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Design Options For Dynamic Captive Ship Model Test Facility

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Abstract:

With increased traffic and operation of ships at sea, it becomes necessary to understand the requirements for the future of safe shipping. In this regard, IMO and other agencies have prescribed standards for surface ship manoeuvring to ensure navigational safety. The performance of surface ships in manoeuvring is estimated numerically and/or experimentally in the early design stage. Present work reviews the experimental methods of estimating manoeuvring characteristics of a ship which includes the conventional captive model tests carried out in the towing tank. Planar Motion Mechanism (PMM) tests are the most popular as they yield both acceleration and velocity dependent derivatives. The paper presents different design concepts of PMM test facility to be installed in the towing tank at IIT Madras. The mechanical and operational details of PMM options, to suit the existing towing tank facility at IIT Madras, will be discussed here in this paper.

1.Introduction

The study of ship manoeuvring gained more importance with the development of ultra-large cargo ships and also high-speed passenger vessels posing greater challenges to the maritime industry. Our understanding of these challenges has enhanced with rapid advance in time and technology.The directional stability and control characteristics of surface ships are generally understood by solving manoeuvring equations of motion. This is a complex and perhaps the most challenging task as it involves the evaluation of the inertia, damping and restoring terms of the equations of motion and the subsequent simulation of the vessel's trajectory. Theterms related to hydrodynamic reaction forces are generally termed as hydrodynamic derivatives or coefficients. These coefficientsdepend on the hull form and ship motion characteristics. The form and presentation of these coefficients are determined by the type of mathematical model chosen for the ship manoeuvring analysis. The hydrodynamic coefficients are derived from the force time history measured from ship model tests or from those estimated using theoretical or numerical methods. Accurate prediction of hydrodynamic coefficients is essential for the correct determination of the manoeuvring characteristics of a vessel.

2.Captive Model Test

Various empirical relations or expressions are available in literature using which some of the hydrodynamic coefficients can be estimated. These relations are mainly drawn from analytical or numerical techniques in a limited way. More reliable values of these derivatives are obtainable through experiments. Static and dynamic captive model tests are largely carried out in the towing tanks for the experimental determination of the hydrodynamic derivatives. The straight line tests carried out in a towing tank and the rotating arm tests performed in a rotating arm facility are static types which give only velocity dependent derivatives, whereas dynamic model tests such as PMM are more versatile as they provide data for both velocity and acceleration dependent derivatives. In this facility, a ship model is oscillated in different modes of motion in the horizontal plane while it is towed along the tank at a pre-determined speed, oscillation frequency and motion amplitude. The hydrodynamic forces and moments acting on the ship modelare measured, recorded and processed to get the hydrodynamic derivatives appearing in the manoeuvring equations of motion.

3.Historical Background Of Pmm

The origin of PMM dates back to late 1950s when Horn(1958) and Walinski(1959) used a pair of cranks of 20cm of radius to generate nearly pure harmonic sway and yaw motions of a captive ship model in the *Versuchsanstalt fur Wasserbau und Schiffbau,* Berlin. Later,Gertler M.(1959) and Goodman A.(1960) coined the word PMM to denote an indigenous two-point slider crank oscillator of 1-inch radius designed mainly for testing submarine modelsat the David Taylor Model Basin (DTMB) near Washington D.C.Subsequently, many improved versions of PMM are reported to have been operationalized at different places all over the world [2]. In the last five decades, other similar devices which followed up since the first published results of PMM tests are: Pauling and Sibul(1962) devised a PMM at the University of California, Berkeley; Keil and Thiemann(1963) at the *Institut fur Schiffbau,* Hamburg; Zunderdorp and Buitenhek(1963) at the Technological University, Delft; Motora and Fujino(1965) at the University of Tokyo, Tokyo; Strom-Tejsen and Cheslett(1966) at the *Hydro-ogAerodynamiskLaboratorium*, Lyngby; Cardases(1987) at the University of Southampton; and so on. However, it was Leeuwen(1964) who thoroughly investigated the frequency effects, Froude number effects and effects of rudder and propeller on a standard Series 60 model [5] followed by Strom-Tejsen(1966) continued in Chislett and Smitt(1974) who obtained predictions for full scale manoeuvres in fair conformity with sea trip results. In 1969, Leeuwen proposed a horizontal PMM of large amplitude(375cm) operating at low frequency to generate realistic ship motions. Bishop et al. emphasised on the non-linear effects in modelling hydrodynamic forces in a series of publications.

4.Theoretical Background Of Pmm

The conventional PMM consists of two oscillators, producing a transverse oscillation at the bow and other at stern, either in-phase or out-of phase while the model is towed in the tank at a constant velocity along the centre-line of the towing tank. The PMM imparts sinusoidal motion to the model in the desired degrees of freedom[1]. Hydraulic drive is selected owing to its excellent controllability of motion.The forces and moments acting on the model are measured using suitable dynamometry and special instrumentation. The Fourier series representation of the force and moment time histories recorded from the model during PMM test leads to the determination of the hydrodynamic derivatives.

The oscillators in the conventional PMM concept are realised using two hydraulic pistons. The phasing between the two oscillators decides the mode of motion given to the model moving forward at some prescribed velocity. When the phase difference is zero, the model undergoes pure sway motion and in other cases, the motion can be pure yaw or combined sway-yaw. When the mechanism is operating to produce the yaw motion, the axis-to-axis distance between two pistons has to change. Hence there is another mechanism to control this motion of the hydraulic piston towards each other(or away from each other, depending on the direction of rotation of the model). Also, it is to be ensured that the yaw motion is always about original centre point. This requires a synchronized symmetric motion of the hydraulic pistons.

5.Design Considerations

5.1.Towing Hardware

The towing tank at IIT Madras has the facility to conduct ship model tests. It has a length of 82.5m, width of 3.2m and depth of 2.8m. The ship models are towed using a towing carriage of length 4m and width 3.75m. The maximum forward velocity of the towing carriage is 5m/s. It is actuated with the help of DC servo-motors which are controlled by a Ward-Leonard system.

Figure 1: Towing Tank

Figure 2:Towing Carriage

The towing carriage has a test well of dimensions 3m by 1.35m of which location is not symmetric with respect to the centre-line of the towing tank. This asymmetry is due to the space occupied by the control console on one side of the carriage which is the housing for the automation and the data acquisition unit. This asymmetry of the test well poses certain constraints to the option of retrofitting the PMM. The amplitude of the sway is limited by the width of the well and also the model may go very close to one side of the tank wall. In addition, there is not enough space to locate the hydraulic pistons on any of the sides of the test well.

Figure 3: Test Well

Parameter	Design Considerations
Sway Amplitude	Minimum interaction with walls.
Sway Rate	Number of cycles in steady velocity region of the tank.
Yaw Amplitude	Adequate clearance between the model and the wall. Number of cycles in steady velocity region of the tank.
Yaw Rate	Number of cycles in steady velocity region of the tank.
Model Size	Adequate clearance between the model and the wall.

Table 1: PMM design parameters

Parameter	**Dimensions**	
	Ship	**Model(1:80)**
Length Overall	171.80m	2.1475m
Length between perpendiculars	160.93m	2.0116m
Maximum Draft	23.17m	0.2896m
Design Draft	8.3m	0.1037m
Design Displacement	$18541m^3$	$0.0362m^3$
Design Speed	15knots(7.7 m/s)	0.86m/s*

Table 2: Particulars of the Mariner vessel [3]
**Using Froude scale criteria*

5.2.Check For Tank Wall Clearances

Analysis for model-tank clearance is made on a box shaped model of dimensions close to that of the above Mariner class vessel. The model is subjected to different kind of motions in horizontal plane. It is assumed that the centroid of the ship model is executing sinusoidal motion with cycle length equal to 10m. The clearances for different Sway and Yaw amplitudes are calculated thereof.

Consider a rectangular box shaped model shown below with dimensions as shown in Figure 4:

Length, L= 2.5m

Breadth, B= 0.5m

Draft, d = 0.4m

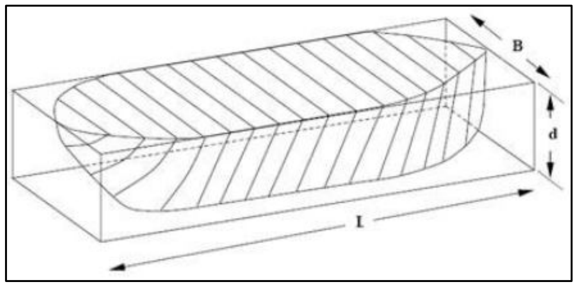

Figure 4: Ship model

5.3.Pure Sway

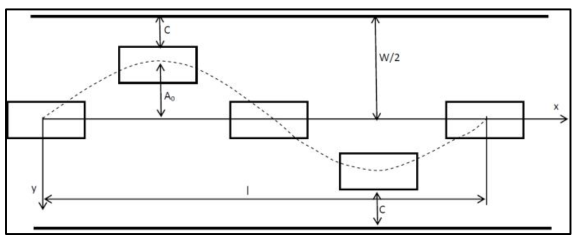

Figure 5: Model subject to pure sway

In this case, the model is subjected to motion in lateral direction to calculate linear velocity and acceleration coefficient terms. Van Leeuwen(1969) considers half the tank width as an upper limit for the trajectory in order to avoid wall effects.

Sway motion is represented by: $y = A_0 \sin(2\pi x/l)$

Given, width of towing tank, W = 3.2m and breadth of the box,B = 0.5m

Available Clearance, $C = W/2 - A_0 - B/2 = 1.35 - A_0$,

Sway Amplitude, A_0 (m)	0.2	0.3	0.4	0.5
Minimum Clearance, C(m)	1.15	1.05	0.95	0.85

Table 3: Available clearances in pure Sway

5.4.Pure Yaw

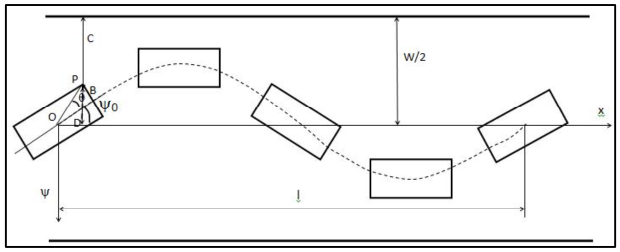

Figure 6: Model subject to pure yaw

In order to impose an angular velocity and an angular acceleration to the body with $\dot{v}$ *and* v both equal to zero, the model must be towed down the tank with the centreline of the model always tangent to its path. This means that the resultant velocity V = u_0.

The yaw motion is given by: $\psi = \psi_0 \sin(2\pi x/l)$

From Fig. 6, the available clearance**,** C = W/2 – PD

wherePD = OP $\sin(\theta + \psi_0)$

$\tan\theta = \frac{BP}{OB} = \frac{0.25}{1.25} = 0.2 \rightarrow \theta \cong 11.31$deg,

OP = $\sqrt{(BP^2 + OB^2)}$ = 1.274m

Hence, Clearance, C= W/2 – OP $\sin(\theta + \psi_0)$

Yaw Amplitude, ψ_0 (deg)	5	10	15	20
Minimum Clearance, C(m)	1.24	1.13	1.03	0.93

Table 4: Available clearances in pure Yaw

5.5. Combined Sway And Yaw

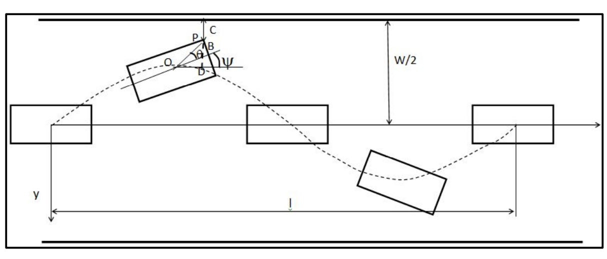

Figure 7: Model subject to combined sway and yaw

A combined Sway and Yaw motion is given to the model. Available clearance is given by;

$C = W/2 - A_0 - PD$

PD is calculated as follows;

$\tan\theta = \frac{BP}{OB} = \frac{0.25}{1.25} = 0.2 \rightarrow \theta \cong 11.31\text{deg}$

$OP = \sqrt{(BP^2 + OB^2)} = 1.274\text{m}$

From the Fig. 5, available Clearance,

$C = W/2 - [A_0 \sin(2\pi x/l) + OP\sin(\psi_0 \sin(2\pi x/l) + \theta)]$

Sway Amplitude, A_0 (m)	0.2	0.3	0.4	0.5
Yaw Amplitude, ψ_0 (deg)	5	10	15	20
Minimum Clearance, C(m)	1.04	0.83	0.63	0.43

Table 5: Available clearances in combined Sway and Yaw

5.6.Motion Parameters

Sway Amplitude, $y = A_0 \sin(2\pi x/l) = A_0 \sin(\omega t)$

Sway Velocity, $v = dy/dt = A_0\, \omega\cos(\omega t)$

Sway acceleration, $\dot{v} = d^2y/dt^2 = - A_0\, \omega^2\sin(\omega t)$

Yaw Amplitude, $\psi = \psi_0\sin(\omega t)$

Yaw Rate, $r = \dot{\psi} = \psi_0\omega\cos(\omega t)$

Yaw acceleration, $\dot{r} = \ddot{\psi} = - \psi_0\, \omega^2\sin(\omega t)$

6. Recommended Standard Pmm Test Procedures [4]

6.1.Harmonic Test

- Pure Sway
- Pure Yaw
- Pure Yaw with rudder deflection
- Pure Yaw with drift(Combined Sway and Yaw motion)

6.2.Ittc Guidelines For Pmm Test

1) Scale – as large as possible

2) Model length, L >2m(mean value 4.5m)

$L_{model} = 2.147\text{m}$*

3) Ratios of model to Tank dimensions

- Length of TT $\geq 35L_{model}$

$L_{Tank} = 35 \times 2.147 = 75.145m$*

- Mean ratio of model length to tank width = 0.42

$\frac{Lmodel}{W} = \frac{2.147}{3.2} = 0.671$*

4) Water Depth, $h \geq 5T$

$h \geq 5 \times 0.2896 = 1.448m (<2.5m)$*

5) Test speed, $u < 0.75\sqrt{(gh)}$

$u < 0.75\sqrt{gh} < 0.75 \times \sqrt{(9.81 \times 2.5)}$

$u < 3.71m/s$*

At the design speed of 0.86m/s, L_{model} reduces to satisfy Froude's No. criteria.

*Values available for the PMM test for the given carriage and towing tank.

6.3.Kinematic And Ship Control Parameters

- Forward speed
- Propeller Rate
- Amplitudes of lateral velocity and acceleration

Assuming a forward carriage velocity of 3.71m/s

i.e. $u_0 = 3.71m/s$, $\omega = 2.33rad/s$

6.3.1.Pure Sway

Maximum Amplitude = 0.4m

Maximum lateral velocity = 0.932m/s

Maximum lateral acceleration = $2.171m/s^2$

- 4. Amplitudes of angular velocity and acceleration

6.3.2.Pure yaw

Maximum amplitude = 15deg

Maximum angular velocity = 0.6096rad/s

Maximum angular acceleration = $1.4205rad/s^2$

- 5. Non-dimensional circular frequency, $\omega' = \omega l/u$
- 6. Drift angles, β : [0°, ±16°]

- 7. Rudder deflection, δ : [-20°, +30°]

6.4.Operational And Analysis Parameters

- Oscillation frequency, ω = 0.54rad/s for u = 0.86m/s

  $\omega_1' = \omega l/u = 6.28$

  $\omega_2' = \omega\sqrt{l/g} = \omega_1' F_n$

  $\omega_3' = \omega u/g = \omega_1' F_n^2$

- No. of Oscillation cycles, $c \leq \frac{1}{2\pi}\frac{\text{Ltank}}{l}\omega_1'$

  L_{tank} being the available tank length

  $c = \frac{1*40*6.28}{2*3.14*10} = 5$

7.Proposed Design Options

Figure 8 shows the general arrangement of the proposed design of PMM for the towing tank facility at IIT Madras. All the geometries have been drawn based on realistic dimensions. The main components are as follows:

- Towing Carriage
- Rails
- Water(depth = 2.5m)
- Towing tank
- Tank Wall

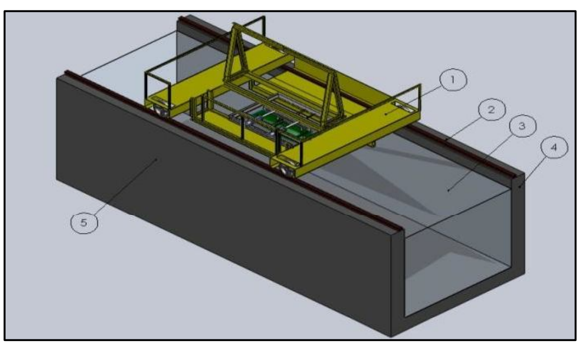

Figure 8: Bird's eye view of the PMM

7.1.Design – I

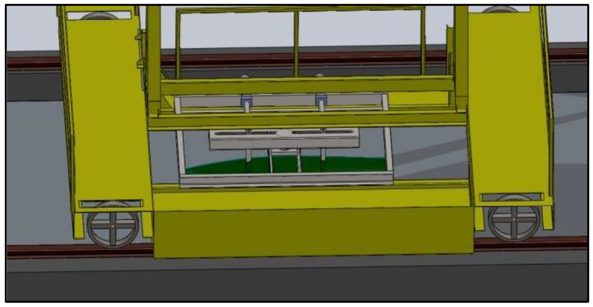

Figure 9: Side-view of the PMM

In this design, the entire mechanism is contained within a frame which rigidly connects the PMM to the towing carriage. Two synchronized linear electronic oscillators impart sinusoidal motion to the ship model while it is towed along the centre-line of the tank. As shown in figure 9, the model is connected to the model via two struts and a beam. The phase difference between the two oscillators determines the mode of motion and the parameters measured.

7.2.Design - II

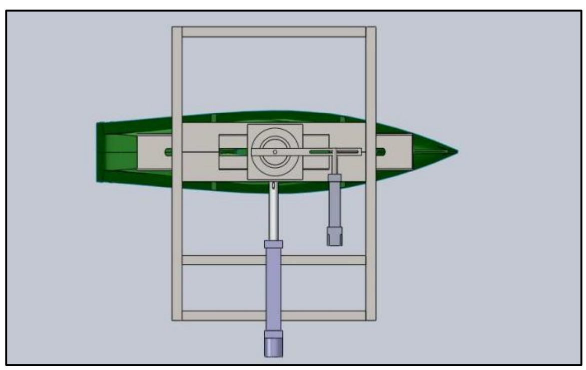

Figure 10: Top-view of the PMM

The two electronic oscillators used in this design operate independently to impart sway and yaw motion to the model. This design is just an improvement over design I from operational point of view. The structural components are more or less same.

8. Calculations

Based on above two designs for PMM, force and moment has been calculated to specify the details of the instrumentation required for this particular vessel.

8.1.Sway Motion

Max. permissible sway amplitude, $A_0 = \pm 0.5$m

Tank wall clearance = 0.85m(for $A_0 = \pm 0.5$m)

Sway velocity amplitude = $A_0\omega$ = 1.165m/s

Sway acceleration = 2.71m/s^2

Total mass subjected to sway(including model) = 200kgs

Force required = 543N

Friction between structural components = $0.14 \times 200 \times 9.81$ = 275N

Total Force = 818N

8.2.Yaw Motion

Max. permissible yaw amplitude, $\psi = 15°$

Tank wall clearance = 1.03m(for $\psi_0 = 15°$)

Max. angular velocity, r = 0.6095rad/s

Max. angular acceleration, α = 1.42rad/s^2

Moment of Inertia, I = 19.608kg-m^2

Torque $\tau = I\,\alpha$ = 27.84Nm

Power, $P = \tau r$ = 16.97W

9. Summary And Conclusion

The proposed design is found to be suitable for the existing towing tank facility. The model is free to heave and pitch during test. The forces acting on the model will be recorded, measured and analysed to get the appropriate derivatives. Installation of PMM test facility in the towing tank for conducting model test will be taken up in due course of time. Results obtained from these tests will be validated with experimental values from other sources and also with the numerical ones. The data will be analysed to estimate the hydrodynamic derivatives to predict the manoeuvring characteristics of the ship.

An attempt has been made to design the PMM for the existing carriage and towing tank facility at IIT Madras. Initial design of PMM has been presented with the details of each component. Once installed, the facility will cater to the needs of academia and industry as well.

10.Acknowledgement

The author sincerely acknowledgesDr. P. Krishnankutty, Professor, Department of Ocean Engineering, Indian Institute of Technology Madras for his constant support and guidance for this work.

- **Nomenclature**

L = Length of the ship (m)

B= Beam of the ship(m)

d =Depth of the ship (m)

T= Draft of the ship(m)

$u_{o=}$ Carriage speed(m/s)

l = Length of one oscillation(m)

y= Transverse displacement(m)

A_0= Sway Amplitude(m)

Y= Sway Force(N)

ω= Oscillation frequency(rad/s)

ψ= Yaw angular displacement(rad)

ψ_0 = Yaw amplitude (rad)

N = Yaw moment(Nm)

ρ= Density of water(kg/m^3)

∇= Displaced volume(m^3)

W = Width of the towing tank(m)

C = Clearance between tank wall and model(m)

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