liquid dampers are supported by the plate. The damper is no longer in parallel with the spring, but is supported by the spring. This made fine-tuning of the TMD difficult, since changing the number of liquid damping cells also changed the mass of the TMD, and therefore the TMD's frequency.

The TMDs were placed so that a TMD support is located at a point of maximum amplitude for a particular mode of vibration. A total of fourteen TMDs were placed on the three bays. Significant improvement in floor response was shown in the acceleration time histories of the uncontrolled versus controlled floor vibrations. The occupants indicated their satisfaction in the reduction of floor vibrations.

Bell (1994) used a tuned mass damper to control vibrations on a museum floor. This particular portion of the floor is a "bridge slab" which has openings on either side to permit natural light from above to penetrate the floors below. The floor framing consists of two dissimilar length girders and floor beams that span between the girders. No information was given about the floor slab.

Prior to the completion of this museum space, the owner complained of excessive floor vibrations on this portion of the floor. Initial floor measurements indicated a floor vibration of 3.7 Hz, and the damping ratio was determined to be 1%.

The tuned mass damper used to control the floor vibrations weighs 1925 pounds, and this results in a mass ratio of 0.0378. The TMD uses fluid dashpots for damping. No further information was given about the design of the tuned mass damper. The TMD was suspended

below the floor at the center of the span. The installation of the TMD increased the damping by 580%. No complaints associated with floor vibration have been reported by museum patrons.

Each floor, on which TMDs were used to control annoying vibrations, had its own unique response characteristics. Only one of the floors was reported as having more than two modes of vibration that required control, and not many details were given concerning the TMD design or placement. The other floors were reported as having only one or two modes of vibration to control, and more details were given on the design and placement of these TMDs. This indicates that the complexity of controlling annoying floor vibrations increases as the number of modes that require control increases. Table 1.1 summarizes the response of these floors before and after the placement of TMDs, and some of the TMD characteristics.

1.2.2.2 The Use of Tuned Mass Dampers to Control Vibrations on Other Structures

Tuned mass dampers have also been used to control annoying vibrations of footbridges and a diving platform. Each of these structures required that only the fundamental mode of vibration be controlled, and each structure initially had a relatively small damping ratio.

Bachmann and Weber (1995) reported the use of tuned mass dampers on a four span girder footbridge. The bridge is constructed of concrete slab resting on two steel girders. The bridge was reported to have excessive vibration under normal use. Bachmann and Weber measured the fundamental frequency of the footbridge to be 2.46 Hz. The damping ratio in the footbridge was determined to be only 0.2-0.4%.

Two identical tuned mass dampers were designed to control the fundamental mode of vibration. The two dampers were located across from each other, one suspended from each girder, near the center of the longest span of the footbridge. The mass of the TMD is supported by four

Table 1.1 Summary of Data and Results from the Use of TMDs to Control Floor Vibrations

Author(s)/ Date	Floor Description	Mode(s) Controlled	Freq. (Hz)	Initial Damping Ratio	Final Damping Ratio	Damping Device	Mass Ratio	Measure of Improvement	Comments
Thorton, et. al./ 1990 (see note 1)	gym floor & neighboring office/ classroom floors	1	4	0.017		"viscoelastic component"		gym & 2nd floor accel. reduced by a factor of 2	dramatic improvement - vibrations perceived as transient rather than continuous
		2	4.6	0.009					
		3	5.3	$0.009 < \zeta < 0.017$					
		4	6.2	$0.009 < \zeta < 0.017$					
Setareh & Hanson/ 1992a	auditorium balcony	1	2.8	0.016	0.008	dashpot	0.0307	vibrations reduced by approximately 20%	
		2	3.9				0.0123		
					(see note 2)				
Webster & Vaicaitis/ 1992	dining/dance floor	1	2.3	$0.028 < \zeta < 0.031$		"variable energy dissipation dashpots"	0.047	vibrations reduced by at least 60 %	guests no longer complain of vibrations
		(2)	3.75						
Shope & Murray/ 1994	office floor	1	5.1/6.5			liquid- filled bladders			positive response from occupants concerning control of vibrations
		2	5.25						
Bell/ 1994	museum floor with openings to floors below	1	3.7	0.01	0.058	fluid dashpots	0.038	580% increase in damping	no complaints from patrons or owner
					(see note 3)				

- notes:
- 1) The frequency and damping ratios are for the third floor of the school, but the results concern the gym and second floor.
 - 2) The damping ratio was determined by a decay curve.
 - 3) The damping ratio was determined by log decrement.

springs and two damping devices, which consist of a rod submerged in viscous fluid. The optimum damping ratio was determined to be 6.5%. The mass ratio with the combined TMDs is 0.0115.

The TMDs were successful in controlling the vibrations on the bridge. Accelerations due to a person jumping were reduced by a factor of 19, while accelerations due to walking were reduced by a factor of 5, and accelerations due to running by a factor of 2.9.

Bachmann and Weber (1995) also controlled annoying vibrations on a cable-stayed footbridge. Before this bridge was constructed it was planned that a TMD would be installed to control vibrations. Through finite element analysis, the first mode of vibration was determined to be 1.98 Hz, and a TMD was designed to control this frequency of vibration. A damping ratio of 0.5% was assumed in the design.

After construction of the bridge, the fundamental frequency was measured as 2.12 Hz. This was very close to the calculated frequency, so very little modification was required of the TMD. Through measurements, the damping ratio was determined to range from 0.5% to 1%, depending on acceleration amplitudes. The TMD has a design similar to the TMDs designed for the girder footbridge described previously. It has a mass ratio of 0.044. To gain optimum control of the vibrations, the TMD was placed at the location of peak amplitude for the fundamental mode of vibration.

This TMD reduced jumping accelerations on this footbridge by factor of 2.4 and running accelerations by a factor of 1.5. These reductions in acceleration were not as dramatic as the reductions in acceleration for the previous case because this structure initially had a larger damping ratio, and the influence of higher modes of vibration were no longer negligible.

Bachmann and Weber (1995) used a TMD to control the vibration of a two-tiered diving platform. The diving platform is concrete and has a platform at 3 m and another at 5 m. The

diving platform had a fundamental frequency that ranged between 2.6 and 2.7 Hz, and had a damping ratio ranging from 1.5% to 2.0%.

The motion of the platform was oblique with the larger component of motion in the horizontal direction. This required a unique design for the TMD. The TMD moves horizontally, and consists of the mass hanging from steel plates, which act as the springs for the TMD. The damping device is once again a rod submerged in a viscous fluid. The final design resulted in a mass ratio of 0.03. The TMD was placed on the 5 m platform, where the largest amplitudes of vibration were taking place.

The TMD was able to diminish the vibrations on the diving platform. The TMD reduced accelerations due to jumping by a factor of 4.3, and accelerations due to shaking on the railing by a factor of 6.4.

1.2.2.3 The Use of People to Control Annoying Floor Vibrations

Humans are excellent damping mechanisms. Because our bodies are composed mostly of water in individual membranes (e.g. cells), the human body is excellent for absorbing and damping vibrations. A laboratory floor at Virginia Tech was utilized to demonstrate the effectiveness of humans as damping mechanisms.

This laboratory test floor was designed to simulate a typical office floor. The single bay floor measures 15 ft by 25 ft, and has a 3½ in. concrete floor resting on 1.0C metal deck. The floor rests on 16K3 open web joists, which span between roller-supported W14x22 hot-rolled steel girders. The joists span 25 ft, and are placed 30 in. on center.

Acceleration measurements of the bare floor resulted in dominant natural frequencies of 7.375, 9.375, and 16.75 Hz. Using the half power method (Meirovitch 1986), the damping ratio of

the first mode of vibration for the bare floor was determined to be 1.6%, while the damping ratio for the second mode of vibration was determined to be 2.3%.

People were placed on the floor to control the first and second mode of vibration. The first mode of vibration has a bowl shape with the peak displacement occurring at the center of the floor. The second mode of vibration is a torsional mode of vibration. The second mode of vibration peaks at two locations in such a way that while one location is at its absolute maximum displacement, the other is at its absolute minimum displacement. These peaks are located at the midspans of the exterior joists. Figure 1.1 shows the mode shapes for the first and second modes of vibration.

Two people were placed at the center of the floor to control the first mode of vibration. Two people, one at each location of peak amplitude, were placed at opposite edges of the floor to control the second mode of vibration. The combination of two people to control the first mode of vibration resulted in a mass ratio of 0.076. The combination of two people to control the second mode of vibration resulted in a mass ratio of 0.130, which was a relatively large mass ratio. If taken individually, the mass ratio of one human damper for the second mode of vibration was approximately 0.065.

Figure 1.2 shows the floor acceleration response to heel drops at the center of the floor and at the center of an edge joist with and without the people in place. With the four people in place, the damping ratio for the first mode of vibration increased by more than a factor of three. The people had only a slight effect on the floor frequency. The first mode frequency peak moved about 0.25 Hz while the second mode frequency peak didn't move, it only decreased in amplitude. The change in frequency was most likely due to the change in mass of the system.

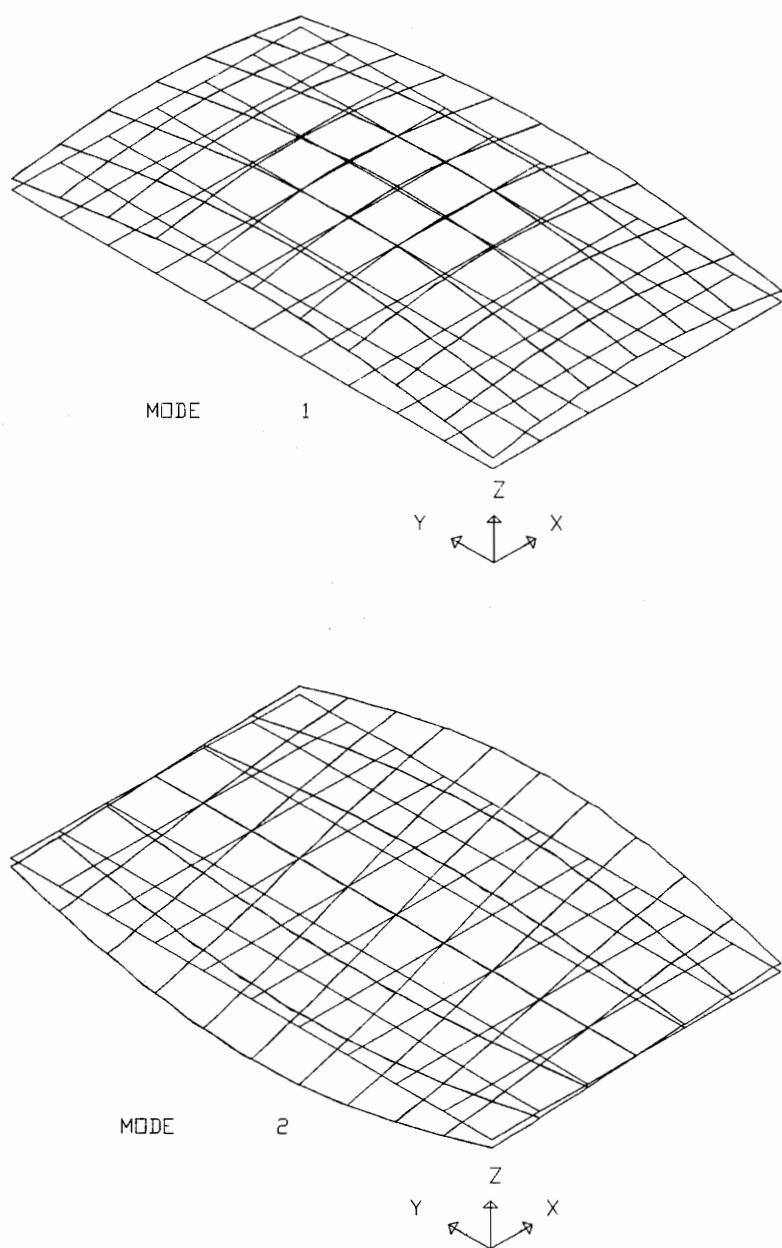
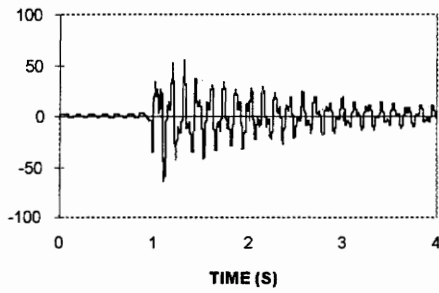


Figure 1.1 Laboratory Floor Mode Shapes One and Two

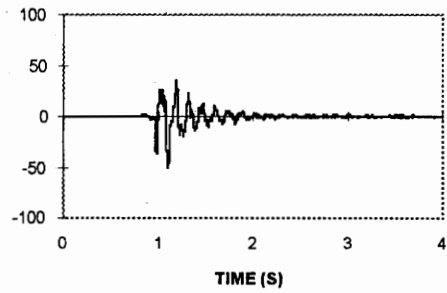
Floor Response Without People

ACCELERATION (IN/S²)

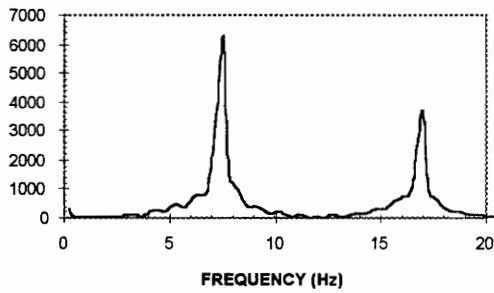


Floor Response With People

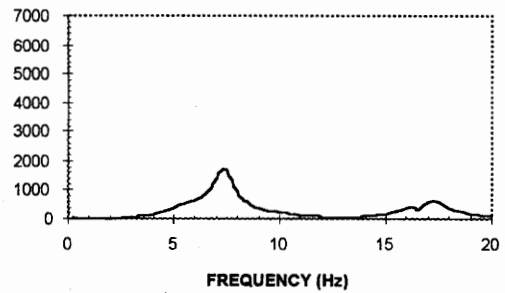
ACCELERATION (IN/S²)



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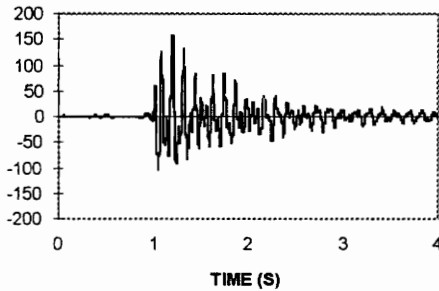


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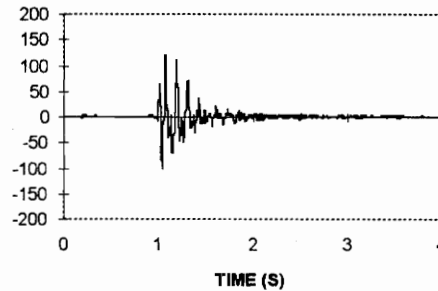


a - Heel drop and response at the center of the floor

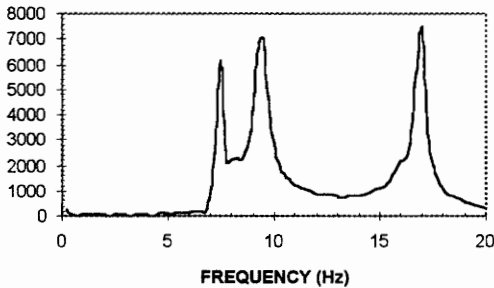
ACCELERATION (IN/S²)



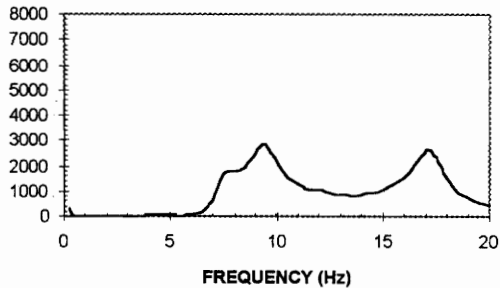
ACCELERATION (IN/S²)



FREQUENCY SPECTRUM AMPLITUDE



FREQUENCY SPECTRUM AMPLITUDE



b - Heel drop at the center of an edge joist and response at the opposite edge joist

Figure 1.2 Acceleration Time Histories and Frequency Transforms

Walking acceleration data was recorded at the center of the floor with a person walking parallel to the joist length along the center of the floor. The root mean square (RMS) of the acceleration data with and without people on the floor was determined. RMS acceleration was reduced by a factor of 1.6 with the people in place. Walking acceleration data was also recorded at the center of one of the edge joists with a person walking perpendicular to the joists along the center of the floor. With people in place, RMS acceleration was reduced by a factor of 2.8. Figure 1.3 shows the walking acceleration traces in the parallel and perpendicular to joist directions, with and without people on the floor.

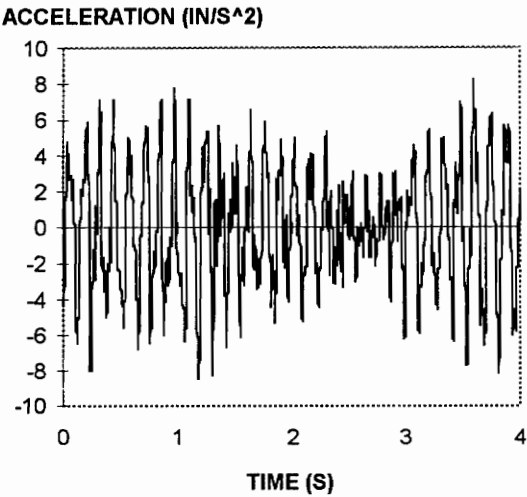
Walking in the perpendicular direction especially excited the second mode of vibration; the larger RMS acceleration reduction factor is most likely due to the larger mass ratio for the second mode of vibration. Even though the acceleration reduction factor may not seem significant, the increase in damping was significant enough so that the floor vibrations were much improved with people at “key” locations on the floor.

1.2.3 Summary

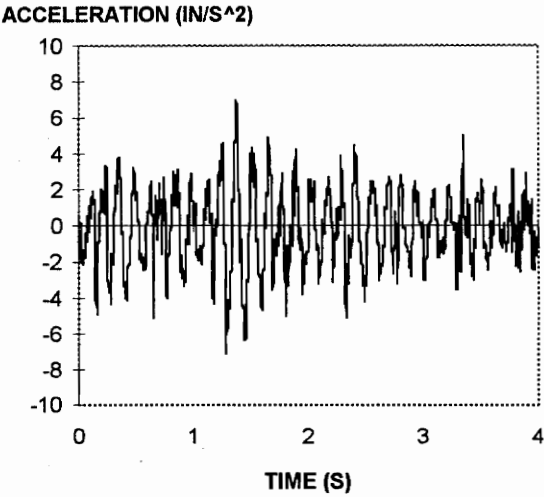
Annoying floor vibrations will continue to be a problem as long as vibration serviceability criteria are not checked by the structural engineer. A number of methods, both active and passive, are available for the control of annoying floor vibrations. A passive control mechanism, such as a tuned mass damper, can be used to diminish floor vibrations, and has been used in the past with some success.

Tuned mass dampers have been used to control vibrations on a variety of structures. When a structure's damping ratio is initially relatively high (i.e. $\geq 3\%$) the success of a TMD is usually minimal. The success of TMDs is also usually minimal when more than one mode of

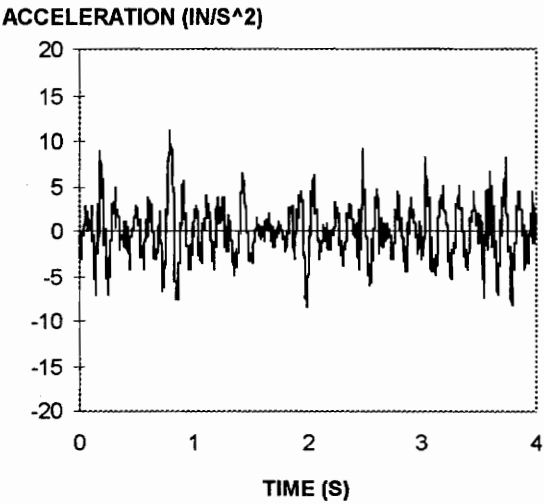
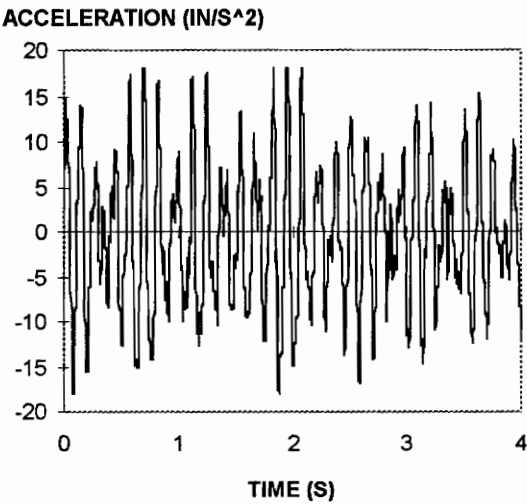
Floor Response Without People



Floor Response With People



a - Walk parallel to joists



b - Walk perpendicular to joists

Figure 1.3 Walking Acceleration Traces With and Without People in Place

vibration needs to be controlled, and the frequencies of those modes of vibration are closely spaced. Bachmann, et al. (1995) and Setareh and Hanson (1992b) have also noted that vibrations on floors with closely spaced natural frequencies are difficult to control using TMDs. The greatest success in the use of TMDs has occurred when the structure initially has a small damping ratio (i.e. $\leq 1\%$) and/or only the structure's fundamental mode of vibration needs to be controlled.

1.3 Research Objectives

The purpose of this research is to perform experimental and analytical studies on the use of tuned mass dampers to control one, two, and three modes of floor vibration. The results of these studies will be presented in Chapters 3, 4, and 5.

Experimental studies will be performed using prototype TMDs on a laboratory floor and several bays of an office floor. Tuned mass damper effectiveness will be determined by performing a visual comparison of acceleration traces and Fast Fourier Transforms (FFTs) of the acceleration data, and by comparing the root mean square (RMS) of acceleration due to a person walking.

Analytical studies will be performed using models of several of the experimental test floors. The analytical results will be compared with the experimental results, and the changes in floor response and vibration characteristics resulting from TMDs in the model will be presented.

CHAPTER 2

PRELIMINARY STEPS FOR THE USE OF TUNED MASS DAMPERS TO CONTROL ANNOYING FLOOR VIBRATIONS

2.1 Determination of Floor Vibration Characteristics

Once a floor vibration problem is identified, the vibration characteristics of the floor need to be studied. To best control a floor's vibration, the dominant and annoying frequencies and modes of vibration need to be determined and controlled as best as possible.

The natural frequencies of a floor can be determined by floor vibration measurements. Vibration measurements allow one to see at which frequencies the natural modes of vibration occur. Tuned mass dampers are designed to control specific frequencies, and because of this the natural frequencies of the floor must be known.

In addition, computer modeling of the floor can be a useful tool. Through an eigenvalue analysis performed using SAP90 v. 6.0 (Wilson and Habibullah 1995), for instance, mode shapes for the natural modes of vibration can be determined. The mode shapes are useful in determining the locations at which tuned mass dampers should be placed to control each mode of vibration. The nodal displacements of each mode shape make up the $\phi^{(i)}$ matrix, where i = a single mode of vibration. Information from the $\phi^{(i)}$ matrix is utilized in determining the equivalent single degree of freedom floor mass for a particular mode of vibration. This effective floor mass is used in determining the TMD-to-floor mass ratio.

2.1.1 Determination of Floor Frequency

To design tuned mass dampers, the floor vibration frequencies which need to be controlled must be determined. These frequencies are the frequencies to which the TMDs will be tuned.

First the areas of a floor with vibration problems must be determined. In most cases, floor vibrations occur in individual bays of a floor (i.e. movement in one bay does not cause considerable vibration in another bay). In that case, vibration measurements can be confined to the individual bay(s) with vibration problems. In some instances floor vibration measurements cannot be confined to an individual bay. Murray (1991) reported the occurrence of floor vibrations that can move from one bay to the next in a wave-like motion transverse to the supporting members. This wave-like motion was found to be prevalent in buildings where the floor was free of fixed partitions, the bays were rectangular, and the supporting members were equally spaced and of the same stiffness throughout the bays. When this wave-like motion occurs, vibration measurements may be taken in one bay while motion is initiated in that bay or in neighboring bays.

To take floor vibration measurements, the proper equipment and a source to initiate the vibration are needed. Heel drop impacts are generally used to initiate floor vibration. A person performs a heel drop by standing on their toes with their knees straight and with the heels of their feet about 2½ in. above the floor. They then relax, allowing the weight of their body to fall freely onto the heels of their feet, resulting in an impact (Murray, 1979). An accelerometer is used to collect floor acceleration data. The data is sent to a spectral analyzer where it is recorded and shown as acceleration over a period of time. The spectral analyzer then takes the acceleration data, which is in the time domain, and performs a Fast Fourier Transform (FFT) to convert the acceleration data to the frequency domain. The plot in the frequency domain shows frequency versus frequency spectrum amplitude. The frequency spectrum amplitude scale is used to measure

the relative change in frequency power of FFTs of acceleration resulting with and without TMDs on a floor. The frequencies at which the frequency spectrum amplitude peaks are the natural frequencies of the floor. Figure 2.1a shows the acceleration of a floor due to a heel drop impact, and Figure 2.1b shows the corresponding frequency transform. When there are several peaks in amplitude, this indicates several natural frequencies of floor vibration.

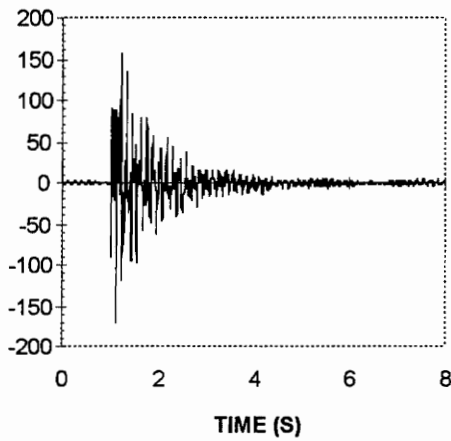
To determine all of the possible annoying modes of vibration, measurements should be taken at several locations on a floor. A floor vibrates differently for each mode of vibration; therefore, floor response may be zero at one location for a particular mode of vibration, while floor response may be at its maximum for another mode of vibration. When floor response for a particular mode is zero at a location, its natural frequency does not appear on a plot in the frequency domain. This needs to be taken into consideration so that all annoying frequencies are identified and TMDs can be designed to control the vibration.

2.1.2 Determination of Floor Damping Ratio

From collected acceleration data, information concerning the damping ratio of the floor can be determined. As stated in Chapter 1, Lenzen (1966) reports that damping plays a very important role in occupant response to floor vibration. The greater the damping in the floor system, the less perceptible the vibration is to the occupant. The damping ratio can be helpful in determining the effectiveness of TMDs by comparing damping before and after placement of TMDs.

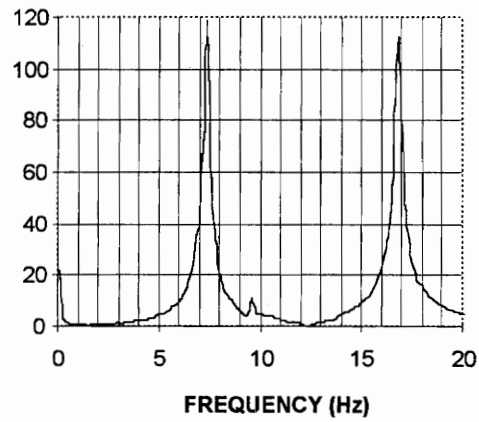
The damping in the floor can be “seen” in the acceleration graph. The greater the damping, the sooner the acceleration decreases. The floor response before and after installation of TMDs can be compared to see the relative increase in damping with TMDs in place. As can be

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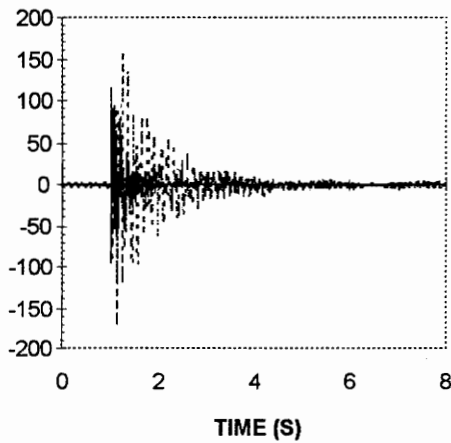
a - Acceleration on a bare floor due to heel drop impact

FREQUENCY SPECTRUM AMPLITUDE

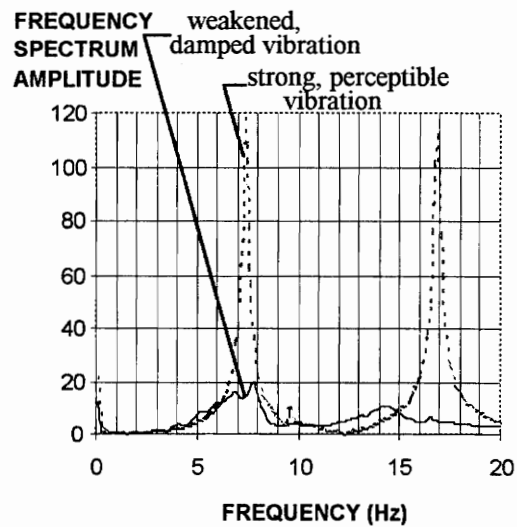


b -Acceleration in the frequency domain

ACCELERATION (IN/S²)



c - Acceleration on floor with and without TMDs due to a heel drop impact



d -Acceleration in the frequency domain with and without TMDs

Figure 2.1 Acceleration Time Histories and Frequency Transforms

seen in Figure 2.1c, the acceleration with TMDs in place diminishes sooner than the acceleration of the floor without TMDs, showing increased damping.

From the plot in the frequency domain, the relative energy of each mode of vibration can be seen. The area under the curve is related to the energy for that particular mode of vibration. Looking to Figure 2.1d, if the peak is relatively sharp with a large amplitude, that mode of vibration is very strong and has very little damping. If the peak is rounded with a relatively small amplitude, that mode of vibration is not very strong and is well damped.

Using the plot in the frequency domain, the floor damping can be determined analytically using the half power method (Meirovitch 1986). This method of calculating the damping ratio of a floor can only be used when there are no TMDs on the floor. To determine the damping ratio, the frequency at the peak amplitude must be determined along with the frequencies on either side of the peak amplitude located at amplitude values $1/\sqrt{2}$ times the peak amplitude value. Figure 2.2 shows a graph in the frequency domain, and the location of the frequencies required to determine the damping ratio. The frequency at the peak amplitude is called f_n , while the frequencies on either side of f_n are called f_1 and f_2 . The graphs in the frequency domain are in terms of cyclic frequency, f . The damping ratio, ζ , can be determined as follows:

$$\zeta = \frac{1}{2} \left(\frac{f_2 - f_1}{f_n} \right) \quad (2.2)$$

The half power method is not very effective when frequencies are closely spaced because energy from one mode of vibration overlaps the energy from a neighboring mode of vibration, distorting the true shape of the individual frequency peaks. *(Different by 20% @ least)*

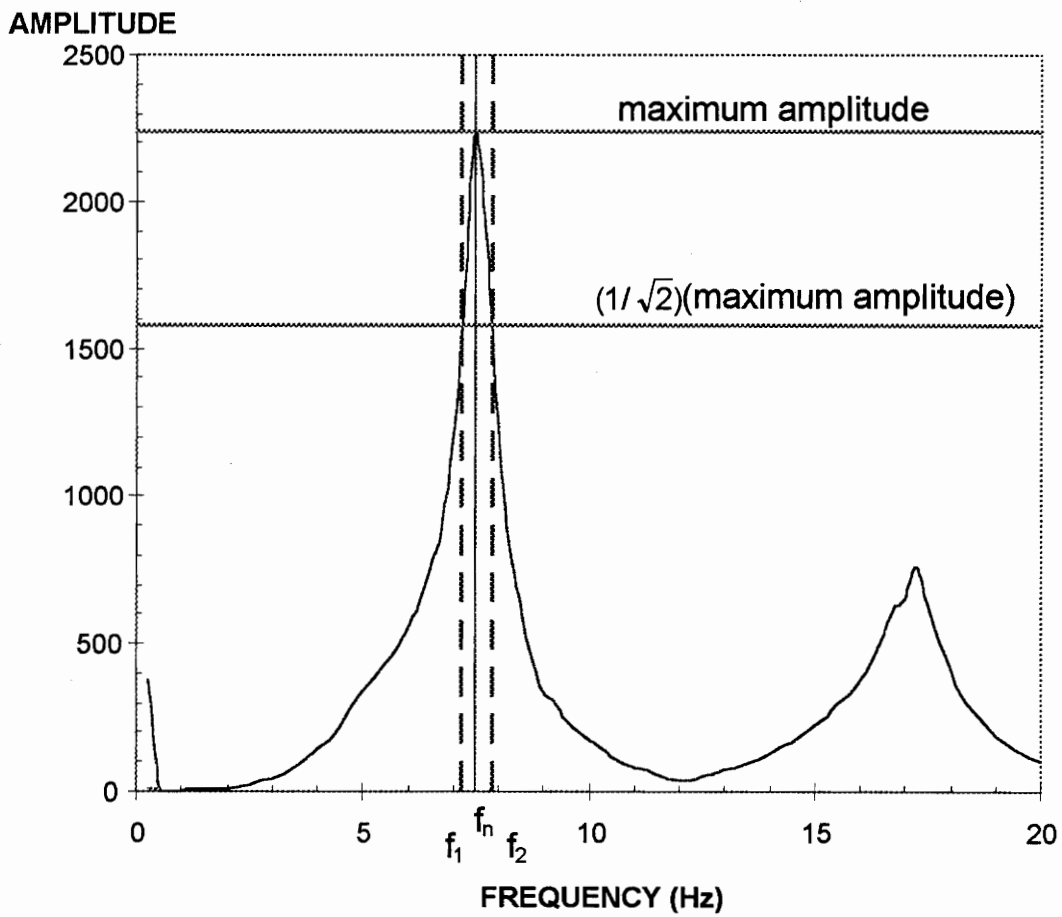


Figure 2.2 Half Power Method (Meirovitch 1986)

2.1.3 Modeling a Floor Area Using a Structural Analysis Program

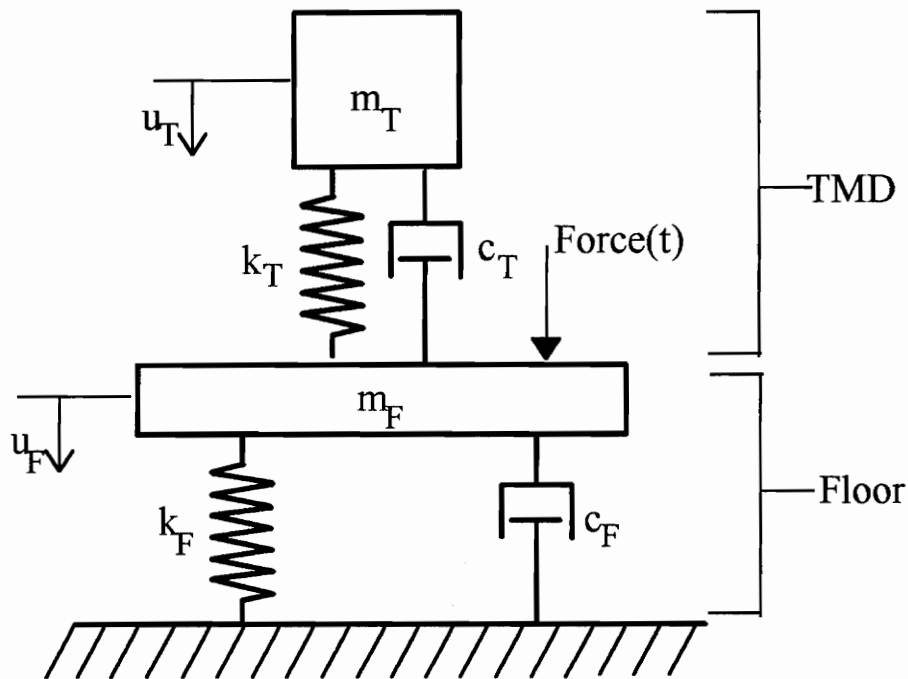
The structural analysis program SAP90 v. 6.0 (Wilson and Habibullah 1995) is used in this study for the analysis of all floor models. The updated version will be commercially available at a date later this year. This program can be used to perform an eigenvalue analysis of the floors with vibration problems. The eigenvalue analysis can output frequencies, mode shapes, and other useful information which can be used in the design of tuned mass dampers.

This program shows how a floor can move in each of its modes of vibration. This information is helpful in determining where to place TMDs for each mode of vibration. To most effectively diminish floor vibration, a TMD should be placed at the location of peak displacement for a particular mode of vibration.

Using results from the eigenvalue analysis, the effective mass for each mode of vibration can be determined. This information is used in the design of a TMD, as will be seen in the following section.

2.2 Designing a Tuned Mass Damper

Once the frequencies and modes of vibration to be controlled have been determined, tuned mass dampers can be designed to control the floor vibration. Theoretically, a tuned mass damper will eliminate all vibration resulting from an impact on the floor. An actual TMD cannot do this; however, a well-designed TMD will control and diminish the floor vibration(s) considerably.



SDOF floor system parameters:

m_F = mass

k_F = stiffness

c_F = damping coefficient

u_F = displacement

f_F = natural frequency

SDOF TMD system parameters:

m_T = mass

k_T = stiffness

c_T = damping coefficient

u_T = displacement

f_T = natural frequency

Figure 2.3 Two Degree of Freedom System: Floor and TMD

2.2.1 Single Degree of Freedom System

A tuned mass damper is a single degree of freedom (SDOF) system. To simplify the design of a tuned mass damper, the floor is modeled as a single degree of freedom system as well, so that together they form a two degree of freedom system as seen in Figure 2.3. A floor has numerous degrees of freedom, and is actually a multiple degree of freedom system. However, modeling the floor as a SDOF system simplifies the floor vibration and constrains the floor vibration to only one mode of vibration. This allows floor vibration to be controlled one natural frequency, or mode of vibration, at a time. A separate TMD is needed to control each annoying mode of vibration of the floor.

2.2.2 Design Equations

The notations used for the equations to follow can be found in Figure 2.3. The equations used for the design of a tuned mass damper assume that the floor and tuned mass damper are both single degree of freedom systems. To model the floor as a SDOF system, the frequency of the floor for the particular mode of vibration must be known, along with the effective, or equivalent SDOF, floor mass, m_F .

Using the $\phi^{(i)}$ matrix, where i = a single mode of vibration, the equivalent SDOF floor mass can be determined for a mode of vibration (Setareh and Hanson 1992b). According to Setareh and Hanson (1992b), the equivalent SDOF floor mass, m_F , is:

$$m_F = \frac{1}{\phi^2} [\phi^{(i)}]^T \mathbf{M} \phi^{(i)} \quad (2.2)$$

where $\bar{\phi}$ equals the modal amplitude of the floor at the location where a TMD is located, and M equals the floor mass matrix. In most cases, $\bar{\phi}$ will equal the maximum modal amplitude for a particular mode of vibration.

For the design of the tuned mass damper, Bachmann, et al. (1995) suggest that the mass ratio be in the range of 1/50 to 1/15, but suggest that the larger the TMD mass, the better. The mass ratio, μ , is defined as:

$$\mu = \frac{m_T}{m_F} \quad (2.3)$$

Knowing the floor mass and a desired mass ratio, an initial estimate for the TMD mass can be made.

The remainder of the equations used to design TMDs are derived from the differential equations of motion of SDOF and two DOF systems. The SDOF equations are simpler and will be introduced first; however, in the design process, equations resulting from the two DOF system are utilized first in determining the required f_{optimal} and ζ_{optimal} . The SDOF equations are utilized later to design the TMD. The parameters, f_{optimal} and ζ_{optimal} , will be defined in a following section.

Single Degree of Freedom Equations

The SDOF equations are derived from the differential equations of motion of a SDOF system (Chopra 1995). The equation of motion for undamped free vibration is as follows:

$$m\ddot{u} + ku = 0 \quad (2.4)$$

The notation, $\ddot{u}$, is the second derivative of the displacement, i.e. the acceleration of the system. The notation, $\dot{u}$, is the first derivative of the displacement, i.e. the velocity of the system, and will be used in a later equation. From Equation 2.4 the following equation, which relates the natural, undamped frequency of a system to the mass and stiffness of a system, can be derived:

$$\omega_n = \sqrt{\frac{k}{m}} \quad (2.5)$$

where

$$\omega = 2\pi f \quad (2.6)$$

Circular frequency, ω , and cyclic frequency, f , are both referred to as frequency, and are used interchangeably.

The equation of motion for viscously damped, free vibration of a SDOF system is as follows:

$$m\ddot{u} + c\dot{u} + ku = 0 \quad (2.7)$$

Dividing this equation by m gives

$$\ddot{u} + 2\zeta\omega_n\dot{u} + \omega_n^2 u = 0 \quad (2.8)$$

where ω_n is as defined in Equation 2.6 and

$$\zeta = \frac{c}{c_{cr}} \quad (2.9)$$

where

$$c_{cr} = 2m\omega_n \quad (2.10)$$

The damping ratio, ζ , is the ratio of the system damping coefficient, c , to the critical damping coefficient, c_{cr} , of the system. The critical damping coefficient is defined as the smallest value of c that stops oscillation completely (Chopra 1995). With damping in the system, the natural frequency of damped vibration, ω_D , can be related to the natural frequency of the system without damping, ω_n , in the following manner:

$$\omega_D = \omega_n \sqrt{1 - \zeta^2} \quad (2.11)$$

If damping in a SDOF system is insignificant, then ω_D can be equated to ω_n .

Using equations derived from the differential equations of motion for a two DOF system, which will be introduced in the next section, f_T and ζ_T will be determined. The value of f_T can be converted to a circular frequency and equated to ω_n for a SDOF system. The value of ζ_T can be converted to ζ for a SDOF system. Knowing the desired mass and ω_n of a TMD, an acceptable damping coefficient, c_T , can be determined using Equations 2.9 and 2.10. This desired damping coefficient can then be utilized in the design of the damping element. The value of ω_D can be determined using Equation 2.11. If damping is significant and ω_D varies greatly from ω_n , ω_D can replace ω_n in Equation 2.6, and the desired TMD stiffness, k_T , can be determined.

Once the desired parameters are determined, the TMD can be designed and built. If the actual TMD parameters differ from the desired parameters, Equations 2.6 and 2.11 can be utilized to “tune” the TMD. For example, if the stiffness and damping coefficient are constant, the mass can be varied so that the TMD frequency will equal f_T . To control floor vibrations it is very important that the TMD be “tuned” to the proper frequency. Slight variations in frequency can be critical, whereas slight variations in damping are not as critical (Bachmann and Weber 1995).

Two Degree of Freedom Equations

The equations of motion for a two DOF system are as follows (Bachmann et al. 1995):

$$m_F \ddot{u}_F + c_F \dot{u}_F - c_T (\dot{u}_T - \dot{u}_F) + k_F u_F - k_T (u_T - u_F) = F \cos \Omega t \quad (2.12)$$

$$m_T \ddot{u}_T + c_T (\dot{u}_T - \dot{u}_F) + k_T (u_T - u_F) = 0 \quad (2.13)$$

where $F \cos(\Omega t)$ represents harmonic excitation of the floor.

From these equations an equation for the “optimum” frequency of the TMD can be derived (Bachmann et al. 1995; Den Hartog 1934):

$$f_{\text{optimal}} = \left(\frac{1}{1 + \mu} \right) f_F \quad (2.14)$$

This frequency, f_{optimal} , is used in the SDOF equations to design the TMD. The value of f_{optimal} usually ranges from 95% to 99% of f_F (Bachmann et al. 1995). When a TMD is “tuned” to f_{optimal} and placed on a floor vibrating at a frequency of f_F , the plot in the frequency domain will no longer have one peak at f_F . With the TMD in place, the plot in the frequency domain will show two peaks, one slightly lower and one slightly higher than f_F . The TMD, one of the two degrees of freedom, is tuned to the optimal frequency and will be slightly lower than f_F . The frequency of the floor, the second degree of freedom, becomes slightly higher than f_F . This increase in floor frequency occurs because the effective mass of the floor decreases with a properly tuned TMD in place. Ideally, a TMD counteracts the motion of the floor, and this should result in a smaller displacement of floor mass. A smaller displaced floor mass results in a smaller effective floor mass, which in turn will cause an increase in floor frequency. Now the plot in the frequency domain will show a split with the first frequency peak corresponding to the TMD frequency and the second frequency peak corresponding to the floor frequency.

Optimal damping of the TMD can be determined by the following equation (Bachmann et al. 1995; Den Hartog 1934):

$$\zeta_{\text{optimal}} = \sqrt{\frac{3\mu}{8(1 + \mu)^3}} \quad (2.15)$$

This equation is only valid for floors with no damping; however, Bachmann and Weber (1995) found that Equations 2.14 and 2.15 are sufficiently accurate for lightly damped structures.

Concerning the use of TMDs, Bachmann, et al. (1995) states that a TMD is more effective the smaller the damping of the floor system, and that TMDs do not work satisfactorily on a floor

with closely spaced natural frequencies. Setareh and Hanson (1992b) state that if the modes which create annoying vibration are within 20% of each other, the use of a tuned mass damper can move the locations of the peak amplitudes (i.e. change the frequencies of the other modes of vibration). These observations will be discussed further in later chapters.

2.2.3 Tuned Mass Damper Design

The tuned mass dampers used in this research were designed by the 3M Company, St. Paul, Minnesota. Figure 2.4 shows an elevation view of the prototype TMD. The TMD stands approximately 3 ft tall, and is 2 ft wide by 2 ft deep. As can be seen, the TMD consists of an outer frame which rests on the floor, connecting elements, and an inner frame. Four springs and a damping element, which were designed by the 3M Company, connect the outer frame to the inner, mass carrying frame. The inner frame can hold a number of 10 lb steel plates which provide the mass for the TMD. Three of these prototype TMDs were provided and used in the research, which is described in following chapters.

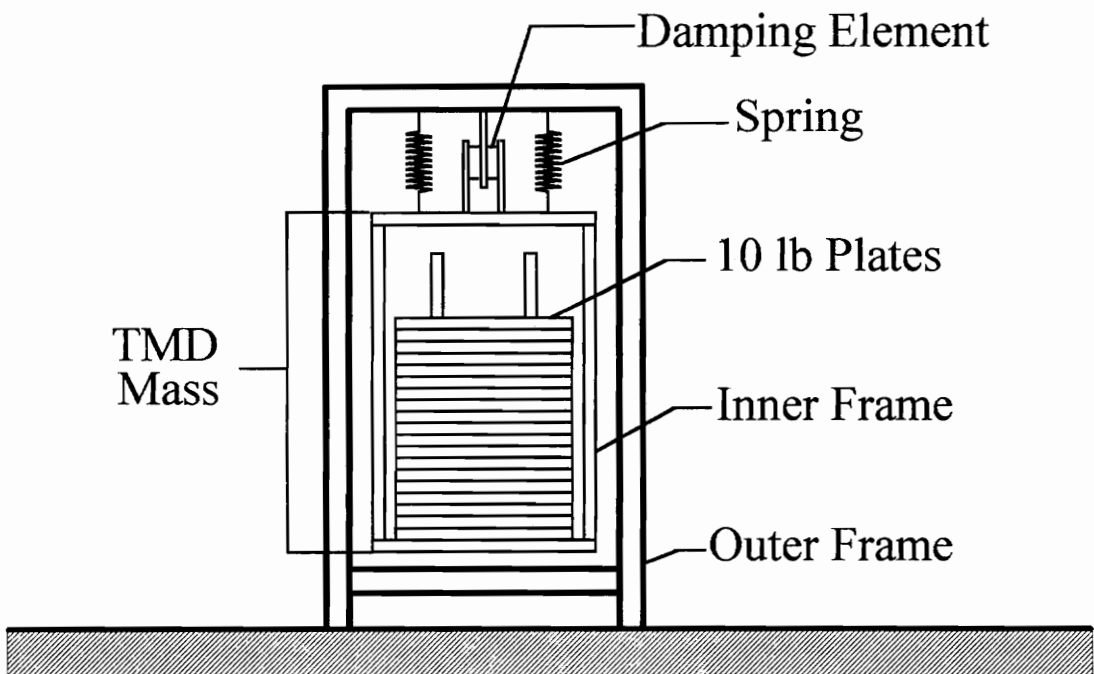


Figure 2.4 Tuned Mass Damper

CHAPTER 3

EXPERIMENTAL USE OF TUNED MASS DAMPERS ON A LABORATORY FLOOR

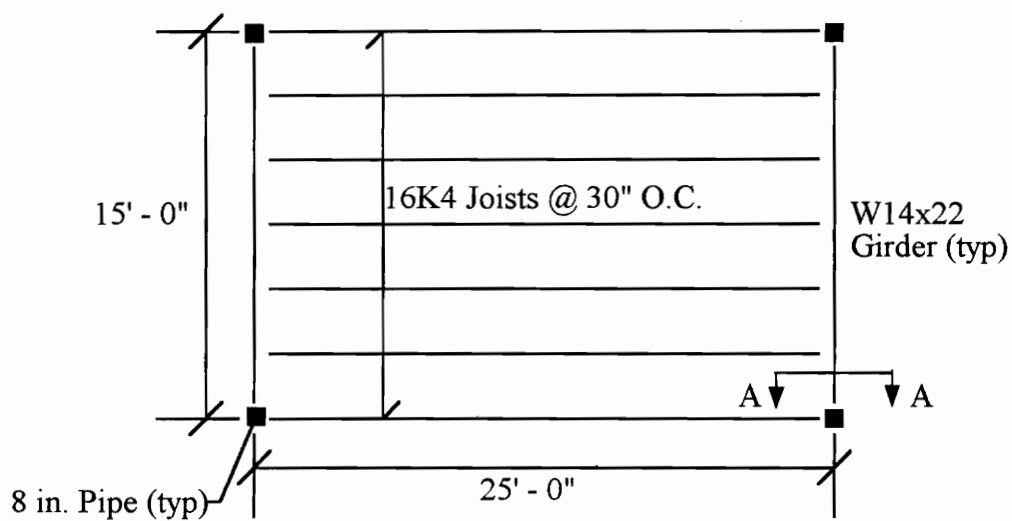
3.1 Description of Laboratory Test Floor

A laboratory test floor at the Virginia Tech Structures Laboratory was utilized for initial testing of the performance of the prototype tuned mass dampers. This floor was designed to simulate floor vibrations of light-weight office floors. The following sections will describe the floor's physical and vibrational characteristics.

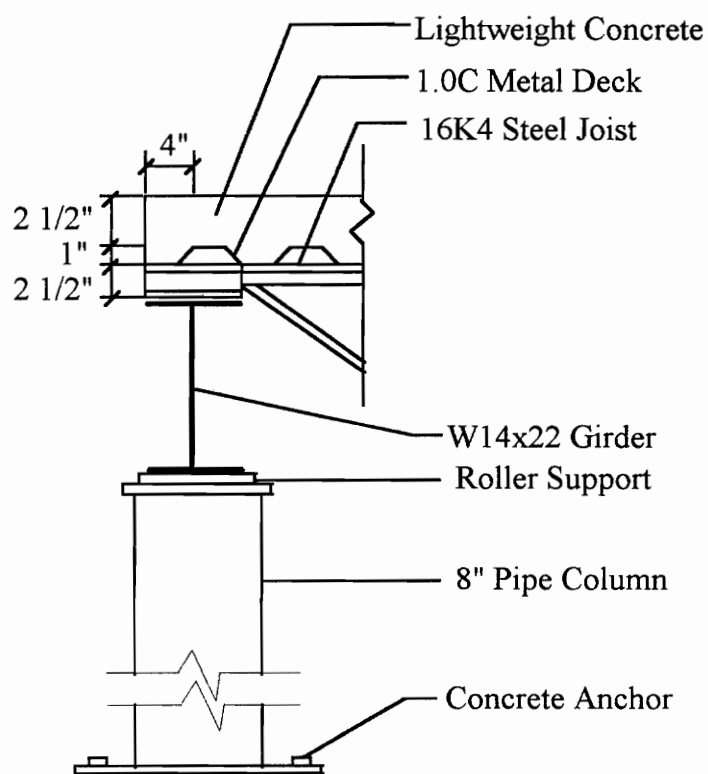
3.1.1 Physical Characteristics

The laboratory floor is a single bay supported at four corners by 8 in. diameter steel pipe. The floor measures 25 ft by 15 ft with W14x22 girders spanning 15 ft. The 16K4 joists span 25 ft between the girders and are spaced 30 in. on center. The lightweight concrete slab, which is supported by 1.0C metal deck, has a total thickness of 3½ in. Figure 3.1 shows a plan and section view of the floor.

Hanagan (1994) designed and built the laboratory floor. Hanagan determined actual member and material properties, and detailed this in her dissertation. From load tests of the joists, the actual moment of inertia was determined to be 80.1 in⁴. This moment of inertia is used in all calculations of this study. The average concrete weight was determined to be 115.7 pcf, and the concrete's average strength to be 4100 psi at 10 days. Through static load tests, Hanagan



Plan - Laboratory Floor



Section A-A

Figure 3.1 Plan and Section of the Laboratory Test Floor

determined that the girders acted non-compositely. This information was utilized when modeling the floor for the computer structural analysis program, SAP90 v. 6.0 (Wilson and Habibullah 1995).

3.1.2 Vibration Characteristics

Vibration characteristics of the floor were determined by floor acceleration measurements, and by results from finite element analysis using the computer program, SAP90.

3.1.2.1 Floor Frequency

Floor frequency was measured at several locations on the floor. At the center of the floor, the fundamental mode of vibration dominated at a frequency of 7.375 Hz. At the center of an exterior joist, the second mode of vibration dominated at a frequency of 9.375 Hz. A third mode of vibration occurred at a frequency of 16.75 Hz.

The 16.75 Hz frequency is above the frequency range that is considered annoying to humans; however, since the amplitude of the 16.75 Hz mode of vibration is relatively strong as compared with the first mode of vibration, it was decided that a TMD would be designed to control this mode of vibration, as well as the first and second modes of vibration. It was also desired to determine whether two or three modes of vibration could be effectively controlled by TMDs.

3.1.2.2 Floor Damping Ratio

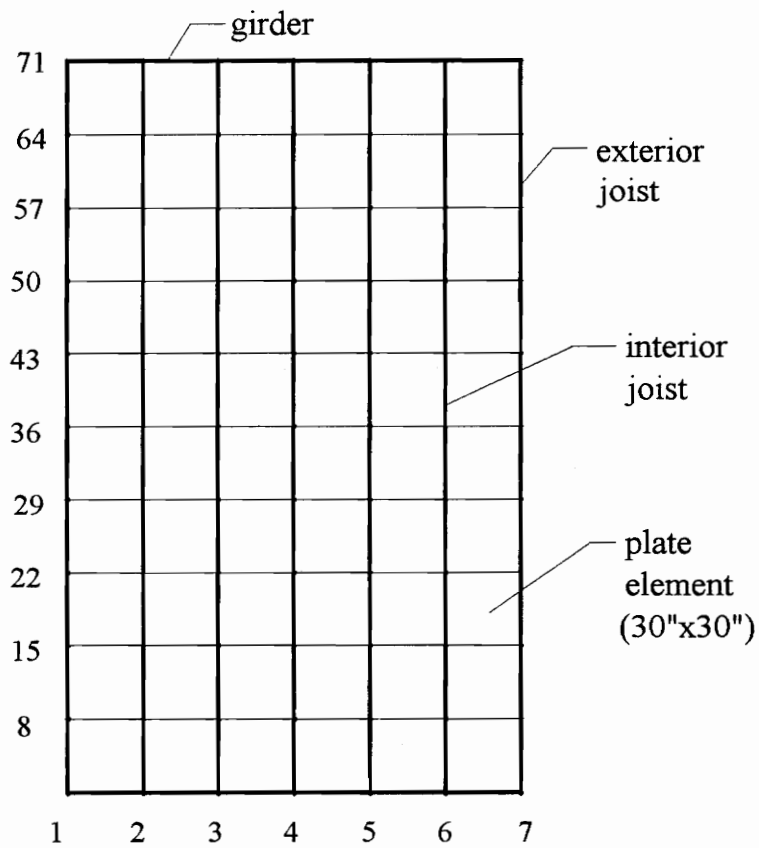
Heel drops were performed on the laboratory floor before placement of the TMDs. FFTs of the acceleration data from the heel drops on the bare floor provided plots in the frequency domain. With information from the frequency domain plots, Equation 2.2 was used to determine the damping ratio of the floor.

With the heel drop located at the center of the floor and the acceleration measured at the center of the floor, the damping ratio was calculated to be 1.6% for the first mode of vibration. The damping contributed by the person performing the heel drop was considered negligible. With the heel drop performed and acceleration measured at the center of an exterior joist, the damping ratio for the second mode of vibration was determined to be 2.3%. From the same acceleration data, the damping ratio for the third mode of vibration was determined to be 1.3%.

3.1.2.3 Other Vibration Characteristics Determined Using SAP90

The structural analysis program, SAP90, was used to determine the mode shapes and the effective SDOF system mass for the modes of vibration. The SAP90 model used frame elements to model the joists and girders, and plate elements to model the concrete floor. All of the model elements were placed in a single plane. Appendix A contains calculations performed to determine the section properties used in the model. Plate elements have dimensions of 30 in. by 30 in. This resulted in 77 spatial coordinates, which gives the model the appearance of a grid. Details of the model can be seen in Figure 3.2. The input for the model can be found in Appendix B.

Hanagan (1994) reported vertical motion of the corner supports of the laboratory floor. She determined that in order to have a more realistic model of the floor, the corners of the floor needed to have vertical spring supports. Spring stiffness was adjusted so that the frequency of the first mode of vibration was approximately 7.3 Hz. With vertical motion restrained at the corners of the model, the first natural frequency was 8.47 Hz. With vertical springs added to the support, the first natural frequency was 7.35 Hz, which compares well with the actual measured frequency of 7.375 Hz.



Frame Elements	I_v (in ⁴)	Area (in ²)	Weight (lb/in ³)	E (psi)
Girder	199	6.49	0.2825	29E10 ⁶
Exterior Joist	166	1.218	0.4789	29E10 ⁶
Interior Joist	171.8	1.218	0.4789	29E10 ⁶

Plate Elements	Bending Thickness (in)	Weight (lb/in ³)	E (psi)
	3	0.08332296	2.63E10 ⁶

Figure 3.2 SAP90 Model of the Laboratory Floor

mode shapes

An eigenvalue analysis of the floor using SAP90 resulted in the natural frequencies of the floor and the corresponding mode shapes. The first six mode shapes and the corresponding natural frequencies are shown in Figure 3.3. The three measured frequencies correlated well with the SAP90 frequencies for the first, second, and fourth modes of vibration. It should be noted that when the SAP90 model had the four corners restrained, the shapes for modes 3 and 4 were reversed (i.e. Figure 3.3 mode 4 became the mode 3 shape, and Figure 3.3 mode 3 became the mode 4 shape). The reason for the vertical corner springs in the model was to gain a better reflection of actual conditions and match the first natural frequency with the measured first frequency of the floor. The mode shapes corresponding with measured frequencies were verified by floor vibration measurements. The frequencies of 7.375 Hz and 9.375 Hz correspond with Figure 3.3 mode 1 shape and mode 2 shape, respectively. The frequency of 16.75 Hz corresponds with the mode 4 shape of Figure 3.3.

Measuring acceleration at several locations on the floor, and looking at FFTs of acceleration traces, gave an indication of which mode shapes correspond to each frequency. When acceleration measurements were recorded at a mode shape node (i.e. no vertical displacement), a FFT of the data showed a small or zero peak amplitude at the frequency corresponding with that mode of vibration. Where the mode 4 shape has nodes, FFTs of actual acceleration at those locations showed little or no peak amplitude at a frequency of 16.75 Hz. When acceleration measurements were recorded at locations of peak displacement for a mode of vibration, a FFT of the data resulted in the largest peak amplitude for the frequency corresponding with the particular mode of vibration. At locations of peak displacement for the mode 4 shape, the frequency amplitude for 16.75 Hz was at its peak. Other floor acceleration measurements did not indicate a

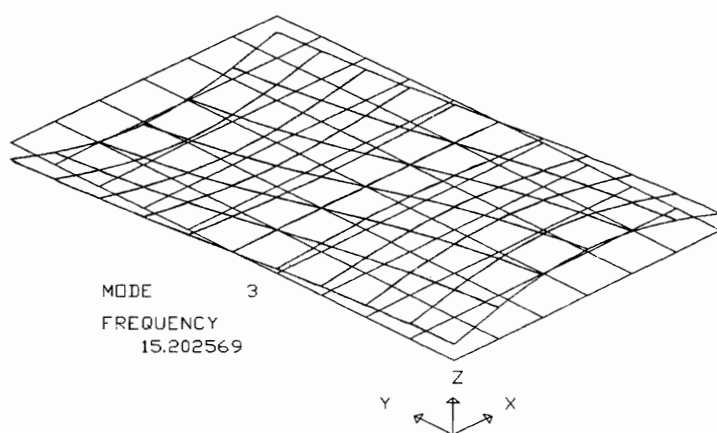
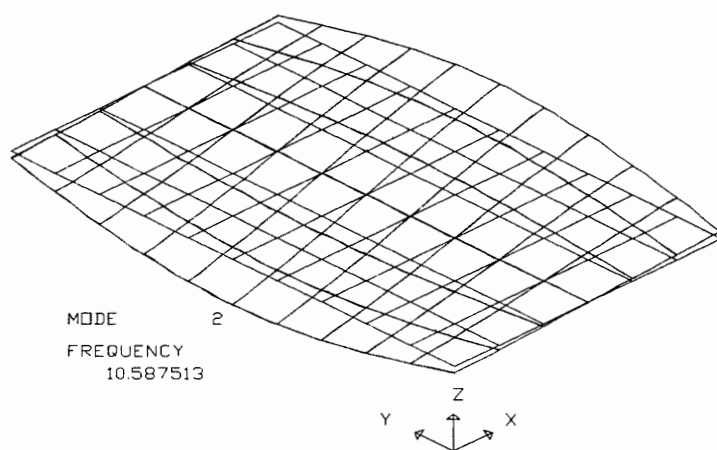
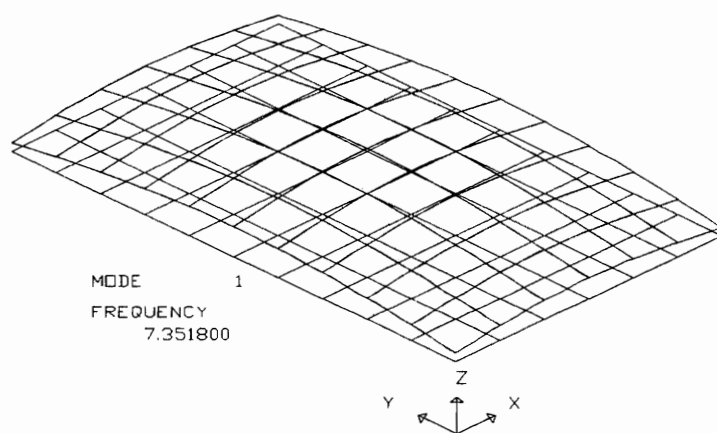


Figure 3.3 First Six Mode Shapes From SAP90 Analysis of the Laboratory Floor

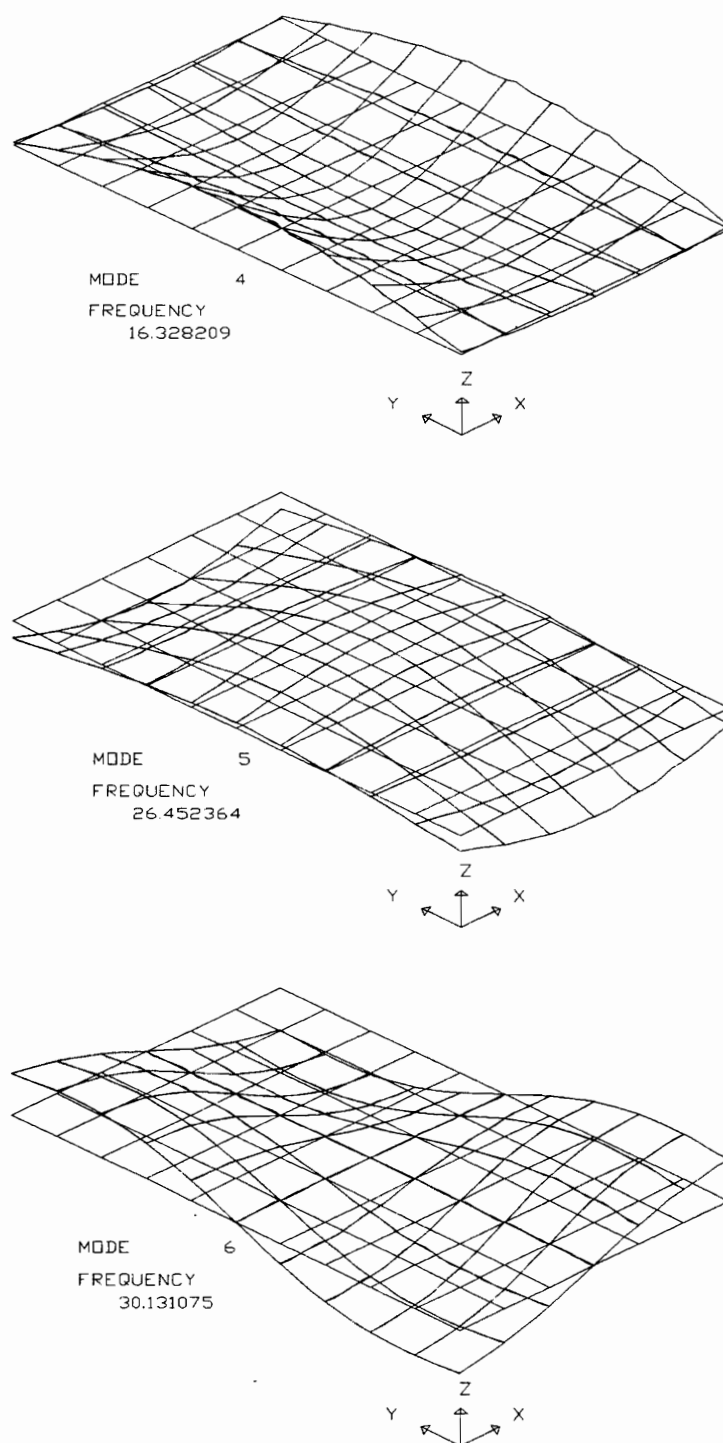


Figure 3.3 First Six Mode Shapes of the Laboratory Floor, continued

strongly perceptible frequency that corresponded with the mode 3 shape of Figure 3.3. From this point forward, the first three modes of vibration will refer to the frequencies of 7.375, 9.375, and 16.75 Hz and their respective vibrational mode shapes 1, 2, and 4 from Figure 3.3.

3.2 Design of Tuned Mass Dampers

Three TMDs, one for each of the first three modes of vibration, were designed by the 3M Company. Using the design equations given in Chapter 2, the required mass, stiffness, and damping coefficient were determined. Appendix C contains calculations of TMD1 properties for the laboratory floor. These calculations are similar for each initial TMD design. Table 3.1 summarizes the input and results of the required TMD properties for the first three modes of vibration.

3.3 Installation of Tuned Mass Dampers

The tuned mass dampers were located at peaks of maximum displacement for their respective modes of vibration. TMD1 was located at the center of the floor to control the first mode of vibration. TMD2, to control the second mode of vibration, was placed at the center of one of the edge joists, and TMD3, to control the third mode of vibration, was placed at the center of the opposite edge joist.

To determine changes in floor vibration frequency and behavior using a TMD, heel drops were performed on the floor before and after a TMD was in place. The force of the heel drop was considered constant, and acceleration and frequency graphs were compared in order to determine if a TMD was optimally tuned for a particular floor frequency.

TMD1, the first TMD to be placed on the floor, was initially placed on a stiff surface and tuned to the proper frequency before placing it on the floor. To tune the TMD, stiffness and

Table 3.1 Laboratory Floor and Initial TMD Parameters

Mode	1	2	3	Remarks
f (Hz)	7.375	9.375	16.75	measured
w_F (lb)	6398	2641	1809	analysis via SAP90
w_T (lb)	365	235	145	chosen by 3M - gives acceptable μ
μ	0.0570	0.0890	0.0802	Equation 2.3
f_{optimal} (Hz)	6.98	8.61	15.51	Equation 2.14
ζ_{optimal} (%)	0.1346	0.1607	0.1544	Equation 2.15
f_D (Hz)	6.91	8.50	15.32	Equation 2.11
k (lb/in)	1782	1733	3477	Eqns. 2.11 and 2.6
c	11.14	10.58	11.29	Eqns. 2.9 and 2.10

Table 3.2 Laboratory Floor and Final TMD Parameters

	Tuned Mass Dampers				Remarks
	TMD1	TMD2	TMD3	TMD1a	
f(Hz)	7.375	9.375	16.75	7.375	
w_F (lb)	6398	2641	1809	6398	
w_T (lb)	365	315	165	445	(plates x 10)+75lb
plates	29	24	9	37	
μ	0.0570	0.1193	0.0912	0.0696	Equation 2.3
ks (lb/in)	1164	1892	2916	1164	
kd (lb/in)	414	885	1496	414	(G'A)/t
ks+kd = k_T	1578	2777	4412	1578	
f_{nTMD} (Hz)	6.50	9.29	16.18	5.89	Equation 2.5
c_T	10.13	15.16	14.72	11.18	k _d /ω _{nTMD}
ζ_T (%)	0.1312	0.1593	0.1695	0.1312	Eqns. 2.8 and 2.9
f_{DTMD} (Hz)	6.45	9.17	15.94	5.84	Equation 2.10
G' (psi)	46	59	68	46	
eta	1	1	1	1	
t (in)	0.5	0.5	0.15	0.5	
A (in^2)	4.5	7.5	3.3	4.5	

note: any property in bold was measured or constant

damping remained unchanged, but the mass was changed by adding and removing 10 lb steel plates. After initial tuning, TMD1 was placed on the floor.

Floor vibration for the first mode of vibration was considered “optimally” controlled when the frequency spectrum amplitude of the fundamental frequency was decreased considerably from its initial value and the frequency was split as discussed in Section 2.2.2. A decrease in the frequency spectrum amplitude indicates an increase in damping for the particular mode of vibration. A frequency split generally indicates that the TMD is “optimally tuned”. In some instances the frequency split does not occur due to the amount of damping in the system; however, the TMD may still be “optimally tuned” in this instance. From Figure 3.4b the split of the first mode frequency, and the considerable decrease in frequency spectrum amplitude, indicated that TMD1 was “optimally tuned”.

After tuning TMD1, TMD3 was initially tuned and placed at the center of one of the edge joists. TMD1 remained on the floor while TMD3 was tuned. Once again, mass plates were added or removed as necessary to tune TMD3 to an “optimum frequency”. Finally, TMD2 was placed on the center of the opposite edge joist and “optimally tuned”. The final parameters of all three TMDs can be found in Table 3.2.

Once all of the TMDs were in place and control of the floor vibrations was considered satisfactory, the mass of TMD1 was altered to see if better control of the floor vibrations could be gained. The frequency spectrum amplitude at first mode frequency and the floor accelerations were decreased when 8 additional 10 lb plates were added to TMD1. With 8 additional plates, the TMD, which will now be referred to as TMD1a, was no longer “optimally tuned” as an individual TMD (i.e. with only TMD1a on the floor it was not optimally tuned). The increased mass did improve the vibration control. Table 3.2 also lists the final parameters of this TMD.

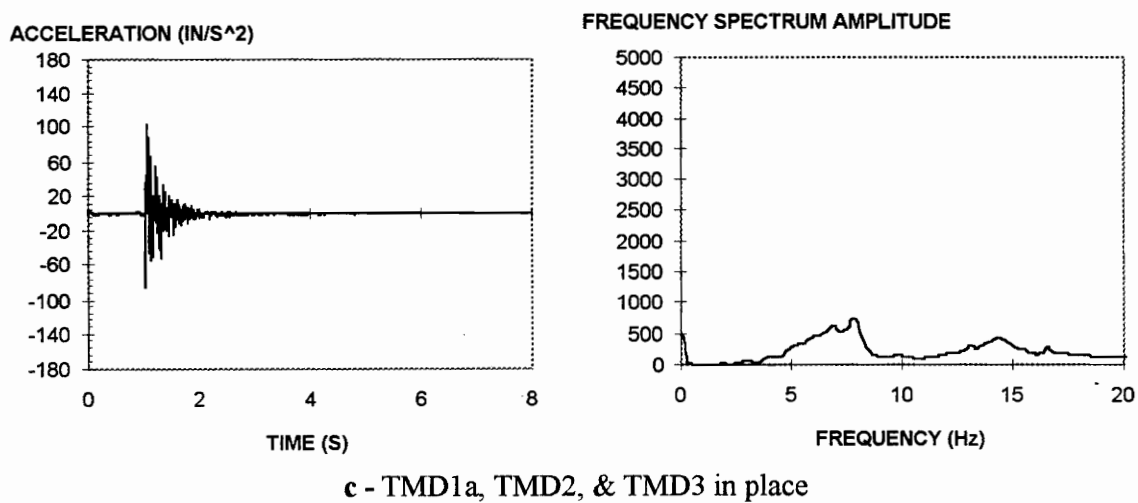
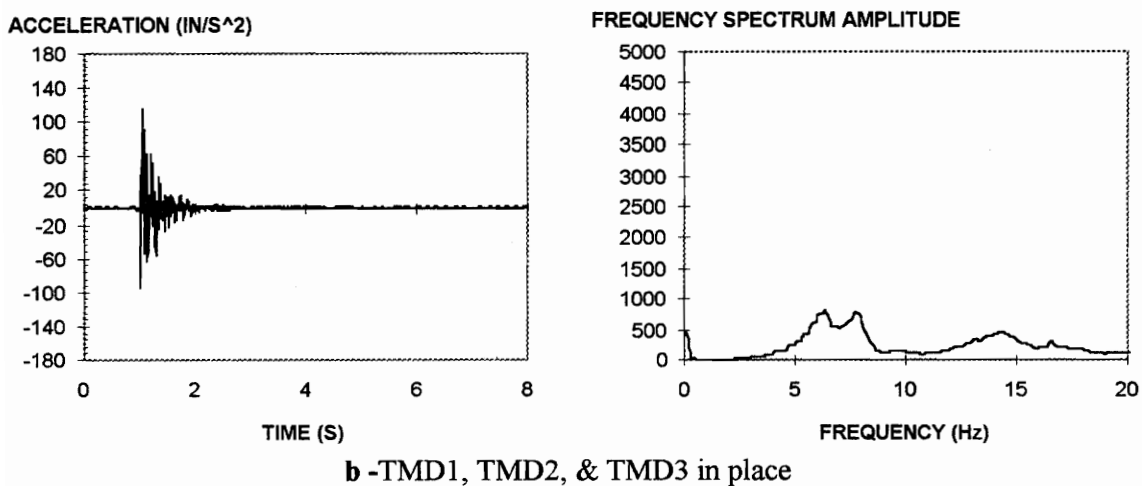
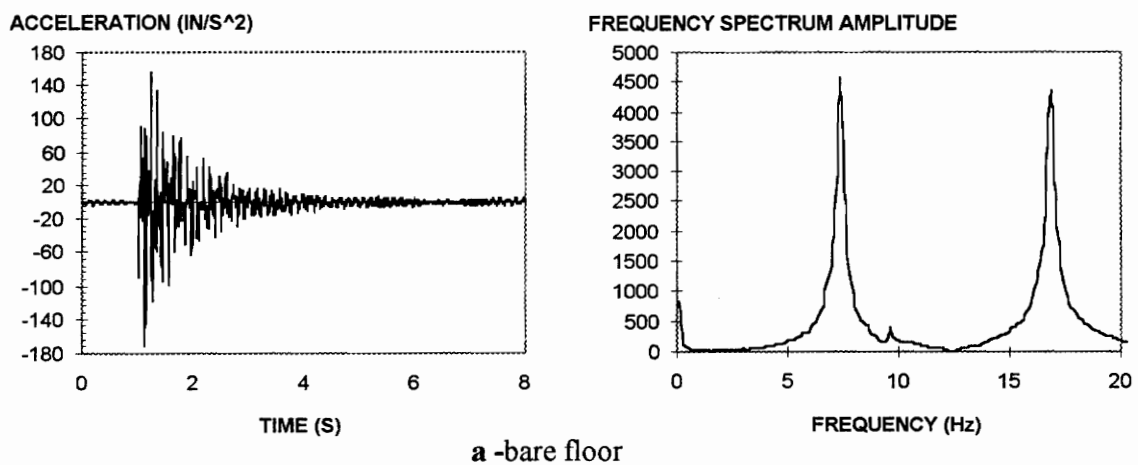


Figure 3.4 Heel Drop Performed and Acceleration Measured at the Center of the Laboratory Floor

3.4 Evaluation of Tuned Mass Damper Effectiveness

To evaluate TMD effectiveness, acceleration data from heel drops and a person walking parallel or perpendicular to the joists were compared. Frequency plots resulting from heel drops reflected the decrease in amplitude and any frequency splits that might have occurred.

3.4.1 Heel Drop Excitation

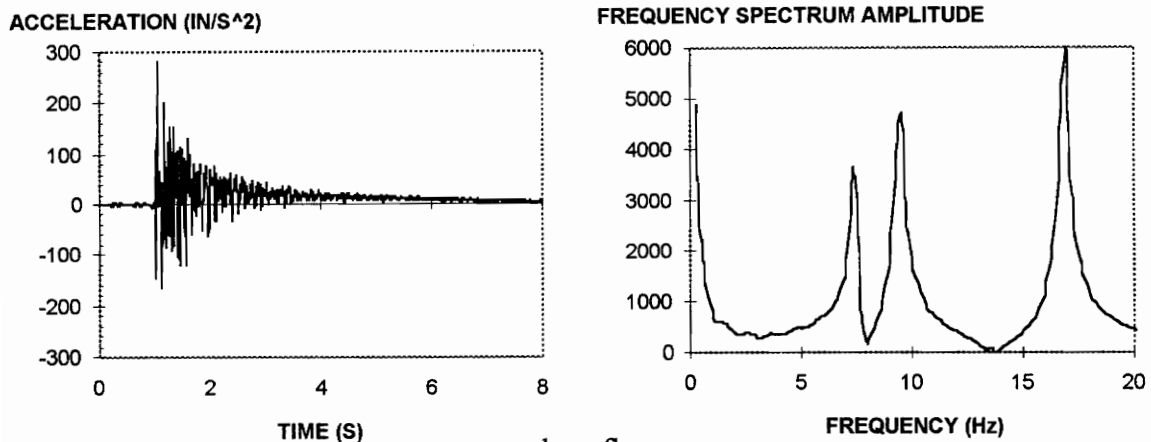
In Figure 3.4 the first and third modes of vibration were excited by a heel drop impact. In Figure 3.5 the first three modes of vibration were excited; however, the second and third frequencies were of major concern when looking at these graphs.

With TMD1, TMD2, and TMD3 in place, the increase in damping and control of the floor vibrations can be seen from Figure 3.4a and b, and Figure 3.5a and b. In Figure 3.4 the frequency split can be seen for the first frequency; indicating optimum control of that mode of vibration.

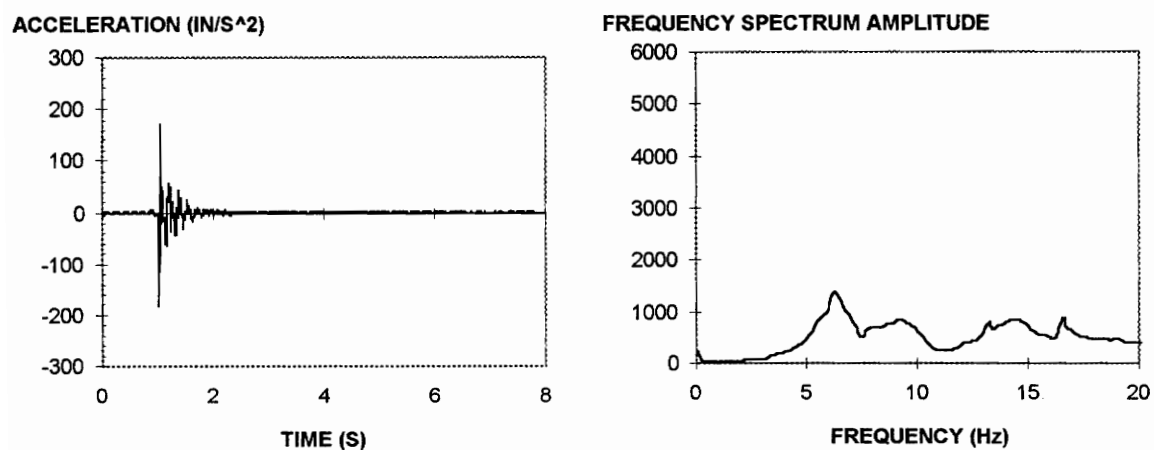
With additional mass added to TMD1, i.e. TMD1a, improved control of the floor vibrations appears to have occurred. This was not apparent from the heel drop acceleration data or frequency plots of Figures 3.4c and 3.5c; however, walking acceleration data reflected improved control of the floor vibrations with the added mass of TMD1a.

3.4.2 Walking Excitation

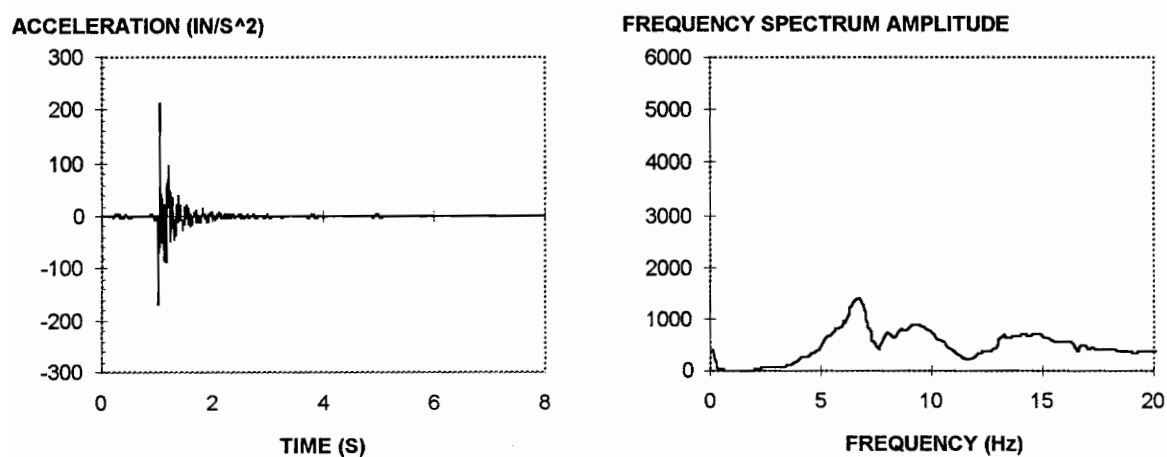
Figure 3.6 shows walking acceleration plots of the floor with and without the TMDs in place. The root mean square (RMS) of the data over the total time interval was determined and was used to compare the change in overall acceleration from one floor condition to another. The



a - bare floor



b - TMD1, TMD2, & TMD3 in place



c - TMD1a, TMD2, & TMD3 in place

Figure 3.5 Heel Drop Performed and Acceleration Measured at the Center of an Edge Joist on the Laboratory Floor

RMS of acceleration is the square root of the variance of the acceleration (Inman 1994), and can be defined as:

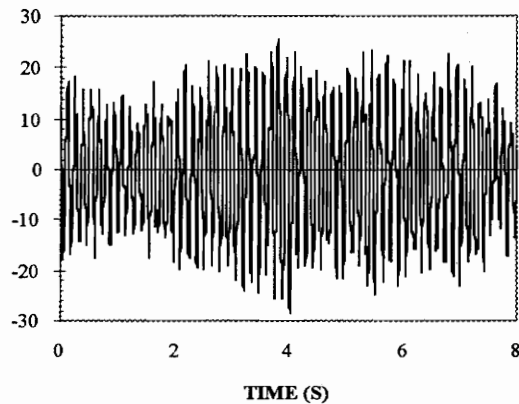
$$\text{RMS Acceleration} = \sqrt{\frac{\sum_{i=1}^N \ddot{U}_i^2}{N}} \quad (3.1)$$

where $\ddot{U}$ is acceleration and N equals the number of data points recorded over a time interval. Using RMS values allows for a better comparison of the data. Instead of looking at just one acceleration peak over the time interval, all of the acceleration peaks over the entire time interval are considered when performing a RMS calculation. Floor accelerations due to walking were measured when a person walked parallel to the joists along the center of the floor and when a person walked perpendicular to the joists along the center of the floor.

Table 3.3 summarizes the RMS acceleration results. From Table 3.3 it can be seen that floor accelerations decreased considerably with TMD1, TMD2, and TMD3 in place; however, when TMD1a replaced TMD1, the RMS acceleration was reduced even further. RMS acceleration due to parallel walking was reduced by a factor of 4.7 and 5.5 for the TMD1 and TMD1a conditions, respectively. RMS acceleration due to perpendicular walking was reduced by a factor of 3.3 and 3.7 for the TMD1 and TMD1a conditions, respectively. The increased TMD mass resulted in the decrease of floor accelerations for the TMD1a floor condition, even though TMD1a was not “optimally tuned” as an individual TMD.

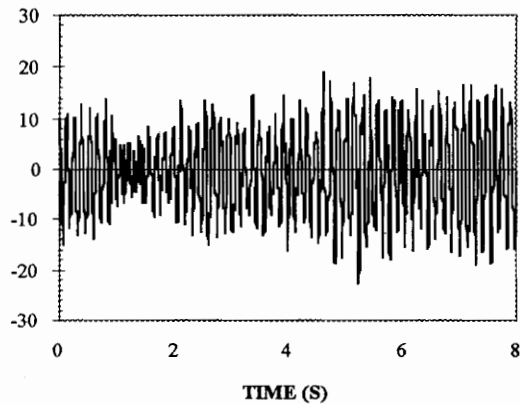
Walk Parallel to Joists

ACCELERATION (IN/S²)



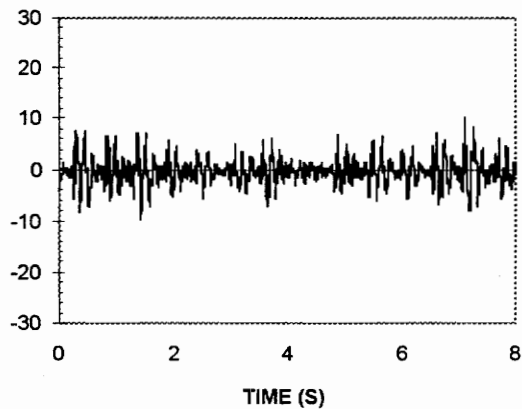
Walk Perpendicular to Joists

ACCELERATION (IN/S²)

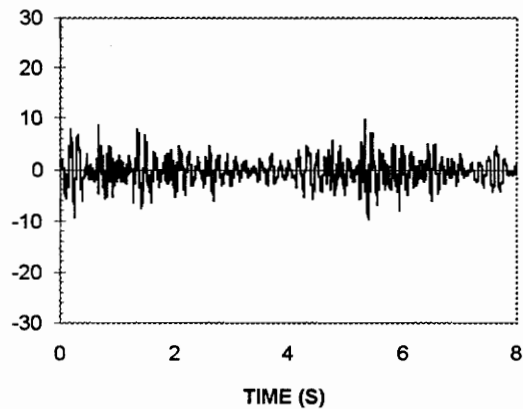


a - bare floor

ACCELERATION (IN/S²)

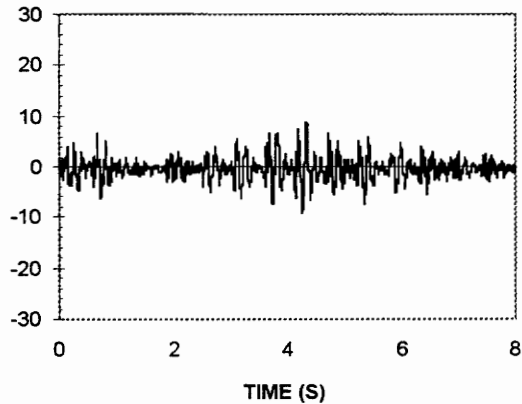


ACCELERATION (IN/S²)

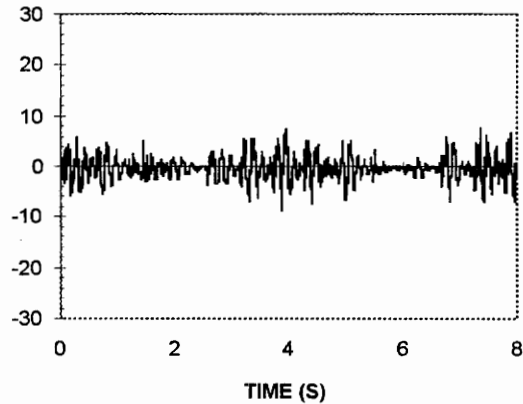


b - TMD1, TMD2, & TMD3 in place

ACCELERATION (IN/S²)



ACCELERATION (IN/S²)



c - TMD1a, TMD2, & TMD3 in place

Figure 3.6 Walk Parallel and Walk Perpendicular to Joists with Acceleration Measured at the Center of the Laboratory Floor

Table 3.3 RMS Acceleration and Ratios of Uncontrolled vs. Controlled for the Laboratory Floor

Walking Direction	Bare Floor RMS	TMD1,2,3 RMS	Ratio	TMD1a,2,3 RMS	Ratio
Parallel to Joists	12.24 in/s ²	2.61 in/s ²	4.69	2.24 in/s ²	5.46
Perpendicular to Joists	8.23 in/s ²	2.50 in/s ²	3.29	2.22 in/s ²	3.71

CHAPTER 4

EXPERIMENTAL USE OF TUNED MASS DAMPERS ON AN OFFICE FLOOR

4.1 Description of Office Floor

Several bays on the second floor of an occupied two-story office building were reported to have annoying levels of floor vibration. Experimental tests to control the vibrations using the previously described tuned mass dampers were performed on three of the bays. Engineers from the 3M Company participated in the testing.

The three bays have measured fundamental frequencies between 4 and 5 Hz, which is about twice the excitation frequency of a person walking or running. Using the half power method on FFTs of acceleration data, the damping ratio for the fundamental mode of vibration for each bay was found to be approximately 5%. Because the higher frequencies were close to one another and the floor had a relatively high damping ratio, it was not possible to calculate the damping ratio of the second and third modes of vibration of the bays. Comparing the 5% damping ratio with damping ratios of other floors on which TMDs have been used (see Table 1.1), a damping ratio of 5% is relatively high. According to Bachmann, et al. (1995), a TMD is more effective if the damping ratio of the floor is relatively small. TMDs were used to control vibrations on three of the bays, and TMDs were left in place for one week on the third bay to gain occupant feedback concerning TMD performance. The test results are presented in the following sections, and summarized in Section 4.6.

Each of the bays varies in size and configuration. Only the floor slab properties are the same for each bay. The normal-weight concrete slab is supported by 0.6C metal deck and has a

total thickness of 2 ½ in. The physical characteristics of each bay will be further described in the following sections.

4.2 Modification of Tuned Mass Dampers

The 3M Company engineers used a fundamental frequency of 4.8 Hz and a secondary frequency of 7.2 Hz to determine the initial design parameters for the prototype TMDs (see Section 2.2.2). Their analysis of the bays resulted in initial equivalent SDOF floor weights of 19,090 lbs and 13,632 lbs for the 4.8 Hz and 7.2 Hz frequencies, respectively. Table 4.1 summarizes the input and initial TMD parameter results. TMD1+ represents two prototype TMDs, sitting side-by-side, used to control the first mode of vibration.

Table 4.2 summarizes the spring and damping element configurations used on the TMDs. Two of the three laboratory TMDs were modified to control the office floor vibrations. Modifications were made to the damping elements and springs, so that the TMDs could be tuned to the various floor fundamental frequencies. Configurations A, B, and C in Table 4.2 were used in TMDs to control the first mode of vibration of the office floor bays. Configuration D was used to control a second or third mode of vibration of a bay. Configurations A, B, C, and D could be implemented in either one of the two TMDs to suit the specific needs of each bay. Configuration E is the TMD1 configuration used for the laboratory floor. The springs and damping element of the laboratory floor TMD1, which was originally tuned to a frequency of approximately 7 Hz, were left unchanged. The laboratory TMD1 was used to control the higher modes of vibration of the office floors.

Table 4.1 Initial TMD Parameters

	TMD1	TMD1+	TMD2	Remarks
f (Hz)	4.8	4.8	7.2	measured
w_F (lb)	19090	19090	13632	determined by 3M
w_T (lb)	365	775	445	chosen by 3M - gives acceptable μ
μ	0.0191	0.0406	0.0326	Equation 2.3
f_{optimal} (Hz)	4.71	4.61	6.97	Equation 2.14
ζ_{optimal} (%)	0.0823	0.1162	0.1054	Equation 2.15
f_D (Hz)	4.69	4.58	6.93	Equation 2.11
k (lb/in)	822	1662	2186	Eqns. 2.11 and 2.5
c	4.60	13.51	10.64	Eqns. 2.9 and 2.10

f = frequency

w_F = floor weight

w_T = TMD weight

$\bar{m}$ = TMD-to-floor mass ratio

f_{optimal} = optimal TMD frequency

ζ_{optimal} = optimal TMD damping ratio

f_D = damped frequency

k = stiffness

c = damping coefficient

Table 4.2 TMD Spring and Damping Element Parameters

	Configuration				
	A	B	C	D	E
ks (lb/in)	720	788	788	1182	1164
kd (lb/in)	138	420	138	420	414
$ks+kd = k_T$	856	1208	926	1602	1578
G' (psi)	40	40	40	40	46
η	1	1	1	1	1
t (in)	0.5	0.5	0.5	0.5	0.5
A (in ²)	1.72	5.25	1.72	5.25	4.5

ks = spring stiffness

kd = damper stiffness

G' = shear storage modulus

η = loss factor for viscoelastic material

t = damping element thickness

A = X-sectional area of damping element

4.3 Bay 1

This bay borders an exterior wall and is not adjacent to any other bay in which annoying floor vibrations were reported. Located in this bay are several cubicles separated by movable partitions. Main walkways through this bay run adjacent to the exterior wall, in a north-south direction, and perpendicular to the joists along the centerline of the bay, in an east-west direction. Figure 4.1 shows a plan view of Bay 1 and some of the adjacent floor area. Framing for Bay 1 consists of 20H6 joists spaced 3 ft on center. The joists span 30 ft between W24x76 girders, which also span 30 ft.

4.3.1 Vibration Characteristics

To determine the vibration characteristics of this bay, motion was initiated in neighboring bays as well as in the bay itself. The strongest floor response occurred when motion was initiated in the bay itself. Motion initiated in bays to the north and south of Bay 1 was barely perceptible in Bay 1; however, motion initiated in the bay just west of Bay 1 resulted in perceptible vibrations in Bay 1. It should be noted that the three bays directly west of Bay 1 have exactly the same type of joists and joist spacing. Without a variance in joist stiffness or spacing, this floor plan contributes to floor vibrations occurring in a wave-like motion transverse to the supporting members, as described in Chapter 2. This wave-like motion occurred in Bay 1, but wave-like motion was diminished by a full-height partition transverse to the joists and located in the bay adjacent to Bay 1. As a result, the reported floor measurements and computer analysis focus on Bay 1, and not on the neighboring bays and their effects on Bay 1.

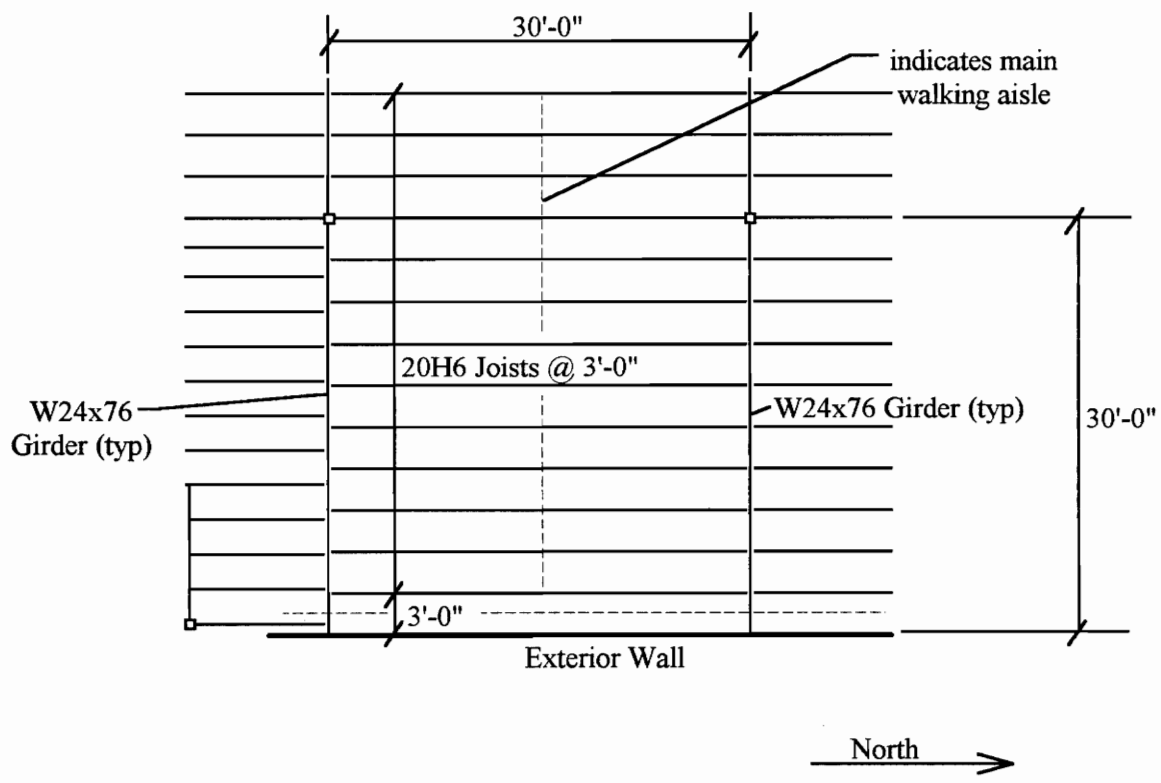


Figure 4.1 Plan View of Bay 1

4.3.1.1 Measured Vibration Characteristics

Fast Fourier Transforms of heel drop acceleration data were used to determine the natural frequencies of the bay. At the center of the bay, the fundamental mode of vibration dominates at a frequency of 5 Hz. The damping ratio for the fundamental mode of vibration is approximately 5%. Other dominant frequencies, determined at various locations in the bay, are 5.75 Hz, 6 Hz, and 8 Hz. The location at which the largest response corresponding with these frequencies occurs is at the center of the edge joist bordering the bay west of Bay 1. Figure 4.2 shows the acceleration and FFT of the acceleration response at the center of the edge joist when a heel drop was performed at the center of the bay. From the figure, it can be seen that the system has significant damping, but there is still considerable energy in the higher modes, i.e. 5.75 Hz to 8 Hz.

4.3.1.2 Analytically Determined Vibration Characteristics

To analytically determine the vibration characteristics of this bay, SAP90 was utilized. The bay model was prepared in a manner similar to the laboratory floor model described in Chapter 3. The model consists of frame and plate elements in a single plane and has 121 spatial coordinates. The plate elements have dimensions of 36 in. by 36 in. Figure 4.3 shows the model layout and properties of the frame and plate elements used in the final variation of the model. The SAP90 input for the final variation of the model is found in Appendix D.

Several variations of the model were studied to determine which model best reflected the actual floor vibration characteristics. Table 4.3 summarizes the variations of the models and the first four natural frequencies resulting from each model. Model NC, the most basic model, considered only the mass of the bay and the mass of neighboring bays, which is carried by the girders and edge joist. Model NC used a non-composite joist and girder moments of inertia.

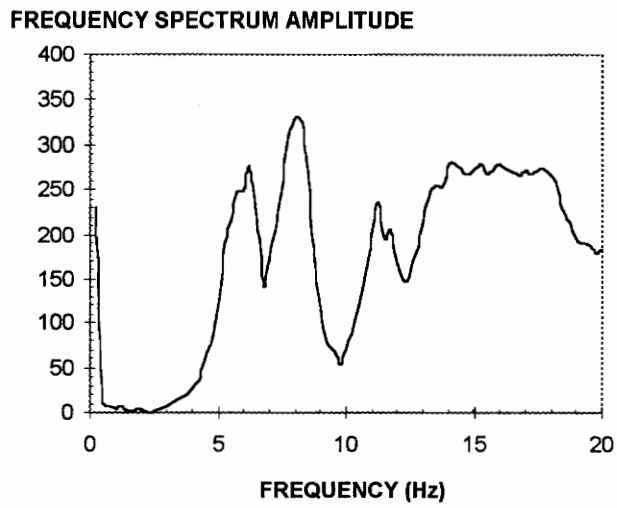
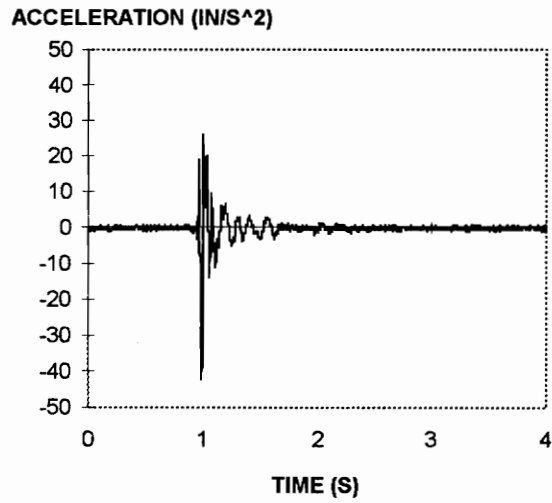
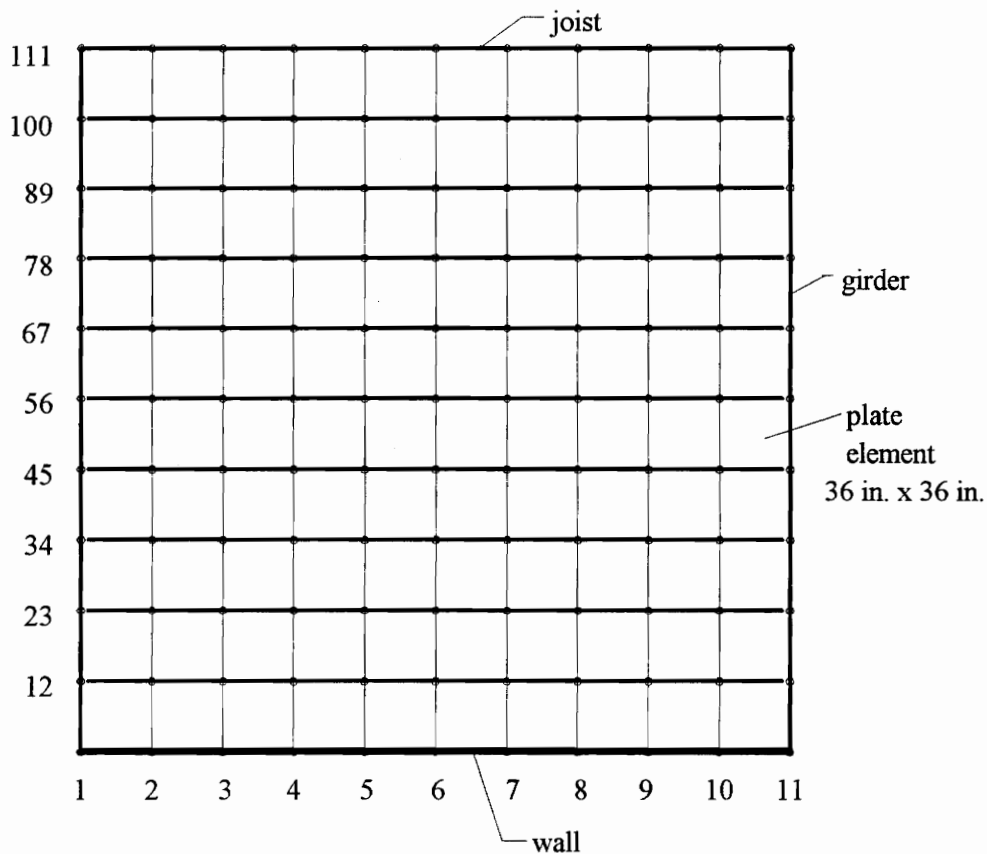


Figure 4.2 Response and FFT of the Response at the Center of the Edge Joist due to a Heel Drop at the Center of the Bay 1



Frame Elements	$I_x(\text{in}^4)$	Area (in^2)	Weight (lb/in^3)	E(psi)
Girder	2100	22.4	0.2827	29×10^6
Joist	270.8	1.491	0.4974	29×10^6
Wall	130000	1	0	29×10^6

Plate Elements	Bending Thickness (in)	Weight (lb/in^3)	E(psi)
	2.25	0.086998	3.409×10^6

Figure 4.3 SAP90 Model of Bay 1

Table 4.3 Frequencies of Bay 1 Model Variations

	Model Version				Measured Floor Vibration Data
	NC	CM	CM9	CM9RK	
Mode	freq. (Hz)	freq. (Hz)	freq. (Hz)	freq. (Hz)	freq. (Hz)
1	4.73	5.63	4.97	5.15	5.0
2	5.71	7.82	6.88	7.38	5.75
3	7.48	8.23	7.33	7.44	6.0
4	8.09	9.17	8.07	8.54	8.0

version	
NC	non-composite joists and girders, only dead weight of floor
CM	version NC, but composite joists
CM9	version CM, but 9 psf added
CM9RK	version CM9, but rotational springs added to the girders

Model CM was the same as model NC, except that a composite joist moment of inertia was used. Model CM9 incorporated a 9 psf sustained component of live load. This live load was included as additional mass at the nodes. Allen and Murray (1993) suggest a sustained component of live load of approximately 11 psf for office floors. A load of 9 psf was considered, since the furniture and other components on the floor were relatively light in weight. Model CM9RK incorporated rotational springs at the nodes along the girder. The rotational springs were added to take into account the continuous effect of the concrete slab between bays. Without these rotational springs, an eigenvalue analysis of the floor resulted in a torsional mode shape for the third mode of vibration. This torsional vibration contrasted with the observed wave-like vibration of the mode shapes. The addition of rotational springs to the model caused the mode shapes to return to a wave-like pattern.

Figure 4.4 shows the first four mode shapes and their corresponding frequencies resulting from an eigenvalue analysis of model CM9RK. The model frequencies for the higher modes of vibration vary from the measured frequencies; however, the small frequency gap between the model mode 2 and mode 3 frequencies correlates well with the small frequency gap between the measured mode 2 and mode 3 frequencies.

From the eigenvalue analysis, the mode shape displacements were used to determine the equivalent SDOF floor masses for the first, second, and third modes of vibration. The equivalent SDOF floor weights are 16,730 lbs, 6171 lbs, and 8500 lbs for the first, second, and third modes of vibration, respectively.

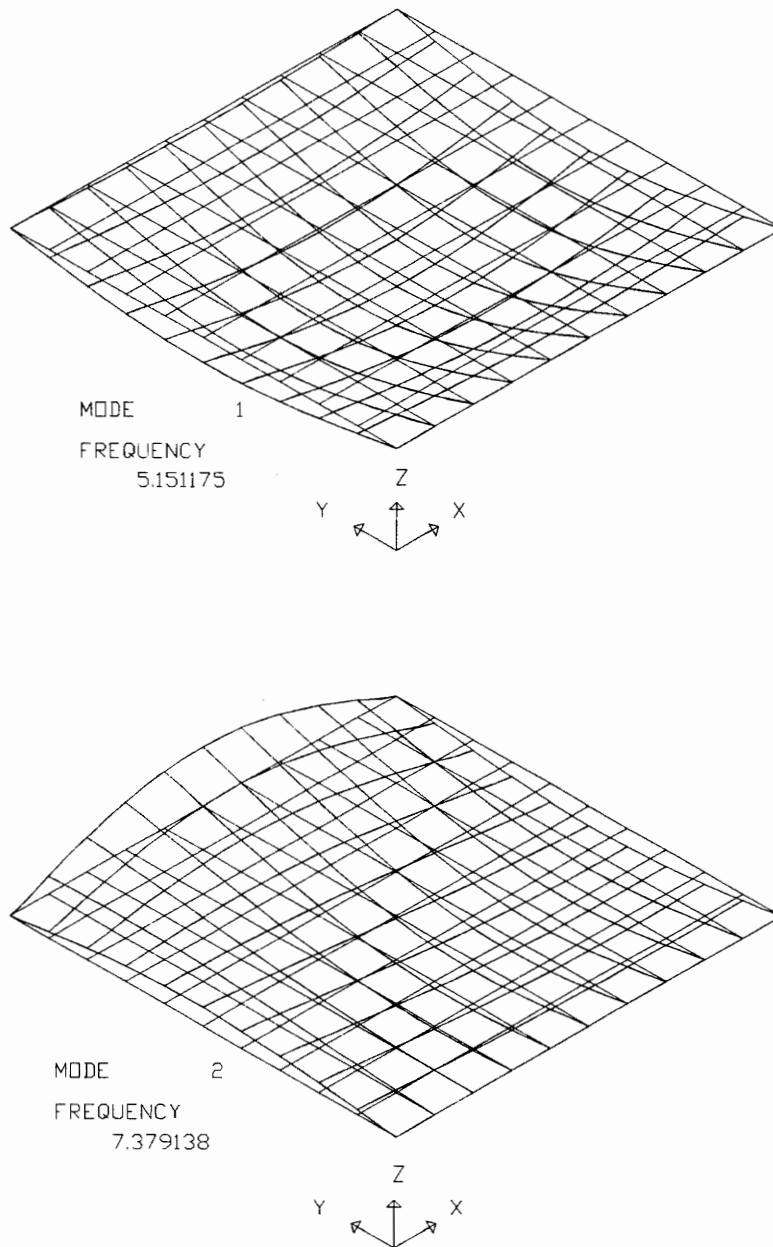


Figure 4.4 First Four Mode Shapes From SAP90 Analysis of Bay 1

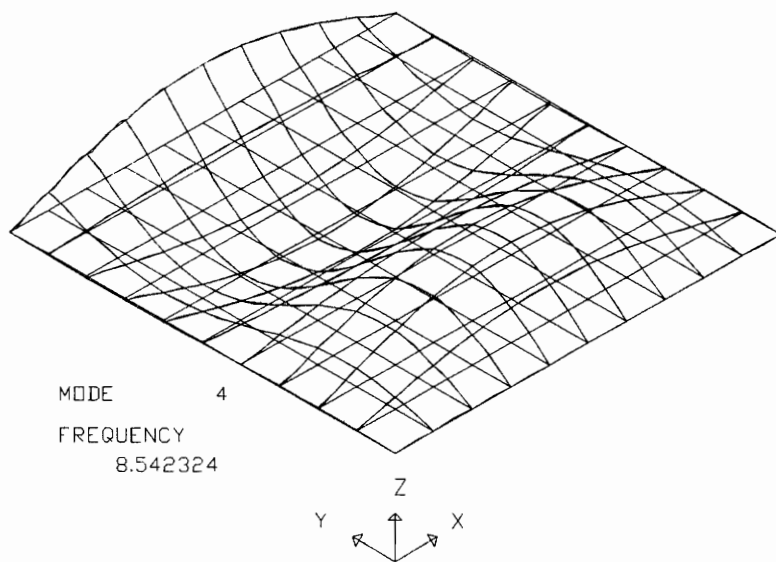
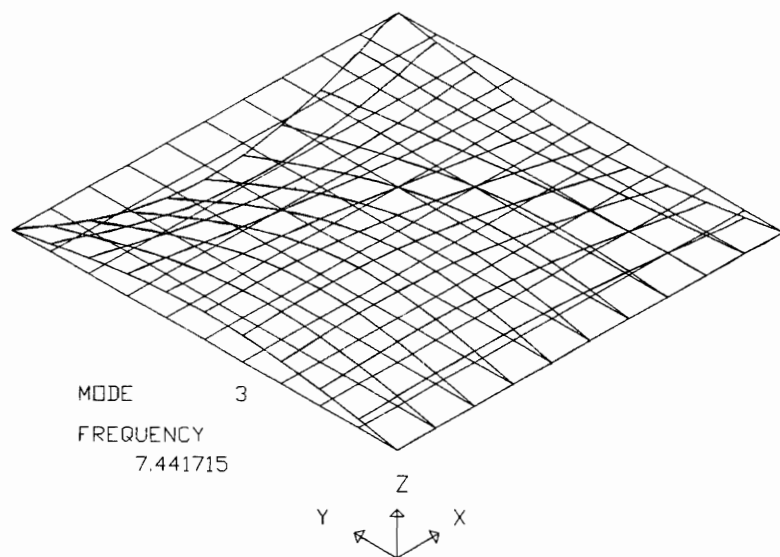


Figure 4.4 First Four Mode Shapes of Bay 1, continued

4.3.2 Installation of Tuned Mass Dampers

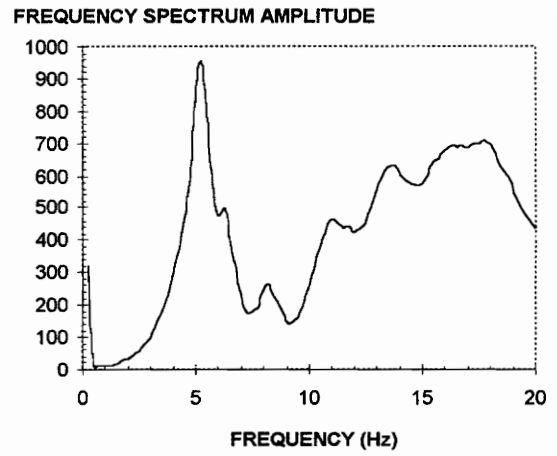
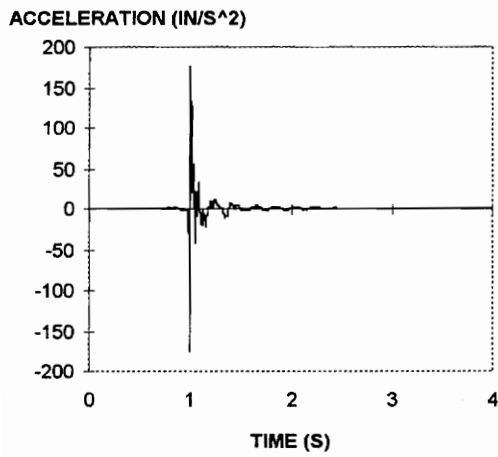
Before a TMD was put in place, the TMD was placed near a column, where the floor is relatively stiff, and tuned to the desired frequency. Once in place, plates were added or removed until “optimum” control of the floor vibration was gained. Vibrations were considered “optimally” controlled when the acceleration amplitudes, and frequency spectrum amplitude of the frequency of concern were reduced as low as possible with the TMD(s).

Several TMD configurations were tested on this bay. One configuration had two TMDs located at the center of the bay to control the fundamental mode of vibration and a third TMD located about 8 ft directly north of the other two TMDs. This configuration reduced the acceleration and appropriate frequency amplitudes; however, another TMD configuration reduced the walking acceleration amplitudes by approximately 25% when comparing RMS accelerations resulting from floor responses with the two configurations.

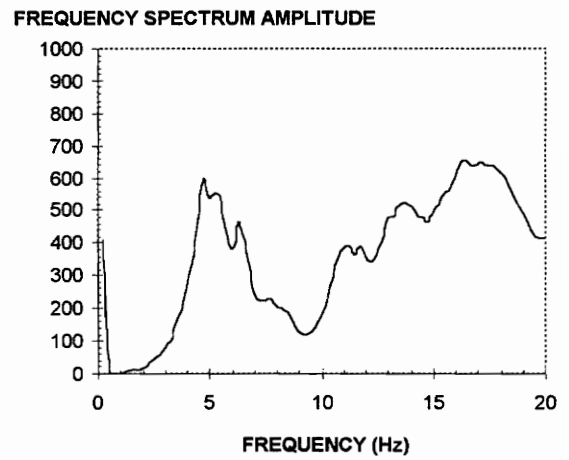
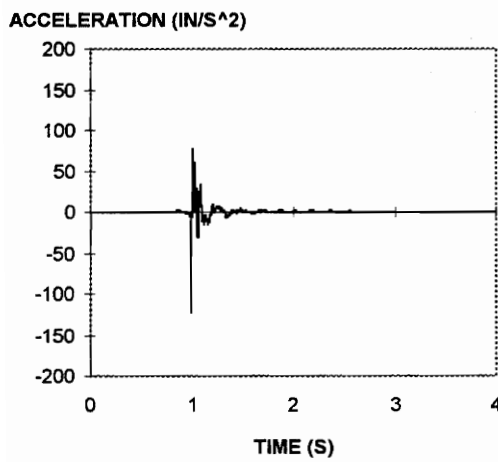
The final configuration used on Bay 1 had one TMD, TMD1, placed at the center to control the fundamental mode of vibration and two TMDs, TMD2 and TMD3, located at the center of the edge joist bordering the bay west of Bay 1. The final parameters of all three TMDs are in Table 4.4. TMD1 controlled the fundamental mode of vibration, while TMD2 and TMD3 controlled the higher modes of vibration. Due to the high system damping and closely spaced frequencies of this bay, final “tuning” of the TMDs in place was difficult. The final TMD frequencies of TMD2 and TMD3 fell between the frequencies of 6 Hz and 8 Hz, and not near an “optimum” frequency for either mode of vibration.

Table 4.4 Final TMD Parameters of Bay 1 TMDs

	Tuned Mass Dampers			Remarks
	TMD1	TMD2	TMD3	
w_F (lb)	16730	6171	8500	
w_T (lb)	325	345	315	(plates x 10)+75lb
plates	25	27	24	
μ	0.0194	0.0559	0.0371	Equation 2.3
ks (lb/in)	720	1182	1164	
kd (lb/in)	137.6	420	414	(G' x A)/t
ks+kd = k_T	857.6	1602	1578	
f_{nTMD} (Hz)	5.08	6.74	7.00	Equation 2.5
c_T	4.31	9.92	9.41	k _d /ω _{nTMD}
ζ_T (%)	0.0802	0.1311	0.1312	Eqns. 2.8 and 2.9
f_{DTMD} (Hz)	5.07	6.68	6.94	Equation 2.10
G' (psi)	40	40	46	
eta	1	1	1	
t (in)	0.5	0.5	0.5	
A (in²)	1.72	5.25	4.5	

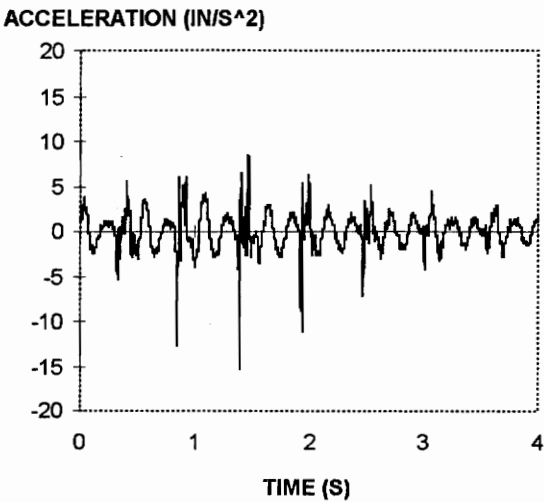


a - bare floor

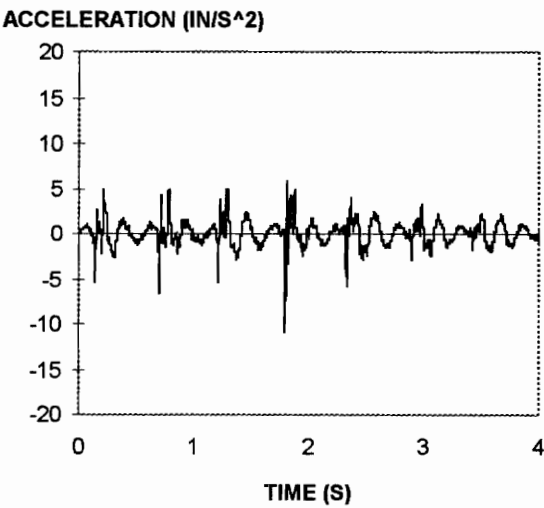


b - three TMDs in place

Figure 4.5 Heel Drop Performed and Acceleration Measured at the Center of Bay 1



a - bare floor



b - three TMDs in place

**Figure 4.6 Walk Perpendicular to Joists
with Acceleration Measured at the Center of
Bay 1**

4.3.3 Evaluation of Tuned Mass Damper Effectiveness

Figure 4.5 shows acceleration traces at the center of Bay 1 caused by heel drops at the center of the bay. Also shown are the resulting plots in the frequency domain. Figure 4.5a is for the bare floor, and 4.5b is with one TMD at the center and two TMDs at the center of the edge joist of the bay. With the TMDs in place, damping was increased and control of the first mode of vibration was improved.

Figure 4.6 shows acceleration traces due to walking in an east-west direction, perpendicular to the joists and near the mid-span of the joists. Figure 4.6a is for the bare floor, and 4.6b is with one TMD at the center and two TMDs at the center of the edge joist of the bay. Walking perpendicular to the joists excited the fundamental and higher modes of vibration. Considering the RMS of the acceleration data, acceleration due to walking was reduced by a factor of 1.4 with the three TMDs in place.

4.4 Bay 2

Bay 2 is an interior bay of the building. This bay, which is rectangular in shape, also had cubicles separated by movable partitions. In addition to the movable partitions, a full-height partition is located along the entire length of one of the edge joists of the bay. The main walkway through this bay runs parallel with the joists in the east-west direction. Figure 4.7 shows a plan view of Bay 2 and some of the adjacent floor area. Bay 2 consists of 24H8 joists spaced at 3 ft on center. The joists span 40 ft between W27x84 girders, which span 30 ft.

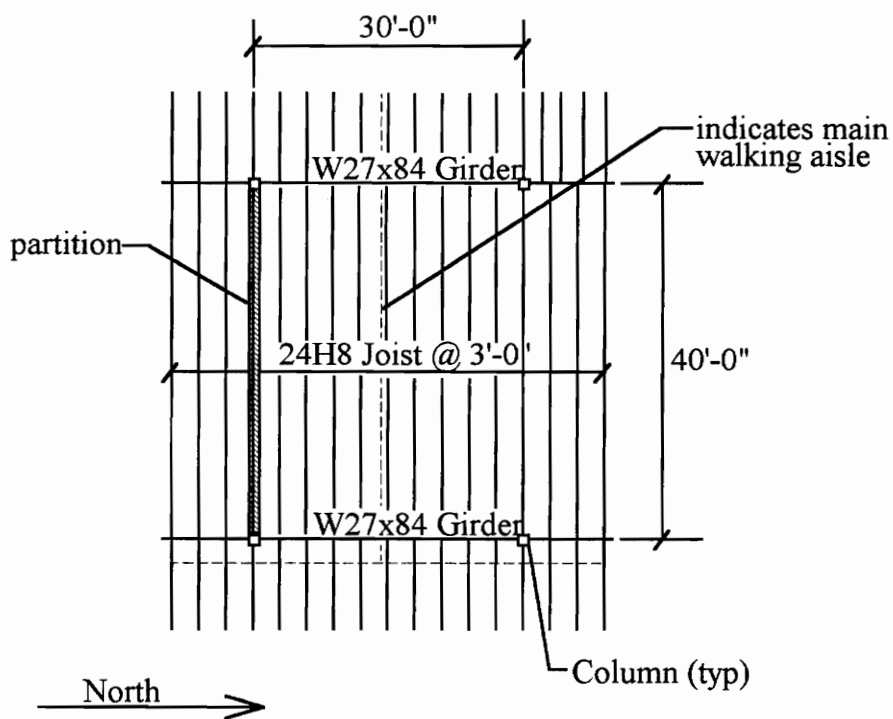


Figure 4.7 Plan View of Bay 2

4.4.1 Vibration Characteristics

Unlike Bay 1, the main walkway for this bay, and bays to the north and south, runs parallel to the joists. In addition, a full-height partition runs the entire length of one of the edge joists of this bay. For these reasons, the wave-like motions which were strong in Bay 1 are not strong in this bay. Walking between cubicles in the north-south direction excited the wave-like higher modes of vibration. Walking along the main walkway primarily excited the fundamental mode of vibration.

4.4.1.1 Measured Vibration Characteristics

Fast Fourier Transforms of heel drop acceleration data were used to determine the natural frequencies of the bay. The fundamental mode of vibration dominated the floor vibrations of this bay, and was excited when a person walked in the east-west direction. The fundamental mode of vibration occurs at a frequency of 4.75 Hz, and is strongest at the center of the bay. The damping ratio for this mode of vibration is approximately 5%.

4.4.1.2 Analytically Determined Vibration Characteristics

This bay was also modeled and analyzed using SAP90. Like the Bay 1 model, rotational springs were placed at the nodes of the girder so that the bay vibrated in a manner that reflected actual floor conditions. Because a full-height partition was located along an exterior joist, additional mass and stiffness were added to this joist to reflect the effects of the partition on the bay. As with Bay 1, a 9 psf sustained live load was added to the model as mass at the nodes. The SAP90 input for this model is in Appendix D. Figure 4.8 shows the grid layout for the model and

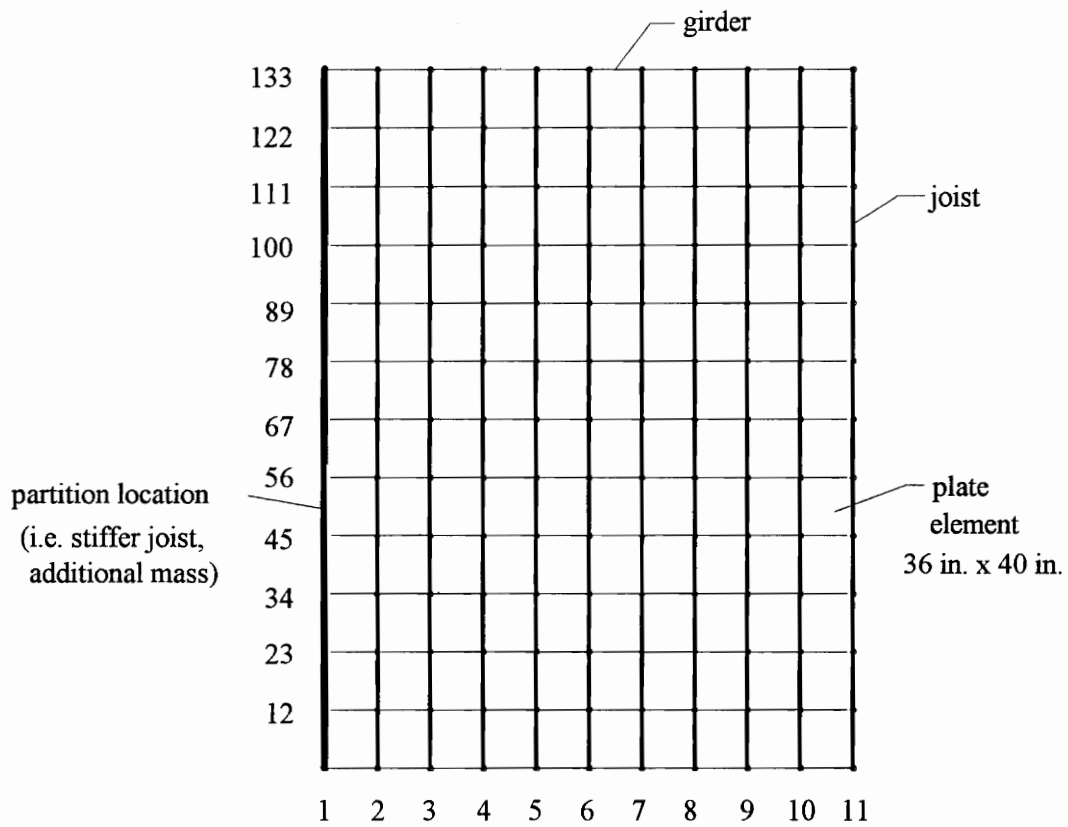
properties of the frame and plate elements used in the model. The plate elements in this model are rectangular with dimensions of 36 in. by 40 in.

The SAP90 analysis of this model predicted a fundamental frequency of 4.81 Hz. The measured floor fundamental frequency is 4.75 Hz. Figure 4.9 shows the first two mode shapes and their corresponding frequencies resulting from the eigenvalue analysis. Only the first mode of vibration was of major concern in this bay, so only the equivalent SDOF mass for the first mode of vibration was determined for this bay. Because of the partition on one side of the bay, the maximum displacement for the first mode did not occur at the center of the bay, but rather about 9 ft north of the center of the bay. At this location the equivalent SDOF floor weight is 17,300 lbs. However, the TMDs were located at the center of the bay so that they would be in the main walking aisle. When TMDs are located at the center of the bay, the equivalent SDOF floor weight is 21,520 lbs.

4.4.2 Installation of Tuned Mass Dampers

Several TMD configurations were tried on Bay 2 to control the fundamental mode of vibration. The first configuration used one TMD at the center of the bay. The second configuration used two TMDs at the center of the bay. The third configuration had the two TMDs spaced 6 ft apart, on an east-west centerline of the bay.

The second configuration was the most successful in controlling the floor vibrations. The parameters of these two TMDs are found in Table 4.5. Using two TMDs versus one TMD has the added advantage of a higher mass ratio. In general, the higher the mass ratio, the greater the control of the floor vibrations. The total TMD-to-floor mass ratio with this configuration is 0.0372.



Frame Elements	$I_y(\text{in}^4)$	Area (in^2)	Weight (lb/in^3)	E(psi)
Girder	2850	24.8	0.2823	29×10^6
Joist	464.7	1.833	0.5228	29×10^6
Stiff Joist	4647	1.833	0.5228	29×10^6

Plate Elements	Bending Thickness (in)	Weight (lb/in^3)	E(psi)
	2.25	0.086998	3.409×10^6

Figure 4.8 SAP90 Model of Bay 2

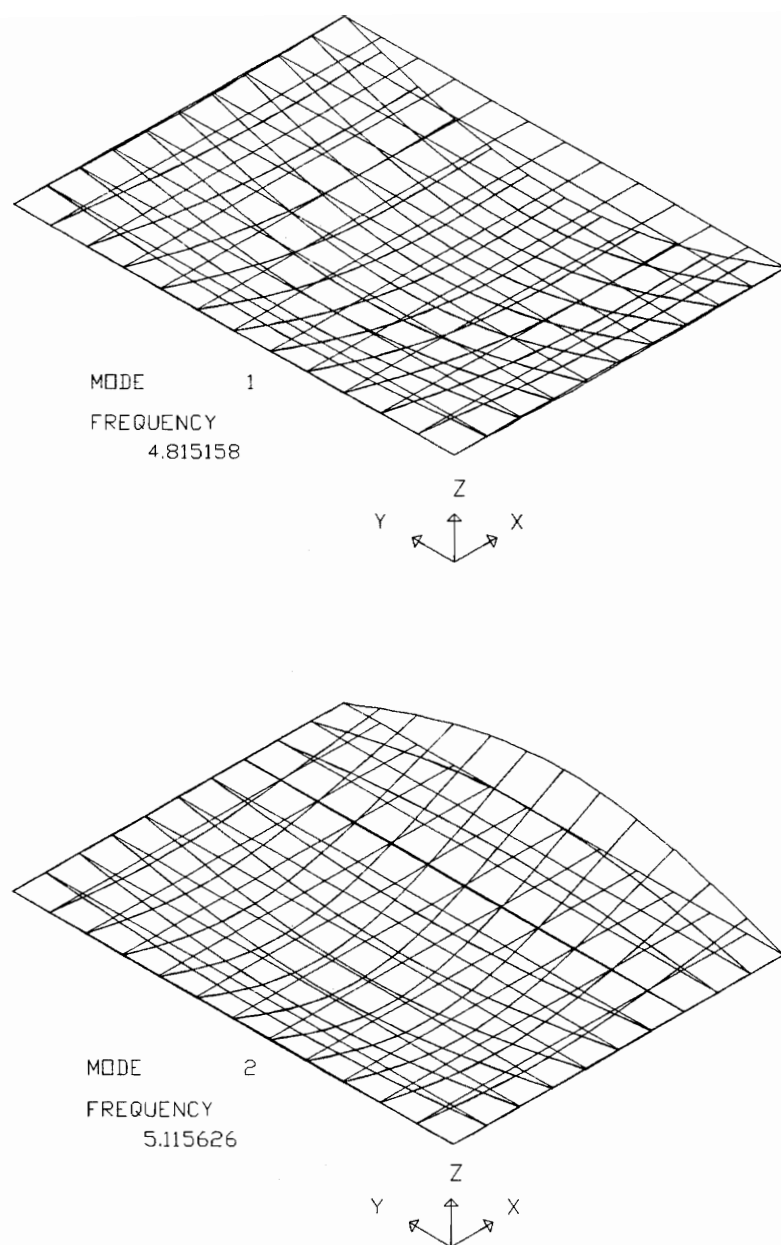


Figure 4.9 First Two Mode Shapes From SAP90 Analysis of Bay 2

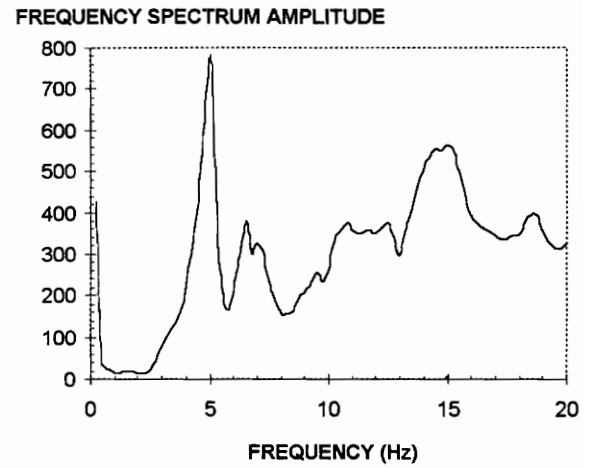
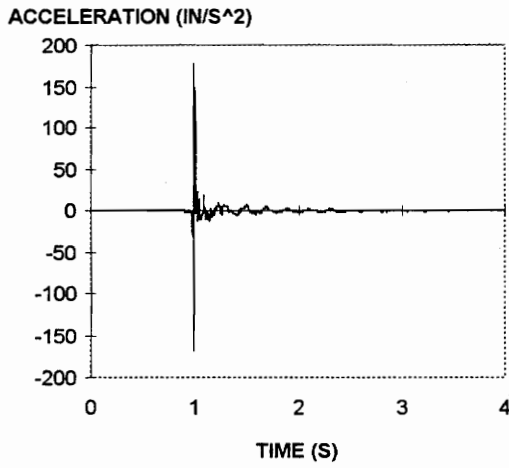
Table 4.5 Final TMD Parameters of Bay 2 TMDs

	Tuned Mass Dampers		Remarks
	TMD1a	TMD1b	
w_F (lb)	21520	21520	
w_T (lb)	325	475	(plates x 10)+75lb
plates	25	40	
μ	0.0151	0.0221	Equation 2.3
ks (lb/in)	720	788	
kd (lb/in)	137.6	420	(G' x A)/t
ks+kd = k_T	857.6	1208	
f_{nTMD} (Hz)	5.08	4.99	Equation 2.5
c_T	4.31	13.40	k _d /ω _{nTMD}
ζ_T (%)	0.0802	0.1738	Eqns. 2.8 and 2.9
f_{DTMD} (Hz)	5.07	4.91	Equation 2.10
G' (psi)	40	40	
eta	1	1	
t (in)	0.5	0.5	
A (in^2)	1.72	5.25	

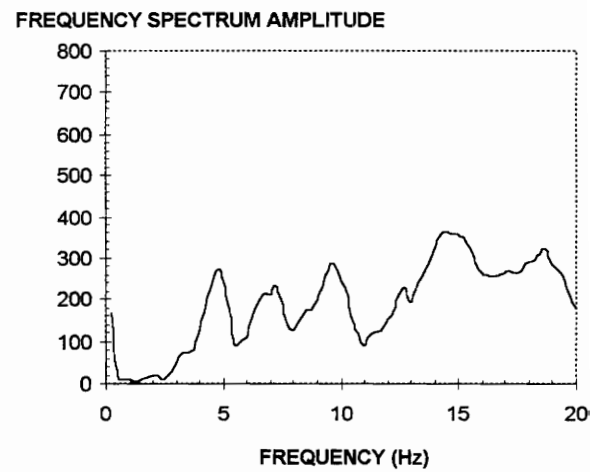
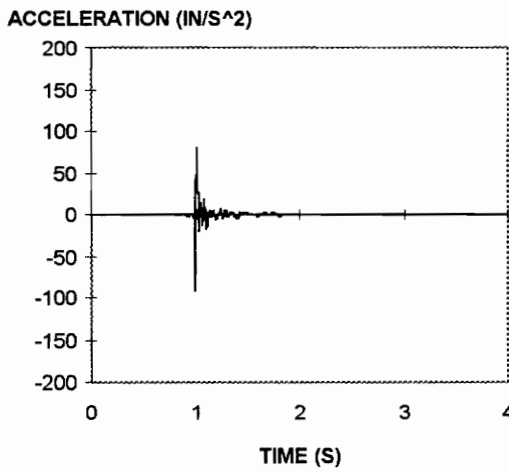
4.4.3 Evaluation of TMD Effectiveness

Figure 4.10 shows acceleration traces due to heel drops and the resulting plots in the frequency domain. Figure 4.10a is for the bare floor and 4.10b is with two TMDs at the center of the floor. With the two TMDs in place, it can be seen that the TMDs increased the damping and decreased the response to an impact on the floor.

Figure 4.11 shows acceleration traces due to walking parallel to the joists in an east-west direction, and acceleration traces due to walking perpendicular to the joists in a north-south direction. Walking perpendicular to the joists required that the person walk in and out of two cubicles that were located across from one another, on either side of the main walkway. The perpendicular walk was located about 3 ft east of the bay centerline. The parallel walk, down the main walkway, was about 3 ft south of the bay centerline. The RMS of the acceleration due to walking parallel to the joists was reduced by a factor of 1.75, while RMS of the acceleration due to walking perpendicular to the joists was reduced by a factor of 1.23. The acceleration reduction factor for walking perpendicular to the joists was not as great as the parallel walking reduction factor because walking perpendicular to the joists excited the higher modes of vibration more than walking parallel to the joists. Since the main walkway runs parallel to the joists, control of the floor vibrations was considered sufficient with TMDs controlling only the fundamental mode of vibration.

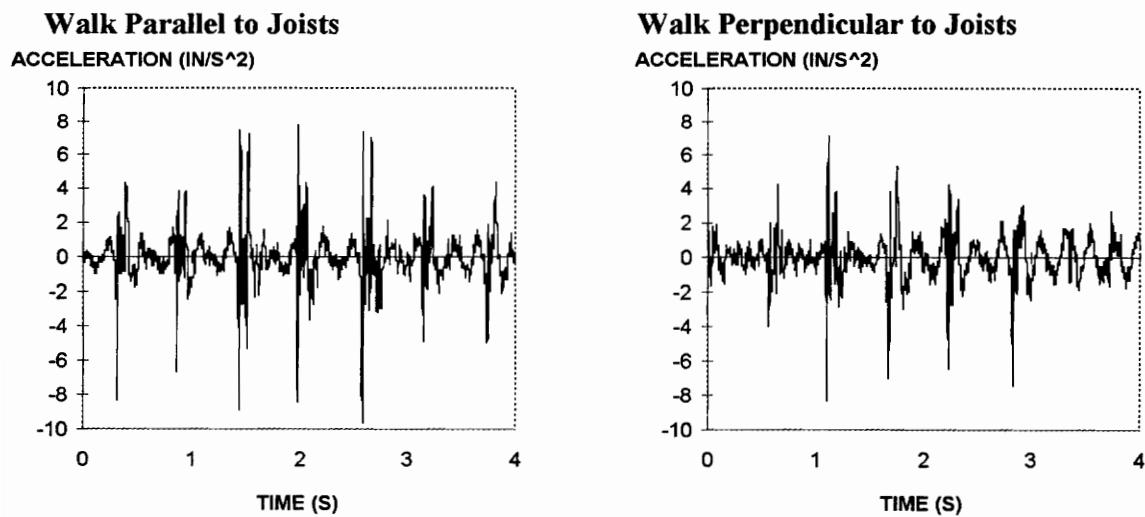


a - bare floor

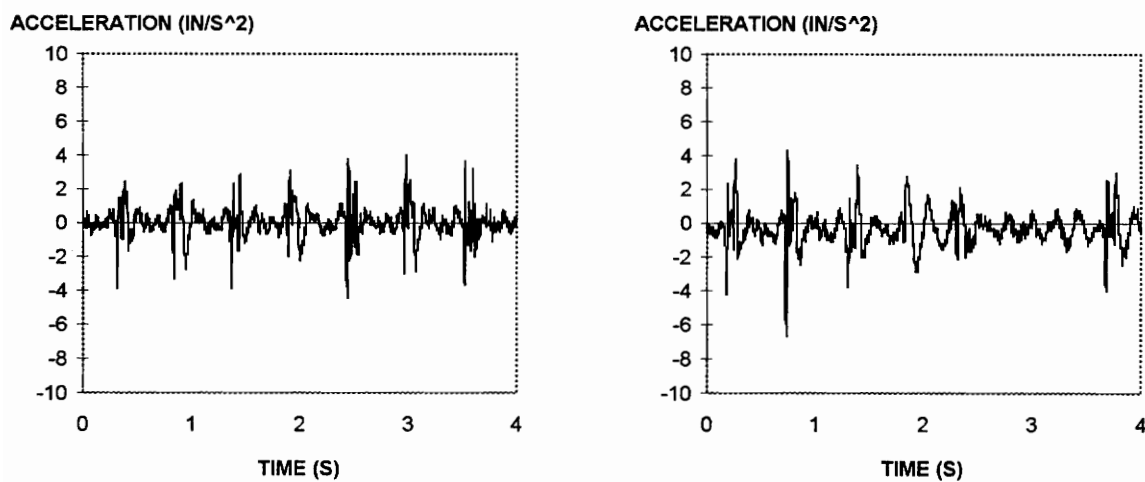


b - two TMDs at the center

Figure 4.10 Heel Drop Performed and Acceleration Measured at the Center of Bay 2



a - bare floor



b - two TMDs at the center

Figure 4.11 Walk Parallel and Walk Perpendicular to Joists with Acceleration Measured at the Center of Bay 2

4.5 Bay 3

This bay borders an exterior wall, and is adjacent to Bay 2. Bay 2 is directly east of the main portion of Bay 3. The main portion of this bay has dimensions of 35 ft by 30 ft. Figure 4.12 shows a plan view of the bay; note that the 35 ft by 30 ft area does not have a column at one corner. This particular bay has the most complex plan of the three bays on which TMDs were tested.

4.5.1 Vibration Characteristics

The main walking aisle in this bay runs perpendicular to the joists, and due to that fact, the higher modes of vibration are excited when people walk through the bay in the north-south direction. Two full-height partitions are located on edge members of this bay, as shown in Figure 4.12. The step shape edge of the north portion of the bay and the locations of full-height partitions complicate the bay behavior. For this reason, correcting a mode of vibration other than the fundamental mode of vibration was found to be very difficult.

4.5.1.1 Measured Vibration Characteristics

Fast Fourier Transforms of heel drop acceleration data were used to determine the bay frequencies. The first mode of vibration occurs at a frequency of 4.25 Hz, and is strongest near the center of the bay. A second frequency of about 6.25 Hz is also prominent at this location. The damping ratio for the first mode of vibration is about 5%.

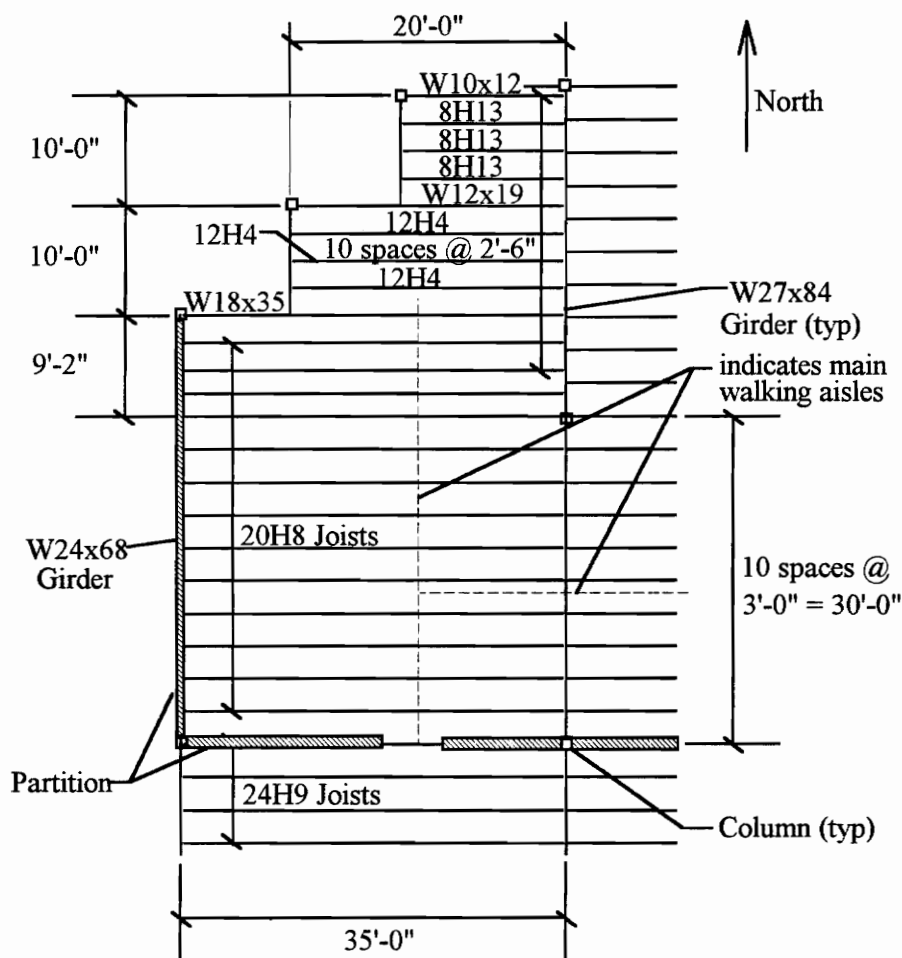


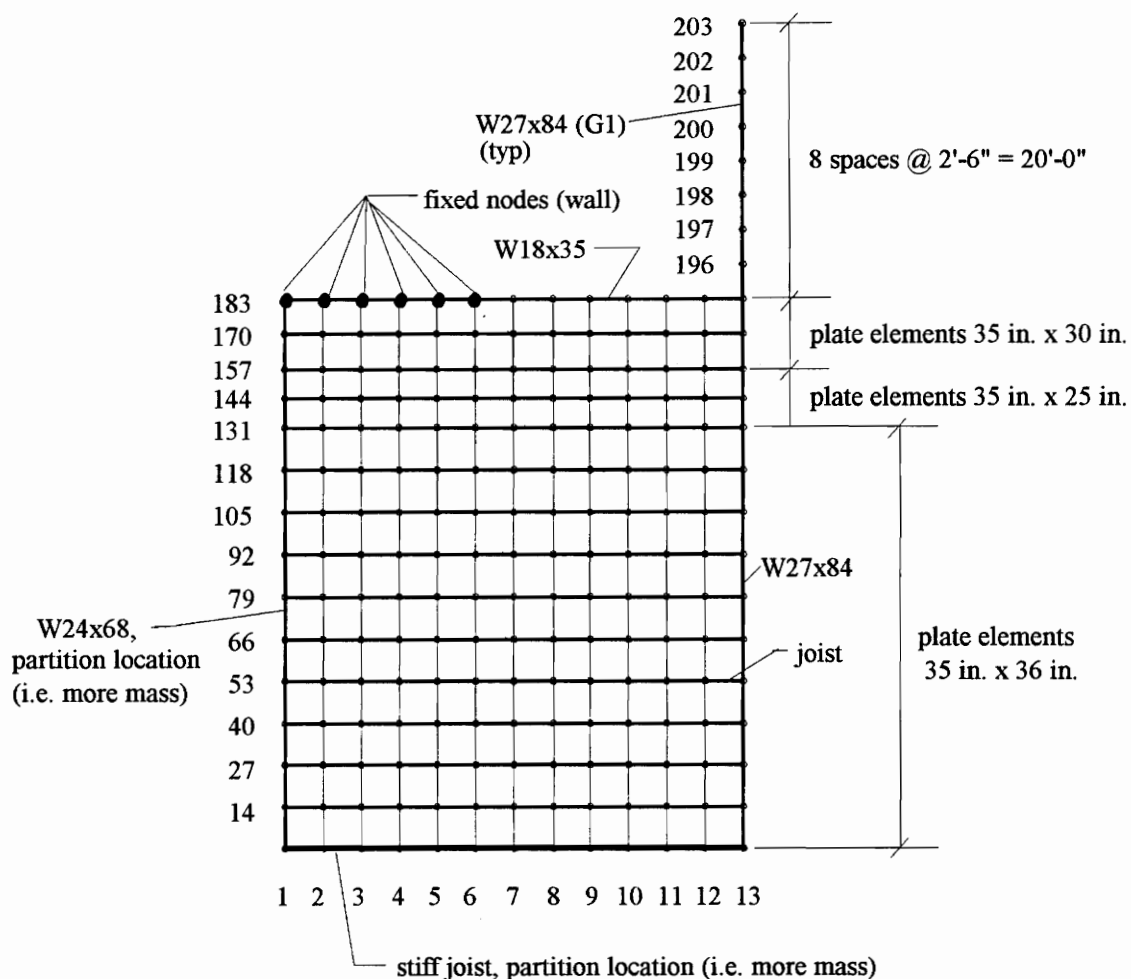
Figure 4.12 Plan View of Bay 3

4.5.1.2 Analytically Determined Vibration Characteristics

This bay was also modeled using SAP90. The model, Figure 4.13, did not include the entire step portion of the exterior portion of the bay. The model measures 35 ft by 39 ft - 2 in. with one of the girders extending beyond the rectangular grid a to connect with a column in the actual structure. For the girder extension, a modified moment of inertia is used to account for the concrete supported by the girder. In addition, the girder extension is loaded with the pertinent tributary mass. The mass of a full-height partition was added to the members which directly support a partition. The moment of inertia of the joist supporting a partition was increased by a factor of about ten to account for the added stiffness created by the partition. The joist which directly supported a partition in the Bay Two model also had a moment of inertia about ten times its original moment of inertia. Increasing the joist moment of inertia by greater factors had little effect on the natural frequencies or mode shapes of the bay. The moment of inertia of the girder supporting the partition was left unchanged; the girder moment of inertia is of the same order of magnitude as the “stiffened” joist moment of inertia. To account for the effect of the exterior wall, nodes of the W18x35 frame elements along the exterior wall were fixed in all directions.

The plate elements for this model were not uniform in size. The 35 in. width remained the same for all of the plate elements; however, the lengths of the elements were 25 in., 30 in., or 36 in. depending on the spacing between the joists.

Figure 4.13 shows the grid layout for this model, and lists the properties of the frame and plate elements. The joist properties listed in Figure 4.13 are an average of all the joist properties used in the model. The joist composite moment of inertia changed when the joist spacing changed. As with the other models, a 9 psf sustained live load was added as additional nodal mass. The SAP90 input for this model is in Appendix D.



Frame Elements	$I_y(\text{in}^4)$	Area (in^2)	Weight (lb/in^3)	E(psi)
W27x84 Girder	2850	24.8	0.2823	29×10^6
W27x84 G1	2866	24.8	0.2823	29×10^6
W24x68 Girder	1830	20.1	0.2819	29×10^6
20H8 Joist	295.6	1.053	0.8705	29×10^6
Stiff Joist	2950	1.137	0.8795	29×10^6
W18x35 Beam	960	10.3	0.2832	29×10^6

Plate Elements	Bending Thickness (in)	Weight (lb/in^3)	E(psi)
	2.25	0.086998	3.409×10^6

Figure 4.13 SAP90 Model of Bay 3

The SAP90 analysis of this model resulted in a fundamental frequency of 4.38 Hz. The measured floor frequency is 4.25 Hz. Once again, the mode shapes demonstrated a wave-like vibration. The first four mode shapes and their corresponding frequencies are in Figure 4.14. The equivalent SDOF floor weights are 14,960 lbs and 12,290 lbs for the first two modes of vibration, respectively. These weights correspond to the locations of maximum displacement for the respective modes of vibration. However, the TMDs were not placed at the locations of maximum displacement. The equivalent SDOF floor weight for a TMD placed at the center of the bay to control the fundamental mode of vibration is 15,560 lbs. The second TMD was placed approximately 14 ft west of the W27x84 girder and approximately 5 ft south of the W18x35 beam. The equivalent SDOF floor weight for this TMD would be 65,600 lbs for the second mode of vibration and 39,430 lbs for the third mode of vibration. The 39,430 lb weight was used in determining the TMD-to-floor mass ratio, since this weight was not as large and seemed more reasonable for determining the mass ratio.

4.5.2 Installation of Tuned Mass Dampers

Two TMDs were used in this bay area. One TMD, TMD1, controlled the first mode of vibration, while the second TMD, TMD2, controlled a higher mode of vibration. (A third TMD on the bay to help control the higher modes of vibration or provide additional damping and mass to control the fundamental mode of vibration would have been ideal; however, there were not enough mass plates available to provide sufficient mass and properly “tune” a third TMD.) The final parameters of the two TMDs can be found in Table 4.6. The mass ratio for TMD2 is lower than desired; however, this mass ratio provided the “optimum” control of the floor vibrations.

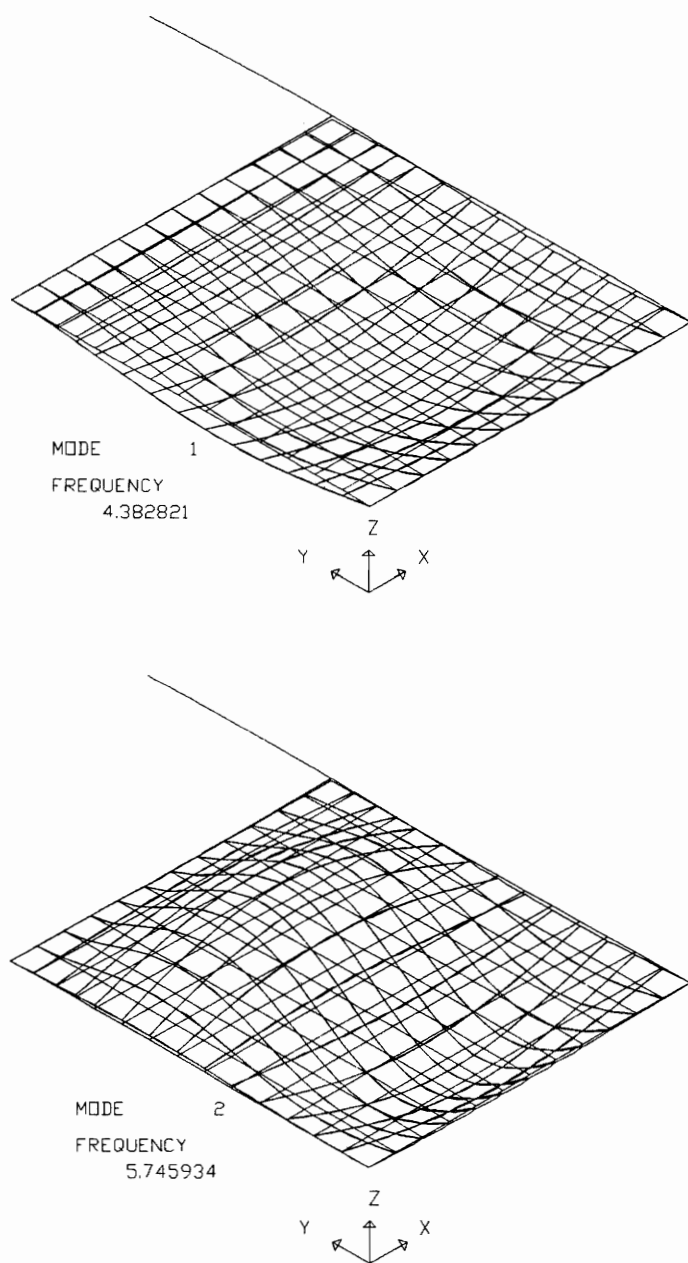


**Figure 4.14 First Four Mode Shapes From SAP90
Analysis of Bay 3**

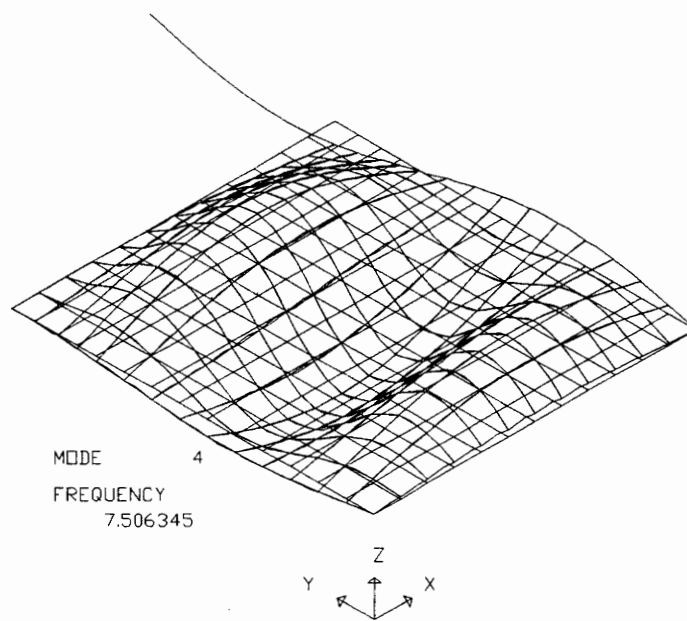
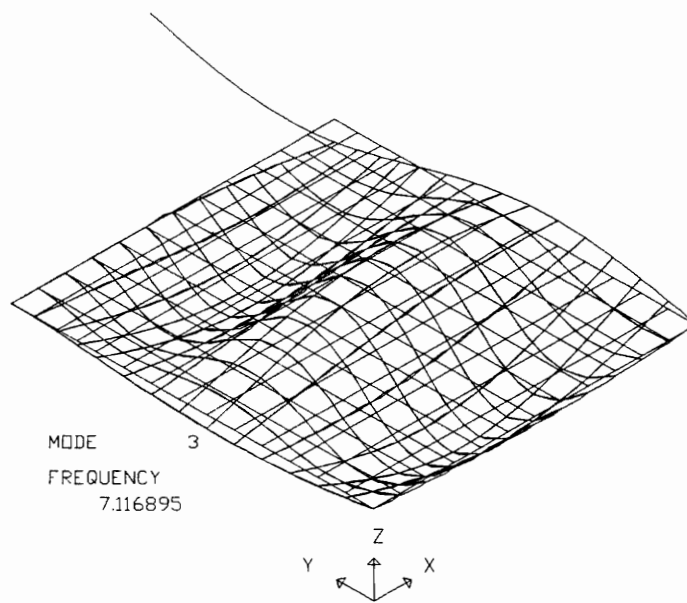


Figure 4.14 First Four Mode Shapes of Bay 3, continued

Table 4.6 Final TMD Parameters of Bay 3 TMDs

	Tuned Mass Dampers		Remarks
	TMD1	TMD2	
w_F (lb)	15600	39430	
w_T (lb)	525	435	(plates x 10)+75lb
plates	45	36	
μ	0.0337	0.0110	Equation 2.3
ks (lb/in)	788	788	
kd (lb/in)	138	414	(G' x A)/t
ks+kd = k_T	926	1202	
f_{nTMD} (Hz)	4.15	5.20	Equation 2.5
c_T	5.27	12.67	k _d /ω _{nTMD}
ζ_T (%)	0.0743	0.1722	Eqns. 2.8 and 2.9
f_{DTMD} (Hz)	4.14	5.12	Equation 2.10
G' (psi)	40	46	
eta	1	1	
t (in)	0.5	0.5	
A (in²)	1.72	4.5	

4.5.3 Evaluation of TMD Effectiveness

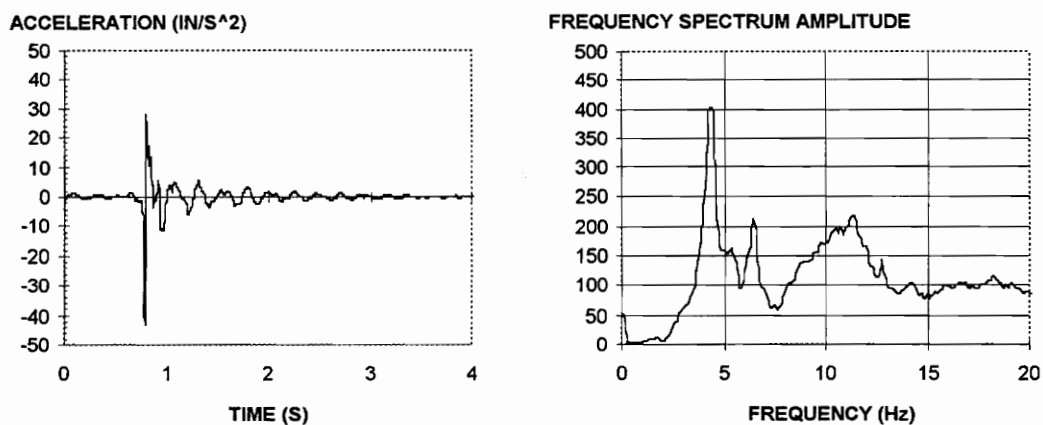
Tuned mass damper effectiveness was measured analytically, as with the previous bays; however, TMD effectiveness was measured by the occupants as well. This TMD configuration was left in place for one week, and the occupants were then asked how well they thought the TMDs controlled the floor vibrations.

4.5.3.1 Analytical Measure of Effectiveness

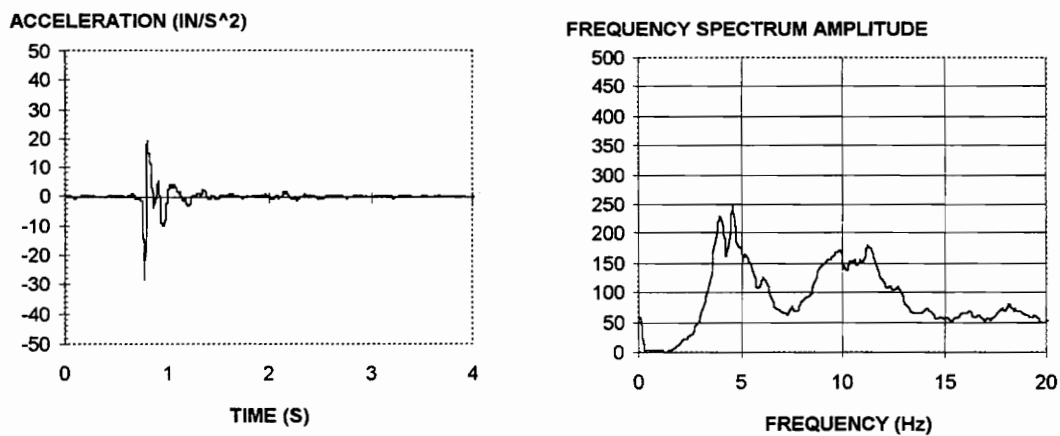
Heel drops and walks for this bay were performed by a person weighing 135 lbs. Heel drops and walks for the previous office floor bays and the laboratory floor were performed by a person weighing 210 lbs. Because of the difference in weight, the initial acceleration peaks resulting from heel drops are no longer of the same magnitude, but this only reduces the amplitude of the acceleration in the frequency domain, and does not change its shape. The magnitude of accelerations due to walking are also lower, but since the ratio of the RMS accelerations with and without TMDs on the floor is used as a measure of TMD effectiveness, this is not a problem.

Figure 4.15 shows acceleration traces due to a heel drop at the center of the main portion of the bay. Figure 4.15a is for the bare floor and 4.15b is for one TMD, TMD1, at the center and the other TMD, TMD2, just outside the main bay area. Damping was increased for the first mode of vibration with the addition of the two TMDs. Comparing the plot in the frequency domain of Figure 4.15b to the plot in the frequency domain of Figure 4.15a, a split of the 4.25 Hz frequency is now visible.

Figure 4.16 shows the acceleration trace due to walking parallel to the joists and the acceleration trace due to walking perpendicular to the joists. Walking parallel to the joists involved walking down a portion of a main walking aisle and continuing between several cubicles. This



a - bare floor

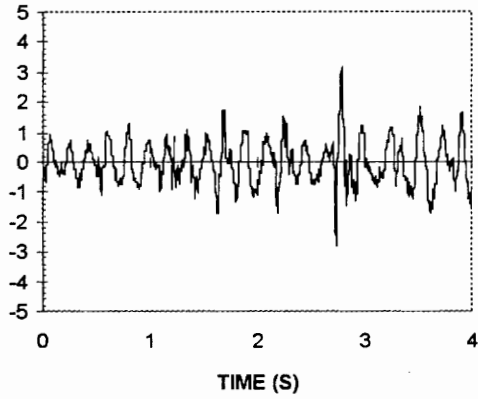


b - two TMDs in place

Figure 4.15 Heel Drop Performed and Acceleration Measured at the Center of the Main Portion of Bay 3

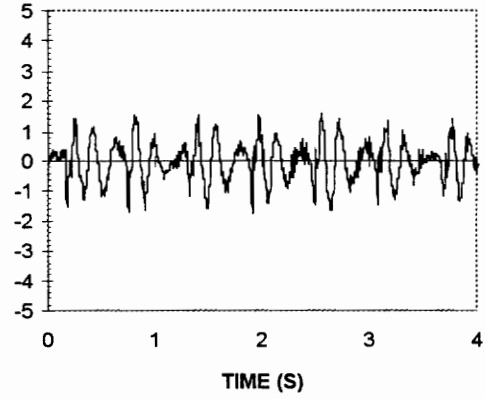
Walk Parallel to Joists

ACCELERATION (IN/S²)



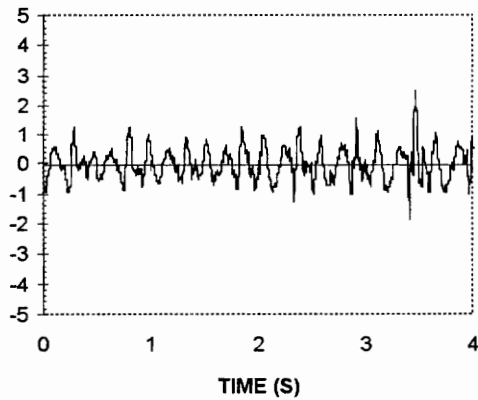
Walk Perpendicular to Joists

ACCELERATION (IN/S²)

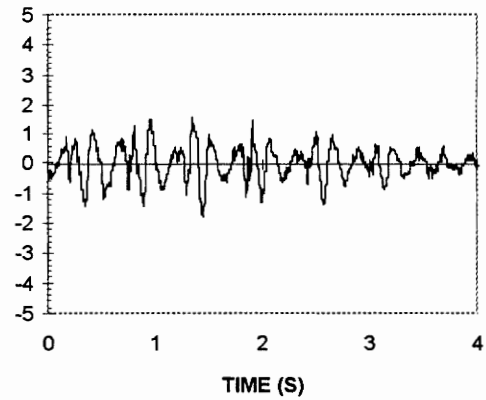


a - bare floor

ACCELERATION (IN/S²)



ACCELERATION (IN/S²)



b - two TMDs in place

Figure 4.16 Walk Parallel and Walk Perpendicular to Joists with Acceleration Measured at the Center of the Main Portion of Bay 3

walkway was located along the centerline of the main portion of the bay. The perpendicular walk involved walking down a main walking path which ran the entire length of the bay area modeled using SAP90. The perpendicular walk was located along the centerline of the bay, as well. The RMS acceleration due to walking parallel to the joists was reduced by a factor of 1.35. The RMS acceleration due to walking perpendicular to the joists was reduced by a factor of 1.23. The RMS acceleration reduction factor for parallel walking is not as high as the RMS acceleration reduction factor for Bay 2 parallel walking. This is due to the fact that a second TMD was used in Bay 2 to control the fundamental mode of vibration. A second TMD to control the fundamental mode of vibration of Bay 3 would have added damping to the system and more mass to counteract the floor vibrations.

4.5.3.2 Occupant Evaluation of TMD Effectiveness

The TMDs were left in place on Bay Area 3 for a week, after which time they were removed from the floor. The occupants of this bay area were asked about the effectiveness of the TMDs at controlling the floor vibrations. One person said that she did not notice the improved control of the floor vibrations with the TMDs in place until the TMDs were gone. Three of the five people in the bay area noted some improvement in the control of floor vibrations. One person said they felt the improvement in floor vibration control, and thought the TMDs worked well. Note that this bay had the smallest improvement in floor vibration control when comparing RMS acceleration results of all the bays, yet the occupants still felt improved floor vibration control with the TMDs in place.

4.6 Summary of Results

Experimental tests using tuned mass dampers to control floor vibrations were performed on three bays on the second floor of a two-story office building. The bays have fundamental frequencies ranging from 4 to 5 Hz. The damping ratio for the fundamental mode of vibration is approximately 5% for all three bays. This damping ratio is relatively high. According to Bachmann, et al. (1995), a TMD is more effective if the damping ratio of the floor is relatively small.

Each bay is unique in its configuration. The direction of the main walking aisle through the bay has a significant effect on the floor vibrations. Table 4.7 summarizes the RMS accelerations due to walking in the bays with and without TMDs in place. Comparing the acceleration reduction factors of each bay, the TMDs in Bay 1 were the most effective. Bay 1 was the smallest bay and had the most TMDs in place. No acceleration measurements for walking parallel to the joists were recorded for Bay 1, but observing the parallel vs. perpendicular walking RMS acceleration ratios of Bay 2 and Bay 3, the perpendicular RMS acceleration ratios are lower than the corresponding parallel RMS acceleration ratios. Based on the RMS acceleration ratios due to parallel walking, Bay 1 TMDs were the most effective at controlling more than one mode of vibration.

As seen in Table 4.8, the frequencies of the TMDs designed to control the fundamental modes of vibration are higher than the actual floor fundamental frequencies for Bay 1 and Bay 2. In theory, the optimum frequency for a TMD should be slightly less than the floor frequency which it is designed to control (Bachmann et al. 1995). Most of the bays have a second natural frequency within 2 Hz of the fundamental frequency. Because of the close proximity of adjacent

Table 4.7 RMS Acceleration and Ratios of Uncontrolled vs. Controlled for the Three Office Bays

Direction of Walking with respect to the Joists		Parallel	Perpendicular
Bay 1	Bare Floor RMS		2.017 in/s ²
	RMS with TMDs in Place		1.445 in/s ²
	Ratio		1.396
Bay 2	Bare Floor RMS	0.175 in/s ²	0.143 in/s ²
	RMS with TMDs in Place	0.100 in/s ²	0.116 in/s ²
	Ratio	1.754	1.232
Bay 3	Bare Floor RMS	0.743 in/s ²	0.695 in/s ²
	RMS with TMDs in Place	0.549 in/s ²	0.565 in/s ²
	Ratio	1.353	1.231

Table 4.8 Calculated TMD Frequencies and Measured Floor Frequencies of the Three Office Bays

Bay	TMD Number	TMD mass ratio	TMD Damped Frequency (Hz)	Measured Frequency (Hz)
1	TMD1	0.0194	5.07	5.0
	TMD2	0.0560	6.68	5.75
	TMD3	0.0371	6.94	6.0
				8.0
2	TMD1a	0.0151	5.07	4.75
	TMD1b	0.0221	4.91	6.0
3	TMD1	0.0337	4.14	4.25
	TMD2	0.0110	5.12	6.25

natural frequencies, the optimum TMD frequency may no longer be near a single natural frequency, but between natural frequencies. This may explain why some of the TMD frequencies are between the frequencies of two natural modes of vibration of the bays. In further reviewing the actual TMD parameters, it should be noted that the TMD-to-floor mass ratios of TMD2 and TMD3 for Bay 1 are larger than the mass ratios of similar TMDs on the other two bays. Bay 2 has no TMDs to control the higher modes of vibration. The higher modes of vibration have a significant effect on the floor vibration, and in order to gain significant control of the floor vibrations, modes other than the fundamental mode of vibration need to be controlled.

CHAPTER 5

COMPUTER ANALYSIS OF FLOOR RESPONSE WITH TUNED MASS DAMPERS AS PART OF THE FLOOR MODEL

The floors on which the tuned mass dampers were tested were modeled and analyzed using SAP90 v. 6.0 (Wilson and Habibullah 1995). These floor models were introduced in Chapters 3 and 4. Eigenvalue analysis of the models gave information concerning vibrational mode shapes, and equivalent SDOF floor mass for each mode of vibration of the floors. SAP90 was used further to determine floor response with TMDs in place.

5.1 Tuned Mass Damper Model

TMD models were added to the floor models. These models shall be referred to as analytical TMDs from this point forward. Figure 5.1 shows the configuration for the analytical TMD. The analytical TMD was designed so that it attaches to a single node of the floor. It has the spring in parallel with the damping element to support the mass. The support members are modeled as stiff elements of no weight. The spring stiffness, k_s , damping element stiffness, k_d , and damping coefficient, c , values are the values of their respective final TMD parameters. The mass of the mass plate was varied in the models. As with the prototype TMDs, the mass was changed to “optimally tune” an analytical TMD.

Before placing a TMD model on a floor model, a TMD model was analyzed using SAP90. The nodes of the TMD were restrained so that they could only move vertically, and constraints were placed on the nodes at either end of the mass plate. These constraints ensured that the mass plate had the same vertical displacement along its entire length. Without this constraint,

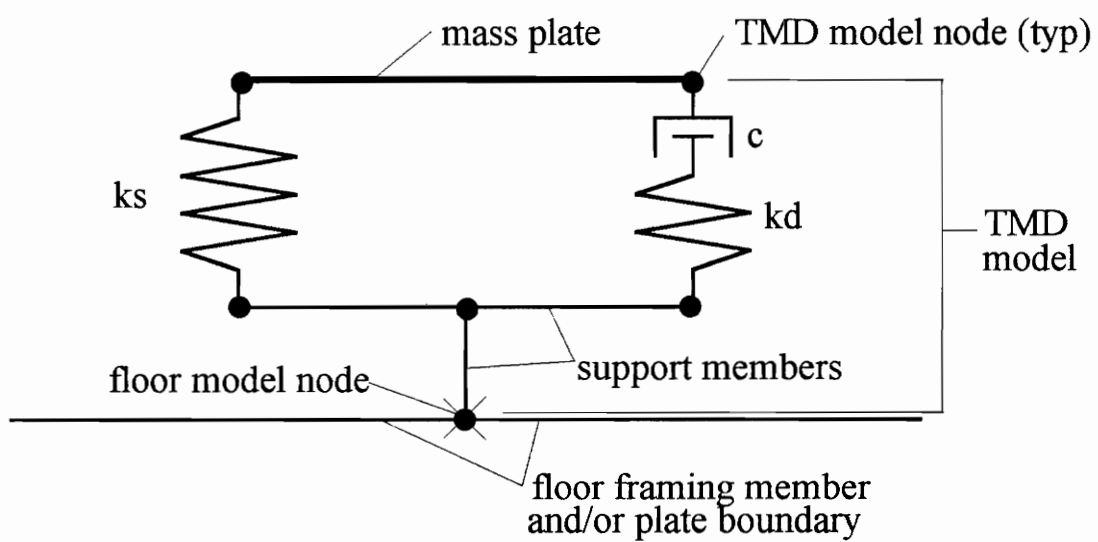


Figure 5.1 Elevation View of a SAP90 TMD Model

the dissimilarity between the members supporting either end of the mass plate resulted in unequal displacement of the ends of the TMD mass plate. The calculated TMD natural frequency of 4.73 Hz, determined using Equation 2.5, was compared with the natural frequency determined using SAP90, 4.73 Hz. The TMD model performed as expected, with the calculated frequency agreeing with the frequency determined using SAP90.

5.2 Analysis Method

Damping had to be included in the analysis of the floor with TMDs. The damping ratios for each mode of vibration were not individually added to the computer model. A damping ratio of 1.5% was assumed for all of the modes of vibration of the laboratory floor. This is slightly less than the average damping ratio of the first three modes of vibration (see Section 3.1.2.2). A damping ratio value less than average was used because Hanagan (1994) found the damping ratios of the laboratory floor to be less than the damping ratios reported in Section 3.1.2.2, and this was considered when choosing the damping ratio for the laboratory floor model. For the office floor bays, only the damping ratios for the fundamental modes of vibration were determined. The higher frequencies were so closely spaced and the damping ratio so high, that use of the half power method to calculate damping ratios was not possible. The average damping ratio for the fundamental frequencies is approximately 5%. A damping ratio of 5% was applied to all of the modes of vibration of the SAP90 models of the office floor bays.

Because damping now had to be included in the floor and TMD models, an eigenvalue analysis of the floor would no longer suffice. A dynamic time history analysis of the floor was performed using SAP90. To initiate this dynamic time history analysis, a dynamic impact was imparted on the model. This dynamic impact was a decreasing ramp function, which was similar

to the actual forcing function produced by a heel drop (Murray 1975). The heel drop ramp function decreased linearly from a 600 lb force to a 0 lb force in a space of 50 milliseconds. The time history analysis was made to span 8 seconds with 1024 data points recorded in the 8 second span.

The RMS of the acceleration data resulting from this dynamic analysis was calculated, and used to determine the optimum TMD parameters. Taking the RMS of the data takes into account the effect of damping on the structure over the 8 second time period. The greater the damping in a system, the sooner the acceleration decreases, and the smaller the resulting RMS of the acceleration data. A series of models, in which only the TMD mass changed, was analyzed. The minimum RMS acceleration in the series of models was chosen as the analytical TMD with “optimum” mass.

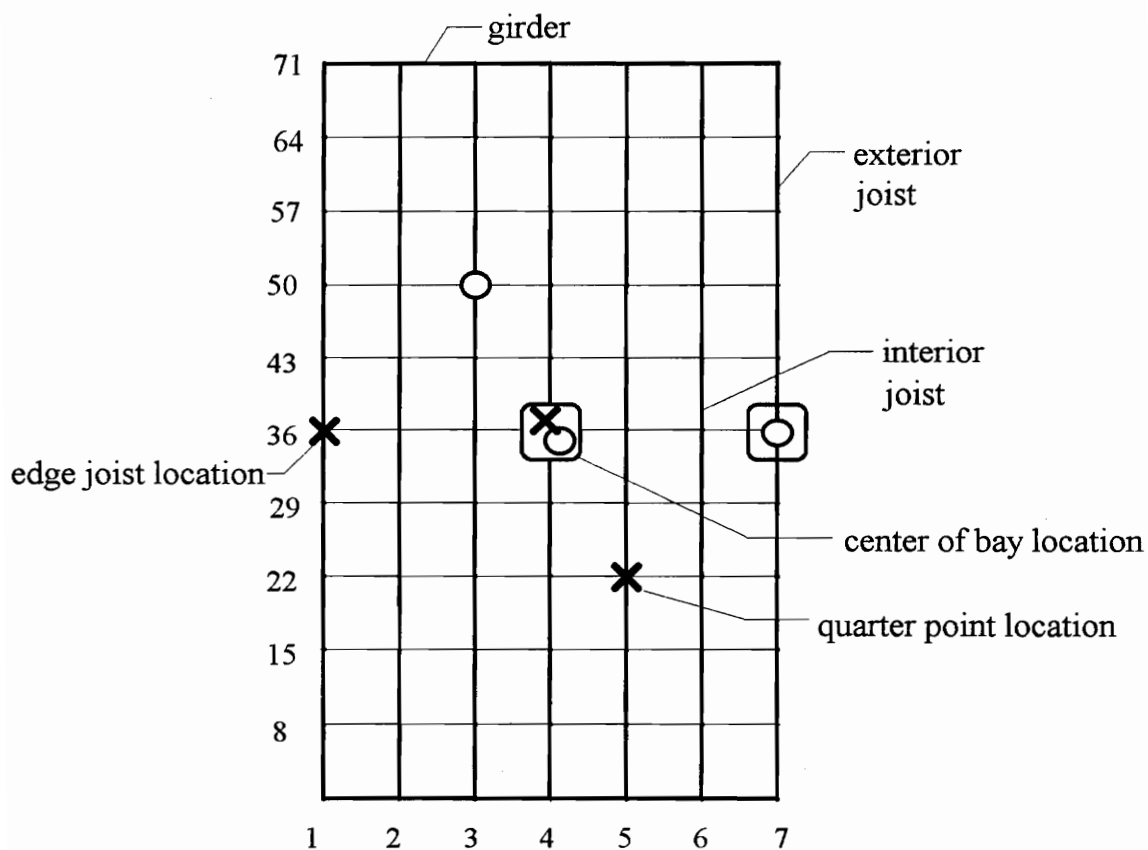
5.3 Laboratory Floor Model

Two analytical TMDs were placed on this floor model to simulate the prototype TMDs used to control the first and second modes of vibration of the laboratory floor.

5.3.1 Determination of “Optimum” TMD Mass

TMD1 was first placed on the floor model, and its “optimum” mass determined. After the “optimum” mass of TMD1 was determined, it remained in the model, and TMD2 was placed on the floor model, so its “optimum” mass could be determined.

Figure 5.2 shows the grid of the SAP90 laboratory floor model and locations at which heel drops were placed and acceleration time history data was calculated by SAP90. Acceleration traces from the three locations, which result from heel drops at their three specified locations, were recorded and studied for the bare floor condition. Plots in the frequency domain of the acceleration



✕ denotes location where
heel drop was placed

○ denotes location where acceleration
data was caculated by SAP90

□ denotes TMD location

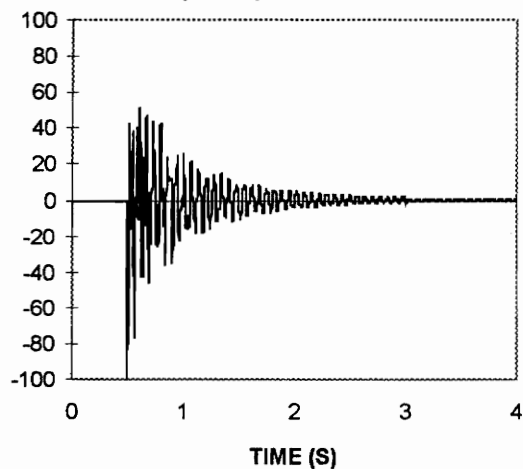
Figure 5.2 SAP90 Layout of the Laboratory Floor and Important Locations

traces were studied to determine which heel drop location and acceleration data calculation point would be the best for determining the “optimum” mass of the TMD. After examining the frequency domain data, it was decided that a heel drop at the center of the bay and a record of the acceleration at that point was the best for determining the “optimum” mass of TMD1. To determine the “optimum” mass for TMD2, acceleration data resulting from a heel drop at a quarter point and recorded at an edge joist was studied. This combination was chosen for TMD2 optimization because the plot in the frequency domain resulting from a heel drop on the bare floor shows peaks for the first and second modes of vibration of similar magnitude. Some of the other plots in the frequency domain resulting from a heel drop on the bare floor show a frequency spectrum amplitude for only one of the floor’s natural frequencies, or a frequency spectrum amplitude of one natural frequency much greater in magnitude than the frequency spectrum amplitude of another natural frequency. Figures 5.3a and 5.4a show the bare floor acceleration trace and corresponding frequency plots used in comparing the bare floor response with the response resulting from “optimum” TMD1 and “optimum” TMD1 and TMD2 in place, respectively.

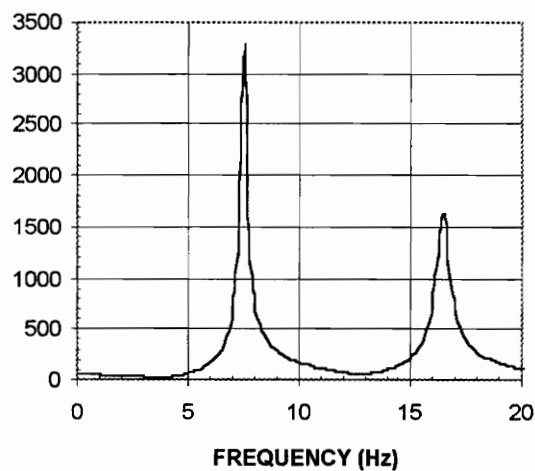
5.3.1.1 TMD1

Analytical TMD1, designed to control the first mode of vibration, was located at the center of the floor, where the maximum displacement for the first mode of vibration occurs. The prototype TMD1 for the laboratory floor was also placed at the floor center to control the first mode of vibration. The stiffness and damping parameters used for the analytical TMD were the same stiffness and damping parameters used in the laboratory floor prototype TMD1. These

ACCELERATION (IN/S²)

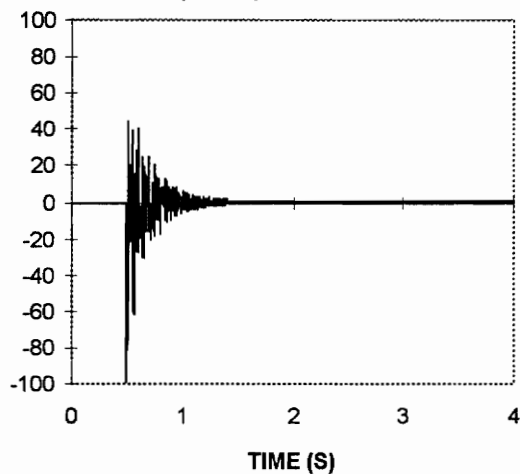


FREQUENCY SPECTRUM AMPLITUDE

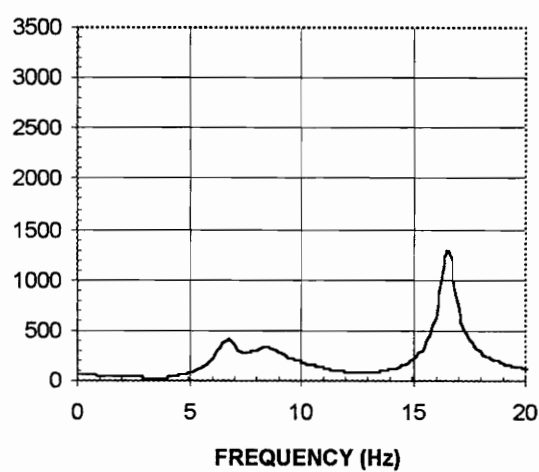


a - bare floor

ACCELERATION (IN/S²)

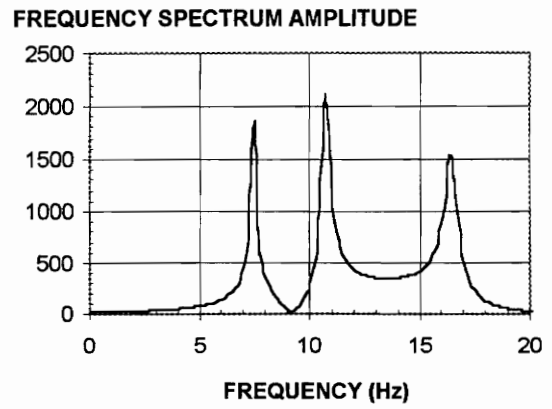
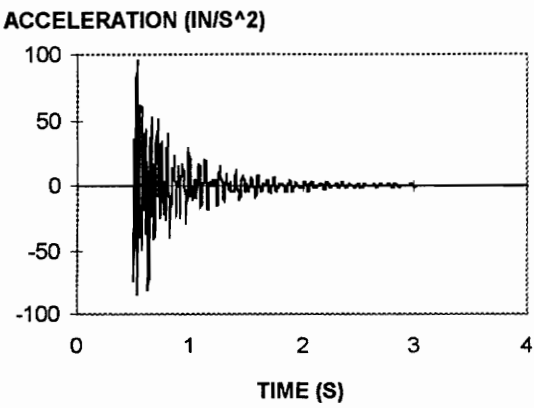


FREQUENCY SPECTRUM AMPLITUDE

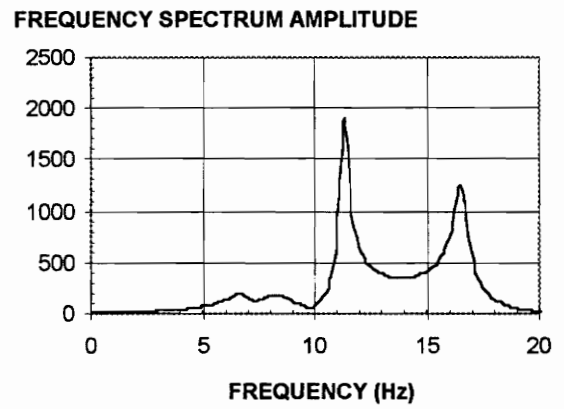
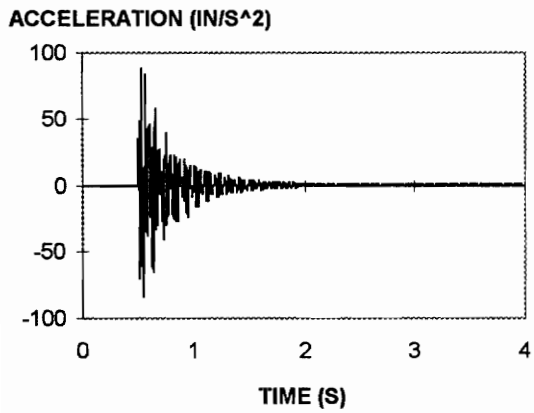


b - TMD1 in place

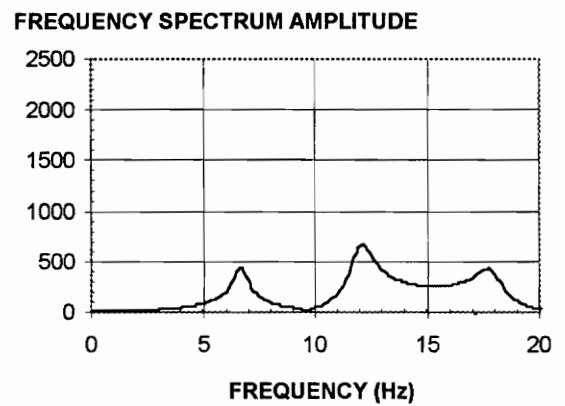
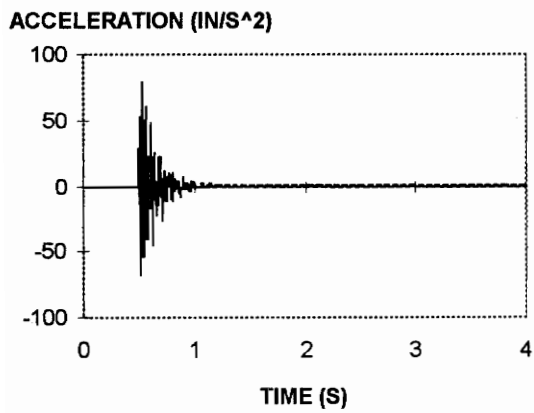
Figure 5.3 Heel Drop Located and Acceleration Calculated at the Floor Center



a - bare floor



b - TMD1 in place



c - TMD1 and TMD2 in place

Figure 5.4 Heel Drop Located at a Quarter Point and Acceleration Calculated at an Edge Joist

parameters are found in Table 3.2. The SAP90 input for the laboratory floor with analytical TMD1 is in Appendix E.

To determine the “optimum” TMD mass, the RMS of the floor acceleration at the center of the bay resulting from a heel drop at the center of the bay was determined for models with various TMD mass. Figure 5.5 shows a graph of RMS acceleration versus the TMD1-to-floor mass ratio. Table 5.1 shows the data used in creating this plot and the TMD frequencies associated with each TMD mass. To calculate the final frequency of the TMD model, Equation 2.11, which takes into account the system damping ratio, was used. The TMD damping ratio is based on the spring and damper stiffness, the damping coefficient, and the TMD mass. This damping ratio is determined using Equations 2.9 and 2.10. The TMD mass is not shown in Table 5.1; rather, the number of 10 lb mass plates placed on the TMD model and the corresponding TMD weight are shown.

The optimal TMD was determined by observing the trend of the RMS of the acceleration time history versus the TMD-to-floor mass ratio. The mass ratio resulting in the lowest RMS acceleration was determined to be the “optimum” mass ratio. The “optimum” TMD1 has a weight of 285 lbs, resulting in a mass ratio of 0.0445 and a floor RMS acceleration of 6.61 in/s^2 . The RMS acceleration of the bare floor was 10.13 in/s^2 . The frequency of this TMD is 7.30 Hz, slightly less than the floor fundamental frequency of 7.35 Hz. The 7.30 Hz frequency is reasonable and expected; a calculated optimal TMD frequency is generally 95% - 99% of the floor frequency which it is meant to control (Bachmann et al. 1995).

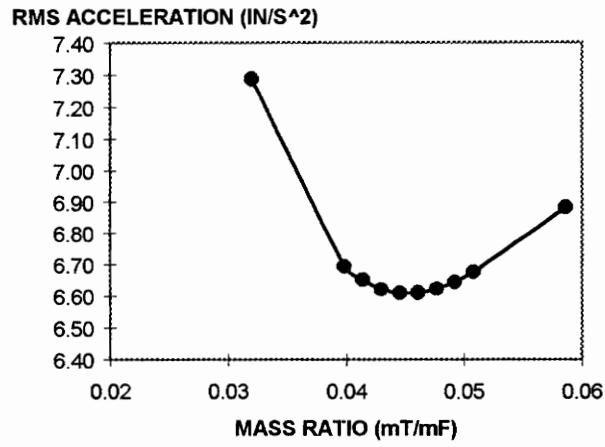


Figure 5.5 Variance in RMS Acceleration with respect to Laboratory TMD1-to-Floor Mass Ratio

Table 5.1 Data Pertaining to the Determination of the “Optimum” Laboratory Floor TMD1

Number of Plates	TMD Wt. (lbs)	Floor Wt. (lbs)	Mass Ratio (m _T /m _F)	RMS Accel. (in/s ²)	TMD1 Freq. (Hz)
13	205	6398	0.0320	7.287	8.60
18	255	6398	0.0399	6.695	7.72
19	265	6398	0.0414	6.650	7.57
20	275	6398	0.0430	6.620	7.43
21	285	6398	0.0445	6.609	7.30
22	295	6398	0.0461	6.610	7.18
23	305	6398	0.0477	6.623	7.05
24	315	6398	0.0492	6.645	6.94
25	325	6398	0.0508	6.674	6.83
30	375	6398	0.0586	6.881	6.36

TMD1 effects on floor vibration characteristics

An eigenvalue analysis was performed on the laboratory floor with the “optimum” TMD1 included in the model. TMD1 caused a shift in the natural frequencies of the floor system. Two eigenfrequencies, one below and one above 7.35 Hz, replaced the bare floor fundamental frequency. These frequencies, 6.60 Hz and 8.13 Hz, are the split frequencies resulting from the placement of the TMD to control the first mode of vibration. The bare floor second natural frequency shifted from 10.59 Hz to 11.19 Hz. The shift of the second bare floor frequency can be seen by comparing the plots in the frequency domain of Figures 5.3a and 5.3b. Figure 5.3a shows the bare floor response, and Figure 5.3b shows the floor response with TMD1 in place.

Figure 5.6 shows the first four modes of vibration with the “optimum” TMD1 on the floor. These modes of vibration can be compared with the bare floor mode shapes of Figure 3.3 to see how the TMD affects the floor response. With TMD1 in place, there are two mode 1 shapes, so to speak. The bare floor mode 2 and 4 shapes correspond with the modes 3 and 4 shapes of the floor with TMD1.

5.3.1.2 TMD2

Analytical TMD2, designed to control the second mode of vibration, was located at the center of one of the edge joists of the bay. The prototype TMD2 for the laboratory floor was also placed at the center of one of the edge joists. The stiffness and damping parameters used for this model were the same parameters used in the prototype laboratory TMD2. These parameters are found in Table 3.2.

The RMS of the floor acceleration was determined for models with various TMD2 mass. The TMD1 mass was held constant while the TMD2 mass was varied. To determine the minimum

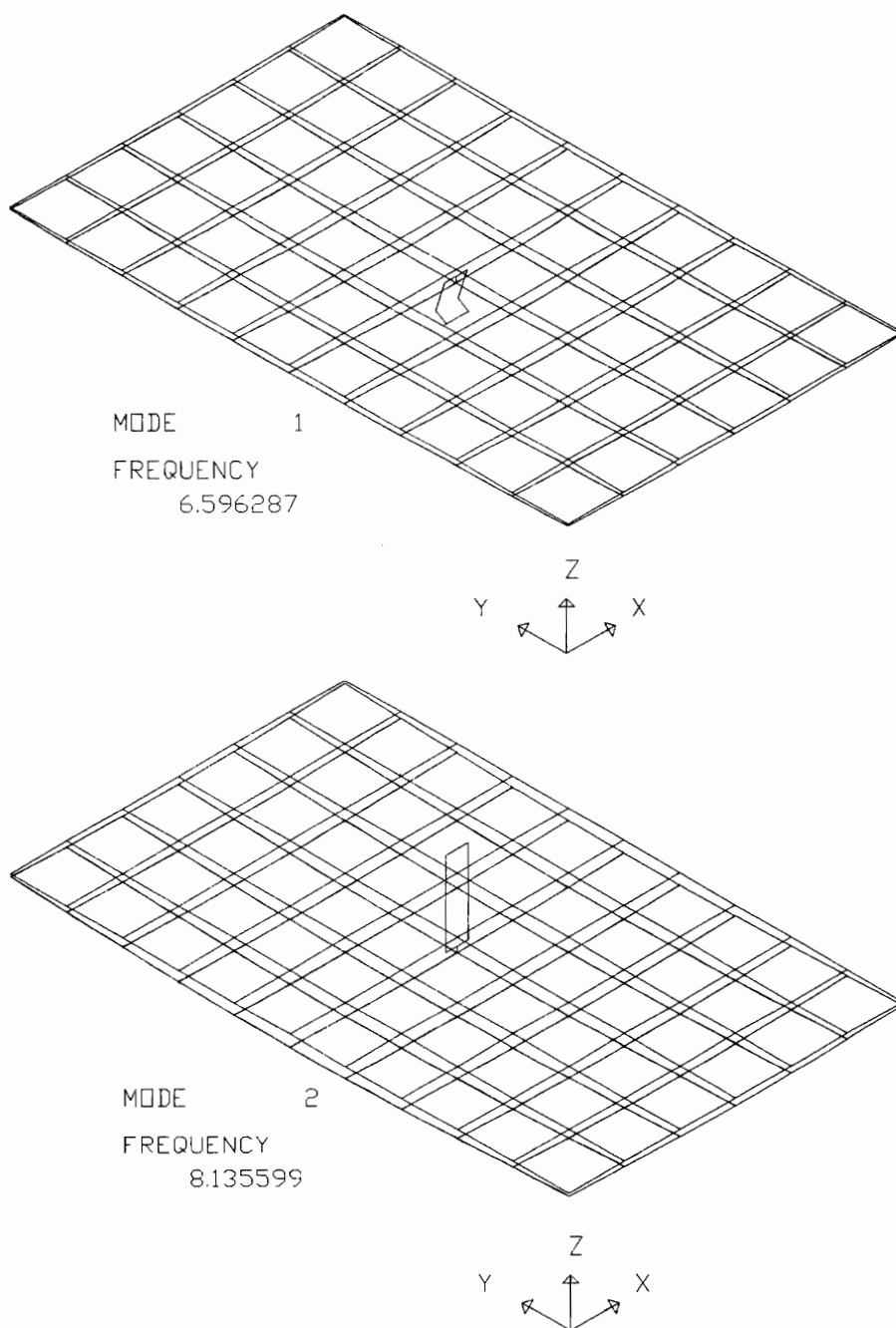


Figure 5.6 First Four Mode Shapes From SAP90 Analysis of Laboratory Floor With TMD1 in Place

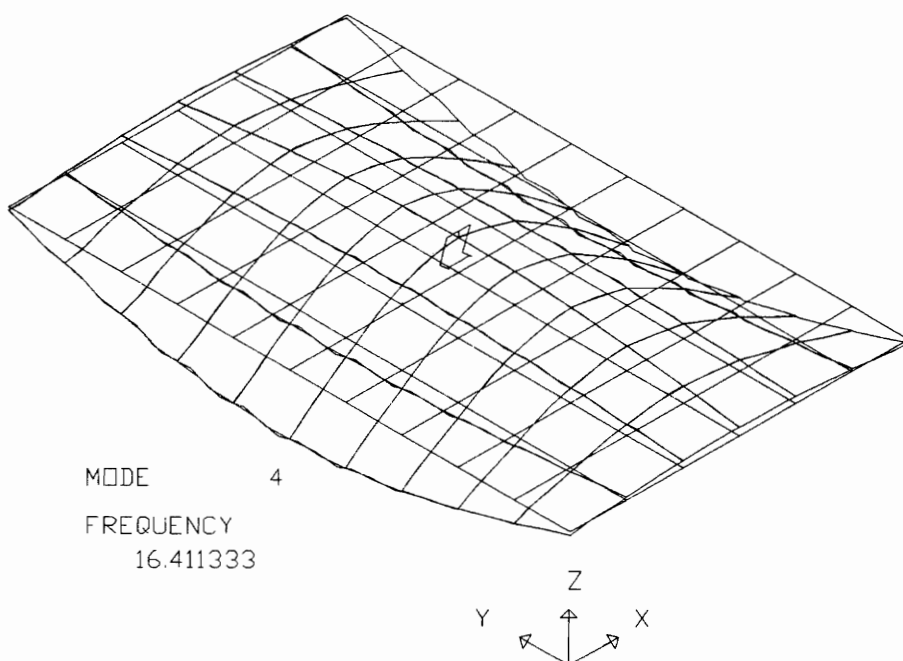
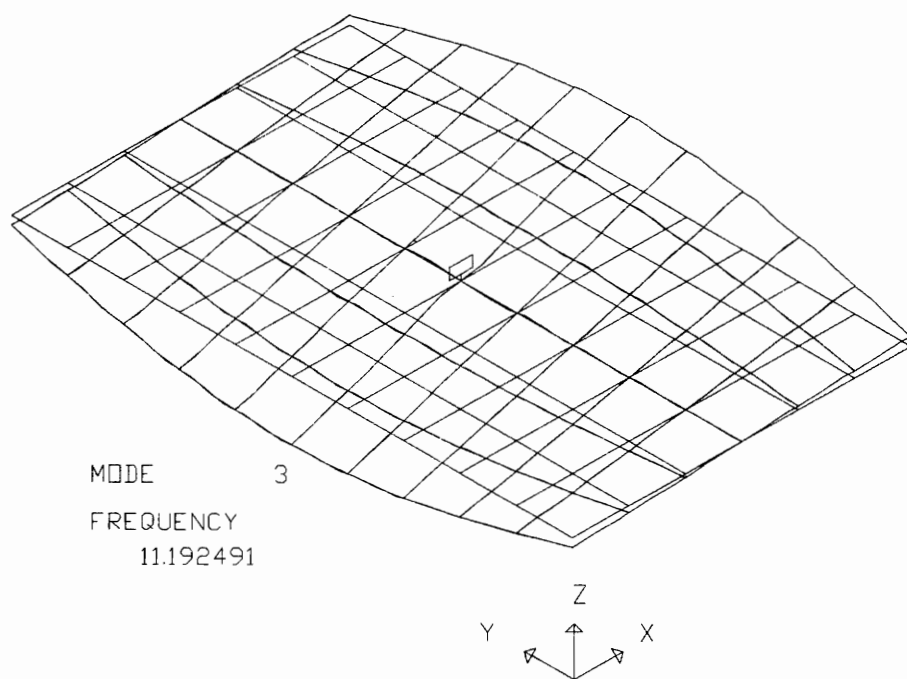


Figure 5.6 First Four Mode Shapes of Laboratory Floor with TMD1 in Place, continued

RMS acceleration, acceleration data resulting from a quarter point heel drop and calculated at an edge joist was recorded. Figure 5.7 shows a plot of RMS acceleration versus the TMD2-to-floor mass ratio. Table 5.2 shows the data used in creating this plot, and the TMD frequencies associated with each mass.

The “optimal” TMD has a mass ratio of 0.1420, and RMS acceleration of 6.76 in/s^2 . This mass ratio is larger than the suggested mass ratio of 0.02 - 0.0667 (Bachmann et al. 1995). To study the effect of spring stiffness on “optimum” mass ratio, a series of models was studied with a spring and damper stiffness half that of the prototype TMD2 spring and damper stiffness. With the stiffness reduced by a factor of two, the mass ratio was reduced to 0.0852. With this smaller mass, the RMS acceleration was higher at a value of 6.82 in/s^2 . This “optimum” analytical TMD2 model resulted in a frequency similar to the other analytical TMD2 frequency; therefore TMD frequency controlled the optimum TMD design, and not the mass or stiffness of the TMD.

The “optimum” TMD2 with prototype laboratory TMD2 parameters (i.e. a mass ratio of 0.1420) has a frequency of 8.40 Hz. This frequency is much lower than the frequency of the floor which it is meant to control. The analytical second mode floor frequency is 10.59 Hz on a bare floor and 11.19 Hz on a floor with TMD1 in place. A frequency above 9 Hz was expected. As can be seen by comparing Figures 5.4a, 5.4b, and 5.4c, TMD2 does reduce the energy in the frequencies related to the second and third modes of vibration. RMS acceleration at this location was reduced from 10.91 in/s^2 to 9.55 in/s^2 with TMD1 in place, and finally reduced to 6.76 in/s^2 with TMD1 and TMD2 in place.

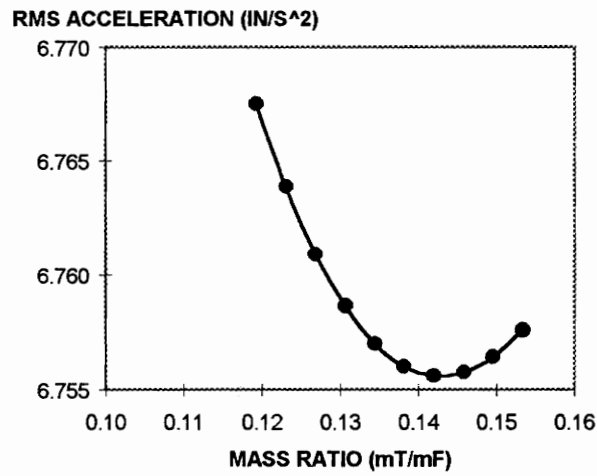


Figure 5.7 Variance in RMS Acceleration with respect to Laboratory TMD2-to-Floor Mass Ratio

Table 5.2 Data Pertaining to the Determination of the “Optimum” Laboratory Floor TMD2

Number of Plates	TMD Wt. (lbs)	Floor Wt. (lbs)	Mass Ratio (m_T/m_F)	RMS Accel. (in/s^2)	TMD2 Freq. (Hz)
24	315	2641	0.1193	6.768	9.17
25	325	2641	0.1231	6.764	9.03
26	335	2641	0.1268	6.761	8.89
27	345	2641	0.1306	6.759	8.76
28	355	2641	0.1344	6.757	8.64
29	365	2641	0.1382	6.7560	8.52
30	375	2641	0.1420	6.7556	8.40
31	385	2641	0.1458	6.7558	8.29
32	395	2641	0.1496	6.756	8.19
33	405	2641	0.1534	6.758	8.09

5.4 Bay 2 Model of the Office Floor

Two TMDs were modeled to simulate the two prototype TMDs placed on Bay 2 to control the first mode of vibration. This floor was the first of two office bay floors to be studied due to the fact that only one mode of vibration was controlled in this bay.

5.4.1 Determination of “Optimum” TMD Mass

Analytical TMD1a was first included in the Bay 2 floor model, and its “optimum” mass determined. Then TMD1b was added to the model and its “optimum” mass was determined.

Figure 5.8 shows the grid of the SAP90 Bay 2 floor model and locations at which heel drops were placed and acceleration time histories calculated. By looking at the various bare floor acceleration traces and their respective frequency plots, it was decided that acceleration data resulting from a heel drop at the center of the bay should be calculated at a node 12 feet north of the center of the bay, node 76. The resulting acceleration trace and plot in the frequency domain clearly show the fundamental frequency of 4.82 Hz, as seen in Figure 5.9a.

5.4.1.1 TMD1a

TMD1a was located at the center of the floor. The maximum displacement for the mode 1 vibration of the model does not occur at the center of the floor, but at a point 9 ft north of the center of the bay, node 75. Analytical TMD1a was located at the center of the Bay 2 model because the prototype TMD1a was located at the center of the bay during the experiments on the in-situ floor. The stiffness and damping parameters used for this TMD model were the same stiffness and damping parameters used in the prototype Bay 2 TMD1a. These parameters are found in Table 4.5.

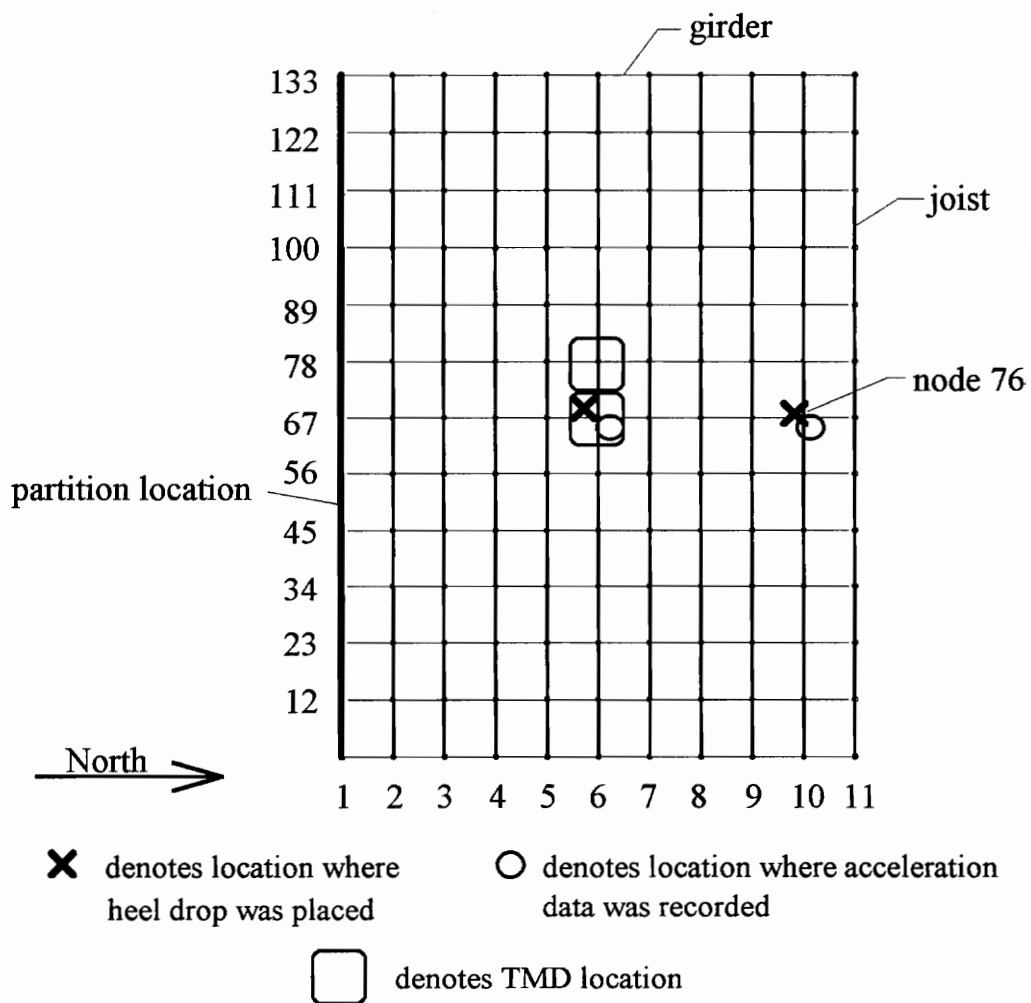
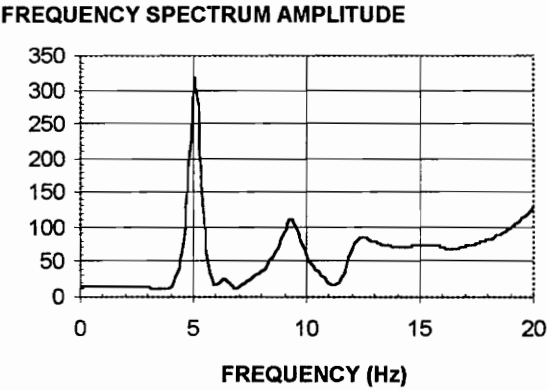
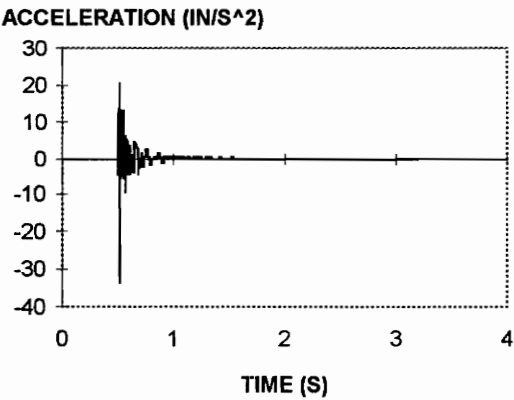
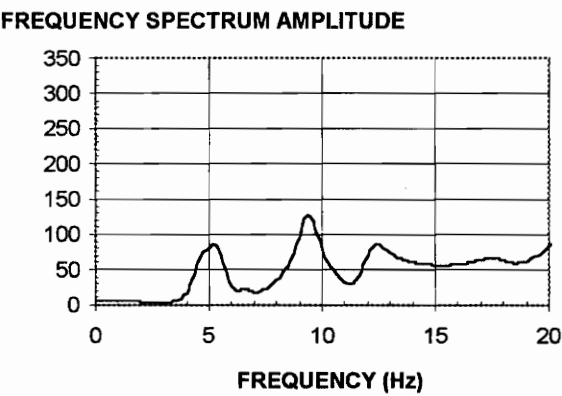
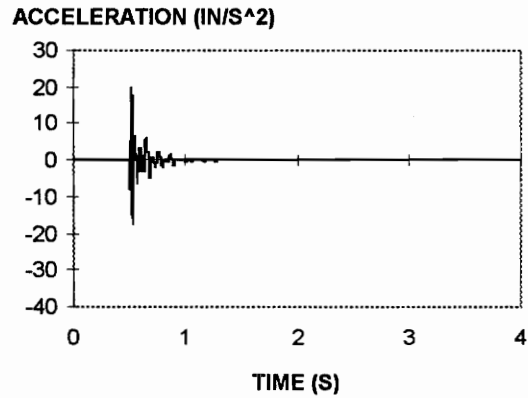


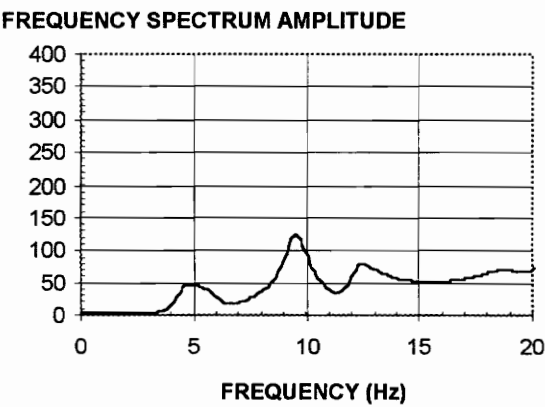
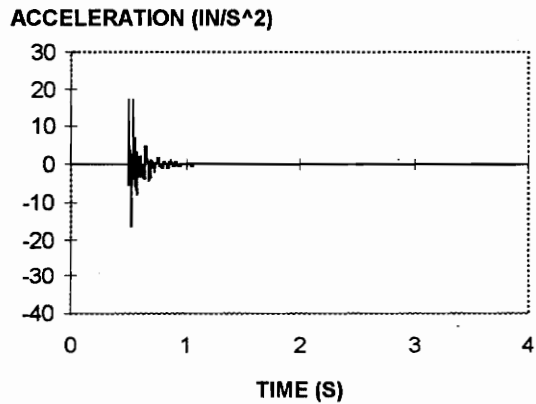
Figure 5.8 SAP90 Layout of Office Floor Bay 2 and Important Locations



a - bare floor



b - TMD1a in place



c - TMD1a and TMD1b in place

Figure 5.9 Heel Drop Placed at the Center of Bay 2 and Acceleration Calculated at Node 76

Figure 5.10 shows the RMS of the acceleration versus the TMD1a-to-floor mass ratio. Table 5.3 shows the data corresponding with the plot and the TMD frequencies associated with each TMD mass. The “optimal” TMD has a mass ratio of 0.0160, and RMS acceleration of 1.53 in/s², which is reduced from the bare floor RMS acceleration of 2.23 in/s². The TMD frequency of 4.92 Hz is higher than the floor frequency, but only by 0.10 Hz.

Analytical TMD1a affected the Bay 2 model vibration characteristics in the same way analytical TMD1 affected the Laboratory Floor model vibration characteristics. With TMD1a in place, the fundamental natural frequency of 4.82 Hz was replaced by frequencies of 4.50 and 5.00 Hz. The bare floor second natural frequency shifted from 5.12 Hz to 5.39 Hz.

5.4.1.2 TMD1b

This TMD was located 3 ft west of the center of the floor at node 83. This is the approximate location at which the prototype Bay 2 TMD1b was placed. The stiffness and damping parameters used for this TMD model were the same stiffness and damping parameters used in prototype Bay 2 TMD1b. These parameters are shown in Table 4.5.

Figure 5.11 shows the RMS of the acceleration versus the TMD1b-to-floor mass ratio. Table 5.4 shows the data corresponding with the plot, and the TMD frequencies associated with each mass. The “optimal” TMD has a mass ratio of 0.0211, and RMS acceleration of 1.34 in/s². The frequency of this TMD is 5.02 Hz, which is higher than the bare floor fundamental frequency or the TMD1a frequency.

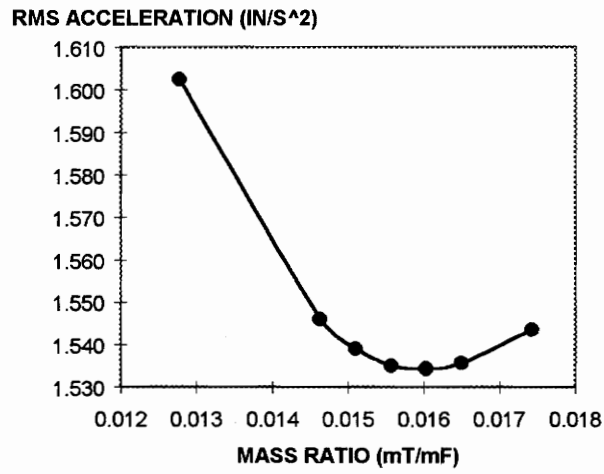


Figure 5.10 Variance in RMS Acceleration with respect to Bay 2 TMD1a-to-Floor Mass Ratio

Table 5.3 Data Pertaining to the Determination of the “Optimum” Bay 2 TMD1a

Number of Plates	TMD Wt. (lbs)	Floor Wt. (lbs)	Mass Ratio (m _T /m _F)	RMS Accel. (in/s ²)	TMD1a Freq. (Hz)
20	275	21520	0.0128	1.603	5.50
24	315	21520	0.0146	1.546	5.15
25	325	21520	0.0151	1.539	5.06
26	335	21520	0.0156	1.535	4.99
27	345	21520	0.0160	1.534	4.92
28	355	21520	0.0165	1.536	4.85
30	375	21520	0.0174	1.544	4.72

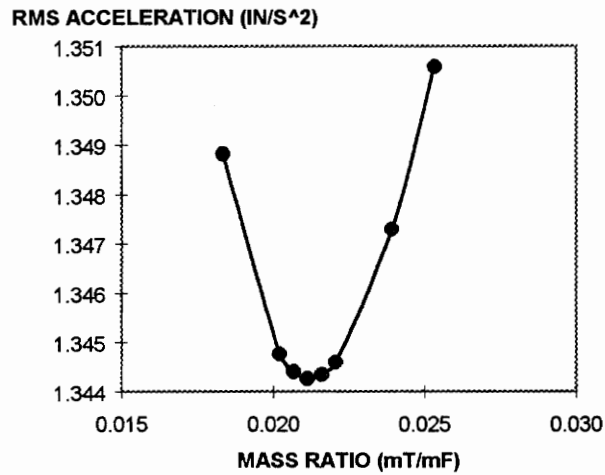


Figure 5.11 Variance in RMS Acceleration with respect to Bay 2 TMD1b-to-Floor Mass Ratio

Table 5.4 Data Pertaining to the Determination of the “Optimum” Bay 2 TMD1b

Number of Plates	TMD Wt. (lbs)	Floor Wt. (lbs)	Mass Ratio (m _T /m _F)	RMS Accel. (in/s ²)	TMD1b Freq. (Hz)
32	395	21520	0.0184	1.349	5.39
36	435	21520	0.0202	1.345	5.13
37	445	21520	0.0207	1.3444	5.08
38	455	21520	0.0211	1.34427	5.02
39	465	21520	0.0216	1.34434	4.97
40	475	21520	0.0221	1.345	4.91
44	515	21520	0.0239	1.347	4.72
47	545	21520	0.0253	1.351	4.59

5.5 Bay 1 Model of the Office Floor

Three TMDs were modeled to simulate the three prototype TMDs placed on the floor to control the first, second, and third modes of vibration. The model bare floor first, second, and third modes of vibration occur at frequencies of 5.15 Hz, 7.38 Hz, and 7.44 Hz, respectively.

5.5.1 Determination of “Optimum” TMD Mass

TMD1 was entered into the model, and its “optimum” mass determined. Figure 5.12 shows the grid layout of the SAP90 Bay 1 floor model and locations at which heel drops were placed and acceleration time histories calculated. After looking at the various bare floor acceleration traces and their respective frequency plots, it was decided that acceleration data resulting from a heel drop at the center of the bay and calculated at the center of the bay would be used to determine the “optimum” TMD1 mass. The resulting acceleration trace and plot in the frequency domain of the bare floor condition is shown in Figure 5.13a.

To determine the “optimum” mass of TMD2 and TMD3, acceleration data resulting from a heel drop placed at node 83 and calculated at node 94 was recorded. This combination was used because the plot in the frequency domain of the bare floor condition shows several frequency peaks below 10 Hz (see Figure 5.14a). TMD2 and TMD3 were designed to control the frequencies of 7.38 Hz and 7.44 Hz.

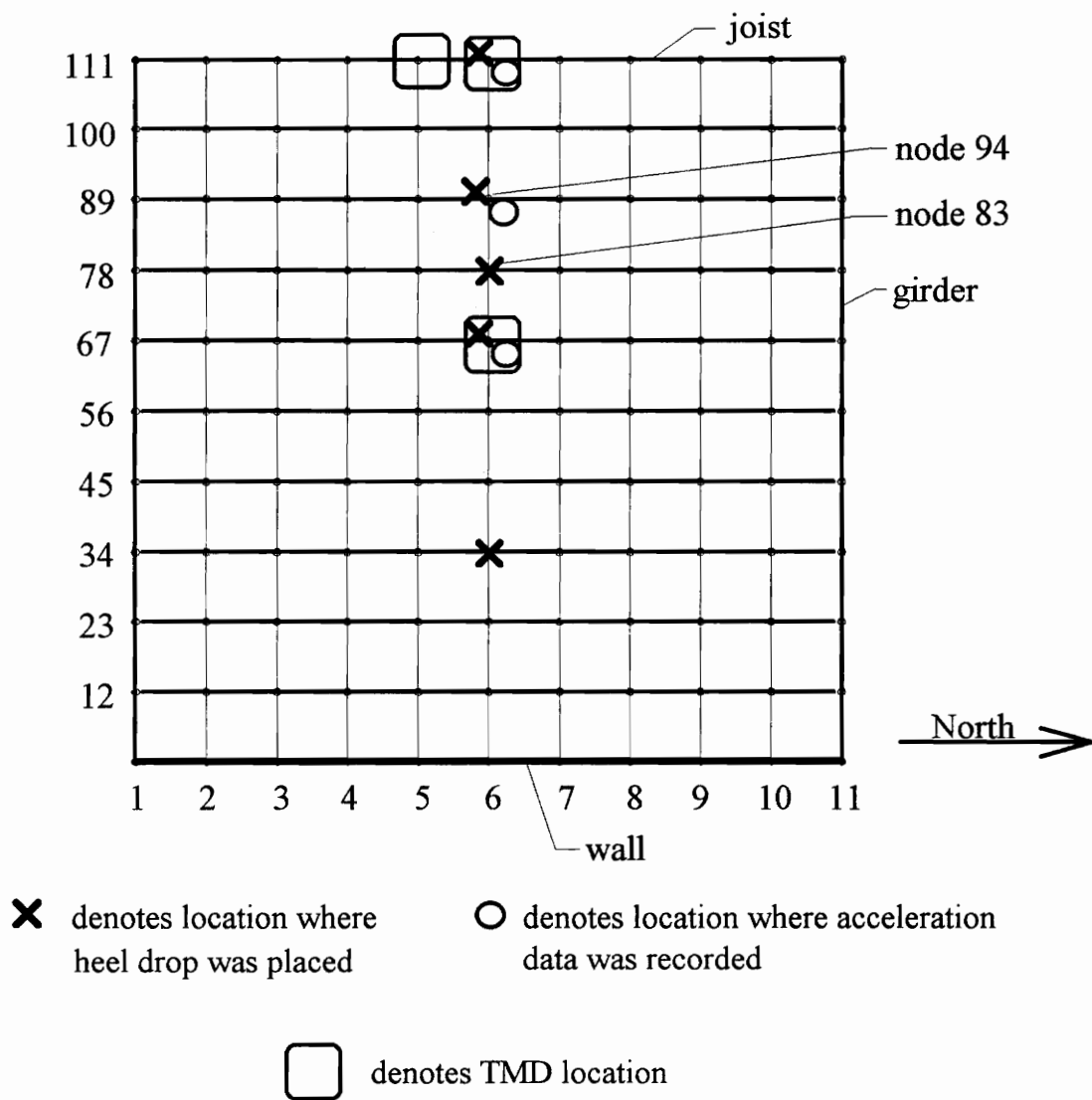
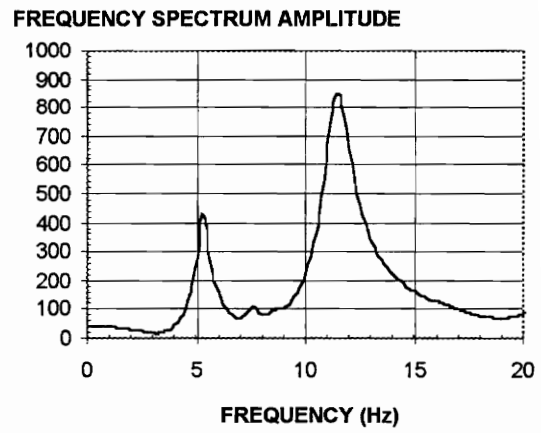
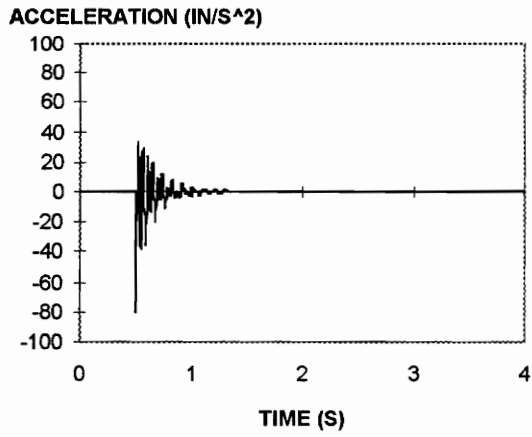
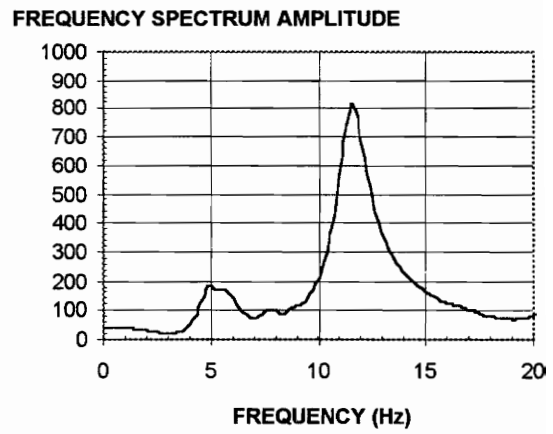
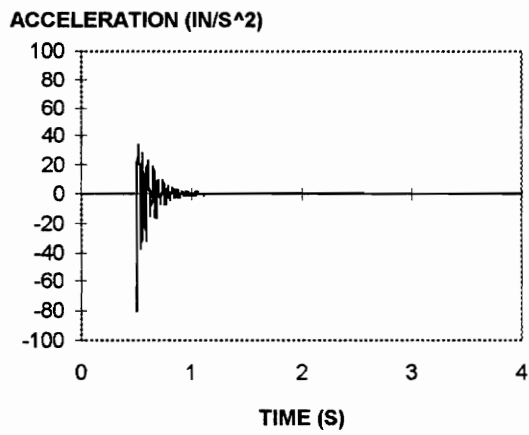


Figure 5.12 SAP90 Layout of Bay 1 and Important Locations

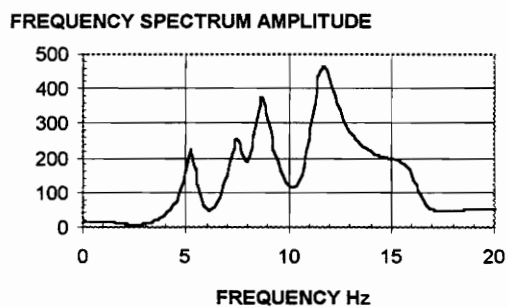
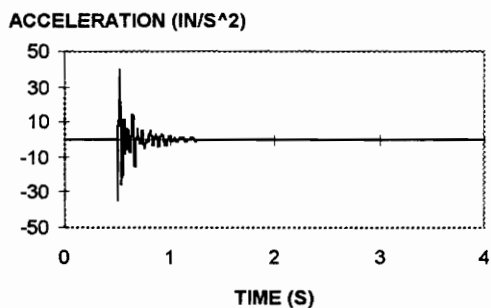


a - bare floor

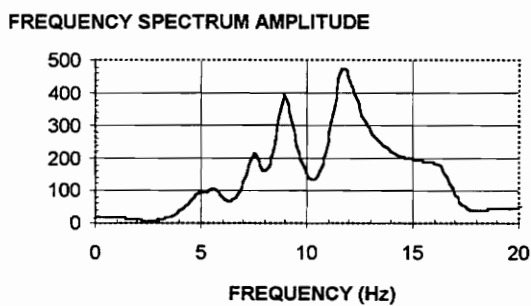
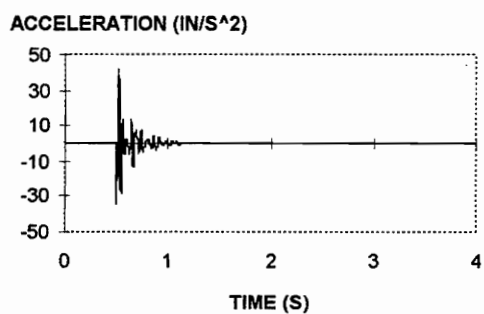


b - TMD1 in place

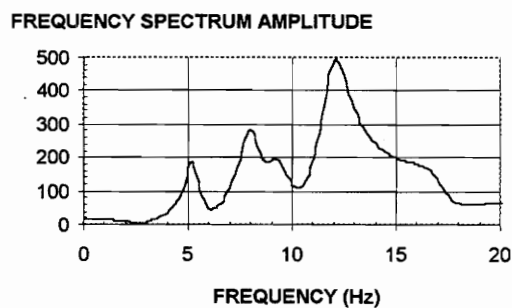
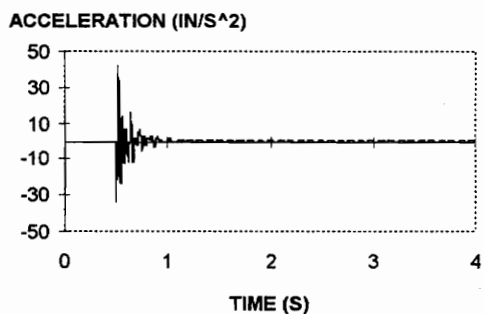
Figure 5.13 Heel Drop Placed and Acceleration Calculated at the Center of Bay 1



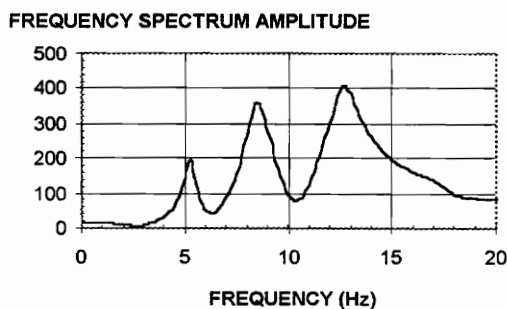
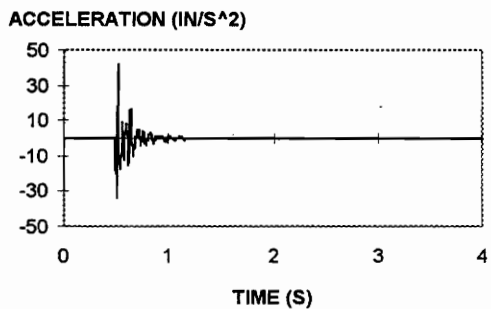
a - bare floor



b - TMD1 in place



c - TMD1 and TMD2 in place



d - TMD1, TMD2, and TMD3 in place

Figure 5.14 Heel Drop Placed at Node 83 and Acceleration Measured at Node 94 of Bay 1

5.5.1.1 TMD1

The analytical TMD1 was located at the center of the floor, where the maximum displacement for mode 1 vibration occurs. The stiffness and damping parameters used for this TMD were the same stiffness and damping parameters used in the prototype Bay 1 TMD1, and are found in Table 4.4.

Figure 5.15 shows the RMS acceleration versus the TMD1-to-floor mass ratio. Table 5.5 shows the data corresponding with the plot, and the TMD frequencies associated with each TMD mass.

The “optimal” TMD1 has a mass ratio of 0.0188 and results in a RMS acceleration of 4.89 in/s^2 , which is reduced from the bare floor RMS acceleration of 5.08 in/s^2 . This mass ratio is just slightly lower than the suggested TMD-to-floor mass ratio range of 0.02 to 0.0667 (Bachmann et al. 1995). This TMD has a frequency of 5.146 Hz, which is just slightly less than the bare floor fundamental frequency of 5.15 Hz.

5.5.1.2 TMD2

Analytical TMD2 was located at node 116, the point of maximum displacement for the second and third modes of vibration. The stiffness and damping parameters used in this TMD are in Table 4.4, and are the same stiffness and damping parameters used in prototype Bay 1 TMD2.

The RMS of acceleration data was calculated at node 94 for a heel drop at node 83. Looking at the RMS versus mass ratio plot of Figure 5.16, an “optimum” TMD mass could not be determined. Several other heel drop location and acceleration data calculation location combinations were tried, but they also did not result in an “optimum” TMD mass. The data corresponding with the plot of Figure 5.16 are in Table 5.6. A plot in the frequency domain of

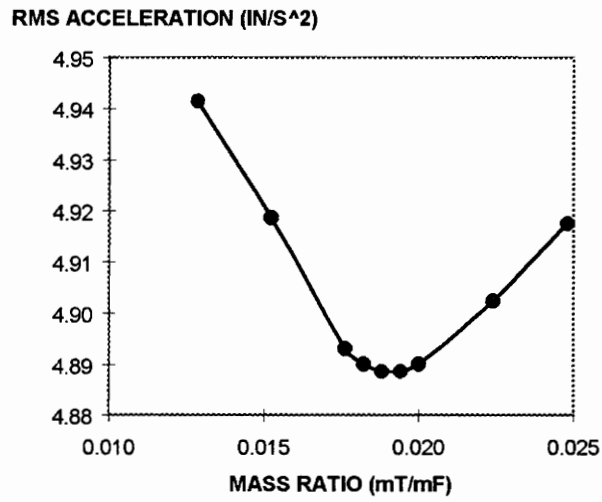


Figure 5.15 Variance in RMS Acceleration with respect to Bay 1 TMD1-to-Floor Mass Ratio

Table 5.5 Data Pertaining to the Determination of the “Optimum” Bay 1 TMD1

Number of Plates	TMD Wt. (lbs)	Floor Wt. (lbs)	Mass Ratio (m _T /m _F)	RMS Accel. (in/s ²)	TMD1 freq. (Hz)
14	215	16730	0.0129	4.942	6.23
18	255	16730	0.0152	4.919	5.72
22	295	16730	0.0176	4.893	5.32
23	305	16730	0.0182	4.890	5.23
24	315	16730	0.0188	4.88866	5.15
25	325	16730	0.0194	4.88873	5.07
26	335	16730	0.0200	4.890	4.99
30	375	16730	0.0224	4.902	4.72
34	415	16730	0.0248	4.918	4.48

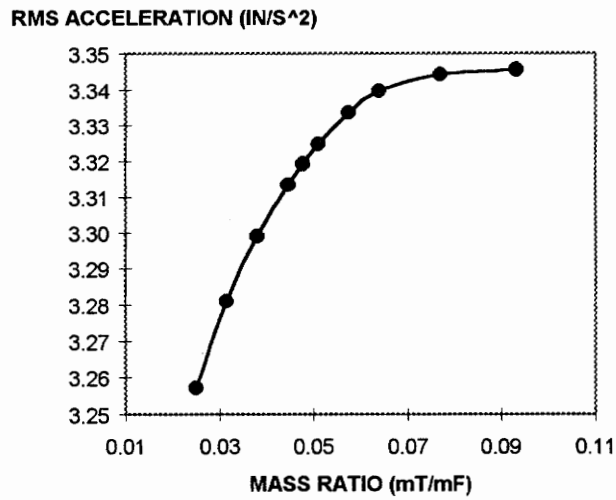


Figure 5.16 Variance in RMS Acceleration with respect to Bay 1 TMD2-to-Floor Mass Ratio

Table 5.6 Data Pertaining to the Determination of the “Optimum” Bay 1 TMD2

Number of Plates	TMD Wt. (lbs)	Floor Wt. (lbs)	Mass Ratio (m_T/m_F)	RMS Accel. (in/s^2)	TMD2 Freq. (Hz)
8	155	6171	0.0251	3.257	9.97
12	195	6171	0.0316	3.281	8.89
16	235	6171	0.0381	3.299	8.10
20	275	6171	0.0446	3.313	7.49
22	295	6171	0.0478	3.319	7.23
24	315	6171	0.0510	3.325	6.99
28	355	6171	0.0575	3.334	6.59
32	395	6171	0.0640	3.340	6.25
40	475	6171	0.0770	3.344	5.70
50	575	6171	0.0932	3.346	5.18

acceleration based on a TMD2 with 22 mass plates is in Figure 5.14c. Comparing Figure 5.14c with Figures 5.14a and 5.14b, TMD2 seems to have dampened the frequency peak just to the left of 10 Hz. With 22 mass plates, TMD2 has a mass ratio of 0.0478 and a frequency of 7.23 Hz.

The RMS acceleration calculated at node 94 increased with the placement of TMD1 and TMD2. The bare floor RMS acceleration at node 94 resulting from heel drop at node 83 was 3.30 in/s². With TMD1 in place, the RMS acceleration increased to 3.35 in/s². With TMD2 in place the RMS acceleration decreased to 3.32 in/s², which is still not below the bare floor RMS acceleration. The TMDs could cause a larger response in the higher modes of vibration (i.e. frequencies larger than 10 Hz); if this is the case, then the floor acceleration would increase, rather than decrease.

5.5.1.3 TMD3

A third TMD was modeled to determine if it could help control the floor vibrations. This TMD was placed at node 115, 3 ft south of TMD2. This TMD has the same damping and stiffness parameters of the prototype Bay 1 TMD3, and the parameters can be found in Table 4.4.

As with TMD2, an “optimum” TMD mass could not be determined for TMD3. Figure 5.17 shows the RMS acceleration versus TMD3-to-floor mass ratio, and Table 5.7 shows the corresponding data and frequencies. The TMD3 model with 16 mass plates was chosen because of its low RMS acceleration, and the mass ratio was comparable with the actual TMD3 mass ratio of 0.03706. The model TMD3 has a mass ratio of 0.0381 and a frequency of 8.04 Hz. The RMS acceleration was reduced to 3.22 in/s², which is less than the bare floor RMS acceleration of 3.30 in/s².

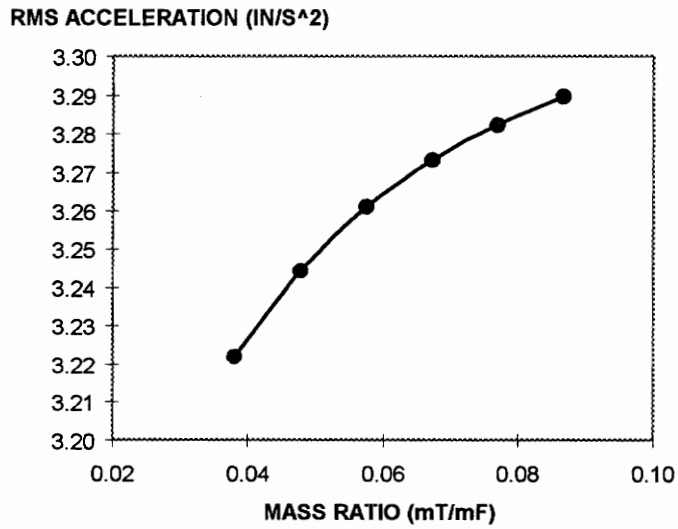


Figure 5.17 Variance in RMS Acceleration with respect to Bay 1 TMD3-to-Floor Mass Ratio

Table 5.7 Data Pertaining to the Determination of the “Optimum” Bay 1 TMD3

Number of Plates	TMD Wt. (lbs)	Floor Wt. (lbs)	Mass Ratio (m _T /m _F)	RMS Accel. (in/s ²)	TMD2 Freq. (Hz)
16	235	6171	0.0381	3.222	8.04
22	295	6171	0.0478	3.244	7.17
28	355	6171	0.0575	3.261	6.54
34	415	6171	0.0673	3.273	6.05
40	475	6171	0.0770	3.282	5.65
46	535	6171	0.0867	3.290	5.33

5.6 Summary and Discussion of Results

SAP90 was used to analytically determine the vibration characteristics of the Laboratory Floor and Bay 1 and Bay 2 of the Office Floor with TMDs in place. The goals of these computer studies were to determine if TMDs could be analytically “optimized” and to examine how their placement would affect the floor response and vibration characteristics.

The proposed TMDs for a floor were “optimized” one at a time. A series of models with a varying TMD mass was analyzed and the model which resulted in the lowest RMS acceleration response was chosen as the “optimum” TMD for a floor. After one “optimal” TMD was determined for a floor, the next TMD was placed on the floor and its mass varied while the “optimum” TMD remained in place. Once all of the proposed TMDs were placed on the floor and their “optimum” mass determined, traces of the calculated acceleration data and their corresponding frequency plots were studied to see the improvement made with the TMDs in place.

For each floor model, locations of heel drops and locations for acceleration data calculation differed. The locations were chosen based on what was shown in the plots in the frequency domain resulting from acceleration calculations and heel drop placement at these various locations. If for the frequencies of concern relatively strong frequency peaks were shown in the plots in the frequency domain, the heel drop and acceleration locations which resulted in the acceleration that was transformed to this frequency plot were chosen. For all three floors studied, the RMS acceleration at the point of calculation was reduced when all of the TMDs were in place.

As the number of TMDs increased, the RMS acceleration at the point of calculation decreased for the Laboratory Floor. Only two TMDs were placed on this floor model. A third TMD was not placed on the model to control the third natural frequency of 15.20 Hz, as was done on the experimental floor. Controlling frequencies below 10 Hz was of main concern; however, the

two TMDs did result in a decreased vibration response for the third mode of vibration, as well. The RMS acceleration at an edge joist decreased from 10.91 in/s^2 for the bare floor to 6.76 in/s^2 with TMD1 and TMD2 in place.

The RMS acceleration at the point of calculation decreased steadily as TMDs were added to the Office Floor Bay 2 model. Two TMDs were placed on the floor model to simulate the two prototype TMDs used to control the first mode of vibration. Together these TMDs reduced the RMS acceleration from 2.23 in/s^2 for the bare floor to 1.34 in/s^2 with both TMDs in place.

Three TMDs were placed on the Office Floor Bay 1 model to simulate the three prototype TMDs used experimentally on Bay 1. With TMD1 in place on Bay 1, the calculated RMS acceleration decreased at the center of the bay, but increased at the node 94 location, which is 6 ft west of the center of the bay. RMS acceleration was calculated at node 94 because the plot in the frequency domain for the acceleration response at node 94 shows the fundamental frequency and other higher mode frequencies of concern. It was desired that these frequencies be controlled. The RMS acceleration at the center of the bay decreased with the placement of TMD1, but the RMS acceleration at node 94 increased with the placement of TMD1. From this maximum RMS acceleration at node 94 with TMD1 in place, the RMS acceleration decreased with the placement of TMD2 and TMD3. However, the RMS acceleration at node 94 was not below the bare floor RMS acceleration at node 94 until all three TMDs were in place. The RMS Acceleration at node 94 decreased from 3.30 in/s^2 to 3.22 in/s^2 with three TMDs in place. An “optimum” mass for TMD2 and TMD3 could not be determined based on the criteria used to determine the other “optimum” TMDs.

The frequencies of the analytical, “optimal” TMDs and calculated floor frequencies (SAP90 frequencies) are shown in Table 5.8. As can be seen, frequencies of the first TMD placed

Table 5.8 Calculated TMD Frequencies and Analytically Determined Floor Frequencies of the Laboratory Floor, and Bay 2 and Bay 1 of the Office Floor

Bay	TMD Number	TMD Damped Frequency (Hz)	TMD Natural Frequency (Hz)	Mode	SAP90 Frequencies (Hz)	
					Bare Floor	TMD1(a) in place
Laboratory Floor	TMD1	7.30	7.36	1	7.35	6.60
						8.14
	TMD2	8.40	8.52	2	10.59	11.19
				3	15.20	16.41
Office Floor Bay 2	TMD1a	4.92	4.93	1	4.82	4.50
	TMD1b	5.02	5.10			5.00
				2	5.12	5.39
Office Floor Bay 1	TMD1	5.15	5.16	1	5.15	4.79
						5.49
	TMD2	7.23	7.29	2	7.38	7.43
	TMD3	8.04	8.11	3	8.54	7.83

Table 5.9 Experimental and Analytical TMD and Bare Floor Frequencies for the Laboratory Floor, and Bay 2 and Bay 1 of the Office Floor

Bay	TMD Number	Experimental TMD Damped Frequency (Hz)	Analytical TMD Damped Frequency (Hz)	Mode	Experimental Bare Floor Frequency (Hz)	Analytical Bare Floor Frequency (Hz)
Laboratory Floor	TMD1/(1a)	6.50 / 5.89	7.30	1	7.375	7.35
	TMD2	9.17	8.40	2	9.375	10.59
	TMD3	15.94	-	3	16.75	15.20
Office Floor Bay 2	TMD1a	5.07	4.92	1	4.75	4.82
	TMD1b	4.91	5.02			
Office Floor Bay 1	TMD1	5.07	5.15	1	5.00	5.15
	TMD2	6.68	7.23	2	5.75	7.38
	TMD3	6.94	8.04	3	6.00	7.44

on a floor (i.e. TMD1 or TMD1a) are almost identical to the bare floor natural frequency that it is meant to control.

The Laboratory Floor TMD2 has a frequency close to the second natural frequency which resulted with TMD1 in place. When trying to obtain a TMD2 frequency that was closer to the second natural frequency of 10.59 Hz, the RMS acceleration only continued to increase. For the prototype TMD2, the frequency was nearer the floor's second natural frequency; however, there was also a third prototype TMD on the actual floor, and this could have had an effect on the prototype TMD2 frequency. Table 5.9 shows the experimental and analytical TMD frequencies and the experimental and analytical floor frequencies.

For Bay 2, TMD1b has a frequency similar to the second natural frequency which resulted with the placement of TMD1a. These TMD frequencies are also similar to the frequencies of the actual Bay 2 TMDs. These analytical results compare well with the experimental results.

The Bay 1 TMD2 and TMD3 frequencies were close to the third and fourth natural frequencies of the floor with TMD1 in place. As with the experimental work, controlling the higher modes of vibration of this office floor bay was difficult.

CHAPTER 6

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

6.1 Summary

Tuned mass dampers (TMDs) have been used with success when a floor or other structure has only one mode of vibration that requires control (Bell 1994, Webster and Vaicaitis 1992, Bachmann and Weber 1995). When two or more modes of vibration require control, effectively controlling the modes of vibration with multiple TMDs can be challenging. Floor modes of vibration that are found annoying to humans usually have a frequency below 10 Hz. Only a few cases concerning the use of TMDs to control two modes of floor vibration have been documented (Setareh and Hanson 1992a, Webster and Vaicaitis 1992). Only one case concerning the use of TMDs to control more than two modes of floor vibration has been documented (Thorton et al. 1992), and not many details were given about the design or placement of TMDs to control more than two modes of vibration. Setareh and Hanson (1992b) have stated that if modes which create annoying vibration have frequencies within 20% of each other, the use of a tuned mass damper can move the location of peak amplitudes (i.e. change the frequency of other modes of vibration). Bachmann et al. (1995) have stated that tuned mass dampers do not work well on floors with relatively high damping ratios and/or closely spaced natural frequencies.

The goal of this research was to perform experimental and analytical studies using TMDs to control more than one mode of vibration for a floor exhibiting annoying floor vibrations. Tuned mass dampers can be used to control more than one mode of vibration with success. However, the amount of success greatly depends on the vibrational characteristics of the floor.

6.1.1 Experimental Study

Three prototype TMDs using viscoelastic damping elements were placed on a Laboratory Floor to control modes of vibration occurring at frequencies of 7.375 Hz, 9.375 Hz, and 16.75 Hz. The prototype TMDs decreased RMS accelerations due to walking on the Laboratory Floor by a factor of 3.7 when walking perpendicular to the joists. RMS accelerations due to walking parallel to the joists were reduced by a factor of 5.5. Well spaced natural frequencies (i.e. at least 2 Hz between adjacent frequencies) and relatively small damping in this floor made it possible for the TMDs to greatly improve vibration response.

Prototype TMDs were used to control anywhere from one to three modes of vibration on three different bays of an office floor. The frequencies which were of concern on these bays ranged from 4.25 Hz to 8 Hz. TMDs decreased RMS accelerations due to walking by factors ranging from 1.23 to 1.75. Closely spaced natural frequencies (i.e. usually less than 2 Hz between adjacent frequencies) and an initial mode 1 damping ratio of approximately 5% made it difficult for the TMDs to control the floor vibrations as effectively as they did on the Laboratory Floor. The greatest vibration control was gained for the bay with the smallest area and the most TMDs in place. For this bay, one TMD was used to control the fundamental mode of vibration and two TMDs were used to control two higher modes of vibration. TMDs used to control the higher modes of vibration had “optimum” frequencies which fell between two floor natural frequencies, and not near a frequency they were designed to control.

6.1.2 Analytical Study

Analytical models of TMDs were placed in the floor models to study control of floor model frequencies less than 10 Hz. The analytical TMDs were similar to the experimental TMDs, in that

they had the same spring stiffness, damping stiffness, and damping coefficient as the experimental TMDs. Analytical studies were performed for the Laboratory Floor and Bay 1 and Bay 2 of the office floor.

The “optimum” analytical TMDs were chosen on the basis of which TMD-to-floor mass ratio resulted in the smallest RMS acceleration response to an analytical heel drop impact. The mass ratio corresponding to the smallest RMS acceleration response was chosen as the “optimum” mass ratio. Knowing the “optimum” mass ratio, the “optimum” mass and frequency of the analytical TMD was determined. The “optimal” analytical TMDs for the first mode of vibration of all floor models had a calculated frequency near the first mode frequency of the bare floor model. This was expected based on theoretical calculations introduced in Chapter 2.

The Laboratory Floor TMD2 had a lower than expected frequency of 8.52 Hz. This frequency was not near the bare floor mode 2 frequency of 10.59 Hz, but near the mode 2 frequency of the floor resulting from the placement of analytical TMD1 in the model; this mode 2 frequency occurred at 8.14 Hz. An optimum TMD could not be determined for TMD2 and TMD3 of Bay 1, but masses for these TMDs were chosen and the RMS acceleration decreased. Only one mode of vibration was controlled in Bay 2. The analytical TMDs for Bay 2 performed as expected.

The analytical work had some results similar to the experimental work. Correlations between the TMD1, TMD1a, and TMD1b frequencies and the bare floor first mode frequencies were similar for the analytical and experimental work for the office floor Bay 1 and Bay 2. Also, the analytical TMD3 for Bay 1 had a frequency higher than the analytical floor frequency that it was designed to control; this similar trend also occurred in the experimental work with TMD3. Had a smaller mass been chosen for the analytical TMD2 of Bay 1, this same trend might have

occurred, and it would have been similar to the experimental TMD2 trend (i.e. a TMD frequency higher than the frequency of the floor mode of vibration which it was designed to control). Results of analytical and experimental work on the Laboratory Floor were not similar; this is most likely due to the fact that a third TMD, TMD3, was used experimentally to control floor vibrations and was “optimized” before the experimental TMD2 was placed on the floor and “optimized”.

6.2 Conclusions

The floor vibrations of a floor with relatively low damping (e.g. ranging from 1.3% to 2.3%) and well-spaced frequencies (e.g. the Laboratory Floor) can be successfully controlled with tuned mass dampers. When more than one mode of vibration needs to be controlled on a floor with relatively high damping (e.g. 5%) and closely-spaced frequencies (e.g. the office floor Bays 1, 2, and 3), gaining control of floor vibration with tuned mass dampers can be difficult, but it is possible.

Tuned mass dampers are able to control more than one mode of vibration with success. The RMS acceleration due to walking on a floor decreases as more TMDs are placed on a floor to control more modes of vibration. This was demonstrated in Bay 1 of the office floor. This bay had three TMDs to control three modes of vibration. The other bays of the office floor only had one or two TMDs to control one or two modes of vibration. The RMS acceleration was reduced by the greatest factor in the bay with the most TMDs and, consequently, the largest cumulative TMD-to-floor mass ratio.

Reduction in acceleration reflects improved control of floor vibration; however, if a floor has relatively high damping and closely spaced natural frequencies, a subjective evaluation of

TMD effectiveness should also be performed by the occupants. If the occupants are not satisfied by the reduction in floor vibrations with the placement of TMDs, the TMDs will not be of value.

6.3 Recommendations for Further Research

It is recommended that further experimental and analytical testing be performed using tuned mass dampers to control more than one mode of vibration. Further experimental testing should be performed with more than three TMDs available. With more TMDs available, more TMD mass can be provided for larger bay areas and more modes of vibration can be controlled. Also, further experimental testing should be performed on other occupied office floors to verify the ability of TMDs to control two or more modes of vibration.

Using the calculated acceleration response to an analytical heel drop impact, the optimum TMD2 and TMD3 models of Bay 1 of the office floor could not be determined. Analytical tests should be performed to see if a dynamic impact representing a person walking is better suited to determining the “optimum” TMD.

Usually the TMD to control the fundamental mode of vibration is first placed on a floor and tuned. Once this TMD is tuned, it is left on the floor and the next TMD is placed on the floor and tuned. Further study should be done using analytical models to determine the best order of TMD placement, and to study the effects of this TMD placement order on a floor’s vibration characteristics.

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APPENDIX A

Determination of Laboratory Floor Section Properties Used in SAP90 Model

Concrete Properties:

from tests (Hanagan 1994): $f'_c = 4100$ psi & $w = 115.7$ pcf

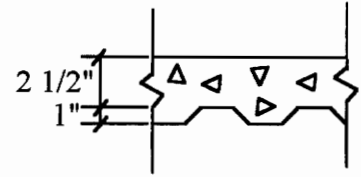
$$E_c = 33w_c^{1.5} \sqrt{f'_c} = 33(115.7)^{1.5} \sqrt{4100} = 2,630,000 \text{ psi}$$

$$n = \frac{E_s}{E_c} = \frac{29,000}{2630} = 11.0$$

average thickness = 3 in.

$$\begin{aligned} \text{weight/volume} = W &= \text{concrete} + \text{steel deck} \\ &= 115.7 \text{ pcf} + (1 \text{ psf})/(0.25 \text{ ft}) = 119.7 \text{ pcf} \\ &= 0.06927083 \text{ lb/in}^3 \end{aligned}$$

$$\begin{aligned} \text{mass/volume} = M &= W/g \\ &= 0.06927083/(386.4 \text{ in/s}^2) \\ &= 0.00017927 \text{ lb-s}^2/\text{in}^4 \end{aligned}$$



Joist Properties:

The joists are assumed to act compositely with the concrete.

$$\begin{array}{lll} 16K4: A_{\text{eff}} = 1.218 \text{ in}^2 & d = 16'' & \text{weight} = 7 \text{ plf} \\ I_{\text{eff}} = 67.3 \text{ in}^4 & \bar{y} = 7.168'' & \end{array}$$

from tests (Hanagan 1994): $I_j = 80.1 \text{ in}^4$

$$\text{weight/volume} = W = \frac{7 \text{ plf}}{12''} / 1.218 \text{ in}^2 = 0.4789272 \text{ lb/in}^3$$

$$\text{mass/volume} = M = W/g = 0.4789272/386.4 = 0.00123946 \text{ lb-s}^2/\text{in}^4$$

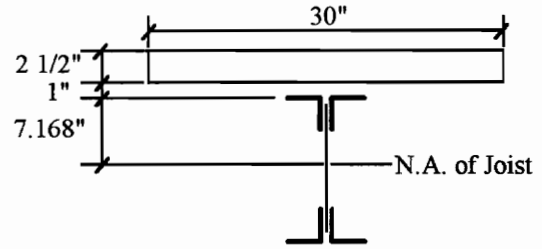
interior joist -

effective width (Allen & Murray 1993), $b = s \leq 0.4 L_b$

$$b = 30" \leq 0.4(25 \times 12") = 120"$$

$$A_c = \frac{1}{11}(2.5" \times 30") = 6.818 \text{ in}^2$$

$$\bar{y}_c = 1.25"$$



$$\bar{y} = \frac{6.818(1.25) + 1.218(3.5 + 7.168)}{6.818 + 1.218} = 2.677"$$

I_{jtr} = moment of inertia of composite section - moment of inertia of concrete

$$= I_{tr} - \left(\frac{1}{12}bd^3\right)_{\text{concrete}}$$

note: SAP90 takes into account the moment of inertia of the concrete plate element; therefore it can be neglected in the calculation of I_{jtr} for the interior and exterior joist.

$$I_{jtr} = 80.1 \text{ in}^4 + 1.218 \text{ in}^2 (3.5" + 7.168" - 2.677")^2 + 6.818 \text{ in}^2 (2.677" - 1.25")^2$$

$$I_{jtr} = 171.8 \text{ in}^4$$

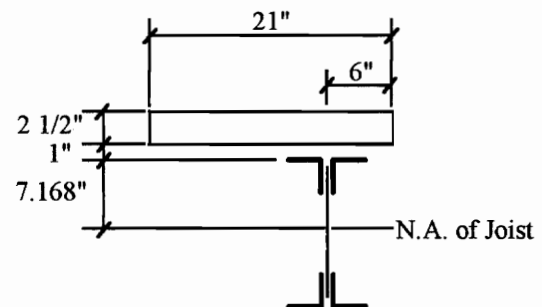
exterior joist -

effective width, $b = \frac{1}{2}b_{\text{interior joist}} + \text{overhang}$

$$b = 15" + 6"$$

$$A_c = \frac{1}{11}(2.5" \times 21") = 4.773 \text{ in}^2$$

$$\bar{y}_c = 1.25"$$



$$\bar{y} = \frac{4.773(1.25) + 1.218(3.5 + 7.168)}{4.773 + 1.218} = 3.165"$$

$$I_{jtr} = 80.1 \text{ in}^4 + 1.218 \text{ in}^2 (3.5" + 7.168" - 3.165")^2 + 4.773 \text{ in}^2 (3.165" - 1.25")^2$$

$$I_{jtr} = 166 \text{ in}^4$$

Girder Properties:

The girder is considered to act non-compositely with the concrete. Hanagan (1994) performed static load tests to verify that the girder acted non-compositely with the concrete.

$$\begin{array}{lll} \text{W14x22:} & A = 6.49 \text{ in}^2 & d = 13.74" \\ & I = 199 \text{ in}^4 & \bar{y} = 6.87" \end{array}$$

$$\text{weight/volume} = W = \frac{22 \text{ plf}}{12"} \bigg/ 6.49 \text{ in}^2 = 0.2825 \text{ lb/in}^3$$

$$\text{mass/volume} = M = W/g = 0.2825/386.4 = 0.00073107 \text{ lb-s}^2/\text{in}^4$$

APPENDIX B

SAP90 v. 6.0 Input for the Laboratory Floor Model

SYSTEM

WARN=Y LENGTH=IN FORCE=LB

JOINT

1,7,1,71,7 X=0,180,0 Y=0,0,300 Z=0

RESTRAINT

ADD=1,7,1,71,7 DOF=UX,UY,RZ

ADD=1 DOF=UX,UY

ADD=7 DOF=UX,UY

ADD=71 DOF=UX,UY

ADD=77 DOF=UX,UY

SPRING

ADD=1 UZ=65000

ADD=7 UZ=65000

ADD=71 UZ=65000

ADD=77 UZ=65000

MATERIAL

NAME=14X22 TYPE=ISO IDES=S M=0.00073107 W=0.2825

E=29000000 U=0.3

NAME=16K4 TYPE=ISO IDES=S M=0.00123946 W=0.4789272

E=29000000 U=0.3

NAME=CONCRET TYPE=ISO IDES=C M=0.00017927 W= 0.06927083

E=2630000 U=0.2

SECTION

NAME=GIRDER MAT=14X22 A=6.49 I=199

NAME=JOIST MAT=16K4 A=1.218 I=171.8

NAME=EXJOIST MAT=16K4 A=1.218 I=166

FRAME

1 J=1,2 SEC=GIRDER PLANE13=-Y

GEN=1,6,1 IINC=1

7 J=71,72 SEC=GIRDER PLANE13=-Y

GEN=7,12,1 IINC=1

13 J=1,8 SEC=EXJOIST PLANE13=+X IREL=R3

14 J=8,15 SEC=EXJOIST PLANE13=+X

GEN=14,21,1 IINC=7 JINC=7
 22 J=64,71 SEC=EXJOIST PLANE13=+X JREL=R3
 23 J=2,9 SEC=JOIST PLANE13=+X IREL=R3
 24 J=9,16 SEC=JOIST PLANE13=+X
 GEN=24,31,1 IINC=7 JINC=7
 32 J=65,72 SEC=JOIST PLANE13=+X JREL=R3
 33 J=3,10 SEC=JOIST PLANE13=+X IREL=R3
 34 J=10,17 SEC=JOIST PLANE13=+X
 GEN=34,41,1 IINC=7 JINC=7
 42 J=66,73 SEC=JOIST PLANE13=+X JREL=R3
 43 J=4,11 SEC=JOIST PLANE13=+X IREL=R3
 44 J=11,18 SEC=JOIST PLANE13=+X
 GEN=44,51,1 IINC=7 JINC=7
 52 J=67,74 SEC=JOIST PLANE13=+X JREL=R3
 53 J=5,12 SEC=JOIST PLANE13=+X IREL=R3
 54 J=12,19 SEC=JOIST PLANE13=+X
 GEN=54,61,1 IINC=7 JINC=7
 62 J=68,75 SEC=JOIST PLANE13=+X JREL=R3
 63 J=6,13 SEC=JOIST PLANE13=+X IREL=R3
 64 J=13,20 SEC=JOIST PLANE13=+X
 GEN=64,71,1 IINC=7 JINC=7
 72 J=69,76 SEC=JOIST PLANE13=+X JREL=R3
 73 J=7,14 SEC=EXJOIST PLANE13=+X IREL=R3
 74 J=14,21 SEC=EXJOIST PLANE13=+X
 GEN=74,81,1 IINC=7 JINC=7
 82 J=70,77 SEC=EXJOIST PLANE13=+X JREL=R3

SHELL

1 J=1,2,8,9 TYPE=P MAT=CONCRET TH=3 PLANE31=0
 GEN=1,6,1,55,6 JINC=1,7

LOAD

NAME=HD
 TYPE=FORCE
 ADD=26 UZ=-600

MODES

TYPE=EIGEN N=10

FUNCTION

NAME=HDRAMP DT=0.005 NPL=11
 1 .9 .8 .7 .6 .5 .4 .3 .2 .1 0

HISTORY

NAME=TIME TYPE=LIN NSTEP=1024 DT=1/128 NMODE=10 DAMP=0.015
 LOAD=HD FUNC=HDRAMP AT=1

```
OUTPUT
ELEM=JOINT OUT=DISP,REAC LOAD=HD MODE=1,2,3,4
ADD=*

END
```

APPENDIX C

Determination of Initial and Final Properties for Laboratory Floor TMD1

Initial Properties:

from floor vibrations measurements: $f_1 = 7.375 \text{ Hz}$

from SAP90 eigenvalue analysis: $w_{F1} = 6398 \text{ lbs}$

chosen by 3M Corporation: $w_T = 365 \text{ lbs}$

note: the inner frame of the TMD weighs 75 pounds. This results in a required plate weight of 290 pounds, which means 29 plates are required for the TMD.

from Equation 2.3, $\mu = \frac{365}{6398} = 0.057049$

from Equation 2.14, $f_{\text{optimal}} = \left(\frac{1}{1 + 0.057049} \right) 7.375 \text{ Hz} = 6.98 \text{ Hz}$

from Equation 2.15, $\zeta_{\text{optimal}} = \sqrt{\frac{3 \times 0.057049}{8(1 + 0.057049)^3}} = 0.134585$

from Equation 2.11 and 2.6, $f_D = 7.375 \text{ Hz} \sqrt{1 - (0.134585)^2} = 6.91 \text{ Hz}$

note: Since the damping ratio has a minimal effect on the TMD frequency, the damped frequency will not be determined using actual TMD properties.

from Equations 2.11 and 2.5, $\omega_D = \sqrt{\frac{k_T}{m_T}} \Rightarrow k_T = m_T \omega_D^2 = \frac{w_T}{g} \times (2\pi f_D)^2$

$$k_T = \frac{365 \text{ lbs}}{386.4 \text{ in/s}^2} \times (2\pi \times 6.91 \text{ Hz})^2 = 1783.4 \text{ lb/in}$$

from Equations 2.9 and 2.10, $\zeta = \frac{c}{2m\omega_n} \Rightarrow c = \zeta_{\text{optimal}} (2m_T \omega_n)$

$$c = \zeta_{\text{optimal}} \left(2 \frac{W_T}{g} \times 2\pi f_{\text{optimal}} \right)$$

$$c = 0.134585 \times \left(2 \times \frac{365 \text{ lbs}}{386.4 \text{ in/s}^2} \times 2\pi \times 6.98 \text{ Hz} \right) = 11.146$$

Final Properties:

number of 10 lb plates = 29: $w_T = (29 \times 10 \text{ lbs}) + 75 \text{ lbs} = 365 \text{ lbs}$

from Equation 2.3, $\mu = \frac{365}{6398} = 0.057049$

manufactured springs: $k_s = 1164 \text{ lb/in}$

damping element properties, as designed by 3M Corporation, (Abbas and Kelly 1993):

$G' = \text{shear storage modulus} = 46 \text{ psi}$

$\eta = 1$ (not used in these calculations, but used by 3M Corporation to determine other viscoelastic damping element properties)

$t = \text{thickness} = 0.5''$

$A = \text{area} = 4.5 \text{ in}^2$

$$k_d = \text{damping element stiffness} = \frac{G'A}{t} = \frac{(46 \text{ psi}) \times (4.5 \text{ in}^2)}{0.5''} = 414 \text{ lb/in}$$

$$k_T = k_s + k_d = 1164 \text{ lb/in} + 414 \text{ lb/in} = 1578 \text{ lb/in}$$

$$\text{from Equation 2.5, } f_{n\text{TMD}} = \frac{1}{2\pi} \sqrt{\frac{k_T}{m_T}} = \frac{1}{2\pi} \sqrt{\frac{k_T}{w_T/g}}$$

$$f_{n\text{TMD}} = \frac{1}{2\pi} \sqrt{\frac{1578 \text{ lb/in}}{365 \text{ lb}/386.4 \text{ in/s}^2}} = 6.505 \text{ Hz}$$

damping coefficient (Abbas and Kelly 1993):

$$c_T = \text{damping coefficient} = \frac{kd}{\omega_{nTMD}} = \frac{kd}{2\pi f_{nTMD}} = \frac{414 \text{ lb/in}}{2\pi(6.505 \text{ Hz})} = 10.12919$$

APPENDIX D

SAP90 v. 6.0 Input for the Office Floor Models

SAP90 Input for Bay 1

SYSTEM

WARN=Y LENGTH=IN FORCE=LB

JOINT

1,11,1,111,11 X=0,360,0 Y=0,0,360 Z=0

RESTRAINT

ADD=1,11,1,221,11 DOF=UX,UY,RZ

ADD=1 DOF=UX,UY,UZ,RZ

ADD=11 DOF=UX,UY,UZ,RZ

ADD=111 DOF=UX,UY,UZ,RZ

ADD=121 DOF=UX,UY,UZ,RZ

SPRING

ADD=1,111,11 RY=1000000000000

ADD=11,121,11 RY=1000000000000

MASS

ADD=1 UZ=9*9.75/386.4+0.579

ADD=12,23,11 UZ=9*19.5/386.4+1.158

ADD=34 UZ=9*19.5/386.4+0.864

ADD=45 UZ=9*34.5/386.4+1.462

ADD=56,67,11 UZ=9*34.5/386.4+2.355

ADD=78 UZ=9*49.5/386.4+4.236

ADD=89,100,11 UZ=9*49.5/386.4+3.6065

ADD=111 UZ=9*454.5/386.4+3.74428

ADD=11 UZ=9*24.75/386.4+1.786

ADD=22,110,11 UZ=9*49.5/386.4+3.572

ADD=121 UZ=9*454.5/386.4+3.72728

ADD=112,120,1 UZ=9*9/386.4+0.31056

ADD=2,10,1 UZ=9*4.5/386.4

ADD=13,21,1 UZ=9*9/386.4

ADD=24,32,1 UZ=9*9/386.4

ADD=35,43,1 UZ=9*9/386.4

ADD=46,54,1 UZ=9*9/386.4

ADD=57,65,1 UZ=9*9/386.4

ADD=68,76,1 UZ=9*9/386.4

ADD=79,87,1 UZ=9*9/386.4
ADD=90,98,1 UZ=9*9/386.4
ADD=101,109,1 UZ=9*9/386.4

MATERIAL

NAME=24X76 TYPE=ISO IDES=S M=0.00073172 W=0.2827381
E=290000000 U=0.3
NAME=20H6 TYPE=ISO IDES=S M=0.00128734 W=0.49742902
E=290000000 U=0.3
NAME=STIFF TYPE=ISO M=0 W=0
E=290000000
NAME=CONCRET TYPE=ISO M=0.00022515 W=0.08699846 IDES=C
E=3.409E+6

SECTION

NAME=GIRDER MAT=24X76 A=22.4 I=2100
NAME=JOIST MAT=20H6 A=1.491 I=270.79
NAME=WALL MAT=STIFF A=1 I=130000

FRAME

1 J=1,2 SEC=WALL PLANE13=-Y
GEN=1,10,1 IINC=1
11 J=12,13 SEC=JOIST PLANE13=-Y IREL=R3
12 J=13,14 SEC=JOIST PLANE13=-Y
GEN=12,19,1 IINC=1
20 J=21,22 SEC=JOIST PLANE13=-Y JREL=R3
21 J=23,24 SEC=JOIST PLANE13=-Y IREL=R3
22 J=24,25 SEC=JOIST PLANE13=-Y
GEN=22,29,1 IINC=1
30 J=32,33 SEC=JOIST PLANE13=-Y JREL=R3
31 J=34,35 SEC=JOIST PLANE13=-Y IREL=R3
32 J=35,36 SEC=JOIST PLANE13=-Y
GEN=32,39,1 IINC=1
40 J=43,44 SEC=JOIST PLANE13=-Y JREL=R3
41 J=45,46 SEC=JOIST PLANE13=-Y IREL=R3
42 J=46,47 SEC=JOIST PLANE13=-Y
GEN=42,49,1 IINC=1
50 J=54,55 SEC=JOIST PLANE13=-Y JREL=R3
51 J=56,57 SEC=JOIST PLANE13=-Y IREL=R3
52 J=57,58 SEC=JOIST PLANE13=-Y
GEN=52,59,1 IINC=1
60 J=65,66 SEC=JOIST PLANE13=-Y JREL=R3
61 J=67,68 SEC=JOIST PLANE13=-Y IREL=R3
62 J=68,69 SEC=JOIST PLANE13=-Y
GEN=62,69,1 IINC=1
70 J=76,77 SEC=JOIST PLANE13=-Y JREL=R3

71 J=78,79 SEC=JOIST PLANE13=-Y IREL=R3
 72 J=79,80 SEC=JOIST PLANE13=-Y
 GEN=72,79,1 IINC=1
 80 J=87,88 SEC=JOIST PLANE13=-Y JREL=R3
 81 J=89,90 SEC=JOIST PLANE13=-Y IREL=R3
 82 J=90,91 SEC=JOIST PLANE13=-Y
 GEN=82,89,1 IINC=1
 90 J=98,99 SEC=JOIST PLANE13=-Y JREL=R3
 91 J=100,101 SEC=JOIST PLANE13=-Y IREL=R3
 92 J=101,102 SEC=JOIST PLANE13=-Y
 GEN=92,99,1 IINC=1
 100 J=109,110 SEC=JOIST PLANE13=-Y JREL=R3
 101 J=111,112 SEC=JOIST PLANE13=-Y IREL=R3
 102 J=112,113 SEC=JOIST PLANE13=-Y
 GEN=102,109,1 IINC=1
 110 J=120,121 SEC=JOIST PLANE13=-Y JREL=R3
 111 J=1,12 SEC=GIRDER PLANE13=+X IREL=R3
 112 J=12,23 SEC=GIRDER PLANE13=+X
 GEN=112,119,1 IINC=11 JINC=11
 120 J=100,111 SEC=GIRDER PLANE13=+X JREL=R3
 121 J=11,22 SEC=GIRDER PLANE13=+X IREL=R3
 122 J=22,33 SEC=GIRDER PLANE13=+X
 GEN=122,129,1 IINC=11 JINC=11
 130 J=110,121 SEC=GIRDER PLANE13=+X JREL=R3

SHELL

1 J=1,2,12,13 TYPE=P MAT=CONCRET TH=2.25 PLANE31=0
 GEN=1,10,1,91,10 JINC=1,11

LOAD

NAME=HD
 TYPE=FORCE
 ADD=61 UZ=-600

MODES

TYPE=EIGEN N=10

FUNCTION

NAME=HDRAMP DT=0.005 NPL=11
 1 .9 .8 .7 .6 .5 .4 .3 .2 .1 0

HISTORY

NAME=TIME TYPE=LIN NSTEP=1024 DT=1/128 NMODE=10 DAMP=0.05
 LOAD=HD FUNC=HDRAMP AT=1

OUTPUT

ELEM=JOINT OUT=DISP,REAC LOAD=HD MODE=1,2,3,4
ADD=*

END

SAP90 Input for Bay 2

SYSTEM

WARN=Y LENGTH=IN FORCE=LB

JOINT

1,11,1,133,11 X=0,360,0 Y=0,0,480 Z=0

RESTRAINT

ADD=1,11,1,133,11 DOF=UX,UY,RZ

ADD=1 DOF=UX,UY,UZ,RZ

ADD=11 DOF=UX,UY,UZ,RZ

ADD=133 DOF=UX,UY,UZ,RZ

ADD=143 DOF=UX,UY,UZ,RZ

SPRING

ADD=1,11,1 RY=1E9

ADD=133,143,1 RY=1E9

MASS

ADD=1 UZ=9*50/386.4+1.9672

ADD=11 UZ=9*50/386.4+1.9672

ADD=133 UZ=9*57.5/386.4+2.2972

ADD=143 UZ=9*59.479/386.4+2.2972

ADD=22,132,11 UZ=9*10/386.4+0.3623

ADD=12,122,11 UZ=10.5*7.5*40/12+90/386.4+0.3623

ADD=2,10,1 UZ=9*50/386.4+3.572

ADD=134,142,1 UZ=9*57.5/386.4+4.232

ADD=13,21,1 UZ=9*10/386.4

ADD=24,32,1 UZ=9*10/386.4

ADD=35,43,1 UZ=9*10/386.4

ADD=46,54,1 UZ=9*10/386.4

ADD=57,65,1 UZ=9*10/386.4

ADD=68,76,1 UZ=9*10/386.4

ADD=79,87,1 UZ=9*10/386.4

ADD=90,98,1 UZ=9*10/386.4

ADD=101,109,1 UZ=9*10/386.4

ADD=112,120,1 UZ=9*10/386.4

ADD=123,131,1 UZ=9*10/386.4

MATERIAL

NAME=27X84 TYPE=ISO IDES=S M=0.0007306 W=0.2823

E=29000000 U=0.3

NAME=24H8 TYPE=ISO IDES=S M=0.001353 W=0.5228

E=29000000 U=0.3

NAME=CONCRET TYPE=ISO M=0.00022515 W=0.08699846 IDES=C

E=3.409E+6

SECTION

NAME=GIRDER MAT=27X84 A=24.8 I=2850

NAME=JOIST MAT=24H8 A=1.833 I=464.72

NAME=STIFJST MAT=24H8 A=1.833 I=4647.20

FRAME

1 J=1,2 SEC=GIRDER PLANE13=-Y

GEN=1,10,1 IINC=1

11 J=133,134 SEC=GIRDER PLANE13=-Y

GEN=11,20,1 IINC=1

21 J=1,12 SEC=STIFJST PLANE13=+X IREL=R3

22 J=12,23 SEC=STIFJST PLANE13=+X

GEN=22,31,1 IINC=11 JINC=11

32 J=122,133 SEC=STIFJST PLANE13=+X JREL=R3

33 J=2,13 SEC=JOIST PLANE13=+X IREL=R3

34 J=13,24 SEC=JOIST PLANE13=+X

GEN=34,43,1 IINC=11 JINC=11

44 J=123,134 SEC=JOIST PLANE13=+X JREL=R3

45 J=3,14 SEC=JOIST PLANE13=+X IREL=R3

46 J=14,25 SEC=JOIST PLANE13=+X

GEN=46,55,1 IINC=11 JINC=11

56 J=124,135 SEC=JOIST PLANE13=+X JREL=R3

57 J=4,15 SEC=JOIST PLANE13=+X IREL=R3

58 J=15,26 SEC=JOIST PLANE13=+X

GEN=58,67,1 IINC=11 JINC=11

68 J=125,136 SEC=JOIST PLANE13=+X JREL=R3

69 J=5,16 SEC=JOIST PLANE13=+X IREL=R3

70 J=16,27 SEC=JOIST PLANE13=+X

GEN=70,79,1 IINC=11 JINC=11

80 J=126,137 SEC=JOIST PLANE13=+X JREL=R3

81 J=6,17 SEC=JOIST PLANE13=+X IREL=R3

82 J=17,28 SEC=JOIST PLANE13=+X

GEN=82,91,1 IINC=11 JINC=11

92 J=127,138 SEC=JOIST PLANE13=+X JREL=R3

93 J=7,18 SEC=JOIST PLANE13=+X IREL=R3

94 J=18,29 SEC=JOIST PLANE13=+X

GEN=94,103,1 IINC=11 JINC=11

104 J=128,139 SEC=JOIST PLANE13=+X JREL=R3

105 J=8,19 SEC=JOIST PLANE13=+X IREL=R3

106 J=19,30 SEC=JOIST PLANE13=+X

GEN=106,115,1 IINC=11 JINC=11

116 J=129,140 SEC=JOIST PLANE13=+X JREL=R3

117 J=9,20 SEC=JOIST PLANE13=+X IREL=R3

118 J=20,31 SEC=JOIST PLANE13=+X

```

GEN=118,127,1 IINC=11 JINC=11
128 J=130,141 SEC=JOIST PLANE13=+X JREL=R3
129 J=10,21 SEC=JOIST PLANE13=+X IREL=R3
130 J=21,32 SEC=JOIST PLANE13=+X
GEN=130,139,1 IINC=11 JINC=11
140 J=131,142 SEC=JOIST PLANE13=+X JREL=R3
141 J=11,22 SEC=JOIST PLANE13=+X IREL=R3
142 J=22,33 SEC=JOIST PLANE13=+X
GEN=142,151,1 IINC=11 JINC=11
152 J=132,143 SEC=JOIST PLANE13=+X JREL=R3

SHELL
1 J=1,2,12,13 TYPE=P MAT=CONCRET TH=2.25 PANE31=0
GEN=1,10,1,111,10 JINC=1,11

LOAD
NAME=HD
TYPE=FORCE
ADD=72 UZ=-600

MODES
TYPE=RITZ N=10
LOAD=HD

FUNCTION
NAME=HDRAMP DT=0.005 NPL=11
1 .9 .8 .7 .6 .5 .4 .3 .2 .1 0

HISTORY
NAME=TIME TYPE=LIN NSTEP=1024 DT=1/128 NMODE=10 DAMP=0.05
LOAD=HD FUNC=HDRAMP

OUTPUT
ELEM=JOINT OUT=DISP,REAC LOAD=HD MODE=1,2,3,4
ADD=*

END

```

SAP90 Input for Bay 3

SYSTEM

WARN=Y LENGTH=IN FORCE=LB

JOINT

1,13,1,131,13 X=0,420,0 Y=0,0,360 Z=0
144,156,1,157,13 X=0,420,0 Y=385,385,410 Z=0
170,182,1,183,13 X=0,420,0 Y=440,440,470 Z=0
196,203,1 X=420,420 Y=500,710 Z=0

RESTRAINT

ADD=1,13,1,183,13 DOF=UX,UY,RZ
ADD=196,202,1 DOF=UX,UY,RY,RZ
ADD=1 DOF=UX,UY,UZ,RZ
ADD=13 DOF=UX,UY,UZ,RZ
ADD=143 DOF=UX,UY,UZ,RZ
ADD=183 DOF=UX,UY,UZ,RX,RY,RZ
ADD=184,188,1 DOF=UX,UY,UZ,RX,RY,RZ
ADD=203 DOF=UX,UY,UZ,RY,RZ

SPRING

ADD=196,202,1 RY=1

MASS

ADD=1 UZ=9*4.375/386.4+0.4642
ADD=14,118,13 UZ=9*4.375/386.4+0.6114
ADD=131 UZ=9*3.707/386.4+0.489
ADD=144 UZ=9*3.038/386.4+0.489
ADD=157 UZ=9*3.342/386.4+0.489
ADD=170 UZ=9*3.646/386.4+0.489
ADD=183 UZ=9*1.823/386.4+0.489
ADD=2,12,1 UZ=9*8.75/386.4+0.317
ADD=15,25,1 UZ=9*8.75/386.4
ADD=28,38,1 UZ=9*8.75/386.4
ADD=41,51,1 UZ=9*8.75/386.4
ADD=54,64,1 UZ=9*8.75/386.4
ADD=67,77,1 UZ=9*8.75/386.4
ADD=80,90,1 UZ=9*8.75/386.4
ADD=93,103,1 UZ=9*8.75/386.4
ADD=106,116,1 UZ=9*8.75/386.4
ADD=119,129,1 UZ=9*8.75/386.4
ADD=132,142,1 UZ=9*7.413/386.4
ADD=145,155,1 UZ=9*6.076/386.4
ADD=158,168,1 UZ=9*6.68/386.4
ADD=171,181,1 UZ=9*7.2917/386.4

ADD=184,187,1 UZ=9*3.646/386.4
 ADD=188 UZ=3.646+50*9/386.4+3.778
 ADD=189,194,1 UZ=9*7.2917/386.4
 ADD=13,130,13 UZ=9*64.375/386.4+4.943
 ADD=143 UZ=9*54.54/386.4+4.943
 ADD=156 UZ=9*44.7/386.4+3.317
 ADD=169 UZ=9*49.175/386.4+3.619
 ADD=182 UZ=9*53.65/386.4+4.218
 ADD=195 UZ=9*64.323/386.4+4.218
 ADD=196,197,1 UZ=9*75/386.4+6.185
 ADD=198 UZ=9*75/386.4+5.888
 ADD=199 UZ=9*93.75/386.4+7.738
 ADD=200,202,1 UZ=9*62.5/386.4+5.188
 ADD=203 UZ=9*56.25/386.4+2.42

MATERIAL

NAME=27X84 TYPE=ISO IDES=S M=0.00073048 W=0.28225806
 E=29000000 U=0.3
 NAME=24X68 TYPE=ISO IDES=S M=0.00072962 W=0.28192371
 E=29000000 U=0.3
 NAME=20H8 TYPE=ISO IDES=S M=0.00225292 W=0.87052865
 E=29000000 U=0.3
 NAME=24H9 TYPE=ISO IDES=S M=0.00227616 W=0.87950748
 E=29000000 U=0.3
 NAME=18X35 TYPE=ISO IDES=S M=0.00073285 W=0.28317152
 E=29000000 U=0.3
 NAME=CONCRET TYPE=ISO M=0.00022515 W=0.08699846
 E=3.409E+6

SECTION

NAME=GIRD27 MAT=27X84 A=24.8 I=2850
 NAME=G1 MAT=27X84 A=24.8 I=2866
 NAME=GIRD24 MAT=24X68 A=20.1 I=1830
 NAME=STFJST MAT=24H9 A=1.137 I=2950
 NAME=JOIST MAT=20H8 A=1.053 I=295.63
 NAME=J1 MAT=20H8 A=1.053 I=293.53
 NAME=J2 MAT=20H8 A=1.053 I=290.63
 NAME=J3 MAT=20H8 A=1.053 I=292.07
 NAME=J4 MAT=20H8 A=1.053 I=293.28
 NAME=STFBM MAT=18X35 A=10.30 I=960
 NAME=BEAM MAT=18X35 A=10.30 I=960

FRAME

1 J=1,14 SEC=GIRD24 PLANE13=+X
 GEN=1,14,1 IINC=13 JINC=13
 15 J=13,26 SEC=GIRD27 PLANE13=+X

GEN=15,28,1 IINC=13 JINC=13
 29 J=195,196 SEC=G1 PLANE13=+X
 GEN=29,36,1 IINC=1
 37 J=1,2 SEC=STFJST PLANE13=-Y IREL=R3
 38 J=2,3 SEC=STFJST PLANE13=-Y
 GEN=38,47,1 IINC=1
 48 J=12,13 SEC=STFJST PLANE13=-Y JREL=R3
 49 J=14,15 SEC=JOIST PLANE13=-Y IREL=R3
 50 J=15,16 SEC=JOIST PLANE13=-Y
 GEN=50,59,1 IINC=1
 60 J=25,26 SEC=JOIST PLANE13=-Y JREL=R3
 61 J=27,28 SEC=JOIST PLANE13=-Y IREL=R3
 62 J=28,29 SEC=JOIST PLANE13=-Y
 GEN=62,71,1 IINC=1
 72 J=38,39 SEC=JOIST PLANE13=-Y JREL=R3
 73 J=40,41 SEC=JOIST PLANE13=-Y IREL=R3
 74 J=41,42 SEC=JOIST PLANE13=-Y
 GEN=74,83,1 IINC=1
 84 J=51,52 SEC=JOIST PLANE13=-Y JREL=R3
 85 J=53,54 SEC=JOIST PLANE13=-Y IREL=R3
 86 J=54,55 SEC=JOIST PLANE13=-Y
 GEN=86,95,1 IINC=1
 96 J=64,65 SEC=JOIST PLANE13=-Y JREL=R3
 97 J=66,67 SEC=JOIST PLANE13=-Y IREL=R3
 98 J=67,68 SEC=JOIST PLANE13=-Y
 GEN=98,107,1 IINC=1
 108 J=77,78 SEC=JOIST PLANE13=-Y JREL=R3
 109 J=79,80 SEC=JOIST PLANE13=-Y IREL=R3
 110 J=80,81 SEC=JOIST PLANE13=-Y
 GEN=110,119,1 IINC=1
 120 J=90,91 SEC=JOIST PLANE13=-Y JREL=R3
 121 J=92,93 SEC=JOIST PLANE13=-Y IREL=R3
 122 J=93,94 SEC=JOIST PLANE13=-Y
 GEN=122,131,1 IINC=1
 132 J=103,104 SEC=JOIST PLANE13=-Y JREL=R3
 133 J=105,106 SEC=JOIST PLANE13=-Y IREL=R3
 134 J=106,107 SEC=JOIST PLANE13=-Y
 GEN=134,143,1 IINC=1
 144 J=116,117 SEC=JOIST PLANE13=-Y JREL=R3
 145 J=118,119 SEC=JOIST PLANE13=-Y IREL=R3
 146 J=119,120 SEC=JOIST PLANE13=-Y
 GEN=146,155,1 IINC=1
 156 J=129,130 SEC=JOIST PLANE13=-Y JREL=R3
 157 J=131,132 SEC=J1 PLANE13=-Y IREL=R3
 158 J=132,133 SEC=J1 PLANE13=-Y
 GEN=158,167,1 IINC=1

168 J=142,143 SEC=J1 PLANE13=-Y JREL=R3
 169 J=144,145 SEC=J2 PLANE13=-Y IREL=R3
 170 J=145,146 SEC=J2 PLANE13=-Y
 GEN=170,179,1 IINC=1
 180 J=155,156 SEC=J2 PLANE13=-Y JREL=R3
 181 J=157,158 SEC=J3 PLANE13=-Y IREL=R3
 182 J=158,159 SEC=J3 PLANE13=-Y
 GEN=182,191,1 IINC=1
 192 J=168,169 SEC=J3 PLANE13=-Y JREL=R3
 193 J=170,171 SEC=J4 PLANE13=-Y IREL=R3
 194 J=171,172 SEC=J4 PLANE13=-Y
 GEN=194,203,1 IINC=1
 204 J=181,182 SEC=J4 PLANE13=-Y JREL=R3
 205 J=183,184 SEC=STFBM PLANE13=-Y IREL=R3
 206 J=184,185 SEC=STFBM PLANE13=-Y
 GEN=206,209,1 IINC=1
 210 J=188,189 SEC=BEAM PLANE13=-Y
 GEN=210,215,1 IINC=1
 216 J=194,195 SEC=BEAM PLANE13=-Y JREL=R3

SHELL

1 J=1,2,14,15 TYPE=P MAT=CONCRET TH=2.25 PANE31=0
 GEN=1,12,1,157,12 IINC=1,13

LOAD

NAME=HD
 TYPE=FORCE
 ADD=85 UZ=-600

MODES

TYPE=RITZ N=10
 LOAD=HD

FUNCTION

NAME=HDRAMP DT=0.005 NPL=11
 1 .9 .8 .7 .6 .5 .4 .3 .2 .1 0

HISTORY

NAME=TIME TYPE=LIN NSTEP=1024 DT=1/128 NMODE=10 DAMP=0.05
 LOAD=HD FUNC=HDRAMP

OUTPUT

ELEM=JOINT OUT=DISP,REAC LOAD=HD MODE=1,2,3,4
 ADD=*

END

APPENDIX E

SAP90 v. 6.0 Input for the Laboratory Floor with TMD1

SYSTEM

WARN=Y LENGTH=IN FORCE=LB

JOINT

1,7,1,71,7 X=0,180,0 Y=0,0,300 Z=0
78 X=84 Y=150 Z=3
79 X=90 Y=150 Z=3
80 X=96 Y=150 Z=3
81 X=84 Y=150 Z=9
82 X=96 Y=150 Z=9

RESTRAINT

ADD=1,7,1,71,7 DOF=UX,UY,RZ
ADD=1 DOF=UX,UY
ADD=7 DOF=UX,UY
ADD=71 DOF=UX,UY
ADD=77 DOF=UX,UY
ADD=78,82,1 DOF=UX,UY,RX,RY,RZ

SPRING

ADD=1 UZ=65000
ADD=7 UZ=65000
ADD=71 UZ=65000
ADD=77 UZ=65000

CONSTRAINT

NAME=TMD TYPE=EQUAL DOF=UZ
ADD=81,82,1

MATERIAL

NAME=14X22 TYPE=ISO IDES=S M=0.00073107 W=0.2825
E=29000000 U=0.3
NAME=16K4 TYPE=ISO IDES=S M=0.00123946 W=0.4789272
E=29000000 U=0.3
NAME=CONCRET TYPE=ISO IDES=C M=0.00017927 W= 0.06927083
E=2630000 U=0.2
NAME=STIFF TYPE=ISO M=0 W=0
E=29000000
NAME=STEEL TYPE=ISO IDES=S M=21*10+75/386.4/12 W=21*10+75/12

E=29000000

SECTION

NAME=GIRDER MAT=14X22 A=6.49 I=199

NAME=JOIST MAT=16K4 A=1.218 I=171.8

NAME=EXJOIST MAT=16K4 A=1.218 I=166

NAME=SUPPORT MAT=STIFF A=10000 J=0 I=100000,100000 AS=50000/6,50000/6 \

SH=R T=100,100

NAME=MASSPL MAT=STEEL A=1 J=0 I=100000,100000 AS=5/6,5/6 SH=R T=1,1

NLPROP

NAME=SPRING TYPE=DAMPER M=0 W=0

DOF=U1 KE=1164

NAME=DAMPEL TYPE=DAMPER M=0 W=0

DOF=U1 KE=414 CE=10.13

FRAME

1 J=1,2 SEC=GIRDER PLANE13=-Y

GEN=1,6,1 IINC=1

7 J=71,72 SEC=GIRDER PLANE13=-Y

GEN=7,12,1 IINC=1

13 J=1,8 SEC=EXJOIST PLANE13=+X IREL=R3

14 J=8,15 SEC=EXJOIST PLANE13=+X

GEN=14,21,1 IINC=7 JINC=7

22 J=64,71 SEC=EXJOIST PLANE13=+X JREL=R3

23 J=2,9 SEC=JOIST PLANE13=+X IREL=R3

24 J=9,16 SEC=JOIST PLANE13=+X

GEN=24,31,1 IINC=7 JINC=7

32 J=65,72 SEC=JOIST PLANE13=+X JREL=R3

33 J=3,10 SEC=JOIST PLANE13=+X IREL=R3

34 J=10,17 SEC=JOIST PLANE13=+X

GEN=34,41,1 IINC=7 JINC=7

42 J=66,73 SEC=JOIST PLANE13=+X JREL=R3

43 J=4,11 SEC=JOIST PLANE13=+X IREL=R3

44 J=11,18 SEC=JOIST PLANE13=+X

GEN=44,51,1 IINC=7 JINC=7

52 J=67,74 SEC=JOIST PLANE13=+X JREL=R3

53 J=5,12 SEC=JOIST PLANE13=+X IREL=R3

54 J=12,19 SEC=JOIST PLANE13=+X

GEN=54,61,1 IINC=7 JINC=7

62 J=68,75 SEC=JOIST PLANE13=+X JREL=R3

63 J=6,13 SEC=JOIST PLANE13=+X IREL=R3

64 J=13,20 SEC=JOIST PLANE13=+X

GEN=64,71,1 IINC=7 JINC=7

72 J=69,76 SEC=JOIST PLANE13=+X JREL=R3

73 J=7,14 SEC=EXJOIST PLANE13=+X IREL=R3

74 J=14,21 SEC=EXJOIST PLANE13=+X
 GEN=74,81,1 IINC=7 JINC=7
 82 J=70,77 SEC=EXJOIST PLANE13=+X JREL=R3
 83 J=39,79 SEC=SUPPORT PLANE13=+X
 84 J=78,79 SEC=SUPPORT PLANE13=-Y
 85 J=79,80 SEC=SUPPORT PLANE13=-Y
 86 J=81,82 SEC=MASSPL PLANE13=-Y

 SHELL
 1 J=1,2,8,9 TYPE=P MAT=CONCRET TH=3 PLANE31=0
 GEN=1,6,1,55,6 JINC=1,7

 NLLINK
 1 J=78,81 NLP=SPRING AXIS1=-Z PLANE13=+X
 2 J=80,82 NLP=DAMPEL AXIS1=-Z PLANE13=+X

 LOAD
 NAME=HD
 TYPE=FORCE
 ADD=39 UZ=-600

 MODES
 TYPE=EIGEN N=10

 FUNCTION
 NAME=HDRAMP DT=0.005 NPL=11
 1 .9 .8 .7 .6 .5 .4 .3 .2 .1 0

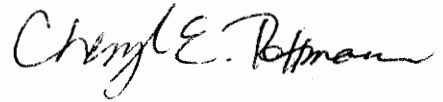
 HISTORY
 NAME=TIME TYPE=LIN NSTEP=1024 DT=1/128 NMODE=10 DAMP=0.015
 LOAD=HD FUNC=HDRAMP AT=1

 OUTPUT
 ELEM=JOINT OUT=DISP,REAC LOAD=HD MODE=1,2,3,4
 ADD=*

 END

VITA

Cheryl E. Rottmann was born in Mt. Vernon, IL on November 20, 1971. She graduated from Highland High School in Highland, IL. While pursuing her Bachelor of Science degree she worked as a civil engineering co-op student with Virginia Power. In 1994 she received her Bachelor of Science degree in civil engineering, with High Honors, from the University of Illinois at Urbana-Champaign. Since August of 1994 she has been pursuing her master's degree in the Structures Division of the Civil Engineering Department at Virginia Tech.

A handwritten signature in black ink that reads "Cheryl E. Rottmann". The signature is written in a cursive style with a large, stylized 'C' and 'R'.