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Harmonic Analysis of a Representative Generation One Tank Waste Retrieval Manipulator

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Harmonic Analysis of a Representative Generation One Tank Waste Retrieval Manipulator

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Pacific Northwest Laboratory
Operated by Battelle Memorial Institute

Introduction

A major task for the Generation One Tank Waste Retrieval Manipulator (TWRM) will be to autonomously move a special-purpose end effector at constant velocity along a predetermined trajectory. The end effector will employ water jets to break up waste in the form of sludge into small pieces which can be conveyed outside the tank in a vacuum-driven air conveyance line. The trajectory to be followed will be determined by the mining strategy used. It has not been determined whether or not this trajectory will be modified using feedback from end effector mounted proximity sensors.

To maintain required retrieval rates, the end effector trajectory deviations in terms of position and velocity must be kept within specific limits. The water jets used to break up the waste will generate (somewhat) predictable forces at the end effector. The scope of this study is to bound the motion induced at the end effector by harmonic forces at the end effector. The approach used was to develop a finite element model (FEM) of a representative generation 1 TWRM geometry in three different configurations, and plot the resulting forced harmonic response of the model.

TWRM Model Description

The kinematic configurations evaluated are shown in Figures 1-3. The mast was assumed to consist of three telescoping sections. The outside diameter of the largest section was 36.8 inches. It is emphasized here that the results are strongly dependent on this maximum diameter. A larger diameter will result in a stiffer system with a higher natural frequency. A smaller diameter will result in a more flexible system which can be expected to have a larger amplitude of response to harmonic excitations.

For the most extended configuration, a total mast height of 50 feet was used, broken down into constant, circular cross section beams of 20, 17, and 13 feet in length. For the middle and retracted configurations these lengths were 20, 1, and 1 feet. For all configurations, the thickness of the mast sections was 1/2 inch. Stresses were not large (on the order of 8000 psi), but 1/2 inch was considered a minimum to satisfy buckling constraints while minimizing the weight of the mast. It may be possible to use a smaller thickness, but this is beyond the scope of this study. The diameter of the second mast section was set to be 0.96 times the diameter of the initial section, and the diameter of the third mast section was set to be 0.92 times that of the initial section. A point mass of 250 lbm was added at the top of the mast to model the moving portion of the mast rotation actuator. A point mass of 600 lbm was added at the bottom of the mast to model the elbow actuator.

The long reach manipulator (LRM) extending from the bottom of the mast was modeled as three beams, each of constant circular cross section and 11 feet in length. In the middle and retracted configurations, the last arm section was reduced to 5 and 4 feet in length, respectively. Past studies at Oak Ridge National Laboratory (ORNL) have indicated potential advantages in making the distal LRM actuator prismatic rather than rotational. Point masses of 300 lbm and 140 lbm were placed between the first and second arm sections and the second and third arm sections, respectively, to model the actuators.

The short reach manipulator (SRM) was modeled as a constant circular cross section beam 5 feet in length extending horizontally from the end of the third section of the LRM. The actual length of the SRM will more likely be about 10 feet. The 5-foot length models an 8-foot long SRM inclined 45° from the horizontal, and places the end effector 38 horizontal feet (roughly the tank radius) from the mast centerline. The total mass of the SRM was assumed to be 1000 lbm. Half of this mass was associated with a point mass located at the end of the third section of the LRM. The diameter of the SRM was determined so that the mass of the beam modeling the SRM would be half that of the end effector. The thickness of the beam modeling the SRM was assumed to be one fourth the diameter.

The end effector was modeled as a 800 lbm point mass at the end of the SRM. The material properties used were those of stainless steel.

The manipulator joints were modeled as spring and damper elements corresponding to PD (proportional + derivative) closed loop control in the direction of active motion. The remaining five axes of each joint were coupled together. This ignores bearing stiffness, which is difficult to determine accurately. The spring and damper elements were determined as follows. The effective inertia of each joint in the extended configuration was calculated. Assuming the inertia was that of a simple second order system, the spring and damper parameters which yielded a natural frequency of 0.8 to 1.5 Hz (higher frequencies for more distal joints), and a damping ratio of 0.7 were found.

The spring rates for all the active joints were then adjusted simultaneously (reduced about 10%) to achieve an overall vertical stiffness of 125 lbf/in at the end effector, and a lowest overall natural frequency of 0.46 Hz. These values (125 lbf/in and roughly 0.5 Hz natural frequency) were taken from the TWRM test bed specification.

Results

The response (both position and velocity) of the manipulator to 100 lbf harmonic loads in the vertical, transverse, and horizontal directions are shown in Figures 4-9. Each plot shows the response of the manipulator in the three different configurations (extended, middle, retracted). For vertical loads, the response in the extended configuration completely bounds the maximum response for other configurations. Vertical loading also yields the maximum response of the manipulator, with the vertical oscillations peaking out at just one inch magnitude (i.e., a peak-to-peak oscillation of just over two inches) and a velocity of 1/2 inch per second at resonance -- about 0.5 Hz. The response drops off rapidly past resonance, to about 1/4-inch magnitude (0.2 in/sec) at 1 Hz, and smaller at higher frequencies.

Transverse and horizontal loading induce a lower amplitude of response, but the peak response (i.e., resonance) is more configuration dependent. The peak response occurs at frequencies in the 1.5-3 Hz range. Also, the largest velocity errors occur at resonance (about 2.2 Hz) in the retracted configuration. This induced velocity is about 0.8 inches/sec. Again, these induced errors drop off rapidly away from resonance.

It should be noted that linearity was implicitly assumed in the FEM. Thus, any nonlinear effects such as modal coupling were not considered.

Conclusion

As always, if exciting the system at resonance can be avoided, it should be. This model can be expected to give an accurate bound to non-resonant frequencies. However, system behavior when excited at resonance is very difficult to predict. Thus, an actual system excited at resonance may deviate significantly from this model. If end effector harmonic forces can be kept above 5 Hz, it is safe to assume that resonance will always be avoided. This could probably be extended down to 3 Hz. If the end effectors induce forces in the 3 Hz range, there is still only a small portion of the tank in which the manipulator will be in the specific configuration at which resonance occurs. Operation at frequencies below 3 Hz can be expected to induce significant oscillations in the manipulator. However, the bandwidth of the SRM is expected to be on the order of 3 Hz or greater, so that oscillations in this range (< 3 Hz) can be actively damped.

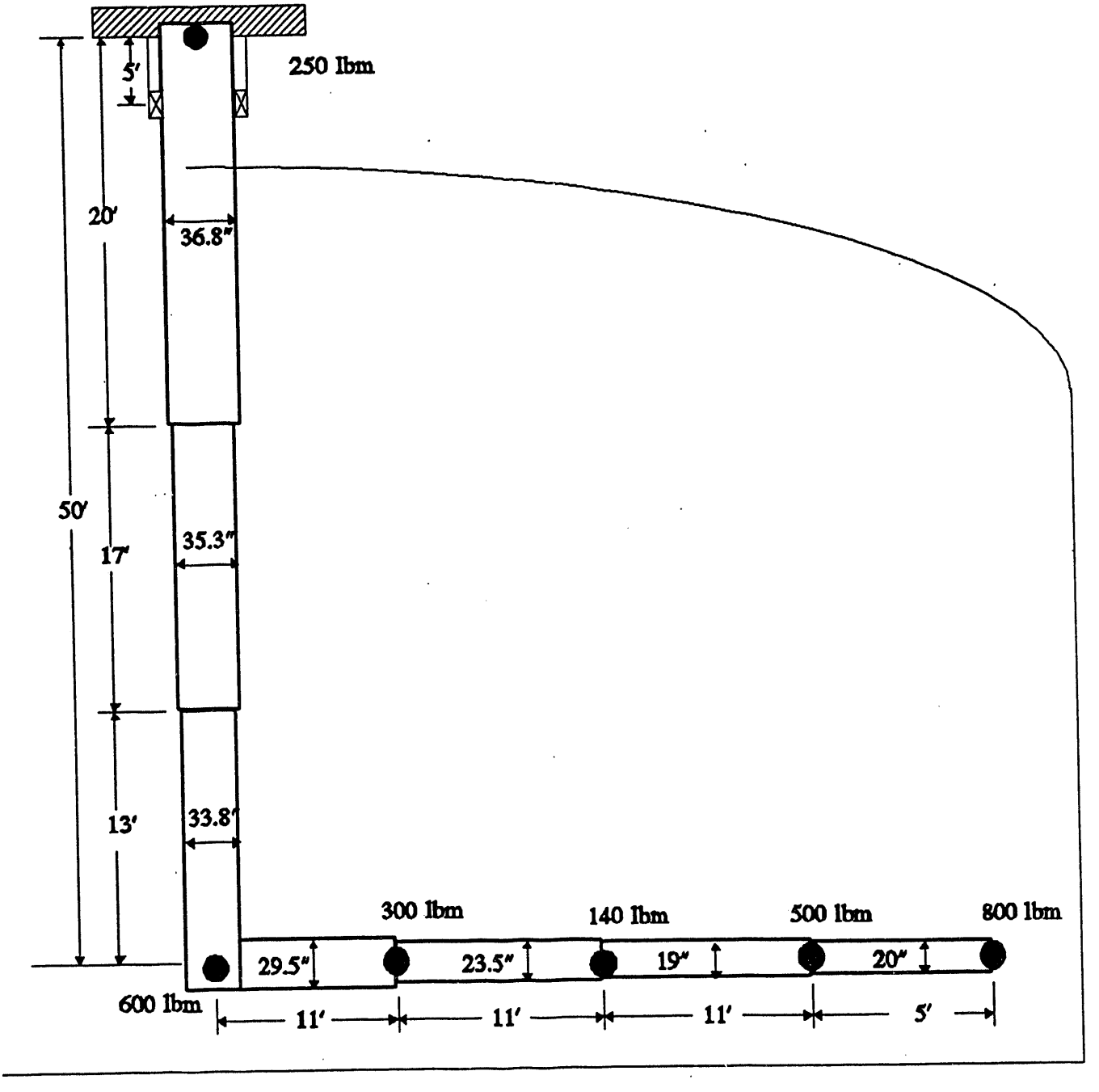

Figure 1 - Extended Configuration

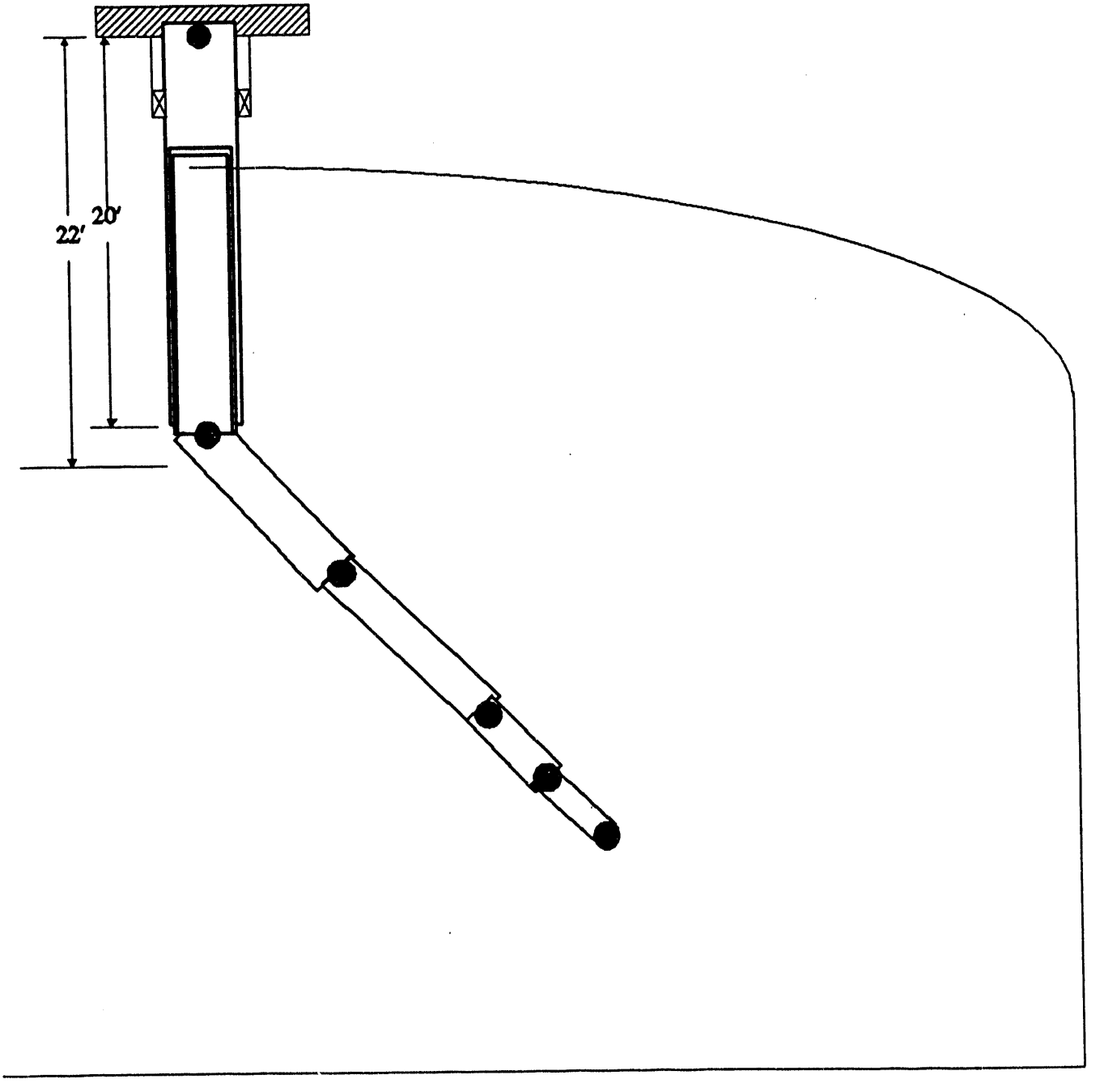

Figure 2 - Middle Configuration

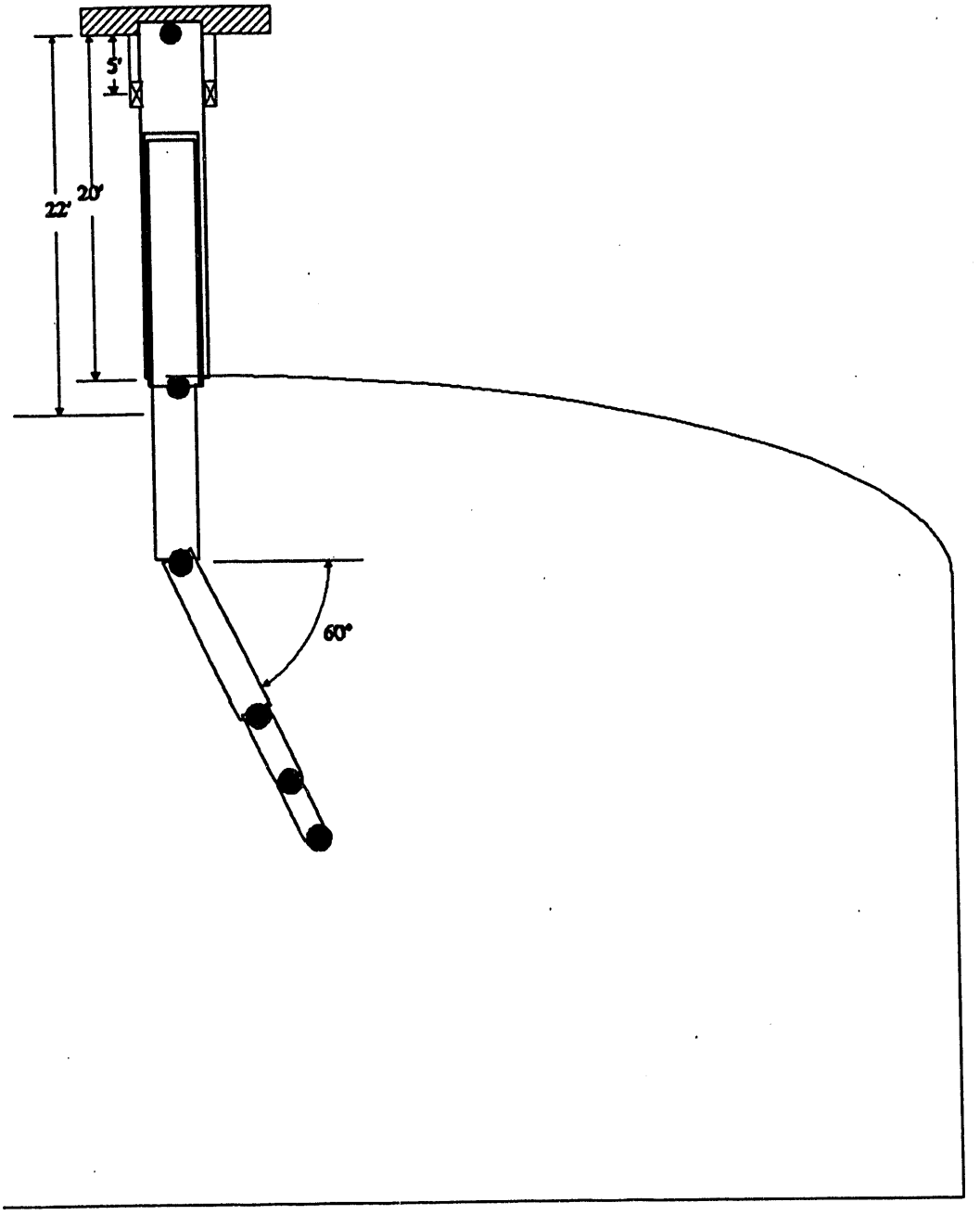

Figure 3 - Retracted Configuration

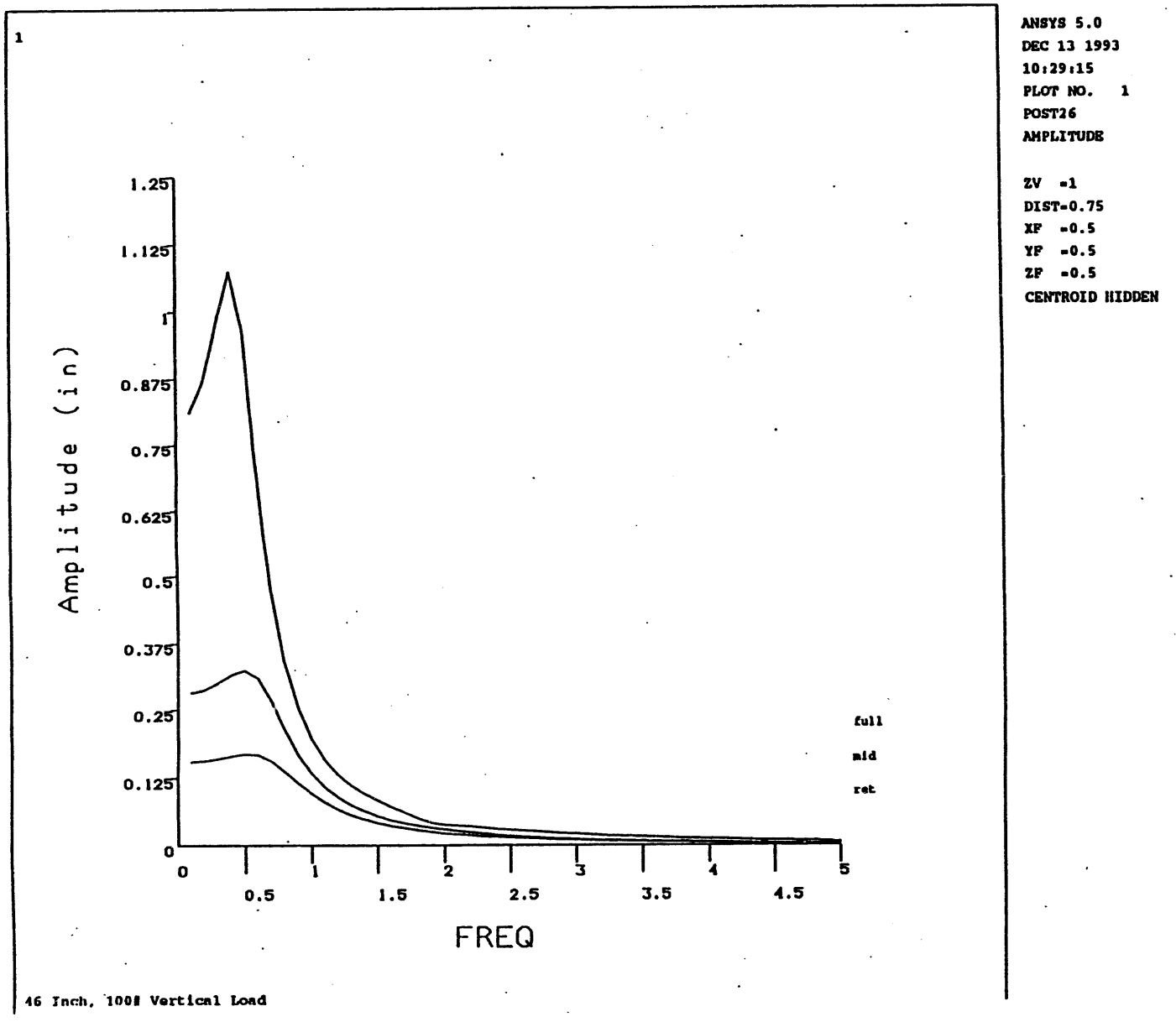

Figure 4 - Frequency Response Amplitude to 100 lb Vertical Force at Tip

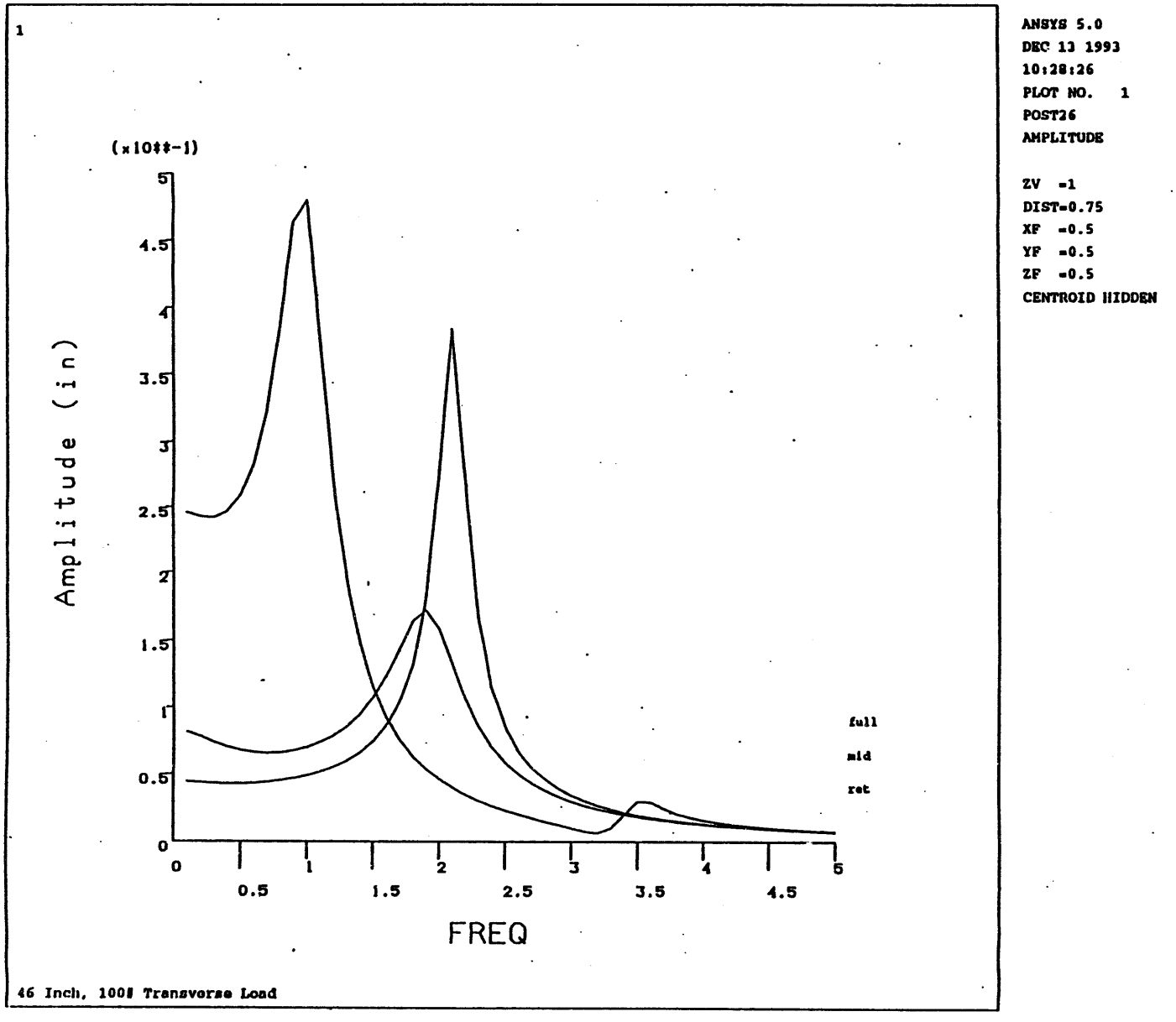

Figure 5 - Frequency Response Amplitude to 100 lb Transverse Load at Tip

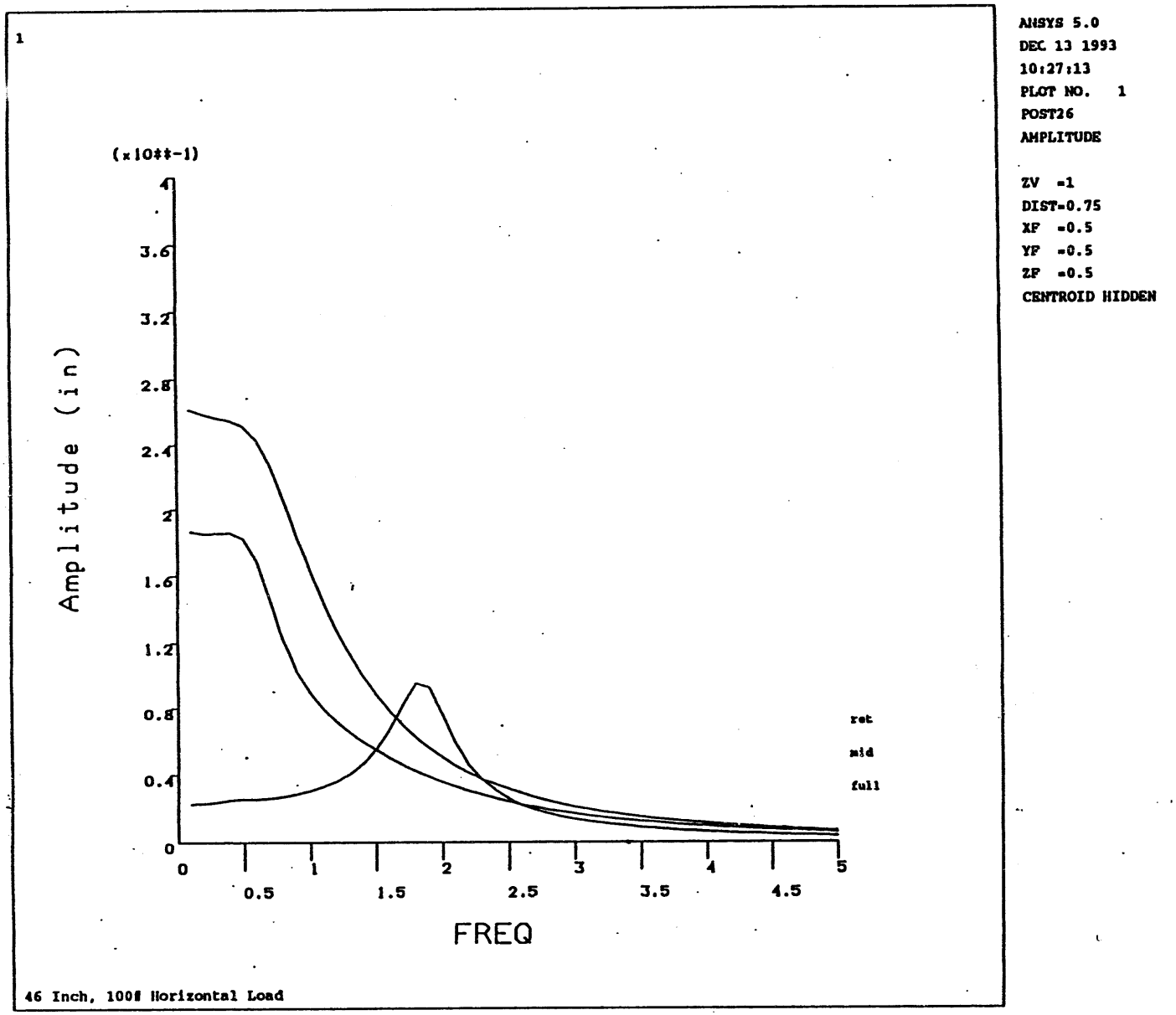

Figure 6 - Frequency Response Amplitude to 100 lb Horizontal Load at Tip

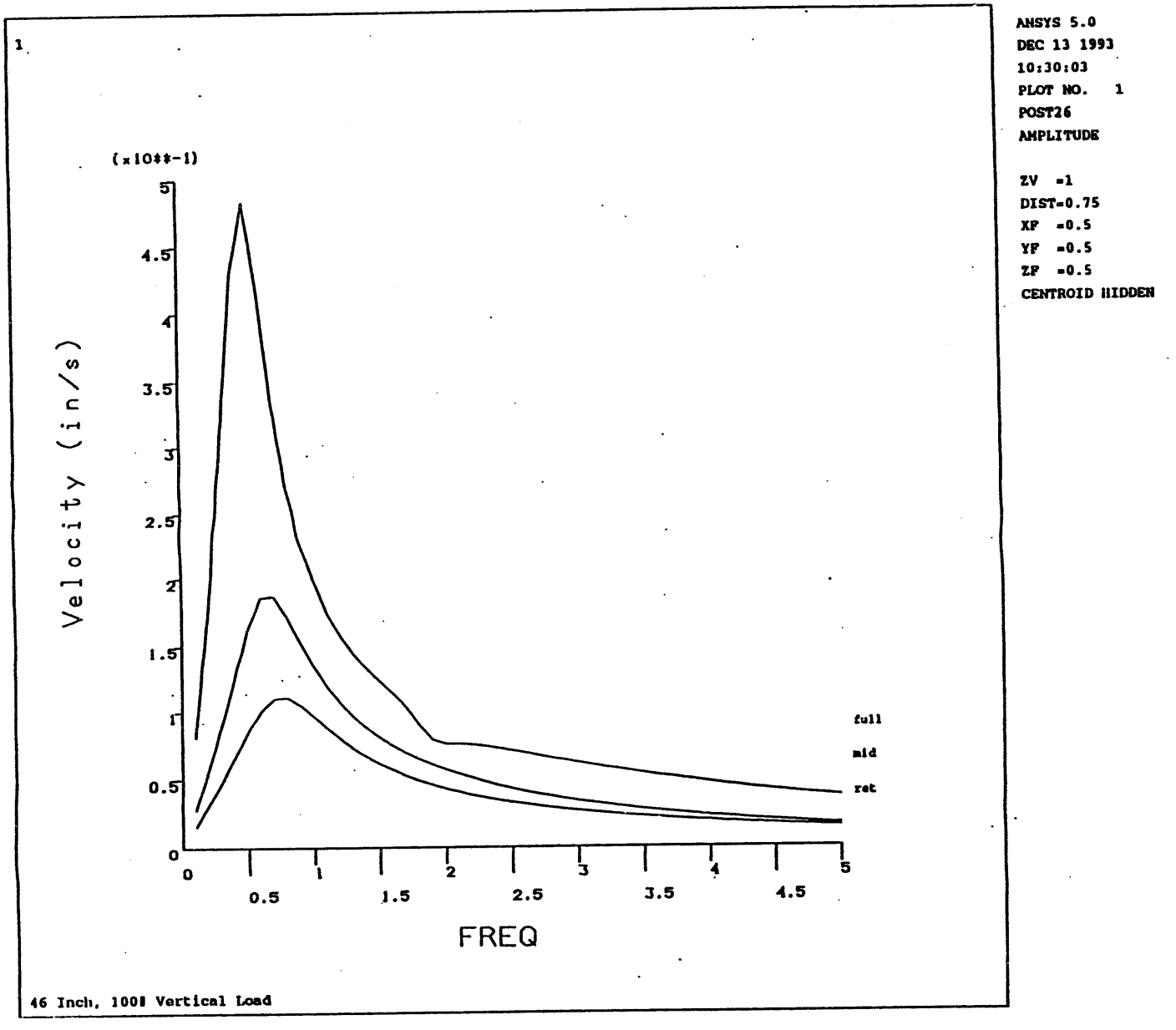

Figure 7 - Frequency Response (Velocity) to 100 lb Vertical Load at Tip

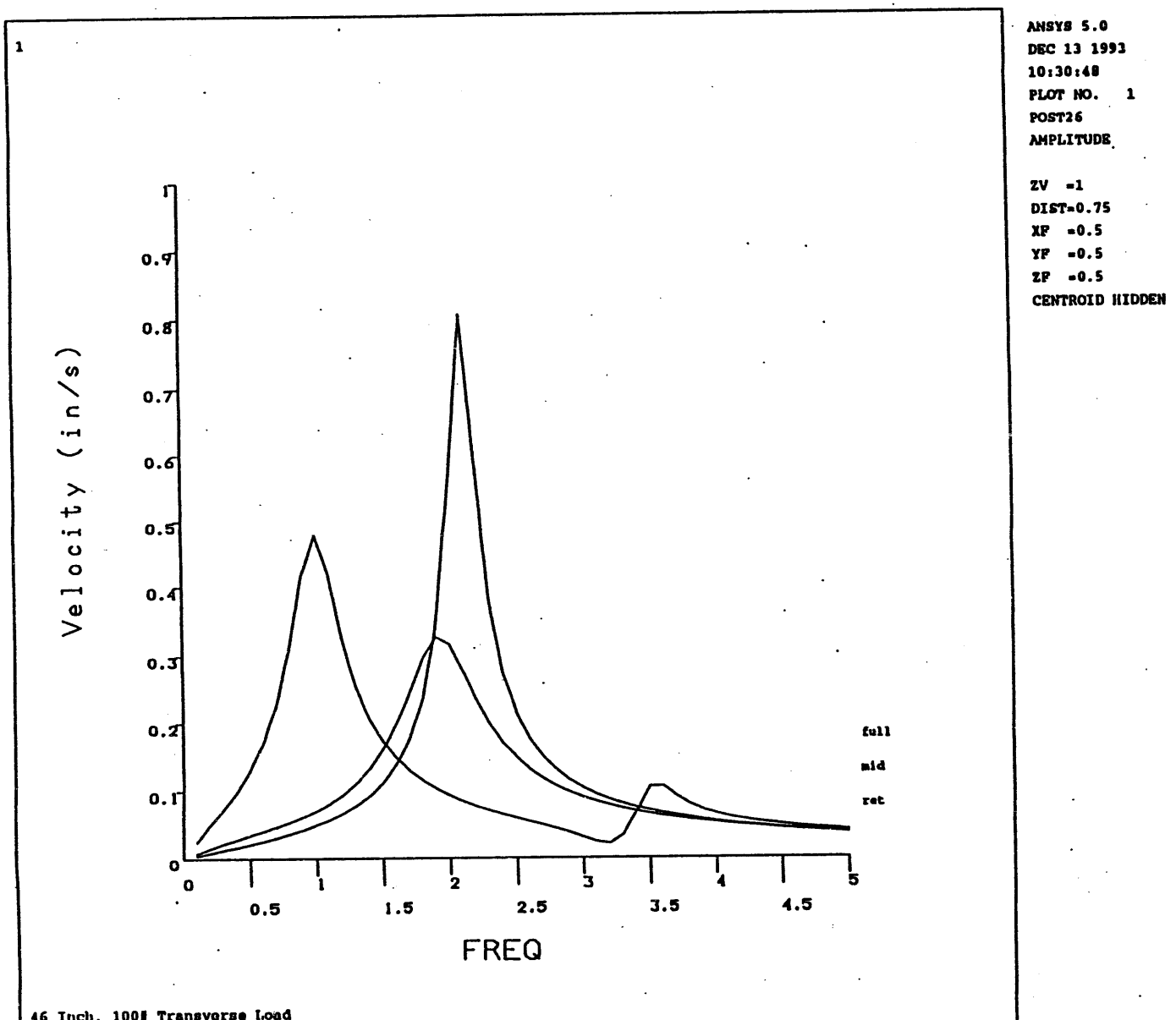

Figure 8 - Frequency Response (Velocity) to 100 lb Transverse Load at Tip

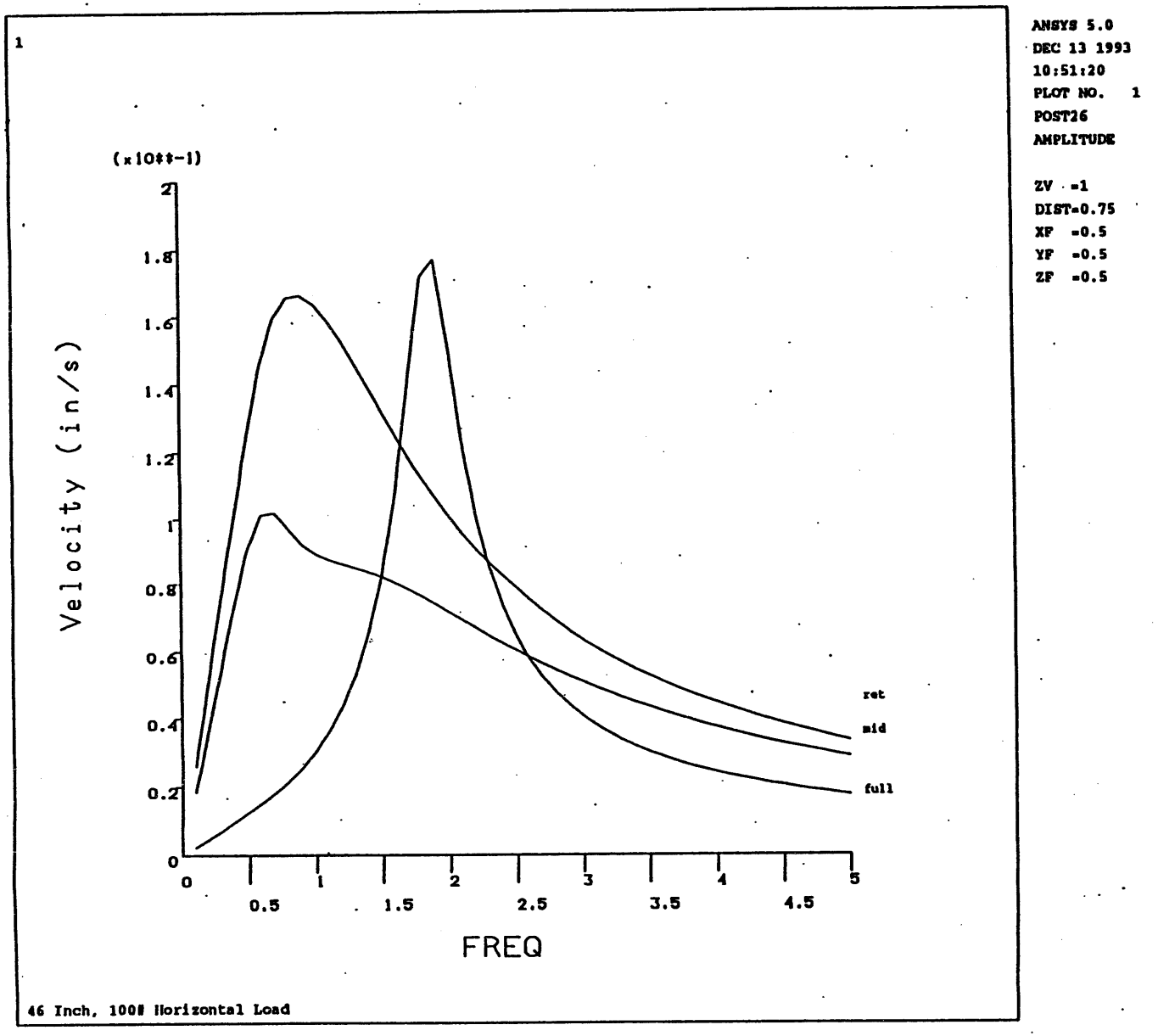

Figure 9 - Frequency Response (Velocity) to 100 lb Horizontal Load at Tip

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