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ROLL MOTION OF YACHTS AT ANCHOR

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SUMMARY

Yachts tend to roll uncomfortably whilst at anchor, causing discomfort to the crew and passengers, generating additional stresses on equipment, and making operations such as embarking and disembarking hazardous activities. A research program is under way to better understand the design factors and environmental influences leading to the rolling problem, with a view to providing effective solutions. Two quite different sets of experiments have been conducted to date - model tests in a wave basin and forced oscillation tests in a calm water tank. The results hold interesting implications for the design of those yachts for which safety and comfort when not under way are important criteria.

NOMENCLATURE

All units are SI unless otherwise stated.

a = roll inertia of the yacht and the surrounding water
A = plate profile area
AR = aspect ratio
b = roll damping constant
c =stiffness constant
C_D = drag coefficient
C_M = inertia coefficient
C_φ = total roll moment coefficient
d = water depth
D = cylinder diameter
f = natural roll frequency (Hz)
dF = force per unit length across the flow
g = acceleration due to gravity
GM = transverse metacentric height
h = distance from the tip of the plate to the bottom of the channel
k = roll gyradius
M = wave exciting moment
s = plate span
u = instantaneous local fluid particle velocity
U = maximum velocity at plate tip
x = instantaneous local sway position
w = dimensionless frequency
φ = roll angle
$\dot{\varphi}$ = roll angular velocity
$\ddot{\varphi}$ = roll angular acceleration
ρ = fluid density
σ = added inertia coefficient

1 INTRODUCTION

Yacht owners invest considerable resource in acquiring a yacht which is comfortable and safe. One of their aims is to be able to anchor in secluded bays in a relaxed atmosphere. This aim is lost if the vessel starts to roll. Roll motion is a nuisance on both motor yachts and sailing yachts for a variety of reasons:

- It causes sea sickness.
- Crew and passengers may fall and hurt themselves.
- Embarking and disembarking become difficult and possibly dangerous.
- Noise is generated through water slap on the hull and motion of inadequately secured objects.
- Some on-board equipment will not perform adequately.

All yachts roll to a greater or lesser extent when subject to waves. When the vessel is on passage and travelling at reasonable speed the roll motion is often limited through forces generated by the flow around the hull, or by the use of fin stabilisers. For sailing yachts, additional roll reduction is obtained from aerodynamic forces. However, when the vessel is moving slowly or is at anchor, those roll-stabilising forces are not present; a different solution is required.

2. BACKGROUND

2.1 ROLL REDUCTION GOALS

The most obvious way of reducing roll motion is to avoid anchoring in waves. Unfortunately most of the more attractive bays are, by nature of their geography, places where waves can work their way into, so there is no avoiding them. If the yacht were anchored exactly head on to the waves there would be no roll motion. However, yachts tend to lie to the wind and current direction rather than wave direction. Furthermore, a wave field exhibits directional spreading about its principle direction, making it impossible for a yacht to remain head on to all the waves. Hence the goal is to minimise the vessel response to waves. Many possible solutions are available but first the roll characteristics of a yacht must be described in a meaningful way.

The equation of motion of a yacht rolling may be written in its simplest form as a linear uncoupled equation:

$$a\ddot{\varphi} + b\dot{\varphi} + c\varphi = M \qquad (1)$$

where

φ = roll angle

$\dot{\varphi}$ = roll angular velocity

$\ddot{\varphi}$ = roll angular acceleration

a = roll inertia of the yacht *and* the surrounding water

b = roll damping constant

c =stiffness constant

M = wave exciting moment

The solution of the equation varies both with wave frequency and height. The roll characteristics of the yacht are described by the coefficients a, b and c. The search for roll minimisation requires an understanding of the design factors affecting the constants in the equation of motion. There are two possible targets that design solutions can aim at - avoiding resonance and increasing damping.

2.2 NATURAL FREQUENCY

Resonance occurs when the wave frequency matches the natural roll frequency of the yacht. For a lightly damped motion such as roll, the natural frequency may be calculated from :

$$f = \frac{1}{2\pi}\sqrt{\frac{a}{c}} \qquad (2)$$

where

f = natural roll frequency (Hz)

The roll inertia of the yacht a may be reconfigured in terms of the gyradius in roll and the roll inertia coefficient:

$$a = \Delta k^2(1+\sigma) \qquad (3)$$

where

Δ = mass displacement

k = roll gyradius

σ = roll added inertia coefficient

and the stiffness term may be written as:

$$c = \Delta gGM \qquad (4)$$

where

g = acceleration due to gravity

GM = transverse metacentric height

leading to a more useable version of the equation (2):

$$f = \frac{1}{2\pi}\sqrt{\frac{gGM}{k^2(1+\sigma)}} \qquad (5)$$

A vessel with 4m beam would have a natural roll frequency of about 0.25Hz. Unfortunately many bays will have a substantial amount of wave energy at this frequency. Can this resonant condition be avoided? It is evident from equation (5) that the natural frequency of the yacht depends on two main factors:

- Transverse stability.

A vessel with a large transverse metacentric height will have a higher natural frequency than an equivalent vessel with a low metacentric height. However there are severe design constraints on transverse stability as the *GM* is usually strongly controlled by stability regulations and, for sailing yachts, sail carrying power. At first glance it might seem that the frequency could be increased by increasing the beam. However, *GM* increases approximately as the square of the beam so a wide yacht will usually have a higher natural frequency.

- Roll mass moment of inertia

The roll mass moment of inertia comprises the roll inertia of the yacht, *and* the inertia of the water particles surrounding the yacht that are accelerated as a consequence of the yacht motion - the added inertia. A yacht with a large mass moment of inertia in roll will have a lower natural frequency than a yacht with a small inertia. So if there are large masses on board which are placed either at the maximum beam, or very high up or very low down, the roll inertia will be large and the frequency low. The mass inertia is controlled by the mass distribution and general arrangement of the yacht, which usually have stronger demands on their function than optimising the natural frequency of roll.

The added inertia of the surrounding water is determined by the underwater shape of the vessel. A yacht with semicircular cross sections and very small appendages will have very little added inertia. A yacht with sections that are more square or triangular in shape will have a higher added inertia, as water must be accelerated as the shape rolls through the water [1]. A keel will contribute significantly to added inertia as some of the water must accelerate with it as it rolls [2], [3].

The added inertia can be changed by changing the hull shape, but the resulting change in frequency is very small. e.g. doubling the added inertia coefficient typically only makes a 5% change in the natural roll frequency.

Note that the mass of the yacht does not enter into the equation; a heavier yacht will have the same roll frequency as a lighter yacht of equivalent shape and mass distribution. However, heavier yachts often do not require such a high *GM* and the extra mass is often in the extremes of the yacht (top, bottom and well outboard) leading to a higher gyradius. So a heavy yacht tends to have a longer roll frequency than a light one because of the gyradius and *GM* change, not because of the mass difference.

2.3 DAMPING

Roll damping is generated by a number of mechanisms. The biggest contribution comes from generating vortices (large eddies) as the yacht rolls. Vortices are most easily generated at sharp edges e.g. chines, keels and rudders. The next most significant contribution comes from generating waves as the yacht rolls. A yacht hull with square or triangular sections will generate more waves as it rolls than does a yacht with circular sections. There is also a damping contribution from the friction between the water and the rolling yacht, but this is usually so small it can be neglected.

2.4 NUMERICAL MODELLING

In order to provide design solutions to the rolling yacht problem a computer technique is required so as to be able to model the numerous "what-if"s in the design process. Unfortunately, most commercial seakeeping software does not deal very effectively with roll motion compared with pitch or heave, particularly the effects of large appendages and non-linearity with respect to wave amplitude. A number of research level codes have been developed based on discrete vortex methods, which are able to model the generation and shedding of vortices from the appendage as it rolls in waves [4], [5]. However, they are only applicable to two dimensional appendages such as very long bilge keels. As yet there is no practical CFD method of dealing with keels and rudders oscillating in separated flow near the free surface. Different techniques have to be used. One of these is under development at Curtin University to overcome this problem, and it was this development that provided the impetus for the experiments described below.

3 SCALE MODEL TESTS

3.1 METHODOLOGY

One of the most immediately effective ways to find out how different design configurations perform in waves is to conduct scale model experiments in a wave basin. A series of tests was conducted with the fundamental aim of determining the effect on roll motion of a substantial change in keel size [6]. The model tests were simplified by using a circular cylinder hull. This hull shape was chosen in order to minimise wave damping and eddy damping from the canoe body. Thus any change in roll response could be attributed directly to the change in keel size. Three appendage geometries were investigated:

- a full depth rectangular flat plate
- a full depth rectangular aerofoil section keel
- a half depth rectangular flat plate.

The model was free to roll, pitch and heave, with the constraint attachment points at the waterline to minimise roll moments caused by sway and yaw restraint.

Since the objective was to measure the hydrodynamic variations between keels, the flotation plane and natural roll frequency in air were kept constant. In this way, the small variations in displaced volume, mass moment of inertia and GM between keel configurations did not influence the comparisons.

Testing was conducted in regular waves at constant amplitude. Wave steepness was well below the limits for wave breaking.

3.2 EQUIPMENT AND PROCEDURE

The tests were conducted in the model test basin at the Australian Maritime College. It was 35m long, 12m wide with water depth set at 0.7m for these experiments. The basin was equipped with a multi-element wavemaker and a beach was situated at the downstream end of the basin. The basin sides were vertical. The instrumentation for these experiments comprised three Linear Voltage Displacement Transducers (LVDTs) and a wave probe. Additional wave probes were used in the preliminary calibration stage.

The model was a circular cross section pipe, 0.315m diameter and 1.3m long, floating at its semi-diameter. A daggerboard case was installed, into which could be slotted one of the three keel configurations:

- a full depth flat plate keel, 300m width and exposed depth (span) 300mm, made of 6mm ply
- a half depth flat plate keel also 300mm wide and 6mm thick, but with exposed depth (span) 150mm
- a full depth aerofoil section keel, made by adding a carefully shaped fairing to each face of the full depth flat plate keel. The foil was based on a NACA 0010 section with the aft portion thickened to accommodate a 6mm wide trailing edge.

The attachment rig for holding the model in the basin consisted firstly of two box-frame square section support tables. These were placed on the basin floor approximately 3m apart. Secondly, two heavy section alloy beams bridged these tables, with the model attachment system and LVDTs connected to the beams. Two of the LVDTs were attached to the port and starboard deck edge amidships and the third was attached on the centreline towards the bow.

The procedure for each run was to acquire the zero datum for all recording channels, run the wavemaker at the predetermined wave amplitude and frequency, then start acquiring data once the waves reached the model. All channels were acquired digitally at 100Hz for 30 seconds without filtering. The time series were later inspected for evidence of wave reflection from the beach and truncated as necessary.

On completion of the beam sea tests (90° to waves), the support tables and attachment system were moved so as to align the model 120° to the waves and further tests were conducted. A comprehensive description of the experiments is provided in [6].

3.2 RESULTS

- Experimental errors

The standard deviation of the wave surface elevation showed a spatial variation of 3% and a temporal variation of up to 2.7% at low frequencies. The motions measurements were subject to errors of about 4% depending on frequency and amplitude. Overall, the 90% confidence limits of the results were approximately 8% i.e. one can be 90% certain that a value lies within 8% of the measured value.

- Wave heading

Both the full depth keel and the half depth keel were tested at wave headings of 90° and 120°. The results for the full depth keel are shown in Figure 1; the half depth keel exhibited similar characteristics. The reduction in roll response as heading angle moves away from beam seas is expected and is found in other work [7].

- Keel geometry

The effect of changing keels is shown in Figure 2. The difference in peak frequency between the half depth and full depth keels is considerable, and is most likely attributable to a very large added inertia change.

The peak roll response for the half depth keel was approximately 20% higher than for the full depth keel, albeit at a different wave frequency - this makes comparison difficult. The wave exciting moment at a particular frequency will most likely be different for the two keels, owing to the difference in lateral area and draft.

The damping of the aerofoil keel is typically 12% greater than for the flat plate keel. This may be a consequence either of a change of flow conditions at the leading edge due to the curvature introduced by the foil section, or the increase in keel thickness. However, given that the experimental error estimate is 8%, caution is urged in interpreting this result.

4 FORCED OSCILLATION EXPERIMENTS

It became evident from the outcomes of the wave basin tests that in order to gain a better understanding of the effect of appendages on roll motion, it was not sufficient just to measure the motion in waves; the roll moment generated by the appendage had to be measured.

4.1 METHODOLOGY

It was decided to build an experimental rig and conduct forced oscillation tests in calm water on a series of stylised keel shapes, measuring both the plate motion and the generated roll moment. The opportunity was also taken to measure the influence of underkeel clearance. The plate motion had to replicate that of a yacht keel in pure roll motion. Given that a vessel pivots approximately about its centre of gravity at small roll angles [8], and that the centre of gravity for many yachts is close to the waterline [9], [10], the hinges for the plates in the experiment were located at the still water level. The motion of the plate was chosen to be a close approximation to sinusoidal. Whilst ocean waves generate a pseudo-random roll motion, the theory of superposition has been successfully applied to small angle linear motions analysis and seakeeping model experiments [11], [12].

Equation (1) describes roll motion in one of many possible forms. It works reasonably well for motions and shapes where the majority of the damping is from wave making and the response varies linearly with wave amplitude. For a yacht with large appendages, the damping is largely due to viscous forces, which are better represented as a velocity squared term. In such circumstances greater insight may be gained by employing the Morison equation [13] used in offshore engineering hydrodynamics to estimate the forces on circular cylinders, presented here as equation (6).

$$dF = (C_M + 1)\rho\frac{\pi}{4}D^2\dot{u} - (C_M)\rho\frac{\pi}{4}D^2\ddot{x} + C_D\frac{\rho}{2}D|u-\dot{x}|(u-\dot{x}) \qquad (6)$$

where

dF = force per unit length across the flow

C_M = inertia coefficient

C_D = drag coefficient

D = cylinder diameter

u = instantaneous local fluid particle velocity

x = instantaneous local sway position

ρ = fluid density

This equation may be adapted for the circumstance of a flat plate undergoing forced oscillation in calm water with a pivot point at the waterline:

$$M = C_M \rho \frac{\pi}{12} s^3 \ddot{\varphi} + C_D \frac{\rho}{8} s^3 |\dot{\varphi}| \dot{\varphi} \quad (7)$$

where

M = hydrodynamic roll moment

$\dot{\varphi}$ = roll angular velocity

$\ddot{\varphi}$ = roll angular acceleration

s = plate span

The total roll moment can also be expressed in coefficient form C_φ. The various coefficients are defined as:

$$C = \frac{moment}{\frac{1}{2}\rho A U^2 s} \quad (8)$$

where

C = coefficient of interest

A = plate profile area

ρ = water density

U = maximum velocity at plate tip

s = plate span

In order to scale the results, frequency is non-dimensionalised as follows:

$$w = 2\pi f \sqrt{\frac{s}{g}} \quad (9)$$

where

w = dimensionless frequency

f = frequency of oscillation (Hz)

s = span

g = acceleration due to gravity

4.2 EQUIPMENT AND PROCEDURE

The facility used was a tank 10m long of cross section 0.3m square. The ends of the channel were blocked off and the channel filled with water to the desired level, usually 0.2m depth. Four plates were tested (Figure 3):

Plate 1: a full width rectangular flat plate, of infinite effective aspect ratio i.e. stretching right across the tank.
Plate 2: a plate approximately 0.1m square.
Plate 3: same area as plate 2, but double span.
Plate 4: same chord as plate 2, but double span.

The attachment rig for holding the plates was suspended from the channel walls. The plate was connected to a crank arm and electric motor and to two hinge supports. The rig could be moved up and down relative to the channel bed to enable tests to be conducted at different water depths whilst keeping the plate hinge at the static water level. The hinge supports and crank attachment were strain gauged in order to measure the forces and moments acting on the plate when oscillated over a range of frequencies and amplitudes. The plate motion was measured with a rotary potentiometer. Full details of these experiments are published in [14].
Tests were conducted for each plate by pre-setting the roll amplitude and frequency, taking a zero-datum measurement then acquiring the data for 20 seconds at 100Hz sample rate, using 20Hz low pass anti-aliasing filters.
The data analysis was complicated by having to remove the effects of buoyancy, mass moment of inertia and bearing friction from the signals. This was achieved by repeating all the runs in air which were then modelled as a normalised series of harmonic Fourier coefficients. The in-air signal for the frequency of an in-water run was then reconstructed and subtracted from the in-water signal.

4.3 RESULTS

- Experimental errors

Only dynamic measurements of strain were required, so problems with slowly varying strain gauge offsets were avoided. Temperature effects were accounted for in two ways: firstly, by choosing gauges with a thermal expansion coefficient similar to that of the attachment plate; secondly, by taking measurements over short duration, which were thus unlikely to experience significant change in temperature.

The constraints of operating in an enclosed channel will influence the flow; this effect is known as blockage. It has been extensively investigated for wind tunnel and towing tank experiments [15], [16] and accurate correction methods have been developed. However, such methods are not appropriate for the conditions in this experiment. Given the uncertainty in estimating blockage effects for the current experiment, the results have been left uncorrected.

Error magnitudes were a function of oscillation frequency and plate dimensions. The largest source of quantifiable error for most conditions was in the correction for buoyancy-induced moment, with errors introduced if the water level was not quite at the hinge level. Hence the percentage error was largest for the plate with the smallest span. Error bars are indicated on all figures; as a guide, the error in roll moment was 0.5% for plate 2 at high frequency, increasing to 9% at low frequency. The equivalent errors for plate 1 (which had the smallest span) were 3% and 24% respectively.

- Effect of underkeel clearance

The underplate clearance was defined as:

$$UKC = \frac{h}{d} \tag{10}$$

where

h = distance from the tip of the plate to the bottom of the channel

d = water depth

The UKC is usually expressed as a percentage. Tests were conducted on plates 3 and 4 at different under-keel clearance ratios down to a value of 1%. The influence on total roll moment coefficient for plate 4 is shown in Figure 4. Similar results were obtained for plate 3. A 1% clearance is far less than any prudent mariner would consider safe (just 50mm clearance for a typical anchorage). Therefore it was concluded that the proximity of the keel to the sea bed does not have any significant effect on the roll motion coefficients. A proviso is added that the wave particle velocities in an ocean environment will be influenced by the presence of the seabed; it is the response to those particle velocities that is not affected.

- Influence of plate geometry

The total roll moment coefficient for plate 4 is plotted as a function of dimensionless frequency for a range of angle amplitudes in Figure 5. The coefficient was, to an engineering approximation, independent of frequency. The other 3-D plates (plates 2 and 3) performed similarly. The 2-D Plate (plate1), however, showed entirely different characteristics from the other three plates (Figure 6). The roll moment coefficient varied with frequency in a highly structured manner and there was evidence of a transitional flow regime at a dimensionless frequency of approximately 0.8. Transitional effects on 2-D plates have been found by other researchers [17]. The significance of this finding is that results from experiments or computational methods for 2-D plates may be pertinent to the long shallow bilge keels of ships but are not applicable to the 3-D shapes of a yacht keel or rudder. Therefore the results for the 2-D plate will not be considered further.

Considering then just the 3-D plates, the total roll moment coefficient was largely independent of frequency. Breaking down the total moment into its inertial and drag components revealed that each component showed a slightly different relationship with frequency and amplitude. The results for plate 4 are shown in Figure 7 and Figure 8. Results for the other plates are omitted for brevity. The entire data set was analysed and the following models were found to represent the 3-D plate behaviour:

$$C_M = 0.4(AR)^{-0.5}\left(\varphi^{0.25}\right) \tag{11}$$

$$C_D = 6AR^{-0.5} + 10w\varphi^{-1} \tag{12}$$

where:

w = dimensionless frequency

AR = aspect ratio

φ = roll angle amplitude (deg)

The goodness of fit for these models is shown for the case of 12.5° amplitude in Figure 9 and Figure 10. This work provides the first engineering estimate of hydrodynamic roll moments generated by typical appendage shapes.

5 CONCLUSIONS

- The difference in roll motion between the flat plate and NACA keel sections tested in regular waves was small compared with other keel geometry changes. It would appear that keel profile shape is the important factor.
- The proximity of the keel to the sea bed does not have a significant effect on the roll motion coefficient for all practical underkeel clearance ratios. However, the water particle velocities around the keel are affected by water depth so there can be differences in roll motion response between shallow water and deep water.
- The relationship between appendage roll moment, area, aspect ratio, span, frequency and amplitude of oscillation has been established to a first order approximation, a result not previously available. This permits an improved accuracy in the prediction of roll motion for yachts.

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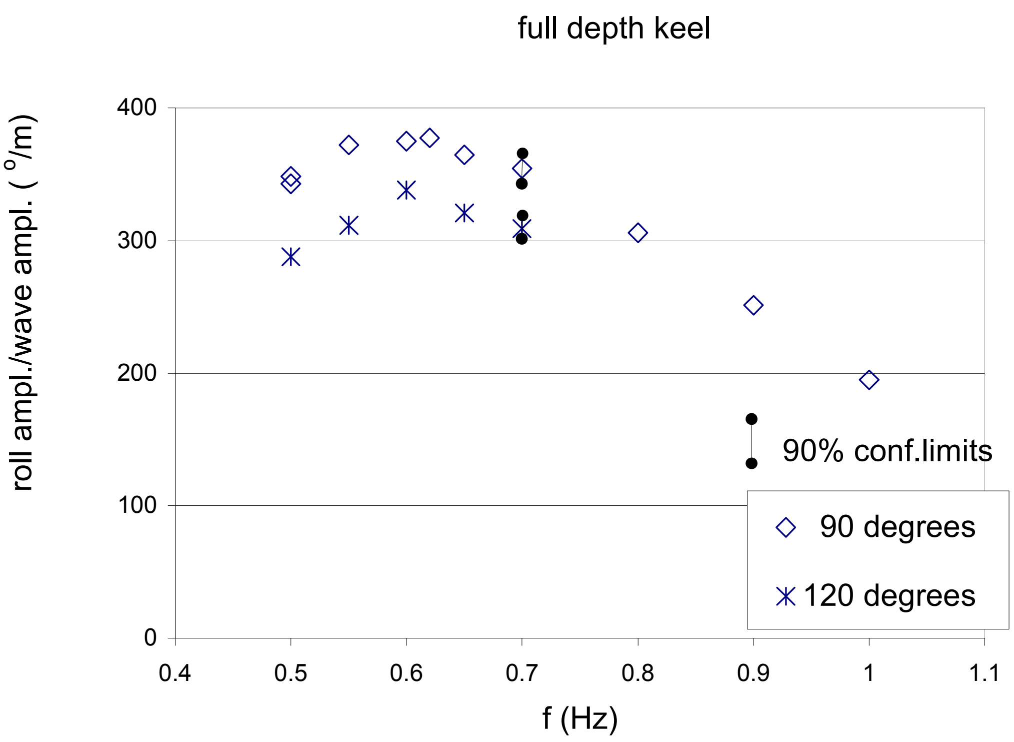

Figure 1 Effect of wave heading on roll amplitude - full depth keel

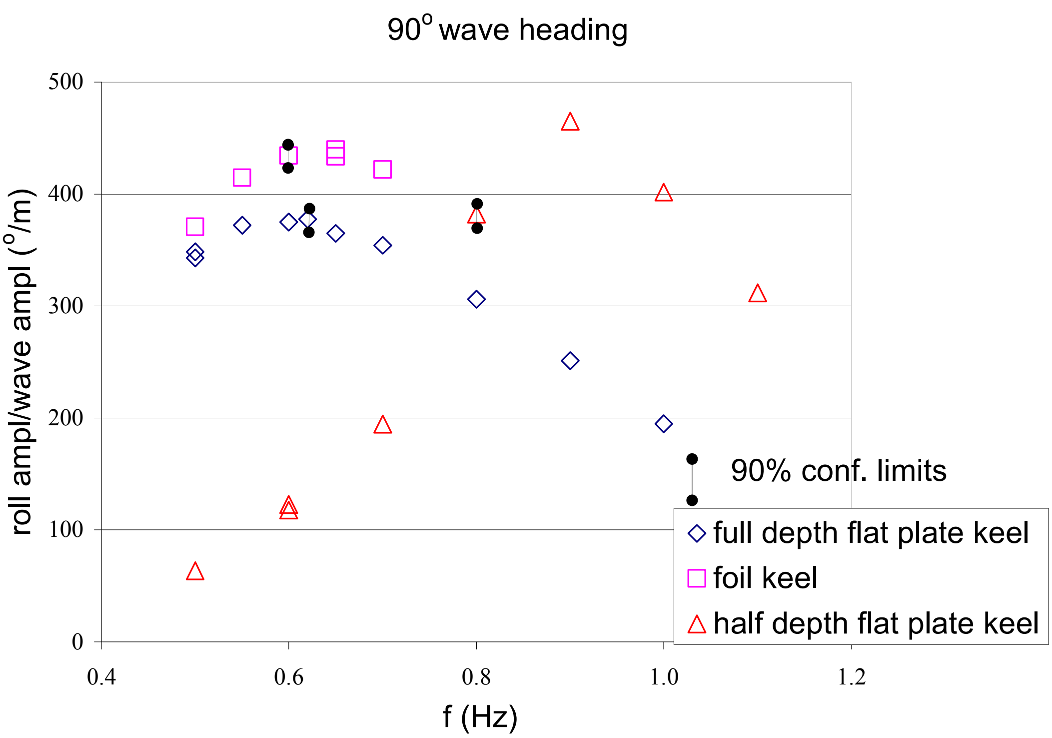

Figure 2 Effect of appendages on roll amplitude

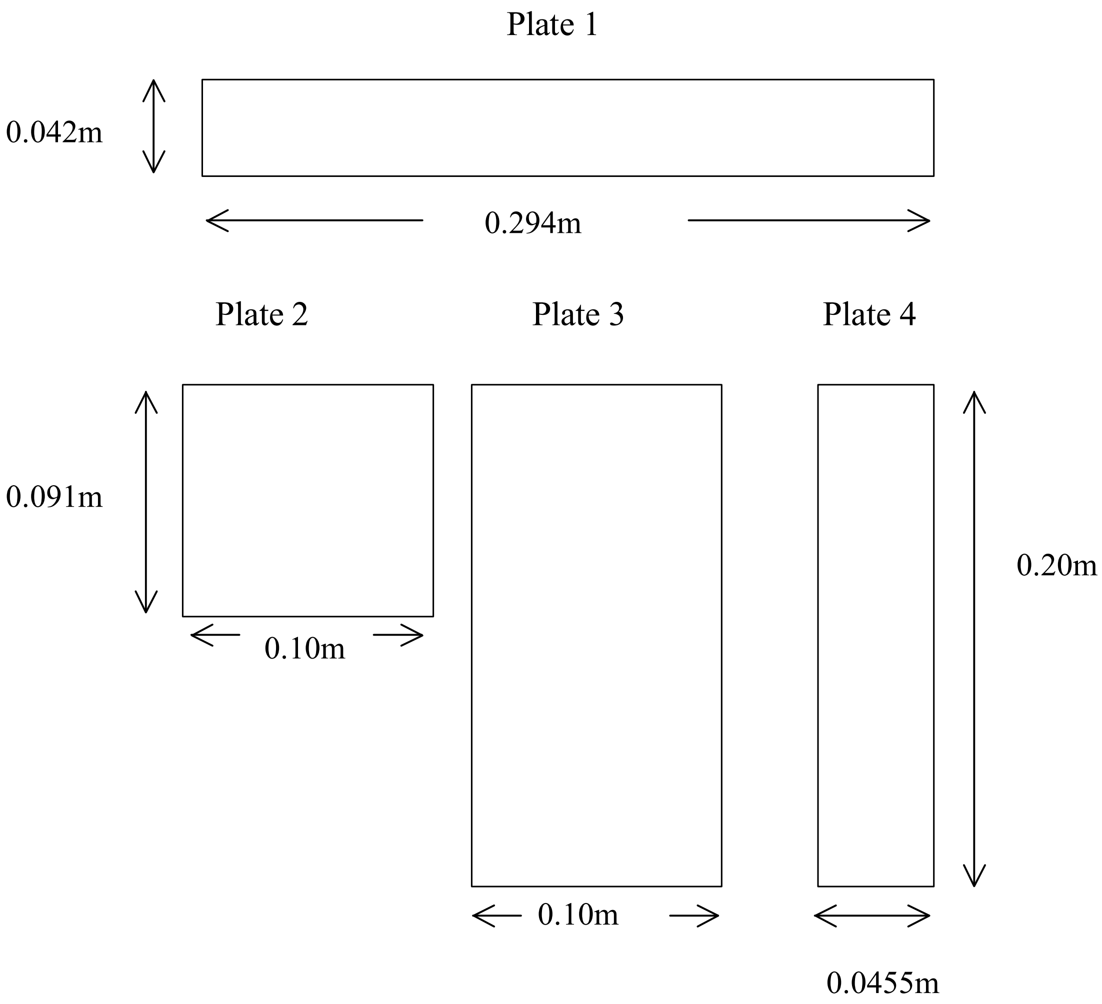

Figure 3 Plate geometry

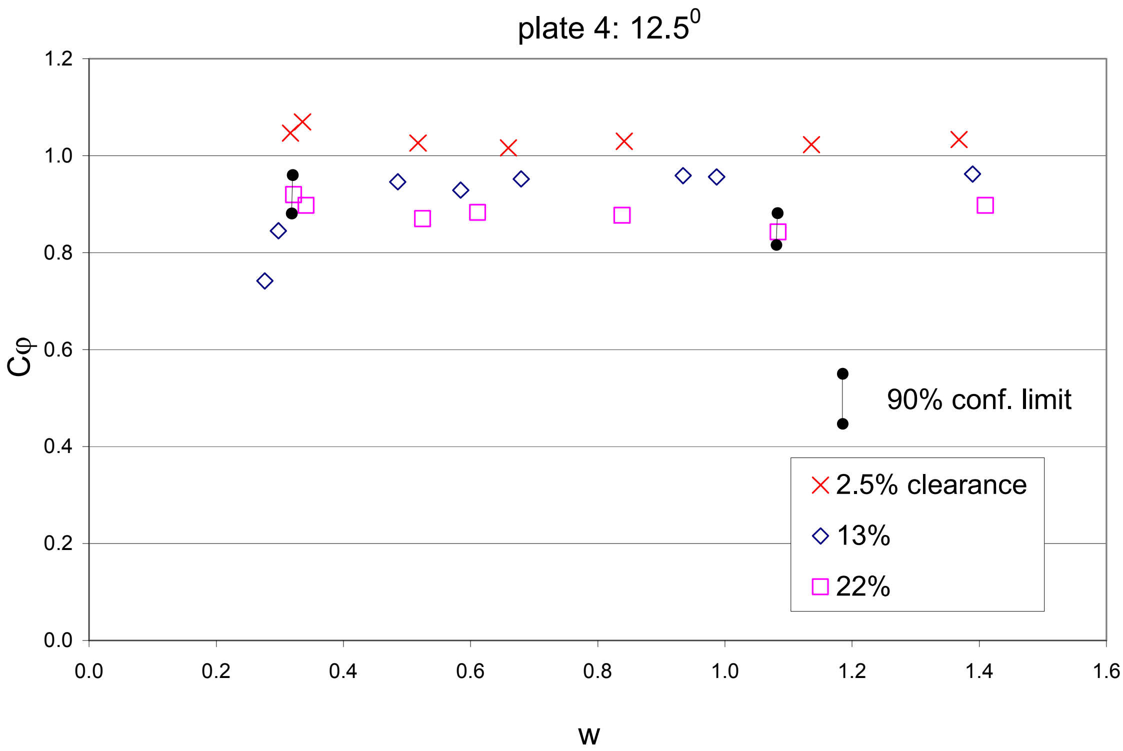

Figure 4 Effect of under-plate clearance on total roll moment

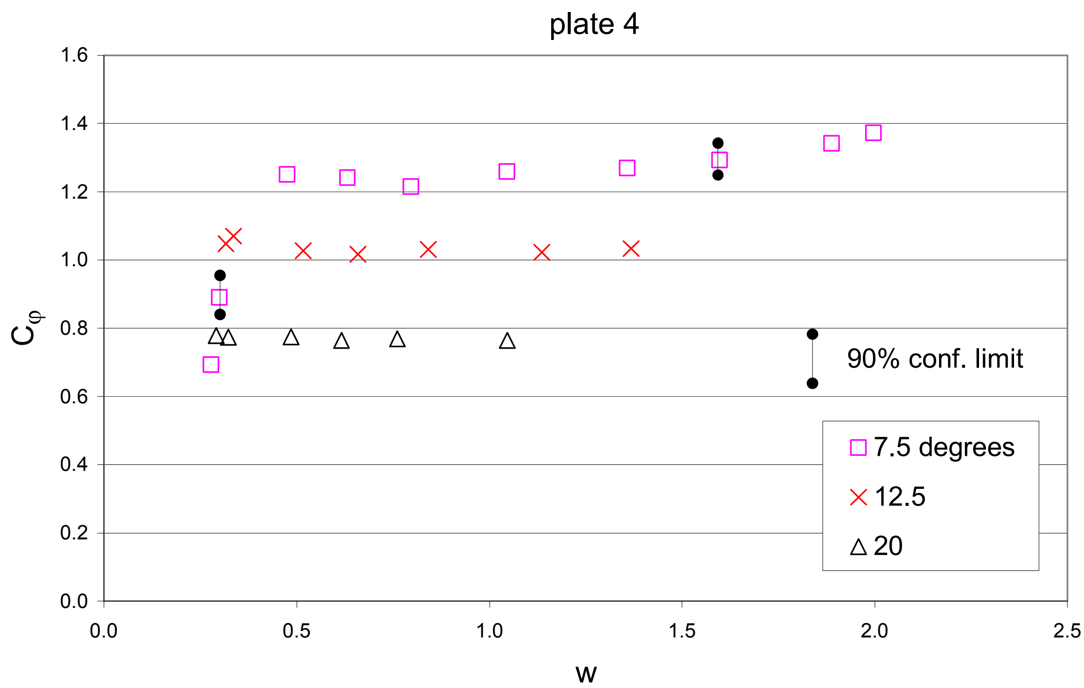

Figure 5 Total roll moment coefficient v. dimensionless frequency, plate 4

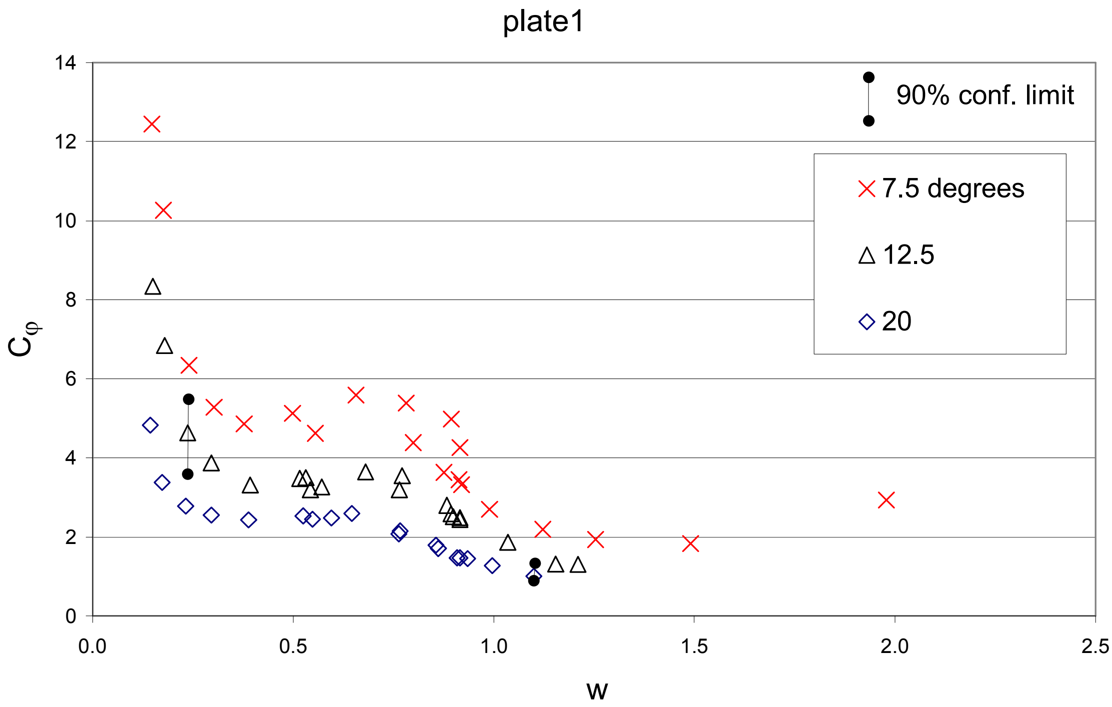

Figure 6 Total roll moment coefficient v. dimensionless frequency, plate 1

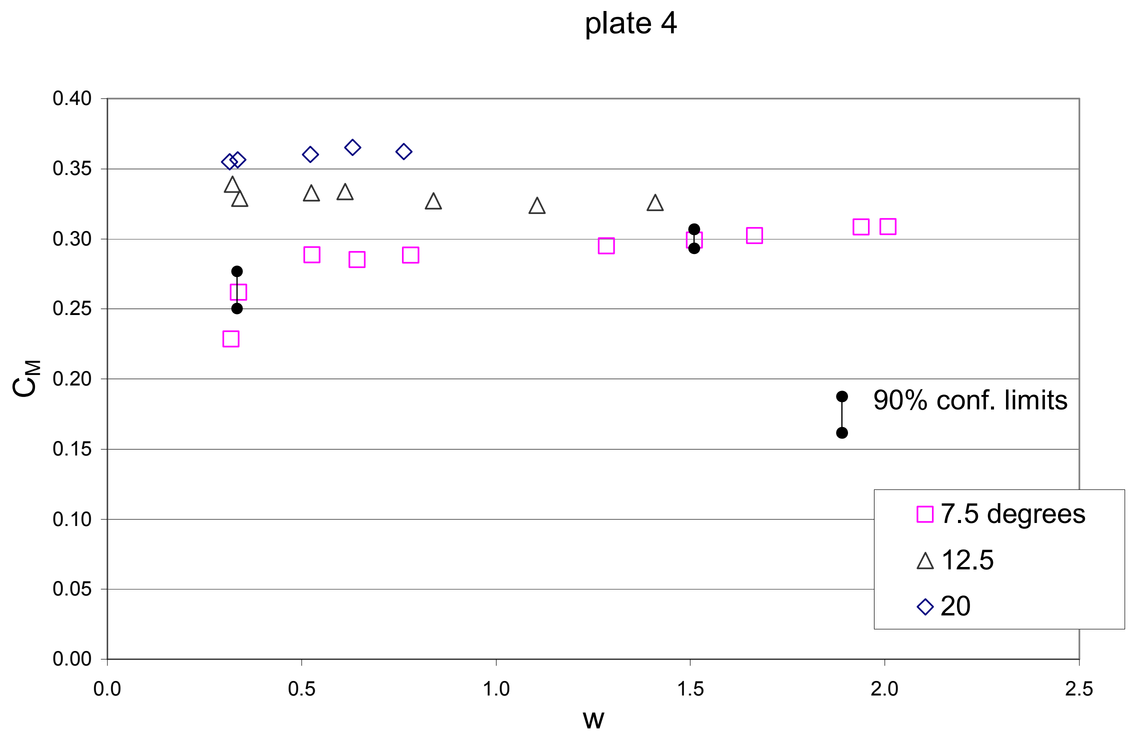

Figure 7 Roll inertia coefficient v. dimensionless frequency, plate 4

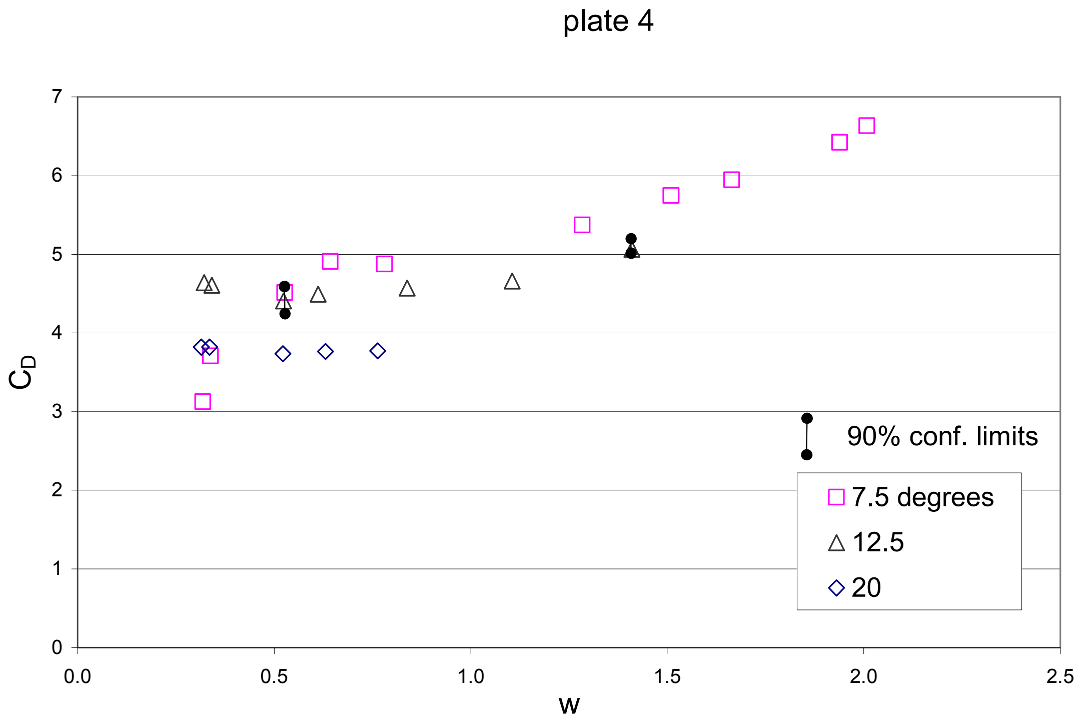

Figure 8 Roll drag coefficient v. dimensionless frequency, plate 4

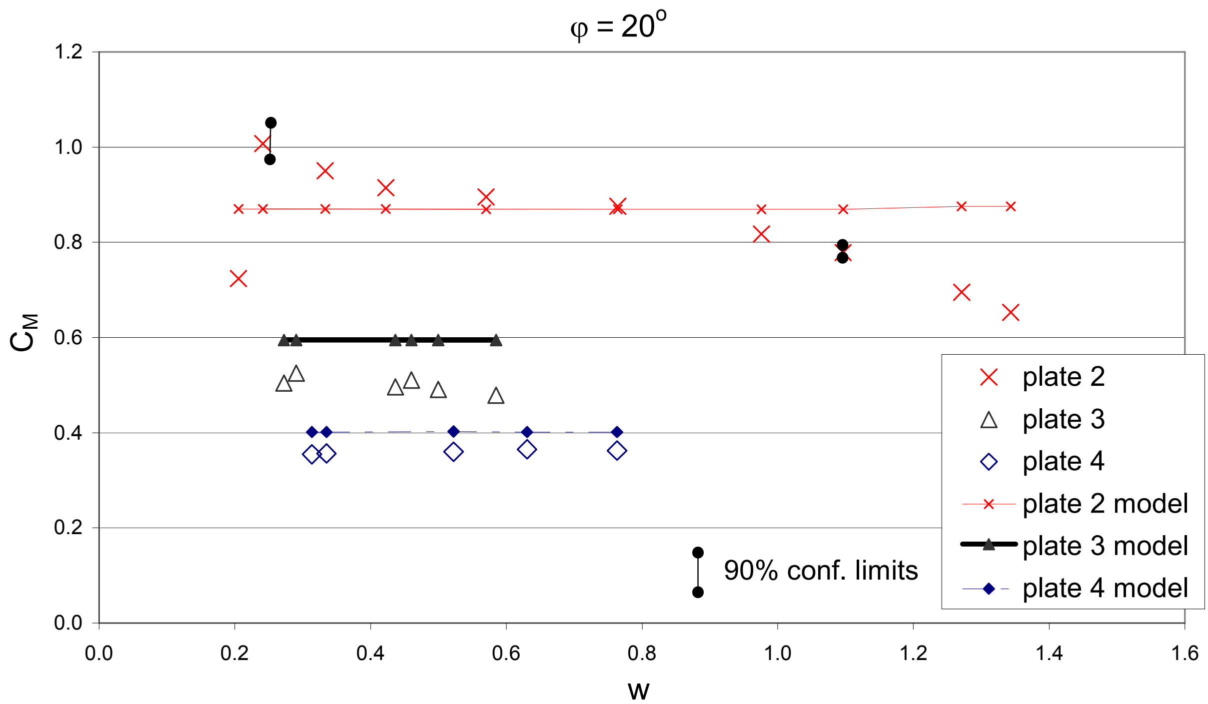

Figure 9 Effect of plate geometry on roll inertia coefficient, 20° amplitude

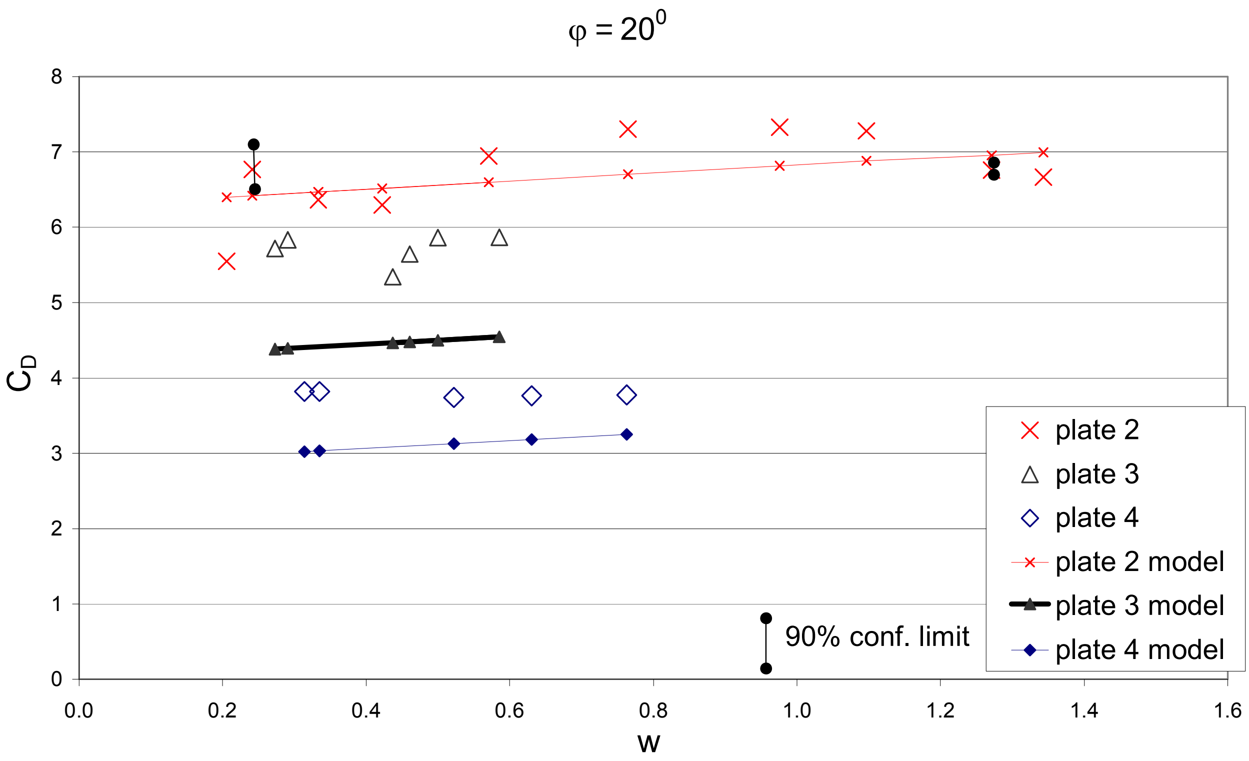

Figure 10 Effect of plate geometry on roll drag coefficient, 20° amplitude