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Theoretical Transfer Function for Force-controlled Wave Machines

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This study concerns the 1st-order transfer function for absorbing force-feedback controlled wave machines. In the context of such devices, this transfer function may be regarded as the equivalent to the progressive wave field coefficient or wave-amplitude ratio for position-controlled machines—the latter linking the demand signal and the amplitude of the progressive wave mode. At present, the operation of force-controlled machines is often based upon an empirical calibration. In contrast, the approach shown here facilitates the computation of a theoretical transfer function. In considering this approach, the influence of the absorption mechanism on the 1st-order transfer function is investigated. A direct comparison of the theoretical transfer function for position-controlled and force-controlled wave machines is made and some important differences are identified. The theoretical framework is based upon published work by the same authors. In contrast to this earlier work, in which the behaviour of flap-type machines was considered, the model has been extended to include the operation of piston-type wave machines. Extensive experimental evidence for a piston-type machine is presented for the first time and reference to further experimental data is made.

INTRODUCTION

An ever increasing number of laboratory wave generation facilities worldwide is equipped with absorbing wavemakers. Given some of the inherent advantages of force-feedback control (Salter, 1982; Naito, 2006; Newman, 2010; Minoura et al., 2009; Spinneken and Swan, 2009a, c), many of these facilities are operated in this latter mode. At present, the wave generation in force-control mode is often based on an empirical transfer function, particularly for those machines following Salter's early design. Such an empirical transfer function has been discussed in detail by Masterton and Swan (2008).

This transfer function relates the desired wave motion in the tank to the demand signal appropriate to the force-controlled paddles. Owing to the empirical nature of the transfer function, an inter-comparison of the operational performance of various laboratory facilities proves extremely difficult. Even for identical facilities, departures in the transfer functions could be due to (i) systematic errors in the calibration procedure, (ii) measurement errors during the calibration routine, and (iii) inaccurate calibration of the force transducer or position transducer. With a common reference being unavailable, errors in obtaining the empirical transfer function are likely.

In contrast, the theoretical behaviour of position-controlled devices is well established and machines are often operated using a theoretical transfer function. In position control, the transfer function links the paddle displacement and the amplitude of the desired progressive wave component (Biésel and Suquet, 1954; Ursell et al., 1960.)

This present paper seeks to establish a similar framework for the operation of force-controlled machines, hence facilitating the inter-comparison of performance data from many laboratory facilities worldwide. In working towards this objective, a *simple* coefficient is introduced which directly links the amplitude of the progressive wave and the force demand signal. Experimental evidence for the successful application of this coefficient is provided, and reference to further experimental data is made.

PHYSICAL MODEL

This study investigates unidirectional, irregular waves based on the superposition of N wave components ranging from $n = 1, \ldots, N$. A boundary-value problem based on a classical perturbation approach combined with Taylor expansions of the boundary conditions at the free surface is adopted. In obtaining the solution, only 1st-order quantities are considered. Within this the following notation is employed: The velocity potential $\Phi = \Phi(x, z, t)$ in Cartesian coordinates (x, z) is defined such that the velocity components are given by $(u, w) = (\partial\Phi/\partial x, \partial\Phi/\partial z)$, where x is defined as positive in the direction of wave propagation, and z is measured vertically upwards from the still water level (SWL). The water surface elevation is given by $\eta = \eta(x, t)$ and the wave board position by $X = X(z, t)$; the wave board position at still water level ($z = 0$) being denoted by $X_0(t)$. Further, θ, ω, g, h and t denote angular wave board displacement, angular wave frequency, acceleration of gravity, still water depth, and time, respectively. The length l is defined so that $z = -(h + l)$ gives the centre of rotation of the wave board, and $d \geq 0$ is the elevation of the hinge above the bed. If the centre of rotation is below or at the bed, then d is set to zero. For the special case of a full-depth piston-type wavemaker, it follows that $l = \infty$ and $d = 0$. Fig. 1 defines both the general wavemaker geometry and the notation adopted. The values for d and l for commonly employed wavemaker geometries are further clarified in Fig. 2.

The boundary conditions appropriate to the problem defined in Fig. 1 are as follows:

a. With the bed assumed to be horizontal and impermeable, $w = 0$ on $z = -h$.

b. The water surface profile is a streamline, defining the Kinematic Free Surface Boundary Condition (KFSBC).

c. The pressure acting on the water surface is constant (atmospheric), defining the Dynamic Free Surface Boundary Condition (DFSBC).

d. On the wave board, the fluid velocity perpendicular to the wave board must be equal to the velocity of the wave board.

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