Lecture 9

Additional Seakeeping Topics

As explained in previous lectures the seakeeping ability or seaworthiness of a vessel is a measure of how well-suited this asset is to operate under casual or extreme conditions. A ship or boat which has good seakeeping ability is seaworthy and able to operate for the benefit of her owners, passengers and crew safely even in different sea states. This practically means overall seakeeping performance uncertainties and effects are well managed at concept design stage. With this in mind this lecture reviews key topics associated with the influence of motions on passenger/crew habitability. It also briefly outlines key information on the important role of model tests at preliminary design stage or for the assessment of ship dynamic performance during operations.

9.1 Seakeeping criteria

Excessive ship motions can have very negative effects on passengers and crew. The people onboard a ship can experience motion sickness and difficulty to move in a controlled and coherent manner so the performance of everyday tasks is impaired. The inner ear detects changes in magnitude and direction of apparent gravitational acceleration. Motion sickness is exacerbated if the person is :

- confined below decks (can't see the horizon)
- facing diagonally across the ship
- anxious
- fatigued
- hungry
- smelling strong smells or eating or smelling greasy foods
- reading
- drinking carbonated or alcoholic drinks

The symptoms of seasickness generally disappear after a few days at sea when the person becomes acclimated. Motions can impair the ability to work effectively even when there are no problems with seasickness. The main cause of motion sickness is believed to be the vertical acceleration experienced by the person (which varies with location on the ship). Other motions can cause motion sickness if sufficiently high, but are not common on conventional ships. It is very difficult to predict the occurrence of seasickness as individuals differ in their susceptibility to motions. Even a single individual's responses may vary from day to day, depending on the other factors mentioned above. Having a job to do versus thinking about how awful you feel can affect how much you suffer. Since a deterministic approach is not realistic (there exists no if this, then that relationship that is always true), a statistical approach is required. There are several measures used currently to assess ship designs in terms of the influence of motions on crew and passengers. A brief summary of these measures is given below. More information can be found in (Lloyd 1998) .

Motion Sickness Incidence (MSI): Experience shows that the principal cause of sea sickness appears to be a result of vertical accelerations, w. Experiments were carried out in the seventies with 300 male volunteers in the USA. The experiments were inside a closed cabin, subject to sinusoidal vertical motion with amplitude up to 3.5m. The MSI is defined as the % of participants who vomited in the first two hours of the experiment. e MSI was expressed in the following form:

$$MSI = 1\left[\quad + \quad \left(\quad \ddot{} \quad - \mu \right)\right] \tag{9-1}$$

where $\mu = - \quad + \quad \left[\quad \omega \quad \right]^2$ and the error function is defined as follows:

$$erf\ (x) = \frac{1}{\sqrt{2\pi}} \int_0^x \exp\left(-0.5z^2\right) dz \tag{9-2}$$

$\ddot{w}$ is the vertical acceleration averaged over a half motion cycle and ω_e is the wave encounter frequency (rad/s). If one assumes that ship accelerations follow a Gaussian distribution, then $\ddot{w} = 0.798\sqrt{m_{\ddot{w}}}$ in (ms^{-2}), where $\sqrt{m_{\ddot{w}}}$ is the root mean square (RMS) value of the vertical acceleration.

Subjective Magnitude (SM): In the mid-seventies a number of pilots were subjected to an experiment of sinusoidal vertical motions using a chair capable of amplitudes of up to 1.5m. The objective of the experiment was to roughly quantify the influence of motions on ability to work effectively. A reference motion at 1Hz with acceleration of 0.6g was assigned a value SM=10. A motion that was judged to be twice as severe was assigned a value SM=20, half as severe SM=5 and so forth. The data obtained were expressed in the following form:

$$SM = A\left(\frac{\ddot{w}}{g}\right)^{1.43} \tag{9-3}$$

where A is wave encounter frequency dependent and expressed as

$$A = \left[\quad - \quad \omega \quad + \quad \omega \quad \right]\left[\quad - \quad - \quad \omega \quad \right] \tag{9-4}$$

The acceleration amplitude can be taken, for example, as one half of the significant acceleration, namely $\ddot{w} = 2\sqrt{m_{\ddot{w}}}$, where $m_{\ddot{w}}$ is the MS value of the vertical acceleration. Using this assumption a plot of SM against RMS acceleration can be generated and the subjective regions: moderate (SM=5), serious (SM=10), severe (SM=15), hazardous (SM=20) and intolerable (SM=30) can be identified.

Motion Induced Interruption (MII): MSI is a reasonably adequate parameter for judging severity of motion for passengers. However, it is not very relevant to the ability of crew to function effectively. On the other hand, SM is a rough measure of how severe a motion feels, but still not relevant to ability of crew to function effectively. MII is based on the frequency that a member of the crew has to stop work and hold on to a suitable anchorage to prevent loss of balance due to sliding or tipping (e.g roughly SM>10). The number of MIIs per minute can be expressed as follows, based on investigations carried out in the 1980s and early 1990s :

$$MII = \frac{60}{T_z}\exp\left[-\frac{\nu}{\quad\ddot{a}}\right] \quad (\min)^{-1} \tag{9-5}$$

where T_z is the average zero crossing period of the seaway and $m_{\ddot{a}}$ is the MS value of the total acceleration, including both lateral and vertical accelerations. The latter are evaluated at the right or left foot of the crew member depending on whether there is sliding or tipping to port or starboard. The MIIs to port and starboard are added together. The aforementioned equation is valid for either sliding, where ν is the friction coefficient between the deck floor and the crew member's shoes, or tipping, where $\nu = l / h$ (h : distance from deck floor to crew member's CoG and l : half-stance distance).

9.2 Experimental methods for seakeeping and wave loads

9.2.1 Model scale facilities

Despite advances in computational methods during preliminary ship design motions and loads are obtained experimentally by model tests in towing tanks for various speeds and headings to obtain RAO curves. This is because numerical codes have not yet reached a completely satisfactory stage, especially for complex hydrodynamic problems laboratory test is still essential to have a reliable relevant design data.

As part of this process pressure measurements at various parts of the underwater part of the hull can also be obtained by integration over the wetted part of the hull. Model tests can be performed also in irregular waves by generating a seaway say from a particular wave spectrum and towing the hull at various speeds and headings to obtain the time record of ship motions. These model tests as well as full scale measurements taken at sea are used to validate theoretical predictions.

In early days experimental model tests aimed to only improve the resistance performance of ships. The first well known recorded model test was performed by Leonardo da Vinci in the 15^{th} century. He tested three models with equal length and different bow and aft shapes. Based on his experiments, he came out with recommendations about the best shape that can give the highest forward speed. Several tests have been carried out after that. All these tests assumed the testing models reflect the full-scale ship performance, until the 18^{th} century when William Froude introduced a scaling concept to obtain the actual ship resistance from the model tests. This approach was a cornerstone in the development of the model testing for ship design. Modern model testing initiated in 1870 in the Froude towing tank in South England. The tank has a length of 85 m, breadth 11m and 3 m depth.

Commercial towing tanks are usually in the range of 250 m length, 10 m breadth and 5 m depth and the corresponding typical model usually has 5:8 m length. Almost all the towing tanks have a towing carriage to move the model through the water with a typical speed 10 m/s. To carry out seakeeping tests or another type of experiments that need surface waves, the tanks should be equipped with wave makers at one end of the tank and a wave absorber (known as wave beach) at the opposite side to prevent wave reflections. Some specialised towing tanks can decrease the pressure in the tank area to 0.04 bar to obtain the same cavitation number of the fully-scaled ship in propulsion tests. Ice tanks can simulate the ice environment for testing icebreakers and other offshore structures exposed to drifting ice. The ice is modelled by freezing, employing high salinity water and chemicals to control its mechanical properties. Example of this facility can be found at Aalto University Ice tank (https://www.aalto.fi/en/research-and-learning-infrastructures/aalto-ice-tank). Typical towing tank facilities are illustrated in Figure 9-1.

One of the largest modern towing tanks today is situated at David Taylor Naval Ship R&D center in USA. The tank has a length of 1,000 m, 6.4 m breadth and 3 m depth and a carriage with a maximum speed of 50m/s for testing high-speed ships. In recent years the rapid development of ship technology research encouraged building more specialized testing facilities such as cavitation tunnels and the maneuvering and seakeeping basins. These new testing facilities made it possible to simulate realistic ocean conditions like wind, current and multidirectional waves and to have a better explanation of the physical problems. Moreover, it has a significant impact on the latest development of the theoretical methods, seakeeping software and other related numerical codes. Cavitation tunnels are used to test propellers at a low pressure to obtain the same cavitation number of the actual full-scale system. The typical size of the conventional tunnels is 1 m diameter with maximum flow speed at measuring section ranges between 10 and 20 m/sec. Berlin tunnel is one of the largest tunnels that may fit an entire ship hull model. It has a test section dimensions of 3 × 6 m with 11 m length. Some tunnels have a free surface to allow for studying ventilation of the propellers, water jets and foil sections. Ocean basins are used frequently to model a realistic ocean environment for testing offshore structures, and in addition, they are used to carry out seakeeping and maneuvering tests of ships. The advanced facilities in the ocean basins are often capable to generate long-crested waves with different directions (head, oblique or beam seas) or short crested waves (multidirectional) as well as current and wind.

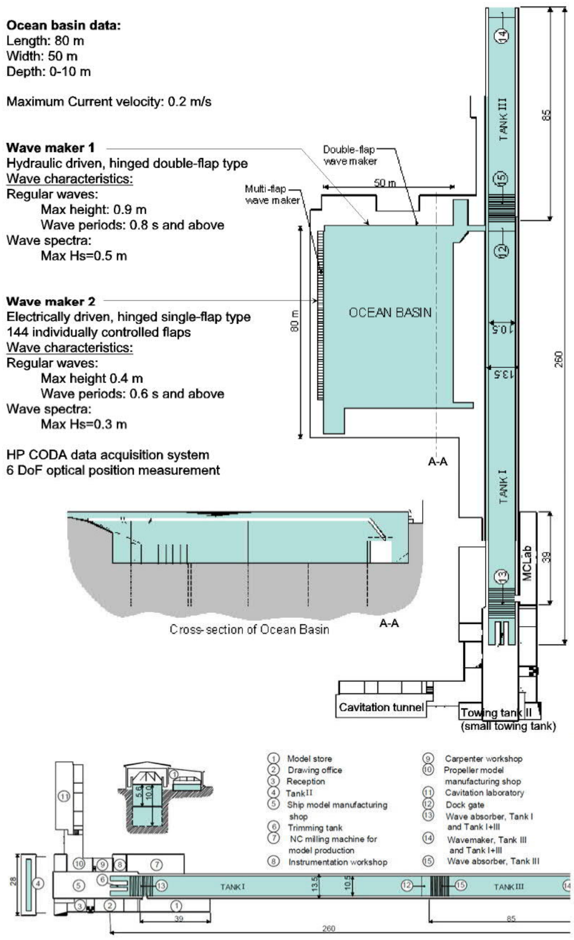

Figure 9-1 Typical layout of the facilities used in ocean basins, towing tanks and cavitation tunnels (Steen 2014)

9.2.2 The importance of similarity laws

Laboratory tests aim to investigate the behavior of the actual full-scale system by using a much smaller scale physical model. The dimensions of the physical models are usually restricted by the available laboratory facilities. These dimensions should be scaled appropriately to obtain similar behavior of the full-scale system, i.e. the results for the model can be transferred to full scale by a proportionality factor. The similarity in forces between the physical model and the full-scale system can be achieved by fulfilling the following conditions namely geometric, kinematic and dynamic similarity. Geometric similarity means that the ratio of the full-scale length L_F (maybe also breadth, draft, deformation etc.) to the model scale length L_M is constant and equals the model scale factor λ:

$$L_F = \lambda \cdot L_M \tag{9-6}$$

By applying the geometrical similarity law we can ensure that the model has the same shape of the full-scale system. However, the target from the geometrical similarity is not to obtain the same shape of the full-scale system but to get the same hydrodynamic performance. Kinematic similarity means that the ratio of full-scale times t_F to the model scale times t_m is constant and equals the kinematic model scale τ:

$$t_F = \tau \cdot t_M \tag{9-7}$$

Dynamic similarity means that the ratio of all forces that acting on the full-scale system (inertial, gravity, and frictional forces) and the ones acting on the model is constant and equals the dynamical model scale κ:

$$\mathrm{F_F} = \kappa \cdot \mathrm{F_M} \tag{9-8}$$

9.2.3 Physical model testing

Design of a physical model is a crucial part for model testing. It depends on the available experimental facilities, type of model test and the instrumentation required. It may also depend on the material used in the construction, wood, wax, aluminium, glass, GRP or foam. The size of the model should be optimized to reduce the errors from viscous effects and to improve the measuring accuracy. At the same time, the building and execution costs should be rational. This is usually restricted by the size of the tank. Tank walls will influence the test results if the ratio between the model and tank size is small, which is defined by the blockage effect. The two main types of physical models, rigid and elastic models are discussed in the following sections.

In dynamic testing of rigid seakeeping models the model is usually free to move as the inertia forces will have a significant contribution to the test results. Hence the mass, centre of gravity (transverse, vertical and longitudinal), moment of inertia and gyration radios of the model should be scaled accurately. In case the test aims to measure the internal loads, the distribution of the weight should be also correctly modelled. The mass is simply measured by weighting the model or by ensuring it is floating at the correct waterline. The center of gravity and the inertia moment is usually measured by pendulum test (see Figure 9-2). First the pendulum period T_0 is measured which is given by:

$$T_0 = 2\pi\sqrt{\frac{I}{Mgh}} \qquad \longrightarrow \qquad 4\pi^2 I - MgT_0^2 h = 0 \tag{9-9}$$

where M is the mass of the model and I is the moment of inertia at point A.

Then two additional masses of weight m are added at each end of the pendulum at a distance a from its center A. The new period T_1 is measured to get the following equation:

$$4\pi^2(I + \Delta I) - (M + 2m)gT_1^2(h + \Delta h) = 0 \tag{9-10}$$

where $\Delta I = 2ma^2$ and $\Delta h = \frac{2mh}{(M+2m)}$. By solving the two equations above we get h and I.

$$h = \frac{8\pi^2 ma^2}{Mg(T_1^2 - T_0^2)}$$
$$I = \frac{2ma^2 T_o^2}{T_1^2 - T_0^2} \tag{9-11}$$

Then the moment of inertia at the center of gravity equals $I_\mathrm{m} = I - Mh^2$ and the metacenter height of the model can be calculated easily. It is a common practice to calibrate the pendulum test by static inclination test, if any deviation is observed, the inclination test is assumed the most accurate. In order to measure the internal loads (as shear force and bending moment), the model should be divided into several sections. At each section, the pendulum test should be repeated to ensure correct mass and moment of inertia of each section.

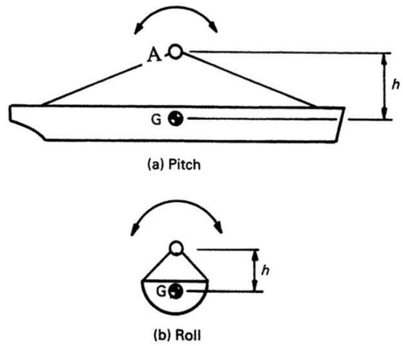

Figure 9-2 Sketch of a pendulum test

There are three different approaches that can be adopted to design and build the elastic models of ships namely backbone model, fully elastic model and hinged model as illustrated in Figure 9-4 For the backbone model, the elasticity is modelled by an elastic beam to which the rigid segments are attached. As the elasticity of the beam is easy to amend, for example by using a material with suitable material properties, this model is relatively simple to construct. The main drawbacks of this type of modelling are the gaps between the rigid segments which may affect the test results especially if the model has a forward speed. The gaps may be filled with an elastic membrane, but it is difficult to avoid the transfer of tension trough the membrane. The fully elastic model is usually built up using 2 materials, for example, fibre-glass and foam, while the thickness can be amended to achieve the required elasticity. This type of elastic model does not have any gaps as in the backbone model; however, it is difficult to achieve the required bending stiffness distribution and in addition, the foam induces too high damping comparing with the backbone model. The high damping may have a considerable effect on springing test results; while the whipping forces are not sensitive to the structural damping level. It is noteworthy that the backbone model and the fully elastic model are usually used with monohull vessels. The hinged model is usually adopted in catamaran model testing. Figure 9-3 illustrates a catamaran hinged model with three hulls built from normal material as the rigid models (foam or GRP), each hull consists of three rigid segments connected by slender steel beams (springs). The dimensions of the beams can be selected to give the required bending and torsional stiffness. The gaps between the segments are usually fitted with rubber membrane to make the model watertight. In seakeeping tests, the gravitational forces (forces from waves) dominate the test results, hence the Froud scaling should be adopted accurately. The waves height should also be scaled with the same geometrical scale ratio, while the wave period follows the square root of the scale ratio. Furthermore, as the dynamic motions are the key result from the seakeeping analysis, the inertia forces will be vital and inconsequence the mass distribution should be scaled precisely. Possible alternative seakeeping test set-ups are illustrated in Figure 9-4. The rigidly restrained model can be used for measuring only the wave excitation forces. In alternatives (b), (c) and (d) the model is free to heave and pitch while the surge motion is restricted in (b) and allowed with different levels in (c) and (d). Notably (d) is a self-propelled model and is usually adopted with (c) in towing tank testing in head and following seas. On the other hand (e) is usually adopted in seakeeping basins tests where the model is self-propelled and only connected to the carriage using very soft cables. Final set up (f) the model is controlled remotely.

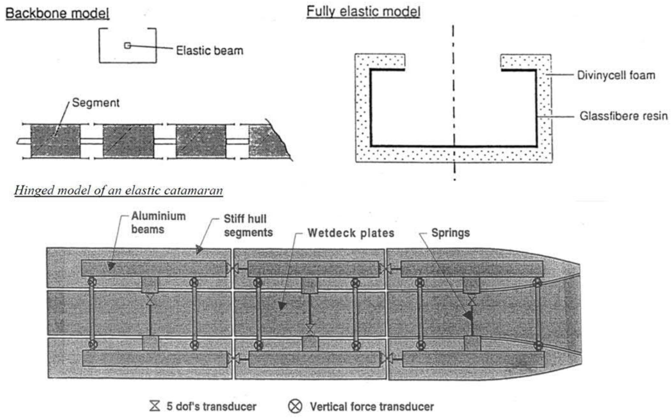

Figure 9-3 Types of elastic models

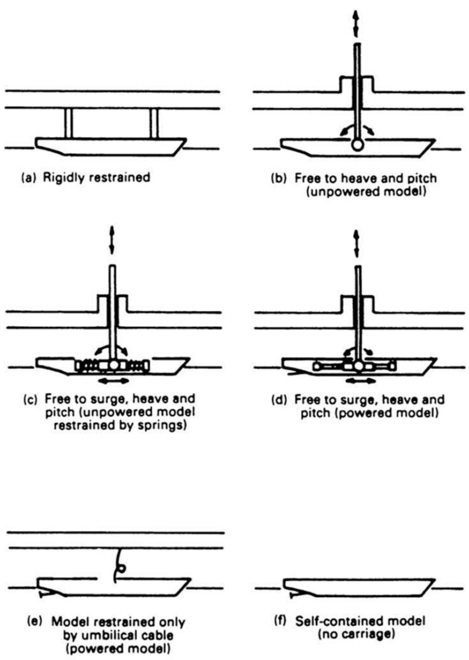

Figure 9-4 Possible alternative of the seakeeping test set-up

Seakeeping tests can be conducted in artificial regular and irregular waves. In regular waves, the RAOs and the added resistance in different regular wave frequencies are the main outcomes from the test. It can be used to validate and calibrate the numerical results. The artificial irregular waves can be adopted to study the performance of the ship in real sea conditions and also to conduct statistical and extreme response analysis.

9.2.4 Full scale measurements

Full-scale measurements are carried out to assess the performance of the full-scale ships in the real environment. Comparing with the tank testing, the uncertainties in the measured response due to the scaling errors do not exist in the full-scale measurements, however, the uncertainties from the stochastic marine environment are much serious and complicated. The full-scale tests are usually essential in three different circumstances:

- Delivery/sea trials
- Malfunction of a ship system(e.g propeller noise or excessive fuel consumption).
- Research purposes

Sea or delivery trials are usually conducted after building the ship and before delivering her to the owner. The main objective of the sea trials is to measure the speed-power characteristics of the ship in still water condition. Almost all ship contracts contain strict requirements for the speed the ship must achieve at certain engine power levels. There are usually penalties if the recorded speed of the ship at a certain engine power level is less than the contract by typically 0.2 knot; while a larger deviation, typically from 0.5 to 1 knot, may give the right to the owner to refuse to take the ownership of the vessel. On the other hand, If the recorded speed is more than the stated in the contract, the yard usually gets a bonus. In case the yard quoted lower speeds in the bidding stage to avoid any penalties in delivery, that may expose it to losing the intense competition for getting the contract in the first place. This background emphasizes why the model testing is substantially important and so common before starting building the ship.

In speed trials, the vessel forward speed is recorded while running the ship in a straight track in both directions Figure 9-5 (double run); the trial should be repeated for at least three different engine power levels. The speed can be measured using GPS or by clocking the time spent per measured mile. The power level of each trial can be obtained by measuring the shaft torque using strain gauges or special torque meters, also it can be determined based on the fuel consumption readings. The test is usually conducted in deep water without waves (still water) nor wind and at sea temperature around 15 °C. The draft or loading condition of the ship during the test is usually stated in the contract (usually the ship design draught). The environmental condition during performing the test (water depth, draft, wind speed, wind direction, wave height, wave direction, current, water salinity, water temperature etc.) should be reported, see Table 9-1 for the typical and max acceptable contractual conditions. In case there is any deviations from the contractual environment, the results of the test should be corrected to get the corresponding test results to the contract actual conditions. Further information about the environmental condition and correction procedures can be found in standard ISO 19019 and in the ITTC standard procedure "Procedure for the Preparation and Conduct of Speed/Power Trials".

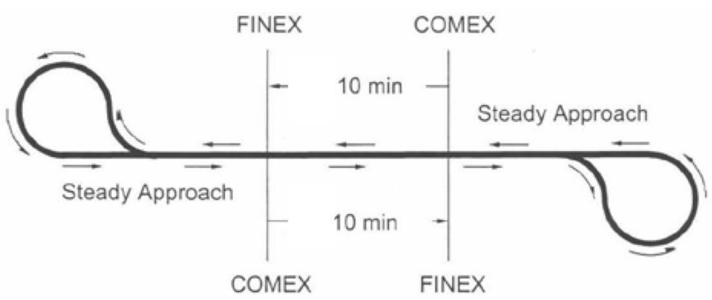

Figure 9-5 Trial trajectory of one double run (ITTC- recommended procedures and guidelines)

Table 9-1 : Typical and max acceptable contractual environmental conditions

Environmental conditions	Typical contractual conditions	Max acceptable trial conditions
Sea state (measured by visual observation, wave buoys, wave radar or Bow-mounted altimeter)	No waves or Beufort 1 (wave height 0.1 m, see Table 9-2)	Preferably ≤ sea state 3 Ultimately ≤ sea state 5 (or up to sea state 6 for ships with L>100 m)
Wind	No wind or Beufort 2 (Wind speed ≤ 6 knots, see Table 9-2)	Wind ≤ Beufort 6 (20 knots) (for ships with L>100 m) Wind ≤ Beufort 5 (for ships with L ≤ 100 m)
Water depth (h)	Deep water or $h > 6\sqrt{A_M}$ and $h > \frac{1}{2}V^2$ where A_M is the midship section are and V is the speed.	Smaller water depth will require correction for shallow water effect
Current	No current. Use of double runs means that corrections are always included	Current of more than a few knots is unacceptable

Table 9-2 The Beufort wind and wave scale

Beaufort	Sea state	Description		Wind speed		Wave height	
		wind	wave	min	max	probable	max
0	0	Calm	Calm	0	1	0	0
1	0	Light air	Ripples	1	3	0.1	0.1
2	1	Light breeze	Small wavelets	3	6	0.2	0.3
3	2	Gentle breeze	Large wavelets	6	10	0.6	1
4	3	Moderate breeze	Small waves	10	16	1	1.5
5	4	Fresh breeze	Moderate waves	16	21	2	2.5
6	5	Strong breeze	Large waves	21	27	3	4
7	6	Near gale	Large waves	27	33	4	5.5
8	7	Gale	Moderately high waves	33	40	6	7.5
9	8	Strong gale	High waves	40	47	7	10
10	9	Storm	Very high waves	47	55	9	12.5
11	9	Violent storm	Exceptionally high waves	55	63	11.5	16
12	9	Hurricane	Exceptionally high waves	63	71	14	16
13	9	Hurricane	Exceptionally high waves	71	80	> 14	> 16
14	9	Hurricane	Exceptionally high waves	80	89	> 14	> 16
15	9	Hurricane	Exceptionally high waves	89	99	> 14	> 16

9.3 References

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