An Investigation into Wave Slamming
Loads on Cylinders (OSFLAG 2A)
Final Report

Supported by the Department of Energy through the Offshore Energy Technology Board.

Report No. 317

March 1977
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Wolfson Marine Craft Unit Report No. 317
Technology Reports Centre No. OT-R-7743

March 1977

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Summary

A new test rig was designed and built on which cylinders of 5, 4, 3 and 2 in. dia. x 32 in. were forced at impact velocities of between 7 and 13 ft/s through a still water surface. A pneumatic/hydraulic system controlled the motion of the test cylinder via a 4 ft. pivot arm. Measurements were made of the transient slamming pressures around the circumference, at angles between 0 and 30° to the vertical, and along the cylinder length. Slamming forces were measured between the cylinder and arm; also some acceleration measurements were made. The data were successfully recorded over time scales of 5 - 50 ms. using Digital Transient Data Recorders with one thousand 8 bit words of store. The test rig and instrumentation have proved to be extremely reliable and to produce repeatable results with little scatter.

Over 350 data plots were analysed to determine the variation of Drag Coefficient with time as the cylinder immersion increased, both from an integration of measured pressure and from direct force measurements.

The pressure integration covers only the first phase of the slam, and over this period direct load measurements appear to be 30% lower than the integrated pressure force. There is no obvious single explanation of this discrepancy, but relevant factors discussed in the report include the non-uniformity of spanwise pressure distribution and the dynamic response of the elastic system to an essentially impulsive load.

The overall level of Drag Coefficient has a maximum value at the beginning of the slam lying between the values of $\pi$ (Von Karman) and $2\pi$ (Wellicome). The coefficient then decays to less than 1.0 at half immersion of the cylinder. Both the Von Karman and Wellicome estimates are based on an expanding plate model of the flow. Von Karman takes the equivalent plate breadth to be the cylinder breadth at the still water level intersection, whilst Wellicome takes the breadth at the spray root.

The force measurements exhibit a strong decaying oscillation about the falling mean line which actually represents the fluid loading. Analogue simulations which exhibit a similar response are discussed in the report. These were used to test the sensitivity of the curve fitting method used in deriving the estimated mean line through the experimental force traces. These analogue simulations were also used to interpret the relationship found to exist between cylinder mass/rig mass ratio and mean measured force levels.
Notation

\( a \) — wetted half beam
\( c \) — damping coefficient
\( C_D \) — Drag Coefficient obtained from total impact load and given by \( L = \frac{1}{2} \rho D U^2 C_D \)
\( C_p \) — peak pressure coefficient given by \( P_{\text{max}} = \frac{1}{2} \rho U^2 C_p \frac{D}{2d} \)
\( d \) — diameter of transducer diaphragm
\( D \) — cylinder diameter
\( E \) — kinetic energy
\( k \) — stiffness
\( L \) — total impact load
\( m \) — mass
\( p \) — pressure
\( q \) — vertical fluid velocity
\( R \) — cylinder radius
\( t \) — time
\( U \) — relative impact velocity
\( x \) — axis along cylinder beam in water plane
\( y \) — vertical axis
\( a, \beta \) — geometric constants
\( e \) — small angular change in \( \theta \)
\( \theta \) — half angle of radius to wetted beam
\( \rho \) — density
\( \phi \) — velocity potential
Introduction

The following report describes the investigation into Wave Slamming Loads on Cylinders (OSFLAG 2A) initiated by a Memorandum of Agreement numbered RD 1043/033 from the Department of Energy through the Offshore Energy Technology Board as part of an overall research programme into fluid loading on offshore structures.

Phase 1 of the investigation led to the commissioning of the test rig and in Phase 2 experiments were successfully performed using the rig. This final report describes these experiments and their analysis.
1. Description of Test Rig and Instrumentation

A new test rig was designed and built for the tests described in this report. The basic components of the rig (see Fig. 1) were as follows:—

i) A 6 ft. long x 5 ft. wide x 4 ft. deep tank with overflow troughs to static water level (S.W.L.) at each end.

ii) A test cylinder 32.14 in. long by 2 in., 3 in., 4 in. or 5 in. dia. with end plates protruding beyond the cylinder wall by a further 1½ in. The test cylinder was connected to the cross-member by two force transducers attached to the bottom of the inside surface of the cylinder (see Fig. 2), and located at the free-free beam nodes.

iii) A 6 in. x 3 in. x ¾ in. cross-member either hollow or lead filled weighing 29 lbs. and 163 lbs. respectively.

iv) A 6 in. x 3 in. x ¾ in. arm attached to the cross-member and pivoted at S.W.L. 4 ft. from the test cylinder centre line.

v) A 12 in. stroke 1½ in. bore double acting hydraulic cylinder attached to the arm 14 in. from the pivot. The arm was driven down by nitrogen gas acting on the top of the hydraulic cylinder, and its descent controlled by a hydraulic circuit operated from the bottom of the cylinder.

vi) The hydraulic circuit consisted of: a solenoid valve for start-stop control operated from one of the failsafe switches on the rig; a mechanical adjustable flow control valve to control the descent speed through the slam; and a hand pump, with pressure relief valve, to raise the arm. The system had a maximum working pressure of 6200 p.s.i. necessary to balance the rig weight and gas pressure with due regard to the piston area ratio.

vii) The nitrogen gas was supplied directly from a 2500 p.s.i. bottle via an on/off valve. In these tests a maximum pressure of 700 p.s.i. was used.

In operation the test rig performed entirely satisfactorily and has considerable scope for future tests with different configurations, such as oblique entry and rough water. or tests on other bodies. Also higher impact velocities might be achieved.

Details of the transducers and associated instrumentation are given in Appendix II. Two pairs of force transducers were used, one with half the stiffness of the other. 30th types were strain gauged and were wired up to yield a signal indicating the summed force from both transducers. The transducers were designed to be as stiff as thought possible; however, even with a gain of 1000 through the processing amplifier the signals had low noise content and gave good resolution, hence it is possible that even stiffer transducers could have been used.

Displacement was recorded from a simple carbon potentiometer driven from the pivot of the arm.

Pressures were recorded from commercial strain gauged transducers mounted through the cylinder wall. Two transducers had a rated pressure of 50 p.s.i. and two of 100 p.s.i.

Accelerations were recorded from two commercial piezoelectric accelerometers, one of which was mounted inside the test cylinder, beside a force transducer, and the other above, on the cross-member.

Following Phase 1 of this programme of work, in which the test rig was designed, built and commissioned, the instrumentation requirements were re-evaluated and three alternative systems identified which were listed in Proposals for Phase 2 (Reference 3). The storage oscilloscope, used in Phase 1 was considered inadequate, and experiments with magnetic tape, as proposed in the Phase 1 report (Reference 2) led to noise problems on the transient signals. So, with approval from Dr. Miller and Mr. Richardson of the National Maritime Institute, a system based around Digital Transient Data Stores was adopted. Each of four instruments provided one data channel in which a signal, of variable time span, could be captured and represented by a thousand 8 bit words of digital store. The signal was then presented on a simple oscilloscope for viewing purposes and could also be plotted
with any recorder at a slowed rate, thus avoiding frequency response and noise problems. Most of the results were plotted on an Ultra Violet (U.V.) recorder although those requiring frequency analysis were put on to magnetic tape. The Transient Data Recorders required accurate triggering in order to capture data from the slam over 5 to 50 ms., and this was achieved by using a water sensitive probe, mounted from the test cylinder. The stores were also linked in master-slave configuration, ensuring that the phases of the recorded signals could be matched.

The instrumentation performed entirely satisfactorily, yielding excellent data of force, pressure and acceleration time histories.
2. **Experimental Procedure**

Whilst commissioning the test rig several effects on the measurements were noted which resulted in the following procedures being adopted to ensure that the most reliable results were obtained.

a) **Warm Up**

Before any calibrations or measurements were taken, all electronic equipment was switched on for at least one hour, and the test rig arm was raised and lowered several times.

b) **Calibration**

All voltage tests were measured using the same digital voltmeter throughout the tests.

At the start of each set of runs, or daily, the calibration of all recording instrumentation was checked and bridge voltages were measured.

The gains of the special processing amplifiers were measured at the start and completion of the test programme. At the start of each set of runs the displacement transducer output was calibrated through all the instrumentation at approximately \( \frac{1}{2} \) intervals, measured by inclinometer to ± 1'.

The force transducers were calibrated at the start of each set of runs using a spring balance to load the test cylinder, and by measuring the output through the whole instrumentation chain.

The pressure transducers were calibrated upon completion of each series of tests and following removal from the test cylinder. A deadweight tester was used and their output measured directly.

The accelerometers were not calibrated since only the frequency from their records was analysed. A manufacturer's calibration was available. Calibration lines were fitted by the method of least squares.

c) **Alignment**

Each test cylinder was aligned to be parallel with the water surface along its length, both by measurement from an inclinometer on the test rig arm and visually by checking the path of the meniscus on contact with the water. The vertical alignment of the centre line of the cylinders relied on manufacturing accuracy and on the water level, which was topped up before each run.

d) **Preparation**

It was found that water drips on the pressure transducers severely affected their response. Fig. 12 shows that both the rise time and peak pressure are affected. To avoid errors from this cause the test cylinder was wiped dry before each run. Similarly, disturbed water affected pressure measurements, so tests were always made on calm water. Attempts were made to achieve similar speeds for slow, medium or fast runs by ensuring that hydraulic and gas pressures and flow valve settings were correctly adjusted. A rough check was made on the speed after each run.

c) **Pressure Transducer Zero Shift**

The pressure transducers were found to exhibit a shift in zero pressure output on immersion, and this was measured after each set of runs by performing a very slow drop.

The zero shift was both positive and negative depending on time and on the particular transducer, as shown in Fig. 11. The characteristics of a particular transducer were generally similar each time they were measured, although there were occasional irregularities. The phase of the shift relative to the penetration of the static water level is not accurately known, but it can be seen to be greatest after approximately 10 ms.

The zero shift did not amount to more than 1.5 p.s.i., which can be compared to a typical peak pressure of about 40 p.s.i. Corrections were not applied for this zero shift, since over the time scale considered these tended to cancel — although individually of significance — and were sufficiently irregular to doubt their application.
3. **Descriptions of Tests**

The majority of measurements were taken over three speed regimes – namely, low 7.0 – 7.4 ft/s, medium 9.6 – 10.2 ft/s and high 11.4 – 13.0 ft/s.

Initially the 5 in. dia. cylinder was tested and measurements made of force, acceleration and pressure at eleven positions around the circumference from 0° to 30° to the vertical. Because of limitations in the number of data channels and transducers, pressures had to be measured in sets of three and the transducers moved after each set of tests. Since one pressure from each set was used as a reference for the phase, results were obtained at the 0°, 3° and 6° positions from more than one set of runs, and in the case of 3° and 6° with different transducers. Following completion of these tests, and for reasons which will be discussed later, the rig mass was increased by substitution of the lead filled cross-member and this was used in all subsequent tests.

A ‘quick method’ of analysis was developed for estimating total impact load from pressures and is described in Section 4 e) ii). Using this method results from pressure measurements were needed at only four positions. It was agreed with representatives of the National Maritime Institute that on the remaining test cylinders pressures would only be measured at 0°, 6°, 12° and 18° on the centre line, but that in addition to the original programme, tests would be repeated on the 5 in. dia. cylinder with increased rig mass, and measurements made of the spanwise pressure distribution. This ensured that the most efficient use was made of the contract budget.

For each of the cylinder sizes the same pressure transducer was kept at the same angular location. The force measurements on the 2 in. and 3 in. dia. cylinders were obtained with the more flexible of the transducers, which thus yielded similar strains to those from the stiff transducers used with the 4 in. and 5 in. dia. cylinders.

A summary of the major tests performed is presented in Table 1, together with the associated transducers used for each measurement. Excluding commissioning, calibration and set up, eighty-nine successful runs were recorded. Pressures were generally recorded over a 5 or 10 ms. period, depending on the impact velocity. This time span was sufficient for analysis and gave a suitable sampling rate for the responses recorded. Forces were recorded over 20 or 50 ms. since the response frequencies were lower than from the pressure transducers, and a longer time span could be analysed.

The following limits of experimental accuracy were obtained from an assessment of calibrations recorded throughout the test programme.

Mean impact velocities were calculated from the displacement record to be ± 2%. Full scale calibrations from the stiff force transducers were all within 5% and from the flexible transducers within 3%. These variations could well be attributed to day-to-day fluctuations in temperature and should not influence the results. Linearity and resolution were within ± 1% of full scale readings.

Sensitivities of the pressure transducers remained within ± 1% of their nominal value over the period of the tests, except in rare cases when close to failure. Most measurements in these instances have been disregarded or were repeated. However, a spot check on the calibration through all the recording instrumentation revealed a 5% discrepancy from that obtained from the product of individual calibrations. This may be attributed to fluctuation in the gain of the special processing amplifier which could not be checked daily. Resolution from the pressure plots was within ± 1/2% of full scale. However, because of the fast decay from peak values and possible inaccuracies in the fit of the smoothed line, values used in the analysis could only be resolved to within ± 5%.
4. **Analysis of Test Results**

The expanding plate theory presented in Appendix I has been used as a basis for comparison with the experimental results and the following points are relevant to their interpretation.

The results have been plotted on a time scale of \( \frac{U t}{D} \) which for a given numerical value indicates similar wetted angular position for all cylinder sizes. Although the geometric assumptions of the theory are only valid to a wetted half angle of approximately 30° the time scale still represents geometric similarity beyond this point.

Theoretical predictions for peak pressures, pressure distributions and total impact loads refer solely to the situation existing at the fluid/cylinder interface, while the experimental measurements represent the response of the whole transducer, cylinder and test rig assembly to the fluid loadings. Under dynamic conditions this response may not be an accurate representation of the actual fluid forces and pressures.

The procedure for estimating fluid loadings from the force responses has been checked by applying the method to the predicted response of two analogue models, whilst the procedure for estimating fluid loadings from the pressure responses has been checked by applying the method to the pressure history predicted by the expanding plate theory. Details of these investigations are given in the following subsections of the report.

a) **Dynamic analogues of test rig behaviour**

The force transducer gave measurements of the force between the test cylinder and the rig arm. This force was not the same as the total impact load or hydrodynamic force exerted on the cylinder by the water, but represented the dynamic response of the test rig to the load, as shown in Fig. 23. Two dynamic analogues, as shown in Fig. 3, have been studied to enable estimations of the total impact force to be made from the transducer responses.

The first, based on a two degree of freedom spring, mass, damper system, tested the accuracy of a decaying forcing function derived from a curve fit of the vibration response and the second, based on two masses in space connected by a spring, suggested a method of correcting the forcing function for mass ratio effects.

i) **Two degree of freedom system**

The equations of motion were solved numerically, using an I.C.L. 1907 computer, which enabled the damped transient response to arbitrary forcing functions to be studied. The predictions for the two degree of freedom system were compared with results from the 5 in. d.a. cylinder with lower rig mass, since this configuration was used in the early part of the test programme.

Values of spring rates and masses were obtained as follows:— the force transducer stiffness and cylinder mass were calculated, after which the two natural frequencies of the system were obtained from force and accelerometer records by frequency analysis, as described fully later in section 4 d). Using these natural frequencies with the known stiffness and mass, the remaining stiffness and mass were calculated by substitution in the undamped steady state free vibration solution. The rig mass thus calculated was consistent with its value computed from scantlings. The constants used in the calculations are shown in Table 2.

Values of damping ratios were then varied for solutions with step and ramp forcing functions and the resulting decays were compared with measured force histories. The following analogue characteristics were required to reproduce, in a qualitative way, the characteristics of the measured force traces:—

- The transducer and rig damping should be small.
- The cylinder damping should be somewhat greater but still well below critical.

The method used in this report to separate the impact load from the oscillations associated with the dynamic response of the rig involved fitting a low order polynomial to the measured force data using a weighted least squares fit by the Forsyth Orthogonal Polynomial method. The validity of this procedure can be judged from Fig. 23 which shows that when applied to the analogue response curves the method accurately reproduces the original forcing function. The forcing function in this case was derived by a cubic fit to an actual experimental force history. The predicted response shows similarities with the original history.
One unresolved feature of the experimental response is the small amplitude of the vibration compared with that predicted, particularly over the first cycle. Increases of damping coefficients in the analogue do not improve the correlation, nor does there seem to be a lack of frequency response in the instrumentation, as the original force plot (Fig. 7) shows a higher order frequency causing a sharp turn down of the first peak. The response is characteristic of limited writing speed in the recorder, but again this was not the case.

There are theoretical grounds for believing that cylinder hydrodynamic damping varies in proportion to the cylinder drag coefficient which itself decays as the impact proceeds. This would provide a higher initial damping without causing an unacceptable decay rate later in the process, and could possibly improve the correlation between measured and analogue responses. The impact load calculated from pressure measurements, which is shown in Fig. 19, can be seen to rise to its maximum value within a value of $1 \times 10^{-3}$, which is fast compared with the frequency response of the force transducers. Attempts to put a significant rise time on the forcing function altered the phase of the two frequencies in the predicted response, such that correlation with measured response was lost.

ii) Two masses in space, connected by a spring

An assumption implicit in the first analogue was that the test rig moved at constant speed, excluding vibrational motion, just prior to and throughout the slam so that by Newton’s first law an equivalent static system, with the rig mass attached by a spring to the ground, could justifiably represent the rig. Results of experiments with the 5 in. cylinder showed an increase in measured force levels when the rig mass was increased, and this was not predicted by such an analogue.

It would be possible for a rig deceleration of short duration to remain undetected by the displacement measuring system. An analogue which would represent the effects of such a deceleration is the two mass-spring system discussed here. The equations of motion for this analogue were solved analytically for step and ramp functions with no damping, and the predicted response in the connecting spring was shown to be vibration at one frequency about a mean force level which was lower than the forcing function by $\frac{1}{1 + \frac{m_1}{m_2}}$.

Prior to the water contact point the hydraulic system is supporting the rig weight. After contact occurs there must be a step change in hydraulic pressure to counteract the impact load on the cylinder. It is reasonable to suppose that either due to the take up of mechanical backlash or the finite response time of the flow control valve, there will be a brief delay in the response of the servo system to the application of the load. The length of such a delay cannot be estimated from the known characteristics of the rig, but it can be shown that the deceleration of the two mass system in response to the impact loads would remain undetected by the displacement potentiometer for periods as long as 5 ms.

At the instant of contact the results of the force measurements can be made to agree reasonably closely by multiplying the mean force estimate by the factor $1 + \frac{m_1}{m_2}$. The mass ratio correction applied to the data assumes, arbitrarily, that the rig characteristics will reduce this correction linearly to zero over a range of $\frac{U_l}{D}$ values up to $35 \times 10^{-3}$.

b) Frequency analysis

Frequency analysis of the force and acceleration measurements from the 5 in. cylinder was performed using the Southampton University Data Analysis Centre PDP 11/45 computer. The tape recorded signals were digitised at a suitable sampling rate after filtering to avoid aliasing. Fast Fourier transforms (FFT) were performed on the data to yield modulus and phase, and in the case of accelerometer records double integration was applied to yield displacement frequency content. Because of the limited length of the transient records the Fourier analysis had limited samples. Attempts were made to improve this by a standard process of increasing the length of the data with a zero signal. Results of the analysis are shown in Fig. 13.
c) Curve fitting to force transducer response

A weighted least squares fit by orthogonal polynomials from the method of Forsyth was applied to the force histories. The computation was performed on an I.C.L. 1907 computer using a standard N.A.G. algorithm. The curve fits were applied to data from an integer number of cycles of vibration starting from the beginning of the rise in force. Various weightings were applied to the data points and the one thought to be most appropriate for the analysis was a linear increase from 1 for the first data point to 4 for the last, on the grounds that the mean line is least well defined where the oscillations are largest.

Experience with the fits showed them to be least accurate over the first and last cycles of vibration, and to be sensitive to the point in the oscillatory cycle at which the data was truncated. As a result some of the fits bent high or low over the last cycle. The computation also indicated the best fit, and for all the data this corresponded to either a quadratic or cubic polynomial. Over the first cycle of vibration these two fits gave significantly different curvature and hence different intersections at the start of the slam. In some cases with a cubic fit the curvature changed sign in this region.

d) Drag Coefficients derived from pressure data

Estimates of the overall drag coefficient of the cylinder were derived by numerically integrating pressures round the cylinder at the mid-span location using pressure data obtained by drawing a smooth mean line through the experimental pressure traces. Typical mean lines can be seen in Fig. 8.

Initially, in the case of the 5 in. dia. cylinder at the lower rig mass, drag coefficients were estimated directly using data from the eleven pressure positions at 3° intervals round the cylinder. Pressures at each position were factored by the projected area of each 3° interval and summed to give a total force force at discrete time intervals corresponding to $\delta \frac{U}{D} = 0.63 \times 10^{-3}$. The force estimates thus found are plotted in Fig. 19. These estimates oscillate about a decaying mean line with a peak value occurring each time the sprayroot crosses a transducer position. The magnitude of the oscillation is reduced as more transducers are immersed and as the pressure peak becomes wider. A similar integration of predicted pressure histories from the theory given in Appendix 1 also shows an oscillation, this time about a mean line given by the theoretical estimate of total force. It is concluded that the oscillations are not genuine but are a result of the peak pressures not accurately reflecting the true mean pressure level over an adjacent 3° interval. Included in Fig. 19 are force estimates based on unsmoothed pressure data and a least squares mean line through the results. There is no significant difference between results from smoothed and unsmoothed pressure data. We would expect the mean line to represent the best estimate of the actual force history over this time period.

Acquiring the pressure data at all eleven pressure positions was time consuming, since only four pressure transducers were available. In order to obtain force estimates from data which could be derived in one test the following 'quick analysis' procedure was devised:

The experimental pressure histories compared well with the theoretical prediction in a qualitative way but were generally of lower magnitude. An average pressure reduction factor between theory and experimental values was obtained for pressure transducers at angles $\theta = 0°, 6°, 12°, 18°$, and a force estimate obtained by multiplying the theoretical force prediction by this factor. Fig. 19 shows the force values obtained by this method and the mean line through these values lies close to that obtained from a full integration.

e) Data analysis

The Ultra Violet plots were marked with scale factors computed from calibrations and xeroxed for analysis.

i) Displacement time measurements

Velocities were computed from an average slope drawn by eye through the plot. The transducer output represented a rotation of approximately 7° (0.489 ft.) of the rig arm spaced roughly equally either side of static water level.
ii) Pressure measurements

Data from eleven circumferential positions obtained from low speed tests on the 5 in. dia. cylinder with lower rig mass were first analysed, both as recorded, and with hand smoothing through the oscillating decay.

Fig. 19 shows the variation of Drag Coefficient with immersion obtained by summing pressures by the methods discussed in section d). In applying the 'quick method' of estimating force on the cylinder four particular time intervals were chosen at \( \frac{U_l}{D} = 10, 16, 24 \) and \( 30 \times 10^{-3} \) to avoid analysing data from close to the pressure peaks, where dynamic magnification factors could lead to errors. Drag Coefficients obtained in this way are plotted in Fig. 20, and this 'quick method' was used for the calculation of Drag Coefficients from all other pressure measurements. No corrections were made to the calculated Drag Coefficients for variation in spanwise pressures or for apparent pressure measurement due to transducer immersion.

Peak pressures and phases are shown in Figs. 15 to 18 and were taken directly from the recorded plots.

iii) Force measurements

Data from force measurements were picked off by hand at discrete time intervals (generally 0.148 ms.) and a cubic curve was fitted to these data as described in section c). Available data beyond \( \frac{U_l}{D} = 200 \times 10^{-3} \) were curve fitted by eye, and are given in Table 3. Where necessary the cubic was corrected to match this fit and the resulting lines are presented in Fig. 24.

A mass ratio correction described in section b) ii) was then applied to the curve fits shown in Fig. 24. Examples showing the effect of this correction on data from each cylinder size are presented in Fig. 25, and corrected data from all force measurements were presented in Fig. 26. It should be noted that these data represent the total load incurred during the slam and include any buoyancy effects which may be present. Corrected force measurements are more consistent between cylinder sizes than uncorrected values at the commencement of the slam, and this provides the justification for making the correction.

iv) Acceleration measurements

Accelerations were recorded on magnetic tape from the Transient Data Stores and were analysed to obtain vibration frequencies as described in section d).
5. Discussion

a) Displacement-time measurements

Fig. 5 shows a typical displacement plot overlaid with its related calibration, suitably scaled, from which it can be seen that there was non-linearity in the transducer output, but no significant variation in mean speed over the full range of measurement. However, comparing the calibration with the displacement transducer output it is evident that a significant speed variation, of say 10%, could still have occurred within a time period of the order of 10 ms. The speed variations associated with the mass ratio corrections from the second analogue described in section 4 b) ii) fall well within this range, but the smooth response from force and accelerometers just prior to impact indicates that there was probably little variation at that time.

Clearly, information from the displacement transducer alone was inadequate for analysing the motion of the test cylinder during the short time interval of a slam. Furthermore, since the transducer was positioned at the pivot of the arm it did not measure directly the motion of the cylinder.

b) Pressure measurements

Figs. 8 to 10 show some typical pressure plots from which it can be seen that:

i) The zero pressure signal prior to impact was generally noise free although often the transducers responded to the impact before the spray root reached them. In the case of the transducers at 0° and 6° this was probably due to air cushioning, whilst at other positions it was more likely to be the response of the diaphragm to acceleration.

ii) The pressure rise was adequately described by a number of data samples, and the recorded peaks were generally within 90% of their probable values with the sampling rate used.

iii) The records of Fig. 10 show several different forms of pressure rise caused by high frequency vibration of the order of 20 to 50 kHz. This was most noticeable on the 0°, 3° and 6° records, and seemed to be a generally repeatable characteristic of particular pressure transducers.

iv) After the sharp rise the pressures decayed with a superimposed vibration of the order of 1 to 2 kHz. Whilst this vibration was not simple harmonic, for a particular run different transducers showed similar responses and by overlaying traces from the Transient Data System on an oscilloscope it was observed that not only were these responses in phase, but also they were of similar shape. Furthermore a separate run at the same condition produced a very similar set of responses.

Insute of the aforementioned characteristics of the pressure transducer response the methods of smoothing and analysing pressure histories yielded consistent results with remarkably little scatter. This can be seen from the Drag Coefficient data presented in Fig. 20 and the spanwise distribution of pressure shown in Fig. 22. In particular, there is close agreement between the C_{D} values for the 5 in. dia. cylinder at two different rig masses as well as a low scatter on the individual condition lines.

Spray root rise times and peak pressures

Figs. 15 and 16 show the theoretical prediction of the spray root rise to be, in general, very close to that measured. The few discrepancies in data from the 2 in. and 3 in. dia. cylinders at 6° and 12° could be due to the transducer not being flush with the cylinder surface through engineering tolerances. However the theoretical predictions of pressure histories were significantly different from the measurements.

Insofar as measured peak pressures were concerned, Figs. 17 and 18 show that at 9° and 12° they were close to theory, Fig. 22 demonstrates that the decay of the 12° pressures was faster than predicted, and this was generally true for all pressures. As a result of this the Drag Coefficient from the integrated pressures also decayed at a faster than predicted rate, as shown in Figs. 19 and 25. Measured peak pressures at positions greater than 12° were increasingly lower than predicted, whilst at 0° and 6° the results were unreliable with increased scatter due to transducer responses.
Spanwise pressure distributions

Pressure measurements along the cylinder length were made in an attempt to determine any non two-dimensional effects, due to 'aspect' or the presence of end plates, which could help to explain the differences between the force measurements and the integrated pressures. Unfortunately the results were inconclusive, as shown in Fig. 22 and Table 4.

The transducers were placed at 12° ± 0.1° since mid span measurements at this position had proved to be consistent. The pressure closest to the end (D) rose well before the cylinder reached static water level, probably due to a wave originating from the end plate, and the pressure was considerably lower than at the centre (A). However, the two middle pressures (B and C) rose slightly after A, for which no justifiable explanation has been found, and remained consistently higher than any previous pressure measurements at the centre. Without further measurements no corrections could be made to the Drag Coefficients calculated from the pressures for the spanwise distribution.

A point which might be significant is that the highest recorded pressures occurred at those transducer positions closest to the free-free beam nodal points chosen for the siting of the force transducers. This would be a relatively 'hard' spot on the cylinder wall.

Drag Coefficient estimates from pressures

Fig. 20 shows the Drag Coefficients for each cylinder size and each test speed derived from pressure histories by the quick method. As mentioned earlier, Fig. 19 shows that this method does accurately reproduce the results of a complete pressure integration. One further advantage of this method is that it makes no use of data close to the pressure peak at which a dynamic magnification could occur due to the transducer response to a rapid pressure rise.

The data shows a consistent trend with increasing cylinder size in that the larger cylinders produced a higher initial $C_D$, but this then decayed more rapidly than was the case with the smaller cylinders. The data is replotted in Fig. 21 to a base of Froude Number. This diagram shows no consistent trend and it must be concluded that this effect stems from some other source. Possible sources are effects due to viscosity, surface tension or aspect ratio of cylinder. An alternative source could be instrumentation error due to the increasing arc subtended by the transducer diaphragm at the centre of the cylinder (2.4° at 5 in. diameter, 5.7° at 2 in. diameter). This error would be most significant close to the spray root.

It is worth noting that no comparable scale effect was observed in the direct measurements of loading via the force transducers.

c) Force measurements

Figs. 6 and 7 show some typical force plots and the curve fits through them are shown in Fig. 24. It can be seen that:

i) The signals were generally noise free in spite of the large amplifier gain of 1000 used with the stiff transducers. Although not shown, the transducers responded at the start of a run as the cylinder began to drop. However, prior to impact this had damped out and the signals were steady at a level taken to represent zero force. In the analysis it has been assumed that this line is associated with constant velocity. This is supported by the low level of accelerometer output during this phase, and by an early test run recording force transducer output from the start to the point of entry.

ii) Different cylinder sizes exhibited different characteristic responses which were generally independent of speed, except in amplitude, and were repeatable from run to run. The response with the 5 in. dia. cylinder was generally predicted by the first analogue described in section 4 b) i) using the mass ratio of cylinder to rig (cross member and arm etc.) computed from the natural frequencies. The response with the 3 in. dia. cylinder was interesting since it exhibited beating.
iii) The force began to rise prior to the cylinder reaching static water level, although at a slower rate. This was possibly due to the presence of a wave initiated by the end plates, as noted in the discussion of spanwise pressures, or due to air cushioning.

iv) The Drag Coefficients obtained from the low speed tests were generally lower at immersion beyond \( \frac{U_t}{D} = 0.1 \) than those obtained from medium or high speed tests. In this respect it is worth noting that a lower flow rate was set on the hydraulic control valve for the slow speed tests, with the exception of those with a low rig mass, and no gas pressure was used to drive the arm. The rig characteristics might require a different mass ratio correction for these tests. Differences in \((speed)^2\) were considerably greater between the slow and medium runs than the medium and high ones.

The force plots represent responses to a total hydrodynamic load resulting from the slam. No attempt has been made to divide this load into components due to inertia and gravitation effects. The impact velocities of the tests were such that the total forces were considerably higher than maximum static buoyancy forces by between 14:1 and 100:1, based on maximum predicted Drag Coefficients. The effect of maximum static buoyancy on the Drag Coefficient is shown in Table 1. However, it is not known how these apply during the slam due to the rise of the spray root.

The corrected curves fitted through the force plots represent the best estimate of the forcing function or hydrodynamic impact load and are shown in Fig. 26.

As with the pressure measurements there was little scatter, the greatest being at \( \frac{U_t}{D} = 0.25 \) where the Drag Coefficient varied between 0.5 and 1.0. However, in the region where results from the force and pressure measurements overlap — ie. \( \frac{U_t}{D} = 0 \) to \( 30 \times 10^{-3} \) — the mean values from force are respectively approximately 30% to 25% lower than pressure integrated loads. Many explanations for this have been considered but none can be verified without further experimental data. Also, unlike results from the pressure measurements, no trend emerged for variation of Drag Coefficient with cylinder size.
6. Conclusions

Before drawing detailed conclusions from the foregoing investigation it is as well to restate the aim of the work. The objective was to obtain good estimates of the fluid loadings applied to a horizontal circular cylinder during a water impact, such as would occur with a wave. The total force history as a wave passes over such a cylinder would combine forces due to slamming, buoyancy, drag due to a separated wake, and fluid inertia forces for a fully submerged cylinder. In many instances however the slamming load is the largest single load. Being an impulsive load it inevitably initiates a structural vibration whose characteristics determine the fatigue life of the structure. The experiments reported here sought to isolate the slamming forces from the rest by operating at high Froude Numbers characteristic of the impact of severe storm waves on relatively small horizontal bracing members of oil rig structures.

Fluid loadings cannot be directly measured in a slamming lasting, typically, 20 ms. (on model scale). Essentially the measurements are of the elastic response of the system to the loading. This is certainly true of the direct measurement of total force, but even pressure measurements are not free of dynamic response effects.

In order to minimise dynamic effects the test rig structure was designed to be very stiff so as to raise all natural frequencies above a level of 0.4 kHz. The lowest frequencies were associated with the stiffness of the force transducers and experience has shown that stiffer transducers operating at lower strain levels would still have had adequate sensitivity. The rig appears to exhibit a delayed response to fluid loading, possibly of the order of 5 ms., which has the effect of reducing the mean measured force history. The mass ratio correction for the ratio of rig mass/cylinder mass is an attempt to mitigate this effect.

Subject to these two comments the performance of the rig was entirely satisfactory. In particular, the close control over cylinder alignment at the point of impact made it possible to reproduce the very short load rise time predicted by expanding plate theories. Measurements of pressure, spray root rise and force histories were consistent and repeatable. Experimental scatter was very small in comparison to other published data on slamming loads.

The following conclusions can be drawn from the measurements made with the rig, and from a comparison between these measurements and predictions from expanding plate theories:

i) At the commencement of the slam the rise to peak pressures and peak loads is virtually instantaneous. There is a slight rise in force prior to water contact occurring (possibly due to air cushioning) but the rise to peak hydrodynamic loads associated with water shock wave transmission is entirely masked by the response time of the pressure and force transducers (ie. it is less than 1 ms.).

ii) Directly measured total forces are consistently less than those obtained by integrating mid-span pressure measurements, being approximately 30% lower even after applying a mass ratio correction. There is as yet no satisfactory explanation of this result. The limited spanwise pressure measurements do not support so large a difference and it can only be suggested that it may be a cumulative result of several factors, such as a slightly oblique water entry, errors in curve fitting and spanwise effects at angles other than 12°.

iii) Spanwise pressure measurements show an increase of pressure near the force transducer locations and a sharp reduction in loading near the ends of the cylinder.

iv) Integrated pressure forces show no dependence on Froude Number, but do show a definite variation with size of cylinder. This could either be an effect of aspect ratio or of some other physical scaling problem. However, this result was not supported by a similar trend in measurements of total force.

v) Because of the singularity at the spray root location expanding plate theories are mathematically ambiguous. In particular the proper choice of equivalent plate breadth is not defined. Von Karman chose an equivalent breadth equal to the breadth of the cylinder at the still water intersection. Appendix I offers an alternative version with the equivalent breadth taken at the spray root level. Von Karman predicts a maximum impact force coefficient of \( C_D = \pi \) whilst the alternative predicts \( C_D = 2\pi \) (max.).
vi) At the beginning of the slam all experimental forces, both from direct measurement and integrated pressures lie between the force levels indicated by these two theories. In particular, predictions of pressure history, integrated pressure load and spray root position are in closer agreement with the theory of Appendix I (Wellicome) than with the Von Karman theory.

vii) The decay of loading as the slam progresses is more rapid than these simple theories suggest. We would not regard the theoretical models as having any validity beyond about \( \frac{U_t}{D} = 0.03 \). Fig. 27 gives our recommended standard loading curve for impact loading at high Froude Numbers. It has been drawn to represent the most plausible compromise between loads obtained from direct measurement and integrated pressure loads. However, it must be born in mind that there is a band of uncertainty of approximately ± 15% either side of this line associated with the residual scatter in the data and the discrepancy between the two sets of force values.

For design use we would recommend that the curve of Fig. 27 be used as a basis for predicting the vibration response of an actual structure to the impact load in order to assess the likely fatigue life of the structure. Appendix III contains guidelines for the interpretation of the curve for design.

7. Acknowledgement

This project was funded by the Department of Energy through the Offshore Energy Technology Board as part of an overall research programme into fluid loading on offshore structures.
8. Proposed Further Work

The following further work is recommended in order to extend the present range of data to cover all physical aspects of slamming on offshore structures, and ultimately to produce a comprehensive design code:—

a) An investigation of the discrepancies between the integrated pressures and forces during the initial period of the slam.

This investigation would be based on the existing test cylinders and would include:

i) Further measurements of the spanwise pressure distribution with and without cylinder end plates.

ii) Further force measurements with improved response.

iii) Further analysis using advanced analogues and improved curve fits from ii) above.

iv) Careful check on symmetry of cylinder entry into the water.

b) An investigation of slamming Drag Coefficients from cylinders at new conditions. This would probably involve only the measurement of forces, and the analysis of these will have been rationalised in item a). The new conditions should include:

i) oblique impact.

ii) cylinders roughened by representative marine fouling.

iii) An investigation of slamming Drag Coefficients from non-cylindrical shapes, such as:

i) pipe intersections.

ii) flats on cylinders.

iii) wedges or flat sided shapes.

d) An investigation of the dynamic response of a representative structural member to a slam. In these tests a suitably strain gauged model structure would be hung from the test rig in place of the normal simple test cylinder. The investigation would aim to check the measured response with that predicted by vibration theory using previously obtained slamming Drag Coefficients, and should include:

i) A check that the fundamental bending mode was the significant mode excited.

ii) The effect of a wave travelling along the structure from an oblique impact.

iii) The possible correlation with full scale trials of data such as from B.P. West Sole Tests.

e) In order to produce a design code based on information from tests conducted in items b), c) and d) above it would be necessary to obtain from existing data a spectra of impact speeds and associated wave directions for, say, the North Sea.

f) A design code could then be produced based on information from the foregoing investigations. As well as ultimate strength considerations the code should include cumulative fatigue damage analysis from representative vibration analysis checked under item d).
### Table 1 – Summary of Major Tests

<table>
<thead>
<tr>
<th>Test Radial pressures</th>
<th>Cylinder size in.</th>
<th>5, low rig mass nos. refer to transducer codes given in Appendix II.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>5</td>
</tr>
<tr>
<td>0°</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td></td>
<td>2</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>9</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>12</td>
<td></td>
<td>3</td>
</tr>
<tr>
<td>15</td>
<td></td>
<td>3</td>
</tr>
<tr>
<td>18</td>
<td></td>
<td>2</td>
</tr>
<tr>
<td>21</td>
<td></td>
<td>2</td>
</tr>
<tr>
<td>24</td>
<td></td>
<td>3</td>
</tr>
<tr>
<td>27</td>
<td></td>
<td>3</td>
</tr>
<tr>
<td>30</td>
<td></td>
<td>4</td>
</tr>
</tbody>
</table>

Spanwise pressure (at 12°) 4.7% from C.L.
38.3%
70.6%
92.4%
Force Acceleration, inside cylinder on bottom by force transducer
Acceleration, on cross member by force transducer

### Table 2 – Vibration Data From Frequency Analysis

<table>
<thead>
<tr>
<th>Source of signal</th>
<th>Natural frequencies obtained from Fourier analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force transducer</td>
<td>Hz</td>
</tr>
<tr>
<td>Accelerometer inside test cylinder</td>
<td>415</td>
</tr>
<tr>
<td>Accelerometer on cross-member</td>
<td>415</td>
</tr>
<tr>
<td>Result used in analogue</td>
<td></td>
</tr>
<tr>
<td>Calculated value of ( \omega_1 = \sqrt{\frac{k}{m_1}} ) (see Fig. 3)</td>
<td></td>
</tr>
<tr>
<td>Resulting computed mass ratio</td>
<td></td>
</tr>
<tr>
<td>( m_2 = 0.5503 )</td>
<td></td>
</tr>
<tr>
<td>Equivalent mass ratio from scantlings = 0.5821</td>
<td></td>
</tr>
</tbody>
</table>

18
### Table 3 – Data from force measurements at immersion beyond $\frac{U_t}{D} = 250 \times 10^{-3}$

<table>
<thead>
<tr>
<th>Cylinder diameter ins</th>
<th>Speed</th>
<th>Total drag coefficient at $10^{-3} \frac{U_t}{D} =$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>250</td>
</tr>
<tr>
<td>2</td>
<td>Low</td>
<td>0.76</td>
</tr>
<tr>
<td></td>
<td>Medium</td>
<td>0.85</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>1.07</td>
</tr>
<tr>
<td>3</td>
<td>Low</td>
<td>0.58</td>
</tr>
<tr>
<td></td>
<td>Medium</td>
<td>0.91</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>0.91</td>
</tr>
<tr>
<td>4</td>
<td>Low</td>
<td>0.53</td>
</tr>
<tr>
<td></td>
<td>Medium</td>
<td>0.89</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>0.80</td>
</tr>
<tr>
<td>5</td>
<td>Low</td>
<td>0.43</td>
</tr>
<tr>
<td></td>
<td>Medium</td>
<td>0.61</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>0.69</td>
</tr>
<tr>
<td>low rig mass</td>
<td>Low</td>
<td>0.61</td>
</tr>
<tr>
<td></td>
<td>Medium</td>
<td>0.61</td>
</tr>
<tr>
<td></td>
<td>High</td>
<td>0.59</td>
</tr>
</tbody>
</table>

### Table 4 – Phase of Pressure rise from spanwise tests

<table>
<thead>
<tr>
<th>Speed</th>
<th>Transducer Position</th>
<th>Rise Ref. to Centre (A) $10^{-3} \frac{U_t}{D}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>% ½ length to Centre line</td>
<td></td>
</tr>
<tr>
<td>Low</td>
<td>A  - 4.7</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>B  38.3</td>
<td>1.19</td>
</tr>
<tr>
<td></td>
<td>C  70.6</td>
<td>1.52</td>
</tr>
<tr>
<td></td>
<td>D  92.4</td>
<td>-14.58</td>
</tr>
<tr>
<td>Medium</td>
<td>A</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>B</td>
<td>1.71</td>
</tr>
<tr>
<td></td>
<td>C</td>
<td>2.62</td>
</tr>
<tr>
<td></td>
<td>D</td>
<td>-15.03</td>
</tr>
<tr>
<td>High</td>
<td>A</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>B</td>
<td>1.06</td>
</tr>
<tr>
<td></td>
<td>C</td>
<td>1.23</td>
</tr>
<tr>
<td></td>
<td>D</td>
<td>8.97</td>
</tr>
</tbody>
</table>

$-ve = before$

$+ve = after$
References


3. Wolfson Marine Craft Unit Document. Wave slamming loads on cylinders (OSFLAG 2A) Proposals for Phase 2, programme of work.
DRAWING OF TYPICAL ATTACHMENT OF TRANSUDERS TO TEST CYLINDERS
i) Two degree of freedom system

Equations of motion

\[ x_2 > x_1 \]
\[ m_1 \ddot{x}_1 + c_1 \dot{x}_1 - c_2 (\dot{x}_2 - \dot{x}_1) + k_1 x_1 - k_2 (x_2 - x_1) = 0 \]
\[ m_2 \ddot{x}_2 + c_3 \dot{x}_2 + c_2 (\dot{x}_2 - \dot{x}_1) + k_2 (x_2 - x_1) = F(t) \]

ii) Two masses connected by a spring

Equations of motion

\[ x_2 > x_1 \]
\[ m_1 \ddot{x}_1 - k_2 (x_2 - x_1) = 0 \]
\[ m_2 \ddot{x}_2 + k_2 (x_2 - x_1) = F(t) \]
FIG. 5

TYPICAL DISPLACEMENT PLOT OVERLAID WITH CALIBRATION POINTS

Displacement plot 151, U = 7.2 ft/s, 5 in. dia. Low rig mass

Start of slam

Av. line thro' plot fitted by eye
Scaled points from least sq. fit thro' calibration to fit above line
+ Scaled calibration points

0 1 2 3 4 5 6 7 8 9 10

7.04°
TYPICAL FORCE PLOTS FROM EACH CYLINDER.

Plot 189, force, 7.2 ft/s, 2 in dia.

Plot 223, force, 7.4 ft/s, 3 in dia.

Plot 306, force, 7.0 ft/s, 4 in dia.

Plot 176, force, 7.4 ft/s, 5 in dia.
FORCE PLOTS FROM TESTS WITH AND WITHOUT INCREASED RIG MASS

Plot 15, force, 7.1 ft/s, 5in. dia.
Low rig mass

Plot 174, force, 7.8 ft/s, 5in. dia.
TYPICAL PRESSURE PLOTS FROM FOUR POSITIONS AROUND 5in dia. CYLINDER

26.9 psi
Plot 284, $P_o$, 7.2ft/s, 5in dia.

42.5 psi
Plot 283, $P_o$, 7.2ft/s, 5in dia.

28.2 psi
Plot 294, $P_{12}$, 7.2ft/s, 5in dia.

45.6 psi
Plot 282, $P_{18}$, 7.2ft/s, 5in dia.
TYPICAL PRESSURE PLOTS AT 6° FROM EACH CYLINDER.

42.6 psi
Plot 198, $P_0$, 10.2 ft/s, 2 in dia.

42.8 psi
Plot 232, $P_0$, 10.4 ft/s, 3 in dia.

42.4 psi
Plot 315, $P_0$, 10.1 ft/s, 4 in dia.

106.2 psi
Plot 299, $P_0$, 10.0 ft/s, 5 in dia.
PRESSURE PLOTS AT $0^\circ$ SHOWING CHARACTERISTIC RISES.

61.9 psi
Plot 216, $P_0$, 10.1 ft/s, 4 in dia.

26.9 psi
Plot 296, $P_0$, 7.2 ft/s, 5 in dia.

71 psi
Plot 86, $P_0$, 9.9 ft/s, 5 in dia.

69.2 psi
Plot 90, $P_0$, 12.7 ft/s, 5 in dia.
TYPICAL PLOTS OF PRESSURE TRANSDUCER RESPONSE TO SLOW DROP.

Plot 332, $P_0^0$, $\frac{1}{2}$ ft/s, 4 in dia., code FL-90

Plot 331, $P_0^0$, $\frac{1}{2}$ ft/s, 4 in dia., code QL-55

Plot 330, $P_{12}^0$, $\frac{1}{2}$ ft/s, 4 in dia., code P25

Plot 329, $P_{18}^0$, $\frac{1}{2}$ ft/s, 4 in dia., code N44
TYPICAL PRESSURE PLOTS SHOWING EFFECT OF DRIPS ON TRANSUDER.

Plot 66, \( P_0 \), 7.2 ft/s, 5 in. dia.
Lower rig mass, with drips.

Plot 40, \( P_0 \), 7.1 ft/s 5 in. dia.
Lower rig mass, no drips.
TYPICAL PLOTS OF ACCELERATION AND FREQUENCY ANALYSIS.

Acceleration measured inside cylinder from run 18

Acceleration measured on arm from run 19

Fourier analysis of displacement from acceleration (run 18)

Fourier analysis of displacement from acceleration (run 19)
The image contains a graph with a line labeled "Wellicome prediction." The graph has a vertical axis labeled $\theta^\circ$ and a horizontal axis labeled $10^{-3} \frac{U_t}{D}$. The key explains the symbols:

- $+$: Start of rise from $P_o$ - Plotted at position of edge of transducer
- $\triangle$: First significant point on rise measured from data labs - Plotted as above
- $\bullet$: Peak from $P_o$ - Plotted at spray root position for maximum pressure on diaphragm

The graph appears to be part of a study on phase of pressure rise and peak from slow speed tests on 3 in. dia. cylinder.
PEAK PRESSURE COEFFICIENTS $C_p'$ FROM ALL CYLINDERS

**Key**
- + Low speed
- △ Medium speed
- • High speed

**Transducer codes**
- FL - 90
- QL - 55
- P25
- N44

**Fig. 17**

2 in dia.

3 in dia.

Wellcome prediction

4 in dia.

5 in dia.
PEAK PRESSURE COEFFICIENTS FROM TESTS ON 5in DIA. CYLINDER

Key
+ Low speed
△ Medium speed
● High speed

Transducer codes

θ°
DATA FROM INTEGRATION OF PRESSURE MEASUREMENTS FROM TESTS ON 5in. DIA. CYLINDER AND LOWER RIG MASS.

\[ C_D = \frac{2\pi}{\left( 1 + \frac{3U}{D} \right)^{1/2}} \]

- Least squares fit to \( x \)
- Least squares fit to \( A \)
- Individual pressure records smooth and unsmoothed
- Average of selected pressures compared with theory
DRAG COEFFICIENTS OBTAINED BY 'QUICK METHOD' OF ANALYSIS
FROM PRESSURE MEASUREMENTS AT 0°, 6°, 12° and 18°.

FIG. 20

- **2in dia.**
- **3in dia.**
- **4in dia.**
- **5in dia.**

**Key**
- + — Low speed
- Δ — Medium speed
- * — High speed
- — — Average all speeds

All lines fitted by least squares.
VARIATION OF DRAG COEFFICIENT FROM FIG 20 vs FROUDE No.
DATA FROM PRESSURE MEASUREMENTS ALONG SPAN AT 12° FROM TESTS ON 5 in DIA. CYLINDER.
ANALOGUE OF FORCE TRANSDUCER RESPONSE

Curve fit through response from low speed tests on 5 in diam. cylinder

- Cubic curve fit
- --- Mod. for longer time

Analogued response to cubic forcing function from above graph

- Cubic forcing
- --- Cubic curve fit
DATA FROM CURVE FIT THROUGH FORCE MEASUREMENTS

Key:
- L: Low speed
- M: Medium speed
- H: High speed
- •: End first cycle vibration

3 in dia.

4
3
2
1
0
CD

20
40
60
80
100
120
140
160
180
200
220
240

10^-3 Ut
D

5 in dia.

4
3
2
1
0
CD

20
40
60
80
100
120
140
160
180
200
220
240

10^-3 Ut
D

Data from lower rig mass

2 in dia.

4
3
2
1
0
CD

20
40
60
80
100
120
140
160
180
200
220
240

10^-3 Ut
D

4 in dia.

4
3
2
1
0
CD

20
40
60
80
100
120
140
160
180
200
220
240

10^-3 Ut
D

FIG. 24
EXAMPLES OF MASS RATIO CORRECTION TO FORCE DATA

**FIG. 25**

- **3 in dia.**
- **5 in dia.**
- **2 in dia.**
- **4 in dia.**

Key:
- Medium speed curve
- Curve corrected for mass ratio effect

Graphs showing relationship between $C_D$ and $10^{-3} U/D$ for different diameters.
DRAG COEFFICIENT FROM PRESSURE DATA (FIG 20) AND CORRECTED FORCE DATA FROM FIG 24
BEST ESTIMATE OF DRAG COEFFICIENT.
APPENDIX I
Expanding Plate approximation for water entry

ACTUAL FLOW

The simplest approximate model for the flow generated by a body entering the water is to replace the actual body by a moving flat plate and to allow the width of the plate to increase with time as the wetted beam of the body increases. From the simple formulae given below the vertical velocity of the free surface can be obtained and integrated to obtain the elevation of the free surface above the still water level. As the size of the plate is increased the kinetic energy of the fluid set in motion by the plate will increase and thus, since this energy comes from work done by the plate, the total force on the plate can be obtained from the rate of change of this kinetic energy.

Finally, we can obtain the pressure distribution on the expanding plate and use this as an approximation to the pressure distribution on the wetted part of the body itself. To apply the plate results to the body it is necessary to assume that at any given time the free surface velocity depends only on the wetted beam at that time and is not too much affected by the body shape – a not unreasonable assumption at a distance from the body itself. It is also necessary to assume that the kinetic energy of the flow depends only on wetted beam and not shape – and indeed for a fully immersed elliptic body the energy is only dependent on beam and not on body depth so that again this is not an unreasonable first approximation. The use of this model to predict actual pressure distributions is probably less justified than its use to predict the position of the sprayroot and to predict the total integrated force.

The basic results for the flat plate in an unbounded fluid are

\[ q = U \left( \frac{x}{\sqrt{x^2 - a^2}} - \frac{a}{2} \right) \]  \hspace{1em} (1) \text{ where } q = \text{vertical velocity of fluid on } y = 0

\[ E = \frac{\pi}{2} \rho a^2 U^2 \]  \hspace{1em} (2) \text{ E = Kinetic energy of fluid below } y = 0

\[ \phi = -U \sqrt{a^2 - x^2} \]  \hspace{1em} (3) \text{ } \phi = \text{Velocity potential along the plate (x < a)}

From these equations we obtain the total impact load \( L \) as

\[ L = \frac{dE}{dt} = \frac{\pi}{2} \rho U^2 \cdot 2a \cdot \frac{da}{dt} \]

or \[ L = \pi \rho a U \cdot \frac{da}{dt} \]  \hspace{1em} (4)

And the impact pressure on the plate \( p \) is given by
\[ p = -\rho \frac{\partial \phi}{\partial t} = -\rho \frac{\partial \phi}{\partial a} \cdot \frac{da}{dt} + \frac{\rho U a}{\sqrt{a^2 - x^2}} \cdot \frac{da}{dt} \quad (5) \]

In order to apply equations (4) and (5) we need to determine the wetted half beam 'a' as a function of time from the geometry of the body as it enters the water. For the specific case of a circular cross section we have

\[ a = R \sin \theta \]
\[ y = Ut + \eta(a) = R \left\{ 1 - \cos \theta \right\} \]

\[ \therefore \quad 1 - \frac{y}{R} = \cos \theta = \sqrt{1 - \left(\frac{a}{R}\right)^2} = 1 - \frac{1}{2} \left(\frac{a}{R}\right)^2 - \frac{1}{8} \left(\frac{a}{R}\right)^4 \quad \ldots \ldots \quad \text{on using a Binomial expansion} \]

hence \[ Ut + \eta(a) = \frac{1}{2R} a^2 + \frac{1}{8R^3} a^4 \text{ for small } a. \quad (6) \]

It is, in fact convenient to relate wetted ½ beam to time as

\[ Ut = \alpha a^2 + \beta a^4 \quad (7) \]

whence \[ \frac{dt}{da} = 2\alpha a + 4\beta a^3 \]

Then from (1) the free surface velocity is given by

\[ \frac{\partial \eta}{\partial t} = \frac{\partial \eta}{\partial a} \cdot \frac{da}{dt} = U f \left[ \frac{x}{\sqrt{x^2 - a^2}} - 1 \right] \]

\[ \therefore \quad \frac{\partial \eta}{\partial a} = \frac{U dt}{da} \left[ \frac{x}{\sqrt{x^2 - a^2}} - 1 \right] = \left\{ 2\alpha a + 4\beta a^3 \right\} \left[ \frac{x}{\sqrt{x^2 - a^2}} - 1 \right] \quad (8) \]

In order to determine the free surface elevation at any given \( x \), equation (8) must be integrated from \( a = 0 \) up to the current ½ beam. In particular for the free surface elevation at the spray root we have

\[ \eta(a) = \int_0^a \left\{ 2\alpha b + 4\beta b^3 \right\} \left[ \frac{b}{\sqrt{a^2 - b^2}} - 1 \right] \, db \]

on making an obvious substitution of variables.

Now integrals of the Form \[ I_n = \int_0^a \frac{b^n}{\sqrt{a^2 - b^2}} \, db \]

Satisfy the recurrence formulae \[ I_0 = \frac{\pi}{2}; \quad I_1 = a; \quad I_n = \frac{n-1}{n} \cdot a^2 \cdot I_{n-2} \]

Thus \[ I_3 = \frac{3}{2} a^2 I_1 = \frac{3}{2} a^3 \text{ and hence} \]

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\[ \eta(a) = 2a^2 + \frac{3}{2} \beta a^4 - a^2 - \beta a^4 \]

\[ = 2a^2 + \frac{3}{2} \beta a^4 - Ut \]

\[ \therefore \quad Ut + \eta(a) = 2a^2 + \frac{3}{2} \beta a^4 \quad (9) \]

Equation (9) relates the immersed draught to the current half beam from the free surface form and this can be compared to equation (6) for the geometry of the cylinder to give

\[ \alpha = \frac{1}{4R} = \frac{1}{2D} \quad \text{and} \quad \beta = \frac{3}{64R^3} = \frac{3}{8D^3} \]

Thus equation (1) becomes

\[ \frac{8Ut}{D} = \left( \frac{a}{R} \right)^2 + \frac{1}{16} \left( \frac{a}{R} \right)^4 \]

Thus the time for the spray root to reach an angular position \( \theta = \theta_o \) from BDC on the cylinder is given by

\[ t = \frac{D}{8U} \left( \frac{\sin^2 \theta_o}{1 + \frac{1}{16} \sin^2 \theta_o} \right)^{\frac{3}{2}} \quad (10) \]

The local pressure at an angular position \( \theta \) at the time when the spray root has reached \( \theta_o \) can be obtained from (5) as

\[ P = \frac{\rho U \sin \theta_o}{\sin^2 \theta_o - \sin^2 \theta} \cdot \frac{U}{\frac{1}{2} \sin \theta_o + \frac{1}{16} \sin^3 \theta_o} \]

hence

\[ P = \frac{2\rho U^2}{(1 + \frac{1}{16} \sin^2 \theta_o) \sqrt{\sin^2 \theta_o - \sin^2 \theta}} \quad (11) \]

Equations (10) and (11) were used in defining the pressure history curves (p vs t) in figures 4 - 12 of this report. It can be seen that as \( \theta \to \theta_o \), \( \theta \to \infty \) so that the largest pressures occur as the spray root covers the transducer diaphragm. As the spray root crosses the transducer diaphragm we can write \( \theta = \theta_o - \epsilon \) and consider the case where \( \epsilon \) is small to obtain

\[ \sin^2 (\theta_o - \epsilon) = -2 \sin \theta_o \cos \theta_o \epsilon + \sin^2 \theta_o = \sin^2 \theta_o - \sin 2\theta_o \epsilon \]

so that near the spray root

\[ P \to \frac{2\rho U^2 \epsilon^{-\frac{1}{2}}}{(1 + \frac{1}{16} \sin^2 \theta_o) \sqrt{\sin 2\theta_o}} \]

Assuming this variation of pressure we can find the mean pressure on the diaphragm as the spray root crosses the diaphragm and hence determine, graphically at any rate, the maximum mean pressure on the diaphragm. The angle subtended by the transducer at the centre of the cylinder is \( \epsilon_o = \frac{d}{D} \) where \( d = \) transducer diameter and hence
\[ p_{\text{max}} = \frac{4\rho U^2}{(1 + \frac{1}{8} \sin^2 \theta_0) \sqrt{\sin 2\theta_0}} \cdot \frac{D}{d} \quad (12) \]

where the constant 4 has been obtained by numerical integration of pressures on the diaphragm. The predicted peak pressure from equation (12) has been compared with the measured peak values from the experiments.

Finally the total impact load on the cylinder can be obtained from equation (4) as

\[ L = \pi \rho U_a \frac{da}{dt} = \frac{\pi \rho U^2 a}{2\alpha_a + 4\beta a} = \frac{\pi \rho U^2 a}{\frac{a}{2R} + \frac{3}{16} \left( \frac{a}{R} \right)^3} \]

thus

\[ L = \frac{\pi \rho U^2 D}{1 + \frac{3}{16} \left( \frac{a}{R} \right)^3} \]

To the lowest order of approximation from equation (7)

\[ U_t = \alpha a^2 = \frac{a^2}{4R} = \frac{D}{8} \cdot \left( \frac{a}{R} \right)^2 \]

\[ \frac{3}{8} \left( \frac{a}{R} \right)^2 \approx \frac{U_t}{D} \]

hence the mean drag coefficient for the cylinder as a function of time is

\[ C_D = \frac{L}{\frac{1}{2} \rho DU^2} = \frac{2\pi}{1 + \frac{3}{16} \frac{U_t}{D}} \quad (13) \]

This total impact drag coefficient is consistent with the pressure distribution equation (11), and indeed could have been obtained from it by integration. At time \( t = 0 \) the instantaneous value of \( C_D \) from equation (13) is \( C_D = 2\pi \).
APPENDIX II

Instrumentation Specification

Pressure Transducers

Kulite Sensors

<table>
<thead>
<tr>
<th>Type</th>
<th>Code</th>
<th>Rated</th>
<th>Nat. freq.</th>
<th>Sensitivity</th>
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<tr>
<td>XML-1-190</td>
<td>1 N44</td>
<td>50</td>
<td>60</td>
<td>0.184</td>
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<tr>
<td>XML-1-190</td>
<td>2 P25</td>
<td>50</td>
<td>60</td>
<td>0.138</td>
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<td>XTMS-1-190</td>
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<td>80</td>
<td>0.120</td>
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<td>4 QL-55</td>
<td>100</td>
<td>80</td>
<td>0.076</td>
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</table>

force Transducers

2 off 2 leg aluminium compression blocks per set up, each block strain gauged as 2.6 arm bridge

<table>
<thead>
<tr>
<th>Code</th>
<th>Strain Gauge type</th>
<th>Total Stiffness</th>
<th>Sensitivity</th>
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</thead>
<tbody>
<tr>
<td>A</td>
<td>350Ω foil resistance</td>
<td>1.5 x 10⁶</td>
<td>0.346</td>
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<tr>
<td>B</td>
<td>Kulite UFP-50-190</td>
<td></td>
<td>71.8</td>
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350Ω piezo resistive

Accelerometers

Bruel + Kjaer

<table>
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<tr>
<th>Frequency Response</th>
<th>2 – 10,000 Hz</th>
</tr>
</thead>
</table>

Displacement Transducer

R.S. Components carbon track potentiometer

Special Signal Processing Amplifiers.

Gain settings 1, 10, 20, 50, 100, 1000

Input Selection High impedance: Single, Differential (8 channels)

Frequency response at gain 1 – 100 D.C. – 10 kHz – 3dB

gain 1000, D.C. – 3 kHz – 3dB

Input range ± 10V f.s. max., Output range ± 10 V f.s. max.

Transient Data Stores

Data Laboratories Ltd. datalab DL 905

4 Stores in master-slave configuration

5 MHz 8 bit analogue to digital converter

Input amplifier frequency response D.C. – 3MHz

input range 10mV – 50V f.s.

1024 point MOS digital memory

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Recording cycle 0.2ms – 10s f.s.
    modes Delayed sweep, switched sweep pre-trigger record.
Output 1V/f.s. in 20s.
C.R.O. Z modulation for individual data point identification.
Trigger Auto continuous or single — internator external

**Oscilloscopes**
Telequipment D614 (2 channel)
Vertical amplifier frequency response DC – 10MHz – 3dB
Sweep rates (19 rates)
    X1 500 ms – 0.5μs/div ± 5%
    X5 100 ms – 200 ks ± 7%
Trigger internal or external

**U.V. Recorder pre-amplifier**
S.E. labs SE 993 (6 channels)
Frequency response D.C. – 10kHz ± 1dB
Input sensitivity 0.02 – 20 cm/V
Output range ± 10 cm dependant on galvanometer
Shift range ± 10 cm

**U.V. Recorder**
Southern Instruments M 1300 (10 channels)
Paper speeds 0.15 – 100 i.p.s. ± 5%
Galvanometers miniature tubular series SMI/L, nat. freq. 1.6 kHz
Writing speed 30 000 i.p.s.
Maximum deflection 6 ins.

**Tape Recorder**
Teac Corp. R-70A DR/FM Cassette Data Recorder
4 channel / 4 track, tape speed 1½ i.p.s. ± 1%
Frequency response FM, D.C. – 625 Hz + 1–2 dB
Output level ± 1 V
APPENDIX III
Design Implications

The response of the force transducers (Fig. 6) indicates the possibility of vibration of a structural member caused by the impulsive slamming load. Three effects result:

a) One slam will result in a number of cycles of free vibration, and fatigue life should be based on these rather than on the number of impacts.

b) The amplitude of the dynamic response may exceed the static response to the maximum Drag Coefficient depending on, amongst other factors, the natural frequency of the member. Shorter time periods of the vibration compared to that of the decay of the slamming force will result in larger responses.

c) The variation of Drag Coefficient with immersion shown in Fig. 27 indicates that for a particular structural member of the decay is slower at lower impact velocities. This is of significance in b) above since the dynamic response can vary with impact velocity.

Estimation of the stresses due to slamming will also require a knowledge of the spectra of impact velocities. Even so, the results from this programme of work give no indication of the effects of oblique impact, cylinder roughness or disturbed water etc., although these could be evaluated with the existing test rig.
UNIVERSITY OF SOUTHAMPTON REPORT NO 317

AN INVESTIGATION INTO WAVE SLAMMING LOADS ON CYLINDERS (OSFLAG 2A)

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