

A Dynamic Simulation of a Versatile
Model 4025 Swather Header

by

Trever Gordon Crowe

A thesis
presented to the University of Manitoba
in partial fulfilment of the
requirements for the degree of
Master of Science
in
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MODEL 4025 SWATHER HEADER

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TREVER GORDON CROWE

A thesis submitted to the Faculty of Graduate Studies of
the University of Manitoba in partial fulfillment of the requirements
of the degree of

MASTER OF SCIENCE

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Abstract

The purpose of this thesis research was to develop a computer model that would predict the natural frequencies and mode shapes of a Versatile Model 4025 swather header. After the model was created, an experimental modal test was completed. The data from the experiment and the initial model were then compared and the model was adjusted to more closely predict the values obtained by the experiment. The modified computer model was then used to determine an optimal operating speed for the knife drive on the Model 4025 swather header.

The modified computer model predicted the natural frequencies to be 7.6, 9.1, 10.6, and 13.1 Hz while the experimental results showed that the natural frequencies were 7.7, 9.4, 10.5, and 12.5 Hz. In general, the experimental mode shapes were well correlated with the mode shapes that were predicted by the modified finite element model.

The modified model was then used to determine the optimum operating speed. The displacements of the connection points between the swather header and the tractor were predicted because it was thought that the knife speed that caused the lowest displacements of the connection points would cause the least force to be exerted on the swather tractor. The knife should operate near 12 Hz to decrease the displacements of the connection points to 50 percent of the displacement magnitude at the original design speed. With the operating knife speed changed from 10 Hz to 12 Hz, a resonant condition was avoided and the lower force levels applied to the tractor improved the comfort for the operator.

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1. Definitions

Bode plot: a representation of a frequency response function in which magnitude and phase are plotted versus frequency

Coherence function: a frequency-dependent real value that indicates the amount of response that is caused by the supplied input

Cross spectrum: a relation between two random signals calculated by multiplying the Fourier transform of one signal by the complex conjugate of the other signal's Fourier transform

Damping coefficient: a value that indicates the level of damping that is present in the structure at a particular natural frequency

Fourier transform: a mathematical conversion performed on a time-variant signal that identifies the frequency content of the signal

Frequency response function: a complex function which represents the frequency-dependent response of a system per unit input

Frequency spectrum: a frequency domain representation of a time-variant signal

Modal analysis: the process of determining a structure's modal parameters

Modal parameters: three parameters that characterize a natural mode of vibration: natural frequency, mode shape and damping coefficient

Modal testing: the process of determining a structure's modal parameters experimentally

Mode shape: the characteristic vibration pattern assumed by a structure in a resonance

Natural frequency or resonant frequency: the frequency at which resonance occurs

Nyquist frequency or wrap-around frequency: in digital signal processing, a frequency equal to one-half of the sampling frequency

Nyquist plot: a type of frequency-response function display in the Argand plane (imaginary portion versus the real portion)

Power spectrum: a function showing the mean square level of a signal at various frequencies calculated by multiplying the Fourier transform of a signal by its own complex conjugate

Resonance: the condition that occurs when the frequency of a dynamic force applied to a structure is equal to one of the structure's natural frequencies, usually resulting in large amplitudes of vibration

2. Introduction

The methods and machinery used to feed the world have undergone great changes in recent decades. Twenty kilowatt field tractors have come and gone. These have been replaced by tractors with engines in excess of 225 kilowatts. Likewise, other farm machinery has been improved by using more powerful engines, increasing capacity, and providing more operator comfort. The drive for efficiency has been caused by increasing farm size. A single operator must till, sow, and harvest larger areas to provide for family needs. These individuals must then spend long periods of time operating farm machinery regardless of its power, size, or comfort.

The Canada Grains Council (1987) showed that the number of Canadian farms is dwindling. In 35 years (1951 to 1986), the number of farms in Manitoba, for example, decreased from 52 383 to 27 336. Since little previously arable land is being left untilled and because new land has been broken, the average size of the Manitoba farm has increased. The number of people in the agricultural industry has also decreased. The ever increasing hungry world has motivated Canadian farmers to strive for larger yields. These farmers have turned to increased mechanization to assist them in attaining their goal. The increased mechanization has been realized by improvements in material handling and storage equipment, tilling and sowing equipment, and harvesting equipment.

One machine that is especially important in the harvesting process is the swather/windrower. The length of swather units ranges from 3.5 to 22.5 metres while those in the 5 to 9 metre range are most common. The time spent operating these machines may exceed ten hours per day. It is an exception that machines are designed to provide adequate comfort for the operator during long periods of operation. The operating characteristics of such a machine play a large role in determining the comfort that the operator experiences. A dynamic analysis of the machine is required to determine the machine's responses under various operating conditions.

Historically, dynamic analyses have been very complex. To avoid these complexities, engineers usually designed excessively stiff and massive structures. However, today's manufacturing industries look for any possible advantage. The lowest possible structural mass is usually their main goal. With a relatively low mass, less material is required. As well, the cost of transporting their product is lower. The design should ensure that the machinery components cost less, last longer, and are less expensive to operate. These components must carry greater loads, run more quietly, and vibrate less.

A vibration problem was identified with the Versatile Model 4025 swather header when it was attached to the Versatile Model 4750 traction unit. Operators reported that the vibration experienced in the cab was excessive. It made operation of the swather for long periods of time uncomfortable. Mr. Harvey Chorney (pers. com. Sep 1989) stated that the

engineers at Versatile Farm Equipment Operations had located the source of the vibration through a process of elimination. They found that the knife drive was the source of most vibration felt by the operator.

In an attempt to help Versatile Farm Equipment Operations improve its machine, a dynamic analysis of a Model 4025 self-propelled swather header was completed. The objectives of the dynamic analysis were:

1. to create a computer model capable of analyzing the effects of dynamic loading,
2. to perform an experimental modal analysis, the results of which could be used to verify the computer model;
3. to adjust the computer model and achieve reasonable correlation between the analytical data and the experimental results and, finally,
4. to use the computer model to evaluate possible modifications to the structure that may reduce vibration and provide increased operator comfort.

3. Background

3.1. Modal Testing

To fulfil some of their design requirements, engineers have looked to a combination of modal testing and modal analysis. Along with the lowest possible mass, other design constraints may be: a) the minimum value of the fundamental natural frequency b) the upper and lower limits of successive natural frequencies (successive natural frequencies are separated by a required frequency bandwidth) c) the fundamental mode of vibration corresponding to a pre-selected shape. The ultimate goal of the three design constraints listed above is to avoid resonance.

Until the early 1960's, modal testing was called 'resonance testing' and the goal was simply to identify the resonant frequencies. Stroud (1987) indicated that 'twang' testing was very popular in this era. In a twang test, a static load was applied then very quickly removed to set the structure into motion.

Eventually, rotating eccentric mass shakers were developed and the late 1960's saw the introduction of the Fast Fourier Transform (FFT) as a vibration analysis tool. New methods of data acquisition and on-line data analysis were introduced after the advent of the lower-priced minicomputer in the 1970's. For this reason, the 1970's have often been referred to as the digital revolution.

The present trend in modal testing involves efficient data acquisition on minicomputers and then movement of the data to mainframe computers for extensive data manipulation. Also, modal testing and finite element modal analysis techniques are being combined and used in modern Computer Aided Design and Computer Aided Manufacturing (CAD/CAM) computer packages.

Obviously, it would be beneficial to design engineers if the dynamic response of a structure could be predicted. A computer model can be created that will predict such a response. However, an experimental test is more powerful when the validity of the results are considered. Regardless of the method used, Ewins (1987) indicated that healthy scepticism and curiosity in either method is warranted.

When conducting a modal test, the structure is often supported as it is in its service environment. After the specimen has been supported, a known excitation force is applied to the structure at an identifiable location. The response of the structure to this input is measured at various locations on the structure. The units of response measurement may be displacement, velocity, acceleration or strain, for example.

The prime assumption that is associated with a modal analysis is that the test specimen can be treated as a linear elastic system.

The easiest form of analysis to visualize employs an input force which is harmonic in nature (forcing signal is similar to a sinusoidal

waveform). In this case, only one force frequency is represented. The steady-state response of the structure is measured at a number of locations. The frequency of the forcing function is then increased (or decreased), and the measurement process is repeated. This continues until the entire frequency range of interest has been tested.

The ratio of the response signal to the input signal for each of the test frequencies represents the magnitude of the frequency response function. The magnitude of the frequency response function depends on the structural characteristics and the driving frequency. The frequency response function is a complex function that can be represented in a number of ways, a Bode plot, a Nyquist plot (in the Argand plane), or Real and Imaginary versus Frequency plots (these plots will be discussed later).

Peters (1983) defined the Frequency Response Function (FRF) as the ratio of the Fourier transform of the response signal to the Fourier transform of the input signal. This definition applies regardless of the input signal as long as the input signal is Fourier-transformable and the structural system is linear.

Consider an input signal represented by $x(t)$. Newland (1984) indicated that this input's Fourier transform, $X(\omega)$, is calculated as follows:

$$X(\omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} x(t) \cdot e^{-i\omega t} dt \quad 3.1$$

The Fourier transform changes the time-variant expression into a frequency-dependent function. The Fourier transform shows the amplitude and phase that is present at different frequencies. A plot of the Fourier transform of a signal versus the frequency is known as the signal's frequency spectrum.

Three modal parameters completely describe a natural mode of vibration of a linear system. These parameters include the natural frequency, the damping coefficient and the mode shape. The natural modes of vibration are inherent to a system and they are completely defined by the physical properties of the system. Depending on the desired use of the modal test data, all or only one of the modal parameters may be required. The frequency response functions between all response nodes and the forcing location are required in order to define the mode shapes. The natural frequencies and the damping coefficients are global parameters and can be determined by analyzing only one frequency response function.

A useful description was given by Stevens (1985). He said that all three modal parameters can be observed directly. The ringing of a bell was given as an example. Each tone produced by the bell corresponds to a natural frequency. Associated with each tone (natural frequency) is a corresponding mode shape. The duration of the tone is a measure of the modal damping (a longer duration indicates less damping).

Stroud (1987) indicated that modal testing involves four basic tasks: 1) excite the test specimen 2) measure the excitation force and

the specimen's related responses 3) process the measurements (input and output) and create an applicable system description 4) analyze the system description and define the specimen's modal parameters.

3.2. Excitation of Test Specimen

Stroud (1987) indicated that the excitation of a test specimen can be divided into two categories: 1) multi-frequency in which several modes of vibration are excited simultaneously and 2) single frequency in which only one mode of vibration is excited at one time. Each of these different excitation methods will be discussed separately.

3.2.1. Multi-frequency Excitation

When a linear system is subjected to a steady-state harmonic forcing signal, the steady-state response of the system occurs at the same frequency as the forcing signal. It can be expected that a dynamic input that contains a number of different frequencies will cause the structure to respond at the same frequencies. Therefore, to determine the modal parameters of a structure within a frequency bandwidth ω_1 to ω_2 , the input signal should contain the frequency bandwidth at least from ω_1 to ω_2 .

Ewins (1987) stated that a test specimen is affected by all of its modes of vibration and not just those in the range of interest. For this reason, the frequency range of analysis should extend 1.5 to 2.0 times more than the frequency range of interest. This should be used as a rule-of-thumb. More definite limits can be established after considering the

particular test situation. As well, preliminary tests may be warranted to indicate the desirable range of excitation.

3.2.1.1. Ambient Excitation

For large awkward test specimens, an ambient excitation is probably the easiest method of structural excitation. In its service environment, a structure is subjected to certain dynamic forces. The responses of the structure are measured while the structure is subjected to these forces. Examples of ambient excitation may be: wind loads, seismic forces, water wave forces, and passing traffic loads.

The loads that are listed above are all random in nature. A random force signal, as opposed to a periodic force signal, has no repetitive pattern and excites a number of frequencies simultaneously. An ambient excitation need not be random. For example, a machine may have forces exerted upon it by a number of rotating unbalanced shafts. Although the force exerted by one shaft is periodic, not all shafts operate at the same speed. Therefore a number of different frequencies are excited simultaneously.

The method of ambient excitation does have definite advantages. First, no actuation system is required. This is a major saving in capital equipment costs. Secondly, the measured responses are the result of the actual service environment that affects the structure.

However, it is often difficult to determine the magnitude and location of the applied force; therefore, the direct measurement of frequency response functions is usually not possible. A second disadvantage is that there is little control over the applied force. This may cause low response levels. Low response levels require larger volumes of data so that electronic noise problems can be overcome. Ambient excitation is probably least desirable but it may be the only alternative in the case of large stationary structures.

3.2.1.2. Impulse Excitation

The impulse method of excitation provides an input signal that is Fourier-transformable, and it produces a valid frequency response function. Ideally, the supplied impulse load behaves as Dirac's delta function. The ideal impulse load has infinite magnitude and occurs over a duration of zero time. The delta function, $A \cdot \delta(t-t_0)$, is defined such that it is equal to zero everywhere except when it is equal to A when t equals t_0 . The units of A are usually [force]*[time].

Completing the Fourier transform of the proposed input can be achieved by setting the displacement, $x(t)$, equal to the delta function in the Fourier transform as follows:

$$X(\omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} A \cdot \delta(t-t_0) \cdot e^{-i\omega t} dt \quad 3.2$$

The constant A can be removed from within the integral sign. The remaining integral is equivalent to 1.0. This means that the Fourier

transform of a constant multiplied by Dirac's delta function is equal to that same constant divided by 2π ($A/2\pi$).

An ideal impulse force of magnitude A will supply an equal amount of momentum, $A/2\pi$ [mass*velocity], for the frequency interval from $\omega=0$ to $\omega=\infty$ (all frequencies from zero to infinity will be excited by an ideal impulse). The closest approximation to the ideal Dirac delta function could be achieved by impacting the structure with a hammer. Obviously, a hammer impact is not an ideal impulse load. The hammer impact can not be infinite in magnitude and it requires a non-zero time duration to occur.

The use of an instrumented hammer is the most common form of impulse force input. The hammer is instrumented to measure the applied force and the duration that the hammer is in contact with the structure. Instrumented hammers vary in size and hardness of the impacting tip. Larger hammers are able to apply greater forces while softer-faced hammers are able to stay in contact with the test specimen longer. A harder-faced hammer is able to excite higher frequencies and a heavier hammer is able to apply more energy to the system. Choosing the correct hammer mass and face hardness depends on the application. The optimum situation would be to have all the supplied energy distributed equally over only the frequency bandwidth of interest.

Speed is the main advantage of the impulse forcing method. Little set-up time is required, and after the procedure is established, data

acquisition can be completed quickly. During an impulse modal test, it is usually easiest to secure the response measuring device (usually an accelerometer) to one position then to impact the specimen at various other locations. This helps to create the speed advantage because the impact hammer is considerably more mobile than the response measuring device. One further advantage is that, compared to other methods of excitation, the impulse technique is relatively inexpensive.

A major disadvantage of impulse testing is the electronic noise that is encountered when measuring the structure's response. An impact excitation force signal has a very short duration in comparison with the sampling time window. The electronic noise is present during the entire sampling period. Therefore, the signal/noise ratio for the force is low. The noise makes it necessary to complete a number of ensemble averages. These averages take additional time and may have a detrimental effect on the speed advantage. One further disadvantage is that the results depend on the skill of the technician. Using an improper technique while impacting the structure may cause the results to be misleading or invalid. For example, an incorrect technique may cause the hammer to contact the specimen twice and produce incorrect results.

3.2.1.3. Random Excitation

Random excitation is the most widely used form of excitation. Random excitation can be achieved by incorporating a random signal generator (a noise generator) and either an electrodynamic or an electrohydraulic exciter. The signal generator produces a waveform with

randomly varying frequency and amplitude. This waveform is used to "drive" the force applicator. A truly random signal has an equal amount of energy in each finite frequency range.

The use of computer-controlled data acquisition systems is important when an exciter is used. An inline force transducer between the exciter and the structure is used to measure the input force. The supplied force and the acceleration response are sampled digitally. These sampled data sets are manipulated and processed to provide the frequency response functions.

The random signal input process makes nonlinear structures seem linear. This feature of the random signal input technique can be considered an advantage or a disadvantage. It may be a disadvantage in the analysis of an actual nonlinear structure. The results of the test would then be misleading. However, the basic assumption in completing a modal test is that the structure is linear. The random signal input process helps to satisfy the linearity assumption.

The random forcing procedure is relatively fast for a few channels of response. It is also representative of the environments that are frequently encountered in the field. Along with these advantages, there are also disadvantages. Firstly, the random excitation procedure overestimates the damping in nonlinear structures. Secondly, to achieve adequate wideband response levels, relatively larger amounts of input energy are required. Another drawback of the random signal input process

is so-called leakage. Leakage occurs during Fourier analysis of the signal and is caused by the fact that the forcing and response signals are not periodic. Certain data processing procedures can be undertaken to minimize this error. These procedures will be discussed later in subsection 3.4.1.

There are a number of different kinds of random signals. A "true random" signal is a continuous non-repeating signal. A "pseudorandom" signal is a random signal that repeats itself periodically. The "burst random" signal consists of an interval of a random signal that is triggered at the same time or shortly after the initiation of the data acquisition process. The random forcing input is then halted in order that the structural responses halt before data acquisition is complete. The "burst random" process minimizes leakage because the signals are forced to appear "periodic" within a block of sampled data. The response is zero at both the beginning and the end of the sampling period.

3.2.1.4. Fast Sine Sweep or Chirp

The experimental set-up for a fast sine sweep test is similar to the random excitation test. A signal generator is used to drive a mechanical or electrohydraulic actuator. However, the input signal is different. The input signal (force versus time) resembles a simple sine wave that gradually varies in frequency from the lower frequency limit of interest to the upper frequency limit. At all frequencies, the amplitude of the input force is maintained at a constant level. The change in frequency

is gradual, yet the structure is not allowed to achieve a steady-state response.

The swept-sine process is set up so that data acquisition commences as the excitation begins. Assuming a single data collection period of T seconds and a driving force frequency varying from ω_1 to ω_2 Hz, the rate at which the driving force frequency changes can be expressed as $(\omega_2 - \omega_1)/T$ Hz/s. This means that the frequency bandwidth is "swept" within one block of time data.

A clean input spectrum is created and very few ensemble averages are required. This procedure is relatively fast because few ensemble averages are required, and only the frequency band of interest is being excited. One further advantage of this type of input is that the effects of noise and nonlinearities are reduced. A disadvantage of this method is that it does not represent a realistic service environment.

3.2.2. Single Frequency Excitation

The sine-dwell method refers to a test where the shaker is tuned to a natural frequency, and held there while force and response data are measured at all points. The signal generator can be adjusted to produce a sinusoidal forcing wave whose frequency is exactly equal to the structure's natural frequency.

The sine-dwell method procedure is useful if a narrow band of frequencies are to be analyzed. It becomes very inefficient if a broad

band of frequencies is to be analyzed and only a few channels can be analyzed at the same time. One further disadvantage is the possibility of structural damage occurring. Prolonged periods of excitation at a resonant frequency may cause structural damage.

The swept sine and sine dwell procedures have two further advantages. First, nonlinear structures can be tested at different excitation levels. This will reveal the effect of the force level on the structure's dynamic characteristics. Second, a large structure or a structure in a noisy environment require a relatively high level of excitation to achieve a coherent response. Both the swept sine and sine dwell methods allow all the excitation energy to be concentrated at a single frequency or over a narrow frequency bandwidth.

3.3. Measurement of Excitation Forces and Structural Responses

Input force measurement is usually achieved by installing a force transducer inline between the source of the force input and the structure. In the case of impulse testing, a force transducer is attached to the head of the impacting hammer. Similarly, in the case of the electrodynamic or electrohydraulic exciter, a force transducer (load cell) is mounted between the actuator and the structure.

The response of the structure can be measured in a number of ways. An accelerometer is most commonly used. Ideally, a triaxial accelerometer would be used when the structure's response in three directions is

required. With a triaxial accelerometer, the accelerations in all three orthogonal directions can be measured simultaneously (provided that a three-channel signal processing unit is available). Other types of response measurements may include displacement, velocity, or strain.

The input and response measurements can be acquired and saved as time histories. These data sets can then be processed at a later date to create the required spectra. The excitation and response data can also undergo real-time Fourier analysis. In this case, the excitation and response signals are Fourier-transformed as they are collected.

3.4. Measured Signal Processing

Raw input and response signals are analog. The computer-controlled data acquisition system monitors these analog signals and samples a single data point at evenly spaced time intervals. The sampled data are converted from analog to digital form and then they are stored in a data buffer.

Data buffers are situated between the transducers and the computer. They may be capable of holding more than 1024 data points. If a buffer is capable of holding n data points and the time between successive data sample points is t seconds, a buffer is filled every $(n-1)*t$ seconds. These buffers are intermediate storage locations for data. After a buffer is filled, the data in that buffer are processed, then stored in memory, and finally erased before another set of data is collected.

The process of digital sampling can be plagued by two problems:

- 1) aliasing is caused by the finite sampling interval, and
- 2) leakage is caused by the finite length of the observation window (finite number of data points in the buffer).

Setting the rate at which data are sampled is a very important decision. If the sampling period (time between samples) is represented as Δt , the sampling frequency is $1/\Delta t$. In order not to lose any required information, the sampling frequency should be chosen correctly. Van Brussel (1983) quotes Shannon's Theorem to solve such a problem. Shannon's Theorem states that, if the maximum frequency to be detected in a signal is $F_{\max}$ (the Nyquist frequency), the sampling frequency should be at least $2 \cdot F_{\max}$.

If the sampling frequency is too low to identify all frequencies that are present in the analog signal, the energy present at frequencies above the Nyquist frequency will be represented as frequencies below the Nyquist frequency. This causes an error known as aliasing. These false energy levels at frequencies below the Nyquist frequency can be removed from the digitized signal by installing an anti-aliasing filter between the analog source and the analog-to-digital converter. The anti-aliasing filter simply removes the signals that have frequencies above the Nyquist frequency.

The transfer of energy from one frequency to its adjacent spectral lines is known as leakage (energy 'leaks' from one frequency into

another). Leakage is caused by one of the assumptions related to digital analysis of a signal. The digital Fourier transform algorithms assume that the sampled data are periodic and that the period of these data is equal to the length of the buffer window. This means that the difference in magnitude between the value of the buffer's first data point and its last data point should be zero. Especially in the case of a random signal, this assumption is very difficult to satisfy unless the sampled buffer is factored by a weighting function.

3.4.1. Windowing

The multiplication of the sample buffer by a weighting function is a process known as windowing. Less importance is placed on the data at the beginning and the end of the data buffer while the weighting value at the middle of the data buffer is 1.0. Since the weighting factors progress from 0.0 to 1.0 then back to 0.0, the windowed data have less energy than the original data buffer.

De Silva (1986) explains the use of a scaling factor. The windowed data are multiplied by a scaling factor greater than 1.0 to account for the energy that was previously lost in the windowing process. There are a number of different types of windows that can be used. De Silva (1986) listed some common windows, shown in Table 3.1. In this table, the variable "T" represents the time period of the buffer.

Table 3.1: Common windows and their scaling factors.

<u>Window</u>	<u>Describing Equation</u>	<u>Scaling Factor</u>
rectangular	$\omega(t)=1.00$ for $t=0$ to T $\omega(t)=0$ elsewhere	1.00
flat-top cosine	$\omega(t)=0.50(1-\cos(10\pi t/T))$ for $t=0$ to $T/10$ and $t=9T/10$ to T $\omega(t)=1.00$ for $t=T/10$ to $9T/10$ $\omega(t)=0$ elsewhere	1.07
Bartlett (triangular)	$\omega(t)=2t/T$ for $t=0$ to $T/2$ $\omega(t)=-2t/T$ for $t=T/2$ to T $\omega(t)=0$ elsewhere	1.73
Hanning	$\omega(t)=0.80+0.46*(\cos(2\pi t/T))$ for $t=0$ to T $\omega(t)=0$ elsewhere	1.59
Hanning (cosine)	$\omega(t)=0.50*(1-\cos(2\pi t/T))$ for $t=0$ to T $\omega(t)=0$ elsewhere	1.63
Parzen	$\omega(t)=1-6*(2t/T-1)^2+6* 2t/T-1 ^3$ for $t=T/4$ to $3T/4$ $\omega(t)=2.00*(1- 2t/T-1)^3$ for $t=0$ to $T/4$ and $t=3T/4$ to T $\omega(t)=0.0$ elsewhere	2.10

Each window type described in Table 3.1 should be used to process specific types of data. De Silva (1986) also listed the signal type upon which each window works best. The appropriate uses are listed in Table 3.2.

Table 3.2: Guidelines for appropriate window selection.

<u>Signal Type</u>	<u>Window</u>
periodic with period= T rapid transients	rectangular
periodic with period= T quasi-periodic slow transients	flat-top cosine
beat-like signals with period $\sim T$	Bartlett (triangular)
narrow-band random lightly damped systems	Hanning (cosine)
broad-band random	Parzen
heavily damped systems	Hanning

3.4.2. Frequency Response Function Calculation

After the data have been windowed, they must be processed by a digital Fourier transform algorithm. The result of such a transformation is the frequency spectrum of the digitally sampled signal. After both the forcing signal and the response signal have been Fourier-transformed, the frequency response function can be calculated. Exactly as the definition states, the frequency response function can be calculated by dividing the Fourier-transformed response signal by the Fourier-transformed input signal. However, Halvorsen and Brown (1977) stated that it is beneficial if the frequency response function is calculated by dividing the cross spectrum between the input and output signals by the power spectrum of the input.

The cross spectrum between a signal $x(t)$ and a signal $y(t)$ is calculated by multiplying the Fourier transform of $x(t)$ with the complex conjugate transform of $y(t)$. Similarly, the power spectrum of a signal $x(t)$ can be calculated by multiplying the Fourier transform of $x(t)$ by its own complex conjugate pair.

Halvorsen and Brown (1977) indicated that using cross spectra and power spectra help to alleviate some of the noise problems that may be encountered. Also, if the frequency response function is calculated by using cross spectra and power spectra, the calculation of the coherence function is simplified.

The coherence coefficient is a real number which indicates the amount of response that is "coherent" with or caused by the measured input and is a function of frequency. This coefficient is usually represented as $\gamma_{xy}^2(f)$. It is calculated by dividing the square of the cross spectral density magnitude by the product of the two power spectra.

$$\text{that is, } \gamma_{xy}^2(f) = \frac{|S_{xy}(f)|^2}{S_x(f) * S_y(f)} \quad 0 < \gamma_{xy}^2 < 1 \quad 3.3$$

where: $S_{xy}(f)$ =cross spectrum of signals x and y
 $S_x(f)$ =power spectrum of x
 $S_y(f)$ =power spectrum of signal y

The coherence coefficient may vary between 0.0 and 1.0. A coherence of zero indicates that the input and response signals are completely uncorrelated. Halvorsen and Brown (1977) stated that a coherence

coefficient less than unity is mostly caused by the presence of electronic noise and nonlinearity in the structure. Since electronic noise is random, the coherence usually improves as the number of data sampling averages increases. An acceptable range for the coherence coefficient is usually 0.9 to 1.0.

3.4.3. Frequency Response Function Display

The frequency response function is complex (has magnitude and phase). As mentioned earlier, there are a number of ways that the frequency response function can be displayed. In each type of graphical representation, the presence of a natural frequency is detected in different ways.

The most common form of representation is the Bode plot. Both the magnitude and the phase of the frequency response function are plotted versus the log of the frequency. In such a plot, resonant frequencies appear as peaks in the magnitude versus frequency curve. At the same time, the phase shifts 180 degrees. An indication of the amount of damping present in the structure can also be obtained from the plot. The frequency bandwidth that is traversed from the beginning of the phase shift to the end indicates the level of damping. Wider bandwidths indicate a relatively higher level of damping.

Another method used to graphically represent a frequency response function is the Nyquist plot. The imaginary portion of the complex frequency response function is plotted versus the real portion of the same

function. In the case of a structure with little damping and well separated natural frequencies, a resonant frequency appears nearly as a circular arc. This type of plot is often used when completing a curve-fitting analysis of a set of frequency response data.

A third method of displaying the frequency response data is by plotting both the real and imaginary portions of the frequency response function versus frequency. The resonant frequencies appear as peaks in the imaginary plot while the real portion passes through zero (switches from a positive value to a negative value).

3.4.4. Mode Shape Calculation

Consider a structure that can be represented by N different points in space. To complete an experimental modal analysis of this structure, the input excitation could be applied to any one of the points that define the structure. In order to define the natural mode shapes, the response measurements must be performed at all N points, including the point of force application. In this way, a single row of the complex frequency response matrix is measured (rows indicate excitation points while columns indicate points of measurement). In the case of a stationary accelerometer and an impact hammer that supplies an input force at different locations, a single column of the complex frequency response matrix is measured. Because the matrix is symmetric, either a single column or a single row contains all the modal information for the structure.

A complete transfer function matrix then consists of $N \times N$ entries and it can relate forces that may be applied at any or all of the points to displacements at those same points. This set of transfer functions can be represented in the following matrix notation.

$$\{F_j(\omega)\} * [H_{ij}(\omega)] = \{X_i(\omega)\} \quad 3.4$$

where: $F_j(\omega) =$ force applied at j^{th} point
where $j = 1$ to N

$H_{ij}(\omega) =$ transfer function of a response measured at the i^{th} point while the force was supplied at the j^{th} point in which
 $i = 1$ to N and $j = 1$ to N

$X_i(\omega) =$ response of structure measured at point i . $i = 1$ to N .

In order to extract the modal parameters (natural frequencies, damping coefficients, and mode shapes) a curve-fitting process must be completed. Stroud (1987) described two common types of curve-fitting algorithms. These two algorithms are the complex exponential algorithm (CEA) and the polyreference technique.

The complex exponential algorithm is based on the theory that a frequency response function can be expressed as a polynomial of complex exponentials. The constant coefficients of the polynomial are determined by evaluating the equation equal to a measured frequency response function. The roots of this equation are complex values and identify the natural frequencies and relative damping for each mode of vibration. The amplitude at each natural frequency gives a relative mode shape coefficient. The amplitude is calculated by evaluating the square root of the summed real and imaginary squares.

The damping coefficients and natural frequencies are global values, (for a given mode of vibration, they are the same at all points on the structure). Therefore, only one frequency response function must be analyzed to determine them. However, for every point on the structure at which a mode shape coefficient is required, a separate frequency response function must be measured.

Consider a structure that was characterized by exciting it at one location and completing 50 response measurements. If three natural frequencies were detected during the experiment, 150 different mode shape coefficients, that is 3×50 , would result. The coefficients at each particular natural frequency establish the associated mode shape of the structure. Often these mode shapes are normalized with respect to the largest coefficient. A graphic display of a mode shape indicates only relative motion among the members of the structure. Therefore, the actual amplitude of displacement is not known.

The polyreference technique uses the same approach as the complex exponential algorithm. The polyreference technique is more powerful because it can analyze test data that were created using more than one excitation source. It is also faster because all frequency response measurements are considered simultaneously. In the case of only one excitation source, the analysis reduces to the complex exponential algorithm. The difference occurs when the global values, natural frequencies and damping coefficients, are determined. Each global value

is determined by using a least squares approximation of all frequency response functions.

The basic requirement of mode shape calculation is that the frequency response function can be characterized by a polynomial equation. Ewins (1987) warned that a "best-fit" curve does not guarantee a good fit and a good fit does not promise valid modal parameters. Problems arise when a nonlinear structure is characterized as being linear. In such a case, certain fictitious modes of vibration are created to achieve a "good fit". These modes have no physical meaning and only improve the cosmetics of the fitted curve. An analytical modal analysis will predict a structure's natural frequencies within a certain bandwidth. Therefore, a preliminary analytical modal analysis is useful when differentiating between the true and fictitious modes that are determined during an experimental modal test.

3.5. Modal Analysis

Normally, structures are continuous. A continuous system has an infinite number of degrees of freedom. Any analytical model must have a finite number of degrees of freedom. Therefore, a model that has a finite number of degrees of freedom will be only an approximation of the actual system. A system with N degrees of freedom can have N possible vibration modes. The approximation improves if the number of degrees of freedom in the model is large relative to the order of the highest mode being analyzed. The trade-off occurs when time and analysis capabilities are

considered. The required precision of the model will dictate the number of degrees of freedom that are required.

A modal analysis using the finite element method involves the solution of a set of second-order differential equations. The set of second-order differential equations is derived by dividing the structure into a finite number of discrete elements. The equation of motion for each of these discrete elements is included in a set that describes the total structure. Expressed in matrix form, the set of N equations of motion can be described as below:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{f(t)\} \quad 3.5$$

where: $[M]$ = discrete mass matrix (N*N)
 $\{x\}$ = displacement of mass element (N*1)
 $[C]$ = matrix of damping coefficients (N*N)
 $[K]$ = stiffness matrix (N*N)
 $\{f(t)\}$ = force matrix (N*1)

Initially, the matrix equations are solved for free vibrations assuming that no damping is present. Therefore, the damping and forcing matrices are equated to zero and the set of equations simplifies to:

$$[M]\{\ddot{x}\} + [K]\{x\} = 0 \quad 3.6$$

For harmonic motions we assume that

$$x(t) = e^{-i\omega t} \quad 3.7$$

Differentiating $x(t)$ twice, the acceleration can be expressed as

$$\ddot{x}(t) = -\omega^2 \cdot e^{-i\omega t} \quad 3.8$$

By substituting equation 3.7 into equation 3.8, the acceleration can be expressed as

$$\ddot{x}(t) = -\omega^2 \cdot x(t) \quad 3.9$$

where ω = the square root of the natural frequency

Substituting this into equation 3.6 leaves

$$([K] - \omega^2 [M]) \{x\} = 0 \quad 3.10$$

The non-trivial solutions of the set of equations in 3.10 are determined by equating the determinant of $([K] - \omega^2 [M])$ to zero. These solutions, the eigenvalues of the set of equations, identify the natural frequencies of the system.

After the natural frequencies have been found, they can then be inserted individually back into the equations of motion to determine the mode shapes. The mode shapes are evaluated by inserting the natural frequency as the value for ω^2 in equation 3.10. The column vector $\{x\}$, the eigenvector, then identifies the mode shape associated with that particular natural frequency. The mode shapes are in fact eigenvectors and, therefore, they have no absolute magnitude.

Many computer routines exist that can complete an eigenvalue and eigenvector analysis. However, the most important part of an analytical modal analysis is the structural definition. The computer routine is only as accurate as the model that was created, and Schmidtberg and Pal (1986) confirmed that the model's accuracy is often a function of the analyst's abilities. The solutions of an analytical modal analysis should be considered only approximate because the model can never be exactly the same as the actual system. The system parameters at the first (lowest) natural frequency are always most accurate.

There are two reasons that indicate why a model can never be exactly correct. Firstly, as described earlier, a continuous system is being approximated by a simplified version. Secondly, the assumption of no damping is often not satisfied. Damping is present in all designs. However, its impact is not known. Therefore, damping is assumed to be negligible when completing a modal analysis.

It is reasonable to neglect damping from a finite element model. For most structural models, the undamped natural frequencies and mode shapes are very similar to the natural frequencies and mode shapes that are obtained from a model that includes damping.

The computer system that is used to complete a modal analysis of a structure created with beam-type elements requires certain information. This includes spatial coordinates, cross-sectional properties (area and

moments of inertia), and material properties (density, Poisson's ratio, shear modulus) for each structural member.

The computer system can automatically divide the structural elements into finite elements for the analysis. The information about these elements can then be used to create the mass and stiffness matrices as required.

There are a number of advantages that are associated with a modal analysis. First, Prats (1986) indicated that a number of hardware design iterations could be avoided if a dynamic model was created and studied. The development of smaller less expensive computers has increased the number of industries that use finite element design procedures. Finite element analysis packages are usually capable of analyzing static and dynamic loading of three-dimensional structures. Various element shapes have allowed users with little experience to create simple yet effective structural models. Often an accurate dynamic model of a structure indicates the locations of stress concentrations and can help to create an effective solution.

Blakely et al. (1986) stated another beneficial use for a finite element structural analysis. Completed prior to experimental analysis, a finite element modal analysis provides plenty of useful information. First, the optimum locations of excitation and response measurement can be determined by studying the predicted mode shapes.

Excitation should not be supplied at a point on the structure which is static during a natural mode of vibration. If such a point is driven, the characteristics of the structure are altered and the experimental results are incorrect. The optimum excitation location has a reasonably large relative displacement in the same direction in all natural modes of vibration. If an optimal excitation location is found, the excitation equipment need not be moved from one point to another when a different mode is to be excited.

The locations of response measurement should be chosen such that large response amplitudes are measured. This will achieve a large signal-to-noise ratio and help to overcome some of the difficulties created by electronic noise. The ultimate goal of a finite element pre-test is to predict the structural characteristics accurately and to avoid unwanted surprises during the experimental stage.

A finite element modal analysis has other definite advantages. There are a large number of finite element packages that aid in the creation of structural models. Less expensive yet more powerful computers allow harmonic, transient, random and temperature loading. With such capabilities, various loading configurations can be simulated and the model limitations depend only on the analyst's creativity.

A number of disadvantages are also associated with the finite element modal analysis procedure. First, the main assumption behind modal analyses is that the structure is linear. Large nonlinearities in the

system may invalidate results. Rieger (1986) listed three sources of structural nonlinearities. Nonlinearities may result from the presence of nonlinear materials. These may include composite structures, viscoelastic and elastic plastic materials. Large deflections of elements may result in plastic deformations. Obviously, plastic deformations destroy the initial assumption of linearity. Finally, Rieger (1986) indicated that nonlinearities can be introduced by the boundary conditions.

Halvorsen and Brown (1977) listed a number of other sources of nonlinearity. Nonlinearities are often caused by clearances between parts. Prime examples of such cases are represented in gear systems and bearing-mounted shafts. Nonlinearity can also be introduced as a result of certain types of damping. This damping is associated with joints whose connected members undergo relative motion. Finally, members whose stiffness varies with the applied load are highly nonlinear. Nonlinearities complicate the analysis and tend to cause discrepancies between the experimental and analytical values. However, the analyst must be aware of the sources of nonlinearity and make appropriate adjustments to minimize their effects.

The best way to deal with nonlinearities caused by clearances between structural members is to apply a preload that will take up the clearances. The preload must be selected carefully so that it does not change the boundary conditions. Often, the preload is applied through a very flexible spring.

3.6. Mathematical Data versus Experimental Data

Although mathematical models can predict many design values, the accuracy of such a prediction should always be questioned until the model has been verified by experimental results. The use of natural frequencies and associated mode shapes is most common when comparing analytical and experimental results. Other comparisons may include measured and predicted responses of the structure under loading conditions that the structure experiences in its service environment.

In a comparison of the predicted and measured-forced responses, a force must be simulated in the analytical model. The acceleration at a particular point on the structure can then be predicted and compared to the response that was determined experimentally. Again, assuming that the system is linear, the magnitude of the simulated load is not important because the response magnitude is directly proportional to the force's magnitude.

While comparing the two sets of data, it is assumed that the experimental results are correct and the analytical model must then be adjusted to more closely simulate the actual structure. Stroud (1987) stated that model refinement is often based on a trial-and-error approach and is often laborious and time-consuming. On the other hand, Ewins (1984) stated that the source of model error can sometimes be determined by a simple comparison of the natural frequencies and mode shapes.

After the computer model has been verified, it can then be used to evaluate design modifications. Design modifications can be attempted to reduce unwanted vibration or noise that is felt by an operator, for example. The structure can be modified to change the natural frequencies of the structure. The goal of such an exercise is to fix the structure's natural frequencies at values different than the forcing frequencies in the structure's operating environment. Thus, resonance is avoided.

Another way to avoid resonance is to change the driving frequencies that are experienced in the structure's service environment. This is a viable solution in the case of relatively simple machines with few dynamic inputs. Although this exercise is simpler and more economically attractive compared to structural modifications, it does have disadvantages. In a more complex machine, changing the speed to limit the effect of one dynamic load may cause another dynamic input to enter a resonant zone.

There are a number of different methods that can be exercised to decrease unwanted vibration or noise:

- 1) remove or reduce excitation levels (e.g. balancing)
- 2) change frequency of excitation (e.g. speed of operation)
- 3) stiffen the structure or add mass to move the structure's natural frequency away from the excitation frequency
- 4) isolate the source of vibration from the receiver
- 5) increase damping.

3.7. Effects of Whole-body Vibration

When a human body experiences vibration, it is displaced from its original stationary position. Therefore, displacement is one quantitative parameter which can be used to describe vibration. Two other quantitative parameters that can be used are velocity and acceleration. Velocity is the first derivative (with respect to time) of displacement and acceleration is the second derivative (with respect to time) of displacement. The time history of a displacement record has the same shape and period as a time history for either velocity or acceleration. The difference occurs in the phase shift. A velocity history leads the displacement history by 90 degrees and an acceleration history lags (or leads) the displacement signal by 180 degrees.

Vibration, as it affects the human body, can be divided into two categories: whole-body vibration and segmental vibration. Segmental vibration involves the effect of vibration on parts of the human anatomy. For example, segmental vibration work may consider the effects of vibration on an individual's hands and fingers. In the forestry industry, researchers continue to search for a chain saw that transmits less vibration to the operator's hands.

The effects of whole-body vibration are felt most by operators of large machines. These individuals include truck drivers, heavy duty construction workers, and farmers. In all the cases listed above, the individual is seated for long periods of time (12-20 h/day). However,

whole-body vibration can also affect individuals who must stand on a vibrating surface.

Sanders and McCormick (1987) described different physiological effects due to vibration. According to them there are two categories of effects: the effects of short-term vibration and the effects of prolonged periods of vibration.

Short periods of vibration cause a slight degree of hyperventilation and an increased heart rate in the early stages of vibration. There are no significant changes in the chemical composition of any body fluids or body matter.

Prolonged periods of vibration cause above normal occurrences of spinal disorders, hemorrhoids, hernias, and digestive and urinary problems. Because the preponderance of the research information was collected from truck drivers and equipment operators, McCormick and Sanders (1987) found it difficult to separate the effects of vibration from the effects of sitting for long periods of time. However, they stated with certainty that vibration causes muscle tension. A person subjected to vibration strains muscles in an attempt to dampen the vibration and, thus, causes muscle tension and fatigue.

Vibration affects the human body in different ways depending on the direction of application, the vibration amplitude and the vibrating frequency. Wasserman (1987) stated that a human body (like any other

structure) is most susceptible to ill-effects of vibration if the frequency of vibration is equal to or nearly equal to a resonant frequency.

Different parts of a human body have different natural frequencies. For example, Wasserman (1987) stated that the vertical resonant frequency of the human upper body occurs in the 4-8 Hz range. Resonance in the horizontal and lateral directions occurs in the 1-2 Hz range. The described ranges of resonant frequencies are relatively wide to accommodate different individual builds. Some people may be slim and have very little muscle development while others may be very muscular and stout.

Other individual body parts have unique resonant frequencies. This indicates that different body parts do not vibrate in unison. A human's head and shoulders have a resonant frequency in the 20 to 30 Hz range while the eyeballs resonate in the range of 60 to 80 Hz.

The resonant frequencies of a human body vary over such a wide frequency band (from 1 Hz to 80 Hz), and it is very difficult for individuals to avoid exposure to a resonant frequency. The auxiliary approach to safety is to limit the exposure time. The International Standards Organization (1978) has produced a standard that outlines the procedures that should be followed when measuring the levels of vibration to which a human is subjected. In this publication (ISO 2631, Guide for The Evaluation of Human Exposure to Whole Body Vibration), the time

permitted to avoid reduced comfort, fatigue, and health endangerment is shown graphically as a function of frequency.

The Guide for The Evaluation of Human Exposure to Whole Body Vibration, ISO 2631 (1978), provides exposure times that cause first reduced comfort, second, decreased proficiency caused by fatigue and, finally, health endangerment. These three ranges can be shown graphically by three different families of curves that are plotted on the same cartesian plane. Each curve in the family establishes a maximum permissible exposure time, usually expressed in hours. The vertical axis of the graph contains the RMS acceleration while the horizontal axis lists the frequencies.

The ISO Standards for human vibration measurement specify that acceleration be measured. A number of other quantities can be used to describe the same vibration history. The peak value is the maximum instantaneous acceleration measured. It can be used to indicate the magnitude of short duration shocks. The RMS acceleration of a signal is a constant acceleration level that would provide the same amount of energy as the original acceleration signal. The crest factor relates the two previously described parameters. The crest factor is the ratio between the peak value and the RMS value. A highly impulsive signal creates a relatively large crest factor. Impulsive vibrations are considered to be more dangerous than the non-impulsive therefore. Therefore, the crest factor is a good indicator of the harmful content of a vibration signal.

The RMS acceleration of each family of curves (the y-axis variable) is related by standard ratios. The RMS acceleration for the family of curves that describes decreased proficiency due to fatigue can be multiplied by two to produce the health exposure limits. Similarly, the reduced comfort curves can be produced by dividing the fatigue-decreased proficiency curves by 3.15. The three families are related. Therefore, only the fatigue-decreased proficiency curves are plotted. As shown in Figure 3.1, the curves resemble troughs. They have a negative slope in the low frequency range (0 to 4 Hz), then a flat region in the vicinity of 5 Hz, followed by a positive slope in the frequency range beyond 8 Hz.

A work practice guide (supplement to ISO 2631) provides hints that may decrease the effects of vibration on human workers. First, a worker should stay in a vibrating environment only long enough to complete the task required. Wherever possible, machine controls should be located the furthest possible distance from the source of vibration. This will help controllers to avoid exposure to vibration. The most effective exercise to protect equipment operators is to mechanically isolate the operator's housing. Also, preventive maintenance should be established to maintain or decrease the level of vibration that is created.

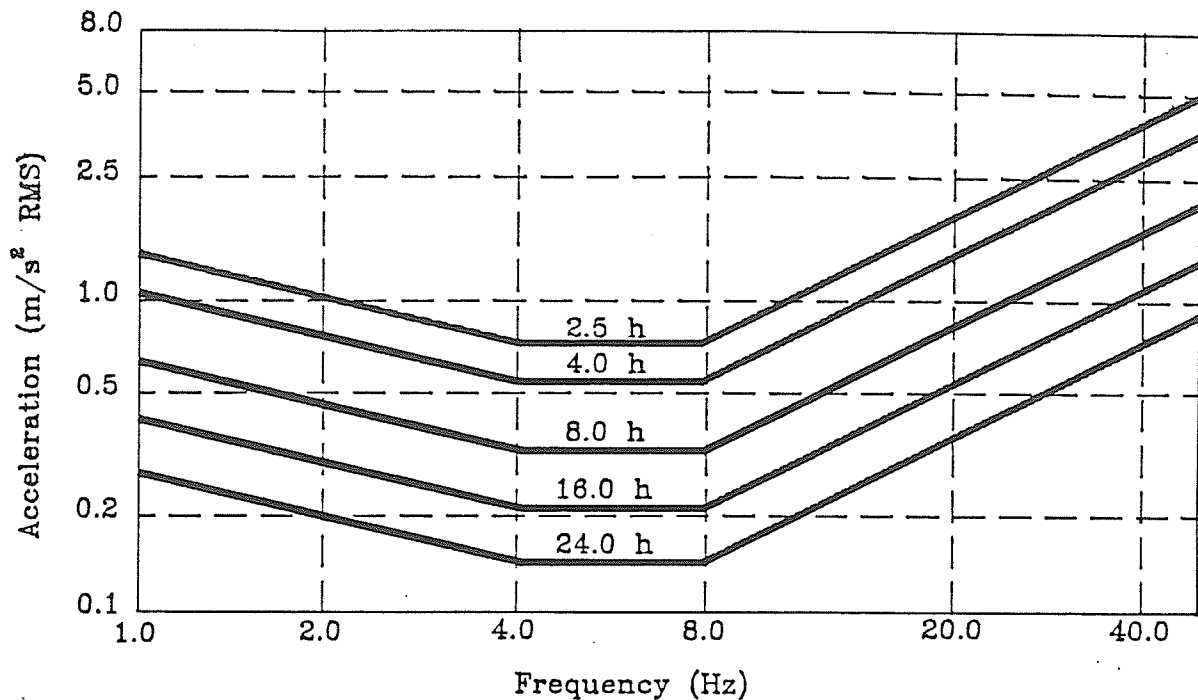


Figure 3.1: Vibration limits to avoid fatigue-decreased proficiency.
(vibration felt in the 'foot-to-head' direction)

The study of subjective responses to vibration is extremely difficult. An individual's comfort is a non-quantitative variable and the difficulty arises when one individual is compared to or grouped with another. Sanders and McCormick (1987) stated that data differ from one study to another because different methodologies, populations and vibration environments are used. The lack of consistent data has created hardships for the ISO.

The publications that were produced regarding the effect of vibration on humans have been criticized. Sanders and McCormick (1987) indicated some areas of criticism that the ISO has received. It is

thought that the time limits for short time exposure in the decreased comfort and decreased proficiency ranges may be excessively high. Another criticism is that the limits do not account for population variability. Also, there is no indicated difference between men and women, for example. Other observers state that the effect of vibration on workers is oversimplified because all three families of curves have the same shape. Finally, the basic criticism is that there is very little information that supports the ISO's limits.

3.8. Summary

The literature cited in this review indicates that the dynamic analysis stage of machine design is indeed a complex matter. Complex or otherwise, a dynamic analysis is helpful to provide necessary information for design engineers.

The results obtained from a modal analysis should always be questioned. The validity of a computer model can only be proven by experimental results. Often computer models that predict dynamic responses can be validated by performing experimental modal analyses. Natural frequencies, mode shapes, and predicted responses are most often used to compare the analytical and experimental results. If discrepancies occur, the computer model must be adjusted to produce results that more closely resemble the experimental values.

In this age of digital technology, the computer has become an important design tool. The ability to predict how a structure will behave in its operating environment before production of the structure begins provides a great advantage for the design engineer. Design iterations performed on a computer save time and money and eventually allow the production of a better product.

Specific guidelines have been established that outline the safe vibration levels to which a human being should be exposed. Machinery designers and manufacturers must be aware of the time that is spent

operating such machines and the vibration levels that operators must endure. Decreasing the vibration felt by the operator would provide increased comfort and safety.

4. Analytical Program

4.1. Introduction

A commercially available computer package was used to create and analyze a finite element model of the swather frame. Version 4.0 of the Integrated Design Engineering Analysis Software (I-DEAS) was used to predict the natural frequencies and mode shapes of the swather. A number of different sub-programs was used to prepare the model, analyze it, and process the results. The entire software package was created by the Structural Dynamics Research Corporation (SDRC) of Cincinnati, Ohio. This program was executed on the computer system at the Industrial Technology Centre, a division of the Manitoba Research Council.

Model creation was simple and easy to accomplish. First, a 'stick model' as shown in Figure 4.1 was sketched that outlined all major structural members of the swather header. After a suitable replica had been drawn, the coordinate origin was chosen. The location of the origin was arbitrary, although it may have been beneficial to locate the origin so that the entire model was in the first quadrant. If the model is in the first quadrant, all nodes have entirely positive coordinates. In this study, the origin was situated half way between each end of the swather at the bottom rear of the header. From the position of the operator, the positive X axis extended from right to left, the positive Y axis pointed vertically upward and the positive Z axis stretched horizontally forward.

To keep the directions consistent, the swather will be described as it is seen from the operator's seat.

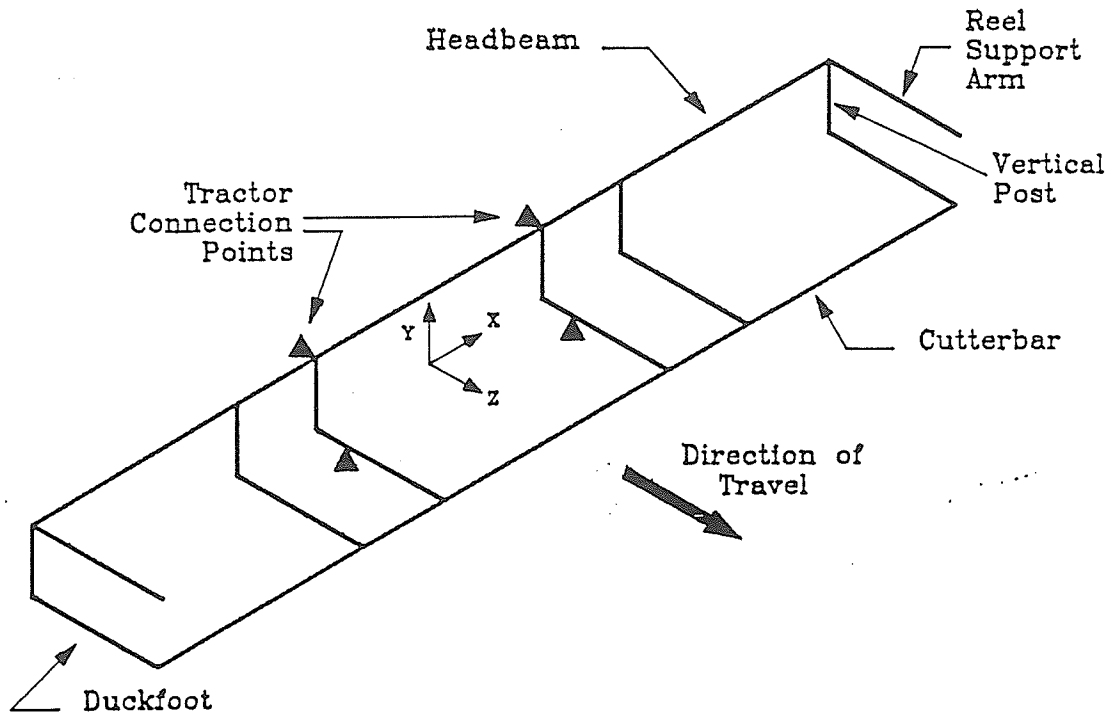


Figure 4.1: Preliminary sketch of test specimen.

The ends of different structural beams were defined through the use of nodes. Nodes were located to identify abrupt changes in beam cross-sections, the intersections of two or more beams, or points that are to be restrained to ground.

During the member's creation, I-DEAS required input for the physical and material properties of each member. The physical properties and material properties for each member were kept separate by entering them into two different tables. The physical properties table contained the cross-sectional area, principal moments of inertia, shear area ratio,

torsional constant, warping constant, degrees of fixity, and eccentricity. These values were not required for rigid beam elements.

The material property tables required different input values depending on the analysis that was to be completed. A modal analysis required that Young's modulus, Poisson's ratio and the density be supplied for each beam.

The modal parameters are independent of the load that is applied to the structure. Therefore, no external load must be applied. An external load must be applied only in the calculation of displacements or stresses.

4.2. Model Development

The swather header was geometrically symmetric. However, the mass distribution was not. Therefore, the entire structure was analyzed to simulate its mass eccentricity. I-DEAS possesses an efficient reflection option. This allows a structure to be reflected about any desired plane. This option was used in the creation of the swather header model. One-half of the structure to the left of the origin was completely modelled then it was reflected about the Y-Z plane at X equal to zero.

4.2.1. Rigid Beam Elements

A modelling problem arose when the intersections of various structural members were considered. The beams were being modelled as simple lines that were located at the beams' centroidal axes. This

created difficulties when the centroidal axes of connected beams did not intersect at beam junctions. The SDRC (1980) indicated that an appropriate method of simulating such connections was to use rigid beam elements to connect the non-intersecting beams. Consider an end view of the swather header as shown in Figure 4.2.

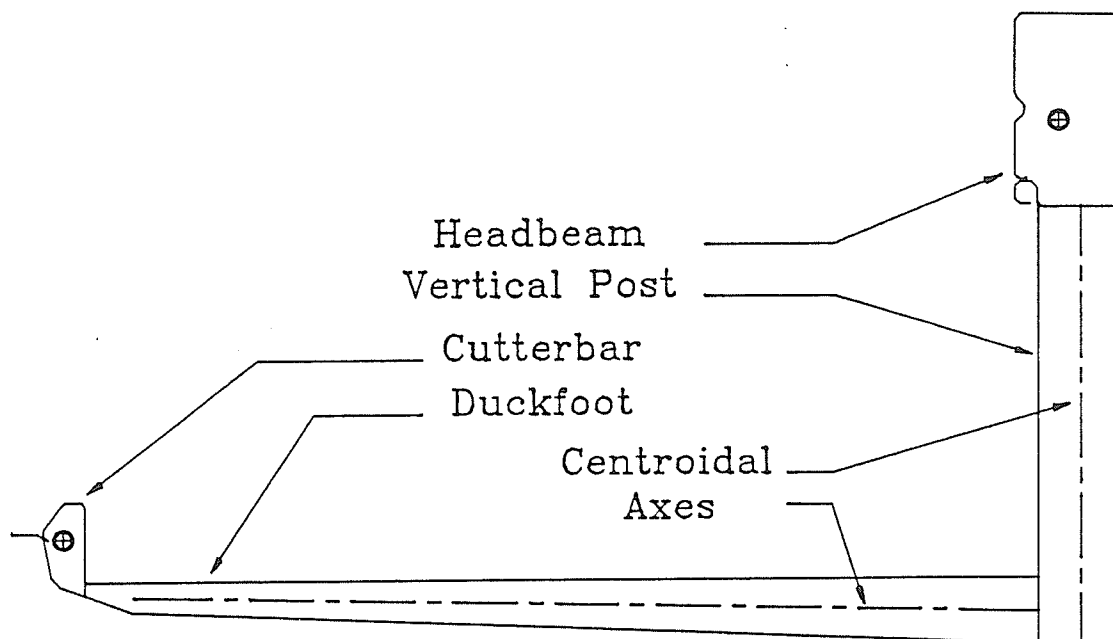


Figure 4.2: Cross-section of swather header.

The centroidal axes of the vertical posts were perfectly vertical. The centroidal axis of the headbeam extended parallel to the X axis above and ahead of the vertical posts. The centroidal axes of the vertical posts and headbeam do not intersect. Similarly, at the front end of the duckfeet, the cutterbar is above and ahead of the end of the duckfeet.

Because the duckfeet were tapered, their centroidal axes were at an upward slope relative to the horizontal as they extend from the vertical posts to the cutterbar. Rather than extending the duckfeet downward and

rearward, a more realistic connection was a horizontal link connecting the duckfeet to the bottoms of the vertical posts.

As shown in Figure 4.3, rigid beam elements were located between the tops of the vertical posts and the headbeam. They were also situated between the rear ends of the tapered duckfeet and the bottoms of the vertical posts. Finally, rigid beam elements were used to connect the front ends of the duckfeet to the cutterbar.

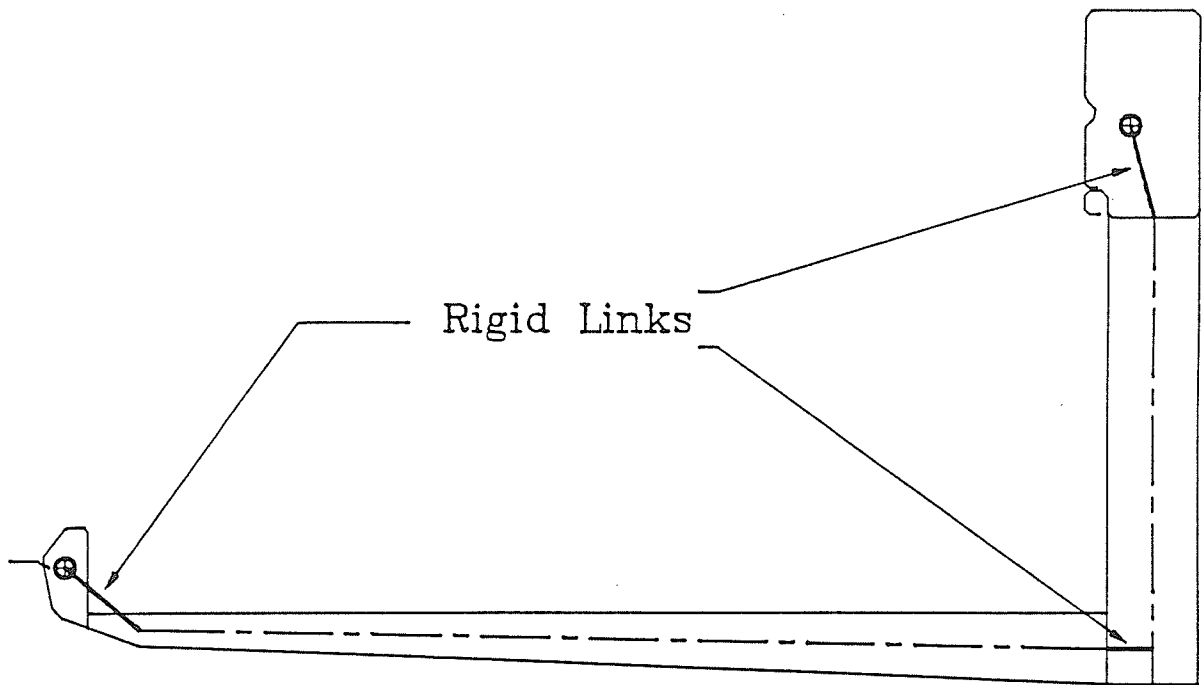


Figure 4.3: Rigid link locations.

4.2.2. Nodes

After the location of the origin was chosen, the nodes with a positive X coordinate were identified. All three orthogonal coordinates were determined for these nodes. The location of the nodes were determined through the use of production drawings supplied by Versatile

Farm Equipment Operations. The coordinates were calculated to the nearest millimetre, then manually entered into the computer using the "Model Preparation" sub-program of I-DEAS. As they were entered into the computer, they were successively numbered from 1 to 26. Figure 4.4 displays the locations of the first one-half of the nodes that were entered.

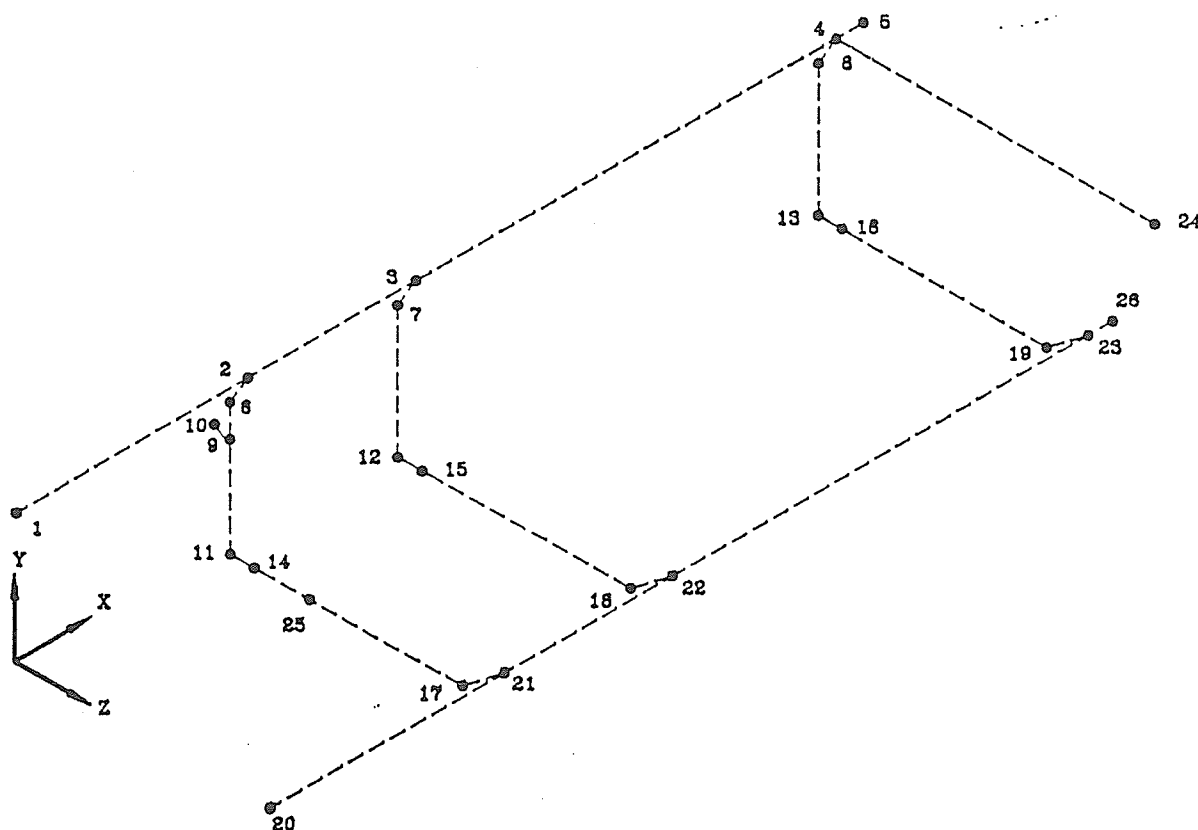


Figure 4.4: Locations for one-half of the nodes used in the model.

4.2.3. Beams

After one-half of the nodes were identified and entered into the computer, beam elements were used to join the nodes and create one-half

of the structural elements. As the beams were entered, they were successively numbered from 1 to 27. SDRC (1980) had specific names for beams of various shapes. A 'straight' beam element had a constant cross-section from one end to the other. The cross-section of a 'tapered' beam element changed linearly from one end of the beam to the other. All beam elements had 12 degrees of freedom. Each end was allowed to move in all three linear directions and rotate about all three axes.

Although the structure consisted of straight and tapered beams, all beams were initially assumed to be the same type. The physical property tables were filled with default values that were adjusted after all beams were created. The locations of one-half of the beams are shown in Figure 4.5.

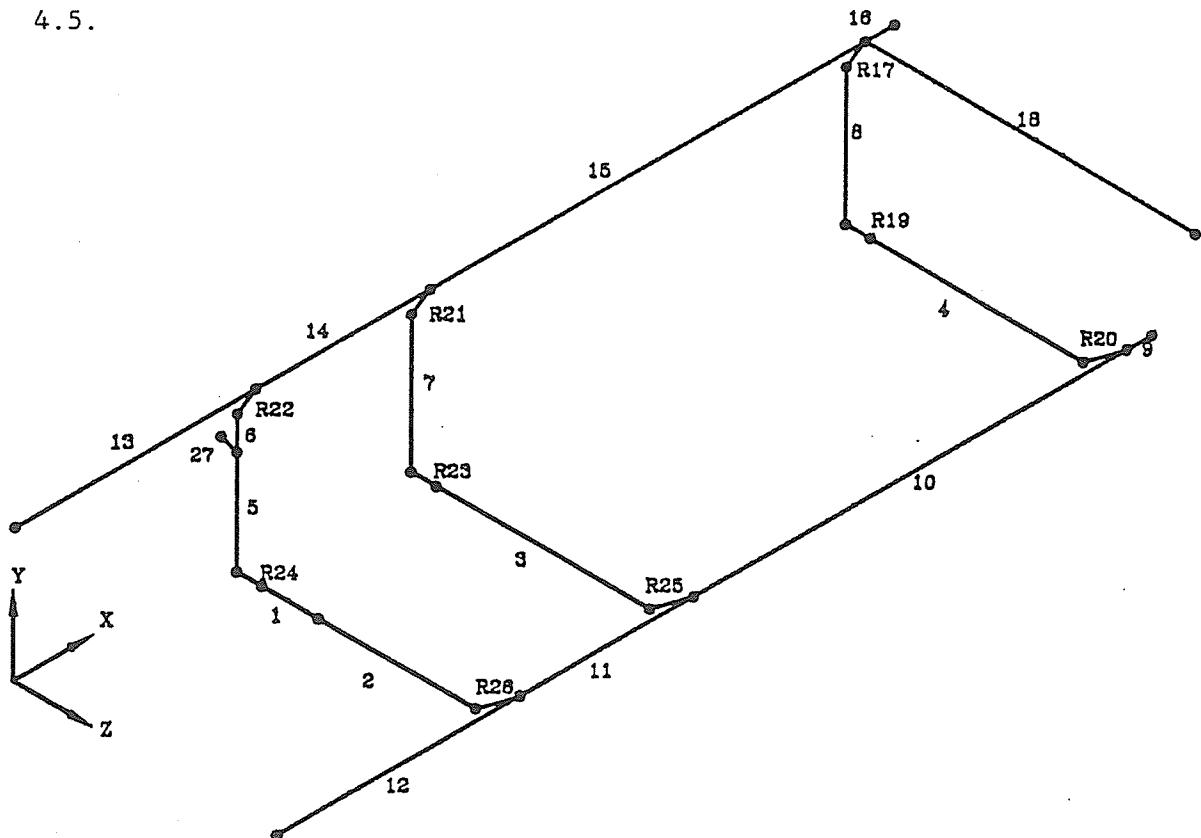


Figure 4.5: One-half of the beams used in the model.

Four different types of straight beams were defined. The headbeam included sections 13 to 16 and extended from node 1 to node 5. The cutterbar was also a straight beam made of beam sections 9 to 12. One-half of the cutterbar extended from node 20 to 26.

Another set of straight beams described the vertical posts. There were three posts that were located below the headbeam. The vertical post closest to the origin was made of two sections to accommodate the upper connection point. These two sections were numbered 5 and 6. The vertical post closest to the origin was located between nodes 11 and 6 with node 9 located between node 11 and node 6, 20 millimetres down from the top of the post. The other two vertical posts were located between nodes 7 and 12, and finally 8 and 13. The last two sections were numbered 7 and 8, respectively.

The final straight beam that was defined simulated the reel support arm. It extended from node 4 to 24. The beam was designated as beam number 18.

Three different tapered beams were defined to simulate the "duckfeet" that extended forward from the vertical posts and attached to the cutterbar. One tapered beam was used to extend from the vertical posts directly to the cutterbar. Examples of these were beams 3 and 4. They stretched between nodes 15 and 18, and 16 and 19, respectively. The second type of tapered beam, beam 1, extended from the vertical post closest to the origin to the lower tractor connection point at node 25.

The final tapered beam type, beam 2, continued from the lower connection point to the cutterbar at node 17.

A special note must be made about the ends of the headbeam and cutterbar. In the case of the headbeam, beam section number 16 extended from node 4 to node 5 and its length was only 30 millimetres. This short beam was used to correctly position the lumped mass that would simulate the drive motor and assembly bracket. Similarly, in the case of the cutterbar, beam 9 extended from node 23 to node 26 and was 10 millimetres long.

4.2.4. Physical Properties

The appearance of these beams on the computer screen did not reveal the beam type. The physical properties of each beam, stored in their respective physical property tables, defined their physical differences. All values for the physical properties tables were calculated by a sub-program of I-DEAS.

A scaled cross-section for each beam was created and entered into the computer. The computer calculated all the required physical properties. In the case of the 'straight' beams, only one cross-section was analyzed because the cross-section at one end was similar to the cross-section at the other. However, the tapered beams had two cross-sections that were analyzed for each type of beam. The beam cross-sections and their physical properties are displayed in Appendix A.

The degree of fixity coefficient quantifies the rigidity of a beam's end connections. It was not specifically designated in any physical property table. The degree of fixity is used when non-circular bars are twisted. In such a case, the cross-sections do not remain plane, but they warp. The ability of a beam to resist warping is determined by the connections at the ends of the beam. An increase in warping resistance also increases the stiffness of the structure. The equation below, as described by Crandall et al. (1987), defines the torsional moment per unit angle of twist of a rectangular shaft:

$$\frac{M_t}{\theta} = \frac{C * G * a * b^3}{L} \quad 4.1$$

where:

- M_t = torsional moment applied
- C = coefficient dependent upon the shaft cross-sectional dimensions
- G = shear modulus
- a = dimension of one side of the shaft
- b = dimension of other side of the shaft
- θ = angle of rotation
- L = length of beam

When including the effects of the end connections, the righthand side of the equation listed above is multiplied by the degree of fixity as shown below.

$$\frac{M_t}{\theta} = D_f * \left(\frac{C * G * a * b^3}{L} \right) \quad 4.2$$

where: D_f = degree of fixity

The default value for the degree of fixity, 0.6, was used in all physical property tables as a first approximation. The SDRC (1980) suggested a

degree of fixity for a number of different connections. These suggested values are displayed in Table 4.1 below.

Table 4.1: Suggested values for degree of fixity for various connections.

<u>Degree of Fixity</u>	<u>Connection Type</u>
0.0 - 0.2	few bolts connecting intersecting beams
0.2 - 0.4	well bolted or welded to a weak member
0.4 - 0.6	welded to a moderately strong member
0.6 - 1.0	continuously welded to plate structure

4.2.5. Material Properties

The material properties for all structural members were assumed to be constant regardless of the beam type or location. Although the structure contained different types of carbon steels with various chemical compositions, Dr. Guoguang Chen (pers. com. Sep 1989) stated that the material properties would not differ substantially between the different types of steel. Therefore, the same material properties could be used for all members.

The following material properties were used based on the text by Crandall et al. (1978). Young's modulus was assumed to be 2.15×10^5 MPa and Poisson's ratio was taken as 0.290. The density of all structural material was assumed to be equal to 7820 kg/m^3 .

After one-half of the model was complete, the structural members were "mirrored" about the Y-Z plane where the X coordinate was zero. The structure was geometrically symmetric about its centerline. The model then consisted of 50 joints connected by 54 straight, tapered, and rigid

beams. Detailed descriptions of initial input parameters and computational details are listed in Appendix B.

4.2.6. Lumped and Distributed Masses

The lumped masses on the model were used to simulate masses due to assembly brackets, drive motors, and associated hardware. The lumped masses were not symmetric about the centerline of the swather. Therefore, they were specified after the entire structure was defined. The brackets and components used on all lengths of the swather headers are similar. Thus, Enns (1988) provided all information required to determine the lumped mass quantities. The calculations to derive these values are shown in Appendix C.

A lumped mass was located at the left end of the headbeam. The mass, 187.0 kg, was located at node 5 to simulate the added mass due to the knife drive motor, the reel arm support bracket, hydraulic cylinders and the rear portion of the crop divider boards. The lumped mass at the opposite end, node 30, accounted for the same mass except for the swather drive motor. The mass applied was equal to 41.7 kg.

Lumped masses were also positioned at the ends of the cutterbar to account for the added mass due to the front portion of the divider boards. A lumped mass equal to 27.2 kg was located at nodes 26 and 50.

Finally, lumped masses were located at the ends of the reel arms to simulate the load due to the reel and the reel motor. The mass at node

24 was equated to 160.0 kg, one-half of the reel mass, while the mass at node 48, 170.0 kg, accounted for one-half of the reel mass plus the mass of the reel drive motor.

In addition to the dead weight load of the structural beams, there were other components which applied distributed loads to various locations on the structure. The drapers applied distributed loads to portions of the headbeam and cutterbar. Also, the guards, knife and associated hardware applied a constant distributed load to the entire length of the cutterbar.

To closely simulate a distributed load, the densities of the structural members were adjusted. The density of the headbeam between the drapers, beams 13 and 40, was 7820 kg/m^3 . The adjusted density of beams 12 and 39, $10\,862 \text{ kg/m}^3$, simulated the loading due to the knife and guards. This density was then increased further in the case of beams 9 to 11 and beams 36 to 38. The density was increased to $18\,542 \text{ kg/m}^3$ to reflect the additional loading due to the drapers. Finally, the density of beams 14 to 16 and 41 to 43 was entered as $15\,824 \text{ kg/m}^3$. This modelled the effect of the drapers on the headbeam. The associated calculations to derive these values are also supplied in Appendix C.

4.2.7. Restraints

SDRC (1980) stated that a rigid beam element can not be connected to ground because one end of the rigid link is eliminated from calculations. It is essential that the reaction forces at the ground be

included in the calculations. Therefore, short beams were positioned from node 9 to node 10 and from node 34 to node 35 to simulate the upper connection points between the swather header and the tractor. These short beams were modelled having excessively high Young's moduli and low densities. Thus, the members would be very stiff yet their mass would be negligible compared to the rest of the structure.

After all structural beams were created and entered, the restraints were defined. The header is connected to the tractor at four different points. The first two points were located near the top of the two centre vertical posts at nodes 10 and 35. The other two connection points, nodes 25 and 49, were located on the duckfeet, 300 millimetres ahead of the two centre vertical posts.

The lower connection points were located at nodes 25 and 49. These two nodes divided the centre duckfeet into two portions and two tapered beams were required to define the entire length. Thus, three different tapered beams were required to model all types of the tapered beams.

As described by Enns (1988), the lower and upper connection points consisted of a bar fitting into a downward opening slot. I-DEAS allowed six different fixities at each boundary point. Each node that designated the connection points could have been fixed in any of the three orthogonal directions and any of the three rotational directions.

A number of different boundary condition configurations were attempted but, intuitively, only one configuration seemed logical. The lower connection points supported the swather header in the Y and Z directions. At the upper connection points, there was only support in the Z direction. To simulate these conditions, the lower connection points were fixed in the Y and Z directions and the upper connection points were fixed in the Z direction. The connections were only loose fitting pins, and no moment-carrying capacities existed.

After all structural members were defined and the boundary conditions were supplied, the model was ready to be analyzed. The model was then written to a file that the "FRAME" sub-program would recognize. The analysis was completed using the "Dynamic Analysis of General Structures (DAGS)" portion of FRAME. Within the 7.5 to 14 Hz frequency range, the structure's natural frequencies and associated mode shapes were predicted. DAGS was also used to process and display the predicted results.

5. Experimental Equipment

5.1. Introduction

An experimental modal analysis was completed to validate the computer model that was described previously. A random force history was used to excite the structure while a tri-axial accelerometer was used to measure the structure's accelerations.

Figure 5.1 illustrates the experimental set-up. The experimental equipment was divided into two different portions. The excitation equipment included all instruments that were used to set the swather header into vibration. The measurement and analysis instruments were those that were used to measure, monitor, and analyze the supplied force and the acceleration responses.

5.2. Excitation Equipment

A Zonic Xcite 1100-6 servo-hydraulic exciter was used as an actuator to shake the swather. The stinger was connected to the top of the actuator's piston. Using an adaptor, the other side of the stinger was connected to one side of the force transducer. The other side of the Zonic model 2522 load cell was secured to a section of square tubing that was then clamped to the swather table.

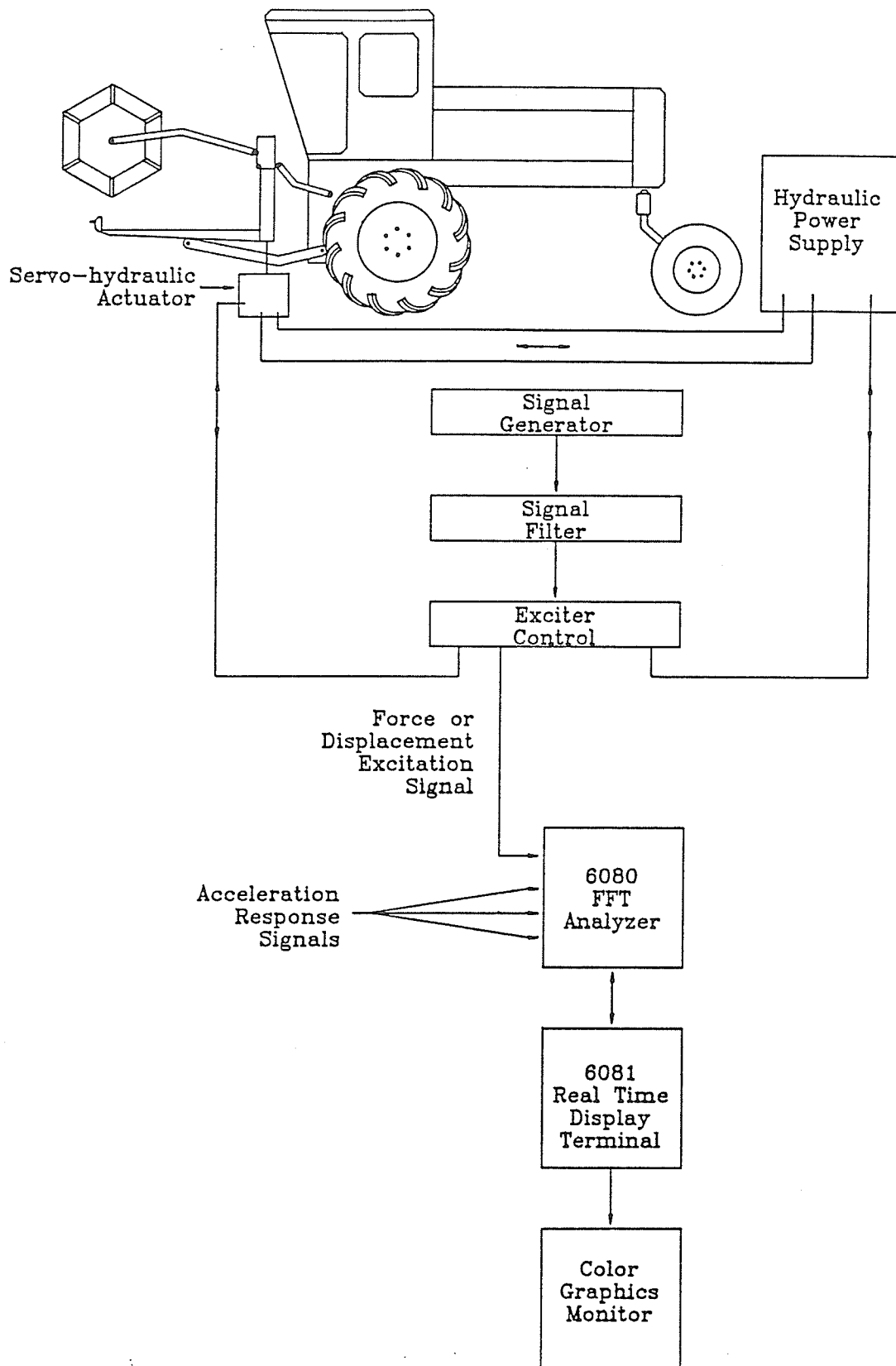


Figure 5.1: Experimental set-up.

Both ends of the stinger were formed by a 3/8"-24 stud. Likewise, a 3/8"-24 stud was located on each side of the load cell. The stinger was turned into a 3/8"-24 receptacle in the top of the actuator's piston. An adaptor was then used to connect the stinger to one side of the load cell. To attach the load cell to the swather header, a 7/16" through hole was drilled in a piece of square tubing. The stud of the load cell was then passed through the hole and a nut was tightened on to the stud to hold it in place. The square tubing was then securely clamped to the bottom of the middle vertical post on the left side of the header. The piece of square tubing was used so that a hole did not have to be drilled in the swather. Supported by six cinder blocks, the actuator applied a vertical force to the swather header. Figure 5.2 displays the orientations of the actuator, force transducer, and stinger.

A Wavetek VCG Noise Generator was used to create the forcing function. The random signal was band-limited by a Wavetek Model 442 Dual Hi/Lo Filter. The filtered signal then passed on to the Xcite Master Controller. The controller monitored and correctly positioned the head of the actuator's piston. The force signal created by the force transducer then resembled the random signal that the controller received.

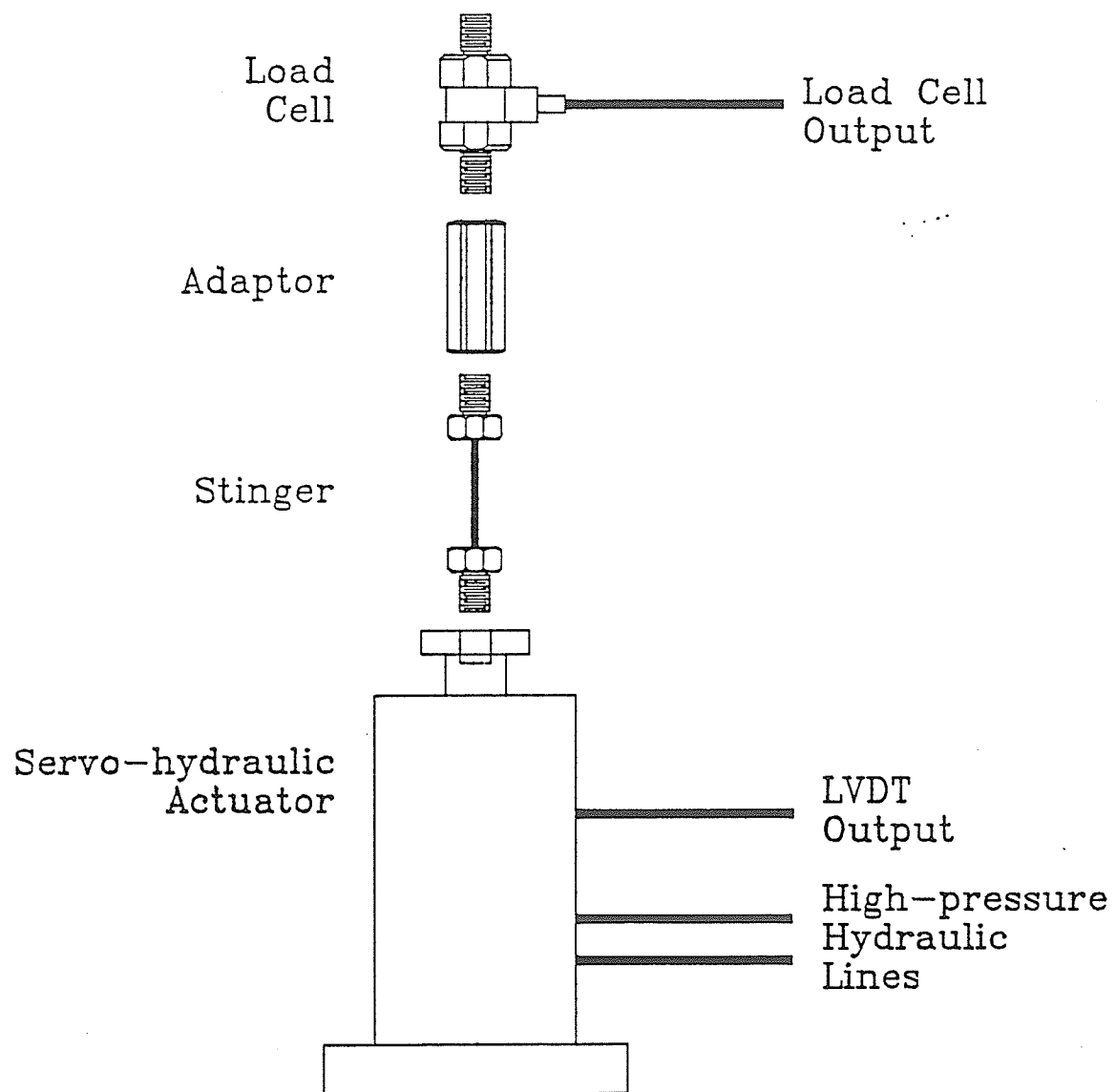


Figure 5.2: Actuator, load-cell and stinger orientation.

5.3. Measurement and Analysis Equipment

The force signal passed from the force transducer back to the master controller then to the Zonic four-channel FFT analyzer. The response of the structure was measured by a tri-axial accelerometer as shown in Figure 5.3.

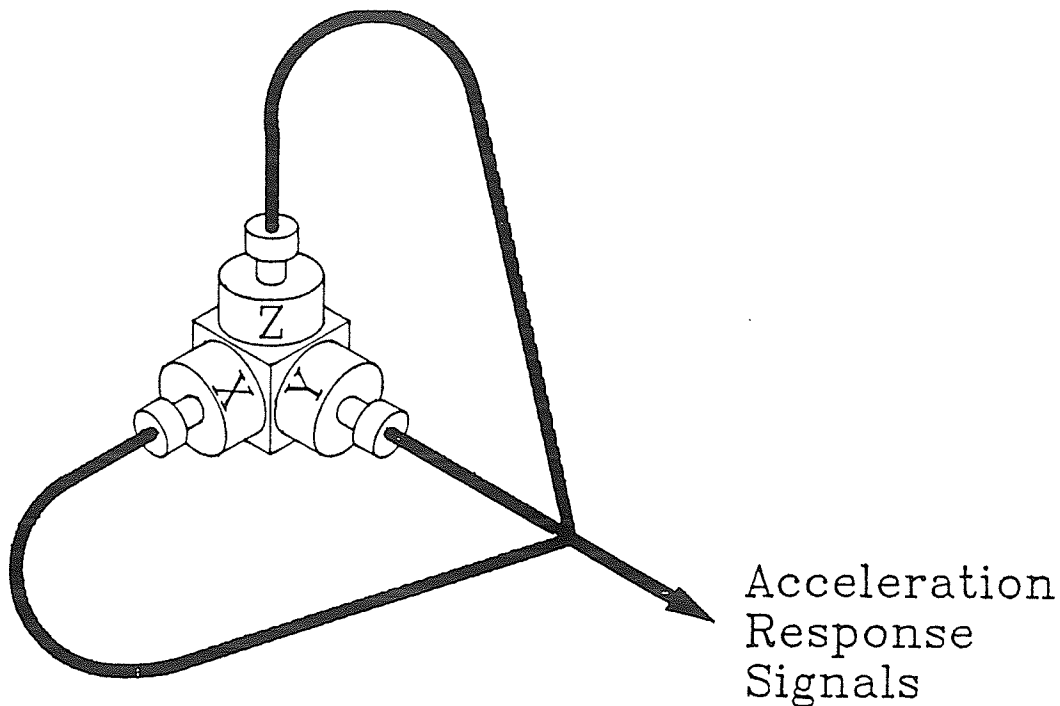


Figure 5.3: Schematic of tri-axial accelerometer.

The acceleration signals from each of the accelerometer's axes passed through a charge amplifier and then on to the FFT analyzer. The four channels, the force signal and three response signals, were first processed by the FFT analyzer. After the signals were averaged appropriately and processed, the data files were stored on floppy disks. The data files that were stored on floppy disks could then be recalled at a later time for plotting or transferring to the VAX minicomputer.

After all data were collected, the files were transferred from the floppy disks to the VAX 11/780 minicomputer. After being transferred to the minicomputer, MODAL PLUS, a modal analysis software package was used to extract the experimental modal parameters.

A detailed description of all experimental instruments is listed in Appendix D.

6. Experimental Procedure

6.1. Introduction

The predicted results from the initial analytical modal analysis were used to help establish the experimental set-up and procedure. The mode shapes were used to locate the optimum excitation location and the response measurement locations. The motion of the header in its natural modes of vibration was primarily in the vertical direction. The joint at node 12 in the model was chosen as the excitation location because it experienced predominantly vertical motion in all natural modes of vibration of interest. The method used to supply the input force was then determined by completing preliminary experimental tests.

6.2. Stinger versus Load Button

It was first assumed that a random force signal would be used to excite the structure. However, the decision to use either a compressive load button or a "stinger" had not been made. Preliminary tests using each mode of force input were completed to help make a decision.

A compressive preload is first applied because the load button must remain in contact with the structure at all times. Therefore, the force varied above and below a non-zero compressive load. At no time could the load cell experience a zero or a negative compressive force. The hydraulic pump controller used an initial DC offset to apply a 1112 N

static compressive load. The random load signal then varied from 1330 N to 890 N. While completing the pretest procedure, the DC offset was decreased so that the static load was 890 N, and the dynamic load varied between 1112 N and 667 N.

The frequency response functions for a similar point under the two different test conditions were compared. Obvious dissimilarities were present between the results under the two different test conditions. It was apparent that a nonlinearity was being created in the system by applying a purely compressive load to the structure.

To correct the nonlinearity problem, a stinger was used. The mean force applied to the structure was zero Newtons. At first the force signal varied between 222 N in tension and 222 N in compression. To test the effect of different force limits, the force magnitude was increased. The force then varied between 334 N in tension and 334 N in compression.

As illustrated in Figure 6.1, the change in force amplitude did not cause significant changes in the measured frequency response functions the phase plots or the coherence functions. This showed that using the stinger did not, in itself, introduce any nonlinearities in the system. All frequency response functions were collected with the maximum force in either tension or compression equal to 334 N.

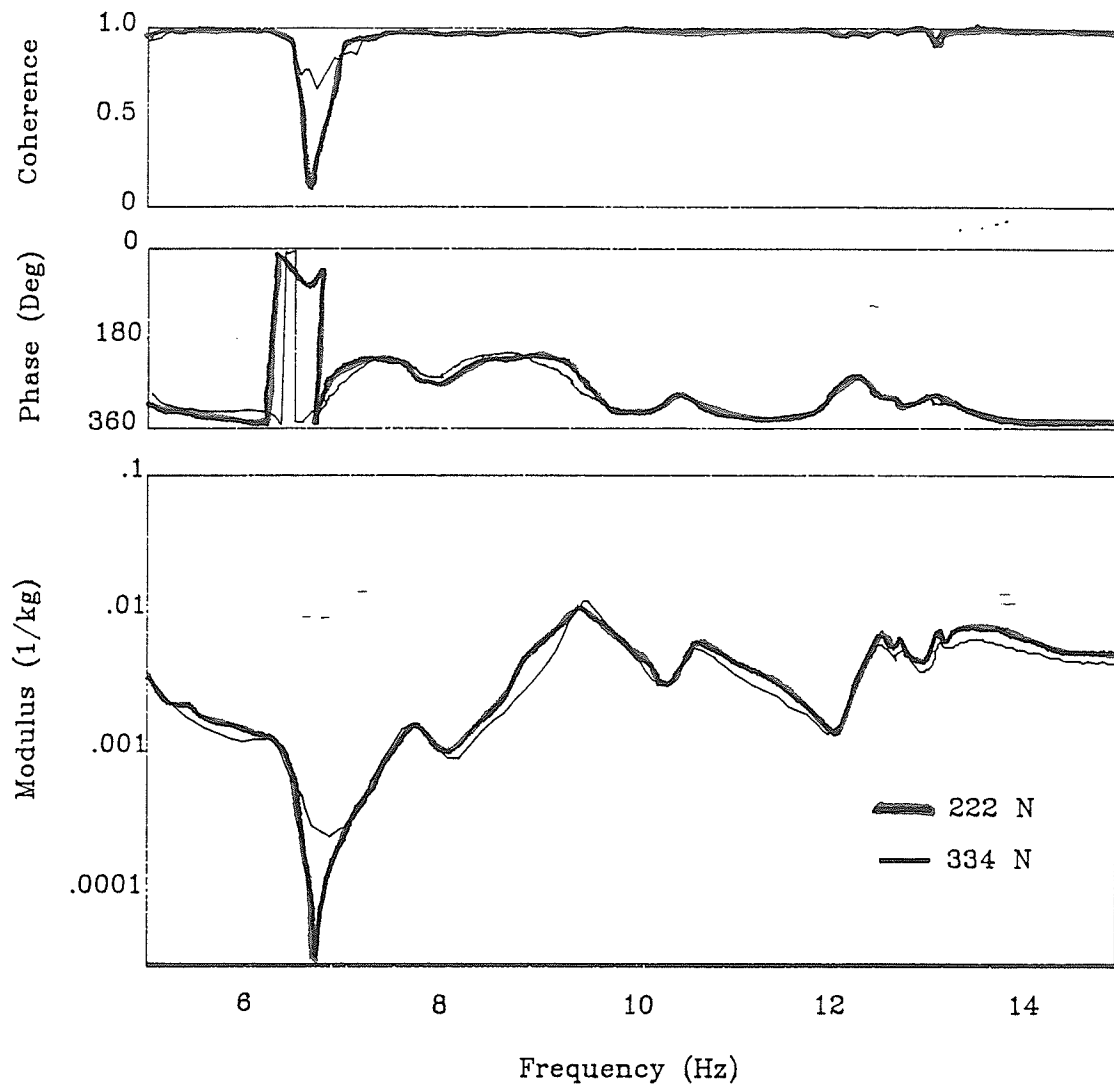


Figure 6.1: Frequency response function plots for two peak force inputs, 334 N and 222 N.

The actuator had a tendency to be lifted when the tensile load was applied. The actuator had to be held down with a force equal to or greater than 334 N. The actuator was bolted to a steel plate that weighed in excess of 556 N. In addition, a mass with a weight of 556 N was placed

on top of the steel plate. Therefore, the effective mass of the actuator was able to overcome the lifting capacity of the forcing signal.

6.3. Testing Procedure

The temperature at the time of testing made it uncomfortable to work outside. The swather was located outside the building at the Manitoba Research Council while the Zonic testing system was located inside the shop. Hydraulic pressure lines and electrical cables connected the hydraulic actuator and the accelerometer to the hydraulic pump and the Zonic test system. The overhead door was closed on these cables and hydraulic lines to conserve heat in the shop.

The noise generator created a random signal that was adjusted first by a low-pass filter then by a high-pass filter. After being filtered, the signal then contained frequencies ranging from 4 Hz to 20 Hz. The filtered random function was then passed on to the hydraulic pump controller and used as the forcing function. The mean force that was applied to the structure was zero Newtons and the peak-to-peak force amplitude was 668 N. The pump controller simply controlled the location of the actuator piston so that the force history resembled that of the supplied signal.

Each axis of the triaxial accelerometer was calibrated. A calibration exciter was attached to the accelerometer and a known input was supplied. A calibration factor for each direction was stored in the

Zonic test system to produce output data scaled in correct engineering units.

After each of the axes of the tri-axial accelerometer was calibrated, the responses of the structure could be measured. Prior to measuring any frequency response functions, a "node map" was created that identified the locations of response measurement. The locations that were measured were at the top and bottom of each vertical post and at the front of each duckfoot. Eighteen different locations were identified and numbered on the swather header. The same numbers that were used in the analytical model were used in the experimental model. These node numbers were also the file names that were used to save the acceleration and force histories.

Along with the nodes on the swather header, four nodes were identified on the swather tractor unit. These nodes were located at the front and rear corners of the tractor. They were used to determine the motion of the tractor at the swather header's natural frequencies.

In addition to node numbering, an imaginary coordinate system was created. The origin location was exactly the same as in the analytical model. The positive X axis extended from the middle of the header towards the left end. The Y axis pointed vertically upward and the Z axis pointed horizontally forward.

After the locations of measurement were determined and numbered, the Zonic system was configured. It was set to accept a continuous data signal. A built-in Hanning window was used to window each data buffer because a continuous random signal was being measured. The data buffer contained 1024 data points. After each buffer was Fourier-transformed, 512 data points remained. Due to roll-off of the anti-aliasing filters, only the first 400 of those data points were used by the Zonic analyzer. The analysis frequency bandwidth was between zero and 15.625 Hz with a frequency resolution of 0.039 Hz.

Although the excitation signal included frequencies from 4 to 20 Hz, the analysis extended from zero to 15.625 Hz. Since no natural frequencies were excited below 4 Hz, the responses below 5 Hz were neglected and the frequency response functions were plotted between 5 and 15.625 Hz. The data were collected using 50 averages. The coherence function was greater than 0.9 for all channels at all accelerometer locations.

After the system was configured and all instruments were set, acceleration and force spectra were collected. The data saved by the Zonic system are in the form of power spectra and cross spectra. For each channel pair, two power spectra and a cross spectrum are saved.

Caution was exercised when relocating the accelerometer at new locations. It was imperative that the orientation of the accelerometer was consistent. As a convention, it was assumed that the cables pointed

in the positive direction as they left the accelerometer. The accelerometer was then located so that the positive direction of each accelerometer coordinate coincided with the positive direction of the structure's coordinate system. If the accelerometer could not be correctly positioned because of some structural constraint, the particular file was identified and it was adjusted later. If the accelerometer could not be correctly positioned, it was positioned so that either one or two of the coordinates were correctly positioned. The remaining coordinate(s) was (were) oriented in the negative direction.

During the data collection process, the force and acceleration signals were monitored by passing them through an oscilloscope. Occasionally, the output from an accelerometer channel exceeded the previously specified limit and caused a signal overload. In such cases, the voltage gain was adjusted to accommodate the increased signal amplitude.

6.4. Uploading the Data for Analysis

After all data files were collected and saved on computer disks, the files were transferred to the VAX 11/780 minicomputer at the Manitoba Research Council. Each frequency response function was separately transferred to the minicomputer. They were designated by the node number that indicated the location of the accelerometer followed by the direction of measurement, either X, Y, or Z.

At this time, the coordinates that were oriented in the negative direction were adjusted by identifying them preceded by a negative sign. For example, assume that, at location number 3, the X and Z coordinates pointed in the negative direction. These acceleration histories were identified as 3X- and 3Z-.

MODAL PLUS was then used to manipulate the experimental data. A complex exponential algorithm was used to identify the global parameters. The frequency response function obtained by locating the accelerometer at the same location as the excitation force (i.e. the driving point) was used to determine the global parameters (natural frequencies and associated damping coefficients). After the global parameters were determined, the polyreference technique was used to determine each mode shape.

7. Analytical Model Adjustments

7.1. Introduction

After comparing the analytical and experimental natural frequencies, it was determined that the analytical model seemed to be too flexible at the second natural frequency. The analytically predicted natural frequency was lower than the experimentally determined value. In addition, all mode shapes were not exactly the same. The predicted deflections that the cutterbar experienced seemed to be excessive.

The boundary conditions that were described in the analytical model were considered to be too restrictive. The headbeam required more flexibility in the model to increase the range of its motion. The model adjustments are described in the following subsections.

7.2. Model Stiffness

To increase the stiffness of the structure, the degree of fixity was changed for all beams except for the headbeam and the reel support arms.

The reel support arms were connected to the headbeam with a pinned joint. Therefore, it did not seem realistic to increase the fixity of these joints. The degree of fixity for the headbeam was not increased because it was determined that the amount of motion that the headbeam had to experience needed to be increased.

The degree of fixity for all other beam connections was increased to 1.0. Table 4.1 indicated that a degree of fixity equal to 1.0 would correspond to a continuously welded connection to a plate-like structure. In the case of the vertical posts, each end was continuously welded to the headbeam or duckfeet. In addition, plate gussets were continuously welded on two sides of the connections to improve their rigidity. Similarly, the duckfeet were continuously welded to the cutterbar on all available sides. It appeared that the designers had hoped that the increased joint rigidity would stiffen the duckfeet and vertical posts. Thus, the motion of the cutterbar would be decreased.

7.3. Modification of Boundary Conditions

A closer inspection of the connections between the tractor and the swather header revealed that the connection points were not truly fixed in any direction. In addition, the motion of the connection points at the top of the swather header depended on the motion of the connection points at the bottom.

Figure 7.1 shows the linkage system that connected the tractor and swather header. Two springs supported each lower connection point in the vertical direction. Assembly drawings showed that the total effective stiffness of the two springs was 68 646 N/m.

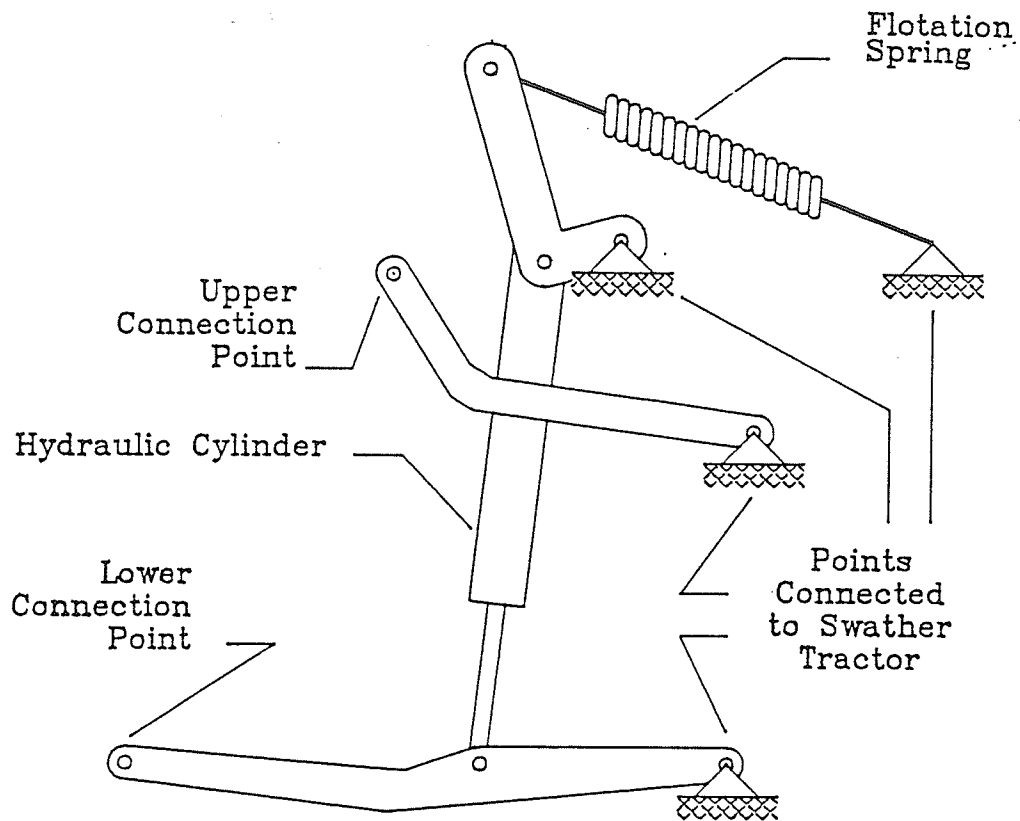


Figure 7.1: Linkage between swather header and tractor.

To more closely simulate these boundary conditions, two additional beams and two spring connections were created in the model. Nodes 51 and 52 were created behind and below the upper connection points. Two beams were used to connect node 51 to node 10 and node 52 to node 35. These beams modelled the connection between the upper connection points on the header and the tractor. At nodes 51 and 52, the beams were fixed in the Y and Z directions but they were free to rotate about all axes.

To simulate the flotation springs, two spring connections to ground were created. The stiffness in any of the three directions could be specified. The stiffness in the Y direction, 110 000 N/m, was determined

through a simple static analysis. This derivation is shown in Appendix E. The springs were made excessively stiff in the Z direction to simulate a rigid beam between the lower connection points and the tractor. All model changes are represented in the modified input listing in Appendix F.

After the modifications to the model were complete, the natural frequencies and mode shapes were predicted again.

8. Forced Response Prediction

8.1. Introduction

After the analytical model was modified to more closely simulate the experimental results, it was used to predict the displacements of the connection points between the tractor and swather header. The response analysis predicted the response due to a provided sinusoidal dynamic input. The dynamic inputs simulated an unbalance in the knife drive flywheel and an inertial load created by the reciprocating knife. It was hoped that the dynamic predictions would indicate an optimum knife speed, and decrease the discomfort felt by the operator.

8.2. Load Application

It was assumed that there were two possible locations in which a dynamic force may be created; first, at the hydraulic motor that drives the knife and, second, at the knife head that connects the knife to the drive linkage.

The hydraulic motor was connected to a flywheel that was in turn connected to a rotating hub. The rotating hub lies in the Y-Z plane. Therefore, if the hub was subjected to eccentric loading, harmonic loading would occur in the Y and Z directions. The phase between the two harmonic loads would differ by 90 degrees.

The rotating hub was responsible for driving the knife in its reciprocating fashion. Therefore, the phase between the rotating hub and knife was constant. A vertical upward force at the rotating hub coincided with a force from right to left at the knife drive.

An I-DEAS sub-program, DAGSINE, was used to locate the dynamic loads and predict the structure's responses. DAGSINE was used to predict structural responses if the drive frequency varied from 7.0 Hz to 14.0 Hz.

The supplied input for the rotating unbalance was the unbalanced mass multiplied by the distance of offset from its centre. From this input, the applied force would be proportional to the drive frequency squared. The rotating unbalance value of 0.15 kg*m, as described by SDRC (1980), was located at node 5 in the model.

The dynamic loading that simulated the motion of the knife was a harmonic linear force in the X direction. A 1000 N force was located at node 26 in the model (the left end of the cutterbar as seen by the operator). The linear force at node 26 acted in the positive X direction when the rotating unbalanced force acted in the positive vertical direction at node 5.

If no damping was assumed to exist in the model, infinite response magnitudes would be expected at the four previously determined natural frequencies. However, damping coefficients (unitless values indicating the level of damping at each natural frequency) were determined

experimentally for each of the natural modes of vibration. These experimentally determined damping coefficients: 0.039, 0.020, 0.026 and 0.013 were respectively assigned to each of the four natural modes of vibration.

Physically, the phase between the rotating unbalance and linear forces may vary. A phase difference may occur due to torsion in the shaft between the rotating hub and the knife head. A number of different analyses were completed when the phase between the two forces, rotating and linear, varied between +10 degrees and -10 degrees. There were no discernible differences between the different predicted responses. Therefore, only the predicted response with zero phase difference between the two forces was analyzed.

It had been stated by the technicians at Versatile Farm Equipment Operations that the discomfort felt by the operator was caused by the operation of the swather header. All forces that are transmitted from the swather header to the tractor unit must go through the four connection points between the two units.

The displacements of the four connection points, nodes 10, 25, 35, and 49, were predicted so that the relative responses of the swather header at various speeds may be compared. Larger displacement amplitudes would indicate more force exerted on the tractor by the swather header. An optimal operating speed could be determined by studying the

displacements of the connection points. The driving frequency that caused the least displacement would be the optimum operating frequency.

9. Results

9.1. Introduction

This section consists of four subsections. The first subsection includes the results from the first analytical model. The second subsection includes the experimental results. The third subsection shows the results that were obtained from the analytical model after it was modified. Finally, the predicted responses from the modified analytical model is shown.

The results from the analytical modal analyses and the experimental modal test include the natural frequencies and their associated mode shapes. A number of different views of each displaced mode are provided to create a clear understanding of the motion that the header experienced.

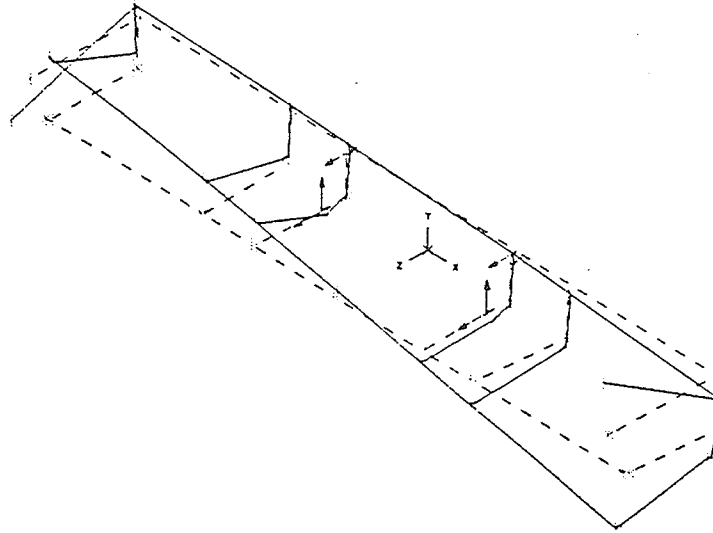
The modified analytical model was used to predict the swather header's response to a supplied sinusoidal dynamic input at various speeds. The displacements of the four connection points were used to compare the amount of force that was transmitted from the swather header to the tractor unit. The phase of each displacement helps to relate the different displacements that are plotted on the same graph.

9.2. Initial Analytical Model

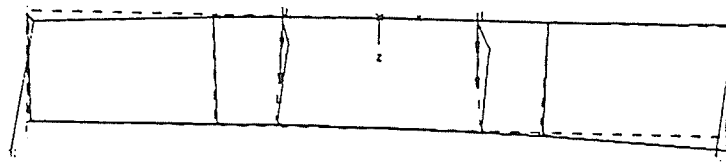
Four natural frequencies were predicted between 7.5 and 14.0 Hz. These natural frequencies occurred at 7.6, 8.9, 10.6, and 12.5 Hz.

The mode shape at 7.6 Hz, shown in Figure 9.1, demonstrates the fundamental natural mode of vibration. Primarily, the cutterbar was "teeter-tottering". The cutterbar remained straight, and rocked about an imaginary fulcrum placed near the centre of the header.

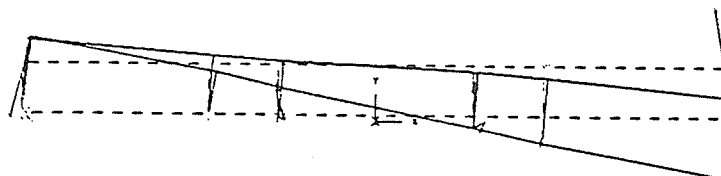
The ends of the headbeam also oscillated vertically, but it exhibited less motion than the cutterbar. The motion of the headbeam was in phase with the cutterbar. The ends of the entire header oscillated vertically. Little rotation or motion in the Z direction was predicted.



a) isometric view



b) plan view



c) front view

Figure 9.1: Predicted mode shape at 7.6 Hz.

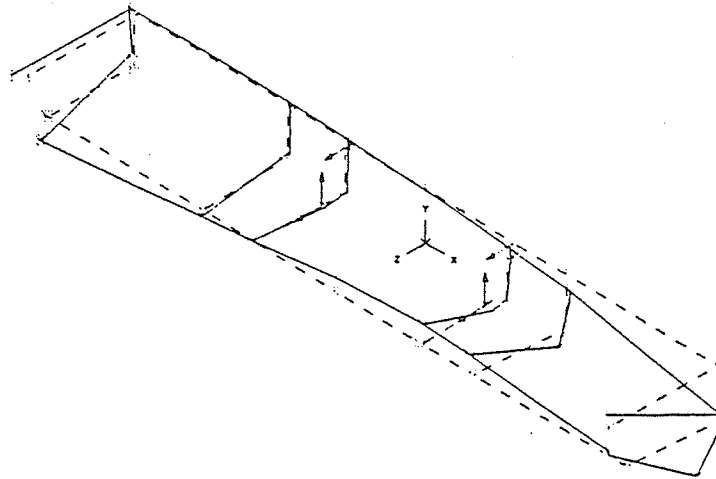
I-DEAS predicted that the second natural mode of vibration was occurring at 8.8 Hz. As shown in Figure 9.2, the ends of the headbeam were experiencing torsional loading. Similar to the first natural mode of vibration, the cutterbar rocked about a point near the centre of the header.

The vertical motion of the headbeam and cutterbar were out of phase (the end of the headbeam moved up while the cutterbar moved down). This motion created torsional moments at the ends of the headbeam.

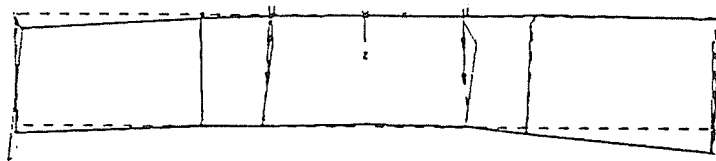
The ends of the header moved out of phase also. The plotted mode shape shows that the left end of the headbeam is displaced downward while the cutterbar is displaced upward. At the right end, the headbeam was displaced upward and the cutterbar was displaced downward. This opposing motion from one end to the other caused opposing torsional moments at each end of the headbeam.

The cutterbar did not remain straight in this mode of vibration. The cutterbar resembled a gentle "S" curve from one end to the other. Similarly, the headbeam curves gently from one end to the other.

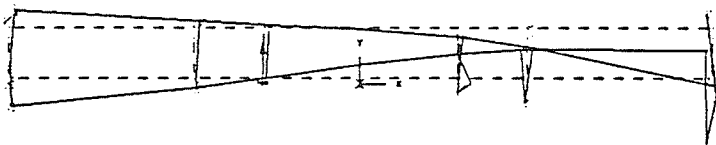
Little deflection in the Z direction was predicted. The ends of the cutterbar moved slightly as the rotation of the headbeam forced them forward. The right end of the headbeam experienced some oscillation in the Z direction, but the left end did not.



a) isometric view



b) plan view



c) front view

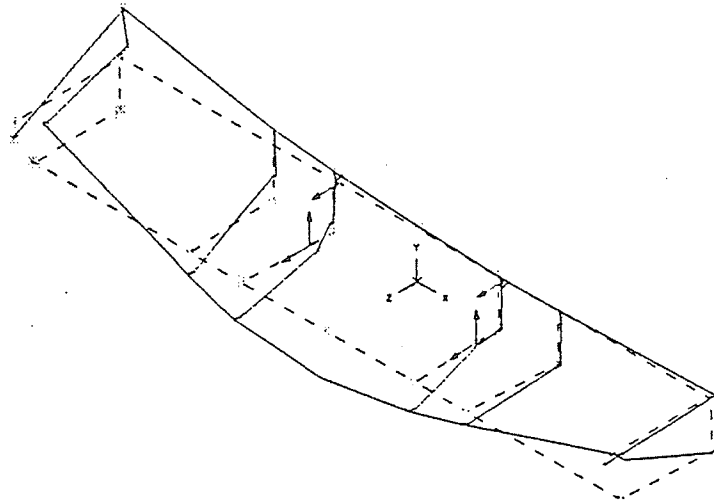
Figure 9.2: Predicted mode shape at 8.8 Hz.

The third natural mode of vibration was predicted to occur at 10.6 Hz. The structural element that experienced the greatest deflection in this mode was the cutterbar. The left end of the headbeam remained relatively static compared to the opposite end.

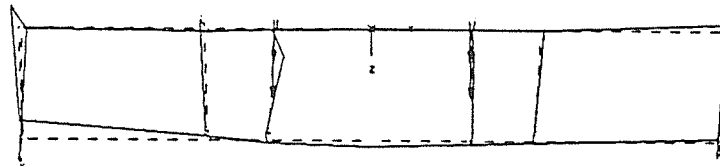
The cutterbar oscillated so that it changed from being concave up to concave down. Figure 9.3 shows that the vertical displacement at the middle of the header was nearly equal the vertical displacement at the ends.

As the right end of the cutterbar moved upward, the right end of the headbeam also moved upward. The left end was not similarly affected. To allow the vertical motion of the cutterbar at the left end, the headbeam was displaced backward, and a negative moment was applied. Also, the duckfoot at the left end acted as a cantilever as it oscillated vertically.

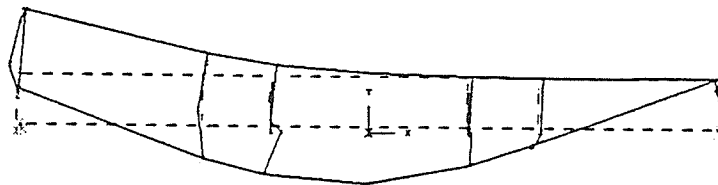
Again, the only oscillation in the Z direction occurred at the ends of the headbeam and cutterbar. These motions were caused by rotation at the ends of the headbeam.



a) isometric view



b) plan view



c) front view

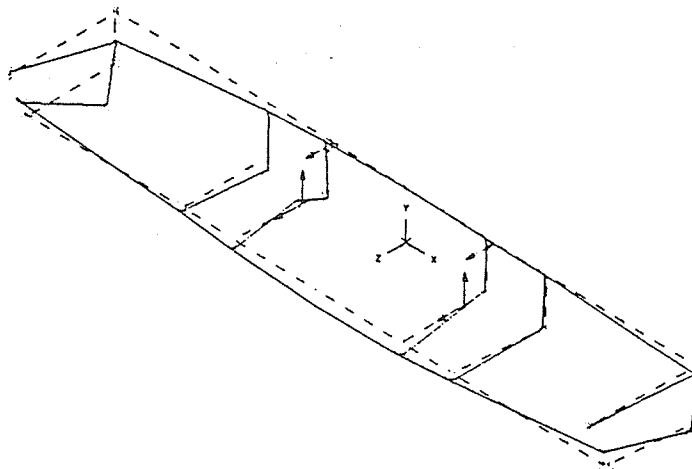
Figure 9.3: Predicted mode shape at 10.6 Hz.

The fourth predicted natural mode of vibration occurred at 12.5 Hz. As in Figure 9.4, the ends of the headbeam were in phase and the ends of the cutterbar were in phase. However, the ends of the headbeam were not in phase with the associated ends of the cutterbar. As one end of the cutterbar moved up, the same end of the headbeam moved down.

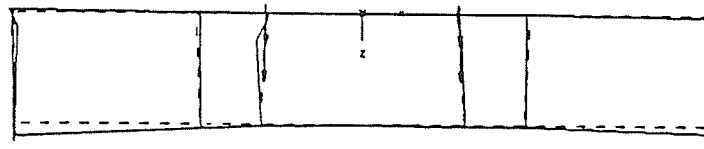
The cutterbar was displaced in the same manner as in the previous mode. During one cycle, it changed from being concave up to concave down. Again, the vertical displacements at the ends of the cutterbar were approximately the same as the vertical displacement at the centre of the header.

To allow the motion at the ends of the cutterbar, the ends of the headbeam were displaced downward and rotated. Compared to the left end, a larger amount of vertical deflection occurred at the right end of both the cutterbar and headbeam.

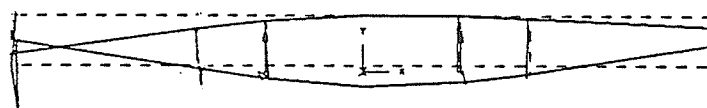
The only oscillations that occurred in the Z direction were experienced by the cutterbar. The ends of the cutterbar moved forward as they moved upward. As the ends moved downward, they moved backward.



a) isometric view



b) plan view



c) front view

Figure 9.4: Predicted mode shape at 12.5 Hz.

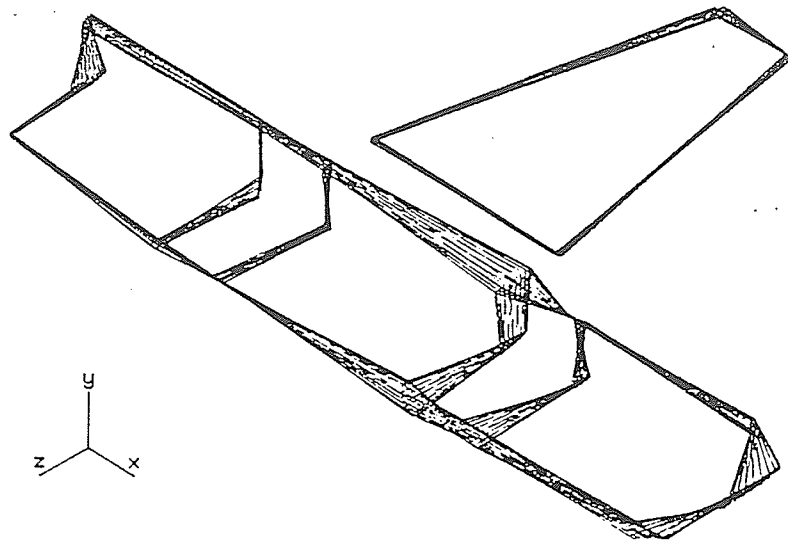
9.3. Experimental Results

Four natural modes of vibration were determined within the 7.5 to 14 Hz frequency range. They occurred at 7.7, 9.4, 10.5, and 12.5 Hz.

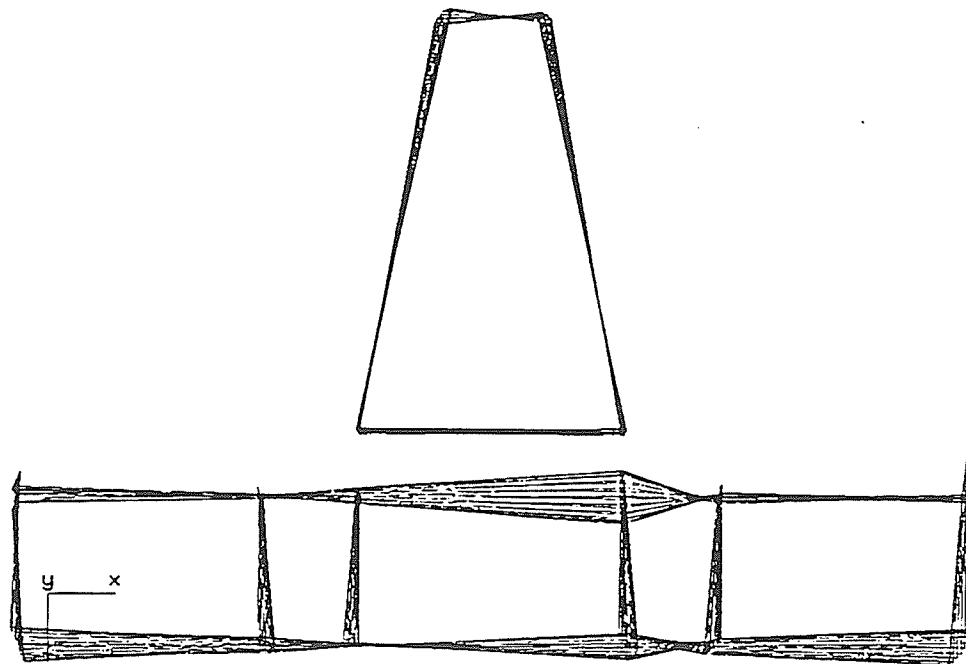
The first mode of vibration at 7.7 Hz is shown in Figure 9.5. The cutterbar rocked like a "teeter-totter". The headbeam moved very little in the vertical direction. The motion of the cutterbar was due, therefore, to rotation of the ends of the headbeam and deflections of the duckfeet.

The duckfeet, welded to the bottom of the vertical posts, acted as cantilever beams. Thus, little vertical deflection by the headbeam was required to make the cutterbar behave as it did.

The top of the third vertical post from the left end experienced a large amount of vibration in the Z direction. This may have been caused by the upper connection link between the header and the tractor. The accelerometer was sensitive to even the slightest impact. If the pin in the upper connection point experienced a slight shock loading, the measurement by the accelerometer would have reflected a large response, as displayed. The rear of the tractor experienced rotational vibration about the Z axis. Little vibration was detected at the front.



a) isometric view



b) plan view



c) front view

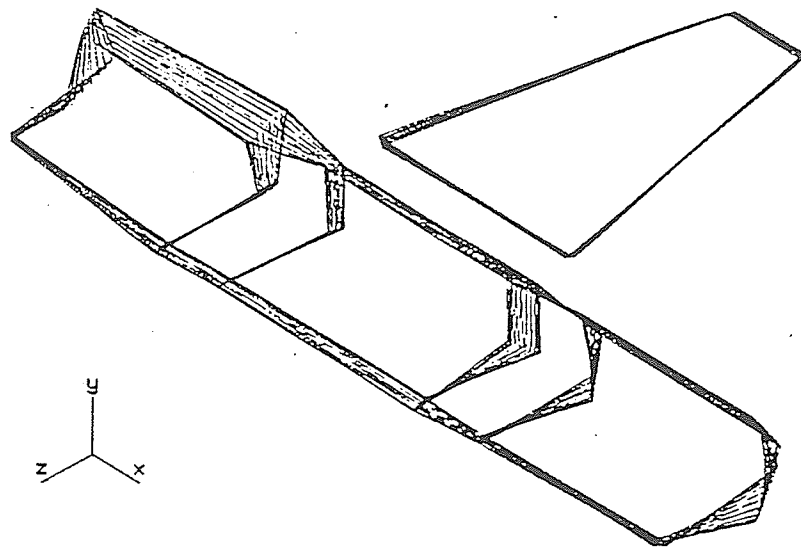
Figure 9.5: Experimental mode shape at 7.7 Hz.

The second natural mode of vibration was detected at 9.4 Hz. In this mode, shown in Figure 9.6, the cutterbar remained relatively straight and oscillated as a rigid body. However, the headbeam oscillated in both the Z and Y directions.

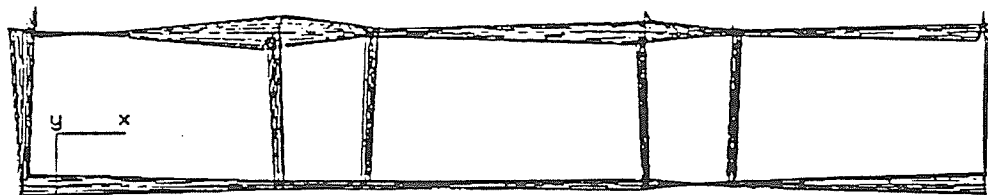
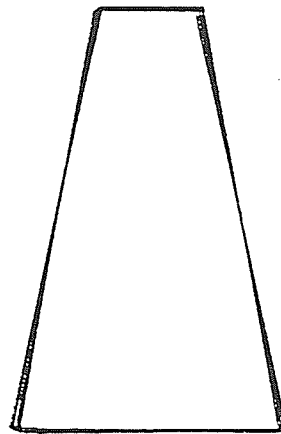
The ends of the headbeam oscillated out of phase with each other (one end moved upward while the other moved downward). As the ends of the headbeam moved in the vertical direction, they also rotated. This rotation reduced the oscillation of the cutterbar in the vertical direction.

The headbeam also oscillated in the Z direction. The right end remained relatively still. However, the middle sections oscillated greatly in the Z direction. From the plan view, the headbeam resembled a gently sloping "S" curve. The points at the tops of the vertical posts, third and fifth from the left end, experienced the greatest oscillation in the Z direction.

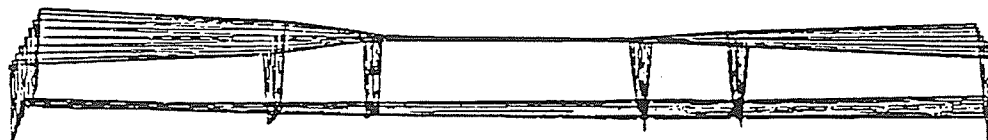
Only slight vibration in the X direction was detected on the tractor. The tractor oscillated side to side at each end. The pivot point of vibration was near the middle of the tractor.



a) isometric view



b) plan view



c) front view

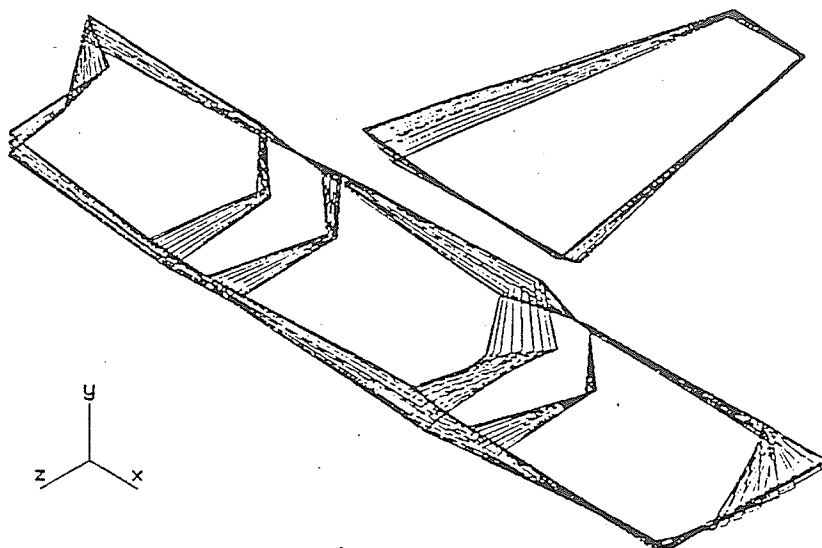
Figure 9.6: Experimental mode shape at 9.4 Hz.

The third natural mode of vibration, shown in Figure 9.7, was detected at 10.5 Hz. Both the cutterbar and the headbeam vibrated in the Y and Z directions.

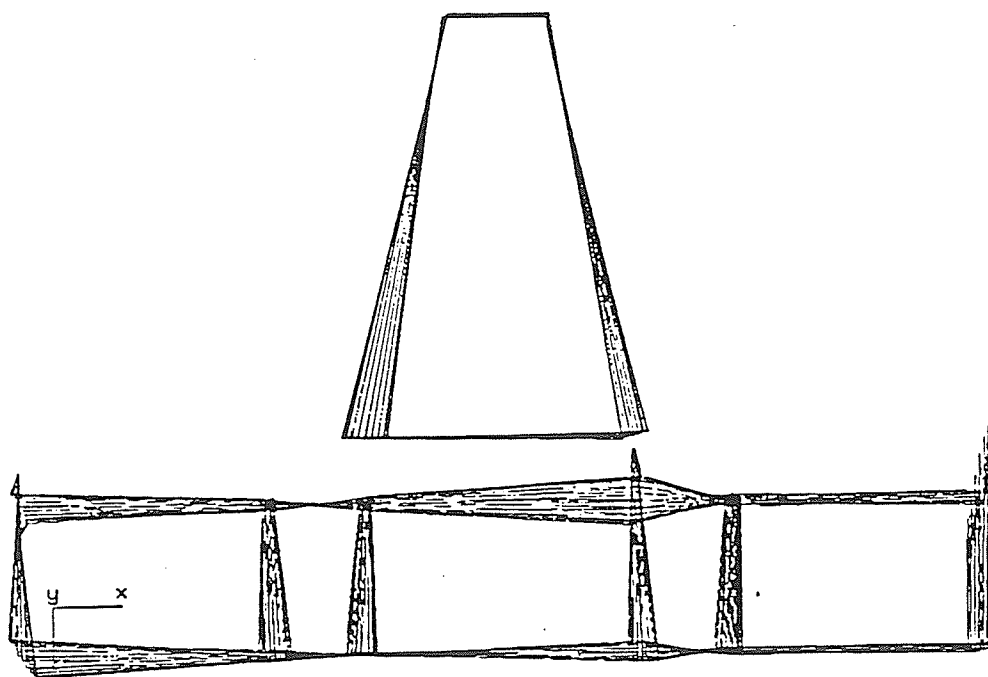
The cutterbar remained relatively still at its ends, but oscillated in the vertical direction near its middle. The greatest vertical motion of the headbeam was detected at the right end while the left end remained still. Oscillation in the Y direction can be studied from the front of the swather. The cutter bar oscillated from being concave up to concave down. In the case of the headbeam, the right hand end remained still while the left end experienced the greatest vertical motion.

The cutterbar and headbeam vibrated in unison in the Z direction. They both resembled gently sloping "S" curves. However, in this mode, the left end remained relatively static. Again, a large response was measured at the top of the vertical post third from the left end. This may have been due to slight shock loading at the connection pin interface.

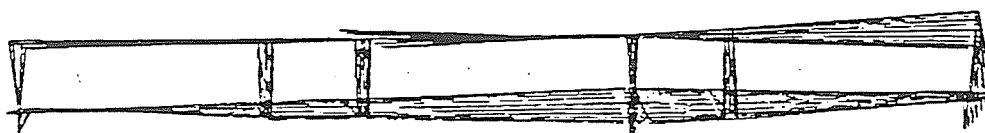
The front portion of the tractor oscillated in the X direction at this frequency. Very little motion was detected at the rear of the swather tractor.



a) isometric view



b) plan view



c) front view

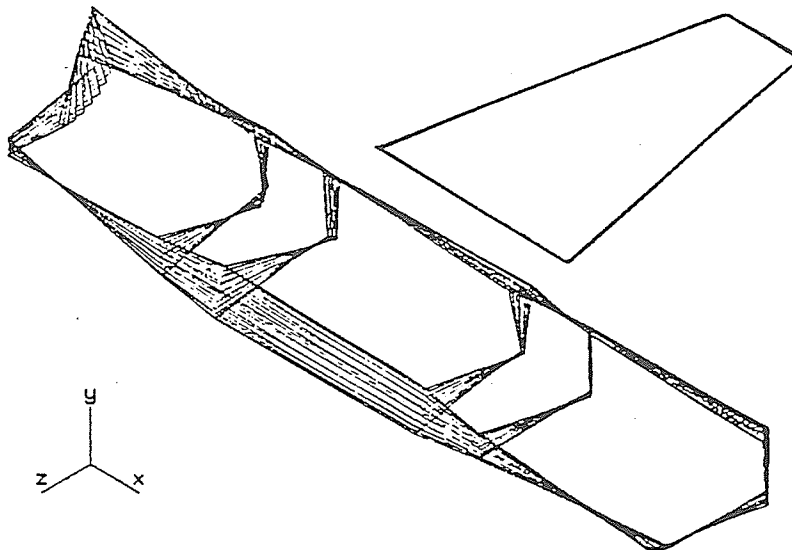
Figure 9.7: Experimental mode shape at 10.5 Hz.

The final experimental natural mode of vibration was detected at 12.5 Hz. Figure 9.8 shows that the majority of the motion was experienced by the cutterbar.

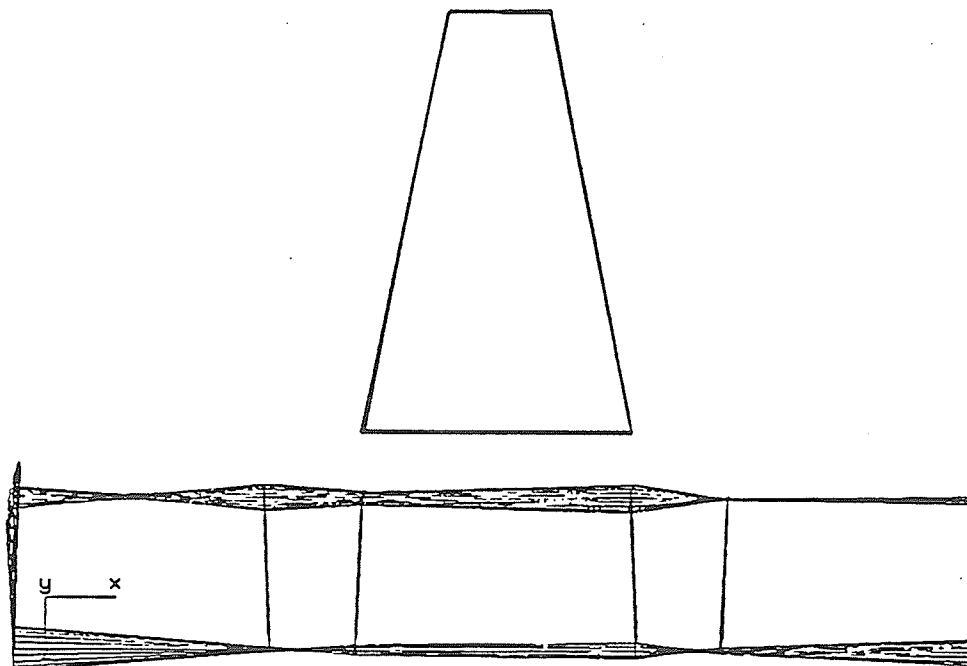
The ends of the cutterbar remained still while the middle oscillated vertically. As with the third mode, the cutterbar oscillated between being concave downward and concave upward. Unlike the previous mode, the left end of the headbeam remained relatively still while the right end vibrated in the Y direction.

Again, the cutterbar and headbeam vibrated in the Z direction in unison. The ends of the swather moved in phase so that both ends moved forward (or back) and the middle sections moved back (or forward).

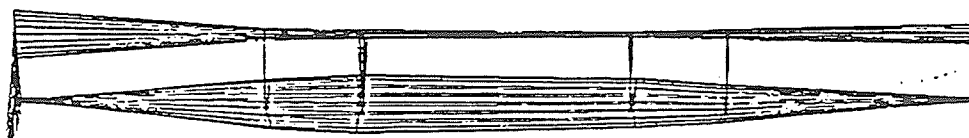
Very little motion of the tractor was detected at this frequency. It remained relatively still at both ends.



a) isometric view



b) plan view



c) front view

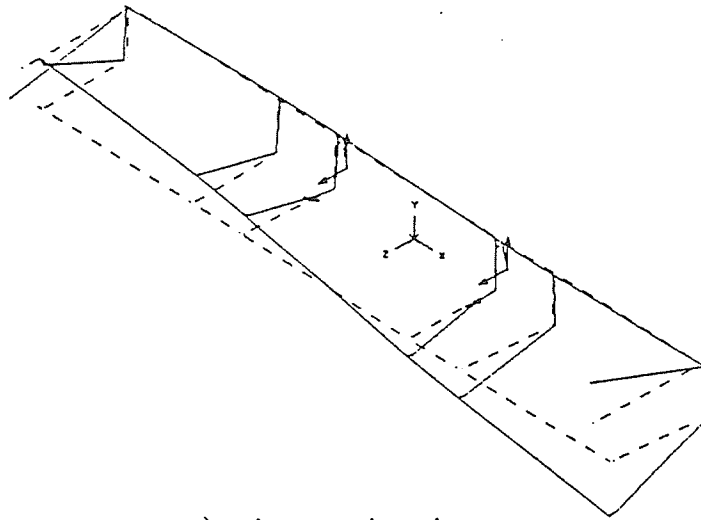
Figure 9.8: Experimental mode shape at 12.5 Hz.

9.4. Modified Analytical Model

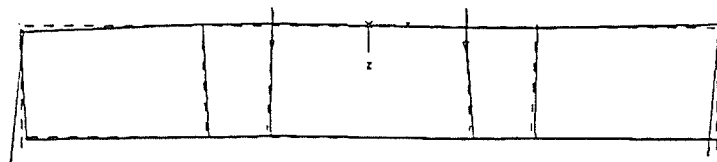
The modified analytical model predicted the four natural frequencies to occur at 7.6, 9.1, 10.6 and 13.1 Hz. In general, the mode shapes were similar to the experimental results.

The mode shape at the first natural frequency is shown in Figure 9.9. The cutterbar acted as a "teeter-totter" and remained relatively straight. The vertical posts remained stationary and the duckfeet moved to allow the cutterbar to oscillate.

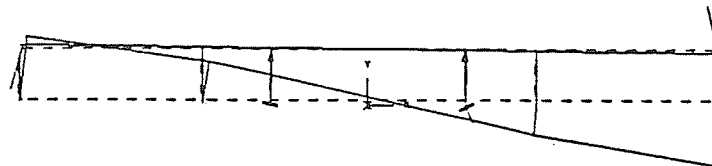
The headbeam also remained relatively straight. However, it vibrated slightly in the Y and Z directions. The headbeam followed the vertical motion of the cutterbar. One end moved down as the opposite end moved up. The motion of the headbeam in the Z direction resembled that of a gently sloping "S" curve. The boundary conditions in the modified model allowed the headbeam to vibrate in the Z direction.



a) isometric view



b) plan view

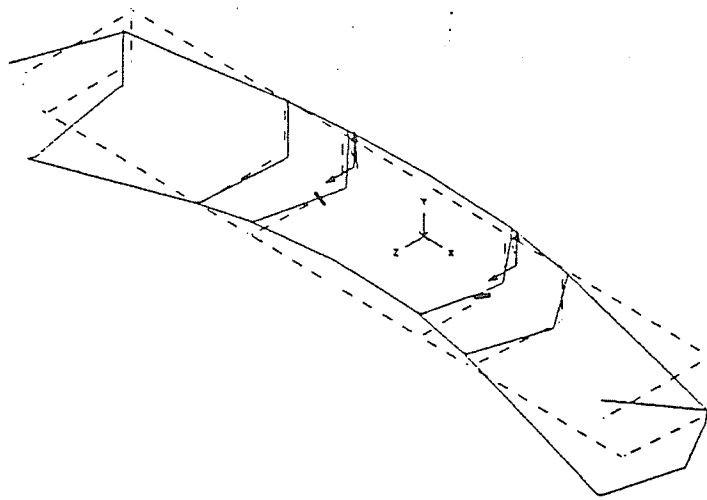


c) front view

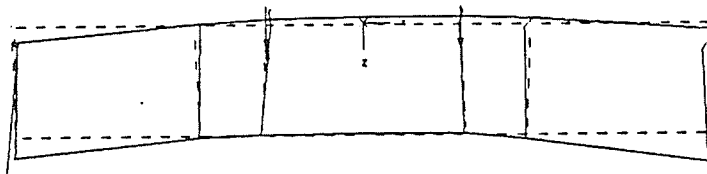
Figure 9.9: Modified finite element mode shape at 7.6 Hz.

The second natural mode of vibration was predicted to occur at 9.1 Hz. Figure 9.10 shows that the majority of the displacement was predicted to occur at the left end of the headbeam. The ends of the headbeam moved vertically downward in phase with each other. Likewise, the ends of the cutterbar also moved vertically downward. As the ends of the header moved downward, the centres of the headbeam and the cutterbar were flexed upward. This gave both the headbeam and the cutterbar a convex upward appearance.

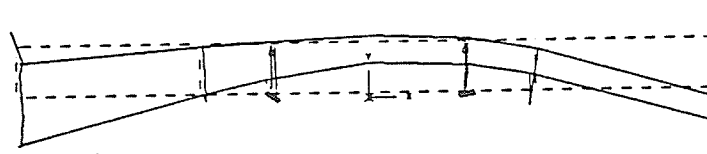
The ends of the header also vibrated in the Z direction. The ends of the headbeam and the cutterbar were in phase with each other. However, the cutterbar was displaced most. The centre of the cutterbar remained relatively stationary while the headbeam experienced larger deflections near its centre.



a) isometric view



b) plan view



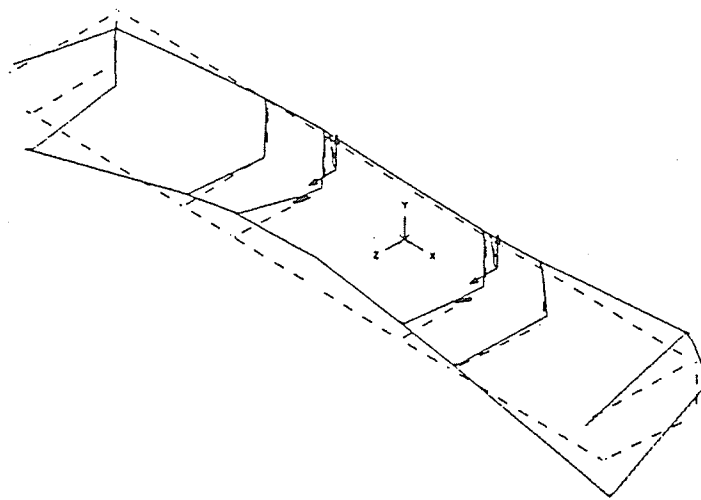
c) front view

Figure 9.10: Modified finite element mode shape at 9.1 Hz.

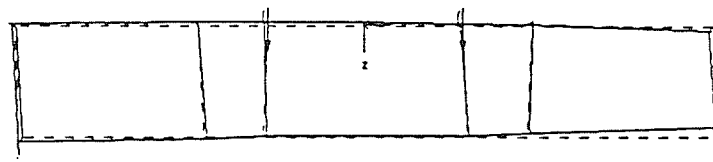
The modified analytical model predicted that the third natural mode of vibration would occur at 10.6 Hz. At this natural frequency, Figure 9.11 shows that both the cutterbar and the headbeam oscillated in the Y and Z directions.

The ends of the headbeam oscillated vertically out of phase. One end of the headbeam moved upward while the opposite end moved downward. Unlike the headbeam, the ends of the cutterbar oscillated vertically in phase. The cutterbar then oscillated between being concave upward and concave downward.

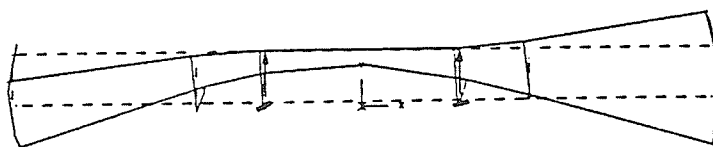
In the Z direction, similar ends of the cutterbar and headbeam moved in opposing directions. The ends of the headbeam and cutterbar moved together then moved apart.



a) isometric view



b) plan view



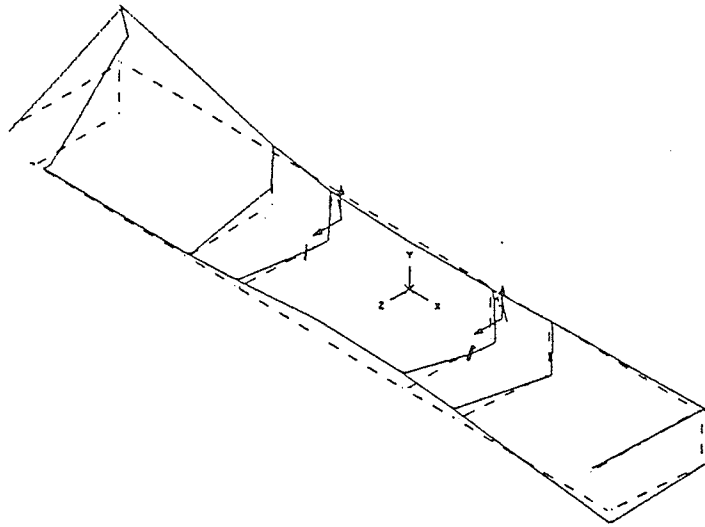
c) front view

Figure 9.11: Modified finite element mode shape at 10.6 Hz.

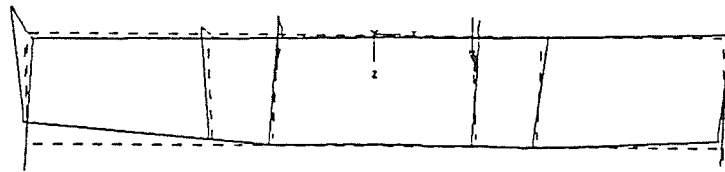
The modified analytical model predicted that the fourth natural mode of vibration would occur at 13.1 Hz. Figure 9.12 shows that the right end of the headbeam experienced a large magnitude of vertical upward motion while the left end moved upward only slightly.

As with the third mode, the cutterbar was flexed upward in the centre and downward at its ends. The cutterbar then appeared to be concave down then switch to being concave up.

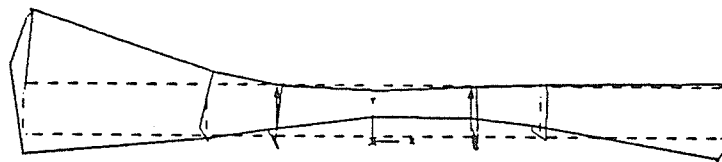
In the Z direction, the left end of the cutterbar and headbeam moved in phase while the right end moved opposite each other. The ends of the cutterbar moved in phase with each other while the ends of the headbeam were out of phase with each other.



a) isometric view



b) plan view



c) front view

Figure 9.12: Modified finite element mode shape at 13.1 Hz.

9.5. Predicted Dynamic Responses

The displacements of the four connection points due to the applied dynamic loads were predicted. The speed of the dynamic loads varied from 7.0 Hz to 14.0 Hz with 0.1 Hz increments. Therefore, the displacements were predicted at 70 different drive frequencies. The displacements in metres were plotted as a function of the drive frequency in Hz.

Figure 9.13 plots the displacements of the connection points on the left side of the header. Likewise, Figure 9.14 illustrates the displacements of the connection points on the right side of the header. The legend in each figure designates the direction as a numerical value. The values 1, 2, and 3, indicate the three orthogonal directions X, Y, and Z, respectively.

In both displacement plots, there is very little evidence of a resonant mode near 7.7 Hz. However, obvious peaks are present near the 9.4, 10.5, and 12.5 Hz resonant frequencies. The maximum deflection was predicted in the vertical direction at the bottom connection point on the left side of the swather, that is, at node 25. The maximum predicted deflection occurred when the drive frequency was about 9.1 Hz. Relatively little displacement was predicted at the higher and lower natural frequencies, 7.7 and 12.5 Hz.

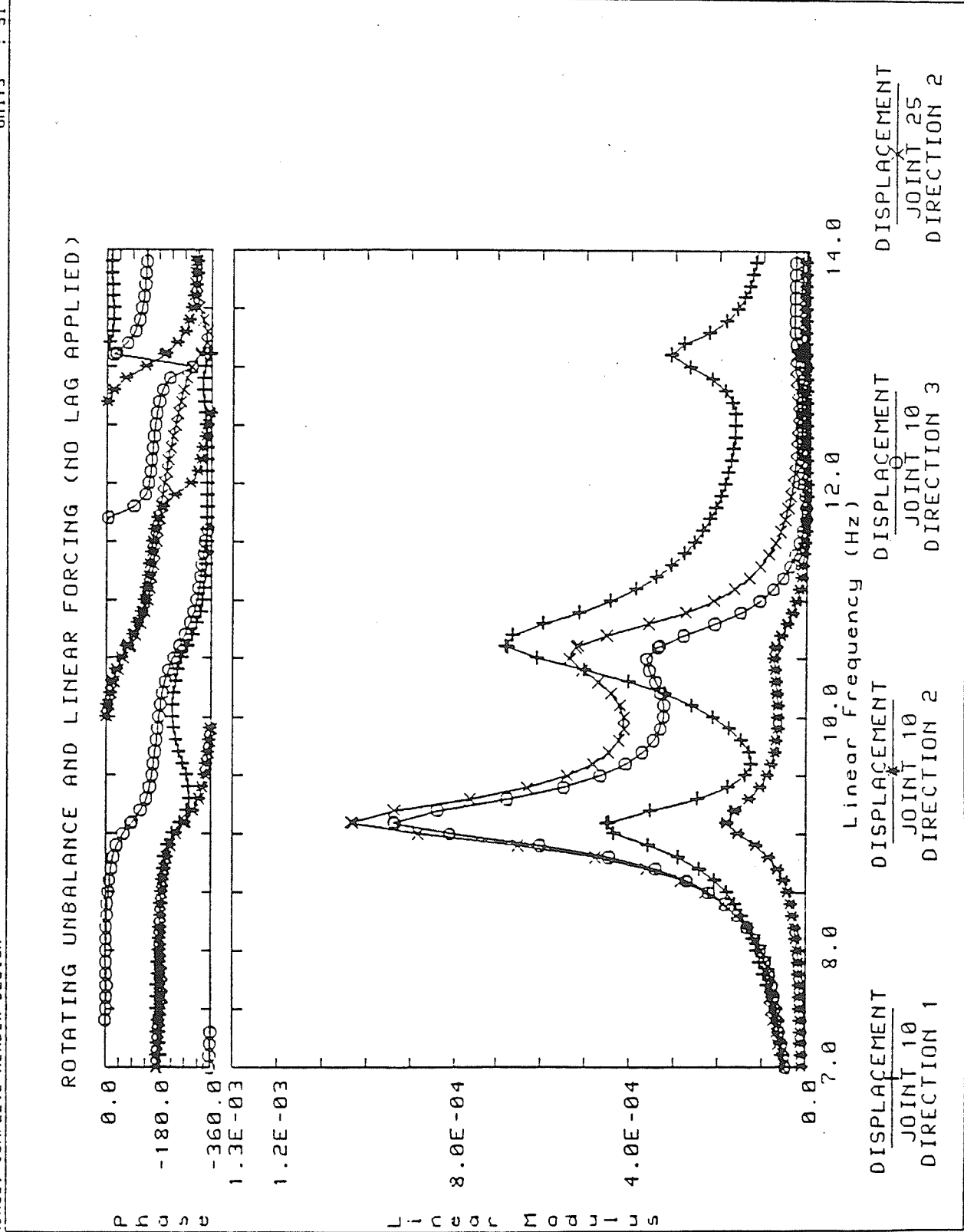


Figure 9.13: Displacements of connection points on the left side of the header.

ROTATING UNBALANCE AND LINEAR FORCING (NO LAG APPLIED)

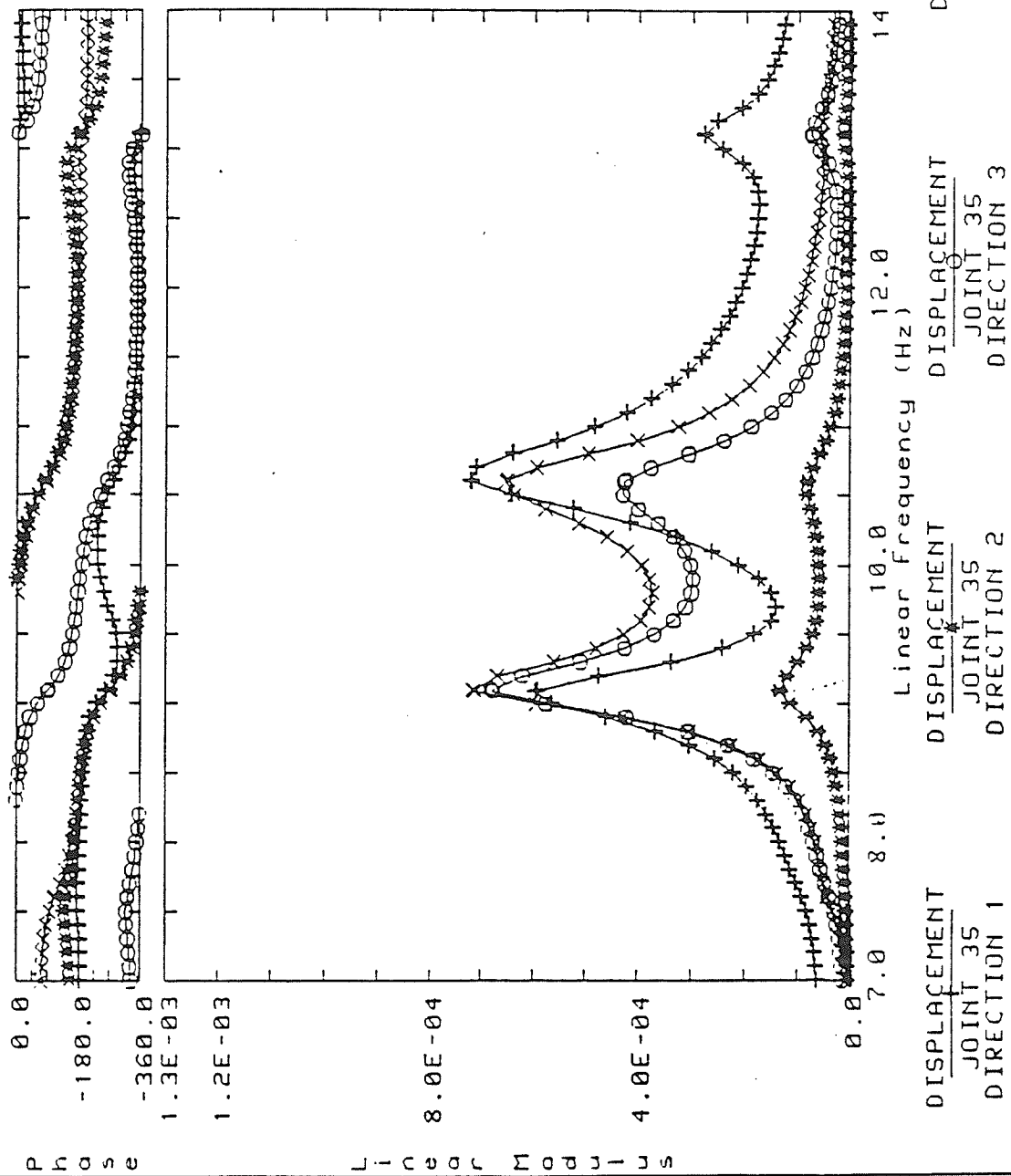


Figure 9.14: Displacements of connection points on the right side of the header.

10. Discussion

10.1. Introduction

Short beams were modelled at both ends of the cutterbar. There were two reasons for their simulation. First, the centroidal axes of the duckfeet on the ends of the header were located so that the cutterbar would be shorter than the headbeam. In addition, the end duckfeet were connected to the cutterbar with a rigid beam element. SDRC (1980) indicated that a lumped mass could not be located at the end of a rigid link. Therefore, the short beams were created to achieve the correct header length and to allow the application of a lumped mass at the ends of the header.

Great care was exercised when the accelerometer was relocated on the structure. However, one particular problem still arose. All of the machine components on the swather were not bolted rigidly or welded to the supporting structure. In one case, the draper rested on the rear of the cutterbar and was retained by two clamps. When the accelerometer was attached to the cutterbar, it was accidentally placed on top of the draper. Not being rigidly fastened, the draper lightly impacted the cutterbar. The impacts were detected by the accelerometer and caused signal overloads. From that point onward, the location of the accelerometer was selected more carefully.

The ultimate goal of this research was to create a computer model that could be used as a design tool. The model was created, then an experimental procedure was completed to determine if the computer model was correct. It was found that a few adjustments could be made to improve the accuracy of the model. The adjustments were completed and the modified model was then used to suggest a design modification.

The accuracy of the analytical model was gauged by its ability to predict the experimental results. Two basic comparisons were made to judge the accuracy of the model. First, the natural frequencies were compared and, second, the corresponding mode shapes were compared. The procedures indicated by Ewins (1984) were used to compare the analytical and experimental values.

The comparison of natural frequencies can be completed easily by plotting the measured natural frequencies versus the predicted natural frequencies. Ideally, all data points should lie on a line with a positive 45 degree slope.

The comparison of the mode shapes is considerably more difficult because more information must be analyzed. The easiest and most revealing technique is to compare the graphical plots of each displaced mode shape. Differences in the mode shapes become very apparent.

10.2. Natural Frequencies

Table 10.1 lists the four initial analytical and experimental natural frequencies. Figure 10.1 shows a plot of these tabular values. The initially predicted and experimental values were very similar. A maximum of approximately 5 percent error existed between the second experimental and analytical natural frequencies. The errors in the other natural frequencies were considerably less.

Table 10.1: Natural frequencies obtained from the initial analytical model and the experimental procedures.

Finite Element Nat. Frequency (Hz)	Experimental Nat. Frequency (Hz)	Error (%)
7.6	7.7	1.3
8.9	9.4	5.3
10.6	10.5	1.0
12.5	12.5	0.0

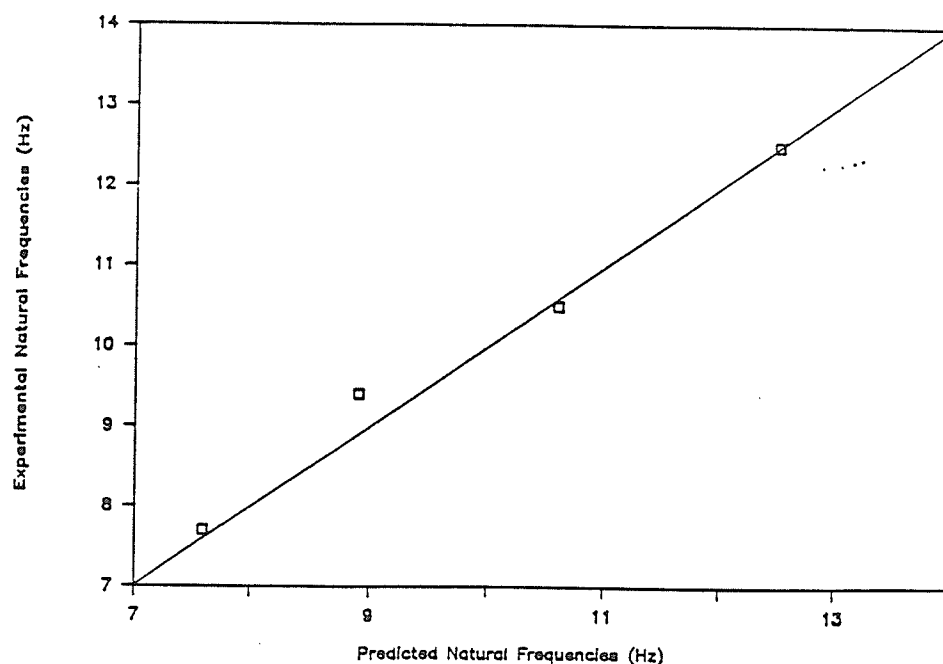


Figure 10.1: Measured versus finite element natural frequencies. (initial model)

The modified analytical and the experimental natural frequencies are listed in Table 10.2 below. The error between the two sets of results is less than five percent for all natural frequencies. Figure 10.2 is a plot of the experimental natural frequencies versus the analytical values. The data points are randomly scattered close to the ideal line.

Table 10.2: Natural frequencies obtained from the modified analytical model and the experimental procedures.

Finite Element Nat. Frequency (Hz)	Experimental Nat. Frequency (Hz)	Error (%)
7.6	7.7	1.3
9.1	9.4	3.2
10.6	10.5	1.0
13.1	12.5	4.8

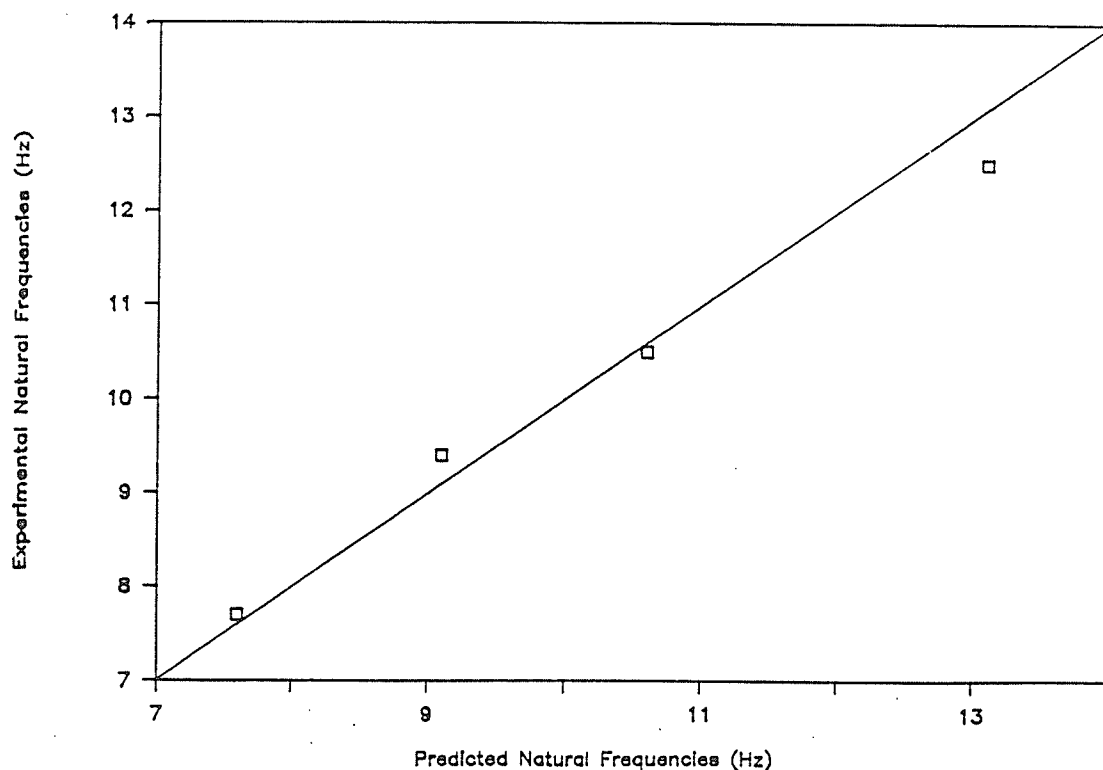


Figure 10.2: Measured versus analytical natural frequencies (modified analytical model).

10.3. Mode Shapes

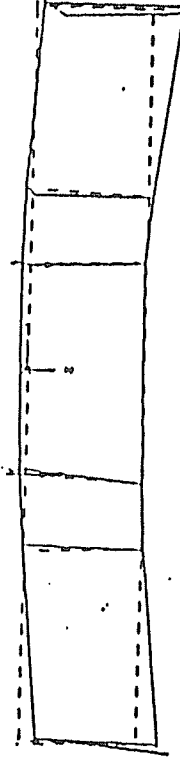
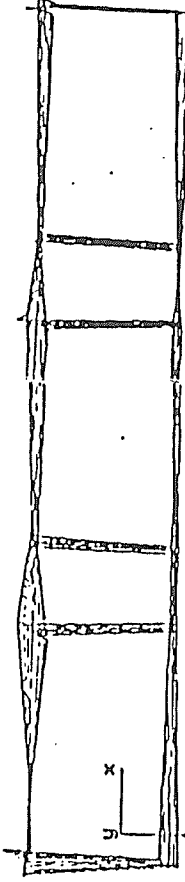
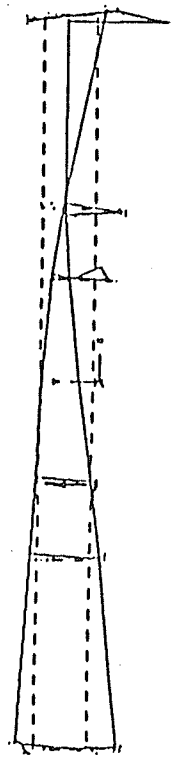
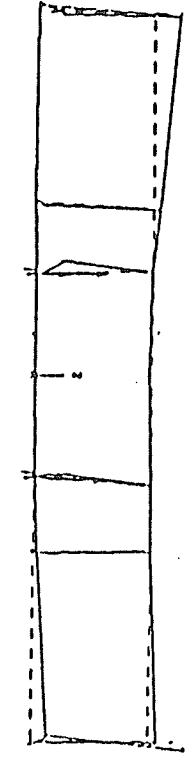
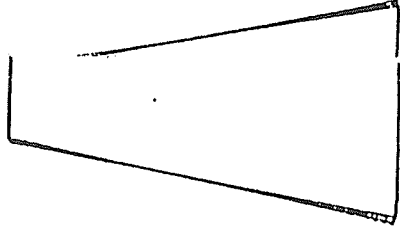
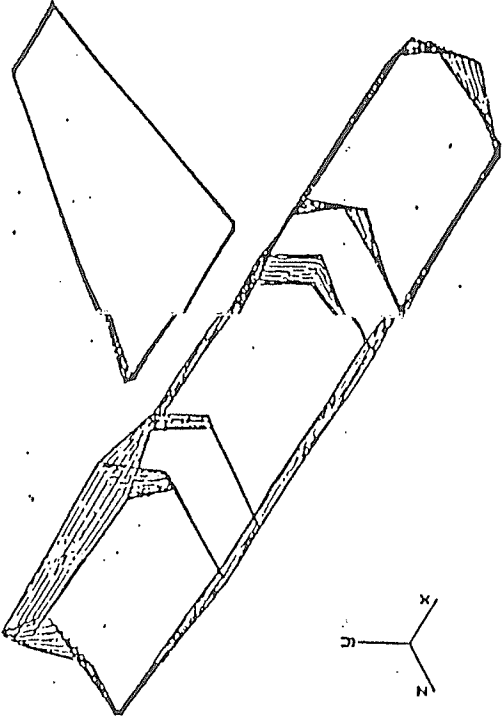
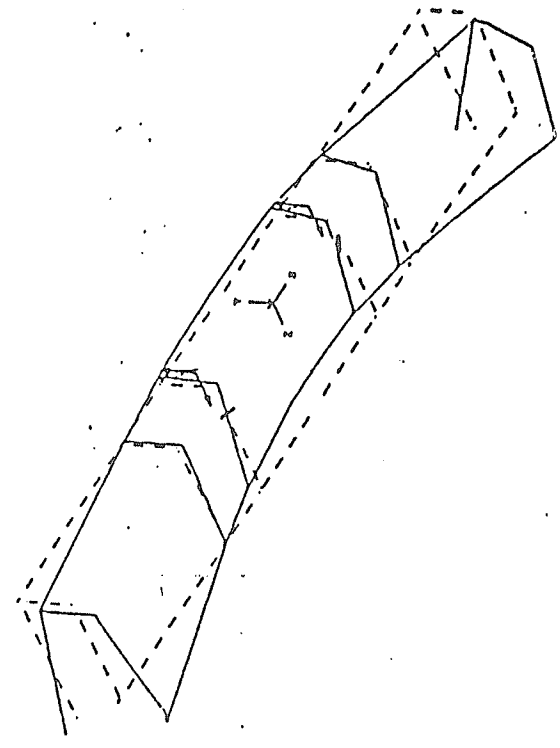
The predicted and experimental mode shapes are discussed individually beginning with the first mode at 7.7 Hz and continuing to the fourth mode at 12.5 Hz. Both sets of analytical mode shapes have been compared to the experimental mode shapes simultaneously.

10.3.1. Mode 1

In general, both analytical mode shapes resemble the experimental mode shape at 7.7 Hz. All three plots shown in Figure 10.3 indicate that the cutterbar acts as a "teeter-totter" with its fulcrum near the centre of the header.

However, the initial analytical results predict that the headbeam would also undergo vertical displacements in phase with the cutterbar. This is not present in the experimental mode shape or the mode shape predicted by the modified model. The headbeam seems to be too flexible in the initial analytical model.

Also, the displacements in the Z direction at the upper connection points are not similar. The experimental and modified analytical mode shapes indicate some motion in the Z direction while this motion is prohibited by the boundary conditions imposed in the initial analytical model. The initial boundary conditions at the upper connection points were too restrictive.



a) initial analytical model, 8.9 Hz

b) experimental results, 9.4 Hz

c) modified analytical model, 9.1 Hz

Figure 10.4: Mode shapes for the second mode of vibration.

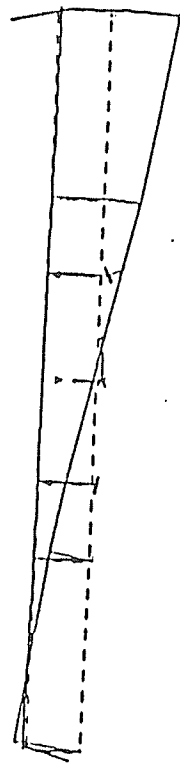
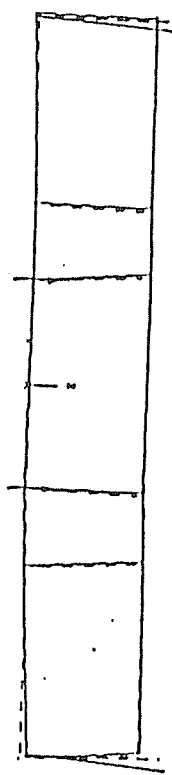
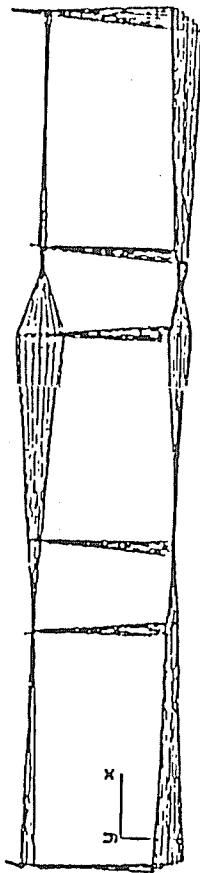
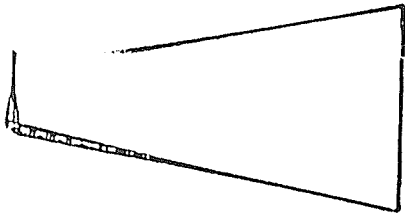
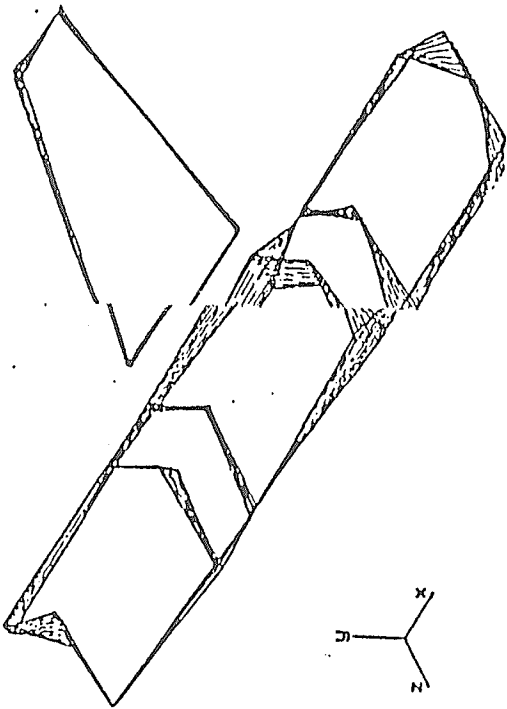
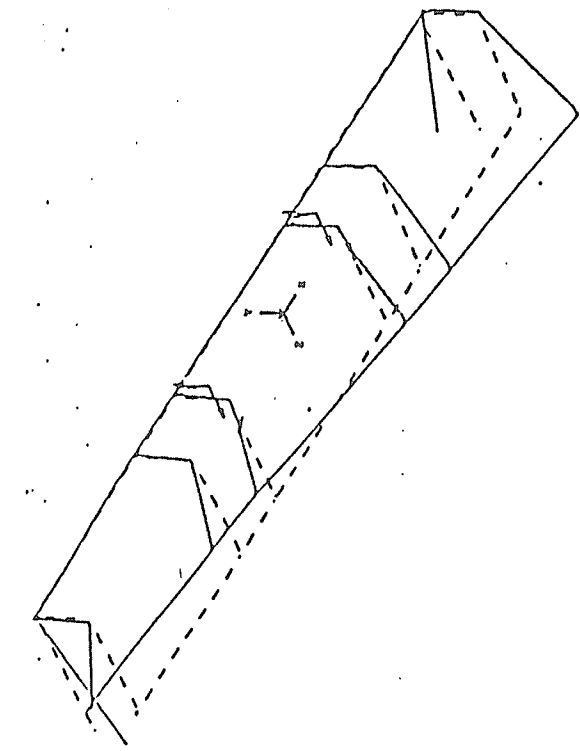
The new boundary conditions and degree of fixity used in the modified analytical model allow the headbeam more freedom in the Z direction. The motion of the headbeam predicted by the modified analytical model more closely resembles the experimental mode shape than the initial analytical model. The modified model is more accurate than the initial analytical model.

10.3.2. Mode 2

Again, all three mode shapes are generally comparable. Figure 10.4 shows the predicted and experimental mode shapes. The experimental procedures determined that this mode shape occurred at 9.4 Hz.

Differences between the three mode shapes are apparent. The cutterbar in the initial predicted mode shape oscillates in the vertical direction with the motion of the opposing ends being out of phase with each other. The experimental mode shape indicates that the cutterbar remains relatively stationary. Finally, the modified analytical model predicted the cutterbar would flex near its midpoint.

The initial analytical model predicts the motion of the headbeam in the vertical direction more accurately than the modified model. The initial model, similar to the experimental results, predicts that the ends of the headbeam will vibrate out of phase. The modified model predicted that the ends of the headbeam will oscillate in phase.



a) initial analytical model, 7.6 Hz

b) experimental ^{vs} ults, 7.7 Hz

c) modified analytical model, 7.6 Hz

Figure 10.3: Mode shapes for the fundamental mode of vibration.

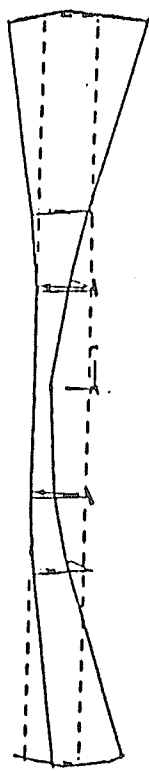
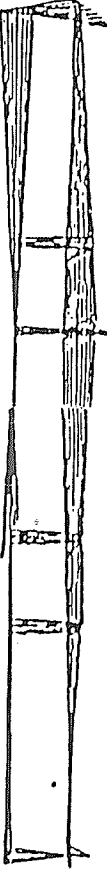
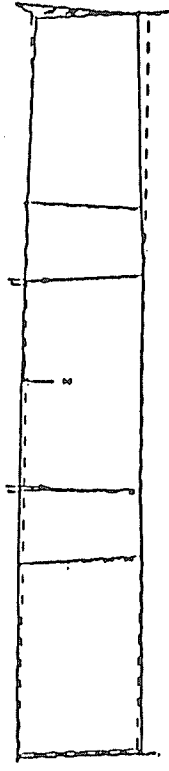
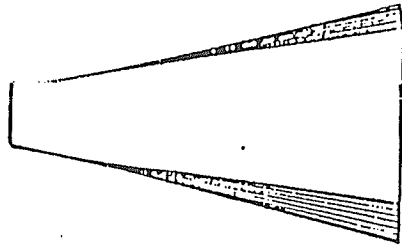
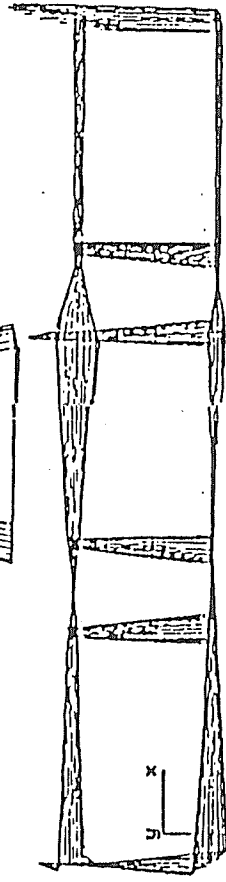
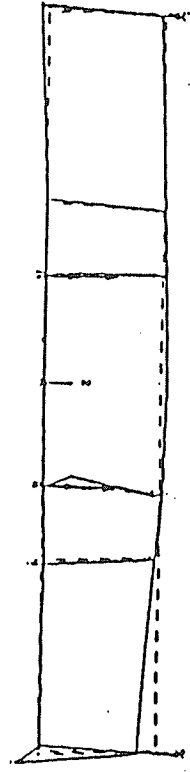
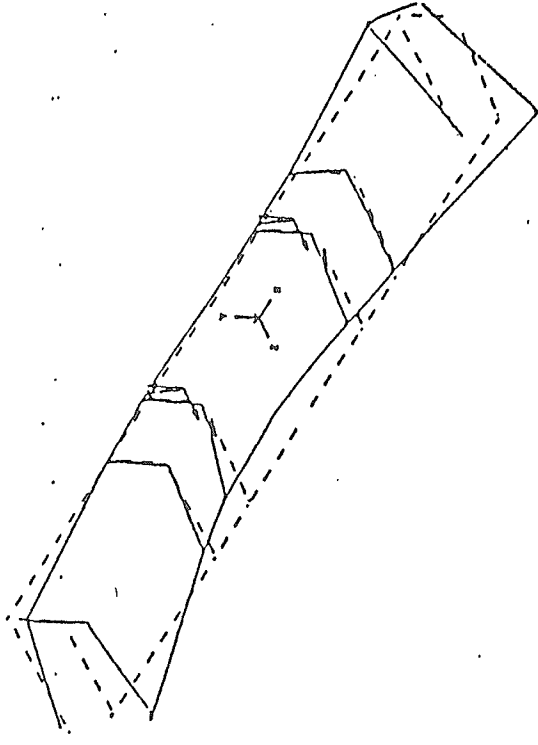
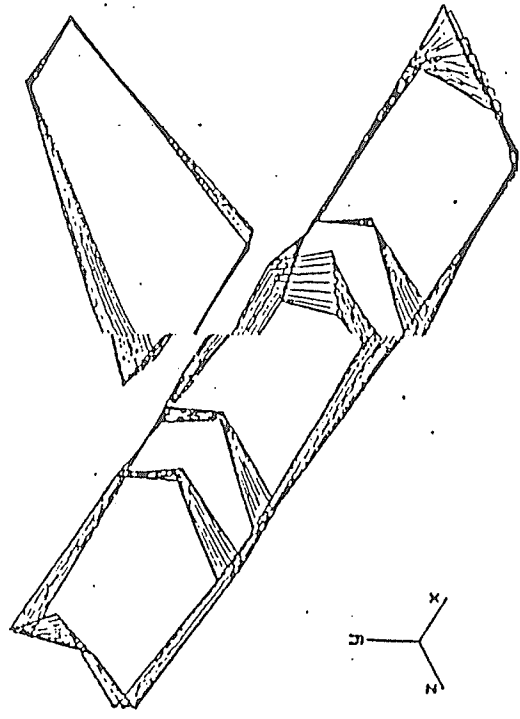
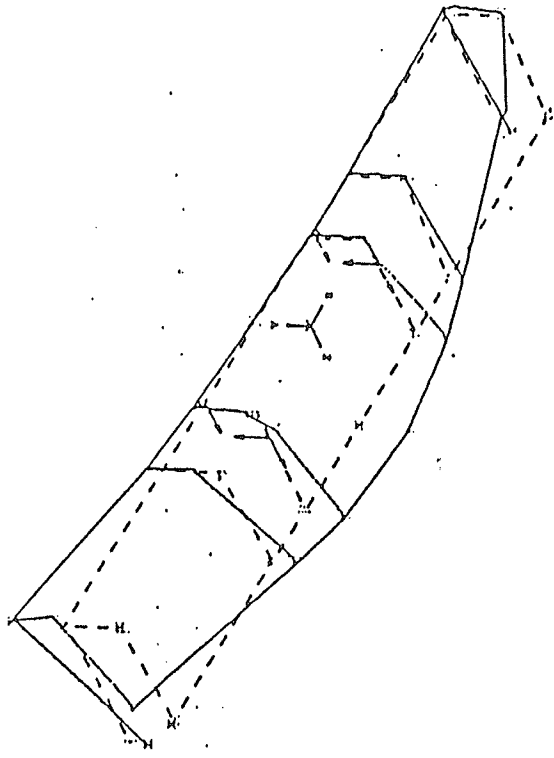
The motion of the headbeam in the Z direction also differs between the plots. The initial analytical model predicts little displacement while the experimental and modified analytical mode shapes resemble a crescent-shaped displacement. The ends of the headbeam move forward in phase while the centre of the headbeam is displaced rearward. The modified boundary conditions allow the headbeam increased mobility in the Z direction and improve the legitimacy of the analytical results.

10.3.3. Mode 3

Figure 10.5 shows the mode shapes that were determined for the third mode of vibration. The third experimental natural frequency was 10.5 Hz.

There are two discrepancies between the initial analytical model and the experimental results. The experimental results show that the right end of the headbeam remains still while the left end experiences a large magnitude of vertical oscillation. The opposite of the results from the analytical model is predicted. The model predicts that the right end oscillates vertically while the left end remains still. The predicted displacement of the cutterbar in the vertical direction is very similar to the experimentally determined mode shape.

Experimentally, the motion of the headbeam in the Z direction shows that the upper connection points are displaced. However, the boundary conditions in the analytical model prohibit such displacement.



a) initial analytical model, 10.6 Hz

b) experimental results, 10.5 Hz

c) modified analytical model, 10.6 Hz

Figure 10.5: Mode shapes for the third mode of vibration.

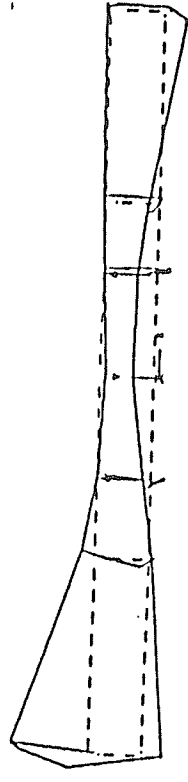
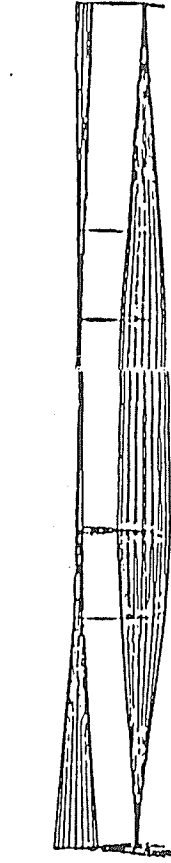
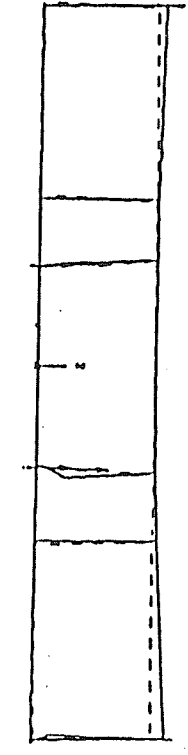
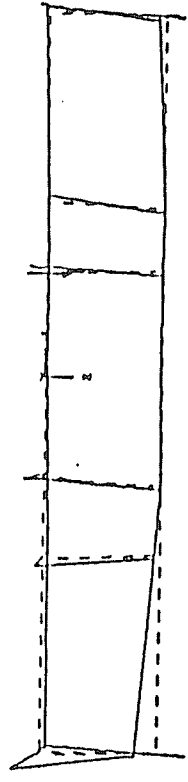
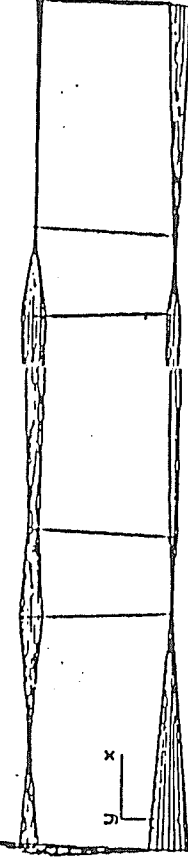
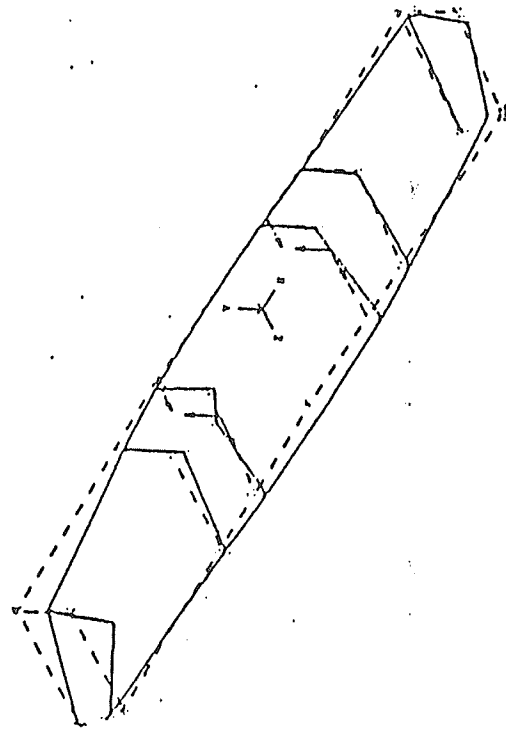
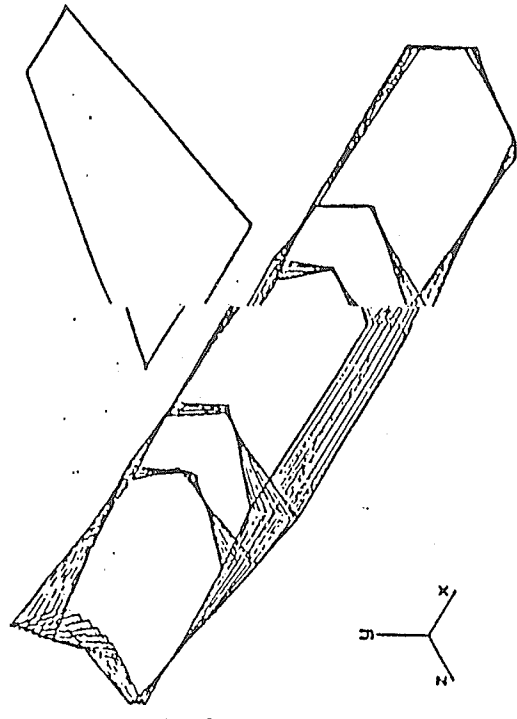
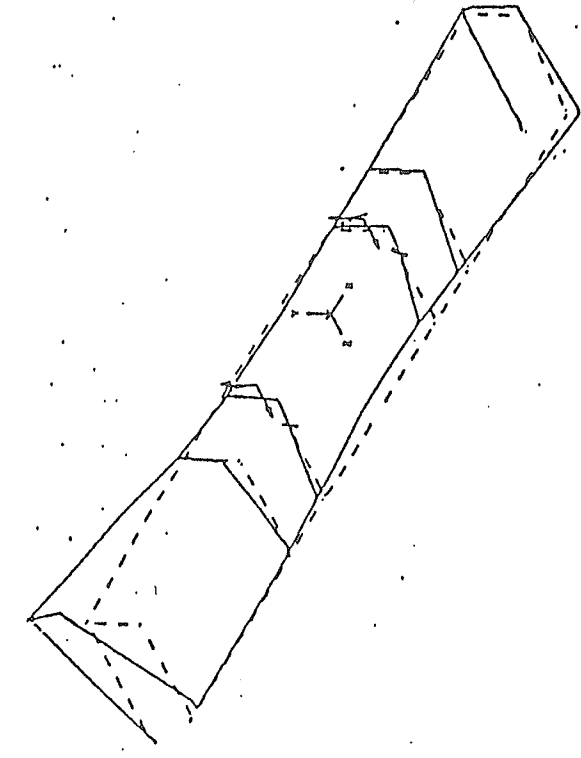
In the third mode of natural vibration, the modified analytical model predicts the motion of the header more accurately than the initial model. Unlike the initial model, the modified analytical model predicts that the left end of the headbeam would experience vertical oscillation. Also, the boundary conditions allow the model to more closely predict the motion of the headbeam in the Z direction.

10.3.4. Mode 4

The mode shapes for the fourth mode are shown in Figure 10.6. All mode shapes are similar. The displacements in the Z direction are very similar in the experiment and the initial analytical model. All three mode shapes indicate that the ends of the headbeam oscillate vertically in unison. The right end of the headbeam experiences more deflection than the left. Also, all three mode shapes indicate that the cutterbar is flexed in the vertical direction near its middle.

The one discrepancy is that the boundary conditions in the initial analytical model do not allow the upper connection points to move in the Z direction. The modified analytical model predicts the motion of the headbeam in the Z direction more accurately than the initial model.

The modified and initial analytical models predicts all four natural frequencies with approximately the same accuracy. However, the mode shapes predicted by the modified analytical model more closely resemble the experimental mode shapes.



a) initial analytical model, 12.5 Hz

b) experimental results, 12.5 Hz

c) modified analytical model, 13.1 Hz

Figure 10.6: Mode shapes for the fourth mode of vibration.

The accuracy of an analytical model can be gauged by its ability to predict the structure's natural frequencies and mode shapes. The predicted mode shapes in the modified analytical model are more accurate than those predicted by the initial model. Therefore, it is apparent that the modified model more accurately simulates the actual structure and Therefore, the modified model was used to predict the response of the structure due to a provided dynamic input.

Originally, the knife drive had been designed to operate in a stationary position at approximately 10 Hz. This speed nearly coincided with an experimentally determined resonant frequency. In addition, operation in the field may not have occurred at 10 Hz. It could be expected to run at a slower speed because a portion of the machine's power would have been used to move the machine along the field. Possibly, the knife drive may have been operating near 9.4 Hz, the second natural frequency.

If the speed of the knife drive had to be changed, it should have been increased rather than decreased. The increased speed should maintain or possibly improve the swather's cutting ability. A relatively wide frequency bandwidth with a low dynamic response is present in the predicted dynamic responses between the third and fourth natural modes of vibration.

If the speed of the knife drive were increased to approximately 12 Hz, the dynamic response amplitude of the swather would decrease from near

0.4 mm to 0.02 mm in the case of node 25 displaced in the Y direction. (These amplitudes are only relative values because the actual amplitude depends on the amount of input force.) A decreased response amplitude should in turn decrease the force magnitude transferred to the tractor by the swather header. The discomfort felt by the operator due to the header should also decrease.

The optimum speed seems to be approximately 12 Hz because the next lower natural frequency occurs at 10.5 Hz. Therefore, the speed of the knife drive would have to decrease by 1.5 Hz before a resonant mode would be encountered. A drive frequency well separated from the natural modes of vibration would be optimal.

11. Conclusions

An initial finite element model consisting of 54 elements and 50 nodes was created to predict the dynamic characteristics of a Model 4025 Versatile self-propelled swather header. The computer model of the swather header was adjusted after experimental data revealed its shortcomings. The adjusted model predicted the natural frequencies to be: 7.6, 9.1, 10.6, and 13.1 Hz. Predominantly, the ends of the header vibrated in the vertical direction in all natural mode shapes.

The experimental procedures also found four natural frequencies in the 7 to 14 Hz frequency bandwidth. The mode shapes were similar to the mode shapes predicted by the finite element model. The maximum variation between the predicted and experimentally determined natural frequencies was less than five percent.

The modified model was then used to predict the swather's responses as the speed of the knife drive varied from 7 Hz to 14 Hz. It was thought that minimal displacements of the connection points between the swather header and the tractor would limit the vibration felt by the operator. The lowest displacement amplitude was predicted when the knife drive was operated in the vicinity of 12 Hz.

Some general conclusions are listed below.

- 1) The modified model predicted the natural frequencies and associated mode shapes with reasonable accuracy.
- 2) The model that has been created, verified, and modified can now be used with confidence as a design tool. It can be used to evaluate the effects of design modifications on the header's dynamic behaviour.
- 3) To decrease the vibration felt by the operator, the knife drive should be adjusted to operate at about 12 Hz.

Appendix A

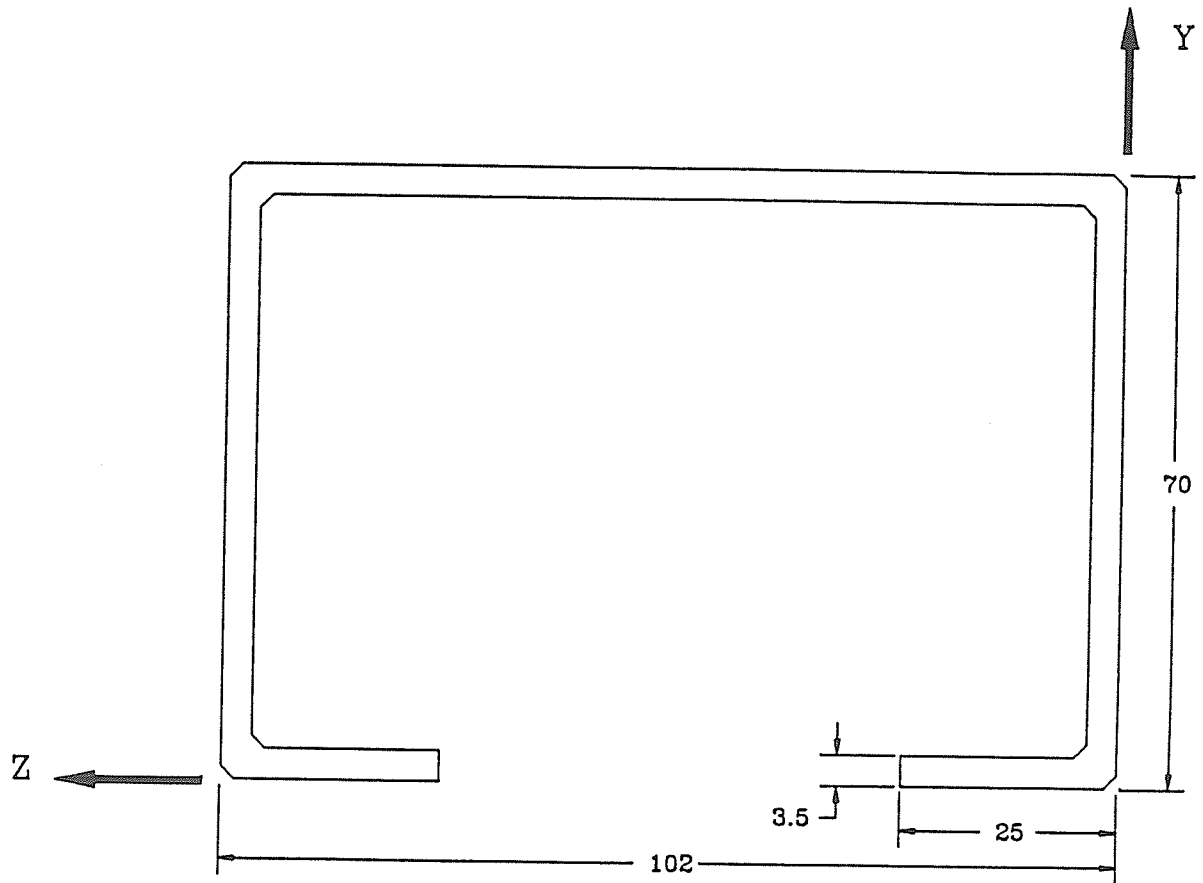
Appendix A

The production drawings that were received from Versatile Farm Equipment Operations displayed in clear detail the cross-sectional dimensions of all members. These dimensions were used to create scaled models of each cross-section. These models were entered into the computer. SDRC (1980) specified that the cross-sectional model lie in the Y-Z plane. In each case the Y axis pointed vertically upward and the Z axis pointed from right to left.

After the models were entered into the computer, the "Beam Cross Section Analysis" sub-program calculated all required physical properties for each beam. In the case of the tapered beams, two different cross-sections were analyzed because they varied from one end to the other. This appendix lists all calculated physical properties for each beam accompanied by a drawing that displays all dimensions.

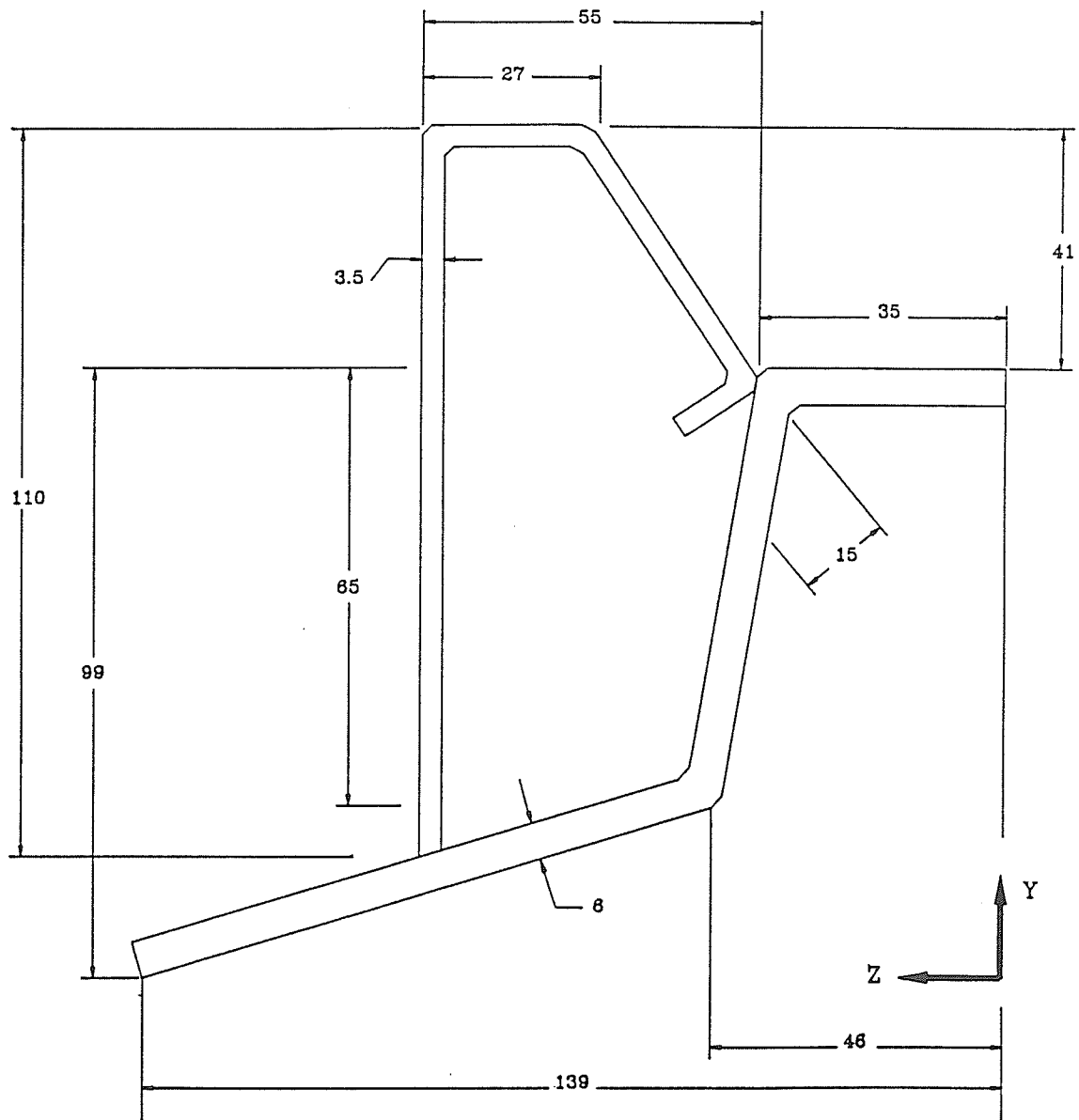
All physical dimensions are listed in millimetres.

Rear of the Duckfeet



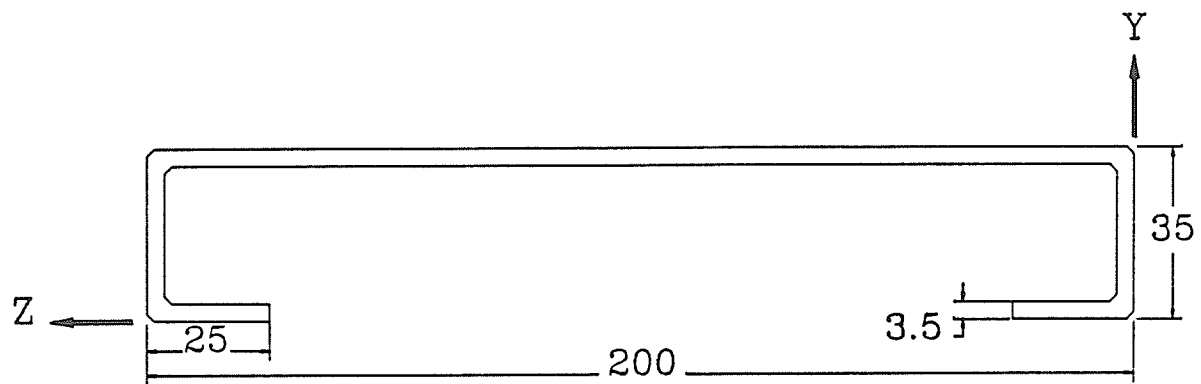
	Y	Z	
Eccentricity	-64.22	0.0	[mm]
Shear ratio	2.32	4.11	
Princ. Mom. Inert.	1.63 E6	6.96 E5	[mm ⁴]
Centroid	41.18	50.80	[mm]
Prod. of Inert.	0.0		[mm ⁴]
Area	971.6		[mm ²]
Torsional Const.	16460.0		[mm ⁴]
Warping Const.	1.95 E9		[mm ⁶]
Rot. of Princ. Axes	0.0		[degrees]

Cutterbar



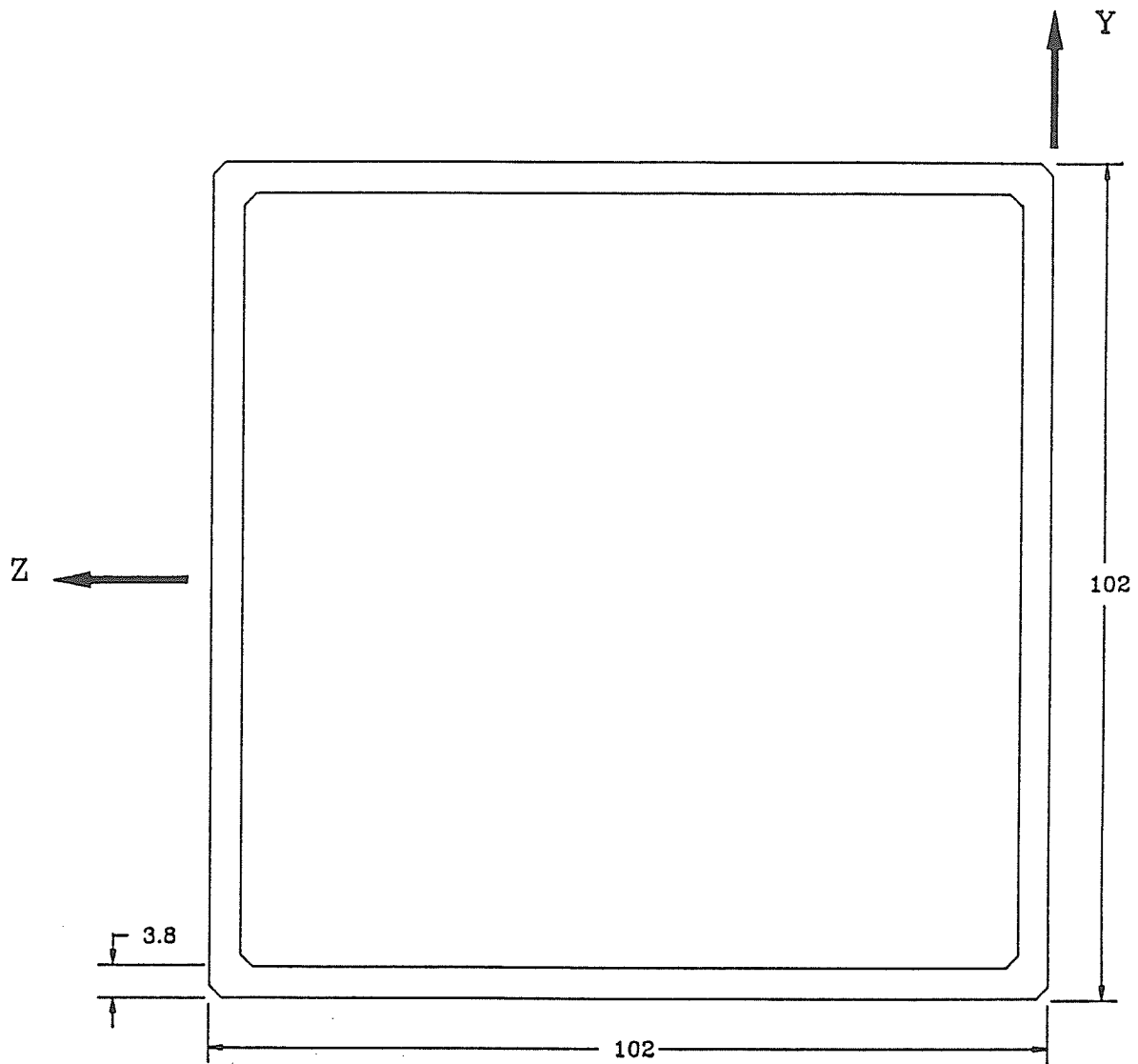
	Y	Z	
Eccentricity	-6.95	11.20	[mm]
Shear ratio	1.80	3.61	
Princ. Mom. Inert.	1.18 E6	3.67 E6	[mm ⁴]
Centroid	65.60	-69.65	[mm]
Prod. of Inert.	-1.15 E6		[mm ⁴]
Area	1843.0		[mm ²]
Torsional Const.	1.41 E6		[mm ⁴]
Warping Const.	3.73 E8		[mm ⁶]
Rot. of Princ. Axes	+ 33.3		[degrees]

Front of Duckfeet

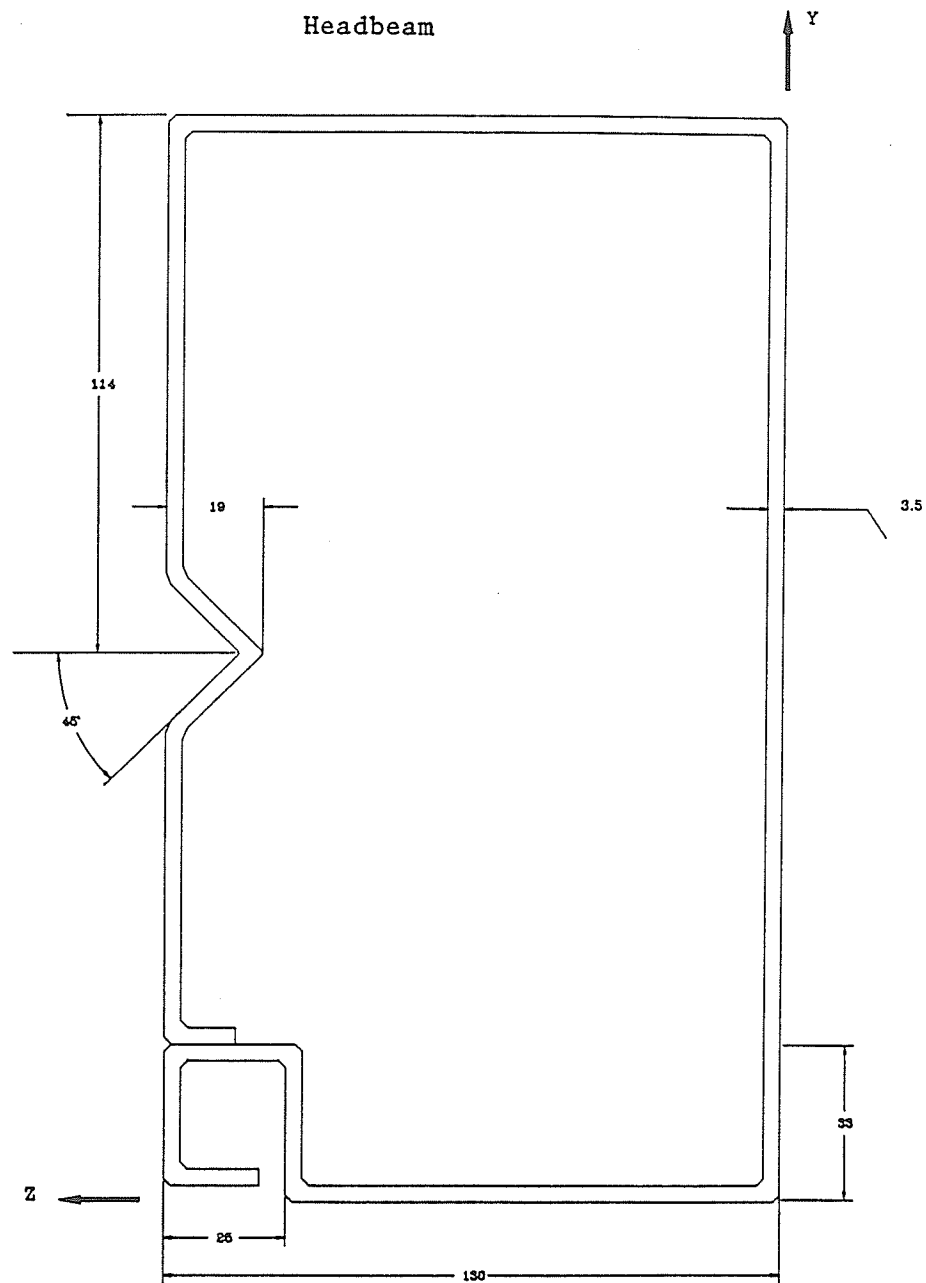


	Y	Z	
Eccentricity	-22.95	0.0	[mm]
Shear ratio	7.52	1.78	
Princ. Mom. Inert.	5.27 E6	1.66 E5	[mm ⁴]
Centroid	25.1	97.75	[mm]
Prod. of Inert.	0.0		[mm ⁴]
Area	1055.0		[mm ²]
Torsional Const.	7688.0		[mm ⁴]
Warping Const.	1.37 E9		[mm ⁶]
Rot. of Princ. Axes	0.0		[degrees]

Square Vertical Post

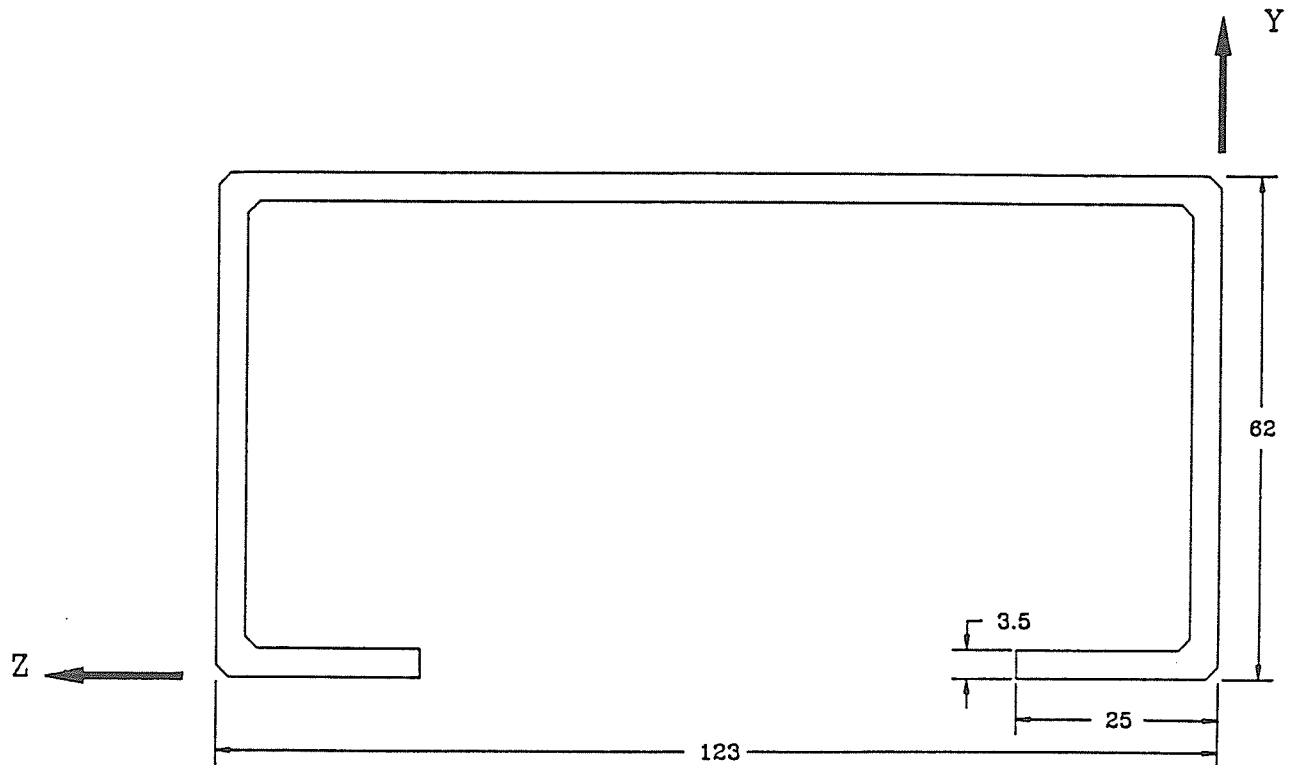


	Y	Z	
Eccentricity	0.0	0.0	[mm]
Shear ratio	2.24	2.24	
Princ. Mom. Inert.	2.38 E6	2.38 E6	[mm ⁴]
Centroid	0.0	50.8	[mm]
Prod. of Inert.	0.0		[mm ⁴]
Area	1490.0		[mm ²]
Torsional Const.	3.67 E6		[mm ⁴]
Warping Const.	2.72 E6		[mm ⁶]
Rot. of Princ. Axes	0.0		[degrees]

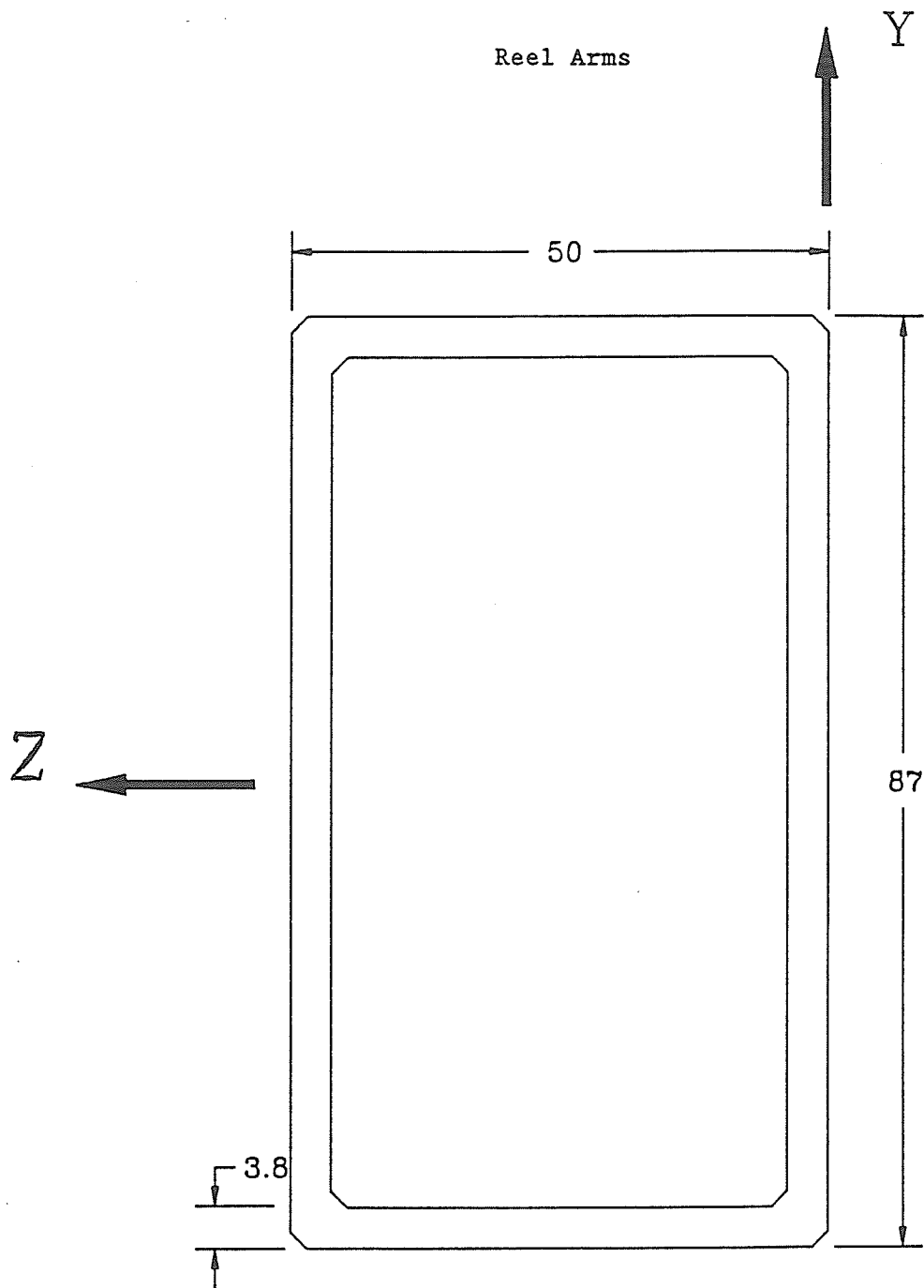


	Y	Z	
Eccentricity	-11.13	7.43	[mm]
Shear ratio	1.87	3.99	
Princ. Mom. Inert.	7.70 E6	1.94 E7	[mm ⁴]
Centroid	109.00	67.65	[mm]
Prod. of Inert.	-6.51 E5		[mm ⁴]
Area	2656.0		[mm ²]
Torsional Const.	1.53 E7		[mm ⁴]
Warping Const.	4.48 E9		[mm ⁶]
Rot. of Princ. Axes	-3.2		[degrees]

Bottom Connection Points



	Y	Z	
Eccentricity	-53.86	0.0	[mm]
Shear ratio	2.89	3.17	
Princ. Mom. Inert.	2.34 E6	5.65 E5	[mm ⁴]
Centroid	38.49	61.39	[mm]
Prod. of Inert.	0.0		[mm ⁴]
Area	993.30		[mm ²]
Torsional Const.	9862.0		[mm ⁴]
Warping Const.	2.13 E9		[mm ⁶]
Rot. of Princ. Axes	0.0		[degrees]



	Y	Z	
Eccentricity	0.0	0.0	[mm]
Shear ratio	1.75	2.79	
Princ. Mom Inert.	1.06 E6	1.85 E6	[mm ⁴]
Centroid	0.0	31.75	[mm]
Prod. of Inert.		0.0	[mm ⁴]
Area	1774.0		[mm ²]
Torsional Const.		2.15 E6	[mm ⁴]
Warping Const.		3.66 E7	[mm ⁶]
Rot. of Princ. Axes		0.0	[degrees]

Appendix B

Appendix B

The following data sheets are copies of the computer printouts received from the minicomputer at the Industrial Technology Centre. All quantities are stated in SI units (kg, m, s).

The printout includes six different sections. These sections are:

- 1) beam descriptions
- 2) node descriptions
- 3) material property descriptions
- 4) section property descriptions
- 5) locations and magnitudes of lumped masses
- 6) restraint descriptions.

The beam descriptions first list the straight beams (constant cross-section from one end to the other). The beam number followed by its length is listed for each beam. Next, for each beam, the fore end (where the beam starts) and the aft end (where the beam ends) are identified by listing the nodes at each of the beam's ends. Finally, the material and section properties for each beam are listed by defining the appropriate code number.

Next, the rigid beams are listed. Like the straight beams, the beams' numbers, lengths and end node numbers are listed. However, the rigid beams have no physical or material properties. Therefore, the slots in the 'Material Code' and 'Section Code' columns are filled with the word 'RIGID'.

Tapered beams (beams with a cross-section different at one end from the other) are listed in a manner similar to that of the straight beams. All columns are the same except for the section codes. There must be two columns because the cross-sections at the ends of the beam are different. The section code must be stated for both the fore end and the aft end.

The node coordinates are then listed in ascending order. The location of each node in space is listed by defining its three-dimensional spatial coordinates.

The material property tables are then listed. The code number identifies a set of values that were used to define a specific beam element. In each list, Young's modulus, the Poisson's ratio, and the density are listed.

The physical property tables are then defined. Again, a code number identifies specific sets of values that are used to define a specific beam cross-section. The area, moments of inertia, shear ratios, torsional constant, warping constant and degree of fixity for each beam cross-section is listed. The eccentricity defines the location of the shear centre from the centroid.

Next the location of the lumped masses are defined by listing the node number at which the mass is applied. For each mass element, the mass in kilograms is listed.

Finally, the restraints are listed. The node that the restraint is applied to is listed first. The direction of restraint for each node is then defined. The directions are identified by a numerical system. Directions 1, 2, and 3 correspond to restraints in the X, Y, and Z directions, respectively.

D A G S

DYNAMIC ANALYSIS OF GENERAL STRUCTURES
STRUCTURAL DYNAMICS RESEARCH CORPORATION

SWATHER TABLE

*** SPACE FRAME ANALYSIS ***

STRAIGHT BEAMS

BEAM	LENGTH	FORE END JOINT	AFT END JOINT	MATERIAL CODE	SECTION CODE	ROTATION ANGLE	TEMP.
5	0.46	11	9	1	3		
6	0.02	9	6	1	3		
7	0.49	12	7	1	3		
8	0.49	13	8	1	3		
9	0.01	23	26	4	5		
10	2.00	22	23	4	5		
11	0.67	21	22	4	5		
12	1.11	20	21	3	5		
13	1.07	1	2	1	4		
14	0.76	2	3	2	4		
15	1.97	3	4	2	4		
16	0.03	4	5	2	4		
18	1.40	4	24	1	17		
27	0.09	9	10	1	17		
32	0.46	36	34	1	3		
33	0.02	34	31	1	3		
34	0.49	37	32	1	3		
35	0.49	38	33	1	3		
36	0.01	47	50	4	5		
37	2.00	46	47	4	5		
38	0.67	45	46	4	5		
39	1.11	20	45	3	5		
40	1.07	1	27	1	4		
41	0.76	27	28	2	4		
42	1.97	28	29	2	4		
43	0.03	29	30	2	4		
45	1.40	29	48	1	17		
54	0.09	34	35	1	17		
17	0.12	8	4		RIGID		
19	0.05	13	16		RIGID		
20	0.08	19	23		RIGID		
21	0.11	7	3		RIGID		
22	0.11	6	2		RIGID		
23	0.05	12	15		RIGID		
24	0.05	11	14		RIGID		
25	0.08	18	22		RIGID		
26	0.08	17	21		RIGID		
44	0.12	33	29		RIGID		
46	0.05	38	41		RIGID		
47	0.08	44	47		RIGID		
48	0.11	32	28		RIGID		

DYNAMIC ANALYSIS OF GENERAL STRUCTURES
SWATHER TABLE

STRAIGHT BEAMS

BEAM	LENGTH	FORE END JOINT	AFT END JOINT	MATERIAL CODE	SECTION CODE	ROTATION ANGLE	TEMP.
49	0.11	31	27		RIGID		
50	0.05	37	40		RIGID		
51	0.05	36	39		RIGID		
52	0.08	43	46		RIGID		
53	0.08	42	45		RIGID		

TAPERED BEAMS

BEAM	LENGTH	FORE END JOINT	AFT END JOINT	TAPER TYPE	MAT. CODE	SECTION CODE FORE AFT	ROT. ANGLE	TEMP
1	0.25	14	25	1	1	8 25		
2	0.83	25	17	1	1	7 24		
3	1.08	15	18	1	1	2 19		
4	1.08	16	19	1	1	2 19		
28	0.25	39	49	1	1	8 25		
29	0.83	49	42	1	1	7 24		
30	1.08	40	43	1	1	2 19		
31	1.08	41	44	1	1	2 19		

JOINT COORDINATES

JOINT	X	Y	Z
1	0.000	0.595	0.017
2	1.067	0.595	0.017
3	1.829	0.595	0.017
4	3.800	0.595	0.017
5	3.833	0.595	0.017
6	1.067	0.486	0.000
7	1.829	0.486	0.000
8	3.833	0.486	0.000
9	1.067	0.462	0.000
10	1.067	0.499	-0.082
11	1.067	0.000	0.000
12	1.829	0.000	0.000
13	3.833	0.000	0.000
14	1.067	0.000	0.051
15	1.829	0.000	0.051
16	3.833	0.000	0.051
17	1.114	0.019	1.127
18	1.782	0.019	1.127
19	3.786	0.019	1.127
20	0.000	0.059	1.196
21	1.114	0.059	1.196
22	1.782	0.059	1.196
23	3.786	0.059	1.196

DYNAMIC ANALYSIS OF GENERAL STRUCTURES
SWATHER TABLE

JOINT	JOINT COORDINATES		
	X	Y	Z
24	3.800	0.595	1.416
25	1.078	0.004	0.300
26	3.800	0.059	1.196
27	-1.067	0.595	0.017
28	-1.829	0.595	0.017
29	-3.800	0.595	0.017
30	-3.833	0.595	0.017
31	-1.067	0.486	0.000
32	-1.829	0.486	0.000
33	-3.833	0.486	0.000
34	-1.067	0.462	0.000
35	-1.067	0.499	-0.082
36	-1.067	0.000	0.000
37	-1.829	0.000	0.000
38	-3.833	0.000	0.000
39	-1.067	0.000	0.051
40	-1.829	0.000	0.051
41	-3.833	0.000	0.051
42	-1.114	0.019	1.127
43	-1.782	0.019	1.127
44	-3.786	0.019	1.127
45	-1.114	0.059	1.196
46	-1.782	0.059	1.196
47	-3.786	0.059	1.196
48	-3.800	0.595	1.416
49	-1.078	0.004	0.300
50	-3.800	0.059	1.196

CODE	MATERIAL PROPERTIES		
	E	POISSON'S	DENSITY
1	21.5E+10	0.290	7.820E+03
2	21.5E+10	0.290	1.523E+04
3	21.5E+10	0.290	3.575E+04
4	21.5E+10	0.290	4.287E+04
8	50.0E+10	0.290	7.820E+03

DYNAMIC ANALYSIS OF GENERAL STRUCTURES
SWATHER TABLE

CROSS-SECTION PROPERTIES								
CODE	AREA	MOMENTS OF INERTIA		SHEAR Y	RATIO Z	TORSION CONSTANT	WARPING CONSTANT	DES. FIX.
		Z	Y					
2	9.80E-04	7.00E-07	1.70E-06	2.33	4.11	1.70E-06	2.00E-09	
3	1.50E-03	2.40E-06	2.40E-06	2.20	2.20	3.70E-06	2.80E-12	
4	3.00E-03	2.00E-05	8.00E-06	1.87	4.00	2.00E-05	5.00E-09	
5	2.00E-03	4.00E-06	2.00E-06	1.80	4.00	2.00E-06	4.00E-10	
7	1.00E-04	5.70E-07	2.40E-06	2.89	3.20	9.90E-09	2.20E-09	
8	9.80E-04	7.00E-07	1.70E-06	2.33	4.11	1.70E-06	2.00E-09	
17	1.78E-03	1.85E-06	1.07E-06	1.75	2.79	2.15E-06	3.66E-11	
19	2.00E-03	2.00E-07	6.00E-06	8.00	1.78	8.00E-09	2.00E-09	
24	2.00E-03	2.00E-07	6.00E-06	8.00	1.78	8.00E-09	2.00E-09	
25	1.00E-04	5.70E-07	2.40E-06	2.89	3.20	9.90E-09	2.20E-09	

CODE	ECCENTRICITY	
	Z	Y
2	0.000	-0.070
4	0.008	-0.020
5	0.020	-0.007
7	0.000	-0.060
8	0.000	-0.070
19	0.000	-0.030
24	0.000	-0.030
25	0.000	-0.060

LUMPED MASS AND INERTIA							
JOINT	MASS	J(X)	J(Y)	J(Z)	J(XY)	J(XZ)	J(YZ)
24	1.60E+02						
5	1.87E+02						
26	2.72E+01						
50	2.72E+01						
30	4.17E+01						
48	1.70E+02						

TOTAL LUMPED WEIGHT/MASS = 6.132E+02

GRAVITATIONAL CONSTANT = 1.000E+00

TOTAL STRUCTURE WEIGHT/MASS = 1.274E+03

C.G. LOCATION: X = 0.407 Y = 0.446 Z = 0.679

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DYNAMIC ANALYSIS OF GENERAL STRUCTURES
SWATHER TABLE

SPECIFIED RESTRAINTS
JOINT DIRECTION VALUE

10	3
25	2
25	3
35	3
49	2
49	3

DEGREES OF FREEDOM = 184
MAXIMUM BANDWIDTH = 45
AVERAGE BANDWIDTH = 16
PROFILE SIZE = 2906

NUMBER OF JOINTS = 50
NUMBER OF ELEMENTS = 54
MAXIMUM JOINT NUMBER = 50
MAXIMUM ELEMENT NUMBER = 54

Appendix C

Appendix C

Non-structural parts such as hydraulic cylinders, drive motors and support brackets were also included in the computer model. Enns (1988) listed forces that were used in his model to simulate the loads associated with the mass of these parts. SDRC (1980) required that these forces be converted to masses. Non-structural parts were located on the model as lumped masses to simulate their effect on the modal properties of the structure.

To more closely simulate the distributed masses due to the drapers, guards and knife, the densities of the affected beams were adjusted. Data to calculate these values were obtained from engineers at Versatile Farm Equipment Operations.

Lumped Masses

node 5:	support bracket	0.058	kN
	knife drive	1.427	kN
	crop divider	0.289	kN
	hydraulic cylinder	<u>0.062</u>	kN
		1.836	kN (187.0 kg)

node 30: same mass as node 5 minus the mass of the knife drive

$$1.836 - 1.427 = 0.409 \text{ kN (41.7 kg)}$$

nodes 26 and 50: front portion of the crop divider boards

$$0.267 \text{ kN (27.2 kg)}$$

node 24: mass of reel

$$1.58 \text{ kN (160.0 kg)}$$

node 48: mass of reel plus motor

$$1.58 + 0.089 = 1.669 \text{ kN (170.0 kg)}$$

Distributed Masses

To adjust the densities, it was first established that the density of all structural material was 7820 kg/m³. The additional density was due to the applied distributed mass.

Drapers

Mr. Ken Frisk (pers. com. Jul 1989), engineer at Versatile Farm Equipment Operations, stated that the mass of one draper was 98.0 kg. Enns (1988) stated that 60% of this mass must be supported by the headbeam and the remaining 40% was supported by the cutterbar. The applied load was distributed over 2.766 metres of the headbeam and cutterbar.

$$\text{headbeam: } \frac{98.0 \text{ kg} * 0.6}{2.766 \text{ m} * .002656 \text{ m}^2} = 8004 \text{ kg/m}^3$$

$$\text{cutterbar: } \frac{98.0 \text{ kg} * 0.4}{2.766 \text{ m} * .001843 \text{ m}^2} = 7680 \text{ kg/m}^3$$

Guards and Knife

The mass of the knife and guards was learned from Mr. Harvey Chorney (pers. com. Oct 1989). The total mass of the knife, guards and pitman

head was 43.0 kg. This mass was supported only by the cutterbar that was 7.666 m long.

$$\text{cutterbar: } \frac{43.0 \text{ kg}}{7.666 \text{ m} * .001843 \text{ m}^2} = 3042 \text{ kg/m}^3$$

The effective densities of the individual members were determined by summing the initial density plus the additional densities as calculated above.

Effective density of headbeam supporting the rear of the draper:

$$7820 + 8004 = 15\,824 \text{ kg/m}^3$$

Effective density of cutterbar supporting knife and guards:

$$7820 + 3042 = 10\,862 \text{ kg/m}^3$$

Effective density of cutterbar supporting knife, guards and the front of the draper:

$$7820 + 3042 + 7680 = 18\,542 \text{ kg/m}^3$$

Appendix D

Appendix D

Noise Generator

A Wavetek VCG/Noise generator (model 132) supplied the input signal to the filter. It had the ability to create sinusoidal, triangular, square or random waveforms. The noise generator's operating range extended from 0.2 Hz to 2 MHz.

Signal Filter

The signal filter was a Wavetek Dual Hi/Lo filter (model 452). It consisted of two single-pass filters. A band-pass filter was created by removing all signals below 4 Hz with one filter and then removing all signals above 20 Hz with the other filter.

Hydraulic Pump Controller

A Zonic Xcite Master Controller (model 1104) was used to control the excitation actuator. Outputs from the load cell and the built-in LVDT were monitored by the controller. Through these two sources of feedback, the controller created a force signal that resembled the random signal from the signal filter.

Hydraulic Pump

The power source for the actuator head was a hydraulic pump. A Zonic model 1201 Xcite Hydraulic Power Supply was connected to the actuator head by two hydraulic lines.

Hydraulic Actuator

The force was applied to the swather by a Zonic Xcite Hydraulic Exciter Head (model 1107-4-T/C). This actuator was controlled through a number of servo-valves which in turn was controlled by a master controller. The exciter system (including the hydraulic pump and actuator) could exert a maximum force of 4448 N and the actuator had a maximum stroke length of 50 mm.

Interface between Actuator and Structure

Two modes of connection between the actuator and the structure were used. First, a "load button" was used. The load button resembled a thimble with a threaded insert. It was threaded onto one side of the load cell while the opposite side of the load cell was threaded onto the actuator. When the piston of the actuator was extended to apply a load to the structure, the load button came in contact with the structure. To avoid nonlinearities and signal overloads, the load button remained in contact with the structure at all times. Therefore, a compressive force was applied at all times.

A "stinger" was the second type of connection that was used. It was a short narrow beam that was inserted between the load cell and the structure. With the use of an adaptor, it was threaded to the load cell on one end and attached to the structure at the other. It was very stiff along its longitudinal axis but flexible in lateral directions. Since the stinger was rigidly attached to the structure, compressive and tensile loads were applied to the structure.

Inline Force Transducer

A Zonic load cell (model 2522) was used to detect the force applied to the swather by the actuator. It was installed between the actuator head and the swather. The load cell had a 11 120 N capacity in tension and compression.

Tri-axial Accelerometer

A Bruel and Kjaer type 4340 tri-axial accelerometer was used to measure all structural responses. Since the mass of this accelerometer was only 36 grams, its effect on the structure's responses was neglected. Three co-axial cables were attached to adjacent sides on the accelerometer and each one transmitted the output in one of the three orthogonal directions. To simplify data analysis, each side of the accelerometer was designated to measure the acceleration in either the X, Y, or Z direction. The accelerometer was then positioned on the structure accordingly.

Mounting threads were located on the three sides opposite the output cables. A magnet with a threaded stud was turned into one of the sides of the accelerometer. The magnet was then used to secure the accelerometer to any metallic surface. To help maintain a consistent orientation the magnet was moved from one side of the accelerometer to another as required.

Calibration Exciter

A Bruel and Kjaer type 4294 Calibration Exciter was used to calibrate the output for each of the three orthogonal directions prior to

completing any measurements. The exciter was turned into each of the three threaded ports on the accelerometer and the output from each accelerometer direction was individually calibrated. The calibrator vibrated at 159.2 Hz with a magnitude of 10.0 m/s² RMS. A calibration factor was stored in the Zonic system to achieve the correct signal amplitude. Consequently, three different calibration factors (one for each orthogonal direction on the accelerometer) were stored in the Zonic system.

Signal Amplifiers

The response signal sensed by each direction of the accelerometer passed through a charge amplifier prior to being received by the FFT analyzer. Since accelerations were measured in the three orthogonal directions, three different charge amplifiers were required. Two battery-powered Bruel and Kjaer model 2635 charge amplifiers were used to amplify the signals in the X and Y directions. The accelerations that were sensed in the Z direction were amplified by a Kistler charge amplifier (model 504E). The Kistler amplifier required an AC power supply.

Two-Channel Oscilloscope

A two-channel oscilloscope was used to monitor the signals that were being measured. The Tektronix Storage Oscilloscope (model 5111) could monitor two data channels simultaneously. Most often, the force signal and one other acceleration signal was monitored.

Modal Testing System

The Zonic modal testing system included three main components. A Zonic 6080 four-channel FFT analyzer accepted the three acceleration and single-force signals. The Color Monitor (model 6081) displayed various signal histories as they were received by the FFT analyzer. A Zonic Real Time Display Terminal (model 6081) controlled the transfer of data from the memory of the FFT analyzer to the color monitor or to storage on disk.

Appendix E

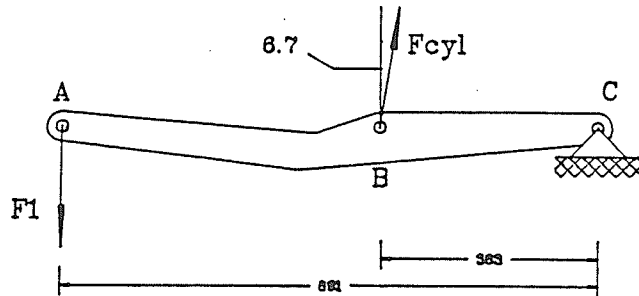
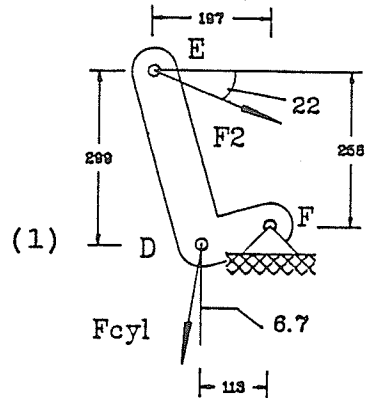
Appendix E

$$\Sigma M_c = 0 \quad (+ \text{ c.c.w. })$$

$$F_1 * (891) = F_{cyl} * (\sin 83.3) * 363$$

$$891 * F_1 = 360 * F_{cyl}$$

$$F_1 = 0.405 * F_{cyl}$$



$$\Sigma M_f = 0 \quad (+ \text{ c.c.w. })$$

$$F_2 * (\cos 68) * 197 + F_{cyl} * (\cos 6.7) * 113 = F_2 * (\sin 68) * 258 + F_{cyl} * (\sin 6.7) * (299 - 258)$$

$$F_2 * (73.8) + F_{cyl} * (112.3) = F_2 * 239.2 + F_{cyl} * 4.8$$

$$F_{cyl} * 107.5 = F_2 * 165.4$$

$$F_{cyl} = F_2 * 1.54 \quad (2)$$

combining equation 1 and equation 2:

$$F_1 = 0.405 * (F_2 * 1.54)$$

$$F_1 = F_2 * 0.624$$

effective stiffness at F_2 is 68 646 N/m

vertical stiffness at F_1 is:

$$68 \ 646 / 0.624 = 110 \ 000 \text{ N/m}$$

Appendix F

D A G S

DYNAMIC ANALYSIS OF GENERAL STRUCTURES
STRUCTURAL DYNAMICS RESEARCH CORPORATION

SWATHER TABLE

*** SPACE FRAME ANALYSIS ***

STRAIGHT BEAMS

BEAM	LENGTH	FORE END JOINT	AFT END JOINT	MATERIAL CODE	SECTION CODE	ROTATION ANGLE	TEMP.
5	0.46	11	9	1	3		
6	0.02	9	6	1	3		
7	0.49	12	7	1	3		
8	0.49	13	8	1	3		
9	0.01	23	26	4	5		
10	2.00	22	23	4	5		
11	0.67	21	22	4	5		
12	1.11	20	21	3	5		
13	1.07	1	2	1	4		
14	0.76	2	3	2	4		
15	1.97	3	4	2	4		
16	0.03	4	5	2	4		
18	1.40	4	24	1	17		
27	0.09	9	10	1	17		
32	0.46	36	34	1	3		
33	0.02	34	31	1	3		
34	0.49	37	32	1	3		
35	0.49	38	33	1	3		
36	0.01	47	50	4	5		
37	2.00	46	47	4	5		
38	0.67	45	46	4	5		
39	1.11	20	45	3	5		
40	1.07	1	27	1	4		
41	0.76	27	28	2	4		
42	1.97	28	29	2	4		
43	0.03	29	30	2	4		
45	1.40	29	48	1	17		
54	0.09	34	35	1	17		
61	0.36	51	10	8	17		
62	0.36	52	35	8	17		
17	0.12	8	4		RIGID		
19	0.05	13	16		RIGID		
20	0.08	19	23		RIGID		
21	0.11	7	3		RIGID		
22	0.11	6	2		RIGID		
23	0.05	12	15		RIGID		
24	0.05	11	14		RIGID		
25	0.08	18	22		RIGID		
26	0.08	17	21		RIGID		
44	0.12	33	29		RIGID		
46	0.05	38	41		RIGID		

DYNAMIC ANALYSIS OF GENERAL STRUCTURES
SWATHER TABLE

STRAIGHT BEAMS

BEAM	LENGTH	FORE END JOINT	AFT END JOINT	MATERIAL CODE	SECTION CODE	ROTATION ANGLE	TEMP.
47	0.08	44	47		RIGID		
48	0.11	32	28		RIGID		
49	0.11	31	27		RIGID		
50	0.05	37	40		RIGID		
51	0.05	36	39		RIGID		
52	0.08	43	46		RIGID		
53	0.08	42	45		RIGID		

TAPERED BEAMS

BEAM	LENGTH	FORE END JOINT	AFT END JOINT	TAPER TYPE	MAT. CODE	SECTION CODE FORE AFT	ROT. ANGLE	TEMP
1	0.25	14	25	1	1	8 25		
2	0.83	25	17	1	1	7 24		
3	1.08	15	18	1	1	2 19		
4	1.08	16	19	1	1	2 19		
28	0.25	39	49	1	1	8 25		
29	0.83	49	42	1	1	7 24		
30	1.08	40	43	1	1	2 19		
31	1.08	41	44	1	1	2 19		

JOINT COORDINATES

JOINT	X	Y	Z
1	0.000	0.595	0.017
2	1.067	0.595	0.017
3	1.829	0.595	0.017
4	3.800	0.595	0.017
5	3.833	0.595	0.017
6	1.067	0.486	0.000
7	1.829	0.486	0.000
8	3.833	0.486	0.000
9	1.067	0.462	0.000
10	1.067	0.499	-0.082
11	1.067	0.000	0.000
12	1.829	0.000	0.000
13	3.833	0.000	0.000
14	1.067	0.000	0.051
15	1.829	0.000	0.051
16	3.833	0.000	0.051
17	1.114	0.019	1.127
18	1.782	0.019	1.127
19	3.786	0.019	1.127
20	0.000	0.059	1.196
21	1.114	0.059	1.196

DYNAMIC ANALYSIS OF GENERAL STRUCTURES SWATHER TABLE

JOINT	JOINT COORDINATES		
	X	Y	Z
22	1.782	0.059	1.196
23	3.786	0.059	1.196
24	3.800	0.595	1.416
25	1.078	0.004	0.300
26	3.800	0.059	1.196
27	-1.067	0.595	0.017
28	-1.829	0.595	0.017
29	-3.800	0.595	0.017
30	-3.833	0.595	0.017
31	-1.067	0.486	0.000
32	-1.829	0.486	0.000
33	-3.833	0.486	0.000
34	-1.067	0.462	0.000
35	-1.067	0.499	-0.082
36	-1.067	0.000	0.000
37	-1.829	0.000	0.000
38	-3.833	0.000	0.000
39	-1.067	0.000	0.051
40	-1.829	0.000	0.051
41	-3.833	0.000	0.051
42	-1.114	0.019	1.127
43	-1.782	0.019	1.127
44	-3.786	0.019	1.127
45	-1.114	0.059	1.196
46	-1.782	0.059	1.196
47	-3.786	0.059	1.196
48	-3.800	0.595	1.416
49	-1.078	0.004	0.300
50	-3.800	0.059	1.196
51	1.067	0.150	-0.150
52	-1.067	0.150	-0.150

CODE	MATERIAL PROPERTIES		
	E	POISSON'S	DENSITY
1	21.5E+10	0.290	7.820E+03
2	21.5E+10	0.290	1.582E+04
3	21.5E+10	0.290	1.086E+04
4	21.5E+10	0.290	1.854E+04
8	50.0E+10	0.290	1.000E-01

DYNAMIC ANALYSIS OF GENERAL STRUCTURES
SWATHER TABLE

CROSS-SECTION PROPERTIES								
CODE	AREA	MOMENTS OF INERTIA		SHEAR Y	RATIO Z	TORSION CONSTANT	WARPING CONSTANT	DEG. FIX.
		Z	Y					
2	9.70E-04	7.00E-07	1.60E-06	2.33	4.11	1.60E-08	2.00E-09	1.00
3	1.50E-03	2.40E-06	2.40E-06	2.22	2.22	3.67E-06	2.72E-12	1.00
4	3.00E-03	2.00E-05	8.00E-06	1.87	4.00	2.00E-05	5.00E-09	
5	1.80E-03	3.67E-06	1.18E-06	1.80	3.61	1.40E-06	3.73E-10	
7	1.00E-03	5.70E-07	2.30E-06	2.89	3.17	9.90E-09	2.10E-09	1.00
8	9.70E-04	7.00E-07	1.60E-06	2.33	4.11	1.60E-08	2.00E-09	1.00
17	1.78E-03	1.85E-06	1.06E-06	1.75	2.79	2.15E-06	3.66E-11	
19	1.00E-03	1.70E-07	5.00E-06	7.50	1.80	7.70E-09	1.00E-09	1.00
24	1.00E-03	1.70E-07	5.00E-06	7.50	1.80	7.70E-09	1.00E-09	1.00
25	1.00E-03	5.70E-07	2.30E-06	2.89	3.17	9.90E-09	2.10E-09	1.00

ECCENTRICITY		
CODE	Z	Y
2	0.000	-0.064
4	0.008	0.020
5	0.011	-0.007
7	0.000	-0.054
8	0.000	-0.064
19	0.000	-0.030
24	0.000	-0.030
25	0.000	-0.054

ELASTIC RESTRAINTS					
JOINT	DIR	TYPE	VALUE	JOINT	DIR
49	Y	LINEAR	1.100E+05		
49	Z	LINEAR	1.000E+11		
25	Y	LINEAR	1.100E+05		
25	Z	LINEAR	1.000E+11		

LUMPED MASS AND INERTIA							
JOINT	MASS	J(X)	J(Y)	J(Z)	J(XY)	J(XZ)	J(YZ)
24	1.60E+02						
5	1.87E+02						
26	2.72E+01						
50	2.72E+01						
30	4.17E+01						
48	1.70E+02						

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DYNAMIC ANALYSIS OF GENERAL STRUCTURES
SWATHER TABLE

TOTAL LUMPED WEIGHT/MASS = 6.132E+02

GRAVITATIONAL CONSTANT = 1.000E+00

TOTAL STRUCTURE WEIGHT/MASS = 1.274E+03

C.G. LOCATION: X = 0.407 Y = 0.446 Z = 0.679

SPECIFIED RESTRAINTS
JOINT DIRECTION VALUE

51	2
51	3
52	2
52	3

DEGREES OF FREEDOM = 200
MAXIMUM BANDWIDTH = 52
AVERAGE BANDWIDTH = 16
PROFILE SIZE = 3248

NUMBER OF JOINTS = 52
NUMBER OF ELEMENTS = 56
MAXIMUM JOINT NUMBER = 52
MAXIMUM ELEMENT NUMBER = 62

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