

FIRST YEAR
INTERIM REPORT

on

EDINBURGH WAVE POWER PROJECT

"STUDY OF MECHANISMS FOR EXTRACTING POWER FROM SEA WAVES"

SEPTEMBER 1975

Department of Mechanical Engineering,
University of Edinburgh

APRIL 1975 DUCK PERFORMANCE

Graph 1 shows the difference in efficiency between the old balsa wood model (D0008) tested by CEGB and the April 1975 (D0011) model with improved ballasting and internal dynamometers. The two curves for the D0011 show how efficiency varies with large and small waves. Curves are $\frac{P_{dyn.}}{P_{incid.}}$ It is not yet clear whether the short term ripples on the D0011 efficiency curves are random or systematic.

Graph 2 repeats the efficiency curve for D0011 and shows the amount of power unaccounted for in reflected waves, transmitted waves and dynamometer output. The lower curve is $\frac{P_{incid.} - (P_{dyn.} + P_{refl.} + P_{trans.})}{P_{dyn.} P_{incid.}} = \frac{P_{turb.}}{P_{incid.}}$

Graph 3 compares efficiency $\frac{P_{dyn.}}{P_{incid.}}$ with the fraction of energy which would have passed through a window of the same depth as the depth of immersion of the model. At the best periods the assumption made by Salter (Nature, Vo. 249, No. 5459, pp.720-724 June 21, 1974) was conservative.

Graph 4 shows the proportion of power contained in reflected and transmitted waves, $\frac{P_{refl.}}{P_{incid.}}$ and $\frac{P_{trans.}}{P_{incid.}}$.

S. H. Salter

D. C. Jeffrey

J. R. M. Taylor

Mechanical Engineering Dept.,
University of Edinburgh,

The attached copy of our original timetable shows completed work crossed out. Most of the apparatus scheduled for completion by February 1976 has been built, and most of the testing scheduled for completion by May 1976 has been done. Descriptions and results of these tests are attached. Duck mechanics are better understood and some performance improvements have been achieved.

There seems no reason to make any major changes to the timetable except to bring forward the second phase of the wave maker building from January 1977. This will allow a commercial company to get into production. Delays in building work at Riccarton may become a problem next year, and plans for an alternative tank have been submitted.

The extraordinary good fortune with which the project has been blessed was interrupted in August with the illness of Mr. Jeffrey which left him paralysed below the waist. He is making a good recovery but the date of his return is not yet known.

The high rate of inflation has forced us to defer orders since June to avoid overspending. New cost planning is based on the projected average of electrical and mechanical engineering costs published by the Department of Industry.

The final decision about the feasibility of wave power needs information about duck performance on the heaving surging rig and on a free floating backbone. On the basis of one year's work I can say that the possibility of economic power from sea waves has not been excluded.

S. H. Salter

30th September, 1975

UNRECORDED

SALTER

JEFFREY

TAYLOR

1974

O
N

~~Moving to K. B. Fixing Lab~~

1975

J

~~Ordering + Planning 1st Design Stage~~

F

Station 'India' ~~Variable Inertia~~

M
A

~~Internal Dynamometer (Fixed Bearings)
Polar Mould Model Building~~

~~Added Virtual
mass tests~~

M

~~Wave height measuring. Pressure
transducer conditioning~~

~~Plotting~~

J
S
O

~~Tank construction (or modification of Armfield). Lighting
test bench, cables~~

1976

N
J

~~Wavemaker improvements to extend
absorbing bend~~

~~Calibrations
and checks~~

F
M
A

Surge, Heave + Pitch mounting Dynamic
variable controls

~~Fixed bearing
tests~~

M
J
J

Analysis, Writing, Thinking, trying
anti-bending moment tricks

S.H.P. tests

S
O
N

Multivane models construction with
scaled compliance setting and bending
moment cells at centre

1977

J
F
M

2nd Phase of wave maker building
Setting up at Heriot-Watt

Plotting
Re-testing S.H.P.

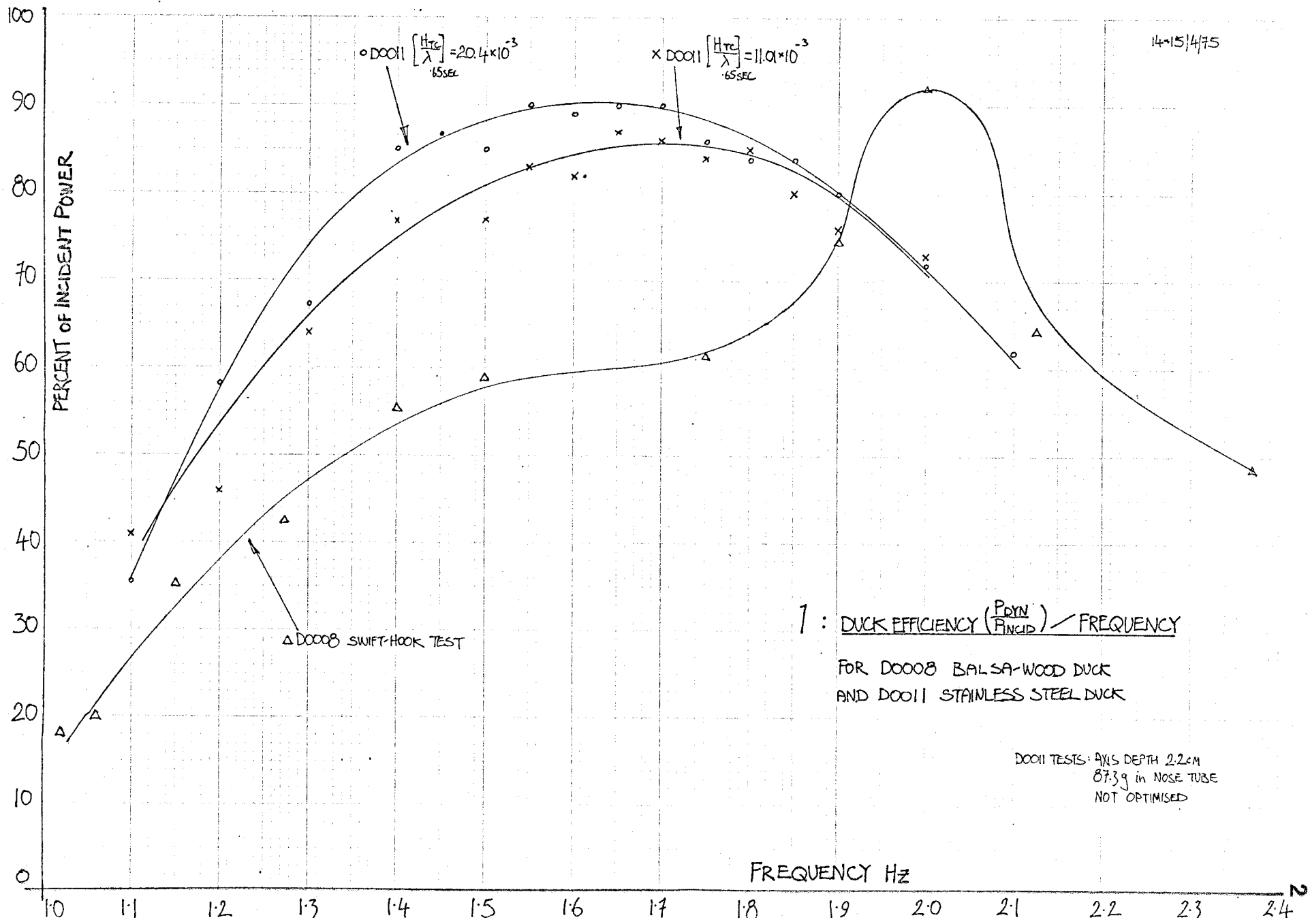
A

Multivane trials in big tank. Bending moment measurements.

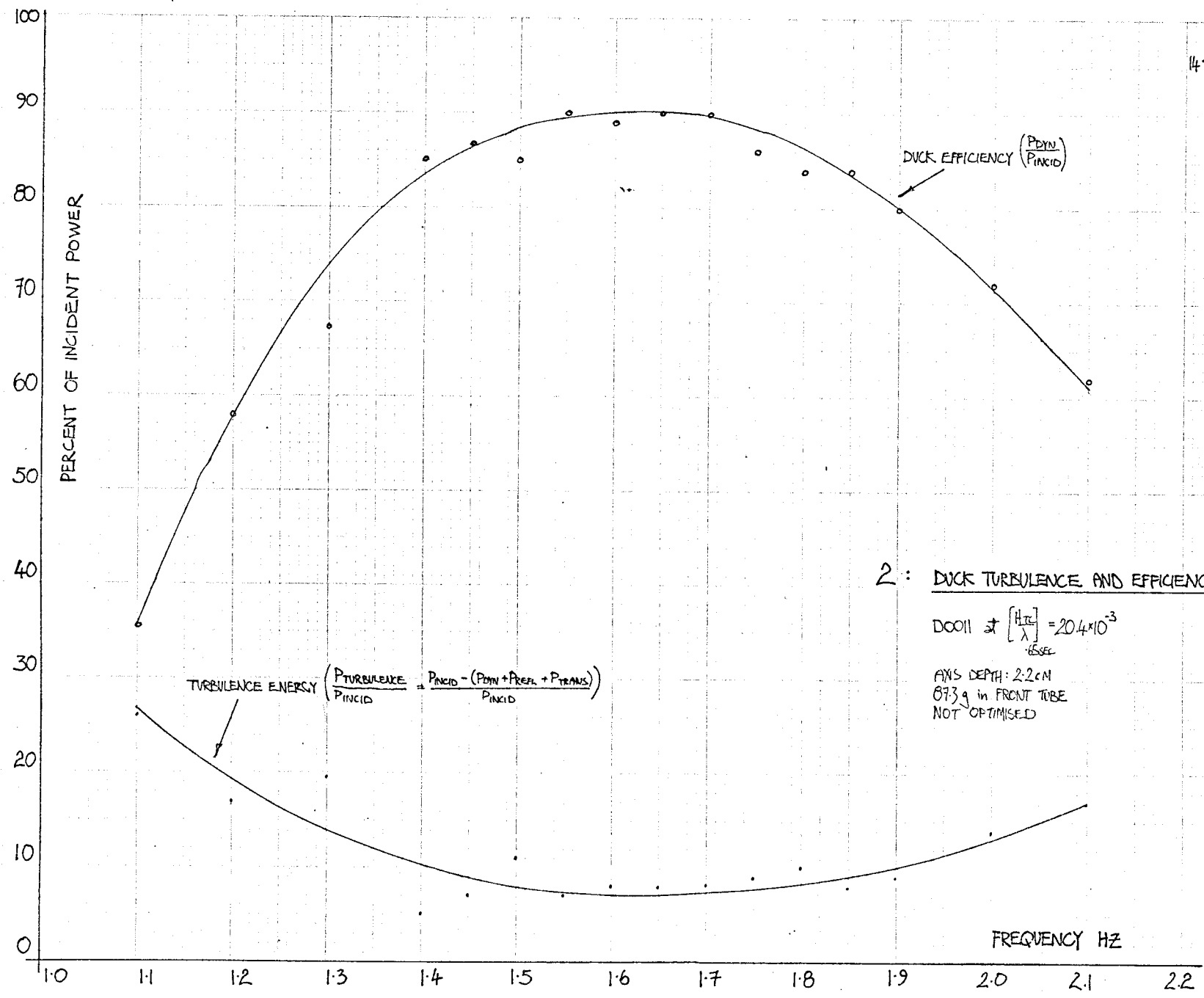
J
J

Writing up

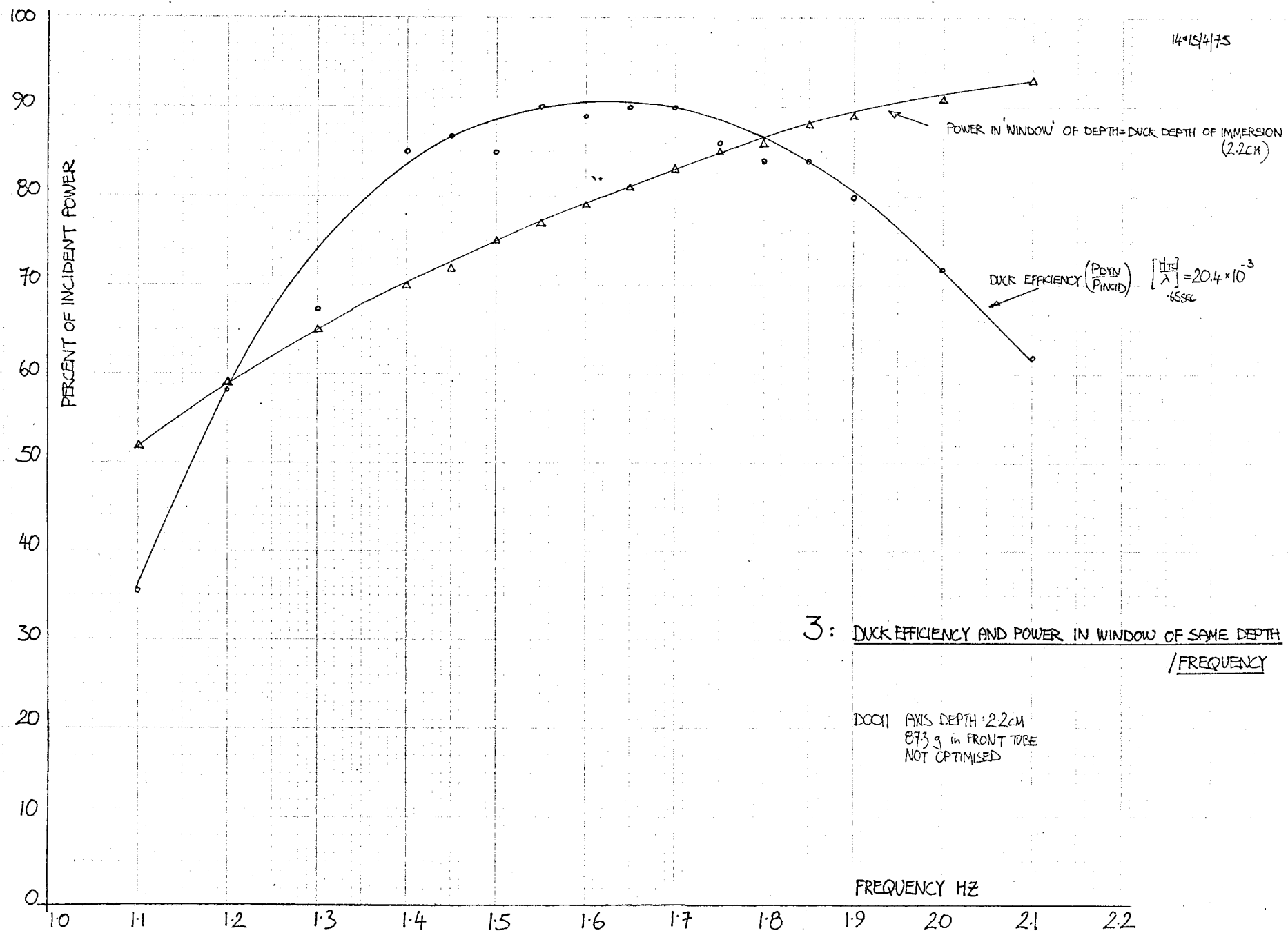
Instrument
Exploitation



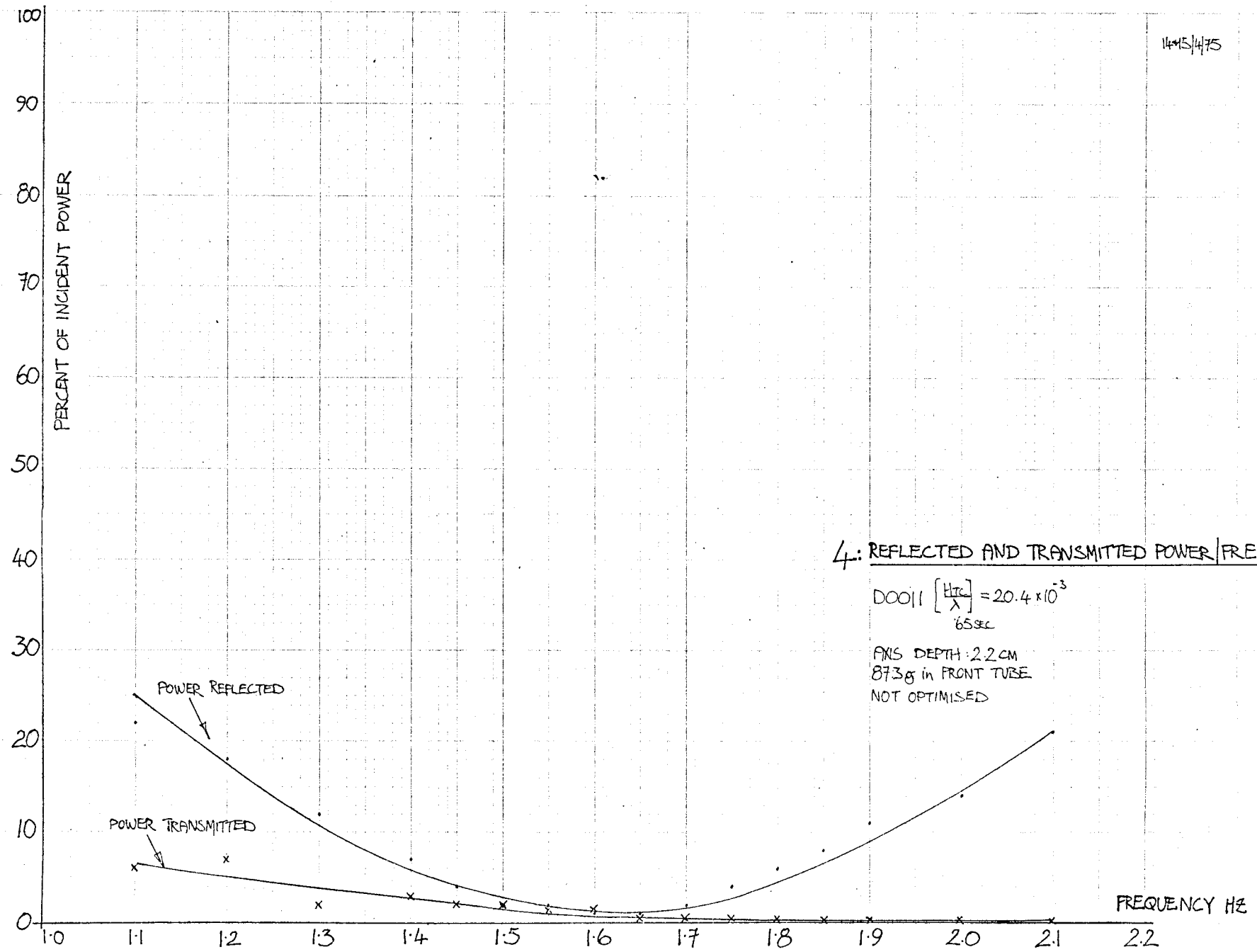
14-15/4/75



14-15/4/75



11/15/475



4: REFLECTED AND TRANSMITTED POWER / FREQUENCY

$$DOO11 \left[\frac{Hz}{X} \right] = 20.4 \times 10^{-3} \text{ '65SEC}$$

AXIS DEPTH: 2.2 CM
87.3g in FRONT TUBE
NOT OPTIMISED

DAMPING, HUB DEPTH EXPERIMENTS

Graph 5 contains a set of efficiency curves for an arbitrary choice of ballast and hub depth using different damping factors. Curves for damping factors 2 are close to those for 3. We conclude that damping factor is not particularly critical.

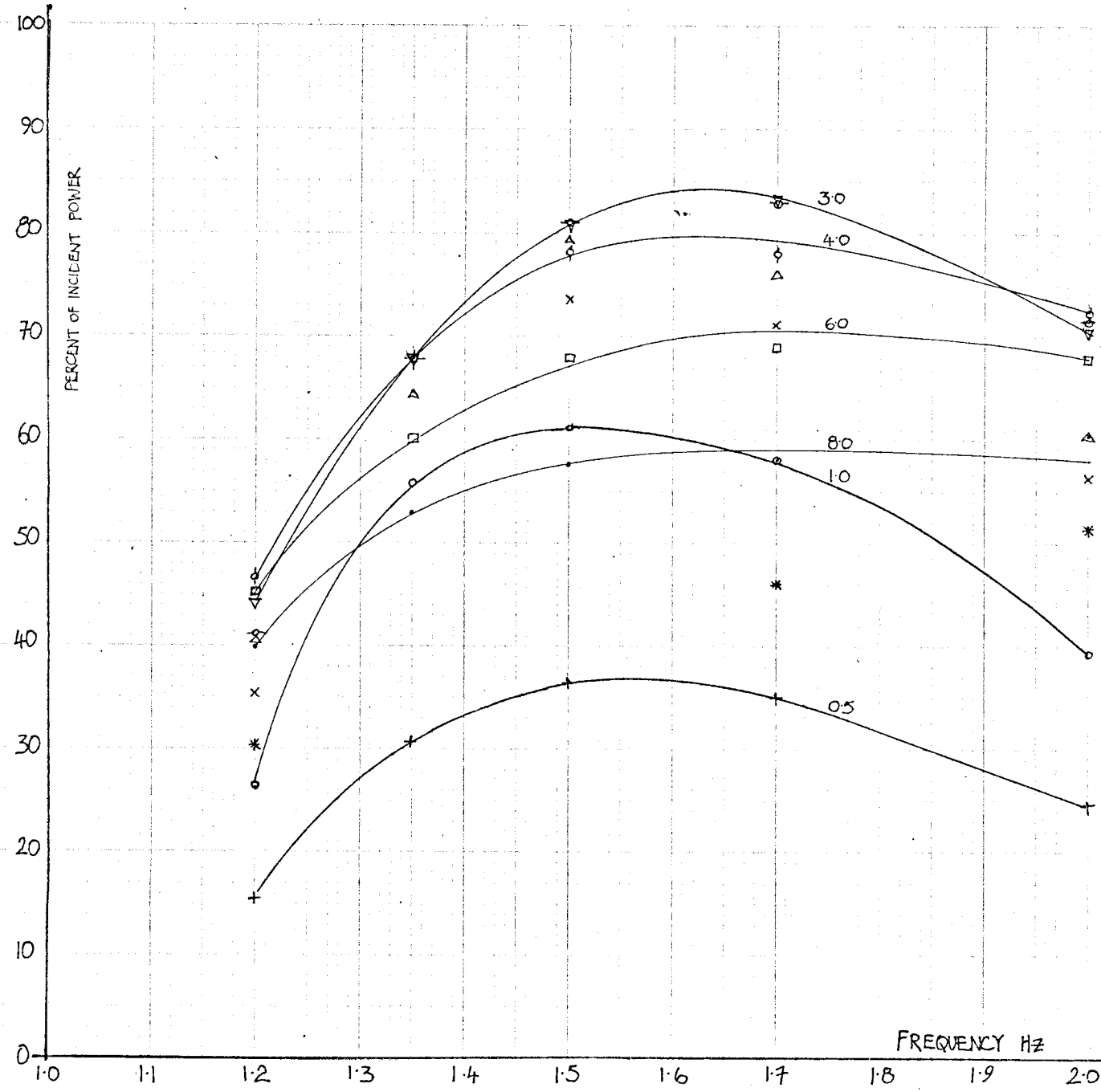
Graph 6 shows efficiency against damping factor for different frequencies for the same ballast and hub depth.

Graph 7 picks out from Graph 5 and 6 the best damping factor at each frequency. We get the lowest optimum damping at resonance. Both above and below resonance the duck prefers higher values of damping factor. We see that doubling or halving optimum damping loses only 15%.

Graphs 8 and 9 were plotted to show the effects of varying hub depth for a fixed ballast. Two damping factors were used. In Graph 8 we chose a factor of 3 newton cm. per radian per second for all points. In Graph 9 we used a value suggested by Graph 7. It proved a good guide for hub depths near 2.2 cms.

12th June, 1975

9+10/675



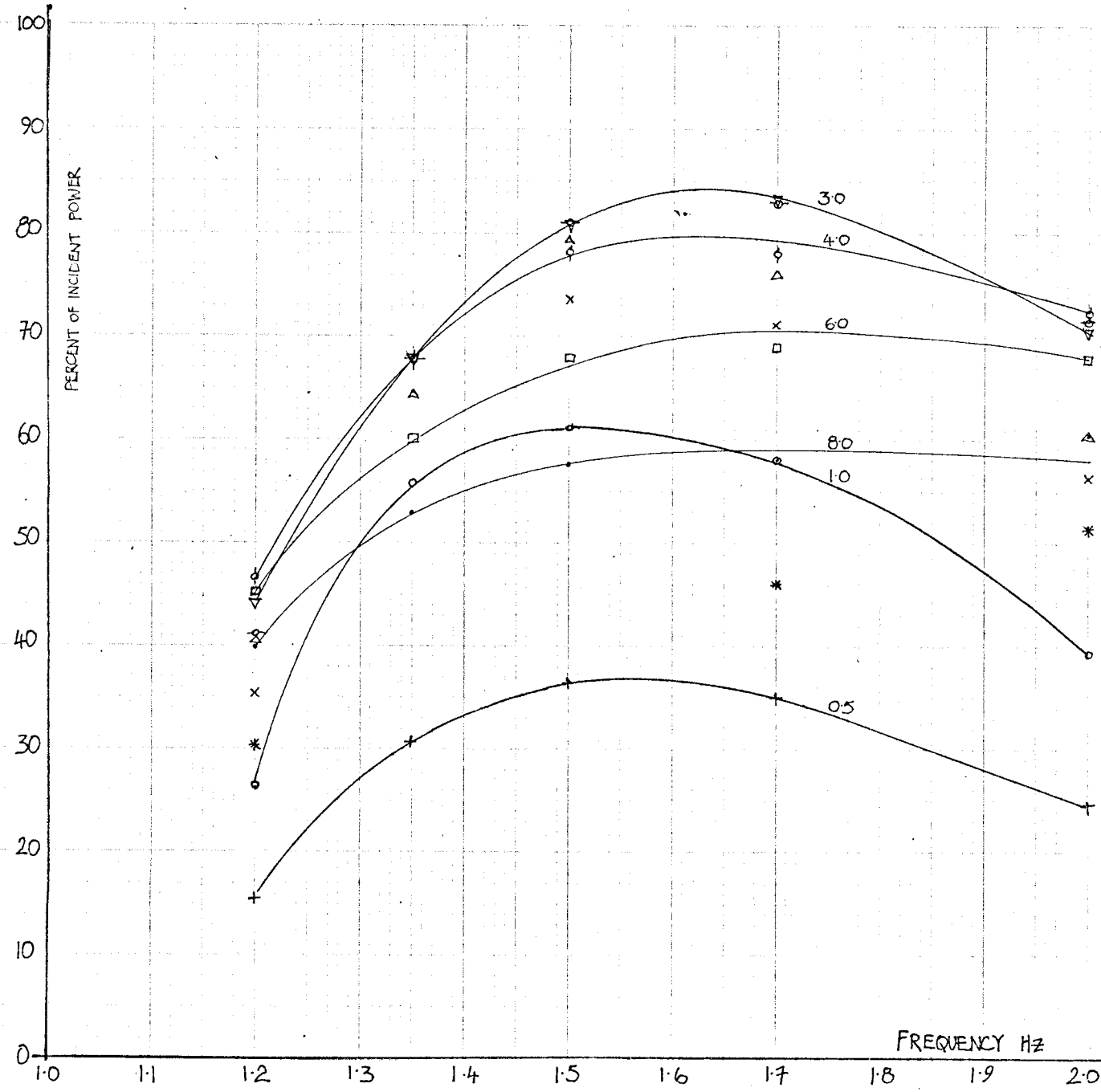
DAMPING FACTORS

- + 0.5
 - o 1.0
 - x 1.5
 - Δ 2.0
 - ◊ 2.5
 - ▽ 3.0
 - ◊ 4.0
 - 6.0
 - * 8.0
 - * 12.0
- N CM
RAD/S

5: DUCK EFFICIENCY - FREQUENCY
FOR DIFFERENT DAMPING FACTORS

BALLAST: 86.5 g in foot tube
AXIS DEPTH: 2.2 cm

9+10/675



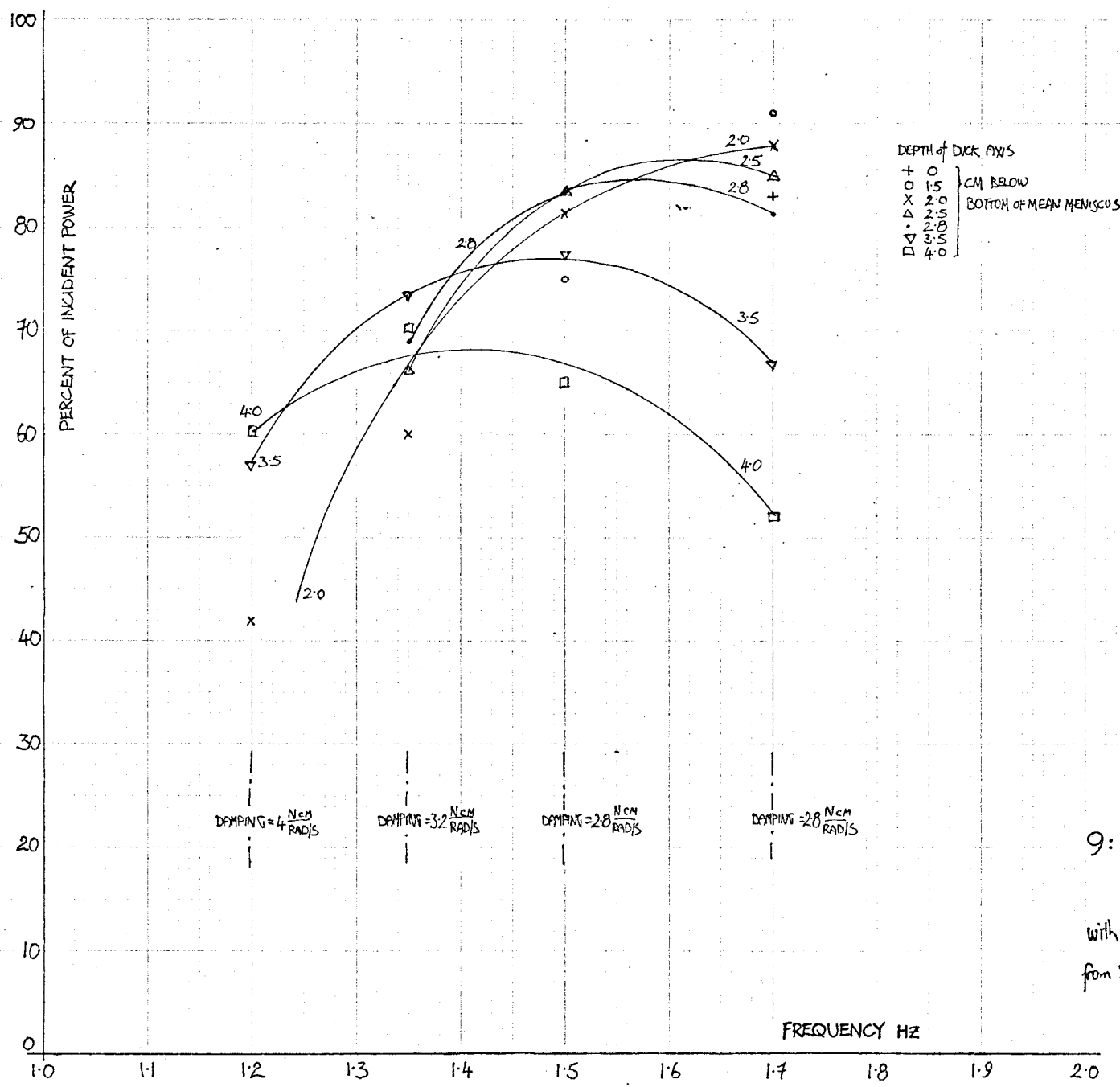
DAMPING FACTORS

- + 0.5
 - o 1.0
 - x 1.5
 - Δ 2.0
 - ◊ 2.5
 - ▽ 3.0
 - ◊ 4.0
 - 6.0
 - * 8.0
 - * 12.0
- N CM
RAD/S

5: DUCK EFFICIENCY - FREQUENCY
FOR DIFFERENT DAMPING FACTORS

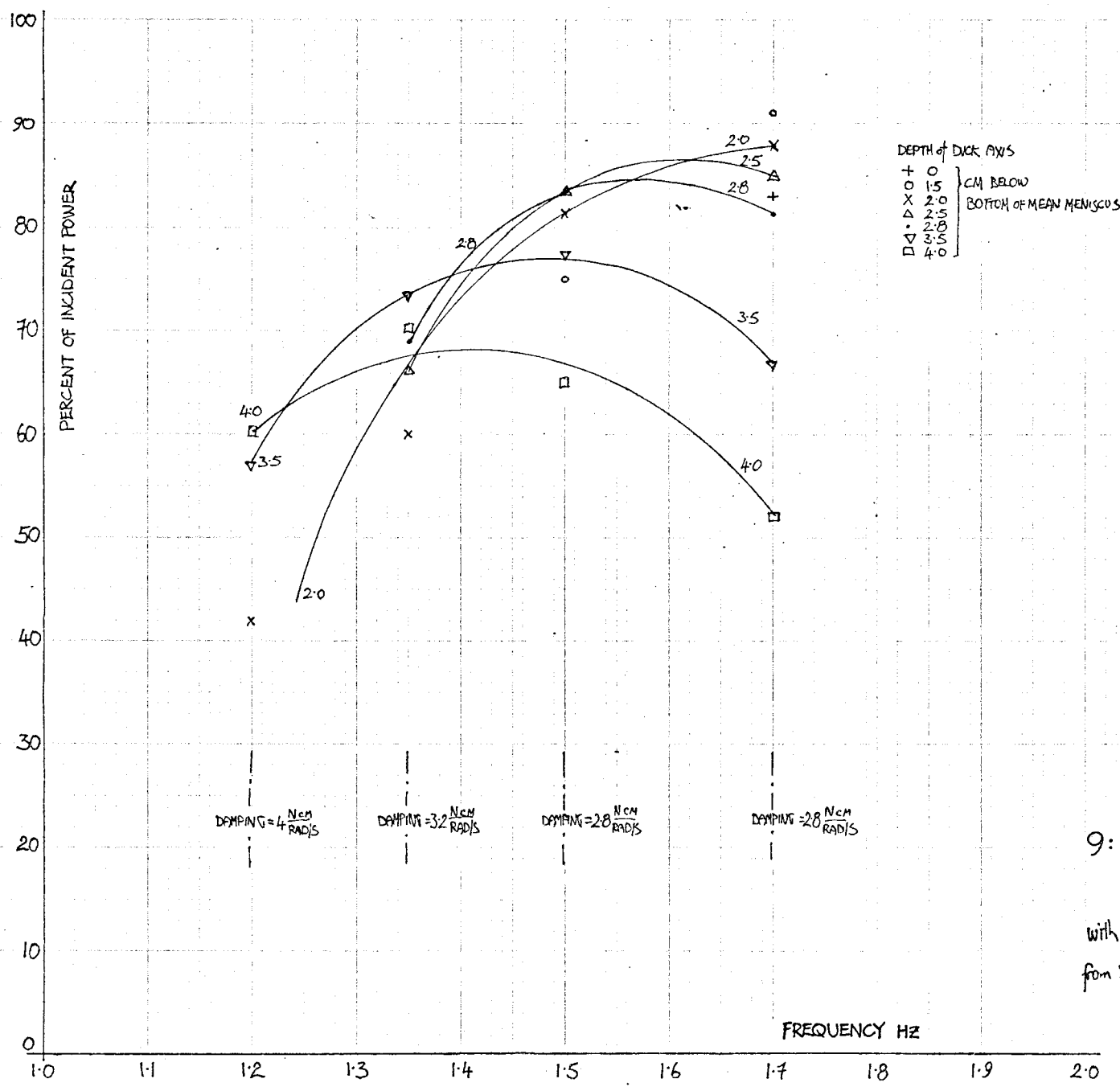
BALLAST: 86.5 g in foot tube
AXIS DEPTH: 2.2 cm

10/6/75



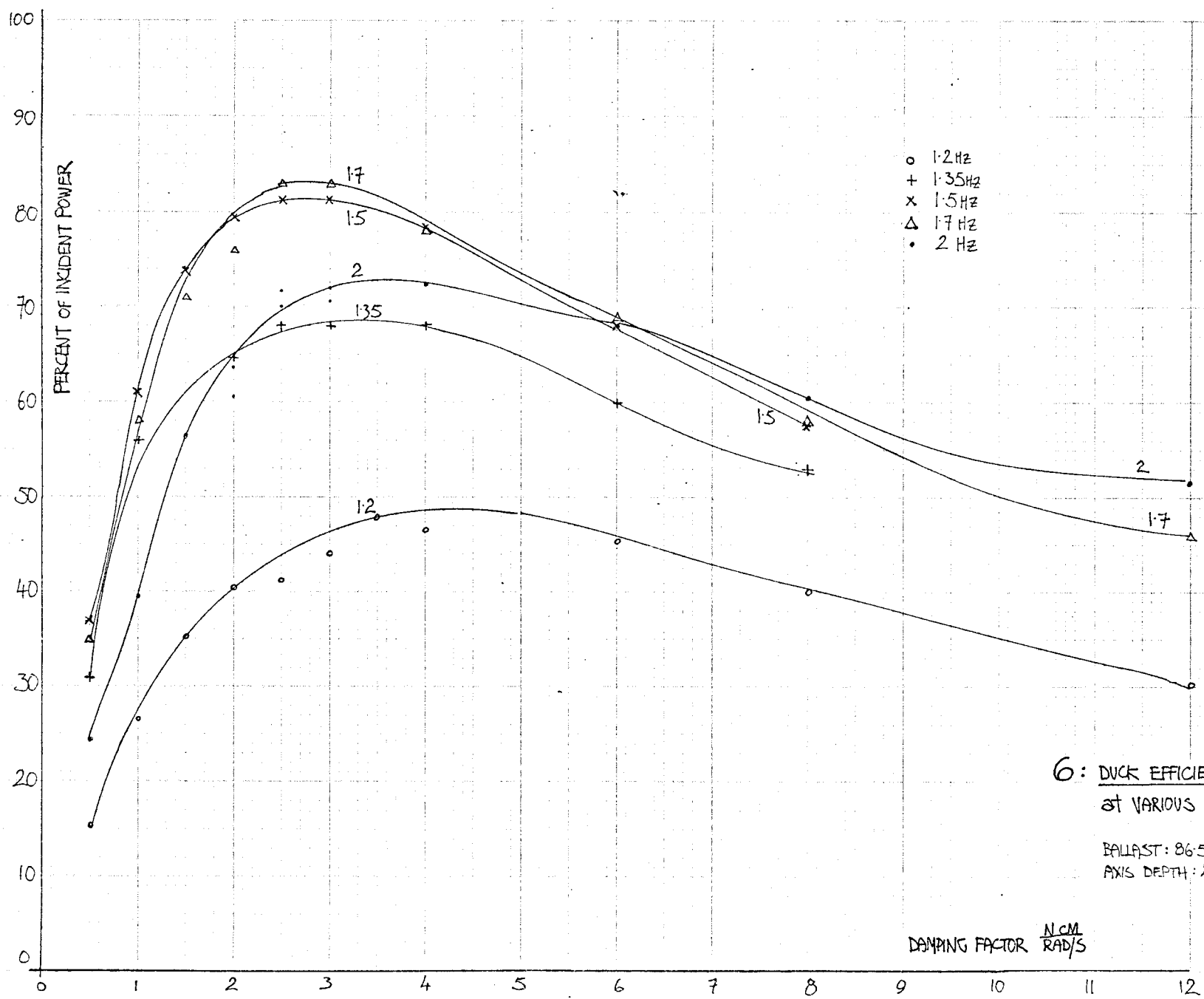
9: DUCK EFFICIENCY / FREQUENCY
 at VARIOUS DEPTHS of IMMERSION
 with DAMPING at each FREQUENCY, CHOSEN
 from DATA of 9-10/6/75 to give HIGHEST EFFICIENCY
 BALLAST: 86.5g in front tube

10/6/75



9: DUCK EFFICIENCY / FREQUENCY
 at VARIOUS DEPTHS of IMMERSION
 with DAMPING at each FREQUENCY, CHOSEN
 from DATA of 9/10/6/75 to give HIGHEST EFFICIENCY
 BALLAST: 86.5g in front tube

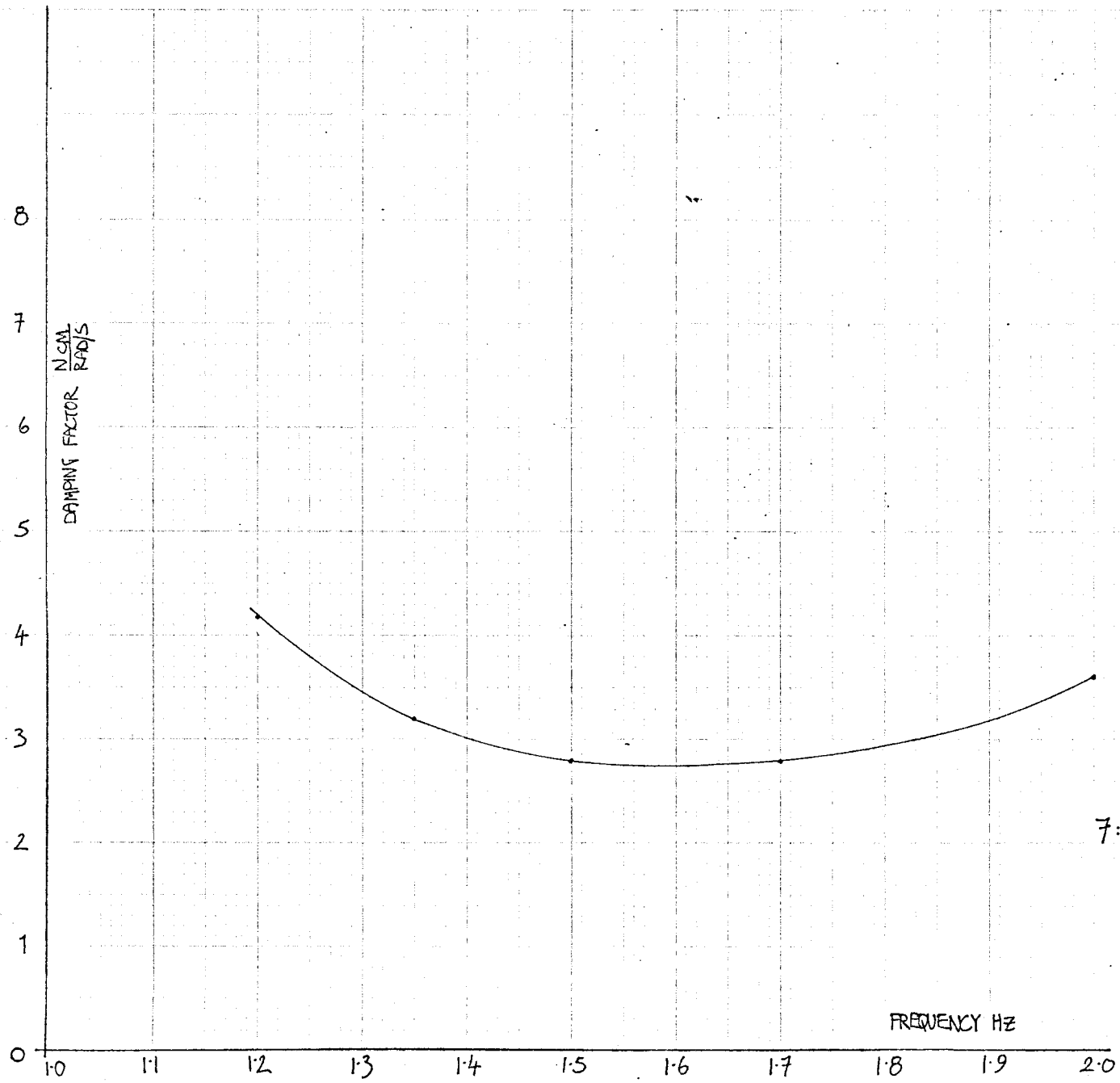
9/10/6/75



6: DUCK EFFICIENCY-DAMPING
AT VARIOUS FREQUENCIES

BALLAST: 86.51 Kg in front tube
AXIS DEPTH: 2.2 CM

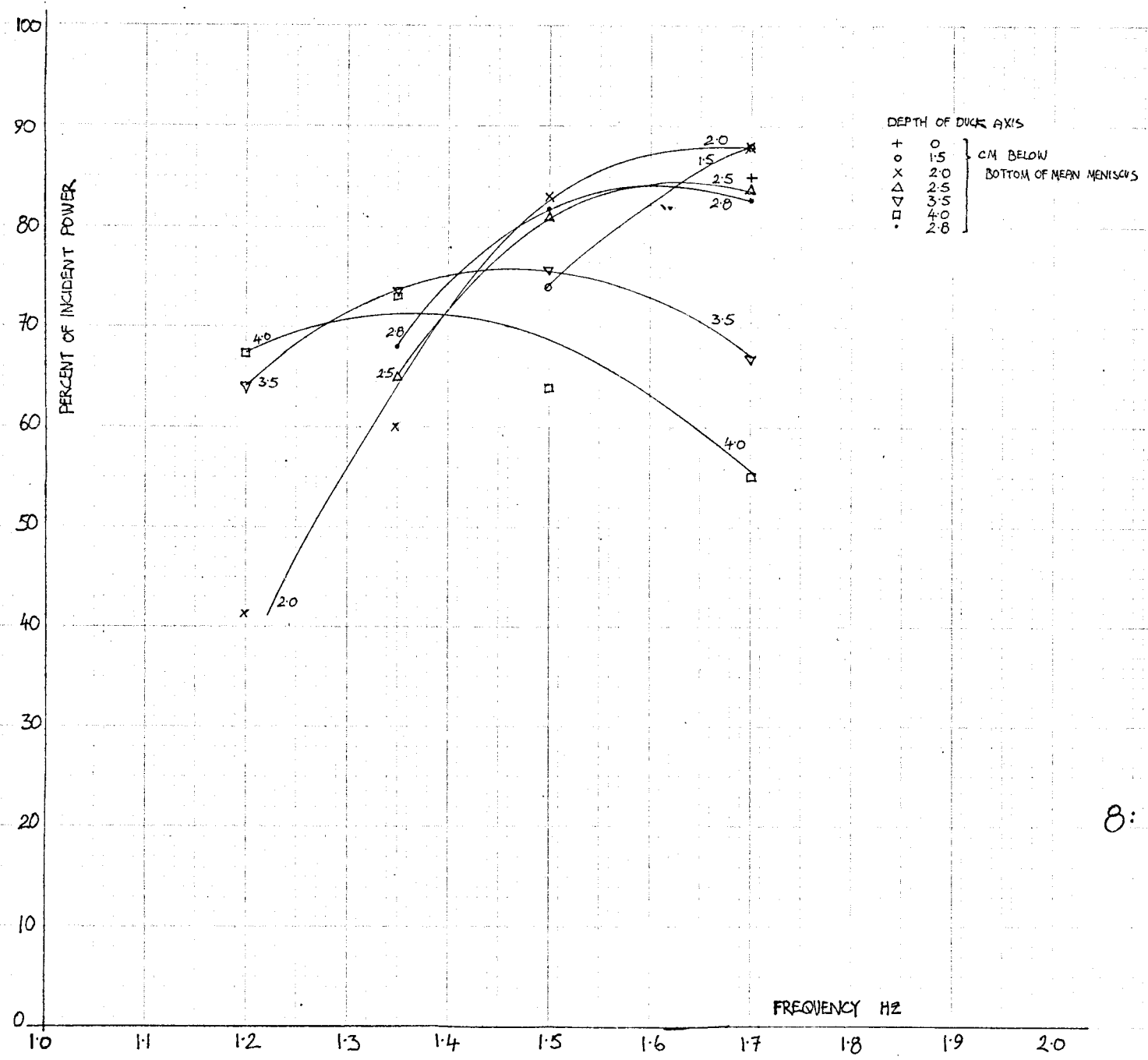
9+10/675



7: DAMPING for HIGHEST EFFICIENCY / FREQUENCY

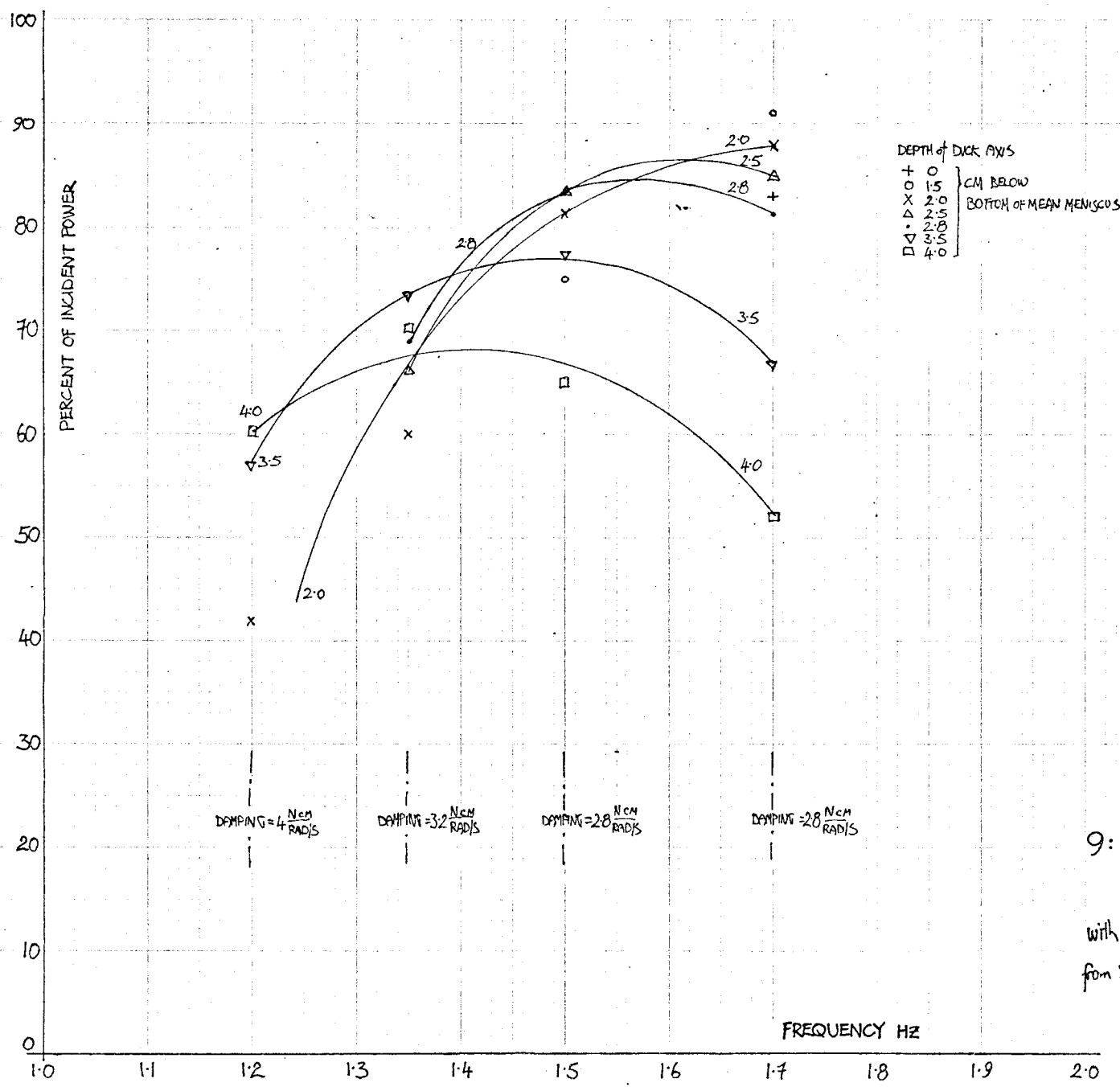
BALLAST: 86.5g in foot tube
AXIS DEPTH: 2.2 cm

10/675



8: DUCK EFFICIENCY - FREQUENCY
 at VARIOUS DEPTHS of IMMERSION
 with DAMPING = $3 \frac{N \cdot CM}{RAD/S}$ throughout
 BALLAST: 26.5g in foot tube

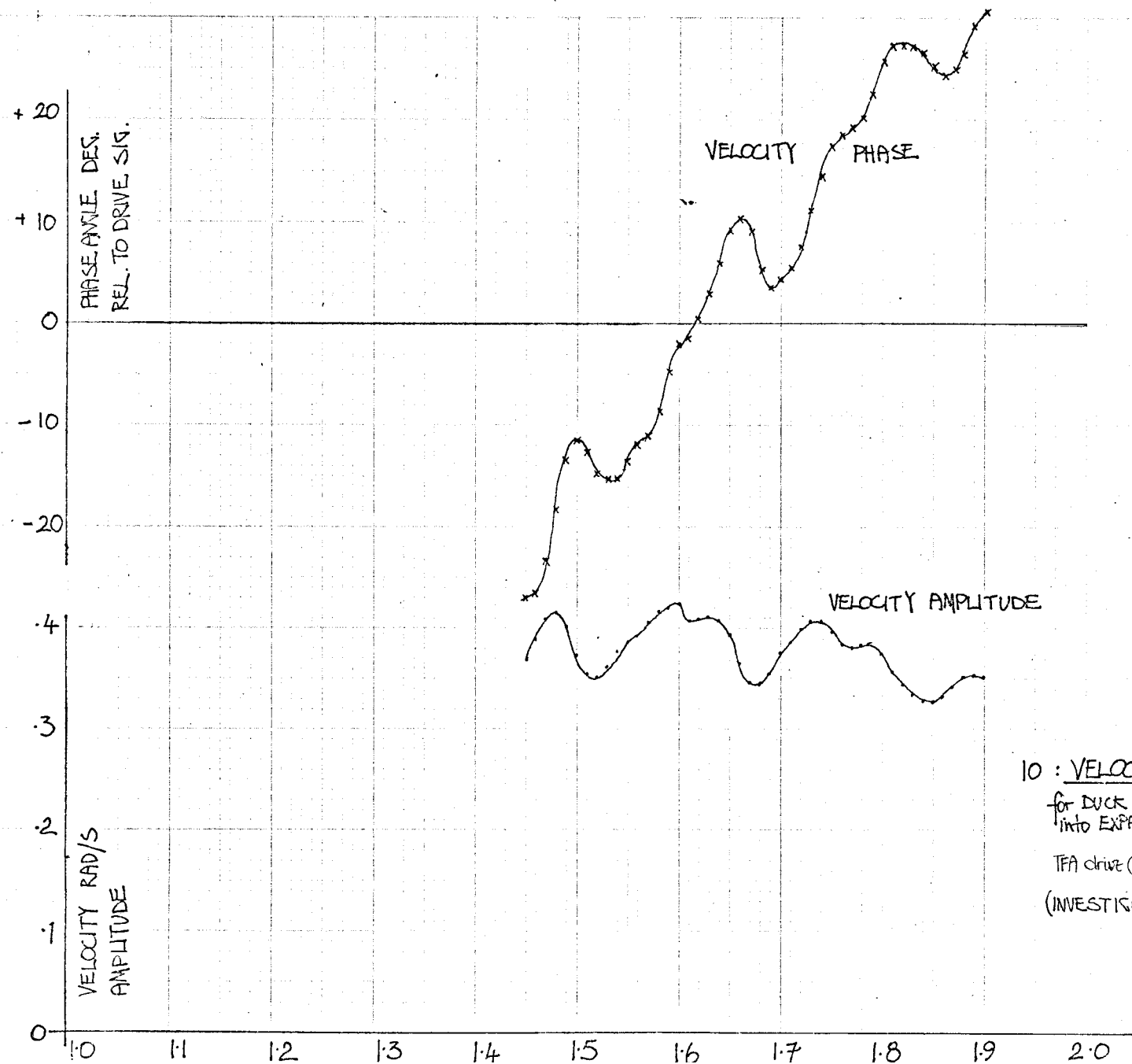
10/6/75



9: DICK EFFICIENCY / FREQUENCY
 at VARIOUS DEPTHS of IMMERSION
 with DAMPING at each FREQUENCY, CHOSEN
 from DATA of 9-10/6/75 to give HIGHEST EFFICIENCY
 BALLAST: 86.5g in front tube

Problems with Frequency Response Measurement

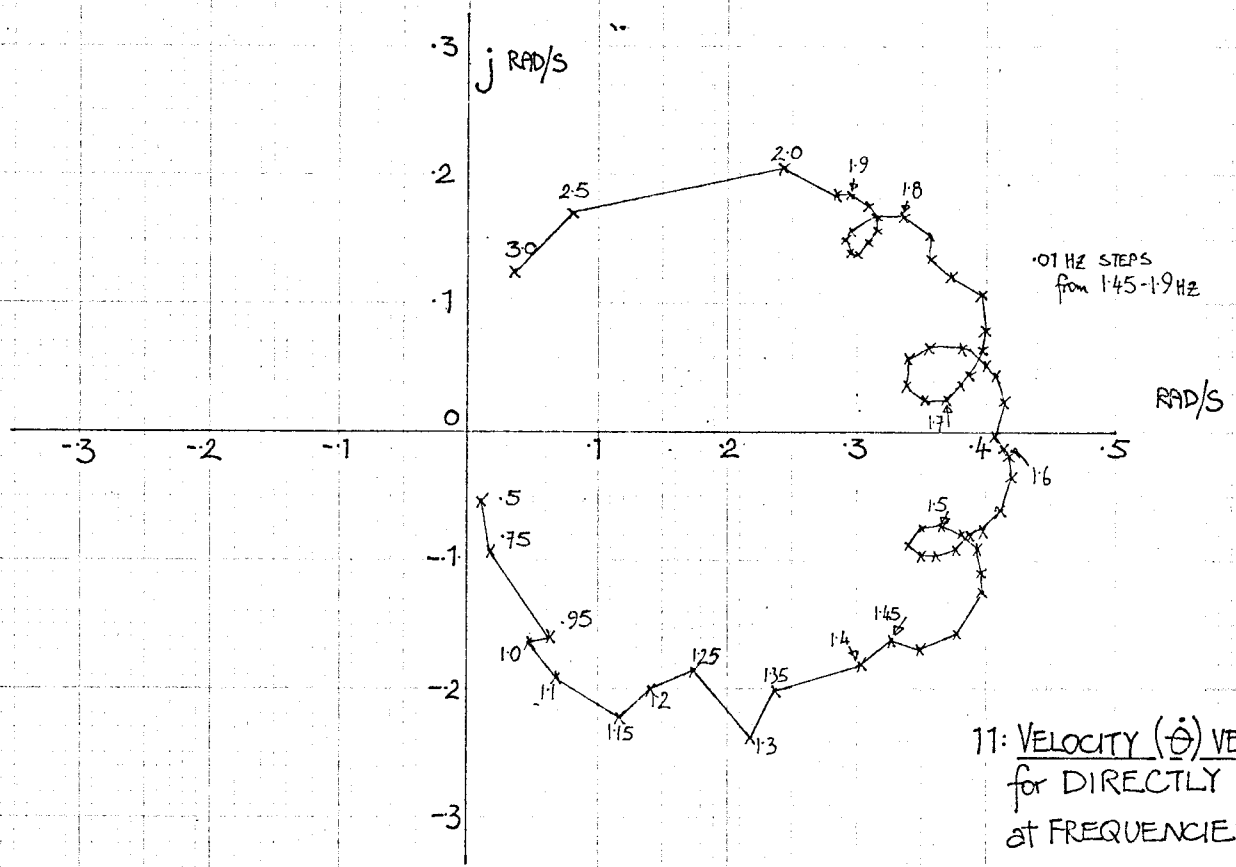
Graphs 10 and 11 reveal a problem with instrumentation. The natural nodding frequency of a duck is of great interest. We have tried to measure it by applying a sinusoidal drive torque and recording response. At resonance the velocity should reach a maximum and be in phase with excitation. The amplitude plot is clearly useless. A statistical mean of the phase readings is more informative. The duck was sending waves into the best beach available to us which was 100 sheets of Expamet. Its reflection coefficient was only 4%. The number of test points shows how much work is involved. The tank needs about one minute per point.



11/6/75

10 : VELOCITY WITH PHASE / FREQUENCY
 for DUCK DRIVEN DIRECTLY BY T.F.A.
 into EXPANET BEACH
 TFA drive @ 1.0 VOLT
 (INVESTIGATING NATURAL FREQUENCY)

13/4/75



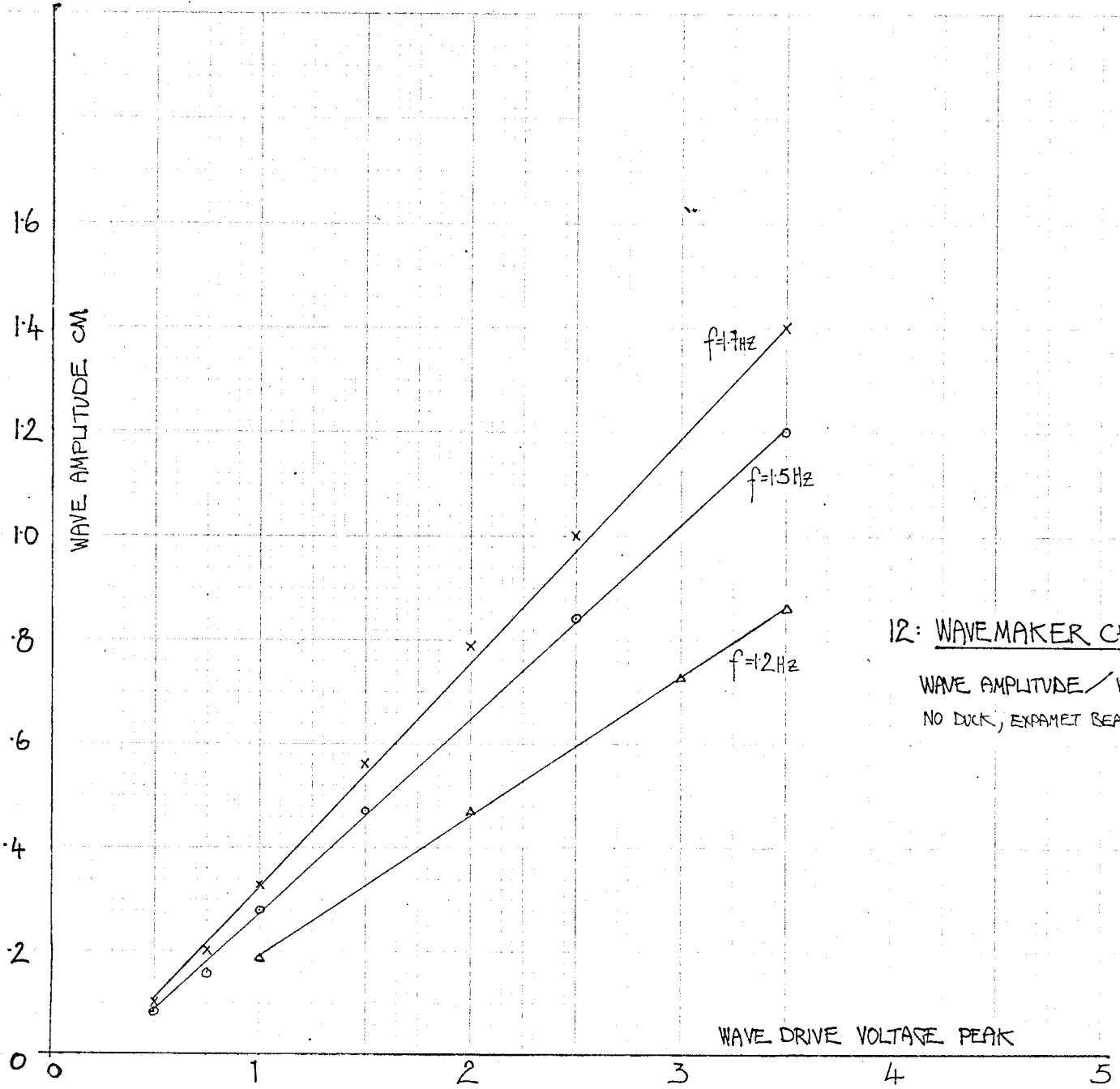
11: VELOCITY ($\dot{\theta}$) VECTOR DIAGRAM
for DIRECTLY DRIVEN DUCK
at FREQUENCIES from .5-3.0 HZ
as indicated

TFA WAVE DRIVE: 1.0 VOLT
(INVESTIGATING NATURAL FREQUENCY)

Mk I Wave Maker Characteristics19th June 1975

There is a fruitful cross fertilisation between putter-inners and taker-outers. Graph 12 shows some tests on the linearity of the duck shape as a wave maker. We sent waves into a 100 sheet Expamet beach with only wave gauges in the tank. The curves show that there is some stiction in the wave maker bearings and that the amplitude response is not linear with frequency. We would like to try cross pivot leaf springs to get rid of stiction and some compensation electronics to flatten frequency response.

18475



12: WAVE MAKER CHARACTERISTIC (MKI)

WAVE AMPLITUDE / WAVE MAKER DRIVE
NO DOCK, EXPANET BEACH

EFFICIENCY AND WAVE SIZE EXPERIMENTS

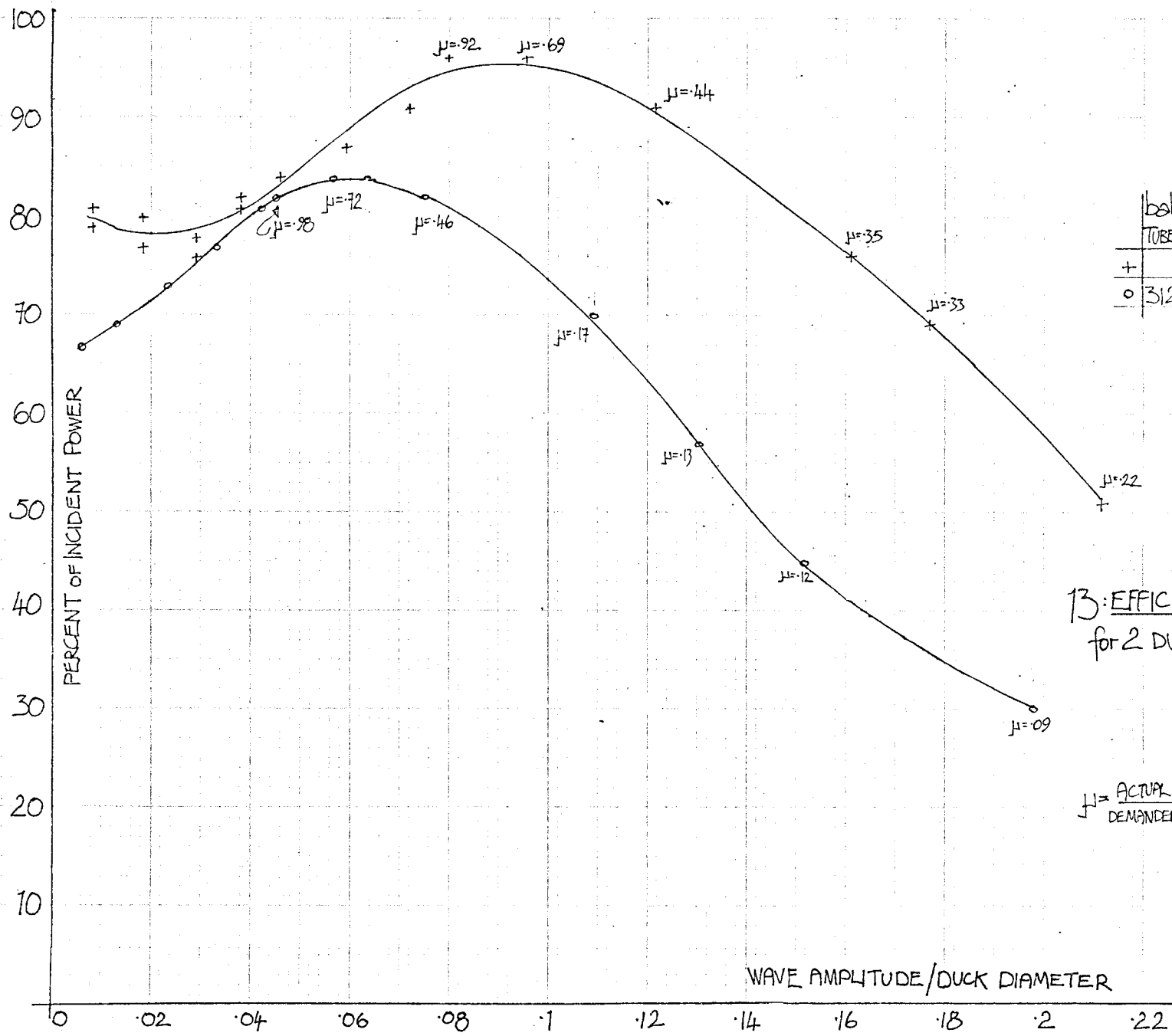
20.6.75

Graph 13 shows the effects of wave size on efficiency at frequencies of 1.7 Hz and 1.2 Hz. The V25Y5 motor is limited in torque to .04 Newton meters. (V25Y6 units with 3 times the torque are on order.)

We define the torque ratio M as the actual torque supplied, divided by the torque demand from the damping setting. The extreme value of .09 occurred.

At wave powers of a few milliwatts stiction probably accounts for the low efficiency. Note that efficiency is still climbing even with value of M less than .75. The abscissa is wave amplitude, i.e. half trough to crest height, relative to duck diameter. So that even with the rather low torque limit the ducks are working tolerably at 100 x scale and 100 kw/meter power densities. But there is room for improvement. V25Y6 curves will be most interesting.

19/6/75 rechecked (23/7/75)



BOULDER	STONE	AXIS DEPTH CM	CLAMPING NEM/INCHES	FREQ HZ
+		312	2.2	2.8
o	312	156	5.0	6.0

13: EFFICIENCY/WAVE AMPLITUDE
for 2 DUCK CONDITIONS DOON

$\mu = \frac{\text{ACTUAL TORQUE}}{\text{DEMANDED TORQUE}}$ ($\mu = 1$ when no limiting)

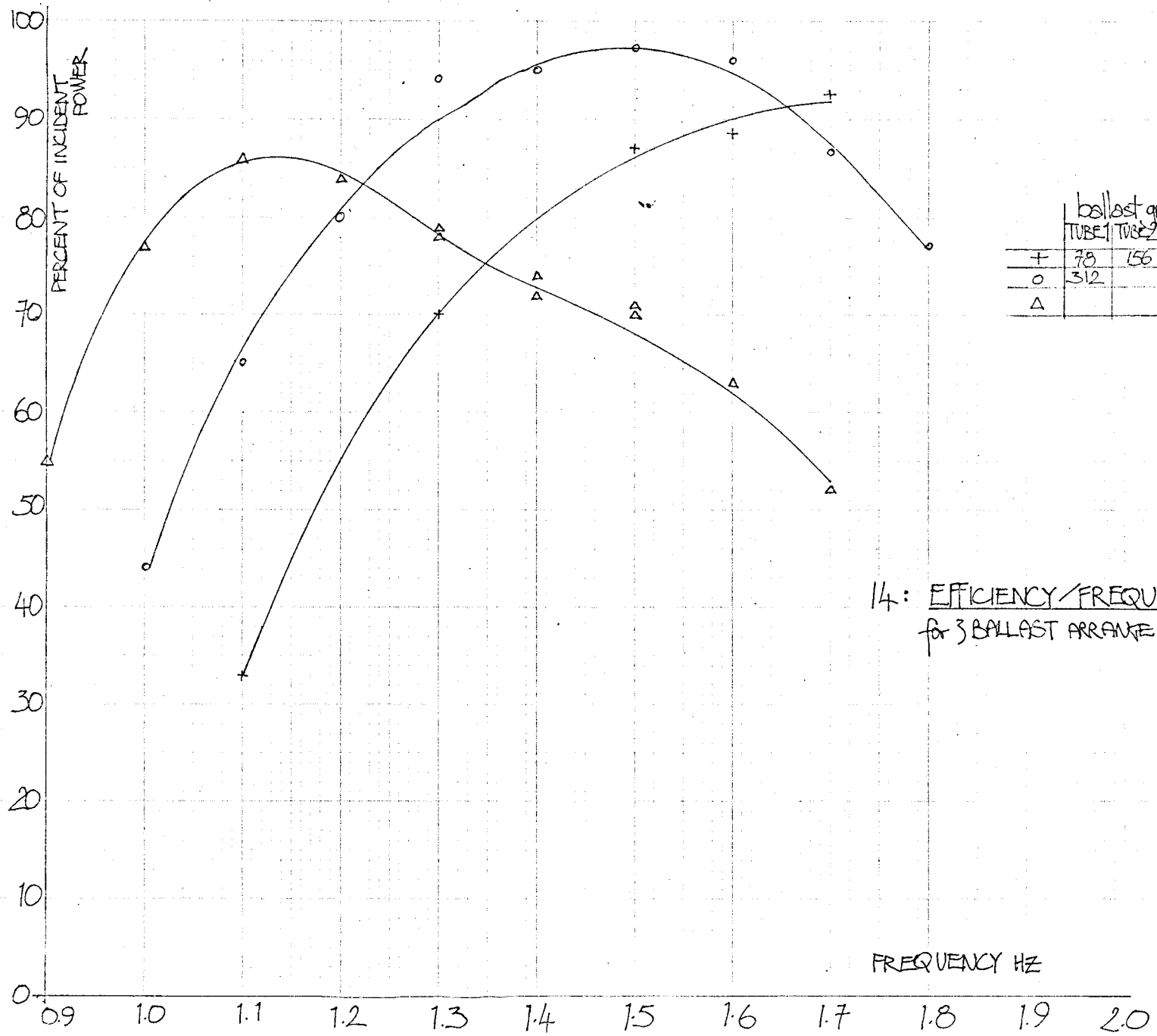
VARIABLE INERTIA EXPERIMENTS

Graphs 14 and 15 show the results of adjusting ballast on D0011. We were trying to push the response peak to lower frequencies. They show some very high efficiencies which we would prefer to keep confidential at this stage. A large number of factors go into the calculation. We believe that the results are accurate to about 4%. They are included to show the shape rather than the absolute values.

The triangular points on Graph 14 were made with a deck cargo of ballast which may not be seaworthy. Future models must be made with more ballast space.

More dramatic extensions to low frequency performance can be produced by electronic trickery which can reduce the 'spring rate' without altering duck shape or inertia. Efficiency of 80% at 1 Hz has been produced. It may be possible to implement this on full scale equipment using 'smart' hydraulics.

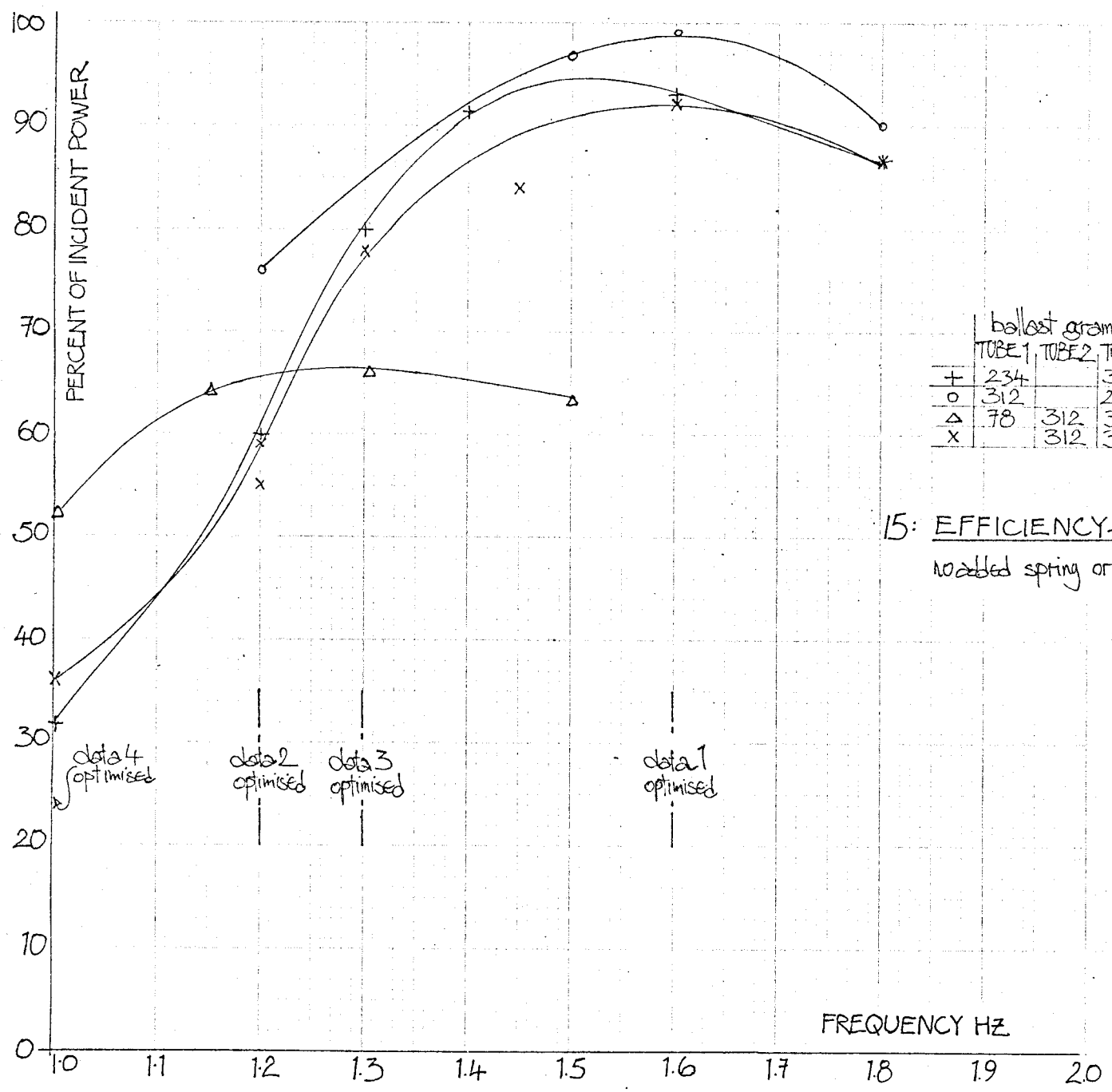
1628/6/75



	ballast arrangement			axial clamp	clamping	obs	obs
	TUBE 1	TUBE 2	TUBE 3	ADDITIONAL	Num. Poles	obs	obs
+	78	156			4.05	2.6	16/75 1
o	312		156		5.1	6.0	16/75 3
Δ			312	3x156 ST	5.1	4.5	25/75

14: EFFICIENCY/FREQUENCY
for 3 BALLAST ARRANGEMENTS D0011

23/6/75



	ballast gramme			axis	damping	data
	TUBE 1	TUBE 2	TUBE 3	depth cm	Nm/roots	
+	234		312	4.5	6.5	1
o	312		234	4.9	6.5	2
Δ	78	312	312	6.5	9.25	4
x		312	312	5.7	6.0	3

15: EFFICIENCY—FREQUENCY

loaded spring or inertia DO011

Instant Masuda

One of the more attractive methods for getting power out of sea waves has been devised by Masuda. He uses a structure consisting of vertical cells open at the bottom to the sea and closed at the top. Water rising and falling inside each tube blows and sucks at the air above it. The alternating air flows can be rectified and used to drive a turbine. There are no primary moving parts. Masuda suggests that a number of tubes could be arranged in a ring of diameter larger than a wave length so that it will form a fairly stable platform. This is in contrast to his earlier work with the single cell suck-and-blow technique which he has used very successfully to power navigation buoys and which has been analysed by McCormick.

On the assumption that the large ring of cells provides a stable reference we tested just a single variable geometry cell made of perspex on a rigid mounting. It has sliding walls fore and aft, an adjustable compression ratio and outlet nozzle area. Work is measured as the product of pressure difference and air flow. Energy is dissipated by turbulence in the nozzle. Flow is sensed by measurements of the water position inside the cell.

Resonant frequency is well predicted by taking into account the added virtual mass of the water below the vessel. The performance shows the familiar shape of a resonant absorber but we could get efficiencies of only about 30%, much less than Masuda. Deflections of the perspex were visible and it would be well worth fitting the mounting with strain gauges. Tracer fluid shows turbulence around the entrance lips despite round edges. Some careful redesign of the entrance is indicated. At resonance there was a 90° phase lag between water level inside and outside the cell. Compression ratio does not seem an important factor.

The gate position for resonance is such that not much energy is transmitted behind but rather a large amount is reflected. If a load could be devised which added some inertia or reduced some stiffness then a smaller gate depth could be used and perhaps some of the transmitted energy taken by extra cells. In this way I would hope to get efficiency of the whole above 50%. It would be interesting to try deeper cell walls to the rear.

It will not be so easy to submerge a large installation and there is little chance of getting water over the top to reduce wave thrust. Masuda agrees that mooring will be more of a problem than with ducks. The drag coefficient will be high.

- 2 -

Our instrumentation would be better with twin wire flow sensors. The single wire units need re-calibration after each change in geometry because the perspex is an electrical insulator. Pressure transducers capable of resolving to less than 1 mm of water are necessary. We had noise and drift problems. Until improvements are made we prefer not to release performance curves.

I feel that the idea will be most useful in applications where some value can be put on the real estate of the top deck.

Mounting Forces

3/23.7.75

Plots 18, 19 and 20 give information about the surge and heave forces on a rigid mounting. A constant scale of 1 cm = 2 Newtons is used throughout. Plots 18 and 19 are one particular duck model at 1.2 and 1 Hz. Plot 20 is a cylinder of diameter 10 cm, the same as the basic diameter of the duck model. Both cylinder and duck are 29.7 cm long.

At small wave levels, for example 19(a), the forces are sinusoidal and the value of the horizontal component is predicted exactly by classical wave force theory. (See Weigel, R.L., Oceanographic Engineering, p. 254, Prentice Hall.)

Vertical forces seem to be less, perhaps because nearness to surface reduces C_m from 2 to 1.5. Orientation of the Lissajous is shown by points a and b. These were determined by setting a wave gauge at one wave length behind the duck and using a squared up version of its output to operate the brightness of the oscilloscope. Note different obliquities between 19(a) and 20(a). As expected the force is dominated by inertia components. It seems to rise more or less linearly with wave height rather than with the square of it as would the velocity forces. Indeed after (d) they seem to grow rather slower than they should. Both inertia and velocity forces should rise with the cube of scale.

The speed of movement of the point round the Lissajous figure is not constant and so the position of the centroid of the figure is not an indicator of mean resultant force. Instead we put the force signals through low pass filters, and mark the resultant at point x on each force plot. We find that there is not as much down wave force as might be expected. Indeed on the cylinder measurement 20 b, c, d there is a small but detectable net force towards the wave maker. When released the cylinder moves slowly but surely up the tank. Perhaps this is caused by water breaking over the cylinder which has a very small positive buoyancy.

The heave force which is downwards in the crest of the wave shows a definite bias such as to sink the equipment. This looks more promising as a self-protection mechanism than does the leading buoy. The pseudopodic protuberances on the higher wave plots are a splendid test for wave theorists.

Plot 18(h) shows the results of large waves on a rigid duck shape. Duck nodding was prevented by lock straps. Wave conditions were the same as for 18(f). Heave and surge forces are nearer to being in phase, so that the resultant peak force is higher than for a working duck.

3.7/75 +23/75

a)

VERY SMALL RIPPLE BEHIND DUCK

"TOP OBS. CENTRE"

$\bar{F}_x = 0$
 $\bar{F}_y = -0.55N$

FREQ. DRIVE	AMPL.	WAVE AMPL.	μ	P _{OUT}	η
12Hz	67V	4cm	No LIMITING	12mW	79%

b)

NOSE JUST GOING UNDER

$\bar{F}_x = -0.6N$
 $\bar{F}_y = -0.16N$

12Hz	97V	6cm	0.63	28mW	85%
------	-----	-----	------	------	-----

DOO11

BLAST FRAME TUBE 1	BLAST FRAME TUBE 2	BLAST FRAME TUBE 3	TID DEPTH CM	DENSITY NCM/RINGS
312		156	5.1	6.0

cf. GRAPH #14 (0) + (11) (75 data)

1.2 HZ

$\mu = \frac{\text{ACTUAL TORQUE}}{\text{DEMANDED TORQUE}}$

displacement of water at duck

c)

$\bar{F}_x = -0.55N$
 $\bar{F}_y = -0.31N$

12Hz	123V	8cm	0.4	46mW	79%
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d)

HF REFLECTION + TRANSMISSION FROM NOSE

$\bar{F}_x = -0.11N$
 $\bar{F}_y = -0.65N$

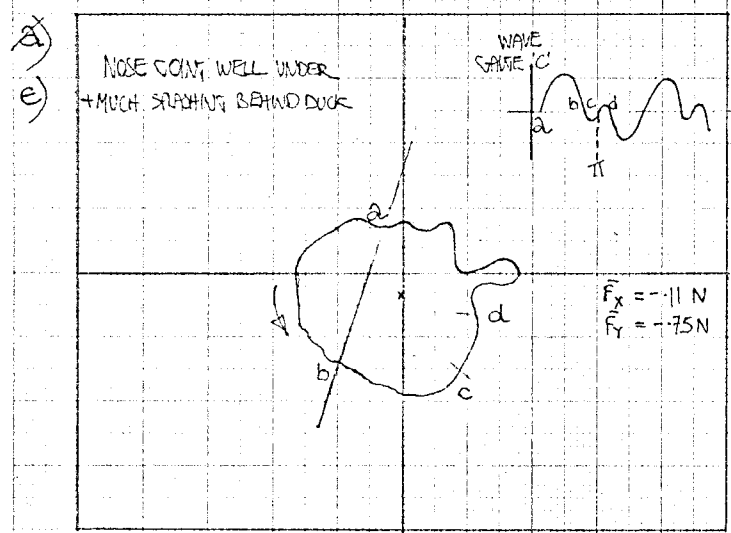
12Hz	187V	12cm	0.22	84mW	64%
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18: X+Y FORCE LISSAJOUS PLOTS FOR VARYING WAVE AMPLITUDES

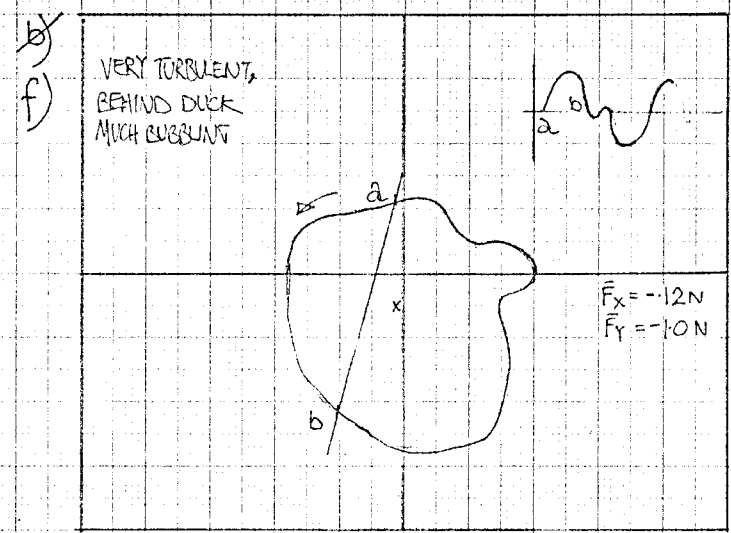
water position at duck shown by reference to WAVE CASE 'c' signal in X direction of duck

X : (\bar{F}_x, \bar{F}_y) 'CENTROID' of X+Y FORCES

3/7/75 (cont)
 +2/7/75

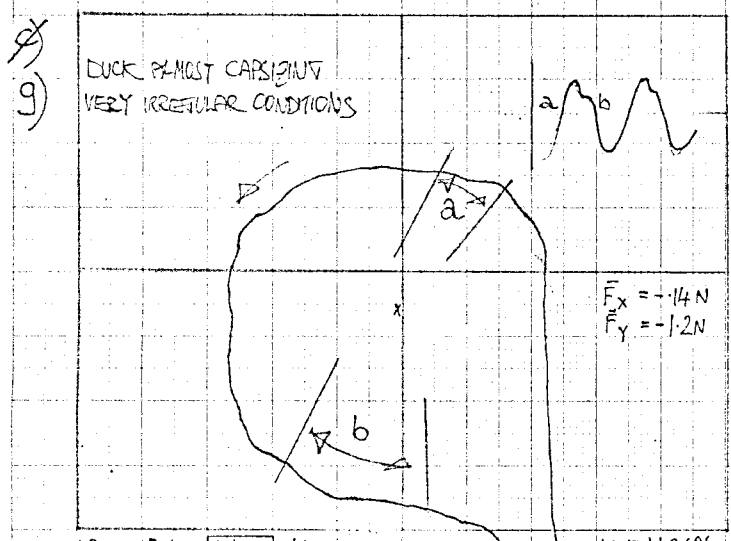


12HZ 2.6W 16CM .19 99mW 43%

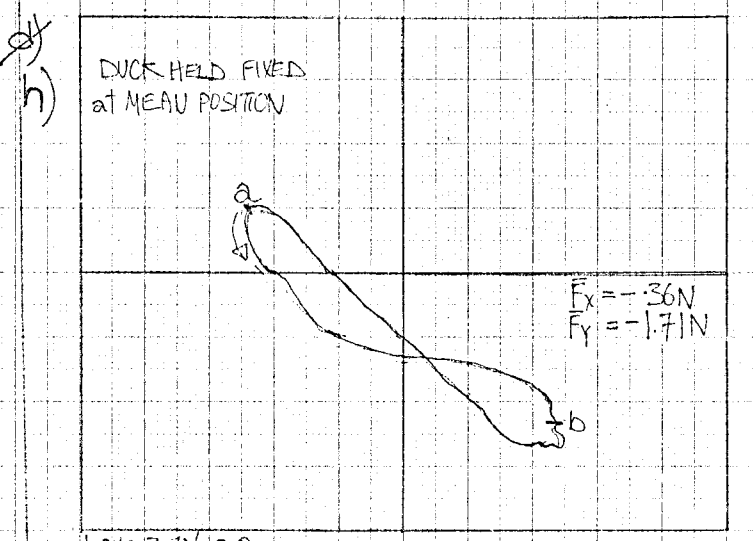


12HZ 3.5W 20CM .17 109mW 30%

B: CONTINUED

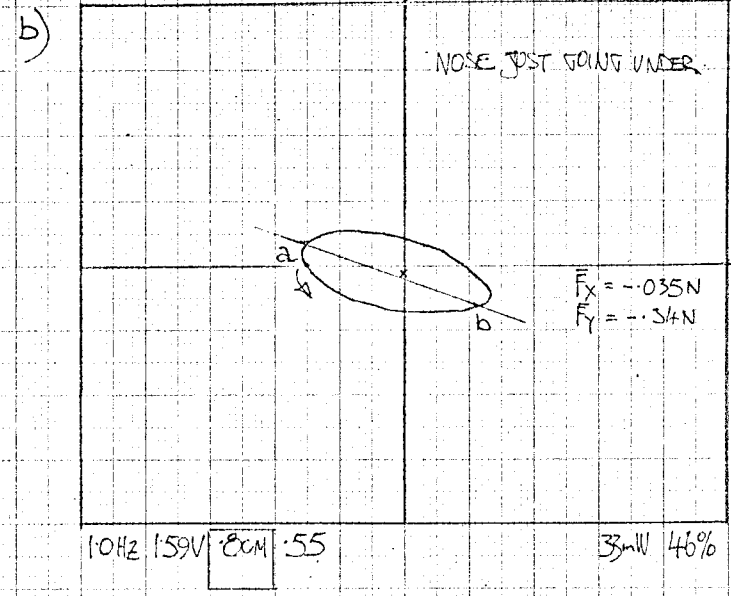
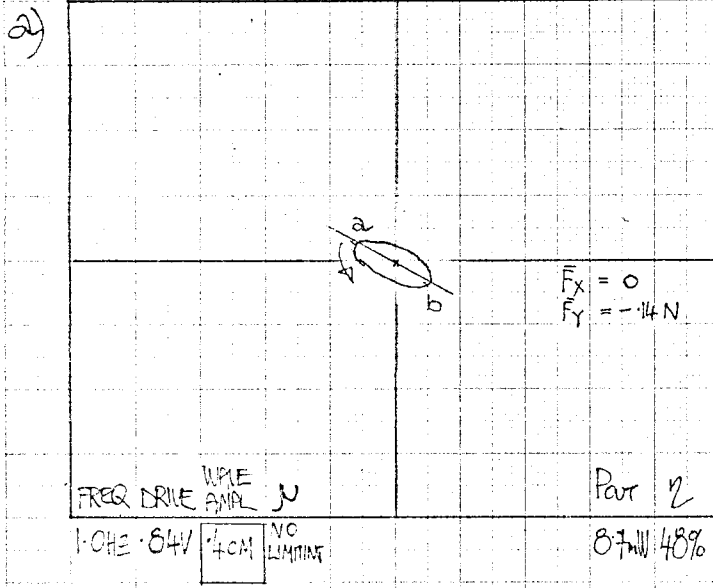


12HZ 3.7V 21CM .16 105mW 26%



12HZ 3.7V 20CM

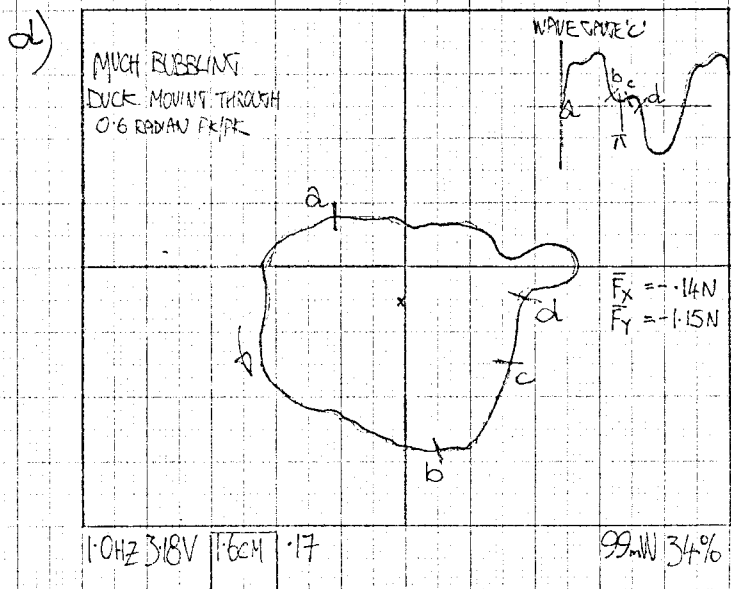
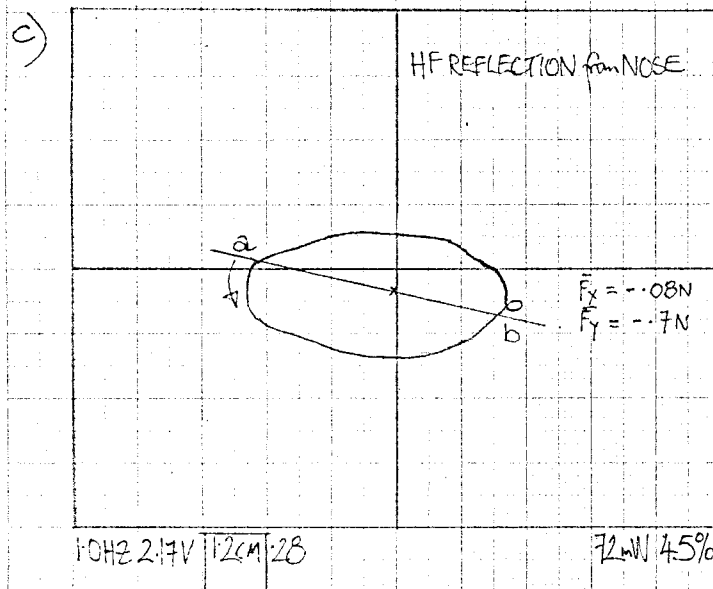
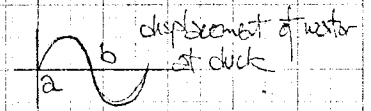
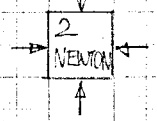
31795, 29775



D0011		BALLAST	DEPTH	SAMPLING
TOUSE 1	TOUSE 2	TOUSE 3	CM	N/CM / RAD/S
312	156	5.1	6.0	

1.0 Hz

$\mu = \frac{\text{DE ACTUAL TORQUE}}{\text{DEMAND TORQUE}}$



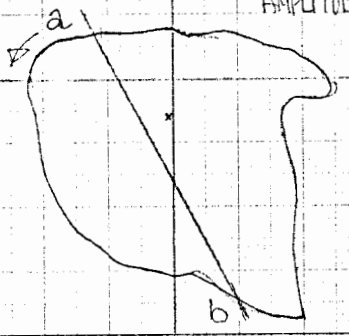
19: X+Y FORCE LISSAJOUS PLOTS
for VARYING WAVE AMPLITUDES

vector position of duck shown
by reference to WAVE GATE 'C' signal
n x diameter of duck
 $x = (\bar{F}_x, \bar{F}_y)$ CENTROID of X+Y FORCES

3775 (CONT)
+23775

e)

DUCK CAPSIZES
ABOVE THIS WAVE
AMPLITUDE



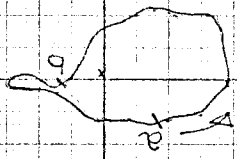
$$\bar{F}_x = -16N$$
$$\bar{F}_y = -1.25N$$

10HZ 375V 18CM 16

113-W 31%

19: CONTINUED

1.2 Hz 4.98V 1.6cm



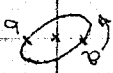
$F_x = +0.5N$
 $F_y = -1.8N$

HIGH PRESSURE BEHIND CYLINDER

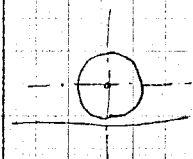
c)

1.2 Hz 1.1V 4cm

FRESH DRIVE FROM WAVE

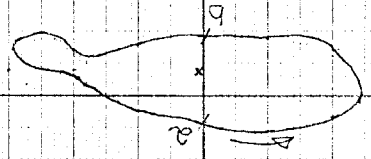


$F_x = 0$
 $F_y = +0.25N$



b)

1.2 Hz 99V 2.3cm

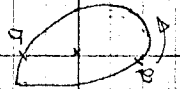


$F_x = +1.7N$
 $F_y = -4.72N$

LARGEST WAVES PURSUABLE WAVE TROUGH JUST REACHES 2-WAY DIAMETER CYLINDER

d)

1.2 Hz 2.05V 8cm



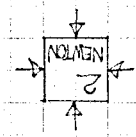
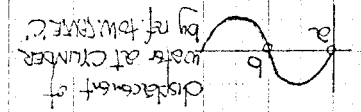
$F_x = +0.15N$
 $F_y = -0.55N$

HF ON TRANSMITTED WAVE AS WAVE CROS BEHIND TOP OF CYLINDER

b)

X: (F_x, F_y) CENTROID OF X+Y FORCES

20: X+Y FORCE LISA PLOTS for 10cm DIA METER NEUTRALLY BUOYANT CYLINDER



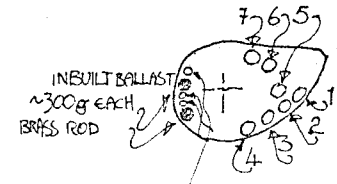
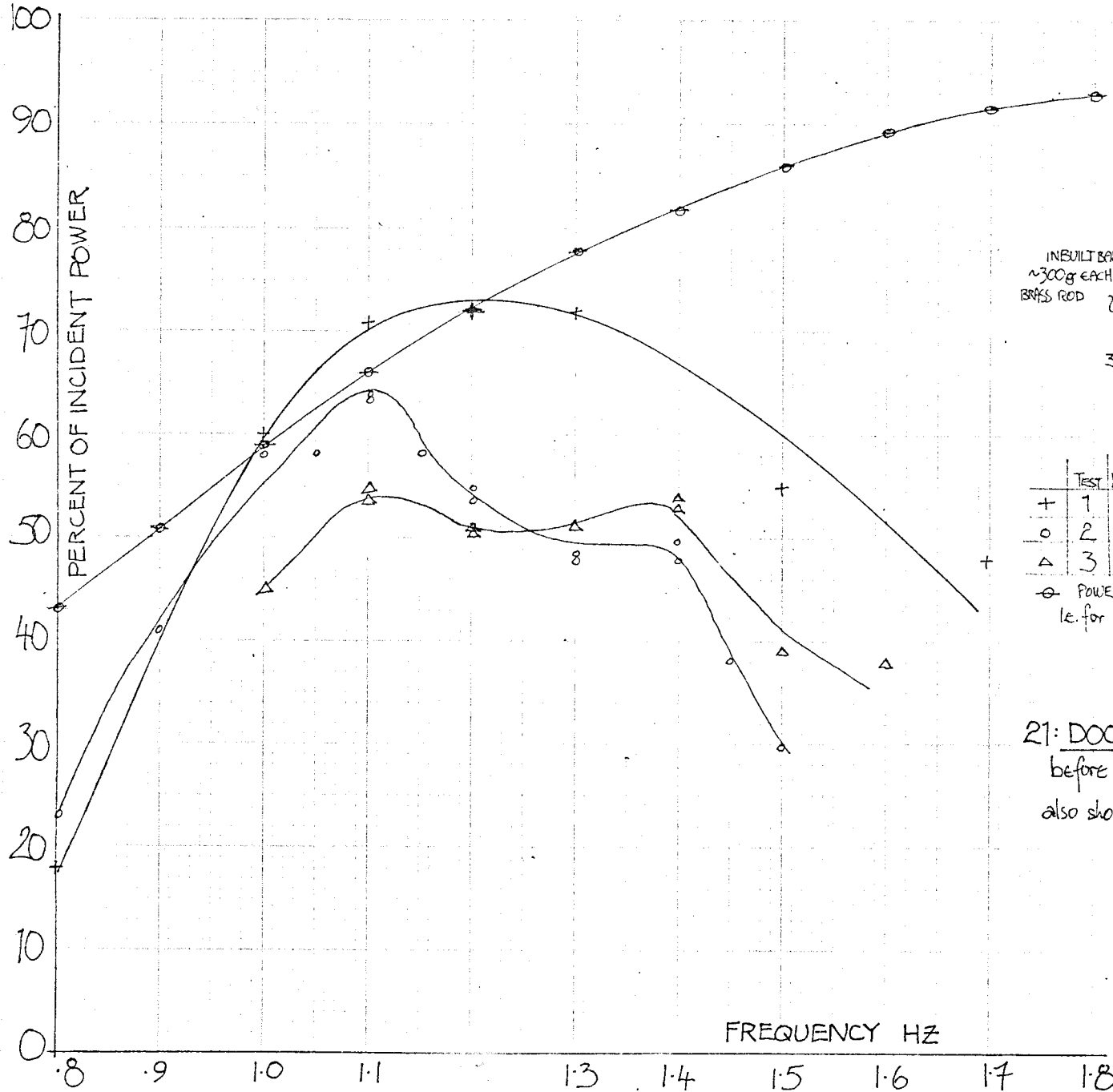
1.2 Hz RMS DEPTH: 5.1cm

J. 23/7/75

Very High Inertia Tests

Graph 21 shows the results of an attempt to extend low frequency performance. D0011 was rather beak heavy with full ballast in all tubes. There is not enough clearance between torque motor housing and rear skin for adjustable ballast tubes so we built D0012 with semi-permanent ballast in its rear section. Restoring trim with forward ballast would produce a large inertia. It was a lousy idea. The heavier ballastings would have sunk the whole duck string.

28/7/15



3 x 29'3cm x 1.4 DA STEEL RODS ADDED FOR TESTS 2+3

TEST	BALLAST	TRAPEZ	TUBE	DAMPING RINGS	NCH	HUB DEPTH CM
+	1567	3124	3127	3		5.7
o	3123	3124	2915 2326 3127	STEEL RODS: 2.5		6.0
Δ	"	"	"	"	60	6.0

o POWER "AVAILABLE" ABOVE 6CM : $[1 - \exp(-0.07 \times \text{depth} / T^2 g)]$
 i.e. for 6CM HUB DEPTH

21: DOO12 EFFICIENCY/FREQUENCY
 before and after REAR BALLASTING MODIFICATION
 also showing POWER IN 6CM DEPTH OF WINDOW

More Wave Efficiency Size and Period Experiments25.8.1975

Graphs 22a to h provide data on D0015 fitted with the new more powerful V25Y6 torque motors. They extend the tests shown in Graph 13 using D0011.

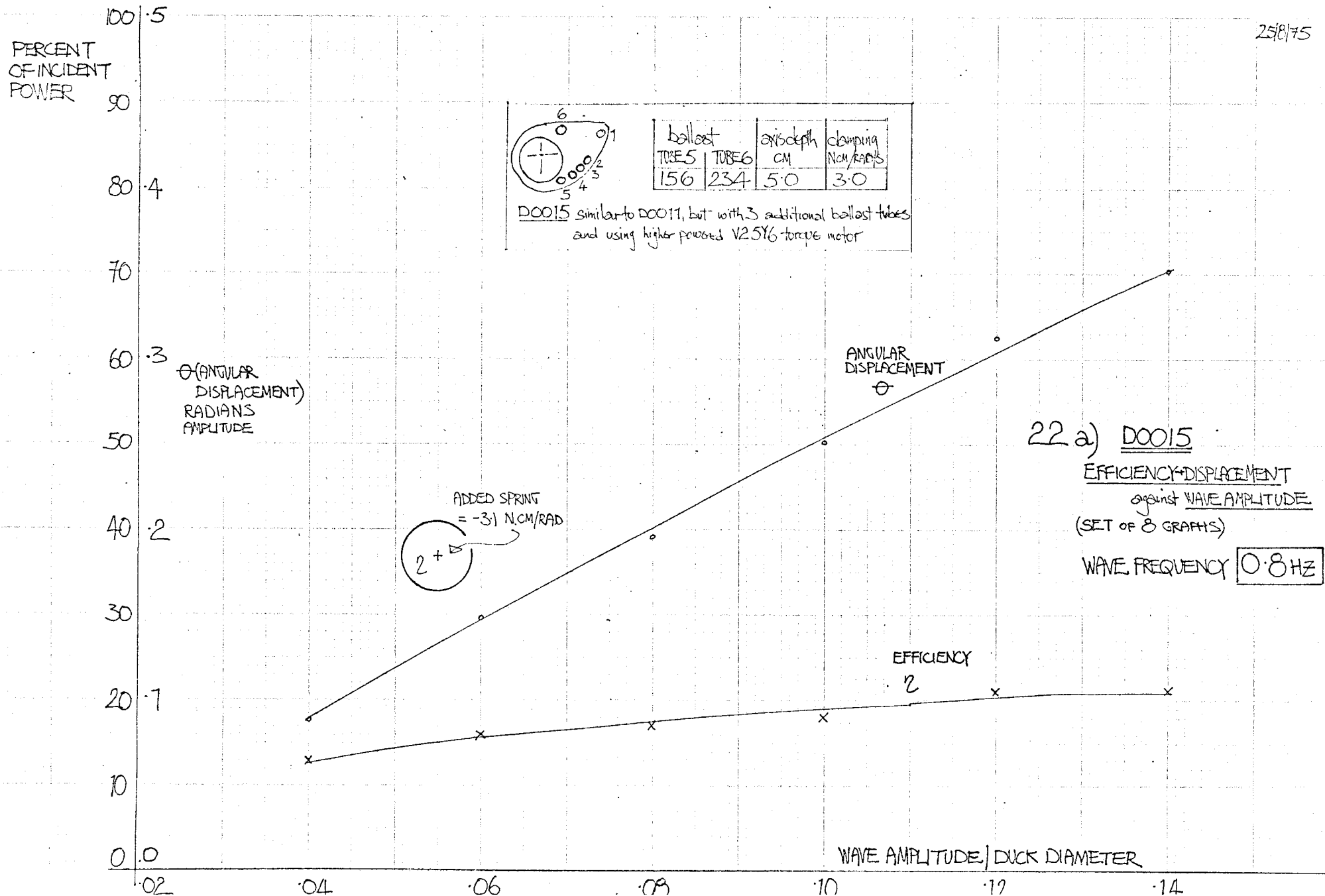
We were trying to go for a broad efficiency band rather than extreme efficiency at one point. The duck was ballasted for best performance at 1.2 Hz. Each curve shows efficiency against wave amplitude up to the biggest stable waves we can produce with the present wave maker. For a hundred to one scale this would represent trough crest heights of about 3 metres and power density levels of 90 kilowatts per metre. The fall at the low amplitude end, where model power is only 2 mw is probably caused by bearing friction. The fall at the high end is probably caused by non-linearity. Also shown is the duck nod amplitude. It is linear with wave size to the point where efficiency begins to fall.

The new torque motors allow us to play with smart hydraulics. The spot points show the results of adjusting spring rate and these are combined with stupid hydraulic data in the efficiency/frequency curves on Graph 23. Dramatic improvements to performance are produced at high frequencies by additional positive spring rate. For a small part of the cycle the system is putting power into the water but it is amply repaid.

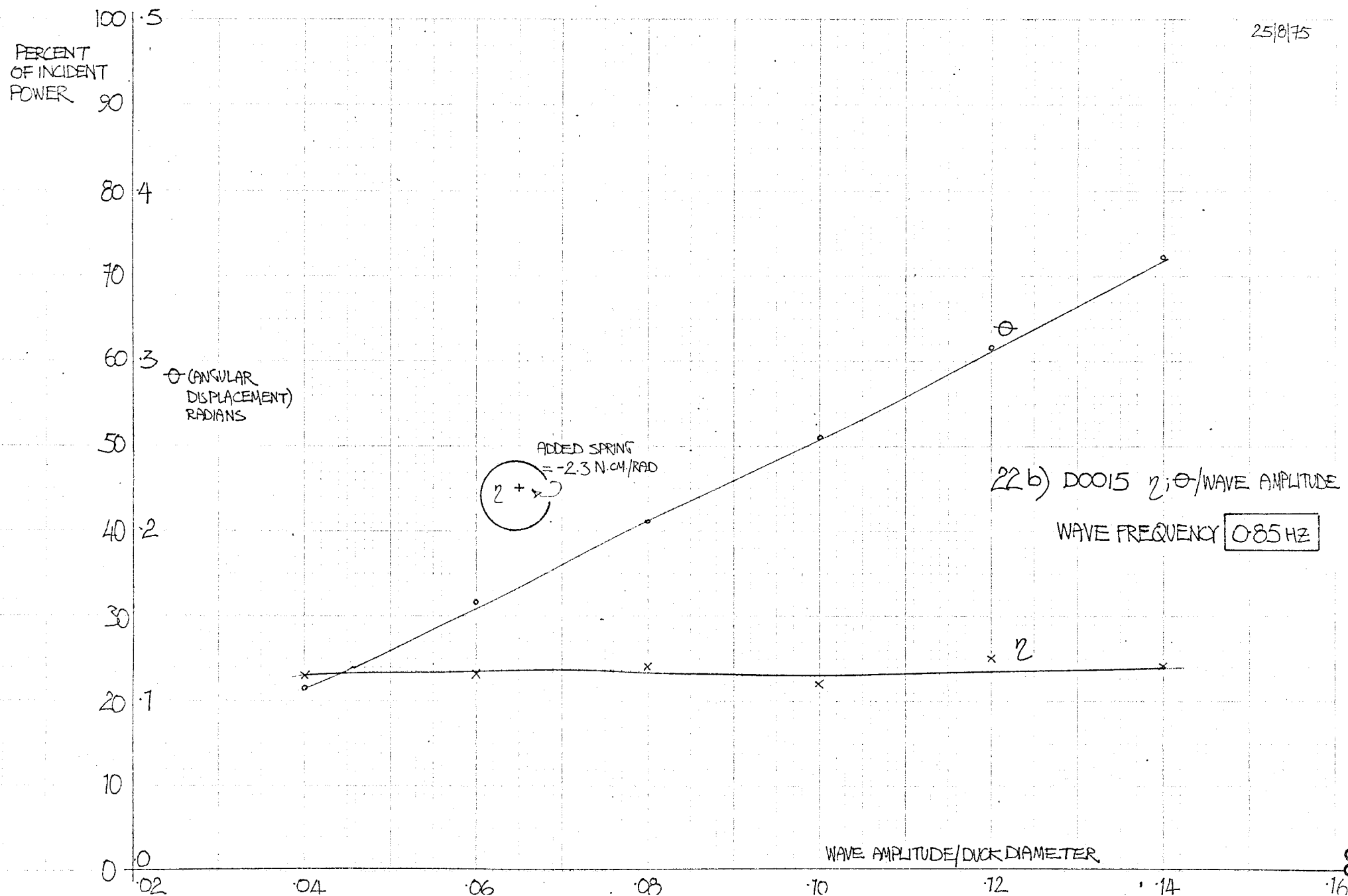
Hydraulics people say that anything we can do with transistors they can do slower but more powerfully. We believe that this field is very promising.

This series of tests was the first outing for D0015. We hope to do a bit better with more experience. The Mollison/Buneman programme should show whether a broad 80% is better than the peaky 95% of D0011. Bigger wave makers are going to be built.

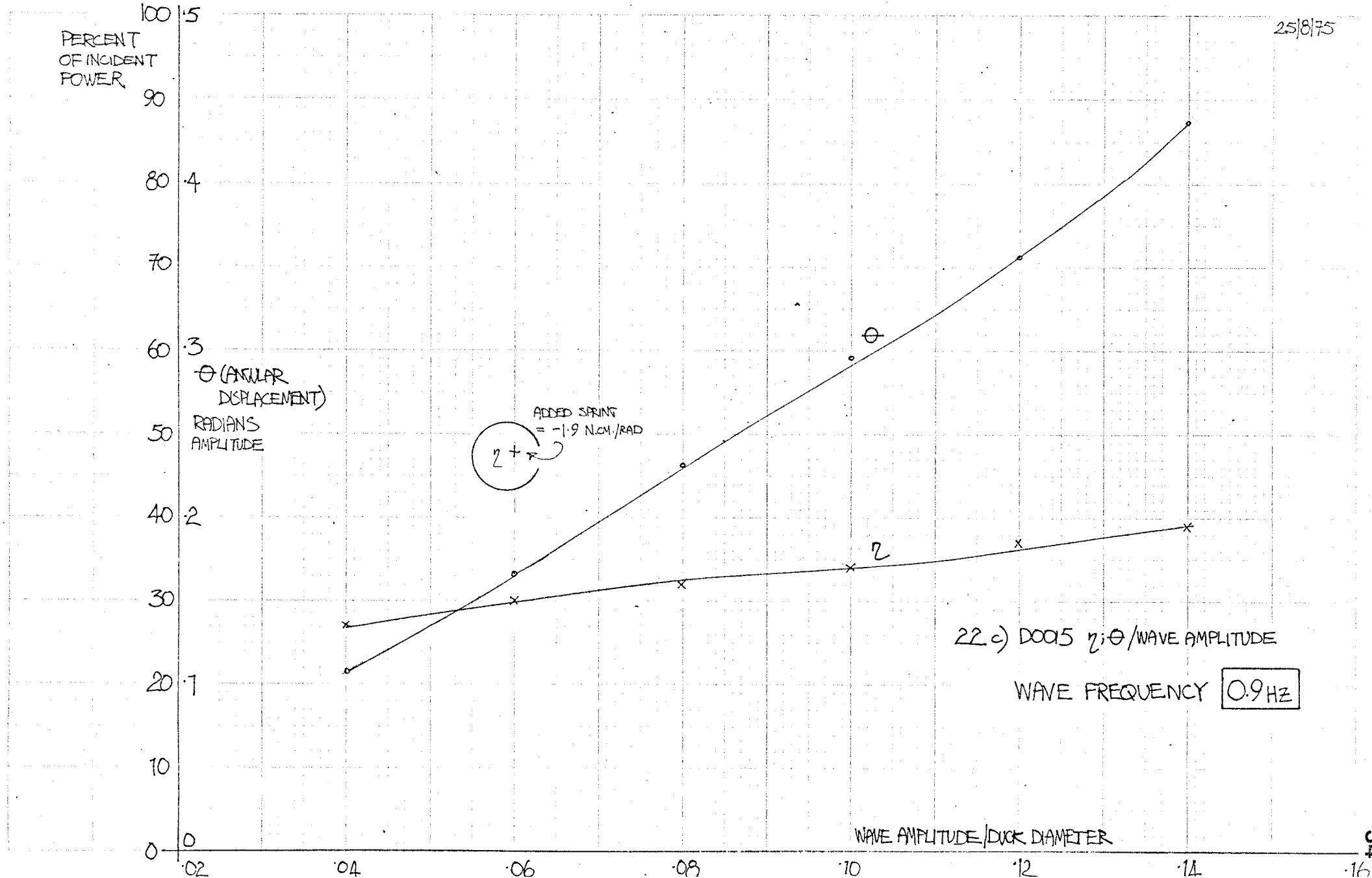
25/8/75



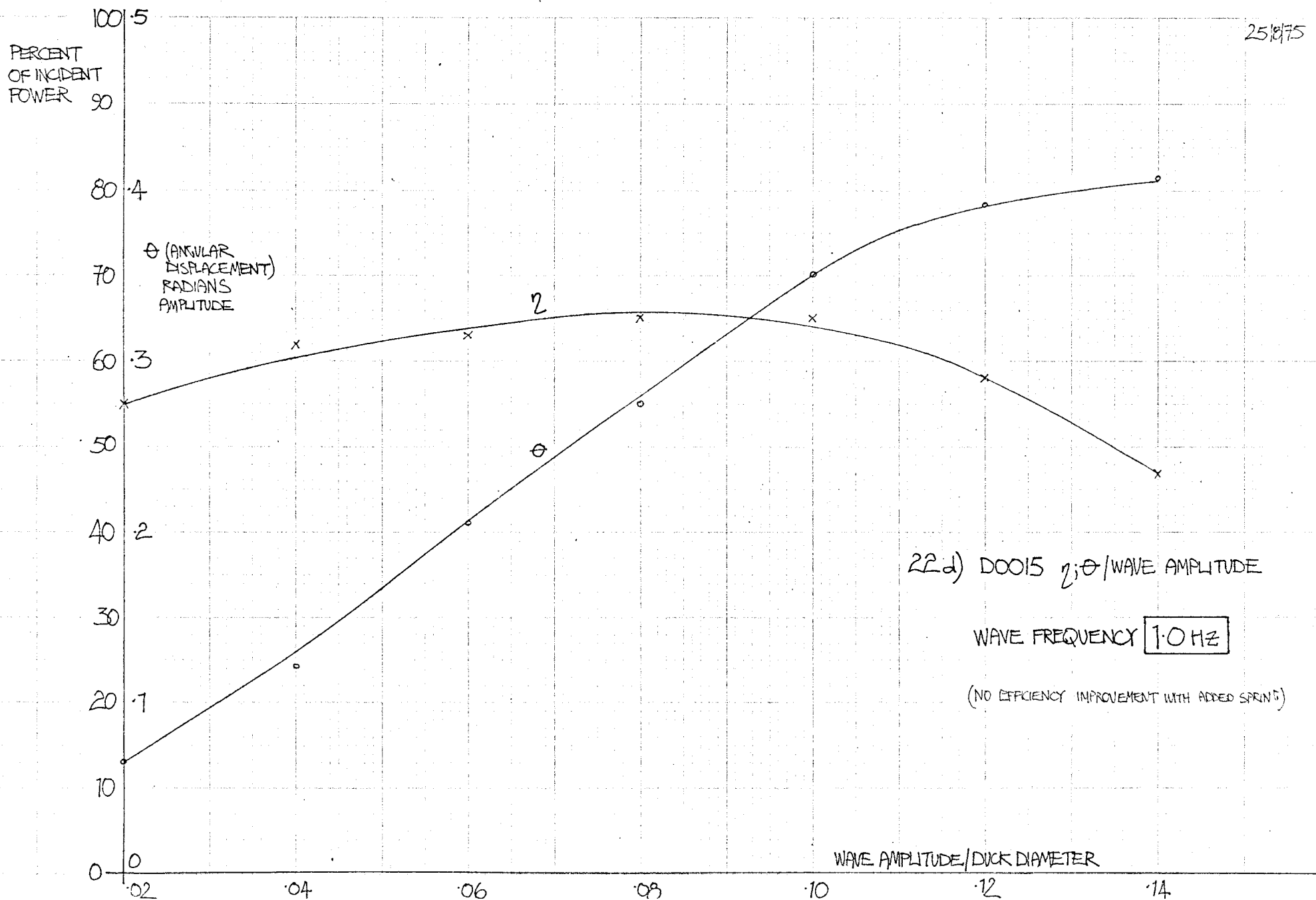
25/8/75



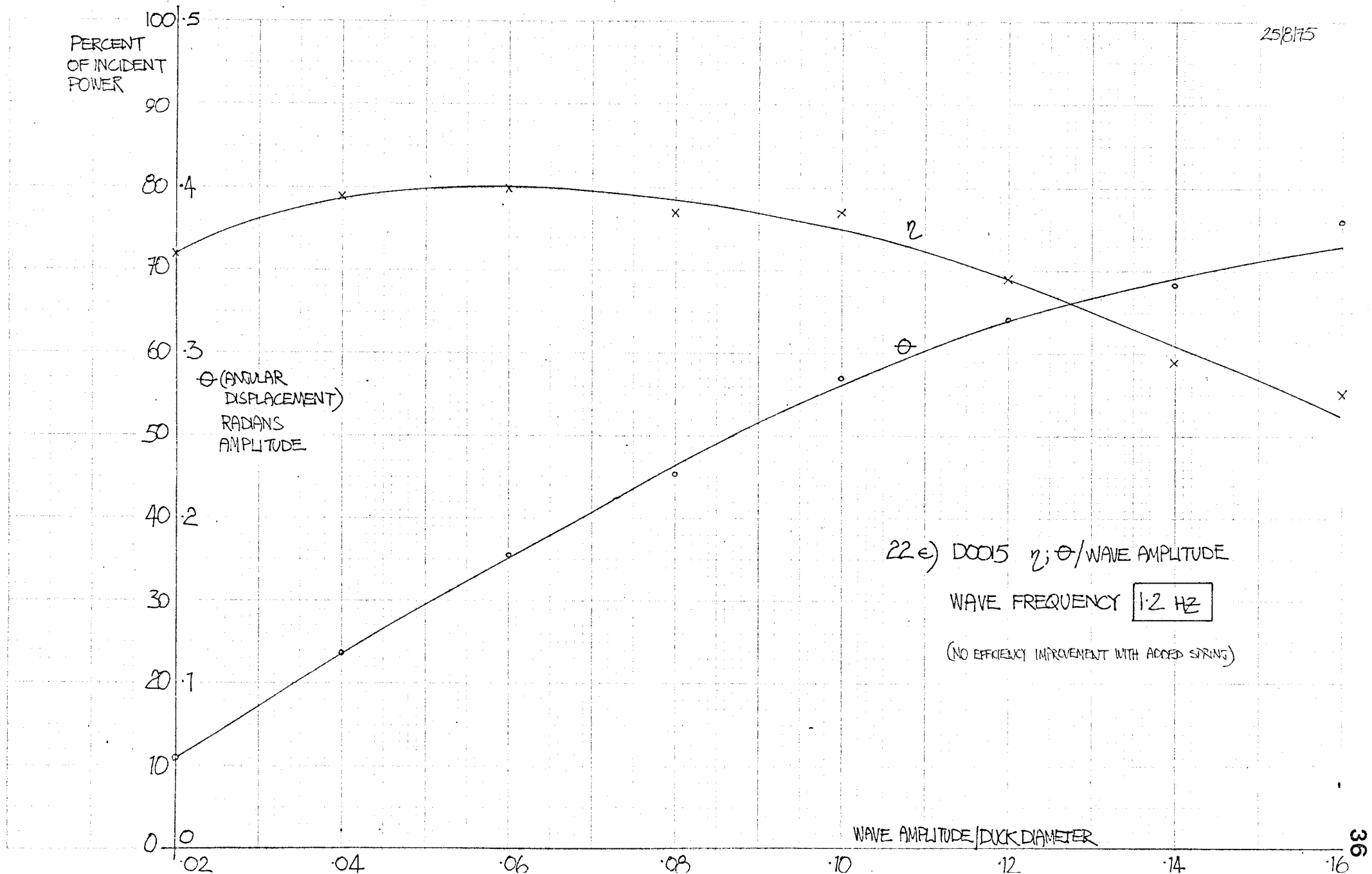
25/01/75



25/875



25/8/75



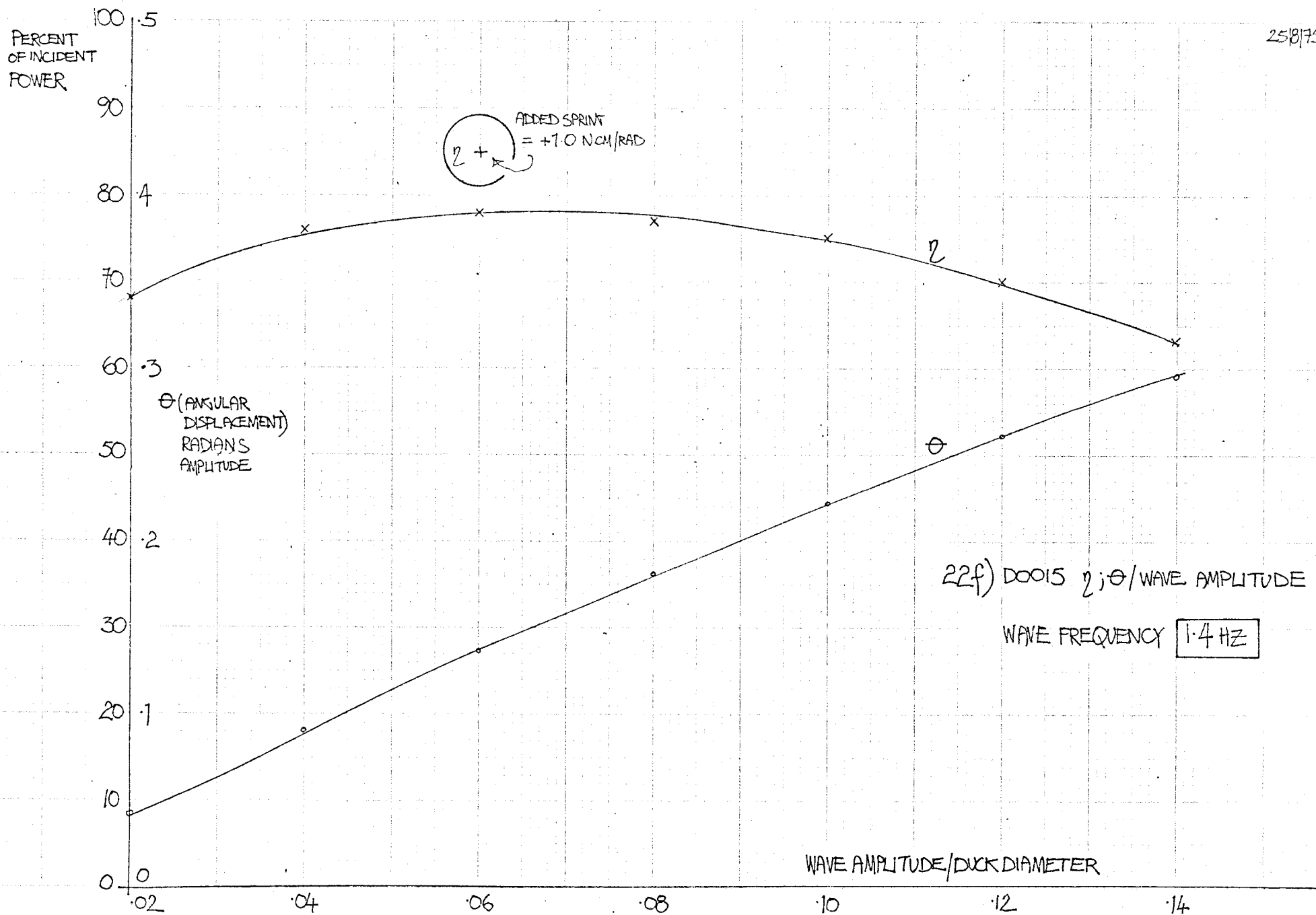
θ (ANGULAR DISPLACEMENT)
RADIAN'S AMPLITUDE

22 e) D0015 η ; θ /WAVE AMPLITUDE

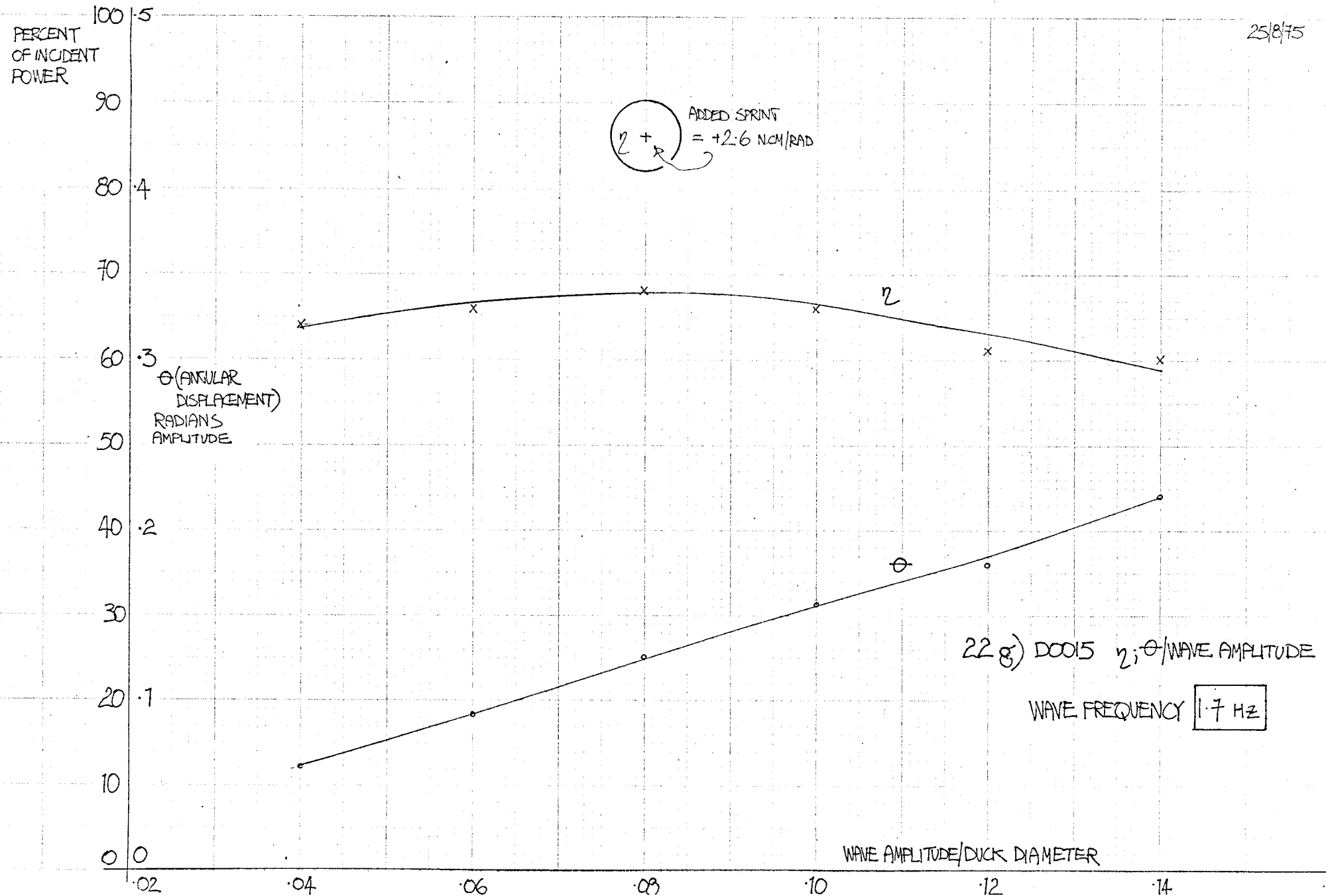
WAVE FREQUENCY 1.2 HZ

(NO EFFICIENCY IMPROVEMENT WITH ADDED SPRING)

25/8/75



25/8/75

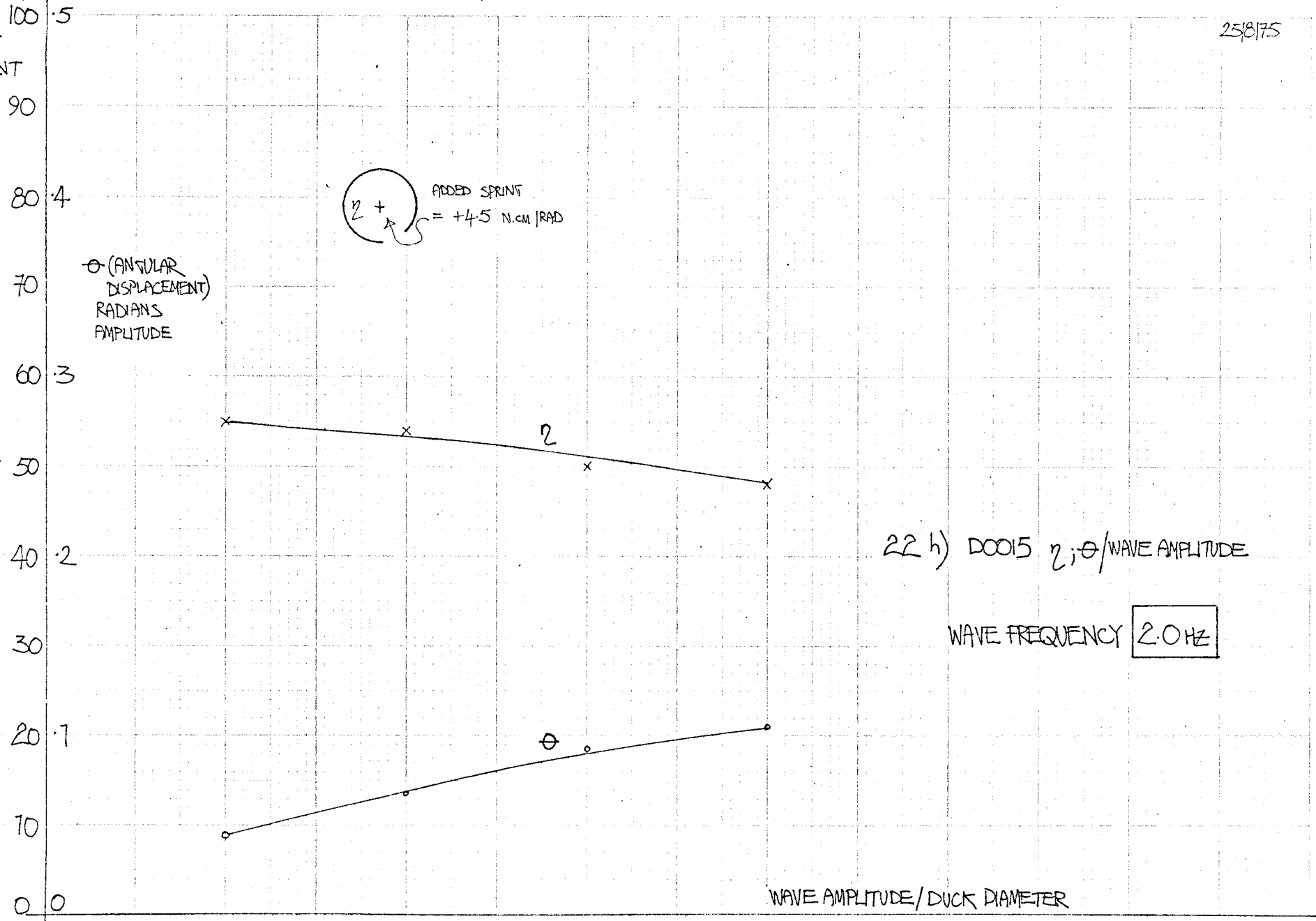


PERCENT OF INCIDENT POWER

100 5
90
80 4
70
60 3
40 2
30
20 1
10
0 0

θ (ANGULAR DISPLACEMENT) RADIANS AMPLITUDE

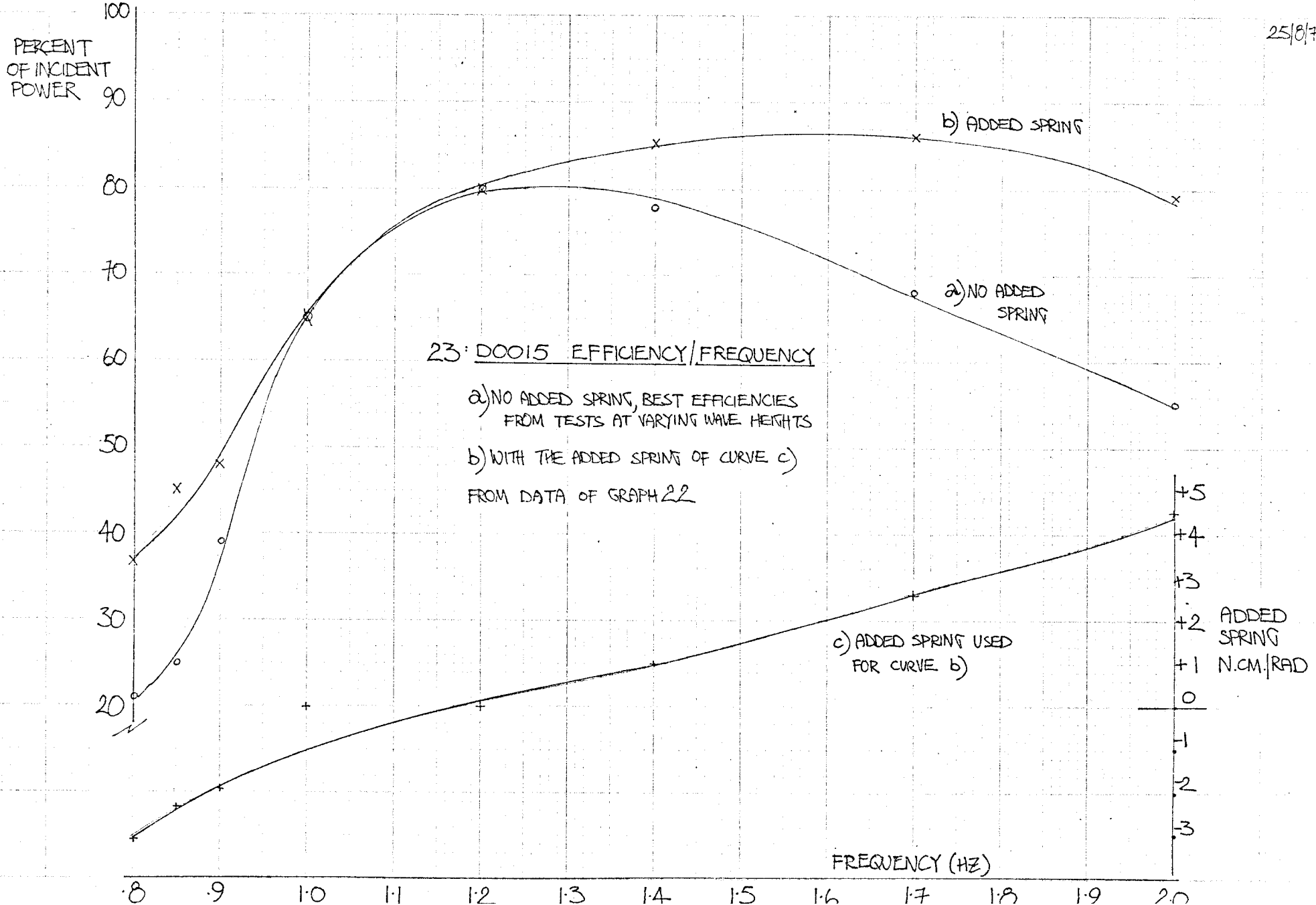
$\zeta +$ ADDED SPRINT = +4.5 N.CM/RAD



22 h) DOO15 $\zeta; \theta$ /WAVE AMPLITUDE

WAVE FREQUENCY 2.0 HZ

25/8/75



Smart Hydraulics Experiments

Graphs 24a, b and c

The previous note showed the results of negative and positive spring rate on widening the absorption band. For these experiments a duck was optimised for performance at 1.4 Hz. It was then tested across the band of frequencies and both damping and spring rate were adjusted for each frequency. The dial settings were noted. The spring rate control is taken from the integral of the duck velocity signal. A variable amount of it can be added to or subtracted from the force signal demand. The torque command signal is computed by an instrument we call the 'dynabox'. The transfer function of this box was measured at each frequency and knob setting and the results shown in Graphs 24a and b. It should be clearly understood that this is a "pink box". A human is twiddling knobs for each frequency and the results would not apply to a different frequency. The technique could only be used to pick out the most useful part of the energy spectrum. Note that the amplitude response rises on either side of 1.4 Hz which was the original optimising frequency. The phase lags with rising frequency and leads with falling frequency.

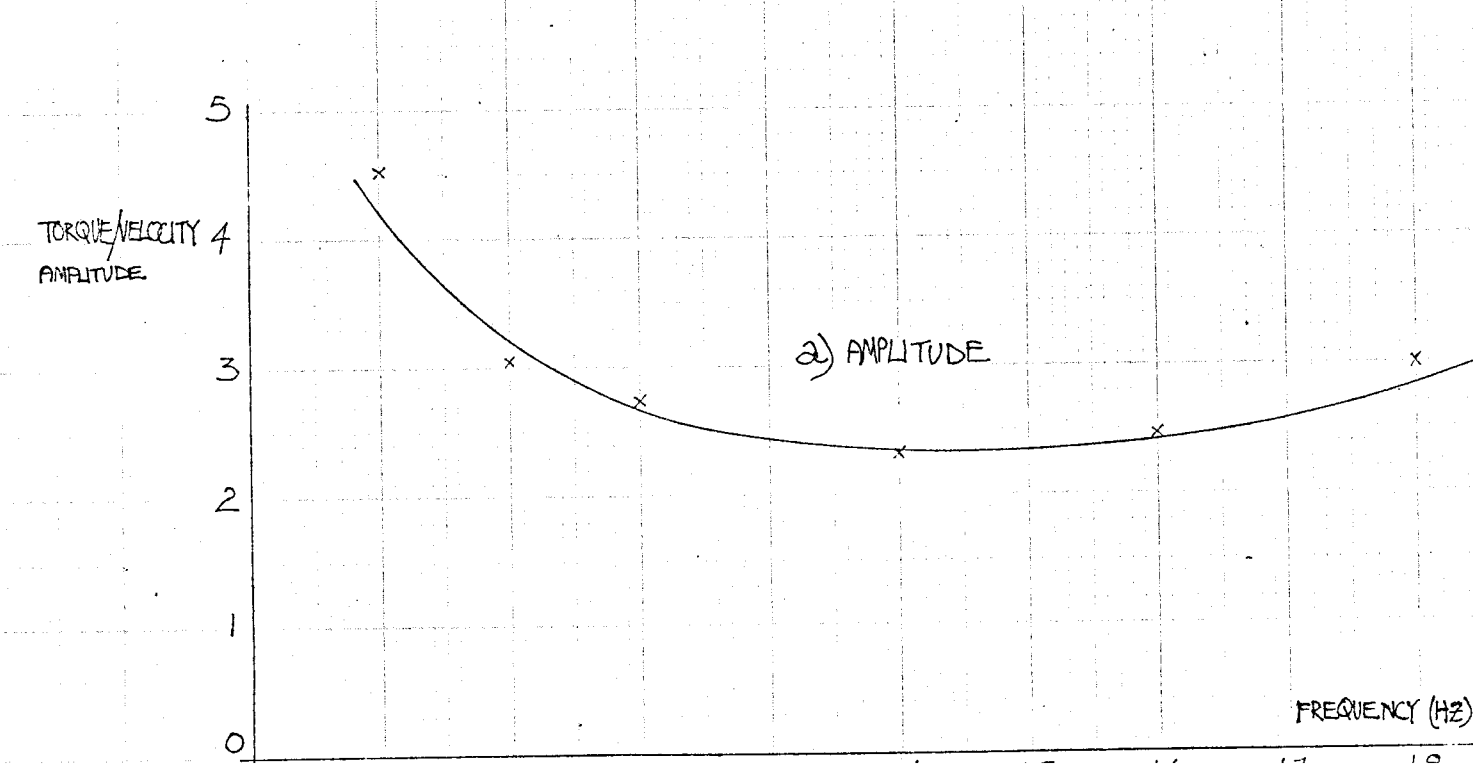
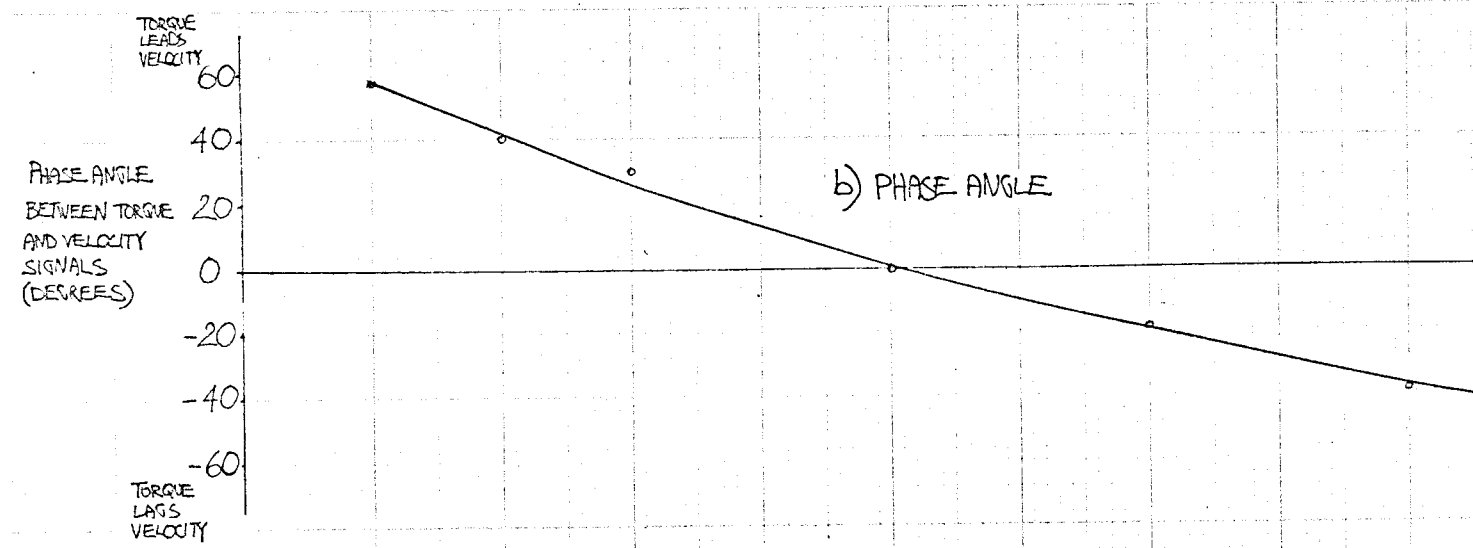
It is widely believed that because of previous arrangements made by Dr. Bode a knowledge of the amplitude response of a network is sufficient to deduce its phase response, and that no network can have a rising amplitude and lagging phase as the frequency rises, or a rising and leading one as it falls. Examination of books on network synthesis showed that the prohibition applied only to 'minimum phase networks'. All passive RCL networks are minimum phase networks but some active ones with more than one signal pathway escape the ban.

Our first attempt was an analogue model of the mass/spring/dashpot with values the same as the duck. The force and velocity connections were used as the feedback network for an operational amplifier. It produced a perfect amplitude response with exactly the wrong phase. One point to Dr. Bode. For our second attempt we replotted 24a and b in Cartesian form on 24c. The result was particularly exciting. Waves seemed to be doing something sensible at last.

The normal connection for an operational amplifier used as an integrator gives a Nyquist plot along the positive imaginary axis with a high gain at low frequencies and zero output at infinite frequency. Each doubling of frequency halves the distance to the origin. Similarly the response of the normal differentiator is along the negative imaginary axis with zero frequency at the origin and a doubling of the response for each doubling of frequency. The Nyquist plot of the old damping alone is just a point on the positive real axis. The right combination of all three is all that is needed for the smart hydraulics black box which works for different frequencies at the same time.

Practical integrators need a leak resistor to stop them drifting. Practical differentiators require a series resistor to limit the noise output at high frequencies. The result of this is to turn the line of 24c into an arc of a large circle which crosses the negative real axis and has a phase such as to produce oscillations when connected to a duck. This tendency is balanced by the falling response of the duck. The theoretically perfect settings are highly strung. Although stable on their own they throb excitingly when their dynamics are changed by the touch of a hand. The non-linearity of the duck's own spring rate curve leads to problems in heavy seas. The duck sometimes rears up out of the water and holds itself poised before plunging down to below its normal position. The torque/angle indicator diagram looks rather peculiar. We expected that the normal ellipse with axes along x and y would just be tilted to one side or the other. This happens for a small amount of smartness. But the best settings give shapes so reminiscent of transformer hysteresis curves that there may be some magnetic artefact. (Mark III dynamometers for the floating string will use high speed motors geared down to duck speed mounted on torque sensors.)

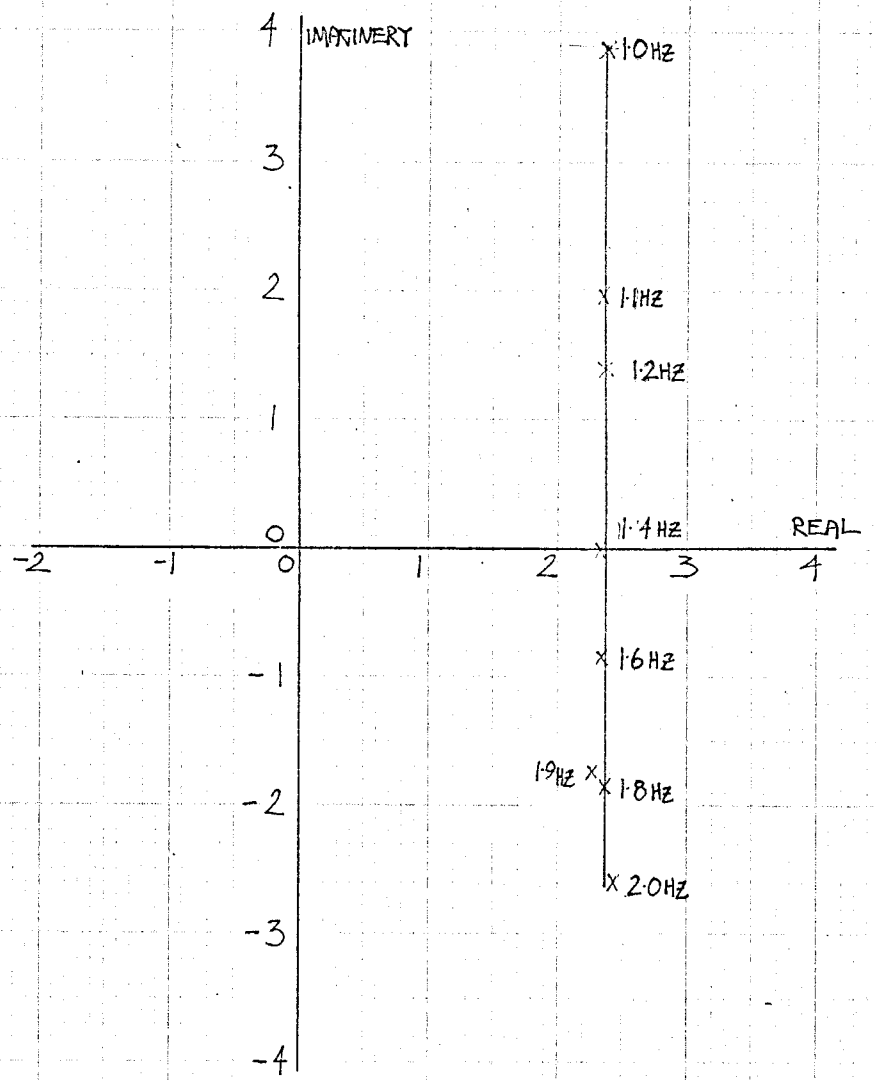
27/8/75



boldest TUBES	axis depth cm	damping Nm/rad/s
156g	5.2	2.4

2.4: TORQUE/VELOCITY

a) AMPLITUDE; b) PHASE ANGLE AGAINST FREQUENCY, for DOO15 optimised at 1.4 Hz without added spring and then at frequencies shown using added spring.



24: TORQUE/VELOCITY

c) CARTESIAN PLOT of DATA in 25a) b)

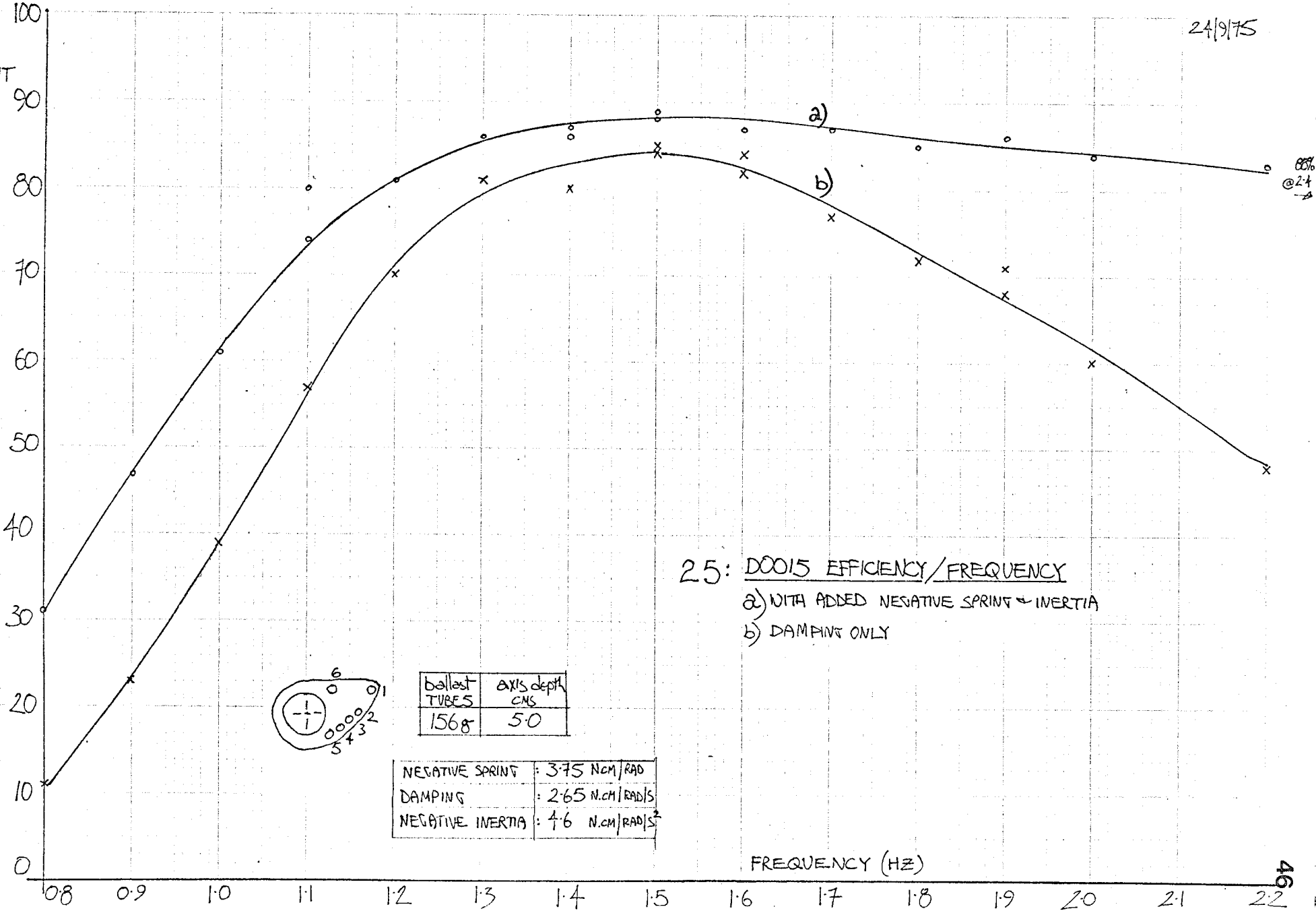
Smart Hydraulics ExperimentsGraph 25

Graph 25 shows a comparison between damping only and damping plus negative spring and inertia.

Smart hydraulics helps both sides of the normal optimum but does best at holding up the high frequency performance of a duck at low power levels. There still remains some sort of barrier for wave lengths of more than fifteen duck diameters.

24/9/75

PERCENT OF INCIDENT POWER



25: DOO15 EFFICIENCY/FREQUENCY
 a) WITH ADDED NEGATIVE SPRING & INERTIA
 b) DAMPING ONLY