An Investigation of Tuned Liquid Dampers for Structural Control

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ABSTRACT

Rectangular and circular tuned liquid dampers were investigated through shaking table tests for large amplitude excitation. The experimental results confirm earlier work undertaken for small amplitude excitation and considerably contribute to the body of knowledge describing tank behavior under large amplitude excitation. In addition to the experimental investigation, extensive numerical modeling was undertaken for the rectangular tanks. In particular, the random choice method of solving the fully nonlinear shallow-water-wave equations was found to capture the underlying physical phenomenon adequately, including wave breaking, for most of the frequency range of interest. Regardless of the geometry, the tank behaves as a hardening spring system due to the liquid sloshing motion. This trend is enhanced as excitation amplitude increases.

1. INTRODUCTION

Tuned liquid dampers have performed successfully in passively mitigating undesirable structural motion under a variety of settings, e.g., Fediw, et al., (1993); Fujii, et al., (1990); Tamura, et al., (1988); Wakahara, et al., (1992); (1993). Because the underlying physical phenomenon of the liquid sloshing motion that mitigates the structural vibration is not well understood, particularly for large amplitude excitation, shaking table tests and numerical modeling of the liquid sloshing motion of tuned liquid dampers (TLDs), with an emphasis on its behavior under large amplitude excitation, were undertaken. With the ultimate objective of providing improved design strategies for TLDs, performance measures were defined to provide a context for test evaluation.

Performance is defined as "the capability of a single damping device to effectively reduce undesirable structural motion in a robust manner." In the case of multiple devices, this definition should be extended to incorporate redundancy also. This definition incorporates both effectiveness and robustness. Measures of effectiveness include the following:

- Reduction in peak or RMS structural displacement;
- Reduction in peak or RMS structural acceleration;
- Significant increase in "effective" damping of the structural system as evaluated through a single-degree-of-freedom structural dynamics analysis; or
- Significant increase in energy dissipation per cycle over a system without a damping device.

Although some of these measures are equivalent under certain loading conditions, they provide the designer with metrics. Along with effective mitigation of structural vibration there is a secondary property of robustness to identify the best design. The quality of "robustness" is a less well-defined term in the structural control community, although it has been investigated for multiple tuned mass dampers by Abe and Fujino (1994) and Yamaguchi and Harnpornchai (1993). Abe and Fujino (1994) investigated the influence of errors in predicting the structural natural frequencies upon vibration mitigation by multiple tuned mass dampers (MTMD) and found the MTMDs to be robust under conditions of mistuning. Yamaguchi and Harnpornchai (1993) investigated the influence of offsets in the structural damping ratio and found that MTMDs were less robust

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than a comparable single TMD in this instance. In this paper, a robustly designed tuned liquid damper is defined as one in which effective vibration mitigation occurs when

- slight mistuning of the device occurs;
- damping estimates of the device are slightly inaccurate; and
- energy dissipation by the device is significant over a wide range of amplitudes of excitation.

More specifically, for sinusoidal excitation, these definitions of robustness would be as follows:

- The frequency response curve of the structural system is "flat" over a large region, implying that motion reduction is not very sensitive to slight mistuning of the tuned liquid damper.
- The frequency response curve of the structural system is "flat" over a large region even if the estimate of structural inherent damping parameter is slightly inaccurate.
- Energy dissipation is high over a large frequency range for a wide range of amplitude of motion (i.e., small to large amplitude excitation).

For multiple tuned dampers, redundancy is an important consideration, although it has not received the same attention as effectiveness and robustness. As in any system in which multiple units act together for a common purpose, if one or more fail, then the resulting effectiveness should be investigated to ascertain whether these losses would significantly compromise the vibration mitigation.

In this paper, the results of the investigations are evaluated in the context of how effective and robust the tanks would be in structures subjected to sinusoidal excitation over a wide range of amplitudes.

2. EXPERIMENTAL INVESTIGATIONS

The rectangular tanks of dimensions given in Table 1 were investigated under sinusoidal loading conditions. This investigation was undertaken using the shaking table facility at the University of Southern California. Figure 1 shows the experimental set-up. The data measured included the shaking table displacement, the sloshing forces in the horizontal or along (x) and across (y) directions, F_{wx} and F_{wy}, respectively; the temporal water surface variations η at the end wall and at the middle of the tank, and a videotape of the water wave motion for a half length of the tank. In presenting these results, the following parameters will be used:

a) The fundamental natural frequency of the water sloshing motion predicted by the linear water wave theory

$$f_w = \frac{1}{2\pi}\sqrt{\frac{\pi g}{L}\tanh(\pi\varepsilon)} \qquad (1)$$

where the water depth ratio ε is defined by

$$\varepsilon = \frac{h_0}{L} \qquad (2)$$

g is gravity, L is the tank length, and h_0 is the water depth.

b) Excitation frequency ratio β

$$\beta = \frac{f_e}{f_w} \qquad (3)$$

where f_e is the frequency of the sinusoidal excitation, and f_w is the theoretical natural frequency of the water in the tank calculated using equation (1).

c) Nondimensional water surface elevation η'

$$\eta' = \frac{\eta}{h_0} \qquad (4)$$

where η is the departure of the water surface from the undisturbed water surface depth, h_0.

d) Nondimensional base shear force F'_w

$$F'_w = \frac{F_w}{m_w\omega^2 A} \qquad (5)$$

where F_w is the reaction force of the tank created by sloshing motions induced by the shaking table movement; m_w is the mass of the water in the tank; ω is the excitation angular frequency of the shaking table; A is the shaking table excitation amplitude; and the product

$m_w\omega^2A$ is the maximum inertia force of the water mass treated as a solid mass. In this analysis, the measured base shear force was obtained by subtracting the inertia force of the mass of the tank from the total force measured by the load cell to identify the force due to sloshing motion only. A low pass filter was also used to eliminate high frequency system noise.

e) Nondimensional phase angle, ϕ'

The phase angle Φ in radians between the shaking table motion and the base shear force has been expressed in units of π, as follows:

$$\phi' = \frac{\phi}{\pi} \qquad (6)$$

f) Nondimensional energy dissipation per cycle E_d'

$$E_d' = \frac{E_d}{\frac{1}{2}m_w(\omega A)^2} \qquad (7)$$

where the numerator is the energy dissipation per cycle is defined by

$$E_d = \int_{T_s} F_{wx}dx \qquad (8)$$

with dx referring to integration over the shaking table displacement per cycle, and the denominator of eqn. (8) is the maximum kinetic energy of the water mass treated as a solid mass.

3. ANALYSIS

Although extensive time history analyses were undertaken, the frequency investigation appears to provide more relevant information in the context of design and will be explored in this paper. For the tank of length L=590 mm with water depth of h_0 = 30 mm, the effect of the base excitation amplitude on the wave motion was examined for the excitation amplitudes A = 10, 20 and 40 mm. The results are shown in Figure 2 for maximum wave height, maximum base shear force, phase angle between the fundamental mode of the base shear force F_{wx} and shaking table displacement x, and the energy dissipation per cycle. All quantities have been placed in nondimensional form. It can be seen that the maximum wave height peaks at an excitation frequency much higher than the tank natural frequency evaluated by (1), which demonstrates the nonlinear behavior of the fluid in the tank. After this peak is reached, there is a sudden drop in value. This phenomenon has been labeled by previous researchers for limited small amplitude experiments as the "jump" phenomenon, and the frequency at which it occurs is called the "jump frequency," $f_{jump.}$; e.g., Lepelletier, et al., (1980); Sun, et al., (1991). The existence of the jump frequency at frequency ratios $\beta > 1.0$ indicates that the water sloshing motion displays a "hardening" or "stiffening" spring-type nonlinearity behavior. These results are qualitatively consistent with the data provided by Fujino, et al. (1992) for similar experiments but with much smaller excitation. The maximum excitation amplitude to tank length ratio in their investigation was A/L = 0.0169, whereas the present experiments included the case with A/L = 0.0678.

The curves in the plot of the phase angle (Figure 2(c)) show gentle slopes. The corresponding energy dissipation plot for the three excitation amplitudes shows that as amplitude increases, the energy dissipation over a broader range of frequencies occurs. The effective dissipation over a broad range in this manner illustrates that the tank is a robust energy-dissipating system.

From Figure 2 (a), it can be seen that as the excitation amplitude increases, the value of the jump frequency ratio and the response increases. This increase indicates that the nonlinearity of the water sloshing motion becomes stronger as the excitation amplitude increases. The hardening spring behavior can be explained physically by the fact that the wave propagation speed in shallow water depends on the wave amplitude. Based on the shallow-water wave theory, the propagation speed, S, for a non-breaking wave is $\sqrt{g(h_0 + \eta)}$. For the quasi-steady response of the sloshing motion to the forcing excitation, the excitation frequency, f, and wave propagation speed, S, must be related by

$$f = \frac{S}{2L}, \qquad (9)$$

where L is the tank length. Equation (9) indicates that as the excitation frequency increases, the wave must propagate faster. Consequently, the wave amplitude must increase, and ultimately, greater response results. A similar observation can be made in the case of wave breaking (Reed, et al., 1996).

The relationship between the jump frequency and nondimensional excitation amplitude is shown in Figure 3. This plot includes previous data of Sun, et al., (1991) for small amplitude motion. Although the damper is assumed to no longer be effective at excitation frequencies greater than the jump frequency, the results clearly show that, when the excitation amplitude increases, the frequency range of damper effectiveness widens. Figure 3 further illustrates that when the excitation amplitude goes beyond the value of approximately $A/L > 0.025$, the slope of the curve increases. This change implies that the rate at which the range of frequencies for which the damper is effective increases. The results clearly demonstrate that the tuned liquid damper performs in a wider frequency range under strong excitation. This behavior also suggests that the nonlinear behavior is responsible for the improved performance and robustness of the rectangular TLD.

4. NUMERICAL ANALYSIS OF THE SLOSHING MOTION IN A RECTANGULAR TANK

A series of numerical simulations were undertaken using the random choice method to solve the shallow-water wave equations (Gardarsson and Yeh, 1994). The results presented here correspond to the simulation for the L = 590 mm tank whose experimental behavior was shown in Figure 2. The length of the tank was meshed with 400 grid points. The hydrodynamic force was limited to the hydrostatic pressure using the wave-heights at the endwalls. Figure 4 contains the frequency analysis comparing the simulations and the measured data. It is seen that the simulated results are in good agreement with the experimental results; however, the jump phenomenon is not captured. The numerical scheme predicts a much gentler, smoother transition over the frequency range for the wave profile and maximum base shear force. The energy dissipation curve compares well for the entire frequency range, with slight discrepancies near the excitation ratio of unity and in the jump region. While the numerical model does not simulate exactly the wave motion over the entire frequency range, it does provide a close approximation of the tuned liquid damper performance through the energy dissipation over the entire range including the conditions under strong excitation motion. This particular result is encouraging, given that the design process may ultimately be formulated upon the maximum performance as defined by the estimated energy dissipation.

5. CONCLUSIONS

Experimental and numerical-modeling investigations of tuned liquid dampers have provided insight into the behavior of these devices in controlling structural vibration, especially under large amplitude excitation. Our conclusions are as follows:

- The TLD exhibits a "hardening" or "stiffening" spring property, i.e. the peak response in wave height and base-shear force occurs at an excitation frequency much higher than the natural frequency f_w based on (1).

- After a gradual build-up of the TLD response, at a certain excitation frequency which is higher than f_w, the response suddenly ceases; this is often termed the "jump" phenomenon. It was found that the TLD's "jump" frequency increases as the excitation amplitude increases.

- In terms of energy dissipation, the greater the excitation amplitude, the wider the range of effectiveness of the TLD.

- The numerical model based on the fully nonlinear, nondispersive, shallow-water wave theory is capable of predicting the observed TLD energy dissipation with satisfactory accuracy. On the other hand, the model failed to predict the "jump" phenomenon in wave amplitude and base shear force.

•Our results indicate that it is the nonlinearity of sloshing motion that makes TLD more effective and robust in structural control under strong motion excitation. In other words, as the excitation increases, the effective frequency range for the TLD becomes wider, hence TLDs are robust in dissipating energy over a broad frequency range.

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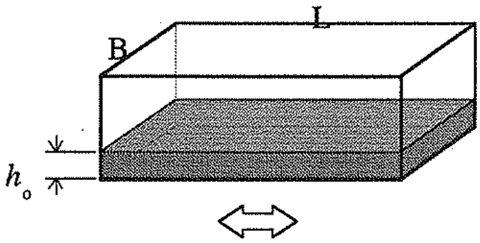
Table 1: Data for the rectangular tanks.

Case ID	*Tank Size* Length L (mm)	Width B (mm)	*Water Depth* h_0 (mm)	*Water Depth Ratio* ε	*Natural Freq.* f_w (Hz)	*Excitation Amplitude* A (mm)	A/L
L335h9.6	335	203	9.6	0.029	0.457	10	0.030
L335h15	335	203	15	0.045	0.571	2.5,5,10,20,30	0.007 - 0.119
L590h15	590	335	15	0.025	0.325	2.5,5,10,20,30,40	0.004 - 0.068
L590h22.5	590	335	22.5	0.038	0.397	2.5,5,10,20,30,40	0.004 - 0.068
L590h30	590	335	30	0.051	0.458	2.5,5,10,20,30,40	0.004 - 0.068
L590h45	590	335	45	0.076	0.558	20	0.034
L900h30	900	335	30	0.033	0.301	2.5,5,10,20,30,40	0.003 - 0.044
L900h40	900	335	40	0.044	0.347	2.5,5,10,20,30,40	0.003 - 0.044
L900h55	900	335	55	0.061	0.406	10,20	0.011 - 0.022
L900h71	900	335	71	0.079	0.459	2.5,5,10	0.003 - 0.011



Excitation direction

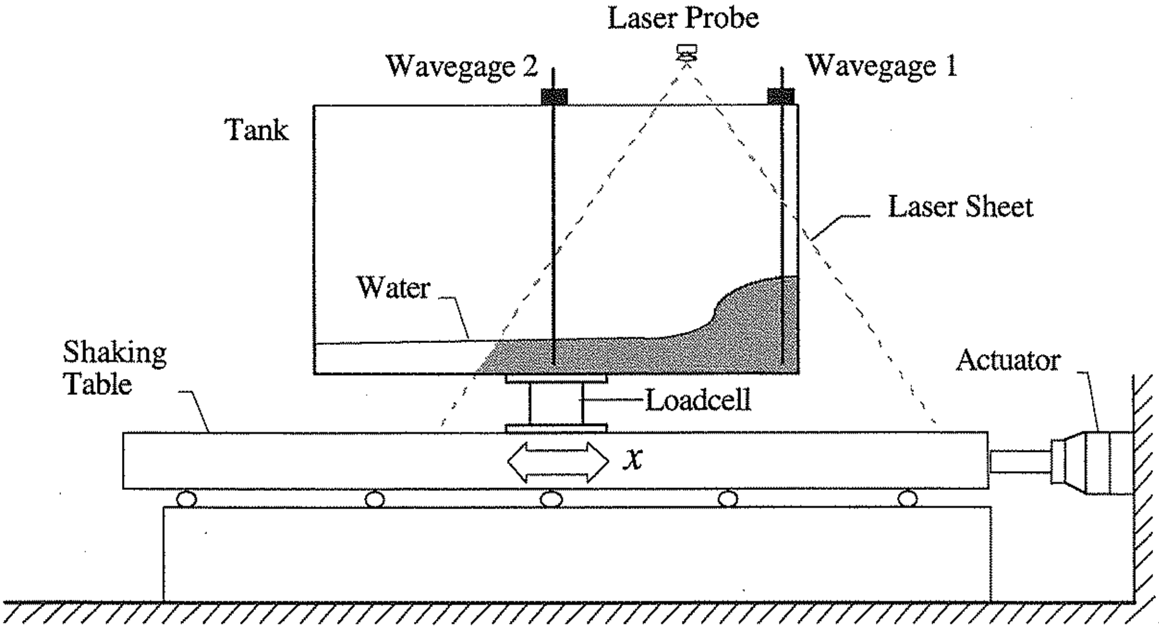


Figure 1: Shaking table experiment set-up

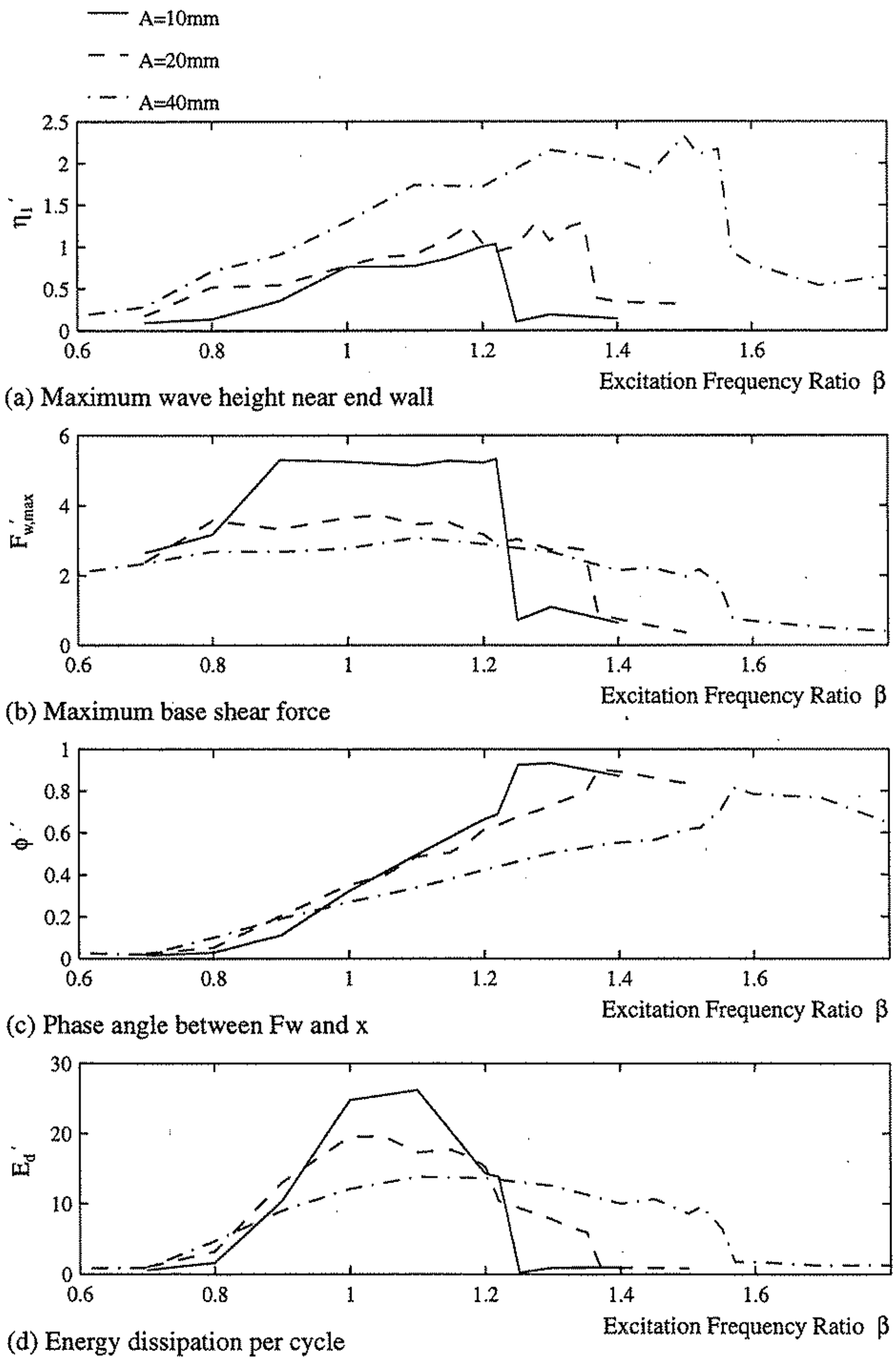


Figure 2: Sample frequency response curves for the water sloshing motion in the shaking table tests of the tank with length $L = 590$ mm, $h_0 = 30$ mm and excitation amplitudes $A =$ 10, 20 and 40 mm. β is excitation frequency ratio.

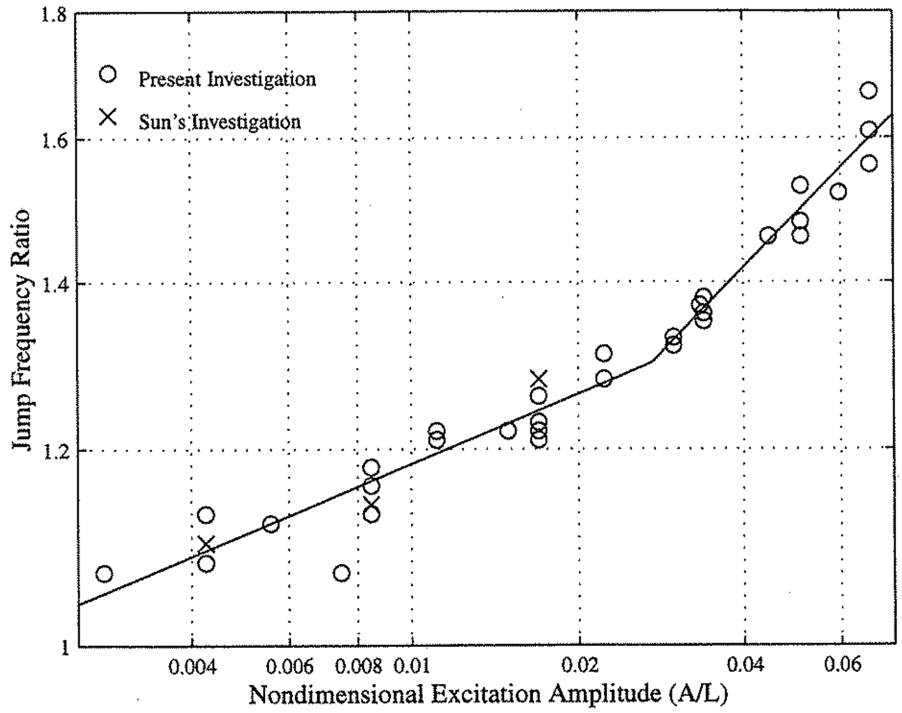


Figure 3: Relationship between the jump frequency ratio and the nondimensional excitation amplitude based on experimental results of Sun, et al. (1991) and the present investigation.

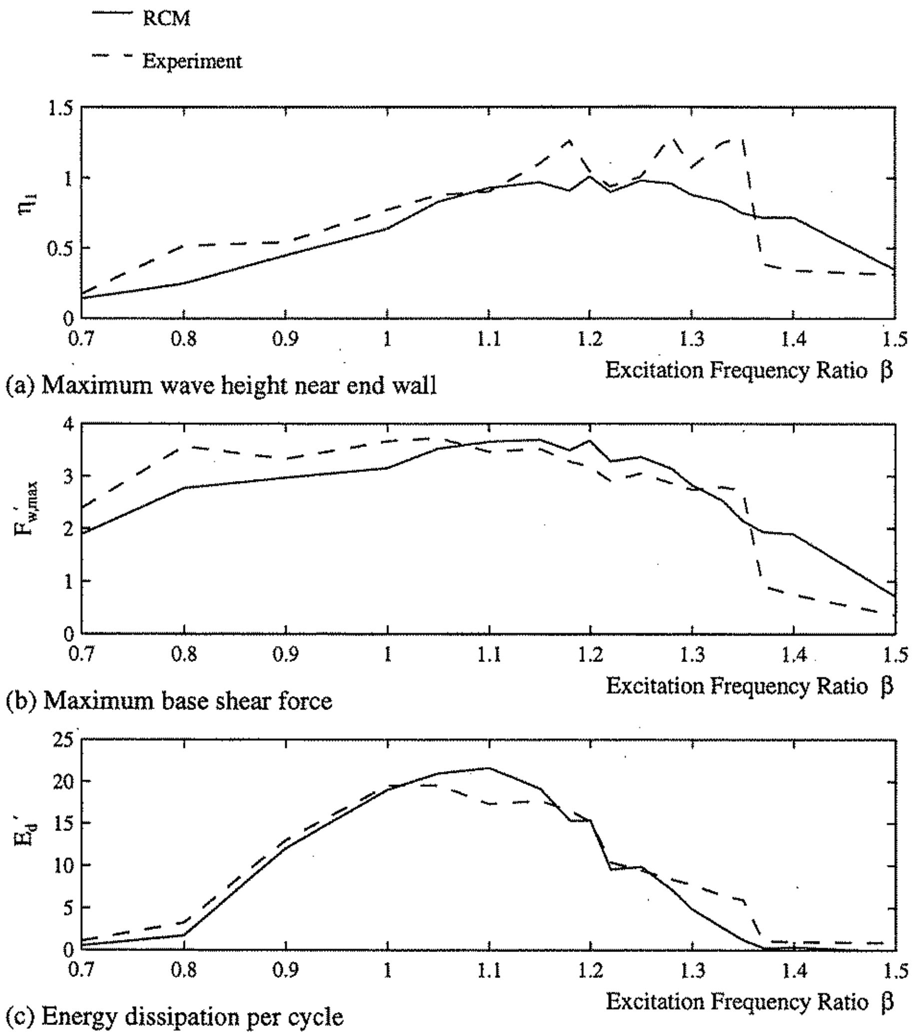


Figure 4: Frequency response comparisons using shaking table data and simulations using the random choice method for the rectangular tank of length $L = 590$ mm, $h_0 = 30$ mm and excitation amplitudes $A = 10$, 20 and 40 mm. β is excitation frequency ratio.