STUDIES OF WOOD PALLET RESPONSE TO FORCED VIBRATION

by

Ira Edwin Lauer III

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APPROVED:

M.S. White, Chairman

J.R. Loferski

J.D. Dolan

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Ira Edwin Lauer III

Committee Chairman: Marshall S. White
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(ABSTRACT)

Wood pallets serve as interfaces between packaged products and transport vehicles. Vertical vibrations are transmitted through pallets into unit-loads. Pallet response to forced vibration affects forces experienced by products. A study was conducted to determine how pallet design influenced the resonant response of a uniformly distributed case goods unit-load. Other studies were conducted to develop a pallet section model to emulate the response of three stringer wood pallets to forced vibration. This model was used to investigate the effects of joint stiffness, deckboard EI, and uniformly distributed load level on the resonant response of pallet decks. Pallets were found to lower unit-load resonant frequencies and increase transmissibilities at resonance. Pallet sections constructed with stiff decks and joints were found to have higher resonant frequencies and lower transmissibilities than sections constructed with less stiff deck components. To prevent pallets from elevating acceleration transmitted through unit-loads, pallets should be constructed with the stiffest decks and joints that are economically feasible.

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Notations

Symbol	Explanation	Units
A	Amplitude	m
C	Viscous Damping	N-sec/m
C_c	Critical Viscous Damping	N-sec/m
E	Modulus of Elasticity	Pa
F	Force	N
f	Frequency	Hz
f_n	Natural Frequency	Hz
F_o	Maximum Applied Force	N
f_r	Resonant Frequency	Hz
g	Acceleration due to Gravity	9.8 m/sec^2
I	Moment of Inertia	m^4
K	Kinetic Energy	J
k	Linear Spring Constant	Kg/m
L	Free Span	m
M	Magnification Factor	dimensionless
m	Mass	Kg
n	Number of Cycles	dimensionless
P	Area Load	Pa
t	Time	sec
T	Transmissibility	dimensionless
$\mathcal{T}$	Period	sec
δ	Logarithmic Decrement	dimensionless
U	Potential Energy	J

w	Angular Frequency	rad/sec
w_d	Damped Natural Angular Frequency	rad/sec
w_n	Natural Angular Frequency	rad/sec
x	Mass Displacement	m
X	Output Displacement to Forced Vibration	m
x''	Acceleration	m/sec^2
x'	Velocity	m/sec
x_i	Initial Mass Displacement	m
x_n	Displacement at n Cycles	m
ϕ	Phase Shift	rad
Y	Input Displacement from Forced Vibration	m
Z	Viscous Damping Ratio	dimensionless

CHAPTER 1

INTRODUCTION

1.1. The Problem

Throughout the world, pallets are used for the automated handling, distribution, and storage of products. Unit-loads, consisting of packaged products fastened to pallets, aid materials handling with forklifts, pallet jacks, and other devices. A diagram of a typical unit-load is shown in Figure 1.1.

During transit, products are subjected to vertical and horizontal base excited vibrations generated by transport vehicles. If dynamic forces are severe enough, products and packaging within unit-loads can be damaged.

The transport environment is complex. A unit-load may be conveyed by tractor trailers, delivery trucks, ships, airplanes, rail cars, forklift trucks, pallet jacks, etc. Each mode of transportation has its own prevalent forcing frequencies and acceleration levels. Shocks and vibrations are transmitted through pallets to harm products; therefore, the dynamic response of pallets should influence product damage.

This work focused on three stringer pallets because they are commonly used in the United States (McCurdy and Ewers, 1986). Three stringer lumber deck,

pallets are made of deckboards, stringers, and fasteners.

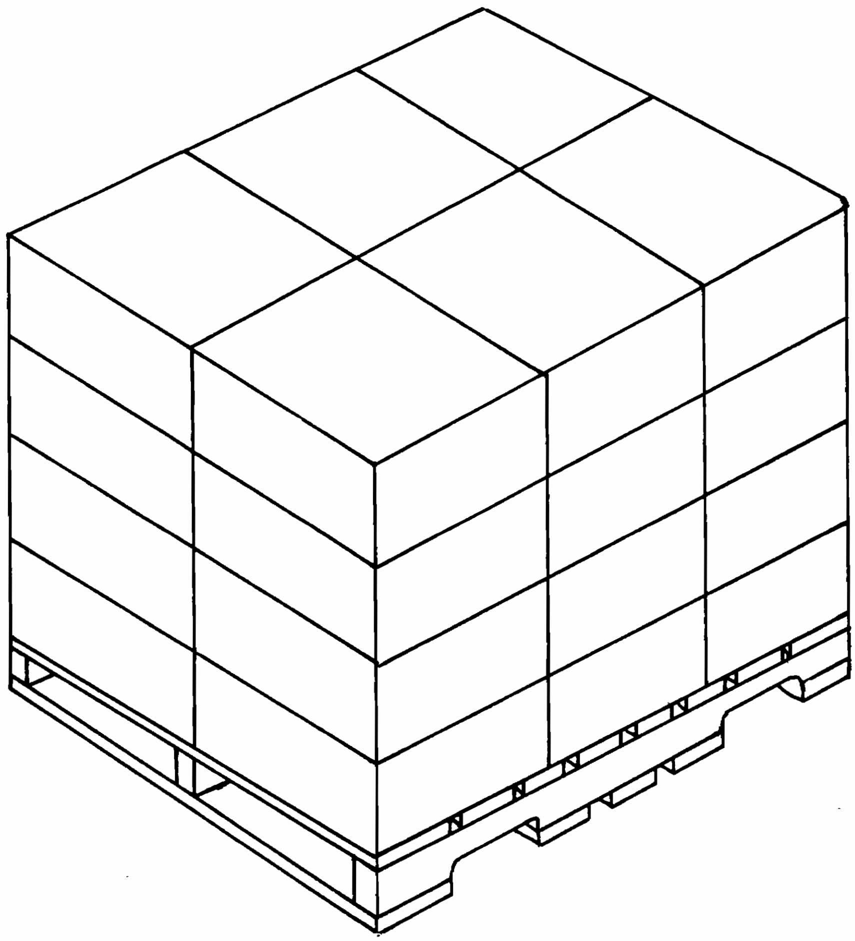

Figure 1.1. Schematic diagram of a typical unit-load used in product distribution.

The structure of a three stringer, flush, lumber deck, pallet is shown in Figure 1.2.

The factors thought to influence pallet vibrational response are deckboard stiffness, joint stiffness, load level, and load configuration. Pallet deckboards are most affected by vibration because they form continuous beams over several spans that are free to move in response to vertical vibration. Stringers act as supports for the top deck when a pallet is rigidly supported at its bottom deck. Fasteners will influence deckboard response through their affect on joint stiffness.

Unit-load resonance occurs when the forcing frequencies of the transport environment equal the natural frequencies of the loaded pallet's components. Resonant conditions are harmful because of the magnification of input accelerations.

If pallets can be designed so that unit-load resonance occurs outside the range of the predominant transport forcing frequencies, or at least different than product resonant frequencies, product damage may be prevented or reduced, lowering expenses for manufacturers, carriers, and consumers.

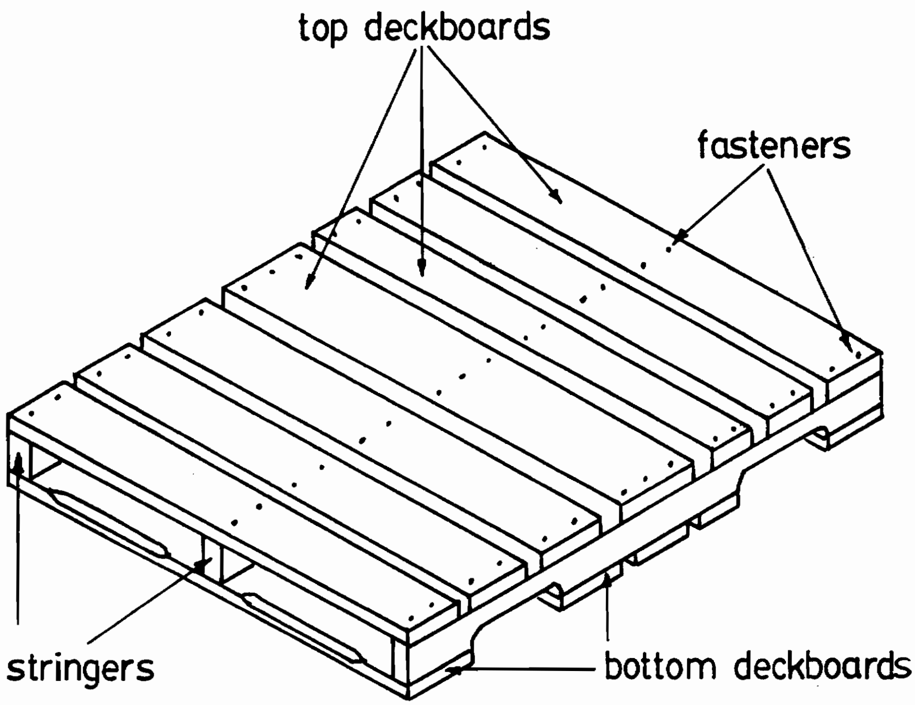

Figure 1.2. Schematic diagram of a three stringer lumber deck pallet commonly used for storage and handling of products.

1.2. Research Objective

The objective of this study was to determine how wood pallets respond to simple harmonic excitation by:

1. Developing a physical pallet section model to simulate the response of full size, three stringer pallets to forced vibration.

2. Examining the effects of deckboard EI, joint stiffness, and load level on the resonant response of three stringer pallets.

CHAPTER 2

Literature Review

2.1. Pallets

White et al. (1986) defined the pallet as "the platform upon which the load rests, which acts as an interface between the handling and shipping device and the package". The wood pallet is the primary load bearing platform, consisting approximately 85% of the pallet population (National Wood Pallet and Container Association (NWPCA), 1986). Pallets aid mechanical handling and protect products.

McCurdy and Ewers (1986) surveyed the pallet manufacturers of the United States and found the pallet trade to be a major forest product industry, representing the largest consumer of domestic hardwood lumber. In 1985, 2340 pallet manufacturers, employing 44,000 people, made 450 million pallets. McCurdy and Ewers said the most common pallet was the 1219mm by 1016mm version used by the grocery industry, which comprised one third of the pallets manufactured. The flush, three stringer, double-faced, non-reversible pallet was the most popular design. The food, paper/fiber, chemical/fluid and steel/metal industries were the major pallet consumers.

Pallets are typically classified as expendable (single use), general purpose (multiuse), and special-purpose.

Expendable pallets are used to ship a unit-load a single trip, general purpose pallets are constructed to be reused for many years, and special purpose pallets are designed for specific material handling applications.

Pallets are also described by style, design, and type. Style refers to the construction of the top and bottom deck, design refers to stringer or block constructions, and type is used to distinguish between the presence or absence of wings (Canadian Forestry Service (CFS), 1976).

Three styles of pallets are single faced, reversible double faced, and nonreversible double faced. Single faced pallets have a top deck only, reversible double-faced pallets have identical top and bottom decks, and nonreversible double-faced pallets have top and bottom decks with different spacings. The three pallet styles are shown in Figure 2.1.

Two pallet designs are stringer and block. A stringer is "a member continuous over the length of a pallet that supports the top deck or separates the top and bottom decks and provides space for entry of a lifting device" (CFS, 1976). A block is "a small, often rectangular, prismatic bearer of a full-four-way-entry or eight-way-entry pallet" (CFS, 1976). Both pallet designs are shown in Figure 2.2.

Pallets are distinguished by how they can be entered by handling equipment. Two-way pallets can be entered by a

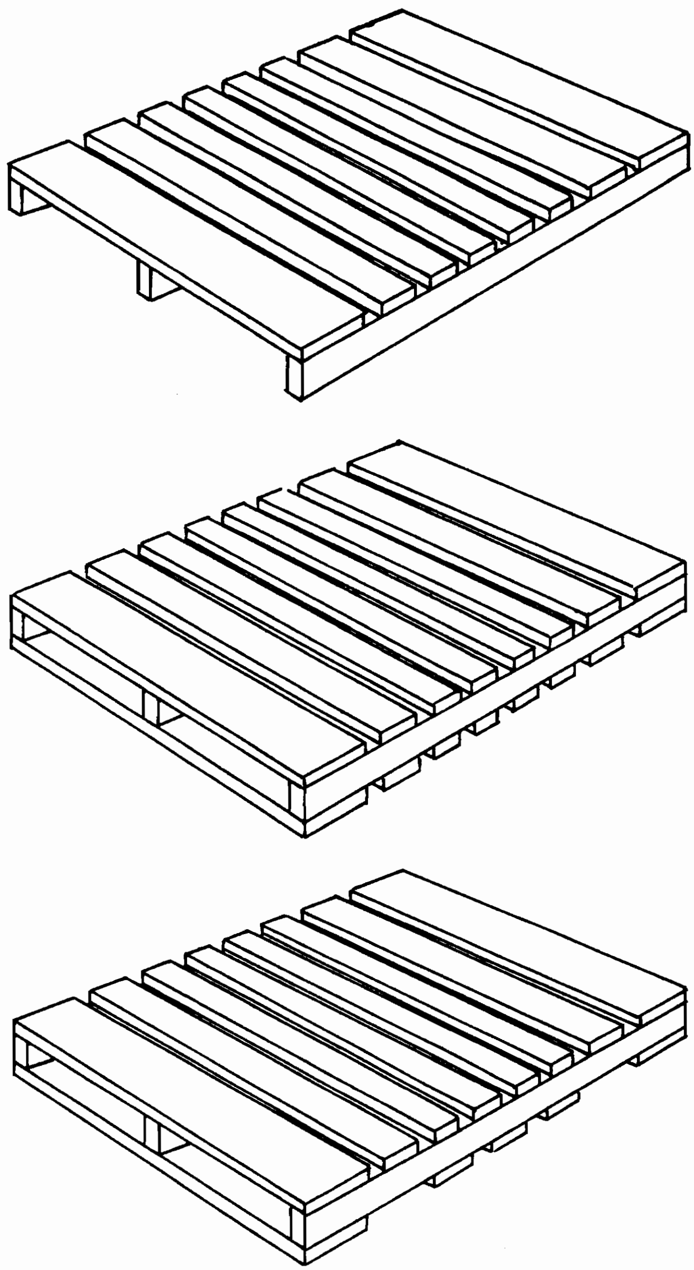

Figure 2.1. Pallet styles: single faced (top), reversible double faced (center), and nonreversible double faced (bottom) (after CFS, 1976).

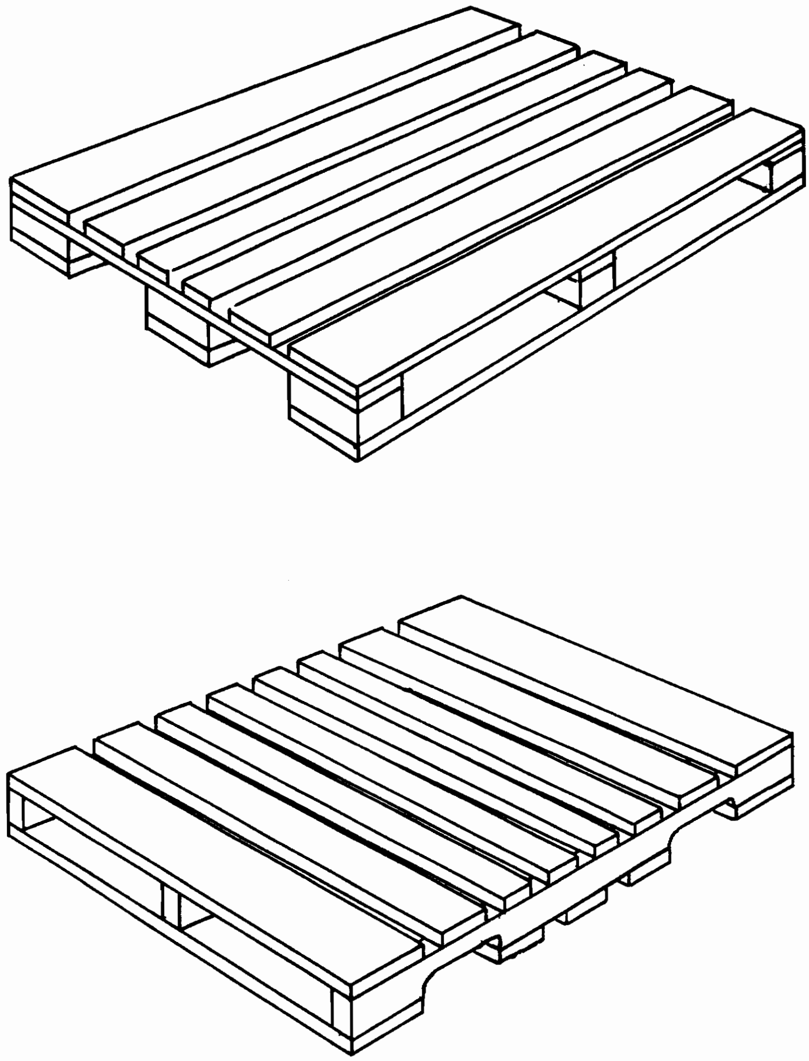

Figure 2.2. Pallet designs: block (top) and stringer (bottom) (after CFS, 1976).

lift truck or pallet jack from two directions. Partial 4-way pallets can be entered by a lift truck in four directions and by a pallet jack in two directions. Full-four-way pallets can be entered by either a pallet jack or a lift truck in four directions. Block pallets are full-four-way pallets, while stringer pallets are either partial four-way or two-way depending upon if their stringers are notched.

Three types of pallets are flush, single wing, and double wing. Flush pallets have no overhanging deckboards at their outside edges, single wing pallets have deckboards that overhang two outside edges of the top deck, and double wing pallets have deckboards that overhang the two outside edges of both the top and bottom decks. The three types of pallets are shown in Figure 2.3.

Pallets are assembled both manually and with machines. The simplest form of pallet manufacture uses workers equipped with hammers, nails, assembly tables, and pallet parts. Mechanized pallet assembly employs sophisticated pallet jigs, nailing machines, pallet turners, and pallet stackers.

The structure of wood pallets seems simple but their form introduces problems associated with wooden members and joints, like load sharing between members and nonlinear joint behavior (Loferski and McLain, 1987). Joints

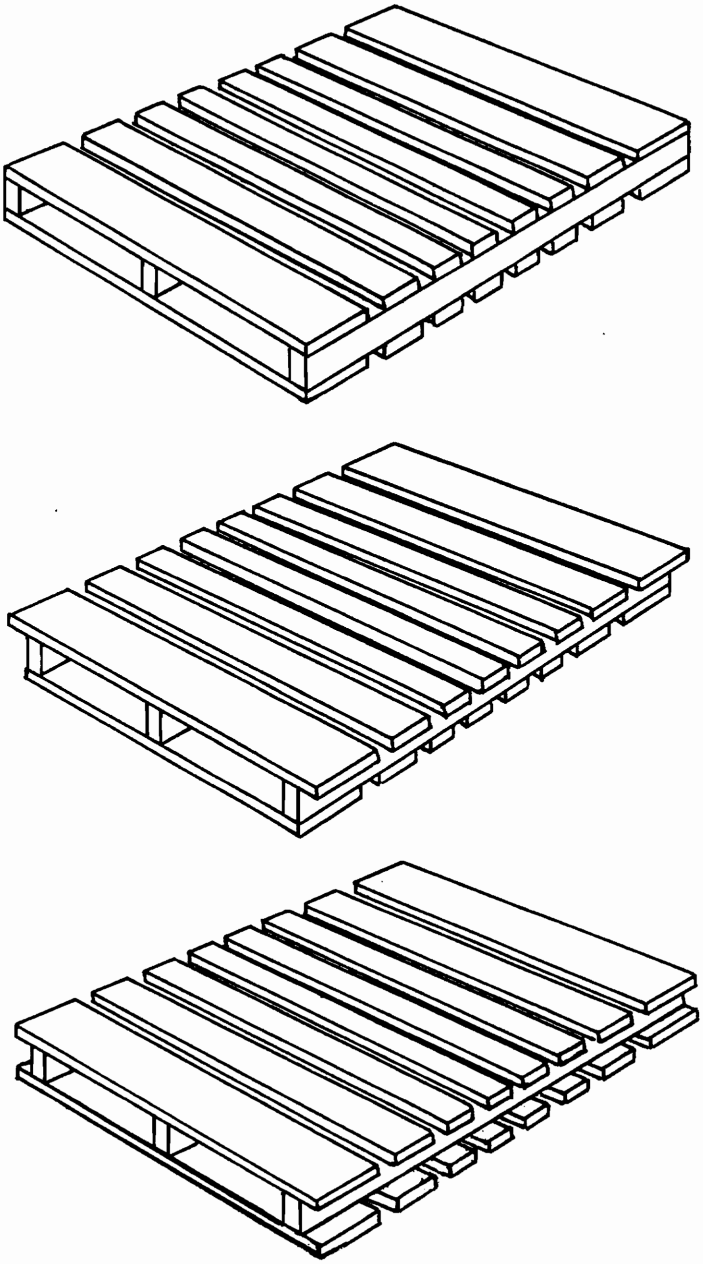

Figure 2.3. Pallet types: flush (top), single wing (center), and double wing (bottom) (after CFS, 1976).

determine pallet mechanical properties and durability (Wallin et al., 1976).

Pallets are made of lumber, plywood, particleboard, plastic, fiberboard, metal, and corrugated fiberboard. The trade name for wood pallet parts is shook. Pallet shook use and sale are governed by grading rules.

Most commercial wood species are used for pallet construction, but species use is often determined by local availability. Pallets are often constructed from mixed shook of several species. The NWPCA (Koch, 1980) divides hardwood pallet shook species into three groups according to density as is shown in Table 2.1. The U.S. Department of Defense species groups are shown in Table 2.2.

Pallet shook properties like modulus of elasticity (MOE), modulus of rupture (MOR), and density are species and shook grade dependent. Loferski (1985) said "the MOR is the computed extreme fiber stress that causes failure in material bending, and "the MOE is the indication of the stiffness of the material in the elastic range".

Many researchers have studied the mechanical properties of pallet shook. McLain and Holland (1982) measured the flexural properties of yellow-poplar, McLain et al. (1986) studied eastern oak shook properties, and Heebink (1959) studied the load carrying capacity of pallet deckboards from assorted species.

Table 2.1. National Wooden Pallet and Container Association hardwood pallet shook groups based on density (Koch, 1985).

Class A (least dense)	Class B (medium dense)	Class C (most dense)
Aspen	Ash (except white)	Beech
Basswood	Butternut	Birch
Buckeye	Chestnut	Hackberry
Cottonwood	Magnolia	Hard maple
Willow	Soft elm	Hickory
	Soft maple	Oak
	Sweetgum	Pecan
	Sycamore	Rock elm
	Tupelo	
	Walnut	
	Yellow-Poplar	

Table 2.2 U.S. Department of Defense shook species groups based on density (Koch, 1985).

Group 1	Group 2	Group 3	Group 4
Aspen	Doug-fir	Ash	Beech
Basswood	Hemlock	Soft Elm	Birch
Buckeye	Southern Pine	Soft maple	Hackberry
Cedar	Tamarack	Sweetgum	Hard Maple
Chestnut	Western Larch	Sycamore	Hickory
Cottonwood		Tupelo	Oak
Cypress			Pecan
Fir			Rock Elm
			White Ash

The most popular and efficient ways of forming pallet joints are with stiff stock nails, hardened nails, or staples. Pallet nails are threaded, either helically or annualarly, to improve withdrawal resistance (Stern, 1969). Pallet nails are rated by their hardness. To meet NWPCA guidelines, pallet nails must be either stiffstock or hardened. Koch (1985) said:

> **Stiff-stock nails** are bright, nonhardened, low to medium-high carbon steel nails.
>
> **Hardened steel nails** are made of medium to medium-high-carbon steel, heat treated and subsequently tempered. Heat treatment and tempering make the nails tough as well as stiff.

Pallet nail bend resistance is a measure of nail quality. The Morgan Impact Bend-Angle Impact (MIBANT) tester drops a weight down a frictionless shaft from a predetermined height onto the head of a nail, fastened in a vice at a 20 degree angle, bending the nail shank. The bend angle, measured in degrees, is used to rate nail shank bend resistance (Stern, 1972). There is a direct relationship between nail bend resistance and pallet rigidity, an indicator of pallet durability (Stern, 1980).

Static rotational characteristics of pallet joints were studied by Mack (1975), Kyokong (1979), Wilkinson (1985) and Samarasinghe (1987). Loferski and Gamalath (1989) developed a model to predict rotational stiffness of nail joints that defines nail joint parameters representing nail withdrawal resistance, nail head indentation, and wood crushing to form a spring system analog model for static joint stiffness using dimensionless springs. A measure of static joint rotational stiffness is rotation modulus. Samarasinghe (1987) defines rotation modulus as "the ratio of applied moment to the corresponding angular rotation".

E.G. Stern has produced a proliferation of literature on wood fasteners, pallet testing, and pallet properties. Kyokong (1979) developed a finite element method model to explain the mechanical behavior of wooden pallets from static loads. Fagan (1983) investigated load and support effects on pallets and developed a computerized pallet test apparatus for testing pallets with static loads. McLeod (1985) derived flexural design values for pallet shook under static loads. Loferski (1985) developed a reliability based design procedure for wood pallets for pallets under static load applications.

The information gathered by these investigators has been used in microcomputer software called the Pallet Design System (PDS) (McLain et al., 1984) produced by Virginia Tech in cooperation with NWPCA and the U.S. Forest Service to help industry use the engineering knowledge obtained by the Virginia Tech investigators to tailor pallet design for maximum material use efficiency, increased pallet life expectancy, decreased pallet cost, acceptable safety standards, and improved pallet marketing.

Most pallet research has been conducted on pallets under static load and support conditions. Pallets help with the transport and storage of products and they protect products. Damaged pallets will not protect products; therefore, most pallet research has focused on designing pallets to resist structural damage. Ultimately, this may not be the best approach for designing pallets to prevent product damage, because structurally sound pallets may damage products by altering the accelerations resulting from vibrations in the shipping environment.

Pallet response to vibration has not been seriously studied. Some unpublished research done by Del Monte Corp. has shown that pallets can cushion packages against shock and vibration (White et al. 1986). Brown (1991) has found that apple bins constructed with stiff decks help prevent vibration caused fruit damage. Trost (1989) mentions pallet response to vibration in ground transport of air shipments and notes that pallets can contribute to cushioning the unit-load.

2.2. Vibration Theory

To understand how loaded pallets react to vibrations, a general knowledge of vibration theory is required. The following sections contain a brief description of vibration theory.

2.2.1. Free Vibration

The simplest form of vibration is simple harmonic motion. This is modeled with a single degree of freedom (SDOF) spring-mass system consisting of a spring and a mass as is shown in Figure 2.4. The SDOF spring-mass system is the building block used to construct complex vibration models.

When a force is applied to an SDOF spring-mass system, the resulting motion can be plotted as mass displacement versus time with respect to the position of the mass at rest and is represented by,

$$x = A \cos(wt + \phi) \tag{2.1}$$

Where $\mathbf{x}$ is displacement, $\mathbf{A}$ is the amplitude, $\mathbf{w}$ is angular frequency, $\mathbf{t}$ is time, and $\boldsymbol{\phi}$ is the phase shift or the position of the mass at time = 0 (Serway, 1983). The period ($\mathcal{T}$) is the time that it takes for a spring-mass system to complete one oscillatory cycle. Frequency ($\mathbf{f}$) is

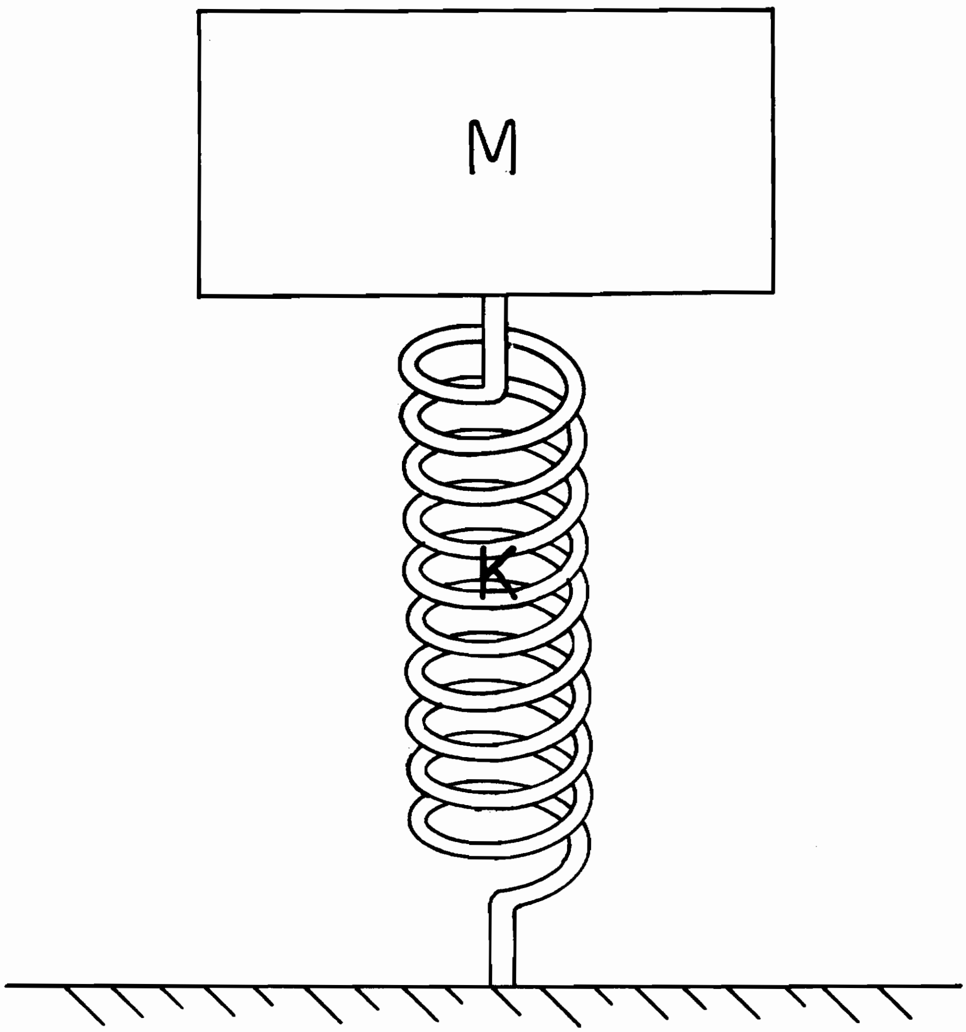

Figure 2.4. A single degree of freedom spring-mass system: K is the linear spring constant and M is mass.

the inverse of the period and is expressed as the number of cycles that occur in a specified time period, usually expressed as cycles per second (**Hz**). The expression for this relationship is,

$$f = 1 / T \tag{2.2}$$

Angular frequency (**w**), representing a rotating vector used in vibration mathematics, is measured in radians per second. Frequency (**f**) is related to angular frequency (**w**) in the following relationship,

$$f = w/2\pi \tag{2.3}$$

The velocity of the mass can be found by differentiating Equation (2.1) with respect to time to obtain,

$$x' = -Aw \sin(wt + \phi) \tag{2.4}$$

Acceleration of the mass is determined by differentiating Equation (2.4) with respect to time to obtain,

$$x'' = -Aw^2 \cos(wt + \phi) \tag{2.5}$$

Which can be further simplified to the form,

$$x'' = -w^2 x \tag{2.6}$$

Maximum values for velocity and acceleration are found from the two relationships,

$$x'_{max} = Aw \tag{2.7}$$

$$x''_{max} = Aw^2 \tag{2.8}$$

The spring-mass relationship is of great significance. When a spring-mass system is displaced from static equilibrium the spring exerts a force on the mass. This force (**F**) is represented by,

$$F = -kx \tag{2.9}$$

Where **k** is the linear spring constant and **x** is the mass displacement. Newton's second law can be applied with,

$$F = mx'' = -kx \tag{2.10}$$

thus,

$$x'' = -(k/m)x \tag{2.11}$$

Where **x"** is acceleration. This can be modified to form the second order differential equation of motion for an undamped SDOF spring-mass system under free vibration in the form,

$$mx'' + kx = 0 \tag{2.12}$$

Where **m** is mass, and **k** is the spring constant, and **x"** is acceleration. A particular solution for this differential equation is given in Equation (2.1).

The ratio of k/m can be expressed as angular momentum squared ($\mathbf{w^2}$). This relationship is used in determining the natural frequencies of SDOF spring-mass systems through the following two equations,

$$w_n = \sqrt{k/m} \tag{2.13}$$

or,

$$f_n = 1/(2\pi)\ \sqrt{k/m} \tag{2.14}$$

One can examine simple harmonic motion through discussing the energy present in an oscillating system. Kinetic energy is influenced by the mass, while potential energy is a property of the spring and the position of the mass. Kinetic energy (**K**) for a single degree of freedom spring mass system under free vibration is expressed as,

$$K = 1/2\ mv^2 \tag{2.15}$$

While elastic (**U**) potential energy for the spring is given as,

$$U = 1/2\ kx^2 \tag{2.16}$$

Serway (1983) noted this about simple harmonic motion:

1. The displacement, velocity, and acceleration all vary sinusoidally with time but are not in phase.

2. The acceleration of a particle is proportional to the displacement, but in the opposite direction.

3. The frequency and period of motion are independent of amplitude.

2.2.2 Damping

Damping is the dissipation of vibrational energy occurring within a vibrating spring-mass system that is caused by the internal molecular friction of the spring or friction within the medium that the system is vibrating. Damping is present in all vibrating systems including unit-loads. Viscous damping, the simplest damping model, is a force that resists vibration proportional to the velocity of the vibration (Brandenburg and Lee, 1985). This is often modeled as a dashpot. A viscously damped SDOF spring-mass system is shown in Figure 2.5. Other damping models are Coulomb damping, quadratic damping, velocity n'th power damping, and hysteretic damping (Ruzika and Derby, 1971).

Damping of a spring-mass system is characterized as having a damping coefficient (**C**). Critical damping (**C_c**) occurs when a spring-mass system is damped to prevent oscillation and is represented by,

$$C_c = 2\sqrt{km} \tag{2.17}$$

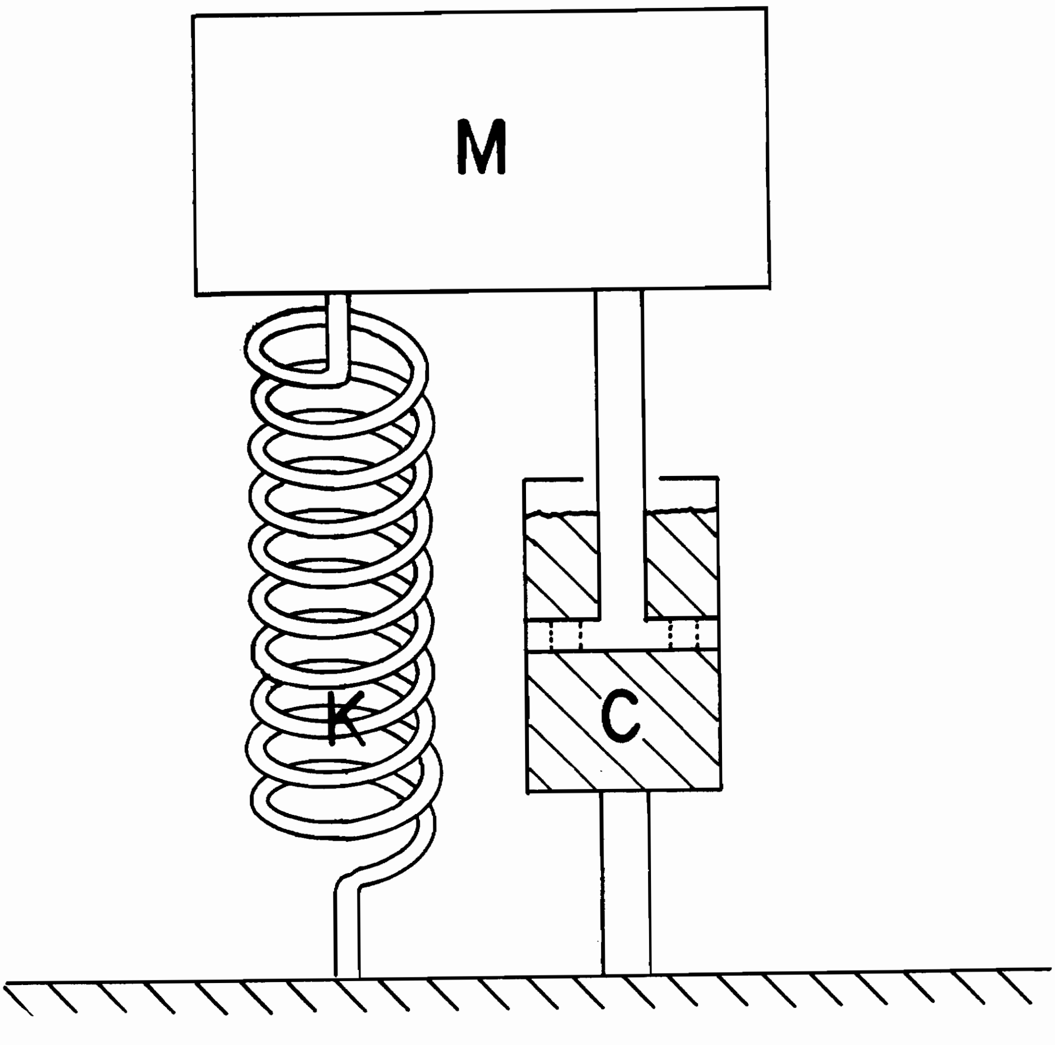

Figure 2.5. A viscously damped single degree of freedom spring-mass system: K is the linear spring constant, C is viscous damping, and M is mass.

The ratio of a spring-mass system's actual damping coefficient to the critical value is called the damping ratio (**Z**),

$$Z = C/C_c \tag{2.18}$$

James et al. (1989) said that damping ratios can be used to define four damping responses: $Z > 1$, overdamped response, $Z = 1$, critically damped response, $0 < Z < 1$, underdamped response, and $Z < 0$, unstable response.

The second order differential equation of free vibration for a damped SDOF spring-mass system is:

$$mx'' + cx' + kx = 0 \tag{2.19}$$

The natural frequency ($\mathbf{w_d}$) of a damped spring mass system is given as,

$$w_d = \sqrt{k/m - (c/2m)^2} \tag{2.20}$$

If Equation (2.19) is divided by the mass, it yields the equation,

$$x'' + (c/m)x' + (k/m)x = 0 \tag{2.21}$$

The natural frequency and damping ratio definitions can be used to modify Equation (2.21) to the form,

$$x'' + 2Zw_n x' + w_n^2 x = 0 \qquad (2.22)$$

A method of quantifying the effect of damping on a vibrating SDOF spring-mass system is logarithmic decrement (**δ**), which describes the decreases of successive cycles of a freely vibrating damped spring-mass system. Bodig and Jayne (1982) give the following equation for the logarithmic decrement of a system,

$$\delta = (1/n)\ (\ln\ (x_i/x_n)) \qquad (2.23)$$

Where **δ** is logarithmic decrement, **x_i** is the initial amplitude, **x_n** is the amplitude at **n** cycles, and **n** is the number of cycles.

2.2.3. Forced Vibration

Forced vibration occurs when an oscillating force is applied to a spring-mass system. The studies included in this thesis concentrated on the response of pallets subjected to simple harmonic forced vibration. To emphasize the importance of the study of structural harmonic excitation, Thomson (1981) said:

> Although pure harmonic excitation is less likely to occur than periodic or other types of excitation, understanding the behavior of a system undergoing harmonic excitation is essential to comprehend how the system will respond to more general types of excitation.

The differential equation of motion for an SDOF system experiencing harmonic excitation is,

$$mx'' + cx' + kx = F_o \sin (wt) \qquad (2.24)$$

Where **w** is the forcing function frequency and $\mathbf{F_o}$ is the maximum applied force. A particular solution for this differential equation is,

$$x = A \sin (wt - \phi) \qquad (2.25)$$

Where **A** is the amplitude of oscillation and $\boldsymbol{\phi}$ is the phase shift of displacement. $\boldsymbol{\phi}$ is derived from the equation,

$$\tan \phi = \frac{2Z(w/w_n)}{1 - (w/w_n)^2} \qquad (2.26)$$

An important relationship occurs between the forcing frequency (**w**) and the natural frequency of the spring-mass system ($\mathbf{w_n}$). This is determined by the ratio of these factors ($\mathbf{w/w_n}$). When $\mathbf{w/w_n}$ is less than 1, the spring is resisting the forces, when $\mathbf{w/w_n}$ is equal to 1 (resonance), the damper is resisting the forces, and when w/w_n is greater than 1, the mass is resisting the forces. These relationships are important to the study of the forced vibration of wood pallets. Unit-loads should be designed so that the ratio of the predominant forcing frequencies in the transport environment to the natural frequencies of loaded pallets (w/w_n) is less than 1.

2.2.4. Resonance

Resonance of a SDOF spring-mass system occurs when a forcing frequency is equal to a spring-mass's natural frequency ($w/w_n = 1$). Resonance of unit-loads causes product damage; thus, the goal of vibration protective packaging is to prevent product resonance. Resonance is defined by the Handbook of Chemistry and Physics (1985) as:

> 1. The phenomenon of amplification of a free wave or oscillation of a system by a forced wave or oscillation of exactly equal period. The forced wave may arise from an impressed force upon the system or from a boundary condition. The growth of the resonant amplitude is characteristically linear in time.
>
> 2. Of a system in forced oscillation, the condition which exists when any change, however small, in the frequency of excitation causes a decrease in the response of the system.

2.2.5. Transmissibility

"Transmissibility is the nondimensional ratio of the response amplitude of a system in steady-state forced vibration to the excitation amplitude. The ratio can be of forces, displacements, velocities, or accelerations" (Harris and Crede, 1976). Transmissibility is also called a magnification factor (**M**) because it can be multiplied by the input amplitude to get a vibrating spring-mass system's output amplitude. Since magnification of acceleration caused by pallets damages products, pallets should be designed to reduce transmissibility. Transmissibility is expressed by the relationship,

$$|X/Y| = \sqrt{\frac{(w/w_n)^2}{[2Z(w/w_n)]^2+[1-(w/w_n)^2]^2}} \qquad (2.27)$$

If damping is small, Equation (2.27) can be converted to,

$$|X/Y| = \sqrt{\frac{(w/w_n)^2}{1-(w/w_n)^2}} \qquad (2.28)$$

Where **X** is the output amplitude and **Y** is the input amplitude.

2.2.6. Vibration Isolation

Vibration isolation is used to reduce the magnification of forces transmitted to an object under forced vibration. The spring-shock absorber systems used in automobile suspensions are examples of vibration isolators.

James et al. (1989) said that isolators are only effective when the ratio of the forcing frequency to the spring-mass system's natural frequency (w/w_n) is greater than $\sqrt{2}$. The isolator is tuned to a specific frequency where the mass is experiencing transmissibilities of less than 1, where it would otherwise experience resonance.

It is the author's opinion that conventional reusable pallets cannot be constructed to act as isolators. Pallets require structural stiffness for static applications; however, special purpose pallets may be constructed to act as isolators. These pallets would be designed for specific products at a given payload.

2.3. Wood and Wood Fastener Dynamics

To understand how pallet components may be affected by forced vibration, a literature search was conducted on how wood and wood connectors are influenced by dynamic forces. The term dynamic used throughout the following text refers to forces applied to masses resulting in motion.

Polensek (1975) examined the damping capacities of wood floors and found that floor damping to be dominated by the quality of the joints. He studied the effect of elastomeric adhesives used in wood joist floor construction and found that the use of adhesives caused slightly lower damping capacities than nailed flooring.

Yeh et al. (1971) found wooden structural damping to be caused by the internal molecular friction of wood, slip between joint surfaces, and the use of elastomeric adhesives. They said "In wood member vibration, shear deformations provide the dominant mode for high internal damping" and "at low amplitudes, wood damping is frequency independent". They found adhesive damping to be caused by shear strain on the interface between the wood member and the adhesive. Yeh et al.'s findings show that the quality of wood joints influences structural damping.

Polensek and Bastendorf (1987) considered damping in nailed joints of light-framed wood buildings. They believed the main sources of damping to be friction between members in nailed joints. Joint properties were highly nonlinear, with joint stiffness and damping ratios decreasing under increasing loads.

Atherton et al. (1980) observed damping and slip of nailed joints. Plywood with 3/8 inch and 5/8 inch thickness, and gypsumboard were tested in single nail slip joints that were cyclically loaded in tension and compression. Energy absorption of the joints was of primary interest. Load level was found to have the greatest effect on energy absorption. Joints were divided into friction and nonfriction groups. Friction joints had members that formed tight joints while nonfriction joints had gaps between members. The 3/8 inch plywood absorbed more energy than the 5/8 inch plywood, and the 5/8 inch plywood had the highest slip moduli.

Chou and Polensek (1987) examined damping and stiffness of nailed joints' responses to drying. They found damping ratios were smaller with joints with gaps than joints without gaps. Increased load level increased damping.

Wilkinson (1976) studied vibrational loading of mechanically fastened wood joints for in-plane shear. His experimental model was a single fastener slip joint using both nails and bolts. He tested his specimens at 5, 10, 15, and 20Hz. One of his significant conclusions was "the stiffness of mechanically fastened wood joints increases appreciably when subjected to vibrational loadings as compared to static loading. This is seemingly due to a rather large increase in the apparent rate of loading". A rapidly loaded joint may be significantly stiffer than the same joint under similar static loading.

Soltis and Mtenga (1985) studied the strength of wood joints subjected to dynamic loads. The mode of joint loading was in plane shear. Their study's objectives were to compare static loading versus dynamic loading of a one nail slip joint, to compare nailed joint resistance over many loading cycles, to find the effect of frequency, and to determine the effect of load history. They tested two joint configurations with two kinds of nails at 1 Hz and 10Hz forcing frequencies. Their conclusions were "that at small deformations the increase in joint capacity due to higher loading is offset by decreased joint capacity due to load cycling" and that "At large deformations and numbers of load cycles, joint resistance decreases".

Kurtenacker (1965) studied pallet nail joints under static and dynamic loads using common wire nails, threaded nails, and pallet staples. He compared fastener static withdrawal with impact withdrawal. The difference between static and impact withdrawal testing was the rate of loading where impact withdrawal used a shock pulse to extract the nail. Pallet fastener dynamic withdrawal resistance was found to be greater than static withdrawal resistance.

2.4. The Transport Environment

To test pallets for their responses to forced vibration, a knowledge of the forcing frequencies that a unit-load may be subjected to is required. Vibration test methods used for products and packaging are based on studies of actual transport vibration conditions.

Ostrem and Godshall (1979) performed a comprehensive survey of the transport environment, studying both shock and vibration. They categorized the forcing frequencies that a product may experience transported by trucks, trains, aircraft, and ships. Vibration data was presented in power spectral density (PSD) form measured in g^2/Hz versus frequency, which is used to describe transport vibration in the frequency domain.

Many variables contribute to truck vibration like the suspension system, load, speed, road condition, trailer condition, and cargo location. Three primary frequency ranges occur in truck transport: 0 to 5Hz attributed to the natural frequencies of the suspension system, 10 to 20Hz due to unsprung mass, and 50 to 100Hz caused by tire natural frequency. Frequencies above 100Hz were attributed to worn suspensions. Average peak accelerations were near 0.5g.

Truck vibration is usually random and it is difficult to quantify frequencies and accelerations that a product may be subjected to. Fast Fourier Transform (FFT) devices are used to transform primary forcing functions from the time domain to the frequency domain in random vibration studies. FFT provides plots of relative power versus frequency to distinguish primary forcing frequencies.

For rail cars, forcing frequencies are usually below 20Hz with less than 0.75g peak excitation. The periodic forcing functions are associated with nonwelded track sections (Ostrem and Godshall, 1979).

Aircraft forcing frequencies range from 1 to 1000Hz. The vibration from turbojet aircraft is typically from 50 to 70Hz depending upon the speed of the propellors during flight. Aircraft forcing frequencies are typically higher and acceleration levels are considerably lower than vibrations in either truck or rail car transport. Expensive fragile products, transported over long distances, are typically shipped by air.

Ostrem and Godshall (1979) said ship vibration has been observed as two distinct levels that are dependent on sea conditions. Ship engines and propellors are responsible for low level continuous vibrations in calm seas. The higher levels of vibration were attributed to rough seas where the ships ride swells and waves impact on the ships' sides. Wave vibration occurs at frequencies from 0.03 to 0.20Hz.

Sharpe et al. (1974) observed the vibration environment of common motor carriers. The purpose of this research was to describe the truck transport environment so that laboratory tests could be developed to test packaged products under appropriate conditions. Their study showed that vertical accelerations are the most prevalent form of truck transport vibration, with peak accelerations occurring as high as 2g.

Ostrem (1980) stated that product tests to simulate truck vibration should consist of frequencies between 3 to 100Hz with peak acceleration levels of 0.5g. Testing for the rail environment should consist of frequencies from 3 to 100Hz with a peak acceleration level of 0.25g. Testing for aircraft vibration uses vibration dwell periods at the frequencies and acceleration levels of interest.

Recent studies of longitudinal and lateral truck vibration have been conducted by Antle (1989). He discovered that below 20Hz, lateral and horizontal acceleration is considerably less than vertical acceleration and that the loading of the truck influenced horizontal vibration. Heavier loaded trucks exhibited higher lateral and longitudinal acceleration levels. Peak accelerations for lateral and longitudinal vibrations occurred at the same frequencies as vertical vibrations.

Pierce (1990) compared vertical vibration levels between leaf spring and air ride trailer suspensions. He found that well maintained air ride suspension systems outperform leaf spring suspensions for product protection because they cushion input vibration better than leaf springs.

Gens (1975) studied vibration found in industrial forklift trucks. He found that lift truck load level had the greatest affect on vibration levels with heavier loads exhibiting the lowest vibrational response. Most vertical vibrations occurred at frequencies less than 10Hz but peak vertical accelerations were recorded as high as 20g. In the transverse direction of lift truck travel, peak acceleration occurred as high as 37g. Vibrations levels encountered in distribution are shown in Table 2.3.

2.5. Packaging Engineering

The pallet acts as an interface between packaged products and the transport vehicle. To understand how the pallet may act as a viable component of a packaging system, one must examine how packaging reacts to forced vibration and how this may influence the response of products, unit-loads, and pallets. Packaging engineers have well established criteria for protective packaging. One must understand packaging and the sensitivity of products to develop methods to design pallets to act as interactive components of packaging systems.

Many products are susceptible to vibration damage. For some products, specific parts, called critical elements, are damaged by vibration. Examples of critical elements may be a central processing unit in a microcomputer, or the filament in a lamp. A goal of packaging is to protect critical elements from experiencing resonant conditions.

Packaging engineers (Brandenburg and Lee, 1985) give

Table 2.3. Periodic vibration found in the distribution environment (Brandenburg and Lee, 1985).

Transport Mode	Description	Frequency (Hz)	Peak g's ♦
Truck	Suspension	2-7	0.50
	Unsprung Suspension	10-20	0.25
	Structural	50-100	0.25
Rail Car	Suspension	2-7	0.50
	Lateral	0.75-2	0.75
	Structural	1	-
Trucks on Flat cars	Vertical	2-4.6	1.0
	Roll	0.70-3.1	10
Aircraft	Propeller	2-10	-
	Jet	100-200	-
Ship	Sea and Engine	10, 100	-

♦ g is acceleration due to gravity ($9.8 m/sec^2$)

five classifications for fragility of typical packaged articles: extremely fragile, very delicate, moderately delicate, moderately rugged, and rugged. Examples of extremely fragile products are missile guidance systems and precision aligned test instruments. Very delicate items may include mechanically shock mounted instruments and electronics equipment. Delicate items comprise aircraft accessories, electric typewriters, cash registers, and electrically operated office equipment. Moderately delicate items are television receivers and other aircraft accessories. Moderately rugged equipment are laundry equipment, refrigerators, and appliances. Machinery is classified as rugged.

O'Brien et al. (1969) used accelerometers in filled, wooden, peach bins to measure how fruit was damaged during transport, and found transport damage was an important factor influencing the quality of fruit before processing. He also found that dissimilar wood bins influenced this damage differently. The resonant frequencies of cling peaches was near 17Hz.

O'Brien et al. (1965) developed a model to find the resonant frequencies of fruit by determining fruit stiffness through static compression testing. Fischer et al. (1989) found strawberries and grapes to be damaged by forcing frequencies between 7.5 to 10Hz. Peleg and Hinga (1986) developed a laboratory method of simulating vibration in the transport environment to test fruit resonant frequencies.

Shabal (1989) stated the primary modes of product damage are:

1. Breakage of the product
2. Lack of containment
3. Contamination
4. Deterioration of the product
5. Loss of marketability
6. Potential safety hazard
7. Loss of communication

Packaging is used for the physical protection and restraint of products and may consist of boxes, bags, plastic wrap, paper wrap, steel banding, metal drums, plastic drums, etc. Because of the vast array of packaging materials, their affects on load distribution and resonant responses of unit-loads are unknown.

Godshall (1971) tested the vibrational response of corrugated fiberboard boxes, developing a method of determining the stiffness of boxes using a nondestructive compression test. The empty corrugated boxes were placed on a vibration table, top loaded, and tested for resonant response. The resonant frequencies of the boxes ranged from 8 to 18Hz with transmissibilities as high as 6.7. Average damping ratios for the boxes were 11.5%.

Urbanik (1978,1981,1984,) has developed mathematical models for box loads represented by multiple degree of freedom spring-mass systems. Urbanik's interests were how vibrational loading changed the corrugated fiberboard strength characteristics and how boxes influence transport environment caused product vibration. He found that vibrational loading may cause box failure. Urbanik (1990) presented a way of modeling stacks of corrugated boxes as single degree of freedom spring-mass systems, where a force plate is used with load cells, and a shaker table to quantify the response of a tier of boxes.

Brandenburg and Lee (1985) discussed vibrations in distribution, mechanical shock, cushion design, vibrational testing of packages, and the use of accelerometers. Shabal (1986) defined the primary functions of packaging as containment, dispensability, and communication. A similar view was held by Fiedler (1985), who said the major purposes of packaging are to contain the product, protect the product, and advertise the product.

Packaging is divided into categories: primary packaging contains the product, secondary packaging contains the primary package, and tertiary packaging contains the secondary packaging. An example of this relationship is a unit-load of canned beverage. The beverage is the product contained by the primary package, the can. The secondary package is the paperboard container enclosing the cans, the case. The tertiary package is the corrugated boxes containing the cases. Complexities of packaging are determined by product fragility.

If the product is rugged, no packaging may be necessary. Some manufacturers have found that by adding to the durability of the product, they need to use less packaging and the result is a better product that requires less expensive packaging materials.

Trost (1989) used accelerometers placed beneath top pallet decks to find input vibrations to unit-loads conveyed in air transport systems and noted that pallet response influenced unit-load response. He found that at the center of the pallet, transmissibilities were not as great as those found at the pallet's outside edges.

Paine et al. (1980) vibration tested palletized unit-loads. Their experimental methods consisted of vibration testing unit-loads for resonance from 3 to 100Hz using 0.35 and 0.5g peak accelerations at sweep rates of 1 octave per minute and resonance dwell testing. For the case goods unit-loads tested in this study, the resonant frequencies of the palletized loads ranged from 8 to 30Hz, with transmissibilities as high as 6.7. They also tested 210l filled steel drums and found resonant conditions to cause drum failure. Paine et al. (1980) stated that testing of unit-loads is important for the following reasons:

1. Stacks of fibreboard cases can show resonance effects in the range of 8 to 30Hz when subjected to vertical vibration. These have been associated with bruising of fruit in transit. Likewise filled metal drums can fail rapidly when vibrated at a resonance frequency.

2. Resonance effects in stacks of cases or in palletised loads can cause the upper, or in more severe examples, all packages to lift from those below; with the loss of contact, frictional effects are also lost, allowing packages to shift horizontally producing instability. These effects occur when the acceleration level exceeds 1g which can occur when low accelerations are applied to the base of the load.

3. Resonance accentuates all effects such as scuffing, fatigue, bruising, and loosening of closures and fastenings.

Vandermeerssche (1981) studied how rail car vibration damaged palletized beverage cans and found that vibration causes can surfaces to become abraded and their seals to fail, causing cans to leak and lose carbonation. Beverage can seals were damaged by frequencies between 40 and 55Hz. Using a horizontal shaking device, Vandermeerssche developed a laboratory method to simulate the rigors of the rail car transport environment for unitized canned beverage. Three types of pallets were evaluated for their affect on a canned beverage unit-load using a vertical shaking device. The pallet that magnified acceleration the least was a double faced, plywood, nine block, 4-way pallet that had 76mm corner posts, 102mm side posts, and a 127mm center post. No descriptions of the other pallet designs were given.

2.6. Vibration Testing

The studies included in this thesis used vibration tests of unit-loads, pallets, and pallet sections. Vibration testing is used extensively by product manufacturers and the packaging industry. ASTM test standards have been developed to find vibrational characteristics of both products and packaging. Most vibration testing is directed at one product or one package at a time and the unit-load including the pallet is seldom tested (White et al. 1986).

DiGeronimo and Rehm (1985) examined dynamic testing to predict packaging performance using vibration testing machines. Commonly used vibration testing equipment are hydraulic shaker devices consisting of vibration tables, actuators, hydraulic power supplies, and closed loop controls. This equipment coupled with accelerometers is used to find packaging and product response to vertical and horizontal forced vibration. Accelerometers are placed on the vibration table to monitor input accelerations, and in or on products to find their responses.

The ASTM vibration testing standards for products and packaging are ASTM D3580-89 "Standard Test Methods for Vibration (vertical sinusoidal motion) Testing of Products", ASTM D999-86 "Standard Methods for Vibration Testing of Shipping Containers", and ASTM D4728-87 "Standard Test Method for Random Vibration Testing of Shipping Containers".

CHAPTER 3

Factors Influencing Wood Pallet Vibrational Response

Before beginning pallet vibration research, the author tried to identify all factors that might affect pallet response to forced vibration. These factors are interrelated and complex, and may include deckboard dimensions, stringer dimensions, location of components, wood material effects, fastener effects, and load effects.

White, et al. (1986) said, "The key variables which are thought to influence (pallet) dynamic response are joint stiffness, deckboard EI/L^3 (effective stiffness), and load level." **E** is the modulus of elasticity, a material property. The moment of inertia value (**I**) and deckboard free span (**L**) are determined by pallet design. Joint stiffness is controlled by joint design and pallet load level and configuration is set by the pallet user.

The elastic modulus (**E**) of wood used in the construction of pallets varies because wood is a natural material that is both heterogeneous and anisotropic. Modulus of elasticity is a function of the species and grade of wood used for pallet components.

Most pallets are assembled with unseasoned shook. The change of shook moisture content during pallet use will influence their member and joint stiffness as unseasoned wood dries and shrinks. Drying causes defects like checks, splits, and cracks, that may alter member stiffness. As pallets are used, they may be exposed to conditions that foster chemical and biological decay, also causing changes in member stiffness. The pallets and pallet sections used during this study were constructed from kiln dried, defect free lumber to remove some of the variability associated with defects and unseasoned wood.

Dynamic forces imposed upon pallets may cause damage or fatigue of the pallet structure, altering the stiffness of pallet joints and members. Deckboards and stringers may be broken, nail joints may be damaged, or pallet shook and nails may fatigue.

Connectors influence joint stiffness. Joint characteristics that may affect pallet dynamic response are the number of fasteners used per joint, fastener stiffness, fastener wire diameter, fastener bending stiffness, fastener penetration depth, fastener thread characteristics, etc.

The load also affects unit-load vibrational response due to the load's mass, type, and stiffness; the unit-load may act as a collection of spring-mass systems. A pallet may support equally distributed loads, unequally distributed loads, concentrated loads, or different variations of line loads. Stiff loads may bridge during dynamic excitation, and pallet load levels may range from a few hundred to several thousand Newtons.

Pallets are not typically fastened to their supporting bases and are free to move in any direction forces may displace them. If pallets are made so that they rock under dynamic loading, excessive load shifting may result if the load is not properly restrained. Pallet rocking may be caused by warped stringers and deckboards or by members with unequal thickness.

Load restraint may influence pallet response. Common methods of load restraint are plastic stretch wrap, metal and plastic banding, and break-away adhesives on boxes. Forces exerted by such restraints may alter unit-load response. These pallet and load characteristics, and their potential interactions, render the study of the response of pallets and unit-loads to harmonic excitation very complex.

The chapters that follow contain a three phase approach to the study of a complex problem. First, using one unit-load configuration, the relative effect of different pallet designs on unit-load resonant response was studied. Second, the characteristics of pallet design that most influence pallet dynamic response were identified. Third, an experiment was developed that permitted the isolation and study of the most significant factors.

CHAPTER 4

Effect of Pallet Design on Unit-Load Resonance

Abstract

Pallet influence on unit-load response to forced vibration has not been seriously studied. Ten pallet designs, one metal, two plastic, one corrugated fiberboard, and six wood, loaded with a 4532N case goods load were tested for unit-load resonance and transmissibility at forcing frequencies between 2 and 100Hz with a 0.5g peak accelerations. Resonant frequencies of the unit-load supported by the ten pallet designs ranged from 9 to 13Hz with transmissibilities at resonance ranging from 4.0 to 5.61. As much as a 4Hz downward shift in resonant frequency and a 34% increase in transmissibility of the unit-load was found by changing the pallet design.

Key Words: unit-load, vibration, transmissibility, resonance, pallets

4.1. Introduction

Vibration found within the distribution environment contributes to product damage. Packaged products are shipped on pallets that act as unit-load supports to ease the handling of products during storage and shipping. Pallet influence on vibration transmitted to the unit-load from shipping devices has not been previously studied.

Sharpe et al. (1974) found vibrations occurring during shipping to be primarily vertical. Vibration of unit-loads is caused by base excitation from transport vehicles. Unit-load resonance occurs when forcing frequencies equal the natural frequencies of components in the unit-load. At resonance (f_r), product damage is most likely to occur.

Transmissibility (**T**), a magnification factor, is the ratio of output acceleration to input acceleration of a forced vibration excited spring-mass system. Transmissibility is used as a relative measure of acceleration magnification.

Paine, et al. (1980) vibration tested palletized unit-loads, using methods similar to those used in this study, and found that box unit-loads had resonant frequencies ranging from 8 to 30Hz with transmissibilities as high as 6.7. For liquid filled metal drums, resonant conditions were found to cause drum failure.

Ostrem and Godshall (1979) defined the vibrational forcing frequencies found within the common carrier shipment environment. They found the predominant vertical forcing frequencies in trucks, rail cars, ships and aircraft to be below 100Hz, with the most damaging frequencies occurring below 20Hz. They found the transport environment to be complex with unit-loads being simultaneously subjected to multiple forcing frequencies. Most transport vibration was found to be random. Brandenburg and Lee (1985) reported maximum acceleration levels of the transport environment to be 0.5g for trucks, 0.75g for rail cars, and 10g for trucks on flat cars.

The produce industry ships products that are susceptible to vibration damage. Fischer, et al. (1989) found grapes and strawberries to be damaged by forcing frequencies from 7.5 to 10Hz. O'Brien, et al, (1969) observed that peaches were damaged by forcing frequencies near 17Hz. Brown (1991) found that apple bins constructed with stiff deck components help reduce fruit vibration damage.

Corrugated fiberboard shipping containers are a major packaging material. Godshall (1971) found the resonant frequencies of top loaded corrugated fiberboard containers to range from 8 to 18Hz.

The pallet's contribution to unit-load dynamic response has not been seriously considered. Trost (1989) noted that different vertical transducers on pallets exhibited different vibrational amplitudes. The center of the pallet was shown to cushion the load.

Pallet designs vary significantly, being constructed from wood, metal, corrugated fiberboard, or plastic in numerous shapes and sizes. The objective of the study reported here was to determine how pallet design influenced the resonant response of a case goods unit-load.

4.2. Materials

The pallet load used in this study was a 4532N, corrugated fiberboard box load consisting of 21 boxes, column stacked three layers high. The box dimensions were 480mm long by 320mm wide by 170mm high, and they contained coated paper.

Ten pallet designs were tested to find their influence on the resonant response of this load. All pallet designs tested in this study are used commercially and their general descriptions are found in Table 4.1. Pallet designs are shown in Figures 4.1 through 4.9.

4.3. Methods

The testing apparatus consisted of a servohydraulic vibration test system including a vibration table, a hydraulic power unit, and closed loop controls. A detailed description of the vibration test system is included in Appendix B.

Piezoelectric accelerometers coupled with vibration meters linked to a three channel plotting/measurement system monitored, recorded, and plotted change in peak acceleration of the unit-load versus frequency. The accelerometers were

Table 4.1. General descriptions of the pallets used to determine the effects of pallet design on the resonant response of a case goods unit-load.

Pallet No.	Dimensions (mm) (l x w x h) *	Style/ Material	Type/ Economic Life
1	1219 X 1016 X 114	3 stringer steel	partial 4-way multiple use
2	1178 X 1016 X 152	9 block ♦ HDPE plastic	full 4-way multiple use
3	1219 X 1140 X 152	9 block HDPE plastic	full 4-way multiple use
4	1219 X 1016 X 127	3 stringer Oak deckboard	partial 4-way multiple use
5	1219 X 1016 X 120	3 stringer yellow pine lumber deck	partial 4-way multiple use
6	1219 X 1016 X 120	3 stringer Douglas-fir lumber deck	partial 4-way multiple use
7	1219 X 1016 X 133	9 block yellow pine plywood panel deck	full 4-way multiple use
8	1219 X 1016 X 159	9 block yellow pine lumber deck	full 4-way multiple use
9	1219 X 1016 X 133	4 stringer corrugated fiberboard	2-way single use
10	1016 X 1016 X 130	3 stringer yellow poplar lumber deck	2-way single use

♦ HDPE = high density polyethylene.
* l x w x h = Length by Width by Height

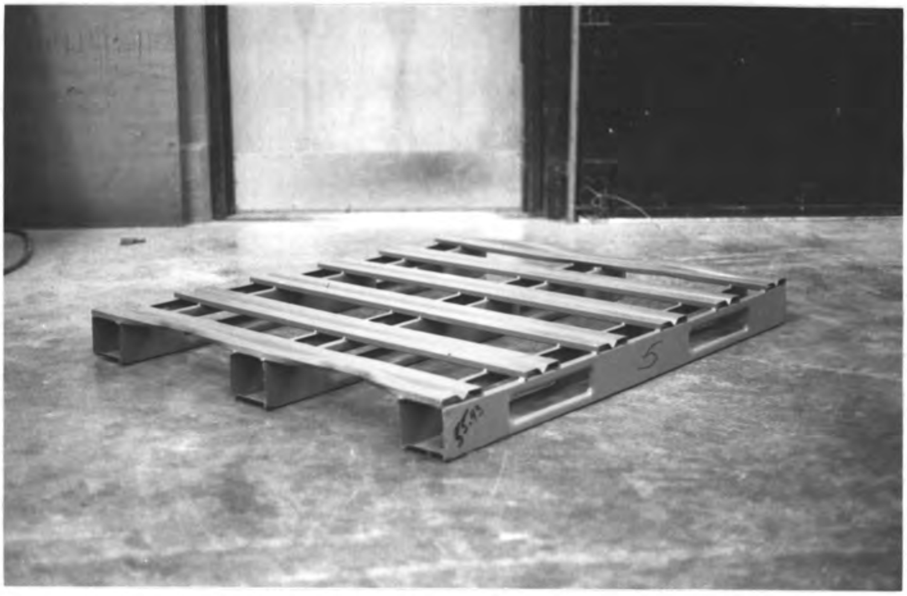

Figure 4.1. Pallet 1, the steel three stringer partial 4-way pallet.

Figure 4.2. Pallet 2, a plastic 4-way block pallet.

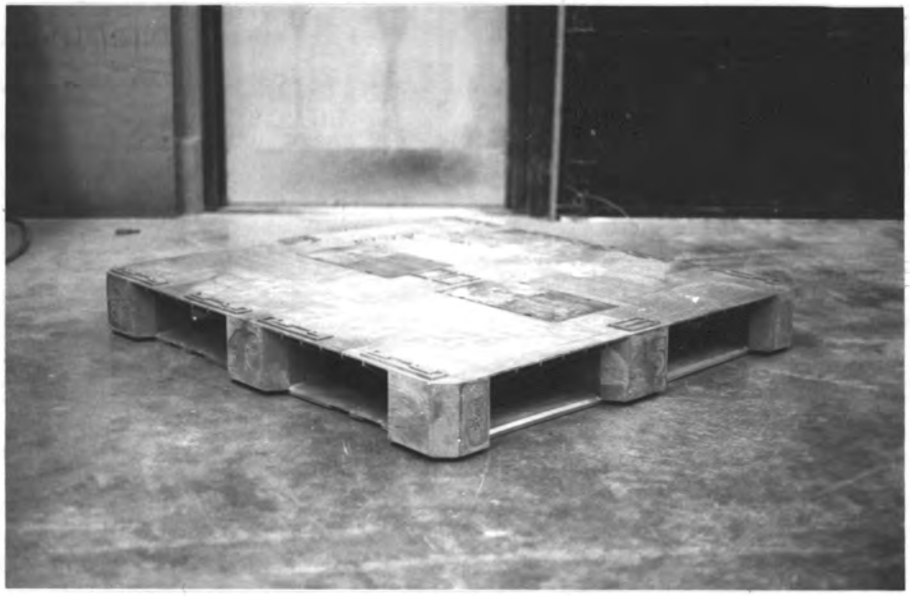

Figure 4.3. Pallet 3, a plastic 4-way block pallet.

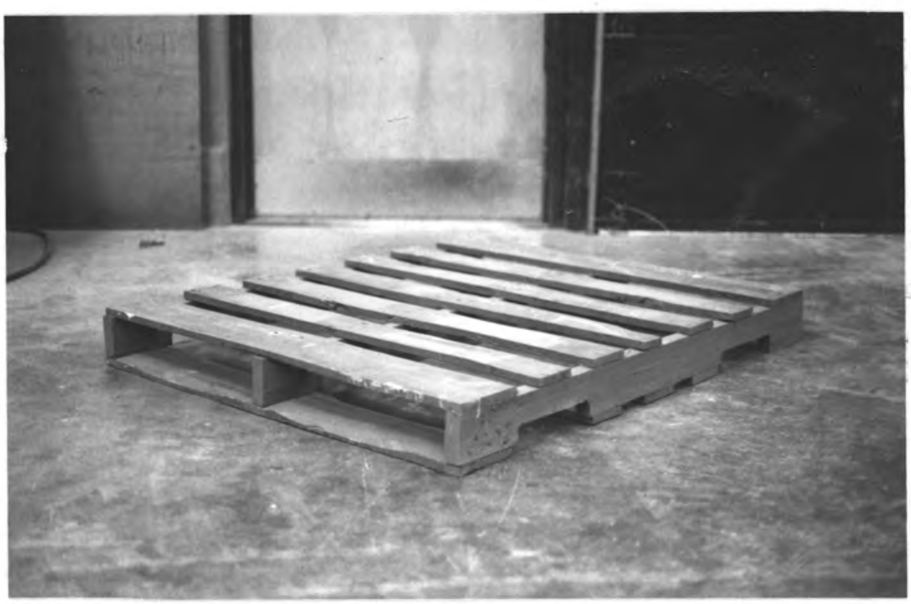

Figure 4.4. Pallet 4, the partial 4-way three stringer hardwood pallet.

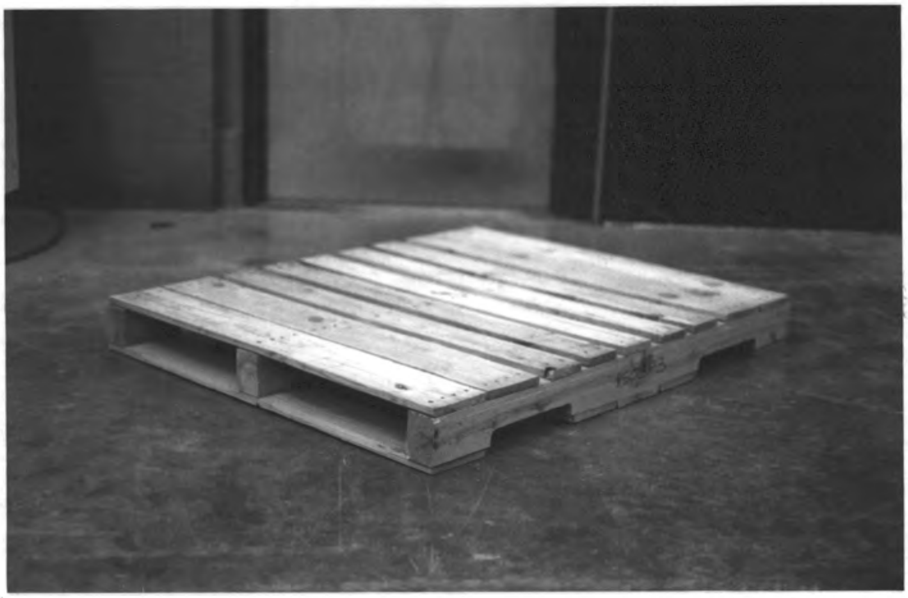

Figure 4.5. Pallets 5 and 6, the partial 4-way three stringer softwood pallet.

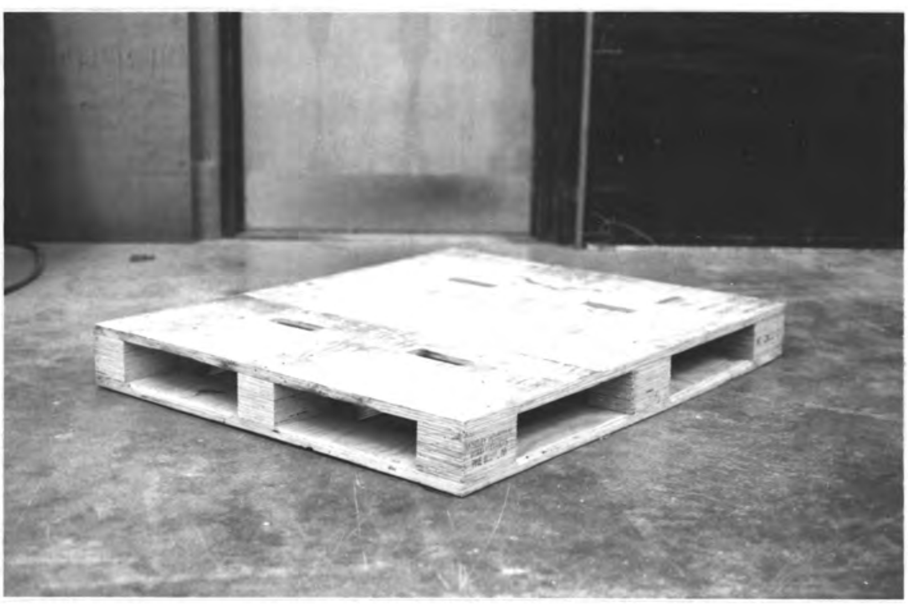

Figure 4.6. Pallet 7, the softwood plywood 4-way block pallet.

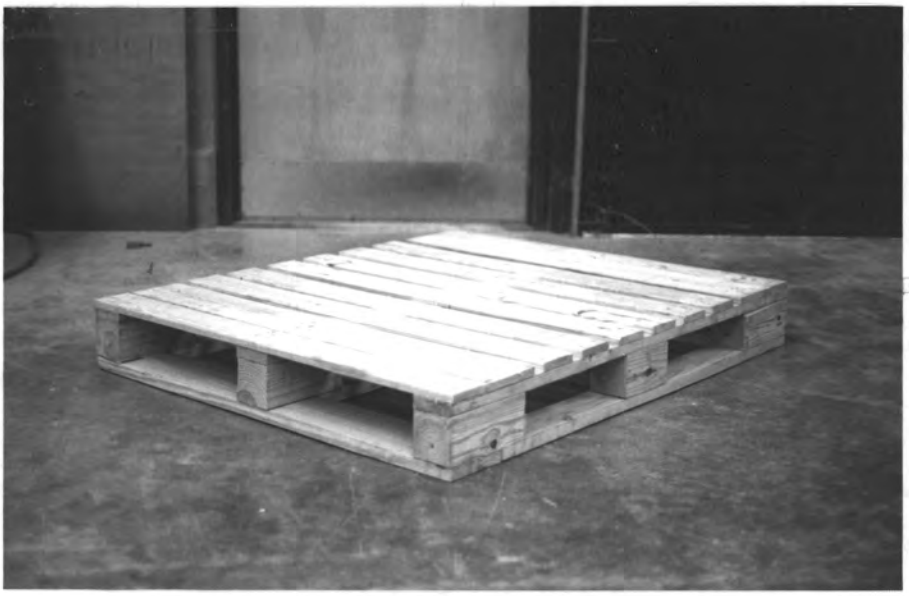

Figure 4.7. Pallet 8, the softwood lumber 4-way block pallet.

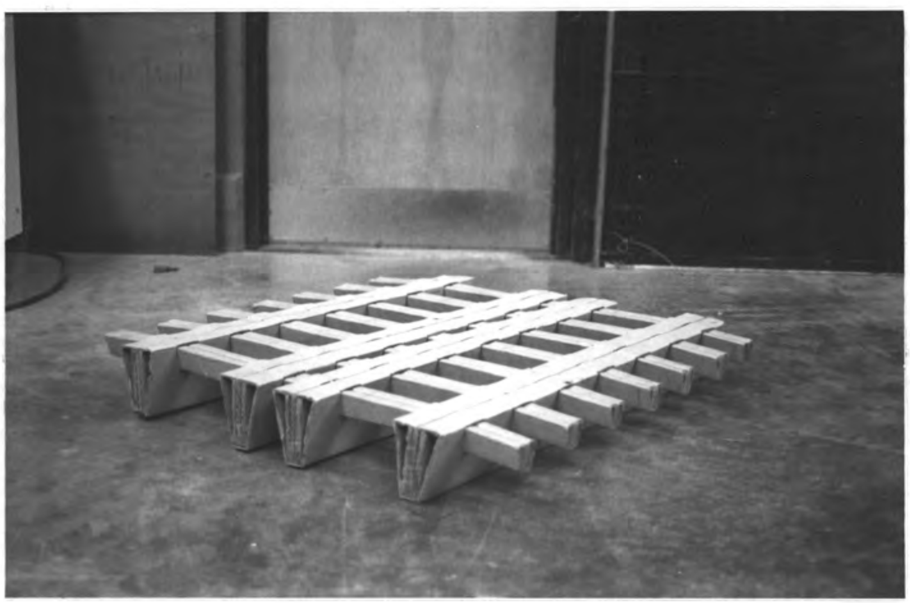

Figure 4.8. Pallet 9, the corrugated fiberboard four stringer 2-way pallet.

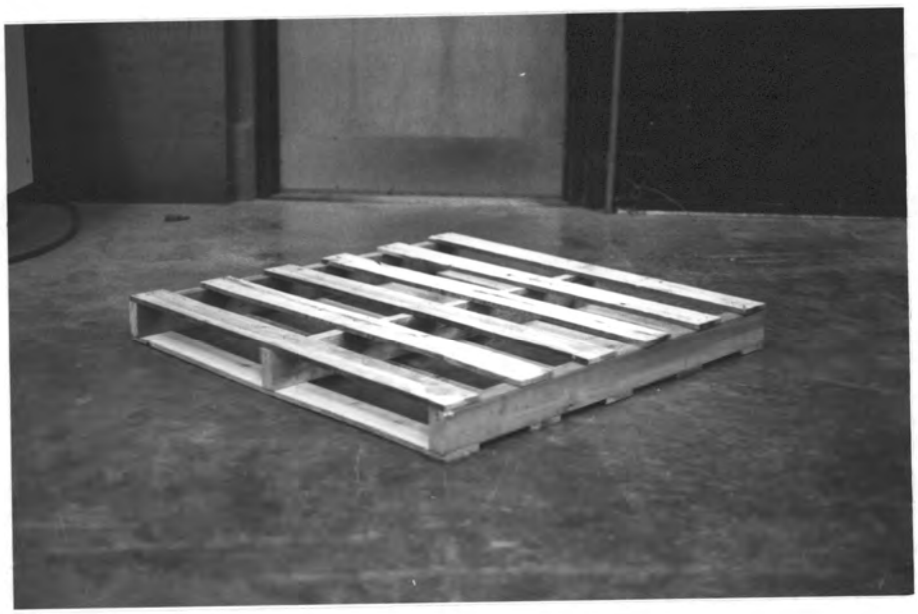

Figure 4.9. Pallet 10, the expendable three stringer hardwood 2-way pallet.

accurate to within 3% over the frequency band and acceleration levels used during testing.

The boxes were hand loaded onto the pallets. The box load was restrained with four wraps of 510mm wide by 0.2mm thick, stretch wrap. The stretch wrap was necessary to keep the unit-load on the pallet during testing.

After the palletized load was placed on the vibration table using a forklift, an accelerometer was fastened in the vertical axis direction at the top layer of the product in the center box. This location was monitored to obtain an indication of unit-load response.

Each palletized load was subjected to a series of five vibration tests. The vibration test was a single sine-log sweep from 2 to 100Hz at 0.5g peak acceleration with a rate of change of 1 octave per minute. Five tests were conducted to determine whether vibration testing would change unit-load resonant response.

Resonance was assumed to occur at the maximum acceleration of the unit-load. The resonant frequency ($\mathbf{f_r}$), is the frequency at maximum acceleration. Transmissibility (**T**) is the ratio of output/input acceleration at the resonant frequency. A representative vibration test plot for a unit-load test used to derive resonant frequencies and transmissibilities is shown in Figure 4.10. In addition to these measures, the shape of the load on the various pallets

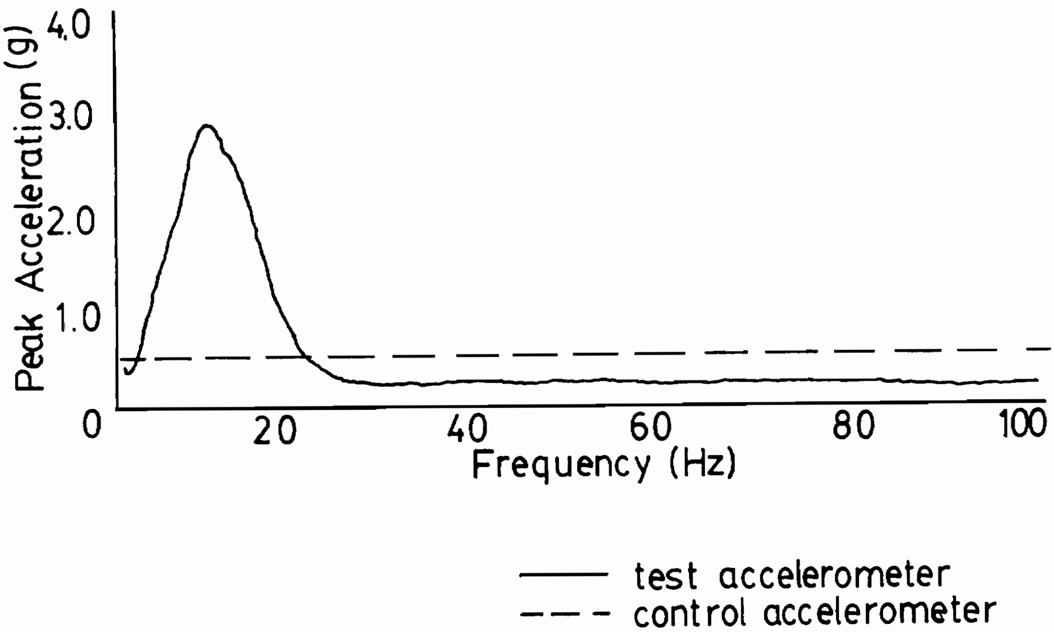

Figure 4.10. A representative vibration test plot for vibration tests used to determine unit-load resonant response.

was noted before and after each test series was conducted to describe load shifting.

4.4. Results

A summary of the test data is shown in Table 4.2, and the results of all vibration tests are included in Appendices A-1 and A-2 for resonant frequency ($\mathbf{f_r}$) and transmissibility (**T**), respectively. Coefficient of variation (C.V.) was used as a relative measure of variation. The C.V.'s for transmissibilities and resonant frequencies measurement for the each pallet tested were typically 2 to 4%. Some exceptions were noted that are worthy of discussion. If pallets did not sit flat on the test table, other modes of vibration significantly affected transmissibility. For example, the wood pallets, during testing, tended to "rock" resulting in C.V.'s as large as 27% for transmissibilities across a five test series. Although only one corrugated fiberboard pallet was tested, transmissibility of the unit-load supported by this pallet design tended to increase through the first, second, and third cycle resulting in a C.V. of 11%. The high C.V. for this pallet may be a result damage occurring to the corrugated fiberboard structure. No other pallet designs indicated any structural change from testing. One could conclude that a single vibration test cycle is adequate for

Table 4.2. The average resonant responses to forced vibration of the unit-load supported by the different pallet designs.

Pallet Design	Number of Replications	f_r ♦ (Hz)	C.V.♥ (%)	T♣	C.V. (%)	Relative Loadshift
No Pallet (Initial)	1	13	-	4.04	-	none
No Pallet (Final)	1	13	-	4.20	-	none
1. Steel Stringer	1	11	-	4.61	-	none
2. Plastic Block	1	9	-	5.60	-	slight
3. Plastic Block	1	13	-	4.33	-	none
4. Oak Stringer	5	11	5	4.98	13	great
5. Y. Pine Stringer	5	11	6	5.05	16	none
6. D. Fir Stringer	5	11	8	4.98	12	none
7. Plywood Block	5	12	4	4.88	9	none
8. Y. Pine Block	4	12	3	4.00	6	none
9. Corrugated Stringer	1	12	-	5.21	-	none
10. Y Poplar Stringer	3	9	-	4.42	-	none

♦ f_r = resonant frequency
♣ T = transmissibility
♥ C.V. = coefficient of variation

most pallet designs made of wood, plastic, or metal to estimate resonant response.

The palletless control unit-load exhibited an average resonant frequency of 13Hz and transmissibility of 4.04. After the palletized unit-load tests were completed, the palletless unit-load was tested again and found to have a resonant frequency of 13Hz and a transmissibility of 4.20, a 4% increase over the initial test, indicating that the box load's resonant frequency did not change but transmissibility did. This may have been caused by the boxes being damaged during testing.

The unit-load supported by pallet 1, the steel pallet, had a resonant frequency of 11Hz with a transmissibility of 4.61, which was 10% greater than the unit-load without a pallet. The box load did not shift with this pallet.

The unit-load supported by pallet 2, a plastic block pallet, had a resonant frequency of 9Hz and an average transmissibility of 5.60, which was 33% greater than the palletless unit-load. Slight load shifting was evident with this design. This pallet was the worst performer because it exhibited the highest transmissibilities. The top and bottom deck had an extended lip designed to help prevent load shifting. The pallet rested on this lip which formed the perimeter of the bottom deck, thus the pallet vibrated as two plates consisting of the top and bottom deck separated by the posts. During resonance, the center post would lift away from and return to the table's surface, impacting on the test table. The shock forces from the center block impacting on the test table may have influenced this pallets transmissibilities. Because of this, this pallet's response was different than those of the other pallets.

The unit-load supported by pallet 3, the other plastic block pallet, had a resonant frequency of 13Hz and an average transmissibility 4.33, a 3% increase in transmissibility over the palletless unit-load. There was no load shifting with this pallet design. This pallet was superior to pallet 2 because it was constructed with a solid top deck. Pallet 2 had a more flexible top deck caused by numerous cut outs in its structure.

The unit-load supported by pallet 4, an oak, three stringer pallet, exhibited an average resonant frequency of 11Hz and transmissibility of 4.98, a 19% increase in transmissibility over the palletless unit-load. Excessive rocking, due to the pallet not resting level on the vibration table's surface, caused extreme load shifting.

The unit-load supported by pallet 5, the yellow pine, three stringer pallet, exhibited an average resonant frequency of 11Hz and a transmissibility of 5.05, a 20%

increase in transmissibility over the palletless unit-load. No load shifting was present with this pallet design.

The unit-load supported by pallet 6, the same design as pallet 5 except that Douglas fir was used in fabrication, had an average resonant frequency of 12 Hz and a transmissibility of 4.98, a 19% increase over the palletless unit-load. There was no load shifting with this pallet design, either. No significant differences in response was noted between pallet 5 and pallet 6.

The unit-load supported by pallet 7, the yellow pine, plywood block pallet, had an average resonant frequency of 12Hz and a transmissibility of 4.87, a 16% increase over the palletless unit-load. No load shifting was present with this pallet design.

The unit-load supported by pallet 8, the lumber deck, yellow pine, block pallet, had an average resonant frequency of 12Hz and transmissibility of 4.00, almost equivalent to that of the palletless unit-load. There was no load shifting with this pallet design. This pallet was the best performer because it caused the unit-load to have a resonant frequency equivalent to the unit-load without a pallet, and it exhibited the lowest transmissibility. It was constructed so that deck free span was less than that of the other pallets, which contributed to higher pallet deck stiffness.

For the unit-load supported by pallet 9, the corrugated fiberboard pallet, the resonant frequency was 12Hz with a transmissibility of 5.21, a 24% increase over the palletless unit-load. No load shifting occurred with this pallet.

The unit-load supported by pallet 10, the yellow poplar three stringer, single use shipping pallet, exhibited an average resonant frequency of 9Hz with a transmissibility of 4.42, a 5% increase over the palletless unit-load.

All pallets tended to reduce the unit-load's resonant frequency and increase its transmissibility. The stiffness of the pallet deck influenced response with the stiffer pallets having higher resonant frequencies and lower transmissibilities.

Pallets that rocked under vibrational loading caused the most load shifting due to induced horizontal motion. To prevent load shifting, pallets should be constructed to prevent rocking, which may be accomplished by designing pallets to rest evenly on the supporting surface. To accomplish this, pallets should be constructed with dry, straight materials to help prevent changes in pallet structures caused by drying.

The simple harmonic forced vibration responses of only one unit-load was tested in this study. Other loads and load configurations subjected to other types of vibration may act differently than the unit-load tested here.

It appears that all of the multiuse wood pallets tested had similar resonant responses. Plastic pallet design influences unit-load resonant response more significantly than wood pallets. The expendable pallets, both wood and corrugated fiberboard, tended to lower the resonant frequencies of unit-loads and increase transmissibilities more than the reusable pallets, because single use pallets are constructed with decks with lower stiffness than reusable pallets. These expendable pallets would be more apt to induce vibration caused product damage.

4.5. Conclusions

This study shows that pallets influence the response of unit-loads to vibration, and that pallet design may affect vibration caused product damage and load shifting. The pallets tested here tended to reduce unit-load's resonant frequency and increased its transmissibility.

The unit-load, when placed on the different pallets, exhibited resonant frequencies ranging from 9 to 13Hz with transmissibilities ranging from 4.0 to 5.61, as much as 34% more transmissibility than what the unit-load experienced without a pallet. From the results of this study, it would appear that pallets constructed with stiff decks perform best for protection of products from vibration by increasing unit-load resonant frequency response, and decreasing unit-load transmissibility at resonance.

CHAPTER 5

Development of a Simplified Test Specimen for Studying Wood Pallet Response to Forced Vibration

Abstract

Three studies were conducted to develop a simple physical model for full size pallets used in material handling environments. The reactions of unit-loads to forced vibration are complex with many factors influencing their response like support conditions, load conditions, and pallet design characteristics. To study pallet design effects, a pallet section model was developed. Full size pallet response to forced vibration was investigated. The effects of different loadings on pallet response were also examined. A pallet section and load model were verified to produce acceptable correlations with the results of testing full size pallets.

Key Words: Vibration, pallet, load effects, modal analysis, pallet sections.

5.1. Introduction

Wood pallets are the bases of unit-loads. Pallets are the typical interfaces between packaged products and acceleration caused by vibration generated by shipping devices. The transport environment has been described by Ostrem and Godshall (1979) and Sharpe et al. (1974).

Products transported by trucks may be excited by periodic forcing frequencies from 0 to 100Hz, with peak acceleration levels as high as 2g.

A common wood pallet design used in the United States is a partial four-way, flush, nonreversible, 1219mm by 1016mm three stringer pallet design (McCurdy and Ewers, 1986). The components of this wood pallet consist of stringers deckboards and fasteners as was shown previously in Figure 1.2. Test results reported in Chapter 4 show pallet deckboards vibrate during handling and shipping. For a three stringer pallet, the top and bottom deckboards are continuous beams over two spans whose stiffness is determined by the span between stringers, deckboard EI, and joint stiffness. Assuming a spring analog, the variation in stiffness, deckboards, and joints should influence the response of pallets and unit-loads to forced vibration.

The objectives of the studies reported here were:

1. To find the significant modes of vibration of three stringer pallets.

2. To choose a load type for further pallet research.

3. To develop a pallet section model to emulate full size pallet response.

5.2. Pallet Vibration Modal Analysis

To find which of the many possible vertical and horizontal modes of pallet vibration that would significantly influence unit-load performance, three sets of comparative tests were conducted on pallets. The tests performed included tests with pallets attached to the test table and tests with unrestrained pallets. The purpose was to isolate through restrained and unrestrained testing, certain modes of vibration and to find which modes correspond to the highest levels of transmissibility or potential hazard to products.

5.2.1 Unrestrained and Restrained Testing of Full Size Pallets

Two pallets were assembled from dry, defect free yellow-poplar (Liriodendron tulipifera), a wood species commonly used for pallet construction. The lumber had an average moisture content of 8%. Pallet deckboards were nondestructively tested for modulus of elasticity using a static deflection test described in Appendix C. One pallet design that was tested is shown previously in Figure 1.2. The top and bottom deckboard thickness was 13mm. The other pallet design was similar to the first except that it did not have a bottom deck. Stringer dimensions were 1219mm long by 38mm wide by 89mm high. For both pallets, the lead deckboards had widths of 146mm and the central deckboards had widths of 95mm. In the latter pallet design, the bottom deck was excluded so that stringers could be bolted to the vibration table. The fastener used to construct both pallets was a 57mm long, 2.8mm shank diameter, helically threaded hardened steel pallet nail. The complete fastener specifications are shown in Table 5.1. The load on the pallet consisted of 4459N crushed lime in 560mm long by 406mm wide by 76mm high bags. Each bag weighed 223N. Bags were column stacked. The configuration for the restrained pallet tests is shown in Figure 5.1.

The vibration testing equipment employed was a servohydraulic vibration test system described in Appendix B that consisted of a vibration table, a hydraulic power source, an actuator and electronic controls. The vibration test consisted of a single sine log sweep from 2 to 100Hz with a 0.5g peak acceleration and a rate of change of 1 octave per minute. A data acquisition system was used to monitor and plot a test accelerometer's and the vibration table's control accelerometer's peak acceleration responses versus frequency.

For the unrestrained pallet tests using the nonreversible double deck pallet, accelerometers were placed beneath deckboards at center of span between stringers and under stringer notches. During 25 successive tests, the accelerometer was moved to various locations on each of

Table 5.1. Fastener specifications for pallet nail used in test pallet construction.

Information	Description
VPI Fastener No:	2349
Length:	57mm
Head Diameter:	6.35mm
Thread Length:	44.5mm
Thread Diameter:	3.22mm
Wire Diameter:	2.82mm
Thread Angle:	62°
Flutes:	4
Number of Helix:	8/38mm
MIBANT Angle:	27°
General Appearance:	bright, blunt, chisel point.
Fastener Withdrawal Index:	81
Fastener Stiffness Index:	80

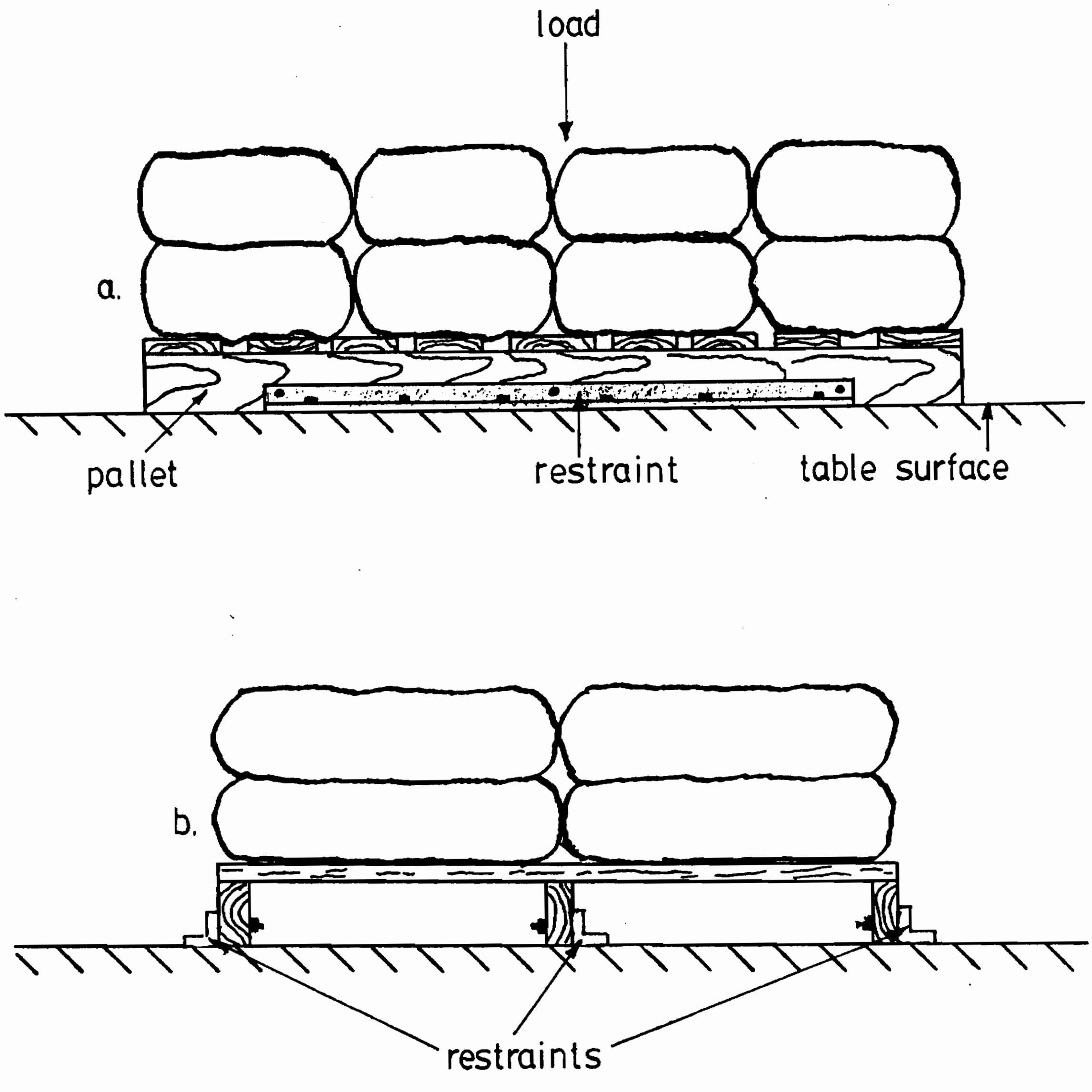

Figure 5.1. Restrained pallet configuration used to determine the resonant response of pallet decks: a. side view, and b. end view.

these components to determine transmissibilities across a pallet top deck.

The loaded pallet's resonant frequency was 16Hz during all 25 tests; thus, pallet resonance testing was found to be replicable with little or no change in resonant response. Transmissibility (T) at resonance, the ratio of output to input peak acceleration, was different at each accelerometer placement ranging from less than 1 at a center stringer notch to greater than 2.0 at the lead deckboard center of span between stringers. Although the resonant frequency of the pallet did not change during the 25 tests, acceleration levels varied across the face of the pallet during resonance. This was caused by the top deckboards exhibiting more vertical displacement in the middle of the deckboard span than near stringers. The pallet responded to vibration as an interactive system of members. Pallet members transferred vibrational energy across the system so that the pallet responded as a unit and not as separate responses of individual members. The conclusion formed from this exercise was that deckboard response between stringers had the greatest transmissibility; therefore, the significant mode of vibration was found to be top deckboard flexure. There were no trends in changes of the pallet's resonant response.

For the restrained tests on the single faced pallet, the stringers were fastened with angle iron brackets bolted to the vibration table as shown previously in Figure 5.1. The pallet was loaded with the same bag load used in the unrestrained pallet test. The test accelerometer was placed beneath the central deckboard at center of span between stringers. The central deckboard exhibited resonance at 25Hz and a transmissibility of 1.52. This resonant frequency was significantly different from the 16Hz observed in the unrestrained pallet.

The accelerometer was then moved to beneath the leading edge deckboard at center of span between stringers, which exhibited a resonant frequency of 16Hz with a transmissibility of 1.31. The dissimilarity in resonant frequencies between the central deckboard and the leading edge deckboard was attributed to differences in load levels and deckboard stiffness caused by differences in deckboard width and restraint.

Restraining the pallet changed the boundary conditions. The restraining connections induced stresses in the pallet's deckboards and joints. This altered the deckboard response; thus, the use of restraints during testing of pallets is not recommended, yet they were used here to try to isolate the response of the top deck.

5.2.2 Transmissibility Across a Single Deckboard Span

For a restrained pallet, the change in transmissibilities across a single deckboard span was investigated. The test accelerometer was placed at center of span and at 76mm and 152mm increments to the left and right of this location on the leading edge deckboard as is shown in Figure 5.2. A final vibration test was performed at a location 38mm left of center toward the outside stringer. One vibration test was performed at each accelerometer placement. The change in transmissibility along the span is shown in Figure 5.3. Maximum acceleration of the pallet top deckboard occurred at 38mm left of center of span toward the outside stringer, which corresponded to the beam theory prediction of the point of maximum static deflection of a double span beam.

5.2.3 Pallet Mode Shape Analysis

The deformation of top deckboards is the primary mode of pallet vibration when a pallet is rigidly supported at its bottom deck. Mode shapes of beams are affected by joint stiffness, number of spans, and load configuration. Primary mode shapes correspond to the shapes of static deflection. Pallet deckboards exhibit both deck flexure and joint rotation, thus vibration of pallet decks would be both translational and rotational. Structural mode shapes are frequency dependent, with the primary mode occurring at

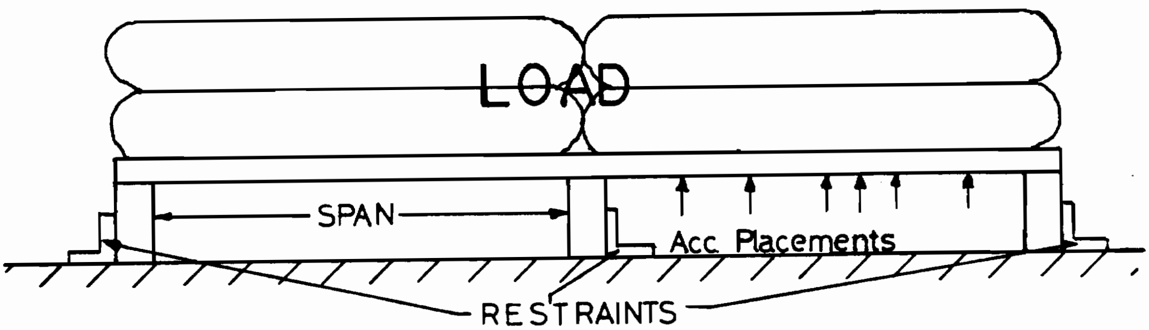

Figure 5.2. Accelerometer placements for vibration test of a restrained pallet to determine change in transmissibility across a single deckboard span.

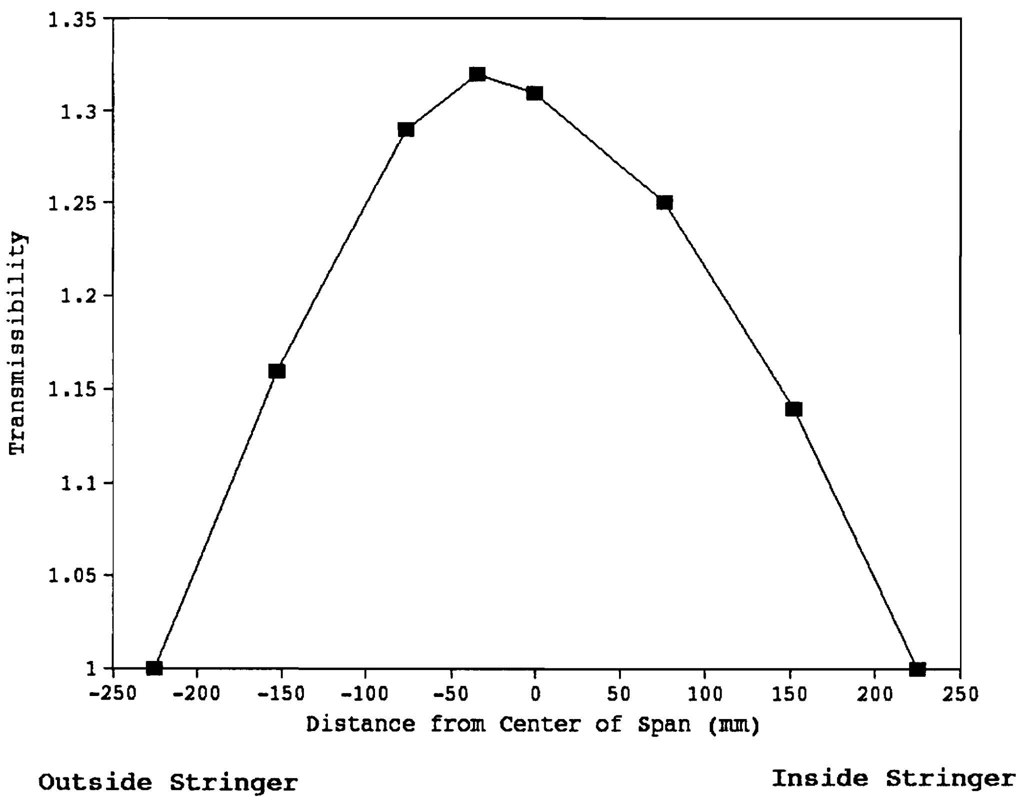

Figure 5.3. The change in transmissibility across a single deckboard span for a restrained pallet.

the lowest natural frequency. The possible primary mode shapes of two span continuous beams with pinned connections may be superimposed on a two dimensional view of a three stringer pallet as is shown in Figure 5.4a. If the center stringer joint is assumed to be fixed, the primary mode shape may be represented by the mode of a pinned-fixed-pinned double span beam as is shown in Figure 5.4b, which is thought to be the most probable mode shape for three stringer pallets.

Pallets have the freedom to lift away from their supporting bases during forced vibration, and tributary loading (the load supported by each stringer) may affect pallet modal response. If the pallet top deck is modeled as a two dimensional, two span, continuous beam with pinned connections, the load supported by the outside stringers is less than the load supported by the center stringer. For a uniformly distributed load, each outside stringer supports 20% of the load while the center stringer supports 60% of the load. It would appear that the outside stringers could lift away from the support more easily than the central stringer, causing the pallet to respond to forced vibration in a "flapping" manner. The primary mode shape of the pallet would then be like a free-fixed-free beam supported at its center.

The single faced pallet, left unrestrained, was used

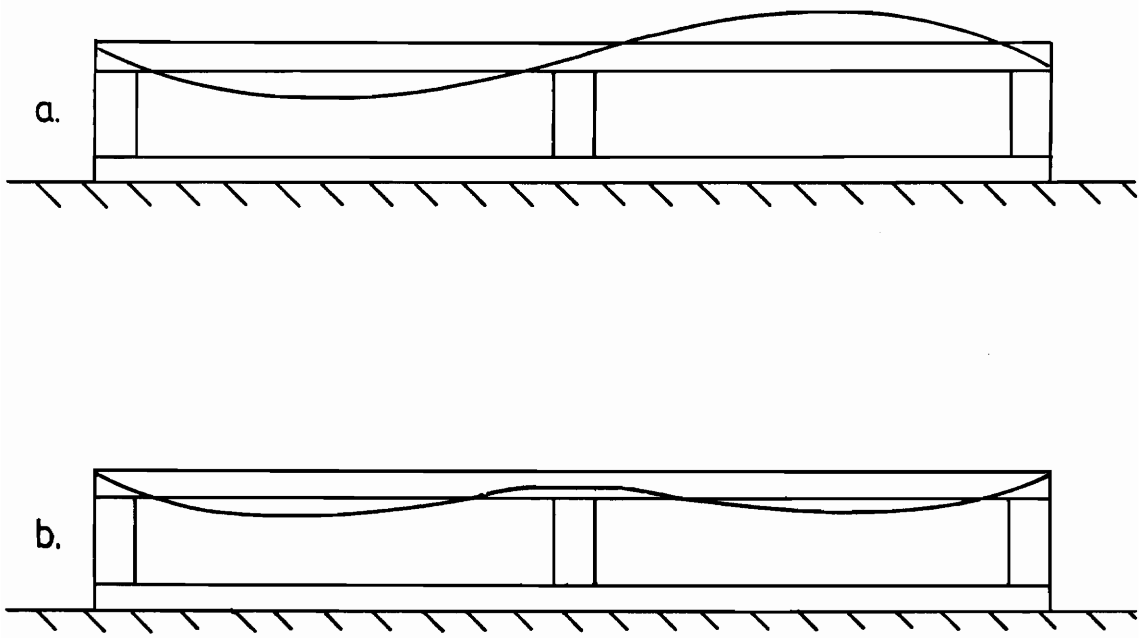

Figure 5.4. Possible pallet deck mode shapes caused by harmonic excitation: a. pinned-pinned-pinned, and b. pinned-fixed-pinned.

for final mode shape experimentation. This pallet was loaded with a 4464N crushed lime bag load. Accelerometers were placed beneath the central deckboard, center of span between stringers, and beneath the inside and outside stringer notches. The accelerometer placements were chosen to be indicators of total pallet motion caused by vibration. Ten vibration tests were performed. The responses of the three accelerometer locations were monitored, recorded and plotted versus time by the data acquisition equipment. Peak accelerations were used to indicate pallet component displacement. The locations with high levels of acceleration were assumed to be coincident with the locations with the most displacement.

Maximum acceleration was found at the accelerometer placed on the deckboard at the center of span between stringers. The next highest acceleration level was found at the outside stringer, with the least amount of acceleration occurring at the center stringer. Average transmissibilities for these accelerometer placements were 1.8, 1.6, and 1.2, respectively. The transmissibility averages had C.V.'s of less than 5%.

In another series of tests, horizontal accelerations were monitored by placing accelerometers on the three stringers perpendicular to the axis of vibration. The vibration test used was the same as that of the previous studies. Ten tests at each accelerometer placement were conducted. For all three accelerometer locations the resonant frequency was 18Hz and peak acceleration was less than 0.2g, with no significant change in acceleration nor resonant frequency during the tests. Transmissibility averages had C.V's of less than 5%. The horizontal acceleration was attributed to vibration table wobble and pallet rocking.

One may conclude that peak acceleration levels vary across a pallet's deck and that the response is symmetrical. Maximum acceleration occurred near the middle of span between stringers, with some magnified acceleration occurring at the outside stringers. The least amount of acceleration occurred over the central stringer. The principal mode of vibration is deckboard vertical displacement between stringers. It was difficult to determine true pallet mode shapes because the response could not be directly observed. With the use of strobe lighting and either still or high speed motion photography, one may be able to capture actual pallet mode shapes.

5.3. Load Effects: Boxes and Bags

Characteristics of static loads on pallets affect the vibrational response of pallets and unit-loads. These characteristics may include load form, load level, and the properties of the container and contents. The objective of this study was to select a test load for further research.

Two loads consisting of 222N crushed lime filled bags, and 222N corrugated fiberboard boxes filled with coated paper were used. These loads represented typical loads used in material handling. The single faced pallet from the previous tests was used and the vibration test was the same as that described in Section 5.2, a 2 to 100Hz sine-log sweep of frequency with a rate of change of one octave per minute.

The effects of the different loads were observed for top deckboard response to vibration. The test pallet was bolted to the table, test accelerometers were placed beneath the deckboards at the center of span between stringers on the leading edge deckboard, and at center of span between stringers beneath the central deckboard. A single layer of boxes or bags was placed on the pallet and five tests were performed. After the first five tests were completed, a second layer of boxes or bags was placed on the pallet and another series of five tests were executed at each accelerometer placement.

The single layer and double layer of boxes weighed 1726N and 2664N, respectively. The single layer and double layer of bags weighed 1354N and 2708N, respectively. The deckboards were affected differently by the two load types.

The box load had lower resonant frequencies and higher transmissibilities than the bag load, even though the second bag load was actually heavier than the second box load. This may be due to the multiple degree of freedom spring-mass system of the box load influencing deckboard response. The data for the pallet load tests is shown in Table 5.2. As expected, the pallet and pallet load reacted as a system.

Boxes were difficult to keep on the pallet during vibration testing, so box stacking height was not extended to more than two layers. To test a larger box load, load restraint would have been necessary. Bag loads did not have to be restrained, but the bag loads did stiffen during testing; therefore, a pretest must be performed where bags are used as static load simulators. A pretest stiffens the bag load and deckboard dynamic responses remain fairly constant thereafter. The bag load introduced fewer testing problems than the box load; hence, the bag load was used for further research.

5.4. Pallet Section Model Development and Verification

Pallet sections are an economical and practical means of testing pallet vibration relationships. The vibrational response of full size pallets is very complex and it was thought that by using a simple physical model, it would be easier to control experimental variation and interpret

Table 5.2. The restrained pallet deck resonant response corresponding to load and accelerometer location.

Load Type	Load (N)	Deckboard	f_r * (Hz)	T ♦
box	1726	central	20	1.75
box	1726	lead	16	1.97
box	2664	central	13	1.47
box	2664	lead	13	2.13
bag	1354	central	20	1.22
bag	1354	lead	22	1.75
bag	2708	central	18	1.24
bag	2708	lead	20	1.81

* f_r = resonant frequency
♦ T = transmissibility

factors influencing pallet response to forced vibration. The objective of the following testing was to develop a physical pallet section model to represent full size pallets.

The requirements of the model were: 1. The pallet sections would have to resemble full size three stringer pallets, 2. They would have a top and bottom deck, and three stringers, and 3. They must have an adequate surface area to support a load. The pallet section model developed is shown in Figure 5.5.

A test pallet was built to compare the responses of sections with those of a full size pallet. A diagram of the top face of this pallet is shown in Figure 5.6. This pallet was a 1220mm by 1016mm, two face, nonreversible, yellow-poplar, three stringer pallet, constructed with 22mm thick by 203mm wide and 1016mm long top and bottom deckboards and 38mm wide by 89mm high by 1219mm long stringers. The pallet was assembled so that three pallet sections could be cut from it. Two 146mm wide deckboards were used as filler deckboards on the top deck between the "pallet section" components. These deckboards were included so that the pallet could support a uniformly distributed load across the pallet's entire deck surface. The use of these deckboards did introduce testing problems because the actual load that

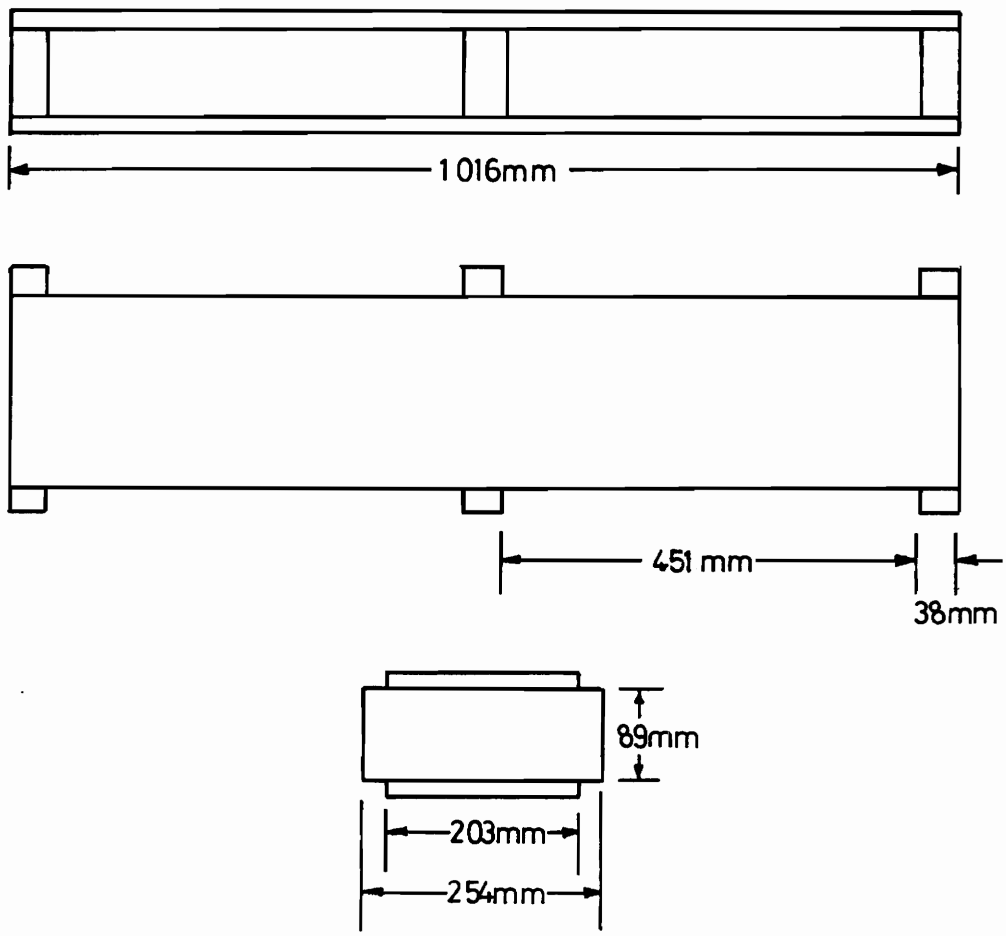

Figure 5.5. Schematic diagram of the pallet section model used for forced vibration testing of pallet decks.

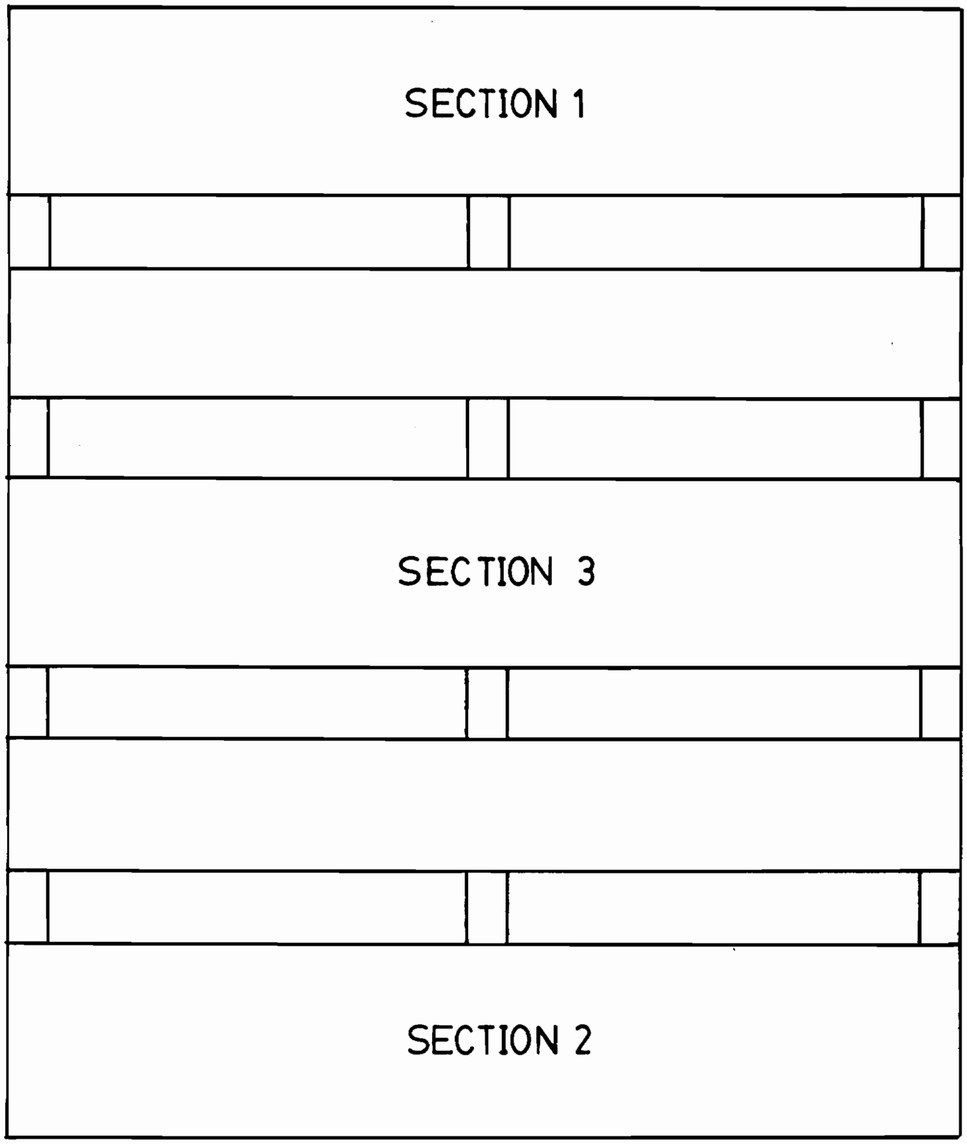

Figure 5.6. Schematic diagram of the top face of pallet used to contain pallet section models.

was supported by each section within the full size pallet could not be precisely determined; therefore, the loads supported by the sections were estimated by dividing the loads by the pallet's surface area.

The pallet was placed on the vibration table and loaded with four loads: 1697N, 3404N, 5111N, and 6847N. The load levels correspond to one, two, three, and four layers of bag loading. Three series of three vibration tests were performed, one series for each pallet section. The test accelerometer was placed at center of span between stringers on the two top leading edge deckboards (Sections 1 and 2) and the central top deckboard (Section 3). The vibration test was a single sine log sweep from 2 to 50Hz with a peak acceleration of 0.5g with a rate of change of one octave per minute. The test accelerometer and table control accelerometer peak acceleration responses were plotted versus frequency. The C.V's for the means of the transmissibilities and resonant frequencies for each test series were less than 5%.

Next, the pallet sections were cut from the pallet. The three sections were tested with an equivalent loading to that of the full size pallet, and retested. The section bag loads were 285N, 627N, 853N, and 1284N, respectively. A comparison of section response in the uncut pallet and as separate structures is contained in Table 5.3. The

Table 5.3 Comparison of average resonant responses of pallet sections in uncut pallet configuration and as independent structures.

Section # (Location)	Load [a] (uncut) (Pa)	Load (cut) (Pa)	f_r [b] (uncut) (Hz)	f_r (cut) (Hz)	T [c] (uncut)	T (cut)
LEDB1 (1)	1380	1425	38	32	2.04	2.09
LEDB1[d] (1)	2767	3043	28	19	2.17	2.17
LEDB1 (1)	4155	4141	15	16	2.32	2.32
LEDB1 (1)	5566	6233	13	15	2.25	2.07
LEDB2 (2)	1380	1425	33	33	2.08	2.29
LEDB2 (2)	2767	3053	25	16	2.39	2.17
LEDB2 (2)	4155	4141	17	16	2.23	2.08
LEDB2 (2)	5566	6233	16	15	2.23	2.05
CDB1[e] (3)	1380	1425	22	31	1.98	2.09
CDB1 (3)	2767	3043	20	18	2.05	1.86
CDB1 (3)	4155	4141	15	17	2.15	1.93
CDB1 (3)	5566	6233	14	16	2.23	1.89

a estimated loads for section deckboards in the uncut pallet configuration.
b f_r = resonant frequency
c T = Transmissibility
d LEDB = Leading edge deckboard
e CDB = Central deckboard

area-loads for the pallet sections were slightly heavier than those estimated for the full size pallet, which may have influenced the observed variations between the pallet sections. The rotational stiffness of the stringer sections may have also been altered by cutting the stringers to separate the pallet sections.

The pallet section responses in the whole pallet were compared to responses of the separate sections. There were differences between the uncut and cut responses. The boundary conditions of the pallet sections within the pallet and as separate structures were different; however, the trends in changes in deckboard response were similar. It was thought that the use of the pallet sections would be an indicator of trends in pallet response to forced vibration.

5.5. Summary and Conclusions

In the first study, the response of a restrained pallet was different from an unrestrained pallet. Since pallets are not usually restrained in real world applications, future studies should use unrestrained pallets or pallet sections. Pallet modal response was investigated; pallet top deckboards flex, and outside stringers lift.

Maximum accelerations occurred at 38mm left of center of deckboard span toward the outside stringers. Deckboard flexure was the primary mode of vibration and the response was symmetrical with respect to pallet width.

In the second study, load effects were discovered for bag and box loads. Box loads were stiffer than bag loads and were hard to keep in place during testing, requiring load restraint. Since bag loads did not shift, bag loads were used for further testing.

During the third study, a pallet section model was developed and verified to approximate full size pallet response. Pallet sections were tested for resonant response within the structure of a pallet and as independent structures cut from a full size pallet. The pallet sections acted similarly, whether they were within the full size pallet or as independent structures and had comparable trends in changes of resonant response due to load level.

From these preliminary studies, a pallet section and load model were developed. The primary assumptions for further research were that dynamic forces must be transmitted through a pallet to affect the contents of a unit-load, the response of the pallet top deck is a good indicator of the forces transmitted to unit-loads, and that there is a correlation between deckboard response and unit-load response.

CHAPTER 6

Effect of Connection Stiffness on the Resonant Response of Wood Pallet Decks

Abstract

The effect of connection stiffness on deckboard resonant response was found by testing pallet sections constructed with connectors resulting in theoretical extremes in joint stiffness. Connections assembled with finishing nails were assumed to simulate pinned joints, while glued connections were assumed to represent fixed joints. Pallet sections with three deck thicknesses were tested with five different static load levels. Comparing the glued with pinned joint sections the "fixed" epoxy joints exhibited 23% higher resonant frequencies and 20% lower transmissibilities than the "pinned" finishing nail joints. It is possible to alter the resonant response of pallets by changing the stiffness of the connections.

Key Words: pallets, connectors, vibration, resonant response

6.1. Introduction

Pallets are used for storage and distribution of products. The transport environment subjects unit-loads to forced vibration. As the interface between the rigors of handling equipment and the packaged product, pallet response to vibration should influence product damage.

Truck transport periodic forcing frequencies may range from 0 to 100 Hz with peak acceleration as high as 2g (Sharpe et al., 1974). Truck vibrations are primarily vertical with most forcing frequencies occurring below 20Hz (Ostrem and Godshall, 1979).

In Chapter 4, pallet designs were evaluated for their influence on unit-load resonant response. Pallets were found to decrease resonant frequencies and increase transmissibilities at resonance of unit-loads. A pallet section model was developed in Chapter 5 to simulate full size pallet vibrational response. The primary mode of pallet vibration was top deck flexure.

A popular pallet design used in the United States is a flush, nonreversible, 1219mm long by 1016mm wide, three stringer lumber deck pallet (McCurdy and Ewers, 1986). The factors thought to affect the dynamic response of three stringer lumber deck pallets are deckboard EI, span between stringers (L), and joint stiffness (White et al., 1986). Wood pallets are constructed with lumber and fasteners. Joint stiffness is controlled by the quality and quantity of fasteners used in pallet joints. Stiff stock nails, hardened nails, and pallet staples are commonly used pallet connectors. Quality pallet nails are threaded to improve withdrawal resistance.

The objective of the study reported here was to find if the stiffness of pallet joints affects the response of pallets to forced vibration.

6.2. Materials and Methods

Pallet sections, as described in Chapter 5, were made from two 1016mm long by 203mm wide, dry, clear, straight grained, yellow-poplar deckboards with thicknesses of 10mm, 16mm, or 22mm, and three 254mm long by 38mm wide by 89mm thick stringer members. The average moisture contents of the deckboards and stringers were 8% and 12%, respectively. The pallet deckboard free span was 451mm. Pallet deckboards were nondestructively tested for modulus of elasticity using the method described in Appendix C, and deckboard EI data is shown in Appendix A-3. A diagram of a pallet section is shown previously in Figure 5.5.

Eighteen pallet sections were constructed. Nine sections were assembled with two 4d finishing nails per joint. Nail holes were predrilled at 100% of fastener diameter into the deckboards to minimize clamping forces from the nails. The other nine sections were made with glued joints using epoxy adhesive to simulate rigid connections. Three replicate sections were assembled for each deckboard thickness/joint stiffness parameter. Neither joint is commonly used in commercial pallet construction but each was chosen to represent a possible extreme in connection stiffness.

The pallet sections were tested for resonant response under five different uniformly distributed crushed lime bag loads: 353N, 706N, 1058N, 1411N, and 1773N. The load levels were chosen to represent a load range that is typically supported by a pallet section of similar dimensions when part of a full size pallet. The vibration test setup is shown in Figure 6.1.

The vibration testing equipment employed was a servohydraulic vibration test system described in Appendix B, consisting of a vibration table, an actuator, a hydraulic power unit, closed loop controls, piezoelectric accelerometers, and a data acquisition/plotting system.

An accelerometer was fastened with hot melt adhesive in the vertical axis direction to a point beneath the spans of pallet top deckboards 190mm from outside stringers. From beam theory, this is the location where maximum static deflection is predicted, and was found in Chapter 5 to be the point of maximum acceleration on pallet top deckboards. Section response was also shown to be symmetrical about the central stringer segment. Accelerometer placement on the pallet sections is shown in Figure 6.2.

The vibration test was a single 2 to 50Hz sine log sweep with a 0.5g peak acceleration and a 1 octave per

Figure 6.1. Pallet section testing configuration used to determine resonant response of pallet top deckboards.

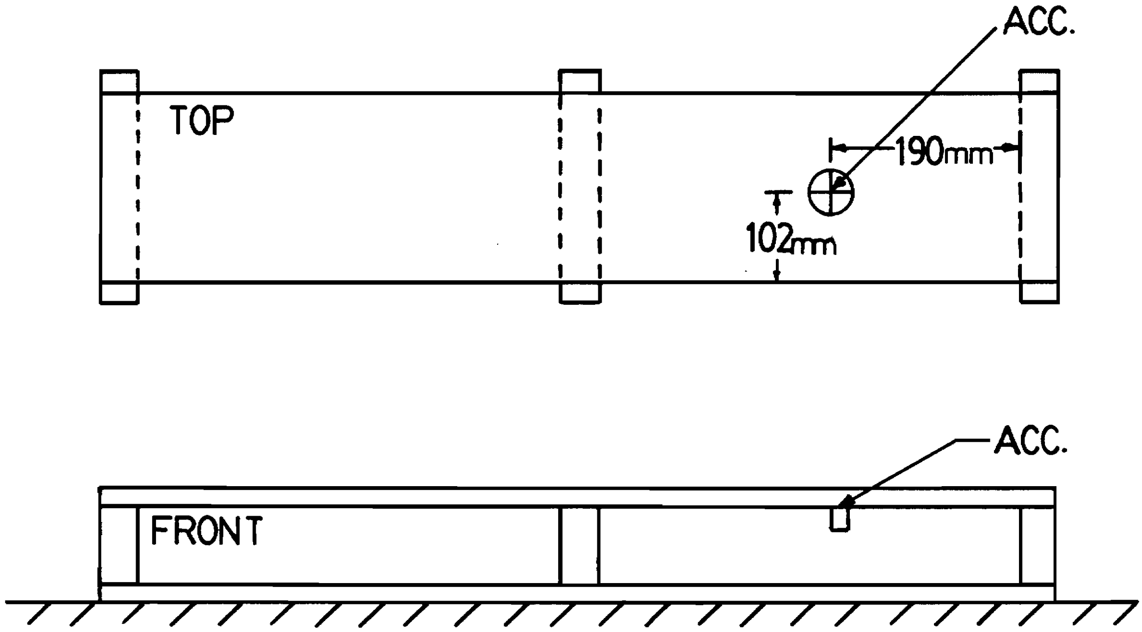

Figure 6.2. Accelerometer placement on pallet sections used to determine resonant frequencies and maximum transmissibility at resonance for pallet decks during vibration testing.

minute rate of change. This frequency band was chosen because preliminary studies found that most loaded pallet decks resonate at frequencies below 50Hz. Pallet sections that exhibited resonant frequencies above 50Hz were subjected to frequency sweeps from 2 to 100Hz.

The pallet sections were loaded to five different load levels and subjected to 4 tests per load level. A layer of bags, weighing 353N, was added after the completion of each test series. The first test, a pretest, was used to settle and stiffen the bag load because it was found that the bags would stiffen during vibration testing and that one test was sufficient to settle the load. A follow up vibration test at the first load level was conducted after testing all five load levels to find if vibration testing had structurally altered the pallet sections.

Resonant frequencies were obtained from a plot of peak accelerations versus frequency generated by the data acquisition system. Both the test accelerometer's response and the control accelerometer's response were monitored during testing. A representative plot is shown in Figure 6.3. Resonance occurred at maximum acceleration of the deckboard. Transmissibility (**T**) at resonance, the ratio of output acceleration to input acceleration, was calculated by dividing the pallet deckboards maximum acceleration response by the response of the control accelerometer at the

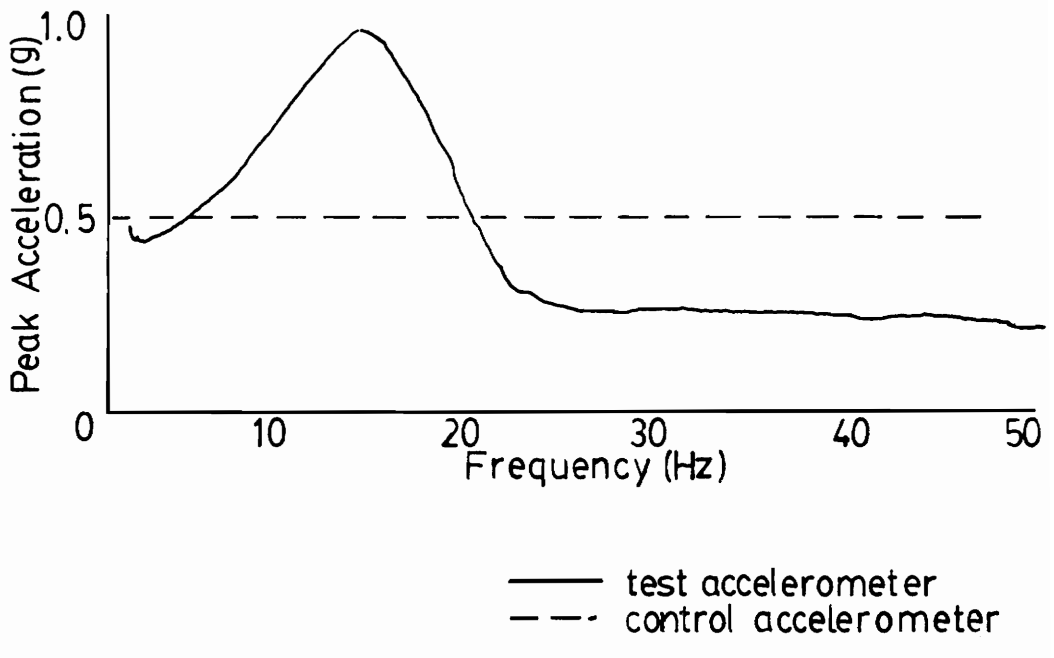

Figure 6.3. A representative test plot of a pallet section vibration resonance test.

resonant frequency. Transmissibility was thought to have been affected by system damping. Damping was not calculated due to the limitations of the testing equipment not allowing measurement of deckboard displacement.

The experimental design consisted of six groups of pallet sections with each group representing a different joint stiffness/deck thickness parameter. There were three replicates per group and each replicate was tested using five different loads and three significant vibration tests per load level. Pallet section load levels were converted to area-loads (uniformly distributed load level supported by an area of pallet deck surface), which corresponded to 1706Pa, 3427Pa, 5141Pa, 6855Pa, and 8617Pa, respectively for each bag load level.

6.3. Results

To determine whether the vibration testing had altered the pallet sections' resonant responses, the initial and final test data for the first load level of the pallet sections were averaged to form initial and final response means. Student's t tests were conducted on the resonant frequency and transmissibility means to determine statistical differences between the initial and final tests. The significance level was 5%. There were no significant differences between the initial and final responses for either resonant frequencies or transmissibilities. Thus, the pallet sections were found not to be significantly altered by vibration testing. Some finishing nails used to construct the "pinned" sections broke during testing. This was caused by nail fatigue from repetitive load cycling.

Average resonant frequencies (f_r) and transmissibilities (T) at each load level were calculated for each section group. Summary resonant frequency and transmissibility data are shown in Tables 6.1 and 6.2. Raw data is shown in Appendices A-4, A-5, A-6, and A-7. Coefficients of variation (C.V.'s) for resonant frequencies and transmissibilities within individual section tests were all less than 5%. There was significant variation across connector/deck thickness groups with C.V.'s as large as 49% for resonant frequencies and 10% for transmissibilities. Large variation was attributed to differences in pallet section structures like twisted and bowed deckboards and joints with dissimilar stiffnesses and possibly to differences in system damping. Also, as load level decreases the influence of variation in deck stiffness increases (this will be discussed later).

Plots of average resonant frequencies and transmissibilities versus area load level are shown in Figures 6.4 and 6.5. As one can see from Figure 6.4, resonant frequencies for the pallet sections decrease as

Table 6.1. Resonant Frequency (f_r) as a function of joint stiffness from the vibration testing of pallet sections with different thickness deckboards supporting different static loads.

Connector	Thickness (mm)	Load (Pa)	f_r ♣ (Hz)	C.V. (%)
pinned	10	1706	16	6
pinned	10	3427	14	7
pinned	10	5141	13	6
pinned	10	6855	11	0
pinned	10	8617	10	0
pinned	16	1706	28	25
pinned	16	3427	25	22
pinned	16	5141	18	3
pinned	16	6855	15	3
pinned	16	8617	14	0
pinned	22	1706	18	11
pinned	22	3427	17	5
pinned	22	5141	16	3
pinned	22	6855	16	3
pinned	22	8617	15	0
epoxy	10	1706	27	25
epoxy	10	3427	17	12
epoxy	10	5141	15	3
epoxy	10	6855	14	3
epoxy	10	8617	13	0
epoxy	10	1706	20	8
epoxy	16	3427	18	0
epoxy	16	5141	17	7
epoxy	16	6855	16	5
epoxy	16	8617	16	6
epoxy	22	1706	35	49
epoxy	22	3427	25	46
epoxy	22	5141	19	13
epoxy	22	6855	17	3
epoxy	22	8617	17	3

♣ f_r = resonant frequency (each resonant frequency value is the mean of three replicates)

Table 6.2. Transmissibility (T) as a function of joint stiffness from the vibration testing of pallet sections with different thickness deckboards supporting different static loads.

Connector	Thickness (mm)	Load (Pa)	T ♠	C.V. (%)
pinned	10	1706	2.78	5
pinned	10	3427	2.85	2
pinned	10	5141	2.73	5
pinned	10	6855	2.72	2
pinned	10	8617	2.59	4
pinned	10	1706	1.17	7
pinned	16	3427	1.84	3
pinned	16	5141	2.14	5
pinned	16	6855	2.16	2
pinned	16	8617	2.00	3
pinned	22	1706	1.83	5
pinned	22	3427	1.79	5
pinned	22	5141	1.76	3
pinned	22	6855	1.95	2
pinned	22	8617	1.86	2
epoxy	10	1706	2.02	10
epoxy	10	3427	2.19	10
epoxy	10	5141	2.28	5
epoxy	10	6855	2.42	5
epoxy	10	8617	2.37	2
epoxy	16	1706	1.58	6
epoxy	16	3427	1.73	3
epoxy	16	5141	1.70	4
epoxy	16	6855	1.80	6
epoxy	16	8617	1.75	6
epoxy	16	1706	1.38	10
epoxy	22	3427	1.49	7
epoxy	22	5141	1.52	10
epoxy	22	6855	1.61	9
epoxy	22	8617	1.56	4

♠ T= transmissibility (each transmissibility value is the mean of 3 replicates)

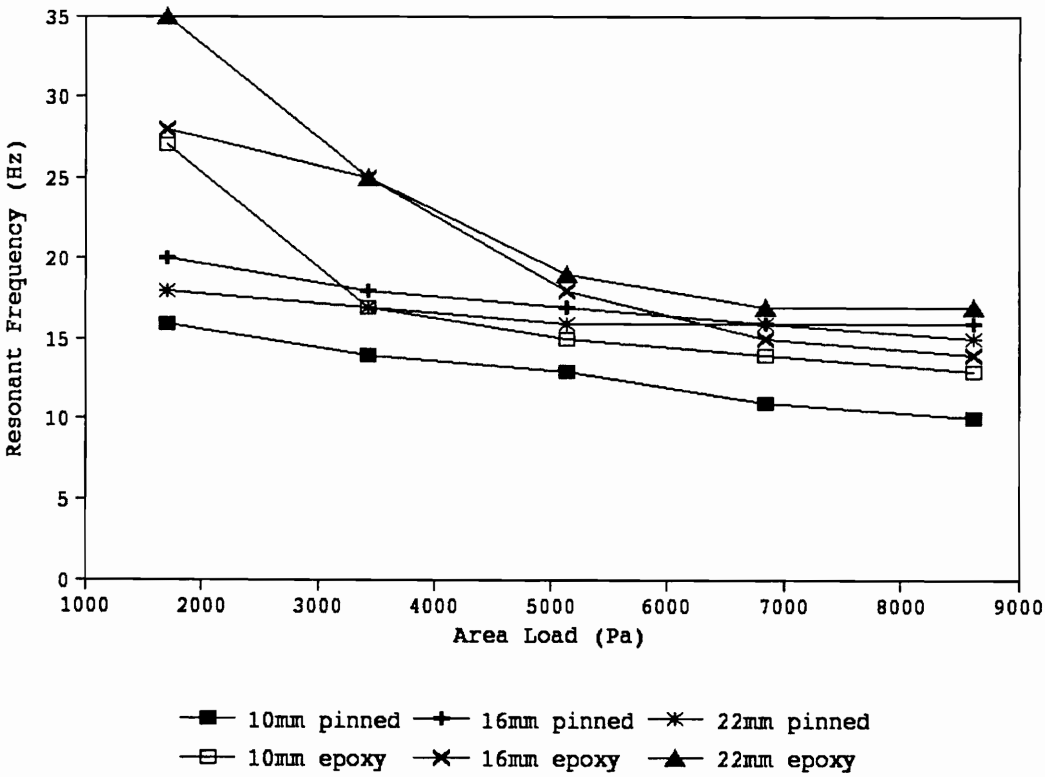

Figure 6.4. Average resonant frequency (f_r) values versus area load level for each joint stiffness-deckboard thickness group of pallet sections used to determine the influence of joint stiffness on pallet deck resonant response.

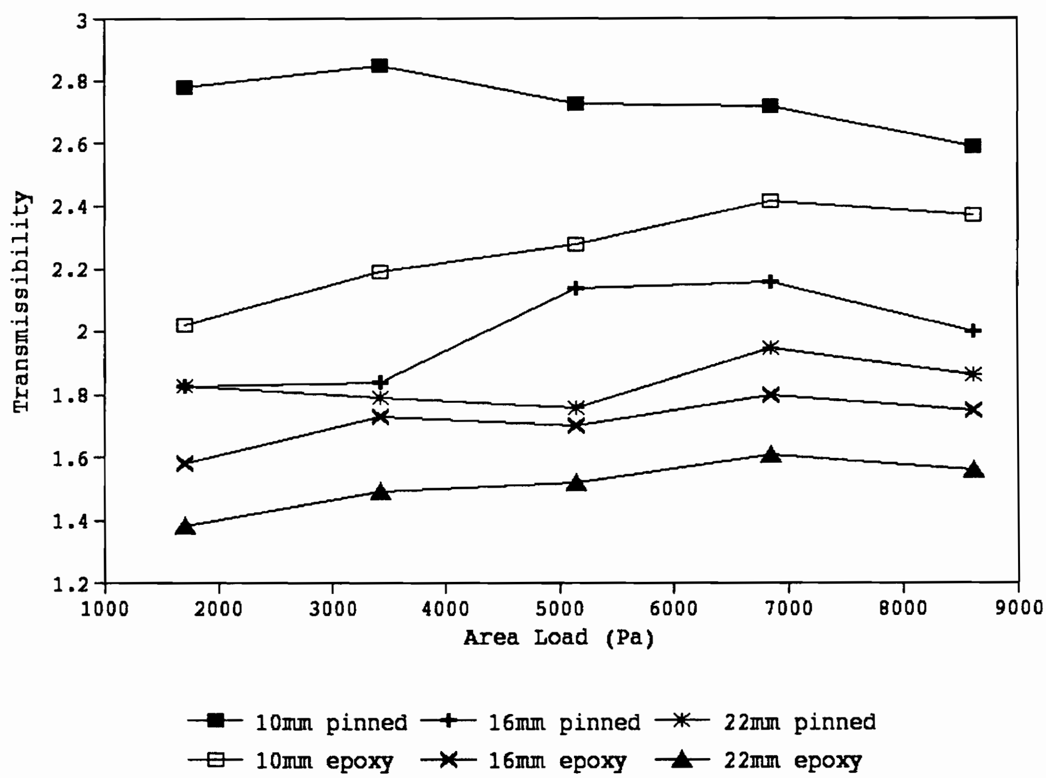

Figure 6.5. Average transmissibility (T) values versus area load level for each joint stiffness-deckboard thickness group of pallet sections used to determine the influence of joint stiffness on pallet deck resonant response.

static load level increases as would be predicted by the natural frequency equation for a single degree of freedom spring-mass system ($\sqrt{k/m}$), where natural frequencies would decrease as mass increases. The sections constructed with epoxy joints exhibited higher resonant frequencies than their pinned counterparts for all the deckboard thicknesses and static load levels, indicating that epoxy joints contributed to stiffening pallet decks. The 10mm deck thickness pinned sections exhibited the lowest resonant frequencies while the 22mm deck thickness epoxy glued sections exhibited the highest resonant frequencies. For transmissibilities, the 10mm deck thickness pinned joint sections had the highest transmissibilities and the 22mm deck thickness epoxy joint sections had the lowest transmissibilities. Thus, joint stiffness and deck thickness were found to influence the resonant frequencies and transmissibilities of the pallet sections.

Completely randomized, three-way analysis of variance (ANOVA) was performed on the means to identify significant variable affects on resonant frequencies and transmissibilities. Summaries of the results for the ANOVAs are shown in Tables 6.3, and 6.4. Since there was little variation across tests of individual sections, the means for the sections at each static load level were thought to be adequate for statistical inference.

Table 6.3. ANOVA Table of connection stiffness (A), deck thickness (B), and load effects (C) on the resonant frequency (f_r) of loaded pallet sections subjected to forced vibration.

Source	DF	Sum-Squares	Mean Square	F-Ratio	Prob>F
A	1	280.9	280.9	10.84	0.0017*
B	2	390.1556	195.0778	7.53	0.0012*
AB	2	106.8667	53.43333	2.06	0.1361
C	4	793.8889	198.4722	7.66	0.0000*
AC	4	208.1555	52.03889	2.01	0.1047
BC	8	107.5111	13.43889	0.52	0.8378
ABC	8	132.5778	16.57222	0.64	0.7414
ERROR	60	1554.667	25.91111		
TOTAL	89	3574.722			

* indicates an effect at a 5% level of significance.

Table 6.4. ANOVA Table of connection stiffness (A), Deck thickness (B), and load effects (C) on the transmissibility (T) of loaded pallet sections subjected to forced vibration.

Source	DF	Sum-Squares	Mean Square	F-Ratio	Prob>F
A	1	2.84089	2.84089	149.60	0.0000*
B	2	11.39051	5.695254	299.91	0.0000*
AB	2	0.1721667	8.608E-02	4.53	0.0147*
C	4	0.4664289	0.1166072	6.14	0.0003*
AC	4	9.668E-02	2.417E-02	1.27	0.2907
BC	8	8.439E-02	1.054E-02	0.56	0.8097
ABC	8	0.3257778	4.072E-02	2.14	0.0451*
ERROR	60	1.1394	0.01899		
TOTAL	89	16.51624			

* indicates an effect at a 5% level of significance.

At a 5% level of significance, the ANOVA showed that there were connection stiffness, deckboard thickness, and load effects on resonant frequencies. For transmissibilities, there were significant deckboard thickness, and connector stiffness effects.

Duncan's multiple range tests were conducted for connector, deckboard thickness, and load level effects on resonant frequencies and transmissibilities. The results of these analyses are shown in Tables 6.5 and 6.6. There were significant differences in the means of resonant frequencies between the joint parameters, but the distinction was not as clear for deck thickness and load levels. For transmissibilities, there were significant joint effects and deckboard thickness effects but the only significant load effect occurred at the light load levels.

There was a direct relationship between joint stiffness and resonant frequency of the top deck. Within deckboard thickness groups, sections constructed with epoxy joints had resonant frequencies as much as 69% higher than their pinned counterparts. There was an inverse relationship between joint stiffness and transmissibility. Within deckboard thickness groups, epoxy joint sections exhibited on average 16% less transmissibility than the pinned joint sections, with a maximum transmissibility reduction of 27% less than the pinned joint sections.

Table 6.5. Duncan's multiple range analysis of connection stiffness, deck thickness, and static load level effects on the resonant frequencies of pallet sections subjected to forced vibration .

Connection	Mean f_r (Hz)	Pinned	Epoxy
Pinned	15.29	.	S
Epoxy	18.82	S	.

Deck Thickness	Mean f_r (Hz)	10mm	16mm	22mm
10mm	14.50	.	.	S
16mm	17.07	.	.	.
22mm	19.60	S	.	.

Load Level (Pa)	Mean f_r (Hz)	8617	6855	5141	3427	1706
8617	14.00	.	.	.	S	S
6855	14.61	.	.	.	.	S
5141	16.33	.	.	.	.	S
3427	18.06	S	.	.	.	S
1706	22.28	S	S	S	S	.

S denotes a statistical difference at a 5% level

Table 6.6. Duncan's multiple range analysis of connection stiffness, deck thickness, and static load level effects on the transmissibilities of pallet sections subjected to forced vibration.

Connection	Mean T	epoxy	pinned		
epoxy	1.82	.	S		
pinned	2.18	S	.		
Deck Thickness	**Mean T**	**22mm**	**16mm**	**10mm**	
22mm	1.67	.	S	S	
16mm	1.85	S	.	S	
10mm	2.50	S	S	.	
Load Level (Pa)	**Mean T**	**1706**	**3427**	**8617**	**5141** / **6855**
1706	1.89	.	S	S	S / S
3427	1.98	S	.	.	. / S
8617	2.02	S	.	.	. / .
5141	2.02	S	.	.	. / .
6855	2.11	S	S	.	. / .

S denotes a statistical difference at a 5% level

As the static load level increased, resonant frequencies decreased for all connector/deckboard thickness parameters. Comparing the two deck stiffness extremes, the 22mm deck epoxy pallet sections' resonant frequencies were on average 35% higher than the 10mm deck pinned pallet sections. For the deck stiffness extremes, the transmissibilities of the 22mm deck epoxy sections were 45% less than the 10mm deck pinned sections.

Variances for resonant frequencies and transmissibilities were greater for the lighter load levels and thicker deckboards. From theory, as load level decreases the effect of the variance of EI becomes more pronounced. Because of this, resonant frequencies of the pallet decks were more sensitive to light load levels. This is illustrated by the SDOF spring-mass system natural frequency equation ($\sqrt{k/m}$), where EI would be indicative of spring stiffness (k). From this equation, one can see that the influence of spring stiffness variation of the deckboards would be more apparent at the light load levels.

Pallet section anomalies could have also affected their resonant responses at the light load levels. If a pallet section had a bottom deck with bow or twist, or if stringer height was not same for the section's three stringer members, modal response would not be exclusively top deck flexure.

6.4. Conclusions

Joint stiffness influences pallet deckboard resonant response. Epoxy joints produced pallet sections with higher resonant frequencies and lower transmissibilities for a given load level than pinned sections. A similar effect was observed for stiff deckboards, with the thicker deckboards having higher resonant frequencies and lower transmissibilities than thinner deckboards. As load level increased, resonant frequencies decreased for all pallet sections groups. For the pinned sections, some finishing nails broke from fatigue, thus pallet joints constructed with slender fasteners like pallet staples may be damaged by in-transit vibration.

From the results of this study, one can infer that by using pallet connectors to increase joint stiffness, resonant frequencies of pallets can be increased and transmissibilities can be decreased. Since neither of the experimental joints is typically used in commercial pallet construction, further research using actual pallet fasteners is necessary to quantify pallet joint stiffness effects for more commonly used connectors.

CHAPTER 7

Effect of Deckboard EI on the Resonant Response of Wood Pallet Decks

Abstract

EI is an indicator of deckboard stiffness. The effect of deckboard EI on pallet resonant response was observed by testing pallet sections constructed with deckboards with three levels of EI. Deck thicknesses of 10mm, 16mm, and 22mm were chosen to represent a range of deckboard EI's present in commercial wood pallet construction. Pallet sections were tested for their resonant response supporting five different static load levels ranging from 1706Pa to 8617Pa. Resonant frequency is directly related to pallet deckboard EI, while transmissibility is inversely related to deckboard EI. Comparing deck EI groups, the sections with the highest EI's exhibited 71% higher resonant frequencies and 48% lower transmissibilities at resonance than the sections with the lowest EI's. Resonant frequency of pallet decks is inversely related to static load level. Pallets with deckboards with high EI values will better protect most packaged products.

Key Words: pallet, EI, deckboard, resonant response,

7.1. Introduction

The distribution environment is complex. Unit-loads may be transported by trucks, rail cars, ships, and aircraft. Vibrations within the transport environment contribute to product damage. Periodic forcing frequencies may range from 0 to 100Hz, with most truck vibration occurring below 20Hz (Ostrem and Godshall, 1979). Products may experience peak accelerations from truck vibrations as high as 2g (Sharpe et al., 1974). Examples of products that are susceptible to vibration damage are fresh produce (Peleg and Hinga, 1986), electronics equipment, and medicines (Dority, 1989).

Pallets are the principal interface between the handling environment and the packaged product, where the pallet functions to protect products. Vibrations are transmitted through the pallet; thus, the dynamic response of pallets should influence product damage. The components of three stringer wood pallets are deckboards, stringers, and fasteners.

The structure of wood pallets seems simple but their components are affected by load sharing and nonlinear joint behavior (Loferski and McLain, 1987). The factors thought to influence wood pallet dynamic response are deckboard **EI**, span between stringers (**L**), and joint stiffness (White et al., 1986). EI is an indicator of deck stiffness. Deckboard span between stringers (L) is a major factor contributing to deck stiffness. Deckboard span was a constant and not an experimental variable in this study. A typical three stringer pallet design is shown in Figure 1.2. In Chapter 5, pallet top deckboards were found to be the members that were most influenced by vibration because they form continuous beams over one or more spans supported by deck spacers such as stringers that are free to move in response to vertical vibration. In Chapter 6, the stiffness of pallet joints was found to influence deck response to forced vibration.

Since the goal of vibration protective packaging is to prevent product resonance, all the components of the unit-load must be collectively designed to prevent products from experiencing resonant conditions and to lower transmissibilities to products. The role of pallets as spring-mass systems resting beneath unit-loads has not been seriously studied.

The objective of the study reported here was to determine the effect of deckboard EI and static load level on the resonant response of pallet decks.

7.2. Materials and Methods

The pallet section model used in this study was developed in Chapter 5. Thirty pallet sections were constructed from dry, defect free, yellow-poplar

(Liriodendron tulipifera) lumber. Yellow-poplar is commonly used in commercial pallet construction. The deckboards and stringers had average moisture contents of 8% and 12%, respectively. The pallet deckboards were nondestructively tested for modulus of elasticity using the static bending method described in Appendix C. Modulus of elasticity data for the top deckboards used in the pallet sections is shown in Appendix A-8.

The pallet section design is shown schematically in Figure 5.5. Section joints were made with helically threaded hardened steel pallet nails. The fastener specifications are shown previously in Table 5.1. Four nails were used per joint using a staggered nailing pattern. To prevent deckboard splitting caused by driving nails, nail holes were predrilled at 70% of nail shank diameter. Ten pallet sections were constructed for each deckboard thickness tested.

The pallet sections were tested for their resonant response under five different uniformly distributed bag loads: 353N, 706N, 1059N, 1412N, and 1776N. The static load levels were chosen to represent a range of loads that would be typically supported by pallet sections of similar dimensions when part of full size pallets.

The vibration testing equipment employed, described in Appendix B, was a servohydraulic vibration test system, consisting of a vibration table, an actuator, a hydraulic power unit, closed loop controls, piezoelectric accelerometers, and a data acquisition/plotting system. During a vibration test, the data acquisition system monitored the frequency generator, the table's control accelerometer, and one test specimen accelerometer. The accelerometers were calibrated to an accuracy of 3% within the frequency band and accelerations used during testing.

Accelerometers were fastened with hot melt adhesive in the vertical axis direction to a location beneath the spans of the top deckboards at a point 190mm from outside stringers. In Chapter 5, this was found to be the location of maximum acceleration for a double span pallet deckboard for a 1219mm by 1016mm three stringer pallet with 38mm wide stringers. Accelerometer placement on the pallet sections is shown previously in Figure 6.2. The vibration test was a single 2 to 50Hz sine log sweep with a rate of frequency change of 1 octave per minute. This frequency band was chosen because most loaded pallets were found in Chapters 4 and 5 to have resonant frequencies of less than 50Hz. Pallet sections that resonated at frequencies above 50Hz were subjected to vibration sweeps from 2 to 100Hz.

The pallet sections were loaded to five different static load levels and subjected to four tests per load level. The first test, a pretest, was used to settle the bag load because it was found during preliminary studies that the bag load would stiffen during this initial test sweep. The data from the first test per series was not used for calculations or statistical inference. A follow up test at the first load level was conducted after testing all five load levels to determine whether testing had structurally altered the pallet sections.

Resonant responses were obtained from a plot of peak accelerations. A representative plot is shown previously in Figure 6.3. Resonance ($\mathbf{f_r}$) occurred at the frequency of maximum acceleration of the pallet section top deck. Transmissibility (**T**) at resonance, the ratio of output to input peak accelerations was calculated by dividing pallet deckboards' maximum acceleration responses by the response of the control accelerometer at the resonant frequency ($\mathbf{f_r}$).

Experimental design consisted of sections constructed with three deckboard EI levels, and ten replicate pallet sections were constructed for each EI level. Each section was tested supporting five different static load levels with three significant resonance tests per load level.

The EI of the top deckboard for each section was calculated. Load levels were converted to deckboard surface area-loads of 1706Pa, 3427Pa, 5141Pa, 6855Pa, and 8617Pa, respectively.

7.3. Results

If for a first approximation, one can assume that loaded top pallet deckboards act as single degree of freedom spring-mass systems, as load increases, resonant frequencies decrease and as spring stiffness increases resonant frequencies increase as is shown by the natural frequency equation ($\sqrt{k/m}$). Transmissibility at resonance is inversely related to spring stiffness with stiffer springs causing lower transmissibility. Transmissibility is also affected by system damping but the equipment used in this study could not be used to determine damping because deckboard displacement could not be measured.

In reality, unit-loads form multiple degree of freedom spring-mass systems. However, for this study, the load was assumed to act as a single mass with no spring components. To investigate the effect of EI on the resonant response, the span (L) and joint stiffness were fixed by section design, but EI was varied.

To determine if vibration had altered the responses of the pallet sections. The sections were tested at the first load level, and then after all vibration tests were completed, they were tested again at the original load level. The test after the pretest was considered the initial test, and the last test was considered the final test. Student's t tests with a 5% significance level indicated that there were no significant differences between the initial and final test means at the first load level, thus it was concluded that vibration testing did not significantly change section resonant response. Comparing the initial and final responses of individual sections, there were some variations in resonant frequencies and transmissibilities but the changes exhibited no specific trends.

The summary data for average deckboard EI's for resonant frequencies and transmissibilities is shown in Tables 7.1 and 7.2. Raw data for resonant frequencies and transmissibilities of the pallet sections is shown in Appendices A-9 through A-14. The variation about the resonant frequency and transmissibility means for the three test cycles at each load level for each section were less than 15%, while the variations for most tests were less than 5%.

Completely randomized analysis of variance (ANOVA) was performed on the data to test the effects of deckboard EI and load level on resonant frequencies and transmissibilities. At a 5% level of significance, deckboard EI and load effects on resonant frequencies were evident. Transmissibilities were affected by the deckboard EI but not load level. There were some notable variations in transmissibilities at the lightest load level in both Chapter 6 and in this study. The variation may have been

Table 7.1. Summary data of average resonant frequencies (f_r) (Hz) of pallet sections as a function of deckboard thickness and static load level.

Load (Pa)	10mm Deck (Hz)	C.V.♦ (%)	16mm Deck (Hz)	C.V.	22mm Deck (Hz)	C.V. (%)
(1) 1706	26	10	45	15	52	15
(2) 3427	16	5	21	6	33	26
(3) 5141	14	6	20	3	23	13
(4) 6855	13	4	17	3	21	13
(5) 8617	12	4	16	3	17	6

♦ C.V. = coefficient of variation

Table 7.2. Summary data of average transmissibilities (T) for the pallet sections used in vibration testing as a function of deck thickness and static load level.

Load (Pa)	10mm Deck	C.V.♦ (%)	16mm Deck	C.V. (%)	22mm Deck	C.V. (%)
(1) 1706	2.09	7	2.13	15	1.84	11
(2) 3427	2.33	4	1.75	6	1.46	5
(3) 5141	2.39	3	1.80	3	1.50	8
(4) 6855	2.46	3	1.93	3	1.54	5
(5) 8617	2.38	6	1.87	3	1.50	5

♦ C.V. = coefficient of variation

caused by the pallet sections losing contact with the table's surface, causing the pallet sections to "bounce", which amplified transmissibility. This load effect on tranmissibilities may have been caused by pallet sections reacting inconsistently to forced vibration at the first load level as is portrayed in Figure 7.6. Because of variation in the data, Duncan's multiple range tests were performed on the data to determine differences between mean resonant frequencies and transmissibilities for each deckboard EI level at each static load level. The results of these tests, as shown in Tables 7.3 and 7.4., indicate that there are strong deckboard EI and load level effects on the resonant response of the pallet sections.

Deckboard EI values were averaged for each deck thickness group. The average EI for the 10mm, 16mm, and 22mm thick deckboards were 229, 897, and 2211N-m^2, respectively. To show the relationship between deckboard EI and resonant response, the resonant frequency data was plotted versus the square root of EI for each load level as is shown in Figures 7.1 through 7.5. Transmissibility was plotted versus 1/EI as is shown in Figures 7.6 through 7.10. Simple linear regressions were performed for the three EI levels at each of the five load levels for resonant frequencies and transmissibilities as is shown in Figures 7.1 through 7.10.

Table 7.3. Ranked comparison of mean resonant frequencies (f_r) as a function of deckboard EI from vibration tests of pallet sections using Duncan's multiple range test .

EI ($N\text{-}m^2$) (Average Values)	Mean f_r (Hz)	EI ($N\text{-}m^2$) 229	897	2211
229	16.18	.	S	S
897	23.93	S	.	S
2211	28.71	S	S	.

Load Level (Pa)	Mean f_r (Hz)	Load Level (Pa) 8617	6855	5141	3427	1706
8617	14.83	.	.	S	S	S
6855	16.53	.	.	S	S	S
5141	19.05	S	S	.	S	S
3427	23.35	S	S	S	.	S
1706	40.94	S	S	S	S	.

S denotes a difference between means at the 5% level of significance

Table 7.4. Ranked comparisons of mean transmissibilities (T) as a function of deckboard EI from vibration tests of pallet sections using Duncan's multiple range test.

EI ($N\text{-}m^2$) (Average Values)	Mean T	EI ($N\text{-}m^2$) 229	897	2211
229	1.57	.	S	S
897	1.89	S	.	S
2211	2.33	S	S	.

Load Level (Pa)	Mean T	Load Level (Pa) 5141	3427	8617	1706	6855
5141	1.91	.	.	.	.	.
3427	1.92	.	.	.	.	.
8617	1.92	.	.	.	.	.
1706	1.94	.	.	.	.	.
6855	1.95	.	.	.	.	.

S denotes a difference between the mean transmissibilities at a 5% level of significance.

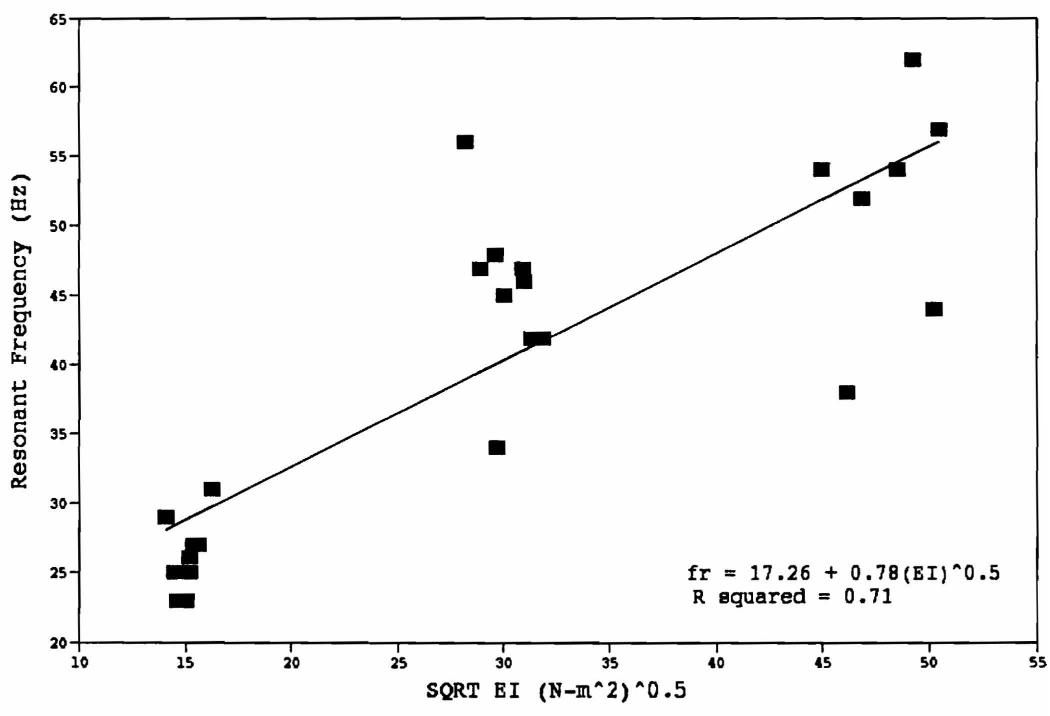

Figure 7.1. Resonant frequency (f_r) as a function of deckboard EI for pallet sections under a 1706Pa static load subjected to forced vibration.

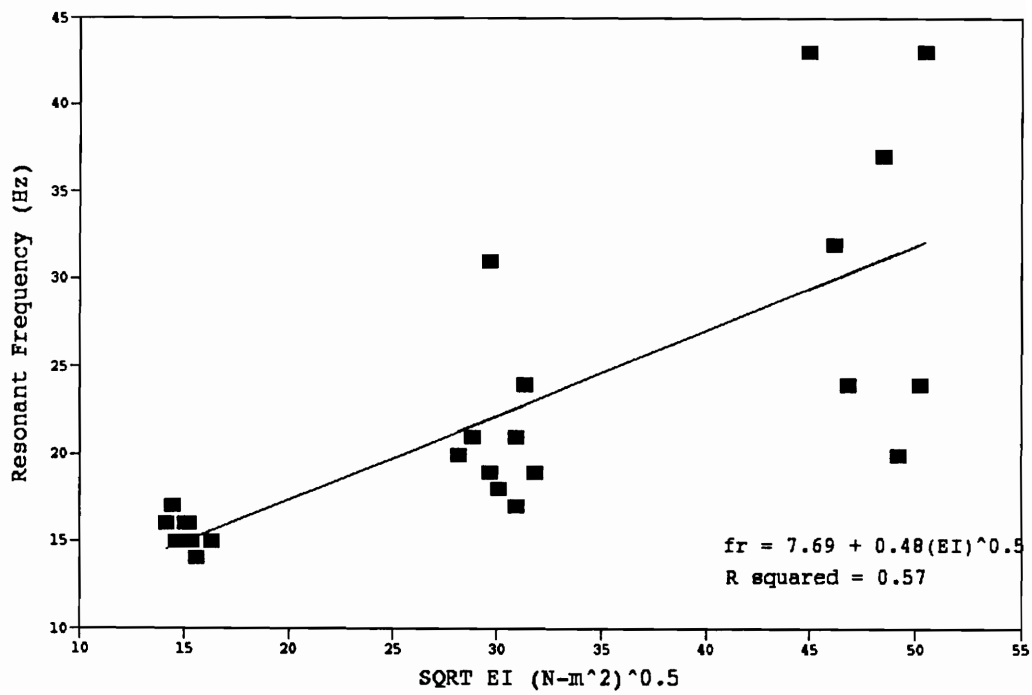

Figure 7.2. Resonant frequency (f_r) as a function of deckboard EI for pallet sections under a 3427Pa static load subjected to forced vibration.

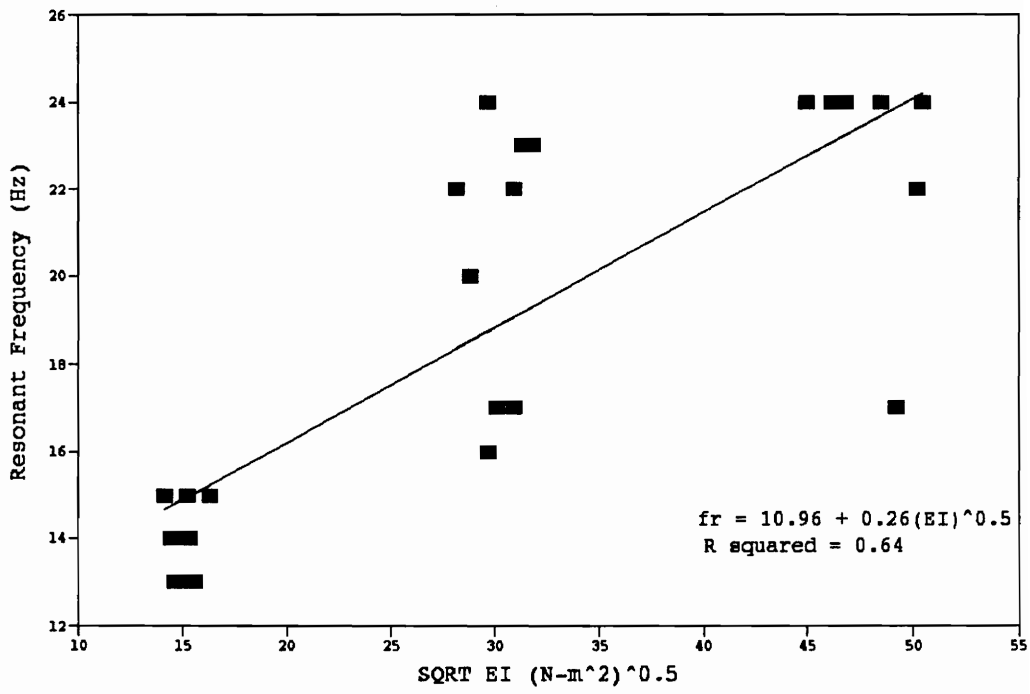

Figure 7.3. Resonant frequency (f_r) as a function of deckboard EI for pallet sections under a 5141Pa static load subjected to forced vibration.

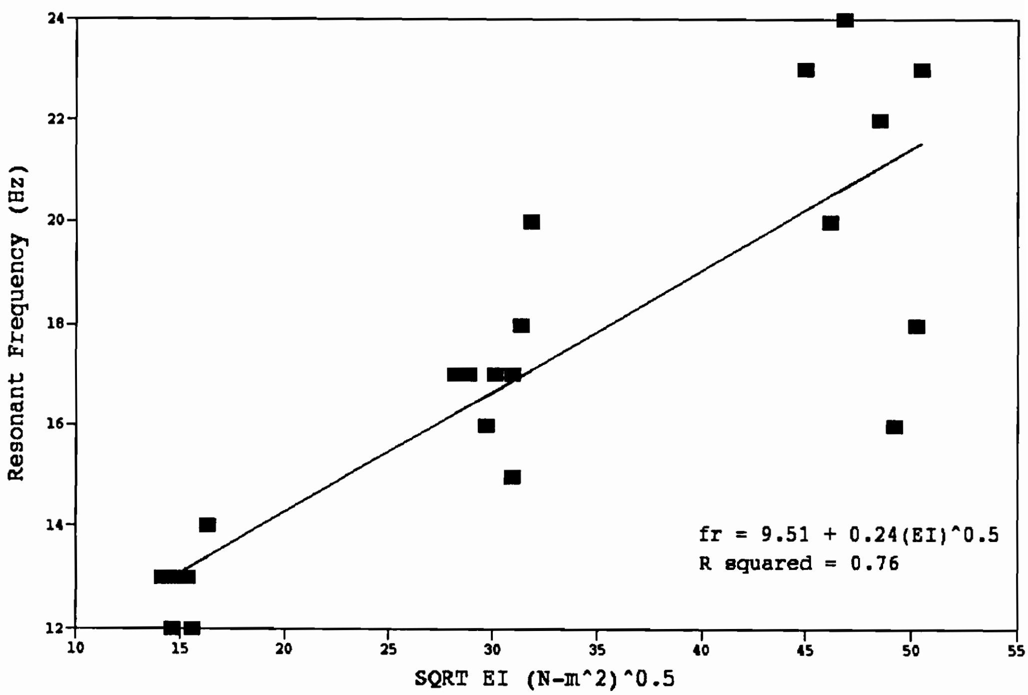

Figure 7.4. Resonant frequency (f_r) as a function of deckboard EI for pallet sections under a 6855Pa static load subjected to forced vibration.

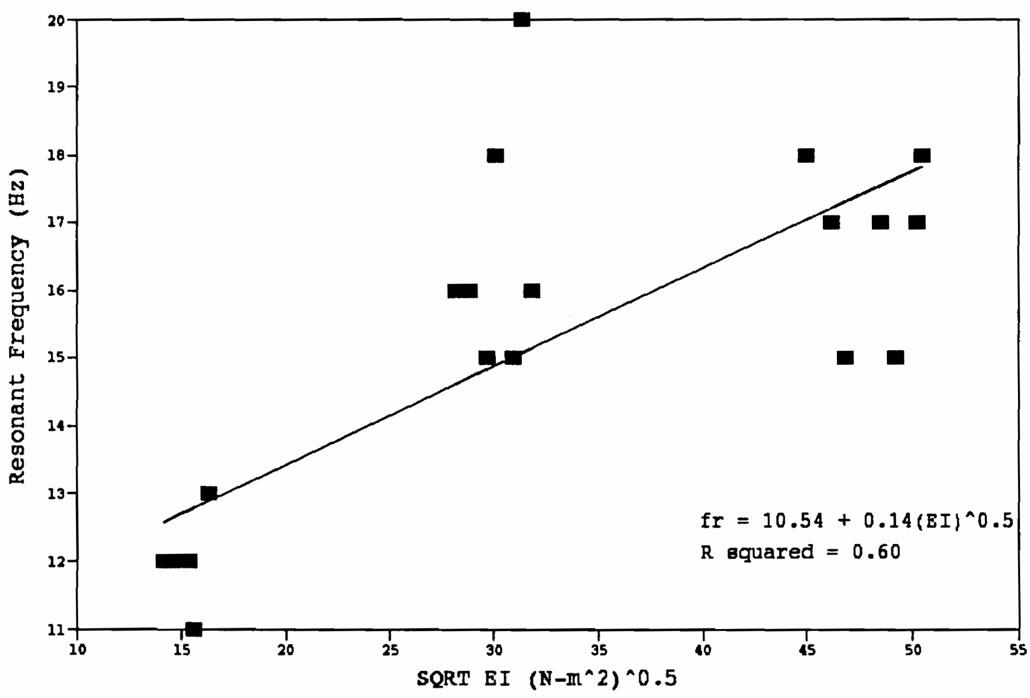

Figure 7.5. Resonant frequency (f_r) as a function of deckboard EI for pallet sections under a 8617Pa static load subjected to forced vibration.

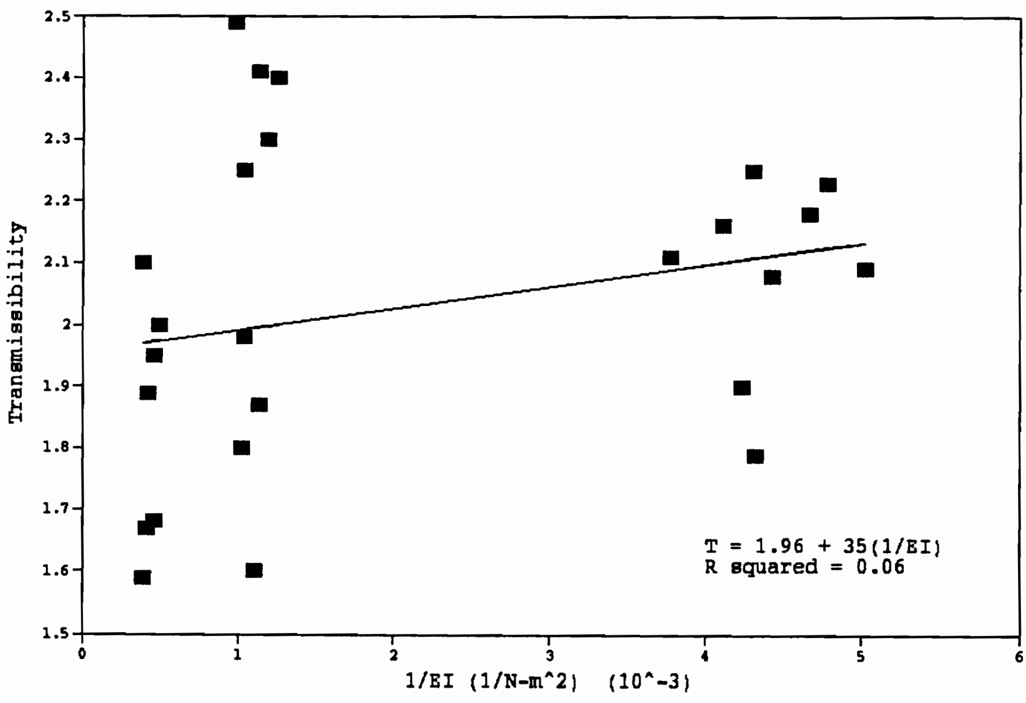

Figure 7.6. Transmissibility (T) as a function of deckboard EI for pallet sections under a 1706Pa static load subjected to forced vibration.

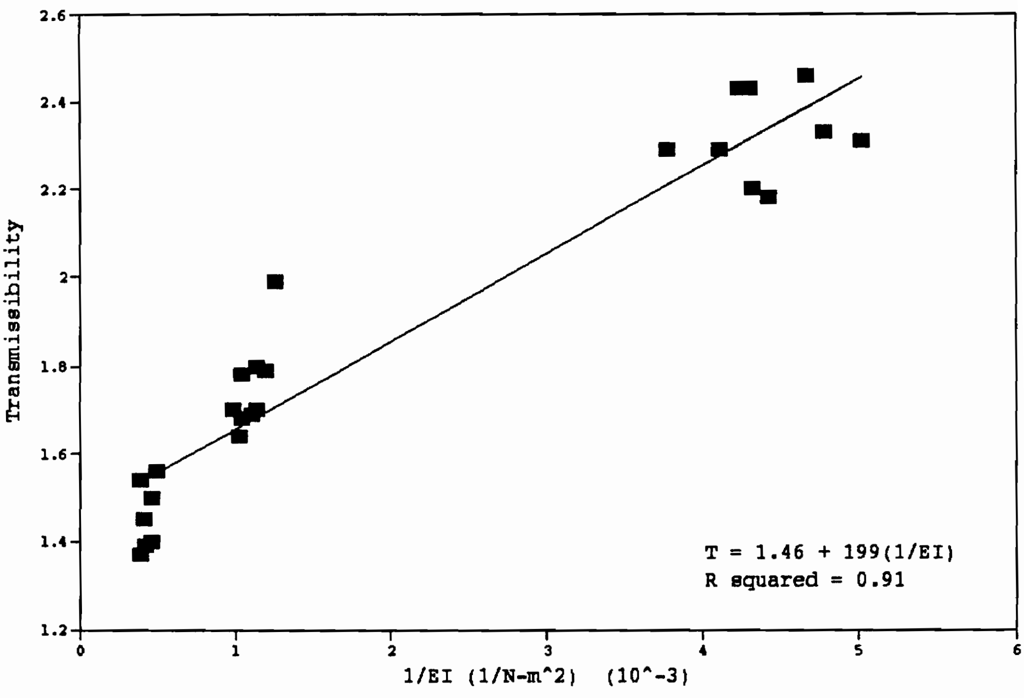

Figure 7.7. Transmissibility (T) as a function of deckboard EI for pallet sections under a 3247Pa static load subjected to forced vibration.

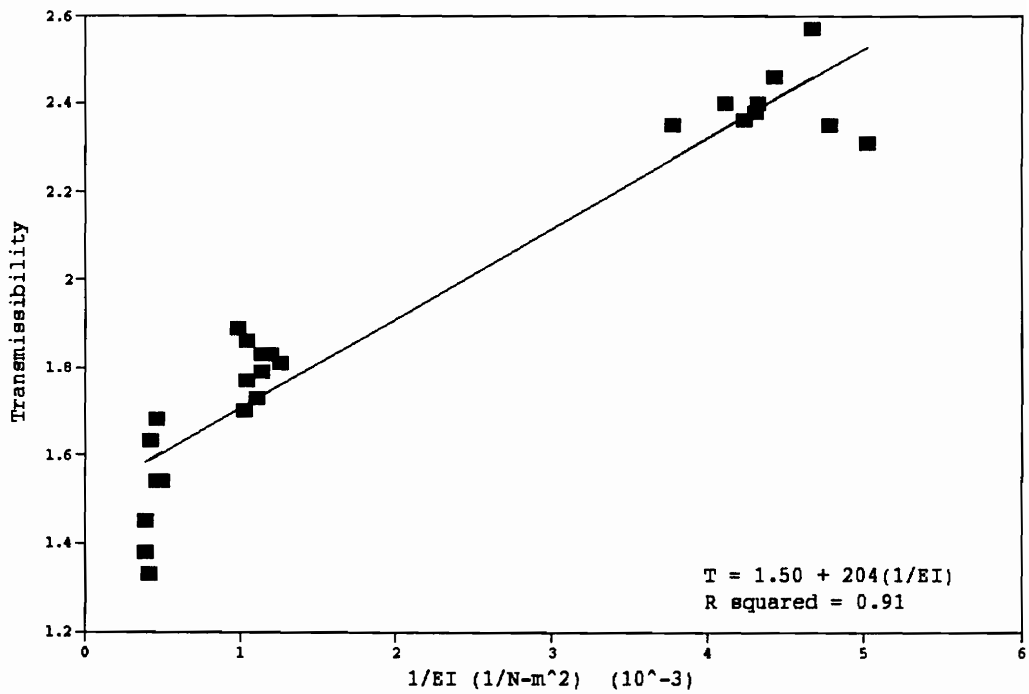

Figure 7.8. Transmissibility (T) as a function of deckboard EI for pallet sections under a 5141Pa static load subjected to forced vibration.

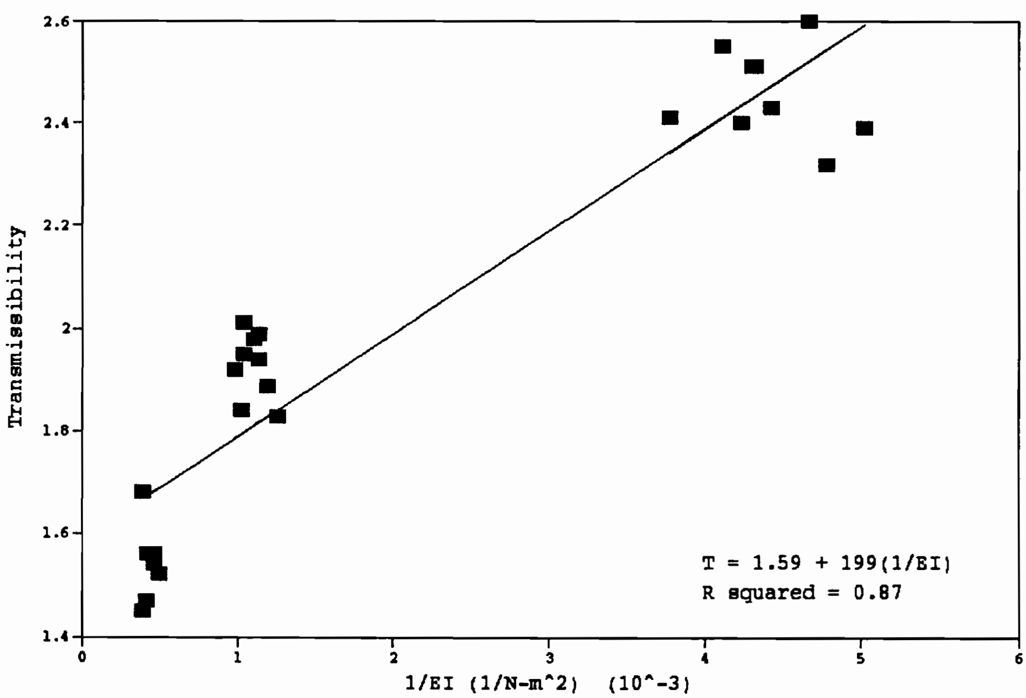

Figure 7.9. Transmissibility (T) as a function of deckboard EI for pallet sections under a 6855Pa static load subjected to forced vibration.

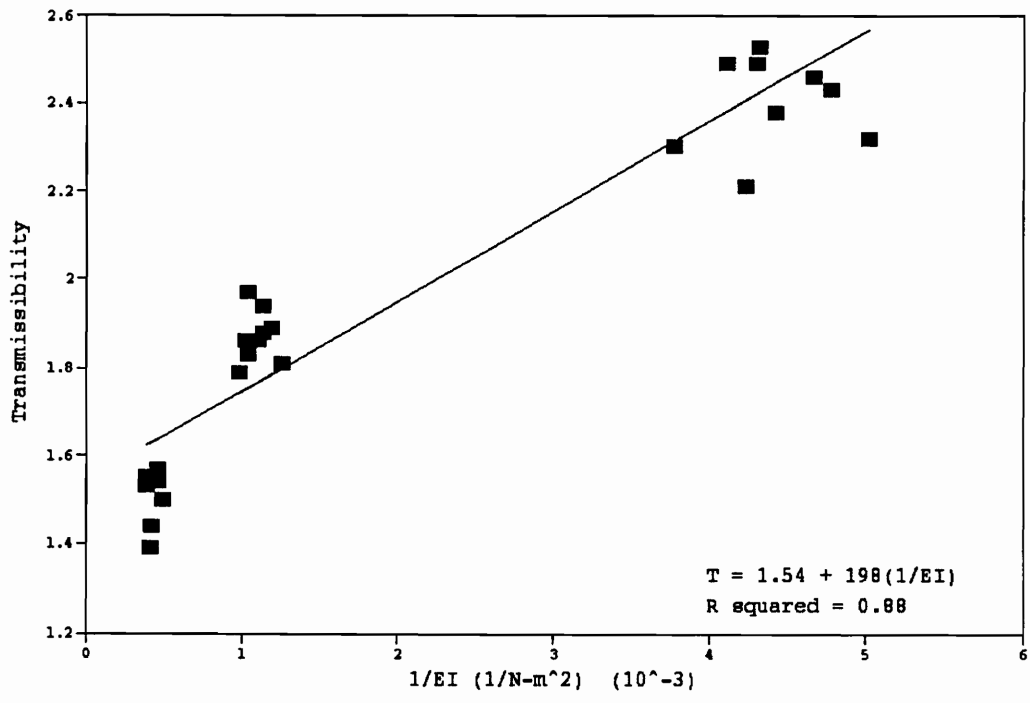

Figure 7.10. Transmissibility (T) as a function of deckboard EI for pallet sections under a 8617Pa static load subjected to forced vibration.

Within the assumptions of this study, these regressions indicate that a SDOF spring-mass system model can be used to describe the relationship between deckboard EI and resonant frequencies and transmissibilities of pallet decks.

For all three deckboard EI levels, resonant frequencies decreased as load levels increased. As the load level increased, the resonant frequencies between EI levels tended to converge; therefore, differences between the EI levels were more pronounced at the light load levels. The slope of the regression at each load level decreased uniformly from 0.78 to 0.14.

The transmissibilities at resonance were inversely related to EI. The decks with high EI's did not amplify input acceleration as much as the decks with lower EI's. The decks with the lowest EI values exhibited the lowest resonant frequencies and the highest transmissibilities. Static load level did not affect transmissibility except at the lightest load level.

A 400% increase in static load level decreased resonant frequencies of the pallet sections, on average, 173%. On the other hand, an 822% increase in EI (a 130% increase in deck thickness) yielded, on average, a 71% increase in resonant frequencies and a 49% decrease in transmissibilities. To significantly raise the resonant frequencies and to lower the transmissibilities of pallet decks, substantial investments must be made into manufacturing pallets with stiffer decks by using thicker or wider deckboards, higher density wood species, or shorter deckboard free spans (using a fourth or fifth stringer or wider stringers), all of which increase pallet cost.

The lightest static load level resulted in the greatest variations in resonant response. This was thought to have been caused by the effect of the variation of EI being more pronounced at the light load levels. This phenomena is due to the angular natural frequency being defined as the square root of the ratio of spring stiffness to mass ($\sqrt{k/m}$). As mass decreases, spring-mass systems are more sensitive to variations in spring stiffness.

Pallet section anomalies like bowed and twisted deckboards may have also influenced section resonant response. Pallet sections did not always sit level on the vibration table at the light load levels. It required heavier load levels to make the bottom decks conform to the vibration table's surface. The sections constructed from the stiffer decks were affected most by deckboard twists and bows because it required a greater load to make these sections conform to the vibration table. The data generated from pallet sections with extreme deckboard twist and bow was not included in the calculations or used for statistical inference.

The sections in this study were assembled with the same type of nail; therefore, fastener penetration varied with the thickness of the deck components. Compression perpendicular to grain beneath the nail heads also varied. The resulting joint stiffnesses between the sections were different with different deck thicknesses; consequently, some of the variation in deck response could be attributed to variation in joint stiffness.

To develop a procedure to predict pallet deck resonant frequencies, a method was devised to use the square root of pallet deck EI divided by deck surface area-load level (**P**). This data and regression for resonant frequency are shown in Figure 7.11. This transformation was based on the theory that the resonant frequency of a single degree of freedom spring-mass system is directly related to spring stiffness and inversely related to load level.

A similar transformation was applied to the transmissibility data where transmissibility was plotted versus the inverse of the square root of EI divided by area load level (**P**). The regression on the data for this transformation is shown in Figure 7.12.

An example application of the regression equations would be with the use of yellow-poplar pallets with deckboards of similar dimensions as those used in this study. Suppose that two pallets were constructed. One

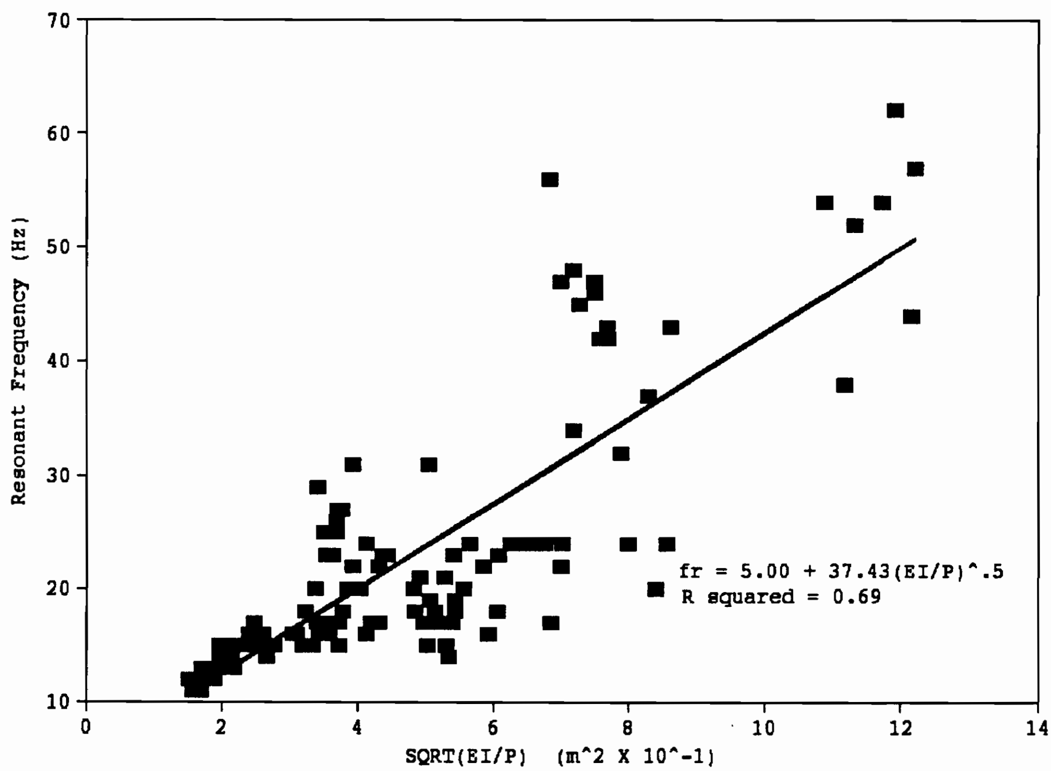

Figure 7.11. Regression of the pallet section resonant frequency data (f_r) for decks subjected to forced vibration as a function of deck EI adjusted for static area-load level (P).

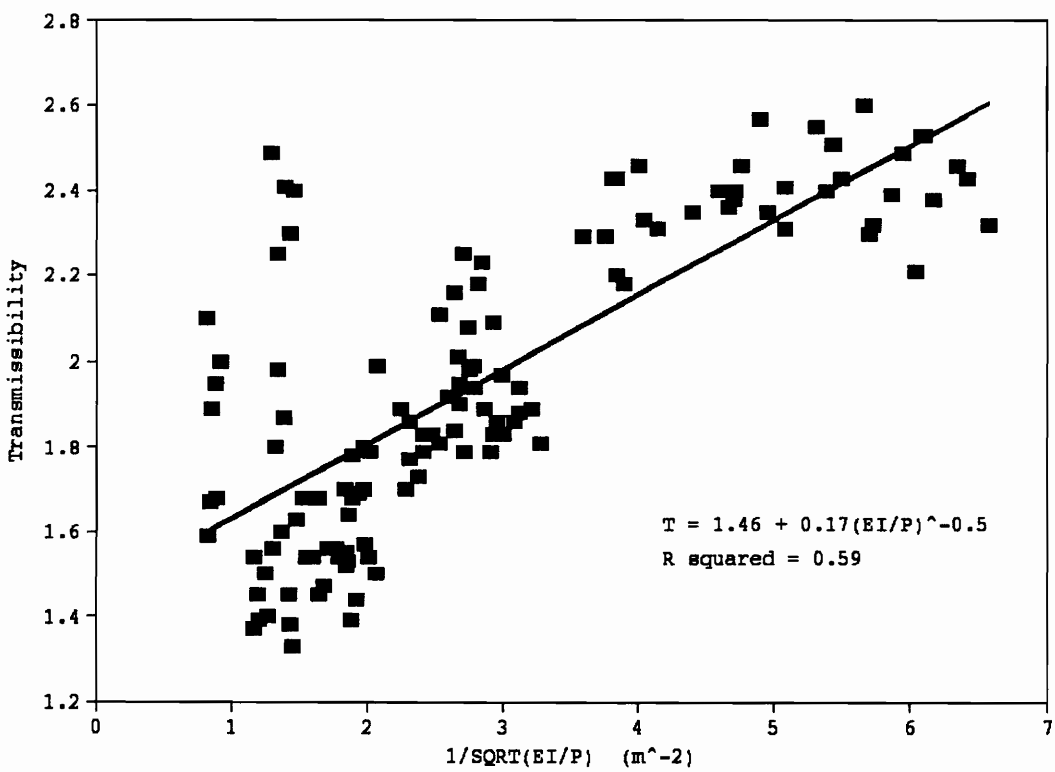

Figure 7.12. Regression of the pallet section transmissibility (T) data for decks subjected to forced vibration as a function of deck EI adjusted for static load level (P).

pallet was made with 10mm thick deckboards and the other with 22mm thick deckboards. The published average modulus of elasticity value for unseasoned defect free yellow-poplar is 8.41GPa (Forest Products Laboratory, 1987). Imagine that both pallets were carrying a 7157Pa load that corresponds to a 1219mm by 1016mm uniformly distributed pallet load of 8838N. The pallet constructed with the 10mm decks would have a predicted resonant frequency of 10Hz and a predicted transmissibility at resonance of 2.76. The pallet constructed with the 22mm thick deckboards would have a predicted resonant frequency of 22Hz and a transmissibility of 1.82. By increasing deck thickness 120%, one could increase the predicted resonant frequency of the pallet by 120% and decrease the predicted transmissibility by 52%.

To illustrate load effects for these regression analyses, imagine the same two pallets supporting a relatively light load level of 717Pa, an 888N, 1219mm by 1016mm pallet load. The 10mm thick deckboard pallet would have a predicted resonant frequency of 20Hz and a predicted transmissibility at resonance of 1.87. The 22mm thick deckboard pallet would have a predicted resonant frequency of 60Hz and a transmissibility of 1.57. Since resonant frequencies are more sensitive to light load levels, the difference in predicted resonant frequencies of these two pallets would be 200% with respect to the 10mm thick deckboard pallet. Differences in predicted transmissibilities would be 19%.

As another example of the use of the regression equations, suppose that two more pallets were constructed. The first pallet was made with aspen (Populus spp.), a low density relatively weak wood species, with 10mm thick deckboards. The second pallet was made from oak (Quercus spp.), a high density relatively strong wood species, with 22mm thick deckboards. Average modulus of elasticity values for defect free unseasoned aspen and oak are 7.82GPa and 9.36Gpa, respectively (Forest Products Laboratory, 1987). If both pallets were supporting a 7156Pa load corresponding to a 8859N uniformly distributed 1219mm by 1016mm pallet load. The aspen pallet would have a predicted resonant frequency of 9.5Hz and a predicted transmissibility at resonance of 2.87. The oak pallet would have a predicted resonant frequency of 23Hz and transmissibility of 1.80. By changing to a higher density wood species and increasing deck thickness, one could raise the predicted resonant frequencies of pallet decks 142% and decrease transmissibilities by 37%.

As these three examples show, significant changes in pallet deckboard EI is required to effect a change in resonant response. The regressions performed on the data are first approximations for the prediction of resonant frequencies and transmissibilities of the pallet decks. The regressions only apply to the pallet sections used in this study and what real products on real pallets experience may differ significantly due to the resonant characteristics of different unit-loads. These regressions do not include span effects. More research is necessary to improve the empirical relationships, and to correlate deck resonance with shocks and accelerations experienced by the product on the pallet.

Unit-loads are complex systems that have components that react together in response to forced vibration. The responses observed in this study may not apply to other types of loads exhibiting different responses to forced vibration.

Although deckboard span between stringers (L) is a significant contributor to deck stiffness, all the pallet sections used in this study had equal spans so that deckboard EI could be investigated. The effect of deckboard span on pallet resonant response should be examined in future research.

7.4. Conclusions

To prevent vibration caused damage, pallets should be designed to elevate unit-load resonant frequencies above the predominate forcing frequencies of the transport environment and to reduce transmissibilities to the unit-load. The conclusions from this study were:

1. The EI of pallet decks significantly influences the resonant frequencies of pallets under load. Decks with higher EI's result in higher resonant frequencies.

2. The EI of pallet decks significantly affects transmissibilities at resonance of forced vibration to the unit-load. Decks with higher EI's result in less amplification of input acceleration.

3. Load level affects resonant response of pallet decks with heavier loads resulting in the lower resonant frequencies. Load level did not significantly influence transmissibilities of pallet decks.

4. Pallet deck resonant response was found to be sensitive to pallet structural geometry; pallet sections constructed with deckboards with bow and twist were found to change modal response from top deck flexure to other forms of total pallet structural flexure.

CHAPTER 8

Summary, Conclusions, and Recommendations for Further Research

8.1. Summary

In Chapter 4, ten different pallet designs were evaluated for their influence on a unit-load's resonant response. All pallet designs lowered the unit-load's resonant frequencies and increased its transmissibilities. Pallet mode shapes were discussed in Chapter 5. Top deck flexure was the primary mode of vibration for three stringer pallets. Load effects were compared between box and bag loads. Box loads were found difficult to use because they shifted during testing; therefore, bag loads were used for further testing. A pallet section model was developed in Chapter 5 that controls experimental variables influencing pallet response to forced vibration.

The effect of joint stiffness on deck resonant response was discussed in Chapter 6. Two theoretical joint extremes, pinned and fixed joints, were compared for their influence on pallet deck resonant response. Stiffer joints resulted in higher resonant frequencies and lower transmissibilities.

Using pallet sections, the effect of deckboard EI was demonstrated in Chapter 7. Three top deckboard EI levels were evaluated for their resonant response and pallet sections with stiffer decks had higher resonant frequencies and lower transmissibilities.

8.2. Conclusions

The objective of this study was to determine how pallets respond to simple harmonic excitation by developing a physical pallet section model to simulate the response of full size three stringer pallets to forced vibration, and by examining the effects of deckboard EI, joint stiffness, and load level on pallet resonant response. A pallet section model was developed to investigate deckboard EI, joint stiffness, and static load level affects on deck resonant response. The following are deductions based on the results of these studies:

1. The primary mode of three stringer pallet vibration is top deck flexure.

2. Pallets tend to amplify input accelerations.

3. Pallets tend to lower the resonant frequency of unit-load systems compared to nonpalletized systems.

4. Pallets with stiffer decks amplify acceleration less and resonate at higher frequencies.

5. Joint stiffness influences pallet deck resonant response; pallets with stiff connections result in pallet decks exhibiting higher resonant frequencies and lower transmissibilities than pallets constructed with less stiff connections.

6. Deckboard EI influences pallet deck resonant response; pallets constructed with deckboards with high EI levels will result in pallet decks with higher resonant frequencies and lower transmissibilities than pallets constructed with low EI values.

7. Uniformly distributed load level affects deck resonant response; as load level increases resonant frequencies decrease.

8. Pallets should rest evenly on supporting surfaces to avoid additional modes of vibration that tend to amplify acceleration and lower resonant frequencies.

9. Packaging designers using tests that include pallets cannot ignore the influence of pallet design on the results of vibration tests.

To reduce the adverse affects of vibrations on unit-loads the pallet deck should be made as stiff as practically possible. A pallet's structure should follow, without amplification, the forcing accelerations from the transport vehicle and not magnify acceleration at periodic frequencies found in the transport environment. Such a pallet would have both a rigid top deck and stiff joints.

8.3. Recommendations

As a preliminary study of the affect of pallet design on the responses of unit-loads to vibrations that occur in the material handling environment, this study used a very simple approach to a complex problem. The following are recommendations for future pallet vibration research:

1. The study of deckboard span effects.

2. An analysis of pallet deck behavior at light load levels to determine how joints, pallet construction anomalies, and variation in deck stiffness influence the resonant response of pallet decks.

3. Detailed vibration studies of pallet joint stiffness effects using different pallet fasteners.

4. Studies of pallet load type and geometry effects using different types of loads, load distributions, and load restraint.

5. Expanded studies of pallet design effects using different pallet designs; the study of block pallet deck response to forced vibration.

6. Closer experimentation of pallet interactions with unit-loads at the interface between pallets and packaged products.

7. Modeling of pallet structures that incorporate a system approach reflecting the resonant characteristics of loads and true interactions at the pallet/load interface.

8. Development of unit-load vibration test using random vibration with power spectral density envelopes from truck vibration, rail vibration, air craft vibration, etc.

9. Development of techniques to find structural damping of unit-loads, including pallets.

10. Economic analysis of the implications of changing pallet design to prevent vibration caused product damage.

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APPENDICES

APPENDIX A

Raw Data

Table A-1. Resonant frequency (f_r) as a function of pallet design from vibration tests of statically loaded pallets.

Pallet Design	Test1	Test2	Test3 (Hz)	Test4	Test5	Mean	C.V. (%)
Boxes I*	13.0	13.0	13.5	13.0	13.0	13.1	2
Boxes F♦	13.0	13.5	13.5	13.0	13.0	13.2	2
1	11.0	11.0	11.0	11.0	12.0	11.2	4
2	9.0	9.0	9.0	9.0	9.0	9.0	0
3	13.0	13.0	13.0	13.0	12.0	12.8	3
4a	10.0	10.0	10.0	10.0	10.0	10.0	0
4b	11.0	11.0	11.0	10.5	10.5	10.8	2
4c	11.0	11.0	11.0	10.0	10.0	10.6	5
4d	11.0	12.0	13.0	12.0	11.0	11.8	6
4e	11.0	10.0	11.0	12.0	11.0	11.0	6
5a	12.0	11.5	11.5	12.0	12.0	11.8	2
5b	10.0	10.5	10.5	10.5	11.0	10.5	3
5c	11.0	12.0	12.0	12.0	12.0	11.8	3
5d	11.0	10.0	11.0	11.0	11.0	10.8	4
5e	10.0	11.0	10.0	10.0	11.0	11.0	4
6a	11.5	11.5	11.5	11.5	12.0	11.6	2
6b	11.0	11.5	12.0	11.5	11.5	11.5	3
6c	12.0	12.0	12.0	12.0	12.0	12.0	0
6d	11.0	11.0	11.0	11.0	11.0	11.0	0
6e	12.0	12.0	12.0	12.0	12.0	12.0	0
7a	11.5	11.5	12.0	12.0	12.0	11.8	2
7b	11.5	12.0	11.5	12.0	11.5	11.7	2
7c	12.5	12.0	12.0	12.0	12.0	12.1	2
7d	11.5	12.0	12.0	12.0	12.0	11.9	2
7e	11.0	12.0	11.5	11.0	11.0	11.3	4
8a	11.0	11.0	11.0	11.0	12.0	11.2	4
8b	12.0	11.5	11.5	12.0	12.0	11.8	2
8c	12.0	12.0	12.0	12.0	12.0	12.0	0
8d	12.0	12.0	12.0	12.0	12.0	12.0	0
9	12.0	12.0	12.0	12.0	12.0	12.0	0
10a	9.0	9.0	9.0	9.0	9.0	9.0	0
10b	9.0	9.0	9.0	9.0	9.0	9.0	0
10c	10.0	10.0	10.0	10.0	10.0	10.0	0

* Boxes I = initial box load testing
♦ Boxes F = final box load testing

Table A-2. Transmissibility (T) as a function of pallet design from vibration tests of statically loaded pallets.

Pallet Design	Test1	Test2	Test3	Test4	Test5	Mean	C.V. (%)
Boxes I*	4.00	3.90	3.90	4.10	4.30	4.04	3
Boxes F♦	4.16	4.16	4.08	4.16	4.20	4.20	1
1	4.77	4.98	4.77	4.42	4.12	4.61	6
2	4.61	4.61	4.58	4.60	4.65	4.61	0
3	4.25	4.34	4.34	4.34	4.38	4.33	1
4a	6.55	7.06	3.82	3.84	3.83	5.02	27
4b	5.91	6.08	5.91	6.00	6.08	6.00	1
4c	4.43	4.34	4.35	4.40	4.35	4.37	1
4d	4.70	4.68	4.68	4.72	4.89	4.73	2
4e	4.25	4.46	5.00	5.06	4.89	4.73	6
5a	4.60	4.87	4.85	4.51	4.93	4.75	3
5b	4.43	4.43	4.60	4.60	4.17	4.45	3
5c	4.17	4.25	4.42	4.51	4.34	4.34	3
5d	4.89	5.32	5.34	5.61	5.53	5.34	4
5e	5.57	7.44	8.08	5.32	5.53	6.39	16
6a	5.00	5.19	5.27	5.10	5.19	5.15	2
6b	4.09	4.00	4.34	4.34	4.51	4.26	4
6c	5.15	5.02	4.93	5.23	5.11	5.09	2
6d	4.47	5.96	5.96	5.64	5.23	5.45	9
6e	4.47	4.47	4.68	4.68	4.68	4.60	2
7a	4.47	4.68	4.77	4.68	5.02	4.72	3
7b	5.06	5.32	5.32	5.19	5.36	5.25	2
7c	4.96	4.90	4.52	4.48	4.48	4.67	4
7d	4.68	4.68	4.80	4.79	4.79	4.75	1
7e	4.89	5.10	4.77	4.72	5.32	4.96	4
8a	3.96	3.92	3.68	4.00	4.08	3.93	3
8b	3.68	3.80	4.08	3.80	3.76	3.82	3
8c	4.16	4.16	4.40	4.40	4.16	4.26	3
8d	4.17	4.18	4.25	4.10	4.00	4.14	2
9	4.17	4.89	5.74	5.53	5.74	5.21	11
10a	4.50	4.50	4.60	4.40	4.40	4.48	2
10b	4.60	4.60	4.60	4.50	4.50	4.56	1
10c	4.25	4.30	4.35	4.20	4.25	4.27	1

* Boxes I = initial box load testing
♦ Boxes F = final box load testing

Table A-3. Physical dimensions, modulus of elasticity, and EI for the top deckboards of the pallet sections used in Chapter 6.

Section	Width (mm)	Thickness (mm)	E (GPa)	EI ($N\text{-}m^2$)
A1	203	10	13.5	202
A2	203	10	13.5	212
A3	203	10	13.0	201
B1	203	16	11.8	812
B2	203	16	13.4	926
B3	203	16	12.4	843
C1	202	23	11.7	2261
C2	203	22	12.9	2454
C3	203	22	12.9	2457
D1	203	10	15.7	240
D2	203	10	13.7	209
D3	203	10	13.1	199
E1	203	16	12.4	860
E2	203	16	12.0	821
E3	203	16	11.7	802
F1	203	22	11.7	2210
F2	203	22	10.8	1960
F3	203	22	11.5	2179

Table A-4. Resonant frequencies (f_r) of pallet sections as a function of deck thickness for pinned connections at five different static load levels.

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
A1♠	1	14	15	15	16	15	3
	2	13	13	13	13	13	0
	3	12	12	12	12	12	0
	4	11	11	11	11	11	0
	5	10	10	10	10	10	0
	R1♦	15					
A2	1	16	15	15	15	15	0
	2	14	13	13	13	13	0
	3	13	13	13	13	13	0
	4	11	11	11	12	11	4
	5	11	11	11	11	11	0
	R1	17					
A3	1	17	17	17	17	17	0
	2	16	15	15	15	15	0
	3	14	14	14	14	14	0
	4	11	11	11	11	11	0
	5	10	10	10	10	10	0
	R1	17					
B1♣	1	18	18	18	18	18	0
	2	16	17	17	17	17	0
	3	18	18	18	18	18	0
	4	14	14	14	14	14	0
	5	16	14	14	14	14	0
	R1	19					
B2	1	22	22	22	22	22	0
	2	16	17	16	17	17	3
	3	18	18	18	18	18	0
	4	14	15	15	15	15	0
	5	15	14	14	14	14	0
	R1	23					
B3	1	32	32	32	32	32	0
	2	17	18	18	18	18	0
	3	19	19	19	19	19	0
	4	14	14	14	14	14	0
	5	15	14	14	14	14	0
	R1	32					

♦ R1 = retest at load 1
♠ A = 10mm pinned
♣ B = 16mm pinned

Table A-4. (Continued)

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
C1♣	1	20	21	22	17	20	11
	2	15	16	15	15	15	3
	3	14	16	17	16	16	3
	4	16	15	15	15	15	0
	5	16	15	15	15	15	0
	R1♦	15					
C2	1	17	18	18	18	18	0
	2	17	18	18	18	18	0
	3	16	17	17	17	17	0
	4	16	16	16	16	16	0
	5	17	16	16	16	16	0
	R1	16					
C3	1	16	16	16	16	16	0
	2	16	16	16	16	16	0
	3	16	16	16	16	16	0
	4	16	16	16	16	16	0
	5	16	15	15	15	15	0
	R1	16					

♠ R1 = retest of load 1
♣ C = 22mm pinned

Table A-5. Resonant frequencies (f_r) of pallet sections as a function of deck thickness for glued connections at five different static load levels.

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
D1♦	1	28	28	28	28	28	0
	2	16	15	15	15	15	0
	3	14	14	14	14	14	0
	4	13	13	13	13	13	0
	5	13	13	13	13	13	0
	R1♦	27					
D2	1	33	33	34	34	33	1
	2	20	20	20	20	20	0
	3	15	15	15	15	15	0
	4	14	14	14	14	14	0
	5	14	14	14	14	14	0
	R1	37					
D3	1	37	39	40	38	39	2
	2	16	16	16	16	16	0
	3	14	14	14	14	14	0
	4	13	12	12	12	12	0
	5	11	11	11	11	11	0
	R1	37					
E1♣	1	16	16	16	16	16	0
	2	19	18	18	18	18	0
	3	18	18	18	18	18	0
	4	19	19	19	19	19	0
	5	17	15	16	16	15	3
	R1	16					
E2	1	22	22	22	22	22	0
	2	18	18	18	18	18	3
	3	20	16	16	16	16	0
	4	17	16	16	16	16	0
	5	16	15	15	15	15	0
	R1	22					
E3	1	20	21	21	21	21	0
	2	19	18	18	18	18	0
	3	20	16	16	16	16	0
	4	15	15	15	15	15	0
	5	16	15	15	15	15	0
	R1	21					

♠ R1 = retest at load 1
♦ D = 10mm epoxy
♣ E = 16mm epoxy

Table A-5. (Continued)

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
F1♣	1	22	22	22	22	22	0
	2	15	15	15	15	15	0
	3	16	16	17	17	16	3
	4	17	17	17	17	17	0
	5	17	16	16	16	16	0
	R1♦	16					
F2	1	24	24	24	24	24	0
	2	18	18	18	18	18	0
	3	16	17	17	17	17	0
	4	17	17	17	17	17	0
	5	16	17	17	17	17	0
	R1	17					
F3	1	60	61	60	60	60	1
	2	42	42	42	42	42	0
	3	21	22	22	22	22	0
	4	19	19	17	19	18	5
	5	17	16	17	17	16	3
	R1	50					

♦ R1 = retest of load 1
♣ F = 22mm epoxy

Table A-6. Transmissibilities (T) of pallet sections as a function of deck thickness for pinned connections at five different static load levels.

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
A1♣	1	2.20	2.60	2.60	2.60	2.60	0
	2	2.70	2.90	2.90	2.90	2.90	0
	3	2.80	2.80	2.80	2.80	2.80	0
	4	2.60	2.90	2.80	2.80	2.83	1
	5	2.80	2.70	2.70	2.70	2.70	0
	R1♦	2.80					
A2	1	2.40	2.90	2.95	3.00	2.95	1
	2	2.60	2.80	2.95	2.95	2.90	2
	3	2.60	2.60	2.60	2.60	2.60	0
	4	2.65	2.65	2.70	2.60	2.65	1
	5	2.40	2.45	2.50	2.50	2.48	0
	R1	2.40					
A3	1	2.60	2.75	2.80	2.85	2.80	1
	2	2.60	2.70	2.75	2.80	2.75	1
	3	2.80	2.80	2.80	2.80	2.80	0
	4	2.50	2.70	2.70	2.70	2.70	0
	5	2.60	2.60	2.60	2.60	2.60	0
	R1	3.00					
B1♠	1	2.00	2.00	2.00	2.00	2.00	0
	2	1.90	1.95	1.95	1.90	1.93	1
	3	2.20	2.20	2.20	2.20	2.20	0
	4	2.00	2.20	2.20	2.20	2.20	0
	5	2.00	2.00	2.00	2.00	2.00	0
	R1	1.95					
B2	1	1.70	1.70	1.70	1.70	1.70	0
	2	1.80	1.80	1.80	1.80	1.80	0
	3	2.40	2.20	2.20	2.20	2.20	0
	4	2.00	2.20	2.20	2.20	2.20	0
	5	2.00	2.10	2.10	2.10	2.10	0
	R1	1.75					
B3	1	1.60	1.60	1.80	1.80	1.73	5
	2	1.70	1.70	1.85	1.85	1.80	3
	3	2.00	2.00	2.05	2.00	2.02	1
	4	1.90	2.05	2.10	2.10	2.08	1
	5	1.80	1.90	1.95	1.95	1.93	1
	R1	1.80					

♦ R1 = retest of load 1
♣ A = 10mm pinned
♠ B = 16mm pinned

Table A-6. (Continued)

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
C1♣	1	2.05	1.90	1.90	1.90	1.90	0
	2	1.80	1.95	1.90	1.90	1.92	1
	3	1.80	1.80	1.80	1.80	1.80	0
	4	1.95	1.95	1.96	1.92	1.94	0
	5	1.80	1.84	1.86	1.90	1.87	1
	R1♦	1.84					
C2	1	1.60	1.70	1.70	1.70	1.70	0
	2	1.50	1.70	1.70	1.70	1.70	0
	3	1.56	1.70	1.64	1.70	1.68	1
	4	1.78	1.90	1.90	1.90	1.90	0
	5	1.80	1.80	1.80	1.80	1.80	0
	R1	1.64					
C3	1	1.50	1.56	1.60	1.60	1.60	1
	2	1.72	1.60	1.80	1.82	1.74	5
	3	1.75	1.80	1.80	1.80	1.80	0
	4	1.95	1.99	2.00	2.00	2.00	0
	5	1.80	1.86	1.90	1.95	1.90	1
	R1	1.70					

♦ R1 = retest of load 1
♣ C = 22mm pinned

Table A-7. Transmissibilities (T) of pallet sections as a function of deck thickness for glued connections at five different static load levels.

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
D1♣	1	1.90	1.95	1.92	1.90	1.92	1
	2	2.06	2.40	2.50	2.50	2.47	1
	3	2.15	2.30	2.40	2.40	2.37	1
	4	2.60	2.90	2.80	2.80	2.83	1
	5	2.30	2.30	2.36	2.40	2.35	1
	R1♦	2.50					
D2	1	2.28	2.30	2.30	2.30	2.30	0
	2	1.90	1.90	1.92	1.92	1.91	0
	3	1.92	2.08	2.12	2.16	2.12	1
	4	2.65	2.65	2.70	2.60	2.65	1
	5	2.10	2.24	2.30	2.30	2.28	1
	R1	2.30					
D3	1	2.00	1.94	1.70	1.85	1.83	5
	2	2.02	2.20	2.20	2.20	2.20	0
	3	2.18	2.32	2.40	2.40	2.37	1
	4	2.26	2.38	2.40	2.44	2.41	1
	5	2.40	2.55	2.60	2.60	2.58	0
	R1	2.00					
E1♠	1	1.60	1.60	1.80	1.80	1.73	5
	2	1.56	1.60	1.70	1.76	1.69	3
	3	1.56	1.60	1.62	1.64	1.62	1
	4	1.60	1.66	1.68	1.70	1.68	0
	5	1.60	1.62	1.60	1.64	1.62	1
	R1	1.70					
E2	1	1.46	1.48	1.48	1.46	1.47	0
	2	1.60	1.68	1.68	1.70	1.69	0
	3	1.70	1.70	1.70	1.70	1.70	0
	4	1.70	1.76	1.75	1.78	1.76	0
	5	1.68	1.72	1.75	1.80	1.76	1
	R1	1.50					
E3	1	1.60	1.70	1.70	1.70	1.70	0
	2	1.72	1.80	1.80	1.80	1.80	0
	3	1.80	1.78	1.78	1.80	1.79	0
	4	1.76	1.90	1.96	1.95	1.94	1
	5	1.80	1.84	1.86	1.90	1.87	1
	R1	1.80					

♦ R1 = retest of load 1
♣ D = 10mm epoxy
♠ E = 16mm epoxy

Table A-7. (Continued)

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
F1♣	1	1.40	1.44	1.44	1.40	1.43	1
	2	1.56	1.50	1.55	1.56	1.54	1
	3	1.60	1.64	1.64	1.68	1.65	1
	4	1.70	1.76	1.75	1.76	1.76	0
	5	1.60	1.60	1.64	1.64	1.63	1
	R1♦	1.66					
F2	1	1.60	1.60	1.60	1.60	1.60	0
	2	1.40	1.40	1.40	1.40	1.40	0
	3	1.34	1.34	1.32	1.30	1.32	1
	4	1.78	1.90	1.90	1.90	1.90	0
	5	1.42	1.46	1.40	1.40	1.42	1
	R1	1.60					
F3	1	1.50	1.52	1.52	1.52	1.52	0
	2	1.56	1.58	1.60	1.60	1.59	0
	3	1.52	1.58	1.60	1.60	1.59	0
	4	1.60	1.60	1.68	1.70	1.66	2
	5	1.50	1.56	1.60	1.60	1.59	1
	R1	1.60					

♦ R1 = retest of load 1
♣ F = 22mm epoxy

Table A-8. The dimensions, modulus of elasticity (E), and EI for the top deckboards of the pallet sections used in vibration testing.

Section	Width (mm)	Thickness (mm)	E (GPa)	EI ($N\text{-}m^2$)
I-1♦	203	10	14.7	226
I-2	203	10	13.9	214
I-3	203	10	15.5	243
I-4	203	10	15.4	231
I-5	203	10	16.4	232
I-6	203	10	15.4	236
I-7	203	10	17.4	265
I-8	203	10	13.7	209
I-9	203	10	13.2	199
I-10	203	10	15.1	233
II-1♣	203	16	11.0	761
II-2	203	16	14.1	981
II-3	203	16	11.9	797
II-4	203	16	12.1	834
II-5	203	16	13.4	905
II-6	203	16	12.7	879
II-7	203	16	12.9	881
II-8	203	16	14.0	957
II-9	203	16	14.8	1014
II-10	203	16	14.0	959
III-1♠	203	22	12.8	2425
III-2	203	22	10.3	1953
III-3	203	22	11.9	2194
III-4	203	22	11.2	2134
III-5	203	22	9.2	1737
III-6	203	22	13.4	2524
III-7	203	22	13.4	2547
III-8	203	22	11.8	2221
III-9	203	22	10.8	2024
III-10	203	22	12.4	2352

♦ I signifies 10mm deck thickness
♣ II signifies 16mm deck thickness
♠ III signifies 22mm deck thickness

Table A-9. Resonant frequency (f_r) as a function of load level for pallet sections with 10mm thick decks subjected to forced vibration.

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
I-1	1	23	23	23	23	23	0
	2	16	16	16	16	16	0
	3	13	13	13	13	13	0
	4	13	13	13	13	13	0
	5	12	12	12	12	12	0
	R1♦	23					
I-2	1	23	23	23	23	23	0
	2	16	16	16	15	15	3
	3	15	13	13	13	13	0
	4	12	12	12	12	12	0
	5	11	12	12	12	12	0
	R1	16					
I-3	1	27	27	27	27	27	0
	2	15	14	14	14	14	0
	3	14	13	13	13	13	0
	4	13	12	12	12	12	0
	5	11	11	11	11	11	0
	R1	27					
I-4	1	36	31	23	23	25	15
	2	16	15	15	15	15	0
	3	15	15	15	15	15	0
	4	13	13	13	13	13	0
	5	12	12	12	12	12	0
	R1	23					
I-5	1	26	26	26	26	26	0
	2	16	16	16	17	16	3
	3	15	14	14	14	14	0
	4	14	13	13	13	13	0
	5	12	12	12	12	12	0
	R1	27					

♦ R1 = retest of load 1, single test

Table A-9. (Continued)

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
I-6	1	27	27	27	27	27	0
	2	16	15	15	15	15	0
	3	16	14	14	14	14	0
	4	13	13	13	13	13	0
	5	13	12	12	12	12	0
	R1♦	21					
I-7	1	31	31	32	32	31	2
	2	16	15	15	15	15	0
	3	16	15	15	15	15	0
	4	14	14	14	14	14	0
	5	13	13	13	13	13	4
	R1	33					
I-8	1	28	26	25	25	25	2
	2	17	17	17	17	17	0
	3	15	14	14	14	14	0
	4	13	13	13	13	13	0
	5	12	12	12	12	12	0
	R1	24					
I-9	1	28	29	29	29	29	0
	2	17	16	16	16	16	0
	3	15	15	15	15	15	0
	4	13	13	13	13	13	0
	5	13	12	12	12	12	0
	R1	31					
I-10	1	14	15	15	15	15	0
	2	15	15	15	15	15	0
	3	15	15	15	15	15	0
	4	14	14	14	14	14	0
	5	13	13	13	13	13	0
	R1	27					

♦ R1 = retest of load 1, single test

Table A-10. Transmissibility (T) as a function of load level for pallet sections with 10mm thick decks subjected to forced vibration.

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
I-1	1	2.12	2.10	2.08	2.06	2.08	1
	2	2.18	2.18	2.18	2.18	2.18	0
	3	2.20	2.42	2.50	2.45	2.46	1
	4	2.36	2.42	2.44	2.42	2.43	0
	5	2.22	2.30	2.44	2.40	2.38	3
	R1♦	1.90					
I-2	1	2.16	2.16	2.20	2.20	2.18	1
	2	2.32	2.40	2.50	2.55	2.48	3
	3	2.32	2.50	2.60	2.60	2.57	2
	4	2.62	2.60	2.60	2.60	2.60	0
	5	2.34	2.44	2.50	2.50	2.48	1
	R1	1.60					
I-3	1	2.16	2.16	2.16	2.15	2.16	0
	2	2.10	2.26	2.30	2.30	2.29	1
	3	2.23	2.36	2.40	2.44	2.40	1
	4	2.38	2.50	2.55	2.60	2.55	2
	5	2.36	2.44	2.52	2.52	2.49	2
	R1	1.90					
I-4	1	1.82	1.78	1.80	1.80	1.79	1
	2	2.06	2.16	2.20	2.24	2.20	2
	3	2.28	2.36	2.40	2.44	2.40	1
	4	2.40	2.50	2.54	2.50	2.51	1
	5	2.40	2.50	2.54	2.56	2.53	1
	R1	2.04					
I-5	1	2.26	2.26	2.24	2.24	2.25	0
	2	2.36	2.42	2.42	2.46	2.43	1
	3	2.26	2.34	2.40	2.40	2.38	1
	4	2.35	2.46	2.50	2.56	2.51	2
	5	2.38	2.46	2.50	2.50	2.49	1
	R1	2.30					

♦ R1 = retest of load 1, single test

Table A-10. (Continued).

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
I-6	1	1.90	1.90	1.90	1.90	1.90	0
	2	2.16	2.38	2.42	2.48	2.43	2
	3	2.10	2.30	2.38	2.40	2.36	2
	4	2.20	2.40	2.40	2.40	2.40	0
	5	2.40	2.18	2.20	2.26	2.21	2
	R1♦	2.22					
I-7	1	2.15	2.12	2.12	2.10	2.11	0
	2	2.06	2.24	2.30	2.32	2.29	2
	3	2.18	2.30	2.36	2.40	2.35	2
	4	2.24	2.38	2.40	2.44	2.41	1
	5	2.20	2.30	2.30	2.30	2.30	0
	R1	2.06					
I-8	1	2.20	2.24	2.22	2.24	2.23	0
	2	2.22	2.30	2.32	2.36	2.33	1
	3	2.26	2.30	2.36	2.40	2.35	2
	4	2.20	2.30	2.30	2.36	2.32	1
	5	2.40	2.40	2.42	2.48	2.43	1
	R1	2.26					
I-9	1	2.16	2.10	2.10	2.06	2.09	1
	2	2.30	2.32	2.30	2.30	2.31	0
	3	2.20	2.30	2.32	2.32	2.31	0
	4	2.24	2.36	2.40	2.40	2.39	1
	5	2.24	2.30	2.30	2.35	2.32	1
	R1	1.95					
I-10	1	1.85	2.10	2.16	2.20	2.15	2
	2	2.20	2.30	2.30	2.30	2.30	0
	3	2.19	2.20	2.24	2.26	2.23	1
	4	2.24	2.30	2.34	2.38	2.34	1
	5	2.20	2.30	2.34	2.36	2.33	1
	R1	2.18					

♦ R1 = retest of load 1, single test

Table A-11. Resonant frequency (f_r) as a function of load level for pallet sections with 16mm thick decks subjected to forced vibration.

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
II-1	1	18	19	19	19	19	0
	2	19	18	20	20	19	5
	3	19	19	19	19	19	0
	4	17	17	17	17	17	0
	5	15	16	16	16	16	0
	R1♦	19					
II-2	1	40	42	42	42	42	0
	2	24	24	24	24	24	0
	3	25	25	25	19	23	12
	4	19	18	18	18	18	0
	5	20	20	20	20	20	0
	R1	49					
II-3	1	52	56	56	56	56	0
	2	20	20	20	20	20	0
	3	22	22	22	22	22	0
	4	17	17	17	17	17	0
	5	16	16	16	16	16	0
	R1	50					
II-4	1	46	47	47	47	47	0
	2	23	21	21	21	21	0
	3	21	20	20	20	20	0
	4	18	17	17	17	17	0
	5	16	16	16	16	16	0
	R1	44					
II-5	1	43	45	45	45	45	0
	2	23	18	18	18	18	0
	3	22	17	17	17	17	0
	4	17	17	17	17	17	0
	5	18	18	18	18	18	0
	R1	27					

♦ R1 = retest of load 1, single test

Table A-11. (Continued)

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
II-6	1	46	48	48	48	48	0
	2	30	31	31	31	31	0
	3	24	24	24	24	24	0
	4	15	16	16	16	16	0
	5	16	15	15	15	15	0
	R1♦	48					
II-7	1	33	32	35	35	34	4
	2	17	19	19	19	19	0
	3	20	16	16	16	16	0
	4	18	16	16	16	16	0
	5	17	17	15	15	15	6
	R1	35					
II-8	1	48	47	47	47	47	0
	2	25	17	17	17	17	0
	3	18	17	17	17	17	0
	4	15	15	15	15	15	0
	5	15	15	15	15	15	0
	R1	46					
II-9	1	41	42	42	42	42	0
	2	18	19	19	19	19	0
	3	23	23	23	23	23	0
	4	20	20	20	20	20	0
	5	19	16	16	16	16	0
	R1	43					
II-10	1	44	46	47	47	46	1
	2	19	21	21	21	21	0
	3	22	22	22	22	22	0
	4	18	17	17	17	17	0
	5	18	15	15	15	15	0
	R1	48					

♦ R1 = retest of load 1, single test

Table A-12. Transmissibility (T) as a function of load level for pallet sections with 16mm thick decks subjected to forced vibration.

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
II-1	1	1.74	1.80	1.80	1.86	1.82	2
	2	1.70	1.64	1.80	1.90	1.78	6
	3	1.66	1.70	1.73	1.70	1.71	1
	4	1.76	1.80	1.84	1.86	1.83	1
	5	1.70	1.84	1.84	1.86	1.85	1
	R1♦	1.90					
II-2	1	1.72	1.80	1.80	1.80	1.80	0
	2	1.62	1.64	1.64	1.64	1.64	0
	3	1.75	1.70	1.70	1.70	1.70	0
	4	1.86	1.84	1.84	1.84	1.84	0
	5	1.88	1.88	1.84	1.86	1.86	1
	R1	1.96					
II-3	1	2.50	2.40	2.40	2.40	2.40	0
	2	1.76	1.96	2.00	2.00	1.99	1
	3	1.80	1.80	1.80	1.82	1.81	1
	4	1.76	1.80	1.86	1.84	1.83	1
	5	1.70	1.80	1.80	1.82	1.81	1
	R1	2.30					
II-4	1	2.30	2.30	2.30	2.30	2.30	0
	2	1.70	1.76	1.80	1.80	1.79	1
	3	1.80	1.82	1.84	1.84	1.83	1
	4	1.76	1.86	1.90	1.90	1.89	1
	5	1.78	1.88	1.90	1.90	1.89	1
	R1	1.90					
II-5	1	1.64	1.60	1.60	1.60	1.60	0
	2	1.76	1.66	1.70	1.70	1.69	1
	3	1.74	1.70	1.72	1.76	1.73	1
	4	1.88	1.95	1.98	2.00	1.98	1
	5	1.80	1.86	1.86	1.86	1.86	0
	R1	1.86					

♦ R1 = retest of load 1, single test

Table A-12. (Continued)

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
II-6	1	2.40	2.40	2.42	2.42	2.41	0
	2	1.80	1.80	1.80	1.80	1.80	0
	3	1.78	1.78	1.78	1.80	1.79	1
	4	1.75	1.92	1.95	1.95	1.94	1
	5	1.80	1.92	1.94	1.95	1.94	1
	R1♦	2.50					
II-7	1	1.90	1.88	1.88	1.84	1.87	1
	2	1.64	1.70	1.70	1.70	1.70	0
	3	1.72	1.80	1.82	1.86	1.83	1
	4	1.94	1.96	2.00	2.00	1.99	1
	5	1.90	1.90	1.84	1.90	1.88	2
	R1	1.90					
II-8	1	1.96	1.98	1.98	1.98	1.98	0
	2	1.58	1.66	1.68	1.70	1.68	1
	3	1.66	1.75	1.76	1.80	1.77	1
	4	1.80	1.95	1.95	1.96	1.95	0
	5	1.92	1.90	1.76	1.84	1.83	3
	R1	2.10					
II-9	1	2.40	2.48	2.50	2.50	2.49	0
	2	1.56	1.70	1.70	1.70	1.70	0
	3	1.86	1.86	1.90	1.90	1.89	1
	4	1.84	1.90	1.92	1.94	1.92	1
	5	1.74	1.78	1.80	1.79	1.79	1
	R1	2.60					
II-10	1	2.20	2.24	2.26	2.26	2.25	0
	2	1.72	1.76	1.78	1.80	1.78	1
	3	1.80	1.86	1.86	1.86	1.86	0
	4	1.82	2.00	2.00	2.04	2.01	1
	5	1.90	1.95	1.96	2.00	1.97	1
	R1	2.05					

♦ R1 = retest of load 1, single test

Table A-13. Resonant frequency (f_r) as a function of load level for pallet sections with 22mm thick decks subjected to forced vibration.

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
III-1	1	60	62	62	62	62	0
	2	20	20	20	20	20	0
	3	15	15	15	15	15	0
	4	17	17	17	17	17	0
	5	16	16	16	16	16	0
	R1♦	62					
III-2	1	15	16	16	16	16	0
	2	12	12	12	12	12	0
	3	15	13	15	15	14	6
	4	14	14	14	14	14	0
	5	14	14	14	14	14	0
	R1	13					
III-3	1	52	52	52	52	52	0
	2	24	24	24	24	24	0
	3	24	24	24	24	24	0
	4	24	24	24	24	24	0
	5	15	15	15	15	15	0
	R1	48					
III-4	1	38	38	38	39	38	1
	2	32	32	32	32	32	0
	3	24	24	24	24	24	0
	4	20	20	20	20	20	0
	5	19	17	17	17	17	0
	R1	48					
III-5	1	29	29	29	29	29	0
	2	20	17	17	17	17	0
	3	17	17	17	17	17	0
	4	20	16	16	16	16	0
	5	15	16	16	16	16	0
	R1	19					

♦ R1 = retest of load 1, single test

Table A-13. (Continued)

Section	Load #	Pretest	Test1	Test2 (Hz)	Test3	Mean	C.V. (%)
III-6	1	42	44	44	44	44	0
	2	24	24	24	24	24	0
	3	24	22	22	22	22	0
	4	23	18	18	18	18	0
	5	17	17	17	17	17	0
	R1♦	40					
III-7	1	56	56	58	58	57	2
	2	43	43	43	43	43	0
	3	24	24	24	24	24	0
	4	23	23	23	23	23	0
	5	23	18	18	18	18	0
	R1	56					
III-8	1	15	15	15	15	15	0
	2	18	18	17	16	17	5
	3	18	17	17	17	17	0
	4	16	16	17	17	16	3
	5	16	16	21	20	19	11
	R1	18					
III-9	1	52	54	54	54	54	0
	2	42	43	43	43	43	0
	3	25	25	25	25	25	0
	4	23	23	23	23	23	0
	5	20	18	18	18	18	0
	R1	54					
III-10	1	54	54	54	54	54	0
	2	37	38	37	37	37	1
	3	24	24	24	24	24	0
	4	24	23	23	22	22	2
	5	23	17	17	17	17	0
	R1	56					

♦ R1 = retest of load 1, single test

Table A-14. Transmissibility as a function of load level for pallet sections with 22mm thick decks subjected to forced vibration.

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
III-1	1	1.60	1.66	1.68	1.68	1.67	1
	2	1.36	1.40	1.48	1.48	1.45	3
	3	1.28	1.30	1.35	1.35	1.33	2
	4	1.40	1.48	1.46	1.48	1.47	1
	5	1.36	1.36	1.40	1.40	1.39	1
	R1♦	1.64					
III-2	1	1.70	1.70	1.70	1.70	1.70	0
	2	1.92	1.96	1.96	1.96	1.96	0
	3	1.80	1.88	1.86	1.90	1.88	1
	4	1.80	1.84	1.84	1.84	1.84	0
	5	1.76	1.80	1.80	1.80	1.80	0
	R1	1.78					
III-3	1	1.92	1.94	1.96	1.96	1.95	1
	2	1.50	1.50	1.50	1.50	1.50	0
	3	1.62	1.68	1.68	1.68	1.68	0
	4	1.52	1.56	1.56	1.56	1.56	0
	5	1.48	1.56	1.58	1.58	1.57	1
	R1	1.90					
III-4	1	1.60	1.68	1.68	1.68	1.68	0
	2	1.36	1.40	1.40	1.40	1.40	0
	3	1.48	1.52	1.54	1.55	1.54	1
	4	1.50	1.52	1.56	1.53	1.54	1
	5	1.44	1.52	1.55	1.56	1.54	1
	R1	1.90					
III-5	1	1.36	1.36	1.36	1.36	1.36	0
	2	1.44	1.48	1.48	1.48	1.48	0
	3	1.50	1.52	1.55	1.56	1.54	1
	4	1.52	1.56	1.56	1.56	1.56	0
	5	1.50	1.60	1.60	1.60	1.60	0
	R1	1.60					

♦ R1 = retest of load 1, single test

Table A-14. (Continued)

Section	Load #	Pretest	Test1	Test2	Test3	Mean	C.V. (%)
III-6	1	1.58	1.56	1.60	1.60	1.59	1
	2	1.36	1.38	1.36	1.36	1.37	1
	3	1.36	1.36	1.38	1.40	1.38	1
	4	1.42	1.40	1.46	1.48	1.45	2
	5	1.50	1.50	1.52	1.56	1.53	2
	R1♦	1.76					
III-7	1	2.06	2.08	2.12	2.10	2.10	1
	2	1.56	1.52	1.56	1.55	1.54	1
	3	1.46	1.46	1.46	1.44	1.45	1
	4	1.66	1.66	1.68	1.70	1.68	1
	5	1.56	1.55	1.55	1.55	1.55	0
	R1	2.16					
III-8	1	1.44	1.52	1.54	1.56	1.54	1
	2	1.55	1.52	1.56	1.55	1.54	1
	3	1.60	1.64	1.68	1.72	1.68	2
	4	1.55	1.64	1.66	1.66	1.65	1
	5	1.58	1.55	1.70	1.60	1.62	4
	R1	1.56					
III-9	1	1.96	2.00	2.00	2.00	2.00	0
	2	1.56	1.56	1.56	1.56	1.56	0
	3	1.52	1.52	1.55	1.55	1.54	1
	4	1.52	1.52	1.52	1.52	1.52	0
	5	1.46	1.50	1.50	1.50	1.50	0
	R1	1.92					
III-10	1	1.86	1.88	1.90	1.90	1.89	1
	2	1.35	1.36	1.40	1.40	1.39	1
	3	1.60	1.60	1.64	1.64	1.63	1
	4	1.56	1.56	1.56	1.56	1.56	0
	5	1.46	1.40	1.46	1.46	1.44	2
	R1	2.00					

♦ R1 = retest of load 1, single test

APPENDIX B

Vibration Testing Equipment

B.1. Mechanical Equipment

The vibration test system was a Mechanical Technologies Inc., LAB, PTV-60, servohydraulic, closed loop controlled, vertical shaker system. This was composed of a 1524mm by 1524mm vibration table, a hydrostatic bearing hydraulic actuator with servo valve and LVDT, a 5500Kg solid steel seismic base, a 22.4 kW hydraulic electric water cooled power supply, a hydraulic manifold, and an air bag isolation system.

The vibration table supported the test specimen while the actuator, driven by the hydraulic power unit, vertically excited the test table. The hydraulic manifold smoothed the operation of the actuator. The seismic base stabilized the actuator and table while the air bags isolated the system from the floor. This system had the capacity to vertically excite a 11kN payload at 5.6g peak acceleration level from 5 to 300Hz. Maximum displacement of the vibration table for this apparatus was 762mm.

B.2. Closed Loop Controls

The electronic equipment required to control the mechanical system was a servocontroller, an automatic pump controller, a sine wave vibration controller, an automatic frequency sweeper, an automatic level controller, a digital vibration monitor, and a control accelerometer.

The table's control accelerometer and the actuator's LVDT are the variables used by the table's controllers. The servocontroller provides a link between the command signal and the hydraulic actuator, while the automatic pump controller monitors and governs the hydraulic power supply. The sine wave vibration controller commands the vibration test as an automatic frequency sweeper controls frequency and displacement sweep rates in either linear or log form. The rate of change in sweep is controlled in octaves per minute, which corresponds to doubling the frequency over a specified period. The automatic level controller maintains a constant peak acceleration level across a vibration sweep and provides the ability to change from acceleration to displacement control. The digital vibration monitor, linked to the control accelerometer, displays peak amplitude, velocity, or displacement of the vibration table. This system's components work together to provide a repeatable vibration test. The sensitivity of the system is ±0.10g.

A key component of the vibration test system is the table's control accelerometer. This device was a PCB Piezotronics, Model 308B, iso-compressive, upright, piezoelectric quartz accelerometer. The accelerometer had a sensitivity of 100mV/g, a resolution of 0.002g, a mounted resonant frequency of 25KHz, and an accurate frequency range from 1 to 3000Hz. The control accelerometer was factory calibrated in compliance with military standard, MIL-STD-45662A. The accelerometer could withstand maximum acceleration levels for vibration and shock of 500g and 5000g, respectively. A block diagram of the vibration table's mechanical and control components is shown in Figure B.1.

B.3 Data Acquisition Equipment

The data acquisition equipment was composed of test accelerometers, vibration monitors, and a data acquisition/plotter system. The test accelerometers were PCB Piezotronic, Model 303A11, quartz, piezoelectric accelerometers that had voltage sensitivities of 100mV/g, a transverse sensitivity of near 3%, a resonant frequency near 70Khz, a peak acceleration range of ±10g and a resolution of ±0.005g. All accelerometers were factory calibrated to military standard MIL-STD-45662 specifications.

The MTI L.A.B., Model 382A, vibration monitors when connected to the test accelerometers were capable of measuring peak acceleration levels from 0 to 40g. Output from these monitors was on a 1V full scale. The vibration monitors could be set on four scales: 0 to 1g, 0 to 4g, 0 to 10g, and 0 to 40g.

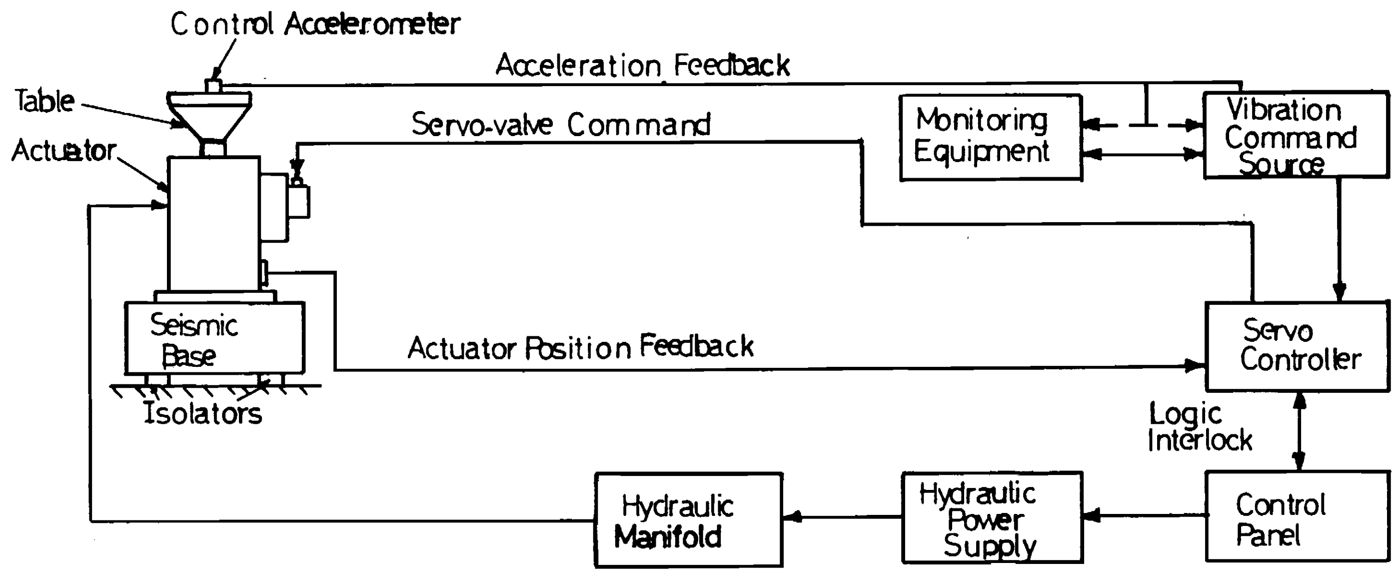

Figure B.1. Block diagram of the components of the vibration test system used in the study of the vibrational characteristics of pallets (after M.T.I. LAB, 1988).

The data acquisition/plotting system was a Hewlet-Packard, Model 7090A, data recorder/plotter capable of simultaneously monitoring and recording three channels of AC or DC current on a buffer with 1000 data points recorded per channel over any time period. The data acquisition system recorded the test vibration monitor's output and the control vibration meter's output, then plotted these with respect to the recorded sweep generator's output. This allowed the peak accelerations of the control and test accelerometers to be plotted versus frequency, generating hard copy graphs that were used to determine resonant responses of the test specimens. A block diagram of the acquisition test apparatus is shown in Figure B.2. A photograph of the vibration system is shown in Figure B.3.

B.4. Equipment Problems

This vibration system was very complex and prone to many installation problems. The equipment was set up by MTI L.A.B. factory technicians. It was soon discovered that the testing equipment was not capable of providing accurate data. The natural frequency of the vibration table interfered with the initial tests performed. Intensive testing showed that the problem was caused by the seismic base that was constructed from laminated concrete within a

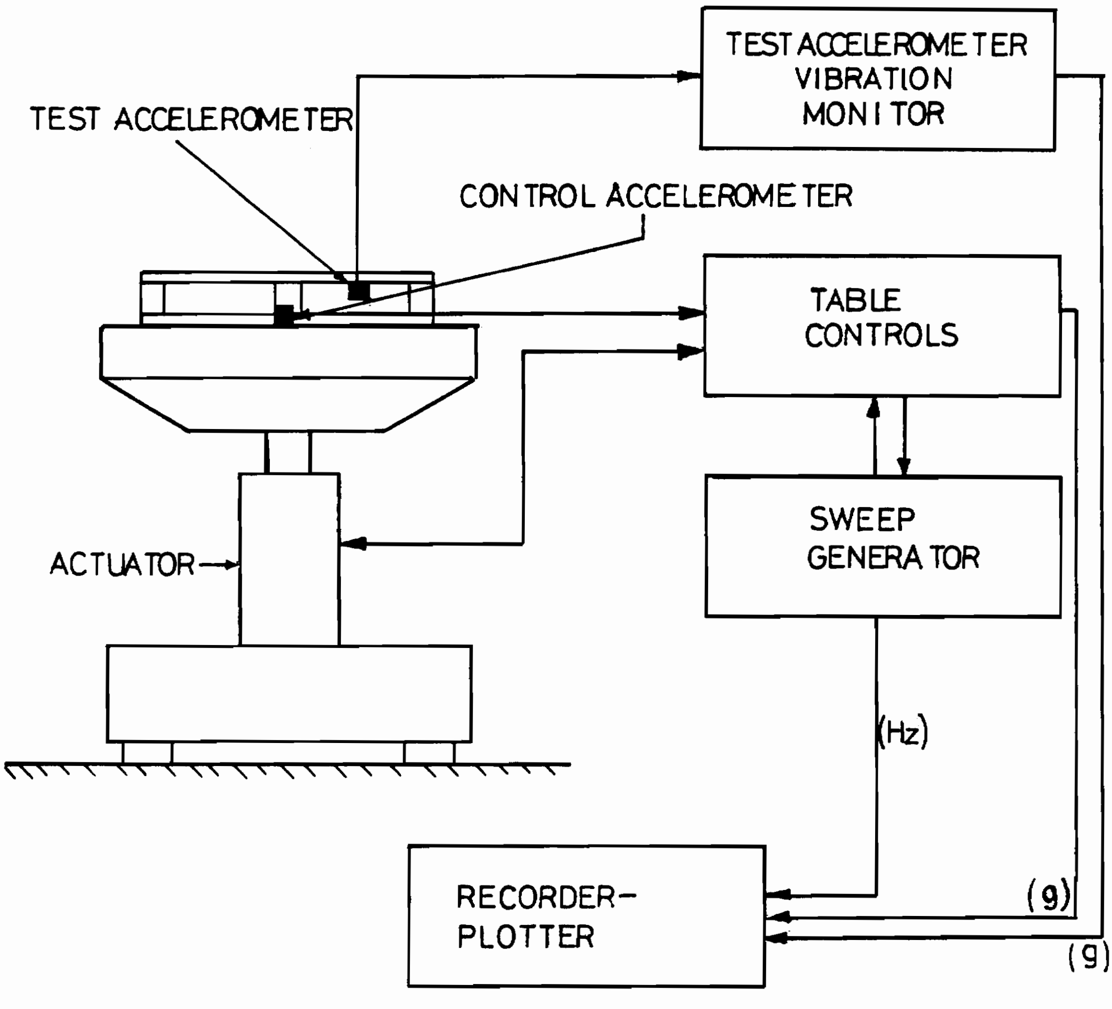

Figure B.2. Block diagram of data acquisition equipment used to determine the resonant response of pallets and pallet sections used in this study.

Figure B.3. A photograph of the vibration testing system used to determine the resonant characteristics of pallet decks.

steel shell. The base was purchased used, and previous shock testing had damaged the concrete laminations. The loose concrete in the shell resonated at frequencies within the frequency range used for testing, causing distortion of the test table response.

A new, solid steel, seismic base was installed. The vibration system supported by the new base was tested and another natural frequency was found at 23Hz. Although this response was not as major as that experienced with the original seismic base, the altered change in the responses of test specimens were significant. The resonant problem was minimized by moving the control accelerometer from its original position beneath the vibration table, to the surface of the table near the test specimen. This phenomena was thought to have been related to the natural frequency of the hydraulic system. The data generated from test specimens with resonant frequencies near 23Hz are therefore questionable.

APPENDIX C

Determination of Deckboard Modulus of Elasticity

C.1. Introduction

All deckboards were tested to determine modulus of elasticity (E) using a nondestructive static bending test. This method was developed by researchers at the Wood Construction and Engineering Laboratory at the Virginia Polytechnic Institute and State University and was found acceptable for the stiffness testing of deckboards (McLeod, 1985).

C.1. Static Bending Test

Deckboard lengths, thickness, and widths were measured and recorded. A moment of inertia (**I**) about the neutral axis was calculated for each deckboard using,

$$I = \frac{b\,h^3}{12} \tag{C.1}$$

Where **b** was deckboard width and **h** was deckboard thickness.

A simple testing procedure was used to measure static deflection of pallet deckboards across a 36in span when loaded with a static center point load. The test apparatus had two cylindrical supports, two load blocks, and a dial indicator. The two load blocks weighed 5 and 25lb, respectively. The dial indicator was accurate to within one thousandth of an inch (0.001in.). A schematic diagram of the device is shown in Figure C.1.

A deckboard was placed on the supports, then the dial indicator was placed beneath the deckboard at center of span and zeroed. The 5lb load block, the preload, was placed above the dial indicator on top of the deckboard. The load was allowed to settle for 1 minute. A deflection measurement was taken from the dial indicator. Next, the 25lb pound weight was placed on top of the 5lb weight. The final deflection measurement was recorded. Both sides of each deckboard were tested. Modulus of elasticity (E) was calculated with the equation:

$$E = \frac{\delta P \, L^3}{48 \, \delta D \, I} \tag{C.2}$$

Where **δP** was the change in load between the first and second load (25lb), **L** was the span between supports (36in), **δD** was the change in deflection from the initial to the final load, and **I** was moment of inertia. The E values for each side of each deckboard were calculated and averaged.

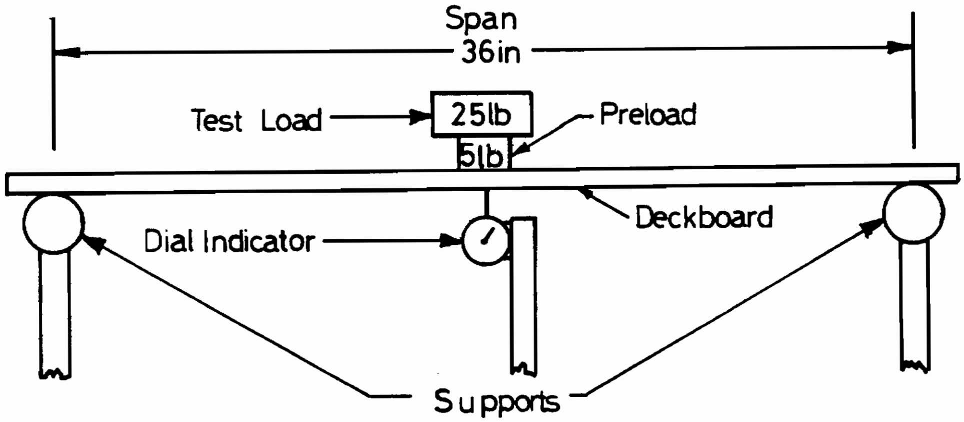

Figure C.1. Static bending test apparatus used to determine modulus of elasticity values for deckboards used in the construction of pallet sections.

Vita

Ira Edwin Lauer III was born November 3, 1965, in York County, Pennsylvania to Mr. and Mrs. Ira E. Lauer Jr. He completed his elementary, intermediate, and high-school education at the Dover Area School District.

After high school graduation in 1983, Ira enrolled at The Pennsylvania State University's York branch campus where he completed his first year and a half of college study. Ira transferred to the Pennsylvania State University's Main campus located in State College, where he earned the degree of Bachelor of Science in Forest Products in December of 1987.

Ira was then employed by a hardwood lumber yard and moulding manufacturer located in south central Pennsylvania where he assumed the responsibilities of lumberyard supervisor, kiln operator, and roughmill foreman.

To further his education, in September of 1989, Ira enrolled in the Wood Science and Forest Products graduate program at the Virginia Polytechnic Institute and State University at Blacksburg Virginia where he worked as a graduate assistant and earned the degree of Master of Science in Forestry and Forest Products.

Ira E. Lauer III

Ira Edwin Lauer III