A PROBLEM IN AFT END VIBRATION :
Interim Report on a Case Study ø

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REPORT NAV/ARCH 74/7 November 1974

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ø Notes for a Lecture to Royal Institution of Naval Architects, Australian Branch, 13th November, 1974.

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I. INTRODUCTION

1.1 The Problem : Cause and Cure

A number of ships developed in different parts of the world in recent years have been found to suffer more or less severe hydrodynamically induced aft-end vibrations and noise. These have presented owners with problems ranging from internal and external structural damage to excessive vibration and/or noise in accommodation* and other manned spaces. The vibration input has usually been hydrodynamic, arising from propeller wake interaction effects. Increased wake fractions have resulted from the blunter aft lines associated with higher block coefficients, more profitable space utilisation and rear movement of the longitudinal centre of buoyancy even when block coefficients were moderate. On the propulsion side, there has been a trend towards single screw installations for obvious reasons and the correspondingly higher powers have been applied increasingly through controllable pitch propellers where limitations of geometry and spindle torque have made it more difficult for the propeller designer to revert to the use of palliatives such as rake and skew to reduce the severity of the effects of operation in non-uniform wakes.

The effect of wake non-uniformity is to produce cyclic changes in flow angle or incidence and this has been found to lead to intermittent cavitation and consequent propagation of strong pressure pulses into the water - thus providing a vibration input to the structure separate from the well understood phenomenon of cyclic variation of thrust and torque. A scientific description of the cavitation phenomenon** is given in Appendix I. The most helpful account for ship purposes is P.Eisenberg's Cavitation Dictionary (1.1). If the water in the wake immediately behind the stern post is sufficiently "dead", the propeller tip vortices may attach themselves temporarily to the counter as the blade tips pass in close proximity to the latter. This so-called propeller-hull-vortex interaction, discovered by E. Huse of the Skipsmodell-Tanken, Trondheim, manifests itself as a series of hammer blows occuring at blade passing frequency.

Although a degree of cavitation is probably inevitable in the present state of propeller technology, the worst effects in respect of vibration input can evidently be avoided by paying attention to the nature of ships' wakes at the design stage of their development. Finer waterline endings will lead to less intensive wake peaks and secondary flow components thus reducing the above mentioned incidence changes on the propeller blades during each revolution. Also, the likelihood of propeller-hull-vortex-cavitation is greatly reduced when there is significant convection velocity in the propeller arch.

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* Ships' houses had been moving aft somewhat earlier and before thresholds of crew tolerance of noise/vibration had tended to approach those of land based work situations.

** A useful summary of "The harmful effects of cavitation on marine propellers" was recently given to the Australian Branch of R.I.N.A. by L. Hawdon of Stone Manganese Ltd. This is to be published.

Because the problem has only emerged in recent years* a good many ships were built with incomplete model tests so that the vibration problem has had to be treated after its appearance during the first trial voyage. The problem then becomes one of finding the most convenient modification of the ship to make it fully operational. A variety of flow deflectors have been used with success to accelerate the wake enough to prevent harmful cavity pulsations. The corresponding power loss is tolerated if the ship is able to make a sufficiently increased number of voyages per annum, if hard-lying allowances can be discontinued or if there is some other adequate trade-off. The present paper is concerned with a case which defied cure by means which had worked for similarly afflicted ships in Europe and details of the case study are given in Section II. Development of a local solution is described in Section III.

1.2 Extent of the Problem : Overseas and in Australia

A comprehensive account of overseas experience was given by Vossnack & Voogd (1.3) to the Second Lips Propeller Symposium in May 1973. This includes a historical development of stern frame design for 17 ships. Some or all of the causative factors listed in 1.1 above were present and flow deflectors, half-tunnels and other cures are described. Interested readers should also consult the very thorough study made on models, full scale and theoretically in respect of a family of 230,000 tdw tankers by a combined team of research staff from S.S.P.A. (Sweden) and Norske Veritas (1.4, 1.5). These appear to be the first ships for which the remarkable "Hydrocalc" programme of Det Norske Veritas was used to predict the amplitudes of pressure fluctuation experienced by a hull when a given propeller turns in a wakefield of given inhomogeneity. The role of this computer programme in the redesign process will be explained in Section II.

A number of vessels have recently entered service on the Australian coastal trade with manifestation of a greater or lesser extent of aft end vibration and/or noise. Some details of these are listed below.

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* This statement has to be qualified, like all others in this complex field : the phenomenon of propeller/wake induced aft end vibration had in fact been observed on some Great Lake ore carriers and its cure by means of horizontal flow deflecting fins was published (Ref. 1.2) in 1952.

Type	RO/RO	RO/RO	Unit Load	Tanker
Length, B.P.	433'	300'	361'	536'
Breadth, mld	74'0"	58'6"	62'	82'0"
Depth, mld	48'6"	39'9"	36'	42'3"
Load Draught	24'0"	17'0"	23'	32'2"
Load Displacement, tons	13170		10300	32411
Load Block coefficient	0.60		0.70	0.80
Machinery	Single screw geared diesel	Single screw geared diesel	Single screw diesel	Single screw geared diesel
Propeller	Controllable Pitch	Controllable Pitch	Controllable Pitch	Controllable Pitch
B.H.P.	16,000	5,280	6,000	16,000
Service Speed, knots	18	15	15.3/4	15

All these 7 ships, sister ships included, shared the common factor of relatively high power being transmitted through a single shaft.

1.3 The Place of Experiment and Theory

In this area, as in a number of others, professional judgments have usually to be made on the basis of limited evidence. Designers tend to rely on specialists who may themselves be severely handicapped when the superposition of a number of small changes produces a qualitatively different situation. The most important information is undoubtedly that gathered from ships at sea. However, this is the most difficult to obtain - quantitatively - from the busy ship operator. Designers therefore derive most of their information from model tests. This is satisfactory if the experimenter is able to model all relevant physical effects and if model/full scale correlation is understood properly. By and large it could be said that this works well enough and the many volumes of proceedings of the International Towing Tank Conference indicate the attention given, particularly to the correlation problem, by the staffs of the major ship model test laboratories in different parts of the world.

The rash of aft-end vibration problems of recent years has highlighted the need for more sophisticated model experimentation than had been used earlier. Resistance, propulsion, manoeuvring and seakeeping tests, until recently the normal range of testing - often made after the design had been well developed and indeed sometimes fixed - are no longer adequate to assure the designer that his ship will be satisfactory in a hydrodynamic sense. Propeller designers also should no longer be satisfied to make recommendations without a fair knowledge of the wake field which provides the operational environment of their propellers. Prediction of wakefield has become imperative.

It is worth recalling that successful modelling is not only based on correct geometric scaling but also, ideally, on the achievement of flow similarity in all respects. This would demand identity for model and full scale of a number of well known ratios such as the Froude number (the speed/length ratio $V/\sqrt{L}$ would serve for this), the Reynolds number, the cavitation number (and hence, indirectly, the atmospheric pressure), the percentage of air in the water, etc. (1.1). Fortunately the coefficients of interest are slowly varying functions of some of the above ratios so that it has been possible to make predictions from model experiments for which full flow similarity has not been provided. In fact, conventional towing tank experiments have been arranged to model ships' speed/√length ratios correctly while all others remained uncontrolled. This has led to the conventional wisdom among naval architects that correct Froude number scaling is a prerequisite of *all* meaningful experimentation. However, there are many quantities of interest which are not much affected by the distortions of the free surface of the order of magnitude found for merchant ships. The so-called zero-Froude number approximation then comes into its own. Modern research into both deep and shallow water manoeuvrability and course stability has been very effective in this respect and the many important results obtained are used successfully at practical speed/√length ratios even though they became fully valid only in the limit as that ratio tends to zero!

The details of the flow into the aperture of a ship are also not much dependent on Froude number while the latter has the low values associated with merchant ship operation. If it is intended to study wake-induced propeller cavitation and hull pressure fluctuations, the provision of a correct ambient pressure, and thus cavitation number, may be much more significant than correct modelling of free surface or Froude number effects. This was recognised by the highly regarded Swedish ship model laboratory Statens Skeppsprovningsanstalt (S.S.P.A.), Goteborg by an investment of 6 m. Kroners in a very large cavitation tunnel facility, see Reference (1.6), with closed working section in which the free surface was represented by a plane wooden board. Models up to 26 ft in length can be accommodated and comparative studies of nominal wake fractions - i.e. in the absence of a propeller - determined on the same model in the cavitation tunnel and also in the S.S.P.A. towing tank showed little difference.

Similar reasoning shows that wind tunnel tests on a double model, viz model of a ship complete with "image" in waterplane, should also yield good information on nominal wake fractions. The wind tunnel is a much more convenient - and generally cheaper - tool for examining boundary layers and wakes than a towing tank or a water tunnel. It is therefore particularly suitable for those designers who only learn of their (hydrodynamically induced) vibration problem during a trial voyage and therefore need to work up ameliorative measures. It is understood that there can be no representation of cavitation number effects - as is also the case for conventional towing tanks - or of Froude number effects - as is also the case for conventional cavitation tunnels. On the other hand, Reynolds number can generally be modelled rather better in a wind tunnel than in the above mentioned water facilities. Although the kinematic viscosity of air is approximately 13 times that of water, the scale speed of models in normal commercial towing tanks is low, so that it is generally possible to exceed it by the factor 13 times (length of towing tank model/length of wind tunnel model) which is needed to equalise Reynolds numbers. The discussion of

Section III illustrates how the wind tunnel was used to design the modifications which were instrumental in curing the symptoms of vibration input described in Section II. It should be remarked however that the authors' reliance on wind tunnel data was greeted with considerable scepticism, not only by the local naval architecture fraternity but also by a number of established overseas experts. This has prompted us to precede our acknowledgement of otherwise helpful discussions with an appropriate remark in Section V. *

It will be clear from the foregoing that the great advances of the last few decades in theoretical, as distinct from physical, modelling have not yet enabled us to predict ships wakefields, either actual or nominal, with sufficient accuracy to displace experiment. A simplified explanation of this is that the fluid stresses associated with turbulent momentum exchange cannot as yet be related to corresponding rates of strain in a consistent manner. Ship boundary layers and wakes are highly turbulent and existing theory is not much more than organisation of empirical observation and hence the continued need for ship model experimentation. At the same time one cannot overstress the importance of ship operators validating model results by revealing full scale data relating to performance, vibration, noise etc.

The situation is rather brighter in the field of theoretical performance prediction of propellers operating in non-uniform wakes. For a long time only averages of peripherally varying quantities were considered so that the radial distributions of inflow velocity used in the design of nominally "wake adapted" propellers were averages over appropriate circles and non-steady phenomena could not appear. Research workers have made determined attacks on the more general problem for many years. For example, one of us holds copy of internal report of 1962 by Dr. N. A. Brown of the Massachusetts Institute of Technology entitled "The Periodic Forces Acting on a Propeller Operating in a Circumferentially Non-Uniform Flow". However, useful working techniques for analysis and design have only emerged recently. These result not only in full performance estimates but they also give somewhat better than qualitative indication of the degree of cavitation present on a propeller blade as it rotates. An excellent summary, together with original work was recently published by Dr. van Oossanen (1.9) of the Netherlands Ship Model Basin (N.S.M.B.). Scandinavian work in this area has gone even further and has led to the Det Norske Veritas "Hydrocalc" programme which gives very reasonable estimates of the magnitude of pressure pulses sent into the water - and their impact on nearby hull structures - by intermittent cavity formation during the rotation of a blade; see Ref. (1.5) for a summary and correlation with full scale measurement on a 230,000 dwt. tanker. An applicable theory of this kind is essential for the evaluation of "how far to go" in wake improvement procedures when wind tunnels are used in the development of flow control structures. If the vibration input needs to be estimated experimentally, the sophisticated facilities at S.S.P.A. and N.S.M.B. (Vacutank) are available. However in the Australian context, the most cost-effective procedure is thought to consist of a judicious mix of simple towing tank and wind tunnel experimentation with use of theory to predict the degree of propeller cavitation and intensity of pressure fluctuation at points on the hull. This is described in the following sections.

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* It is possible that some of the critics were more concerned with our resuscitation of the vortex generator concept as a means of boundary layer/wake control in spite of the failure reported nearly a decade ago - see (1.7), (1.8).

REFERENCES :

1.1	Eisenberg, P.	"Cavitation Dictionary", Proc. 13th I.T.T.C. Berling/Hamburg, September 1972, pp.776-789.
1.2	Baier, L.A., Ormandroyd, J.	"Vibration at the Stern of Single Screw Vessels", Proc. S.N.A.M.E., 1952
1.3	Vossnack, E., Voogd, A.	"Development of Ships' Afterbody-Propeller excited Vibrations", Proc. 2nd Lips Propeller Symposium, May 1973, pp.103-175
1.4	Lindgren, H. et al.	"Propulsion and Cavitation Investigation on 230,000 t.dw Tankers: Full Scale and Model Experiments".R.I.N.A. Spring Meetings 1972, Paper No.8
1.5	Johnsson C.-A., Søntvedt, T.	"Propeller Excitation and Response of 230,000 TDW Tankers : Full Scale/Model Experiments and Theoretical Calculations." Det Norske Veritas Publication No.79, November 1972, 47 pp.
1.6	Fink, P. T.	"Record of Group Discussion on Facilities", Proc. 13th I.T.T.C., Berlin/Hamburg, September 1972, pp. 795-802
1.7	Townsin, R.	"Boundary Layer Separation from Ship Models : A Note on its Detection and Suppression with Some Examples", Q.Trans., R.I.N.A., 107, 3, July 1965, pp.341-349
1.8	Clements, R.E	"The Control of Flow Separation at the Stern of a Ship Model Using Vortex Generators", Q. Trans. R.I.N.A., 107, 3, July 1965, pp.351-366
1.9	van Oossanen, P.	"Calculation of Performance and Cavitation Characteristics of Propellers Including Effects of Non-Uniform Flow and Viscosity", N.S.M.B. 457, June 1974, 259 pp.

II. THE CASE STUDY : DEVELOPMENT OF TWO RO/RO SHIPS, 7000 TDWT

2.1 Design Background and Preliminary Model Testing

In 1969 Shipbuilding Division of the Department of Shipping and Transport began design work on two roll on/roll off ships for the carriage of rolled steel products between Port Kembla, N.S.W. and Westernport, Victoria, by the Australian National Line. The ships were constructed by State Dockyard, Newcastle N.S.W. and they have been described in the technical press (2.1). Essentially, the requirement was for an 18 knot, 7,000 ton deadweight design with facilities for the rapid handling and safe carriage of the cargo. The ship was to be relatively fast for its length and lines had to be fine. Due to heavy cargo, much of it abaft centre, afterbody lines had to be filled out to prevent excessive trimming by the stern. Maximum draft was to be 24 ft. A single screw installation had been specified in the interest of economic operation and reduced building cost. Lines were developed to fit these constraints and a design was evolved. The principal hull and propeller design characteristics are given in Tables 2.1, 2.2 and in Figures 2.1, 2.2.

In accordance with established practice at the time, a comprehensive programme of model testing was arranged, including resistance, self propulsion, stream flow and manoeuvring experiments. These indicated a satisfactory hull form subject to some minor adjustments to aft end lines to improve steering qualities. The tank report stated i.a. that ......" taken overall, the results of the experiments and the designed propeller indicate that the ship could be expected to give a satisfactory performance......"

2.2 Warning Signals, Further Model Tests, Ship Trials

In 1971, Shipbuilding Division's attention was directed to the particular problem of wake non-uniformity by a propeller manufacturer who sought model wake studies for another ship which was being designed at the time. That ship had somewhat finer stern water line endings than the above-mentioned A.N.L. ships. It was therefore decided to obtain wake measurements from the towing tank. Unfortunately this request was made before the current degree of understanding of the possible implications of strong wake peaks was reached and the advice received (see Fig. 2.3 as an example) was that ...." the general character of wake contours for load and ballast conditions although a little unusual, are considered to be reasonable for a stern design of the type considered". However, the propeller manufacturer disagreed with this assessment and asked that a more uniform distribution of wake be obtained.

This request set the stage for a protracted process of further experimentation and consultation over a period of a little more than one and a half years before trials of the "Lysaght Enterprise" took place in May 1973. This is not to say that any of the authorities involved were given to understand that they had 18 months in which to improve the wake characteristics; there were some delays in completing the ship but it would not have been prudent to rely on this. As a first step, some cavitation tunnel tests were

TABLE 2.1

PRINCIPAL SHIP PARTICULARS - FROM MODEL TEST REPORT

Model hull No.			5097C	5097C
Type of bow			Raked Stem	Raked Stem
Length between perpendiculars	Lpp	ft	432.667	432.667
Breadth moulded	B_{mld}	ft	74.0	74.0
Condition			Load	Ballast
Mean draught moulded	T_{mld}	ft	24.0	16.458
Trim at rest, in Lpp		ft	Level	6.25 by Stern
Equivalent mean draught moulded at level trim		ft	-	16.776
Designed rake of keel, in Lpp		ft	-	-
Displacement moulded	Δ_{mld}	tons	13,170	8,310
Wetted surface coefficient	(S)		6.178	6.580
Length-displacement ratio	(M)		5.601	6.530
Block coefficient	C_B		0.600	0.541
Maximum section coefficient	C_X		0.956	0.933
Prismatic coefficient	C_P		0.628	0.580
Longitudinal centre of buoyancy from amidships PP for level trim	LCB	ft	16.202 aft	-
LCB in trimmed condition		ft	-	22.693 aft
½ angle of entrance of waterline	i_E	deg	11	8
Length of entrance	L_E	ft	237.97	237.97
Length of parallel middle body	L_P	ft	-	-
Length of run	L_R	ft	194.70	194.70
Bilge radius		ft	12.0	12.0
Rise of floor		ft	0.5	0.5
Half flat of bottom amidships		ft	0.5	0.5

TABLE 2.2

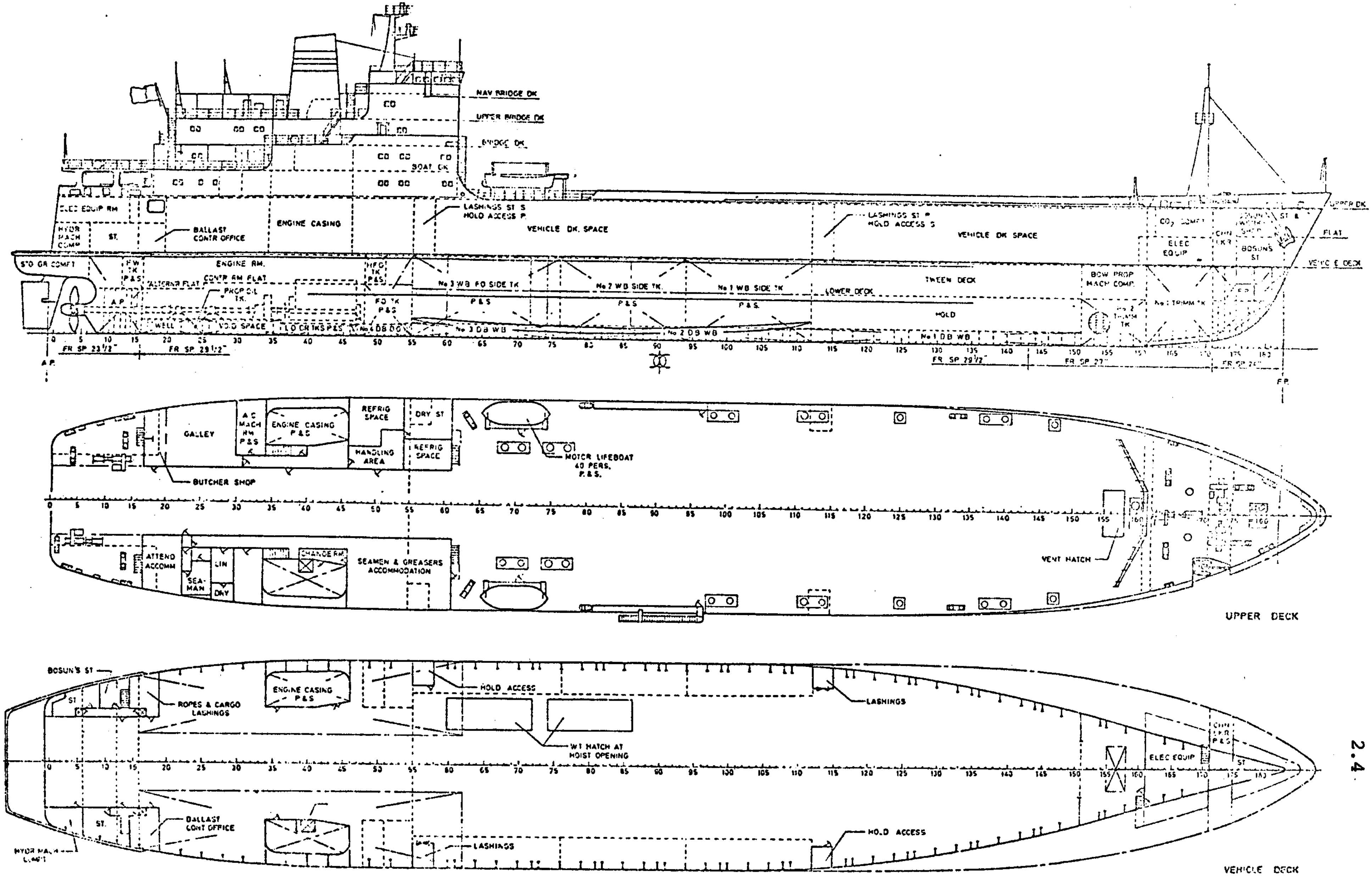
PRINCIPAL PROPELLER PARTICULARS - FROM MODEL TEST REPORT

Model hull No.			5097	
Model screw No.			W.130	U.481
No. of screws			One controllable pitch	
Designed by			NPL stock	Stone Manganese Marine Ltd.
Drawing No.			W.130	18375
Type of boss			Solid	
Material			Bronze	Novoston
No.of blades	Z		4	4
Diameter	D	ft	18.057	16.01
Boss diameter (max)		ft	3.529	4.216
Boss diameter at rake line	d	ft	3.414	4.724
Designed face pitch (max)	P_F	ft	15.857	16.726
Designed face pitch (mean)	P_M	ft	15.076	16.404
Developed area outside boss	A_D	ft^2	158.00	131.586
Cylindrical thickness	t_R	ft	at 1.82 ft rad) 0.719	at 2.362 ft rad) 0.7874
Thickness at shaft axis	t	ft	0.996	1.437
Rake aft		deg	10	0
Boss diameter ratio	d/D		0.1891	0.295
Mean face pitch ratio	P_M/D		0.835	1.0246
Blade area ratio	$4A_D/\pi D^2$		0.617	0.6536
Thickness ratio	t/D		0.055	0.0898
Centre of propeller				
- forward of AP		ft	10.81	5.30
- above moulded base		ft	9.25	8.50
Rake of shaft - up forward			-	-
Clearances :				
Trailing edge and rudder		ft	4.38	3.56
Top of aperture				
- above tips		ft	2.36	3.70
- forward of tips		ft	3.91	6.89
Bottom of aperture				
- below tips		ft	-	-

FIGURE 2.1

LYSAGHT ENTERPRISE / LYSAGHT ENDEAVOUR

432'8" L.B.P. x 74'0" x 48'6"

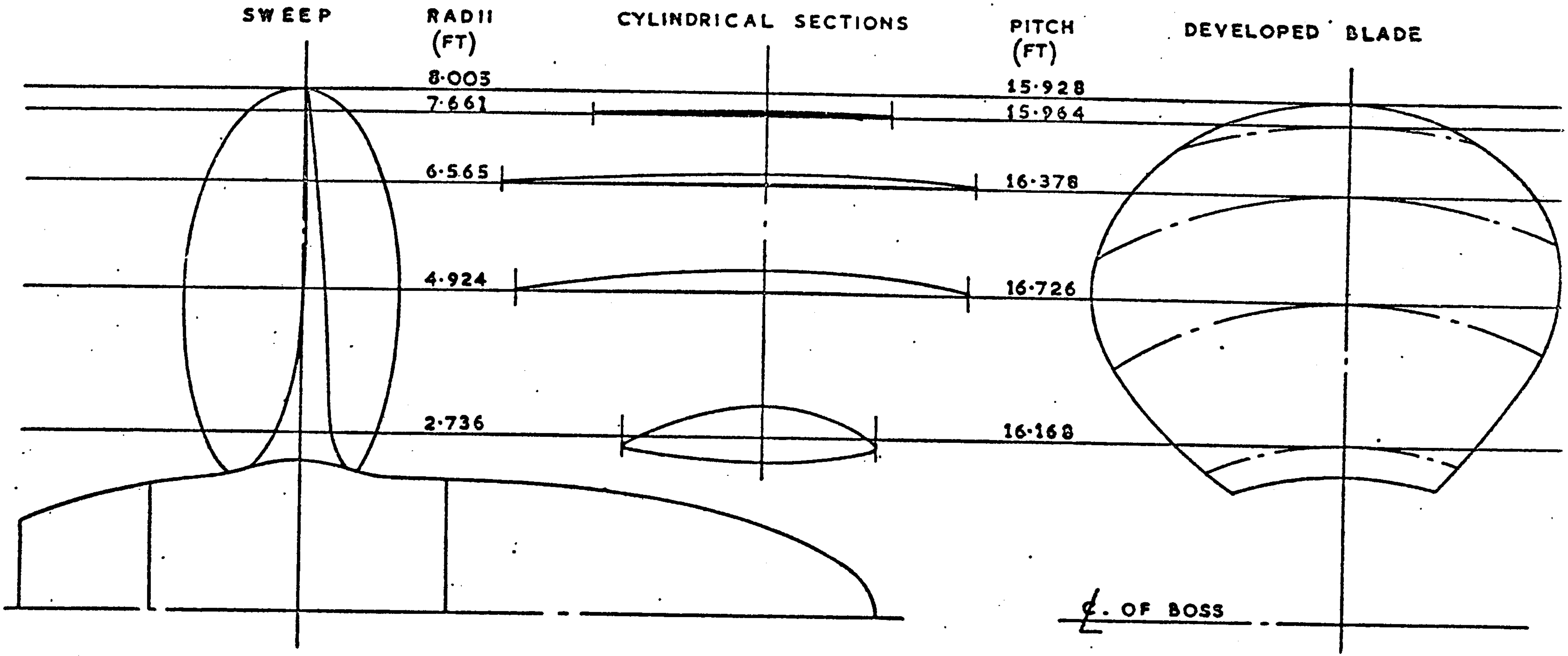


SHIPS DIVISION – N.P.L.

MODEL HULL No 5097.C.

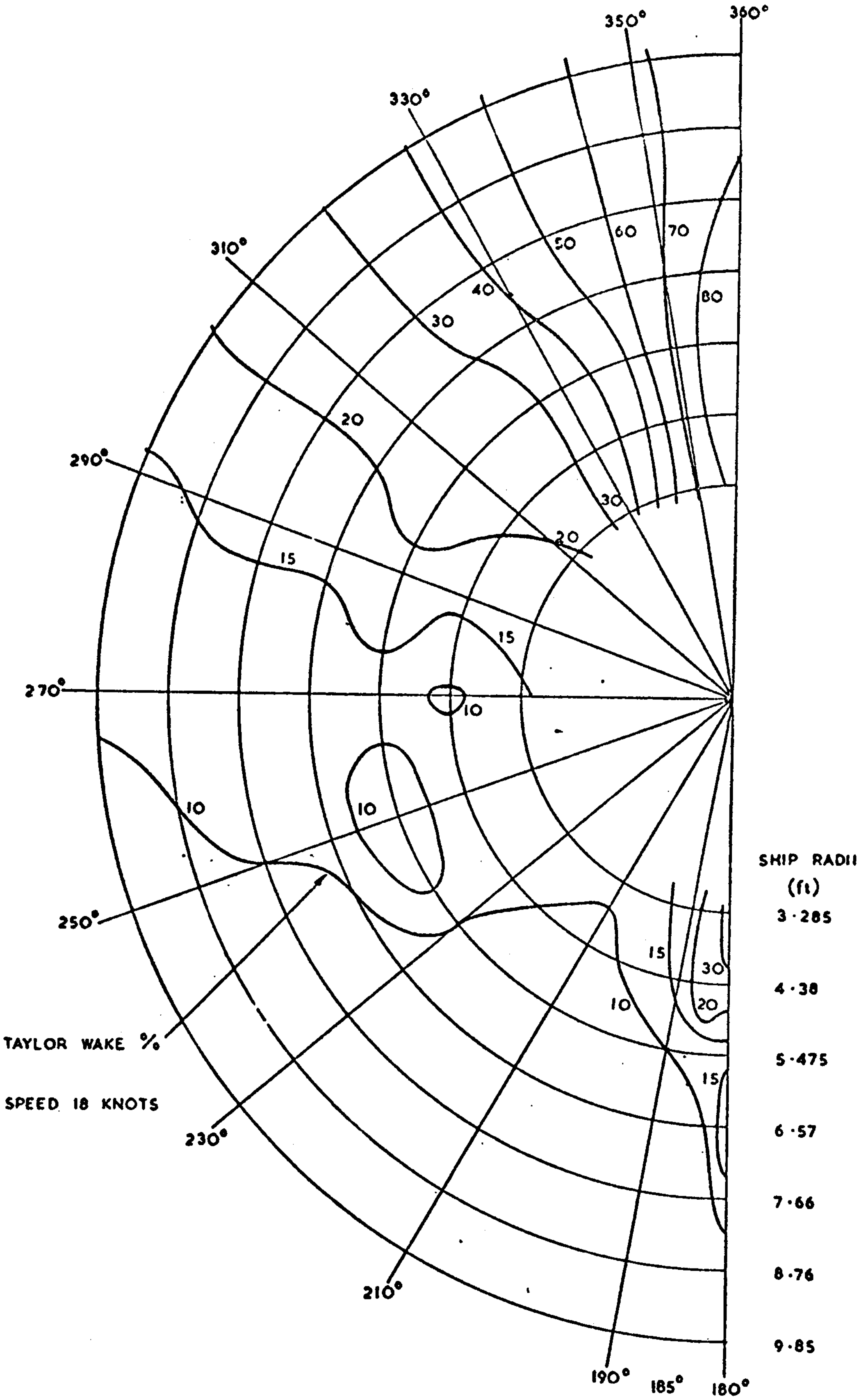
PARTICULARS OF SCREW U.481

SCALE :- 1/26.265 (SHIP)



DIAMETER	16·01 FT.
No OF BLADES	4-R.H.
BLADE AREA RATIO	0·6536
MEAN FACE PITCH RATIO	1·0246

FIGURE 2.2



(Taylor Wake percentage = 100 - Velocity Defect percentage)

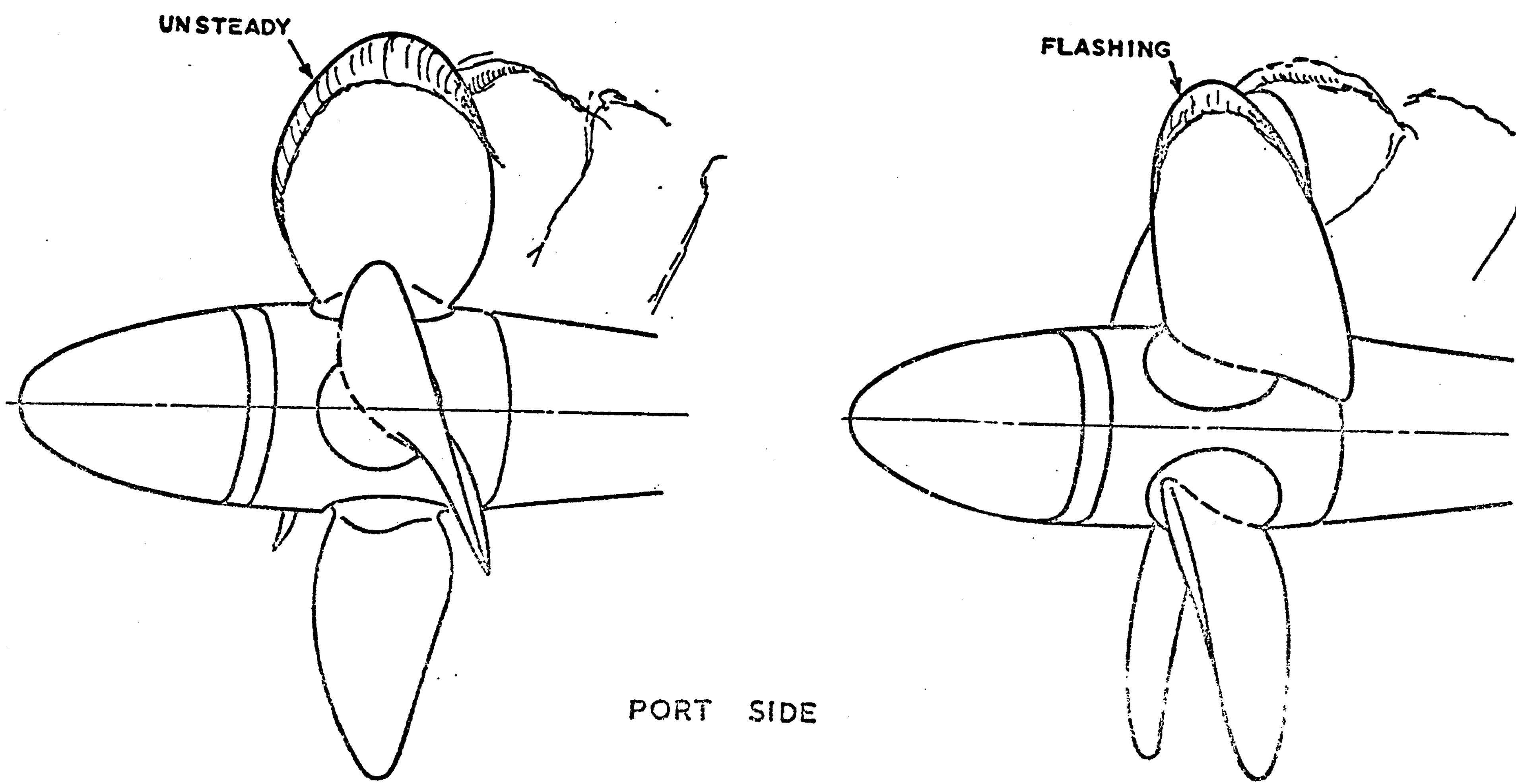
FIG. 2.3

CONTOURS OF CONSTANT TAYLOR WAKE DIAGRAM
LOAD CONDITION
MODEL N° 5097 C — PROJECT N° 51·1·82

commissioned in which the wake was simulated with graded screens.Typical results for two power settings are shown in Figs. 2.4, 2.5. These led to the conclusion that although there was evidence of some cavitation, the operation of the ship would probably be reasonably satisfactory to 10,000 D.H.P. and possibly up to 12,000 D.H.P., beyond which the cavitation was thought to be too extensive for satisfactory ship operation. In view of the advanced state of hull construction, it was not practicable to amend the aft lines for the first ship but the tank investigated revised lines, for possible application in the second ship, with disappointing results. The tank laboratory also investigated the effects of fitting horizontal fin flow deflectors (viz Great Lakes carriers/"Scandinavian" remedy) and a so-called "half tunnel"; i.e. a pair of keels set symmetrically with respect to the plane of symmetry of the ship with an intermediate structure to deflect "clean" water into the propeller disc along the lines of a partial nozzle. This scheme had been successful on other ships but failed to improve the wake of the A.N.L. ship model significantly. The principal author was able to see some of this work during a short visit to U.K. for another purpose in April 1972. Having noted the evidence of flow separation and after consideration of the lines of the ship, he suggested that vortex generators were likely to prove a practical means of boundary layer control; i.e. there seemed to be good prospects that an effective arrangement of vortex generators could be accommodated within the maximum dimensions of the ship. He would have to admit that he also advised in favour of continuing construction of the ship on what he now recognises as somewhat irrelevant grounds, namely that most of the existing model/full scale correlations of cavitation damage had shown model tests to be conservative. It is now recognised that the generation of pressure impulses by fluctuating cavities on propellers need not follow this trend.

Although the decision to continue construction was made in consultation with the owner in April 1972 it was thought prudent to continue to seek advice and to explore the matter experimentally. The towing tank laboratory explored a number of alternatives without success and its reports were received by end of year.

The "Lysaght Enterprise" underwent sea trials in May 1973 and encountered very severe aft-end vibrations at operating speed, both in ballast and in loaded conditions. This was accompanied by very loud noise in aft areas such as the steering flat. A local consultant measured aft deck accelerations up to 1.2 g at propeller design pitch in a later voyage and some other structural vibrations were also observed. Although comprehensive measurements were not made at this stage, trial observations indicated that the vibration was not due to a main hull excited resonance or out of balance forces but was attributable to propeller cavitation. The trials also confirmed that the degree of vibration increased with power, as a result of which a qualification was imposed by the Classification Society restricting propeller pitch to 21^{o}, corresponding to a service speed of approximately 16.5 knots in place of the design speed of 18 knots. This was to ensure the safety of the vessel and an adequate degree of habitability. It was learned later that propeller pitch needed to be set to angles of 17^{o} and less for complete absence of vibration discomfort.



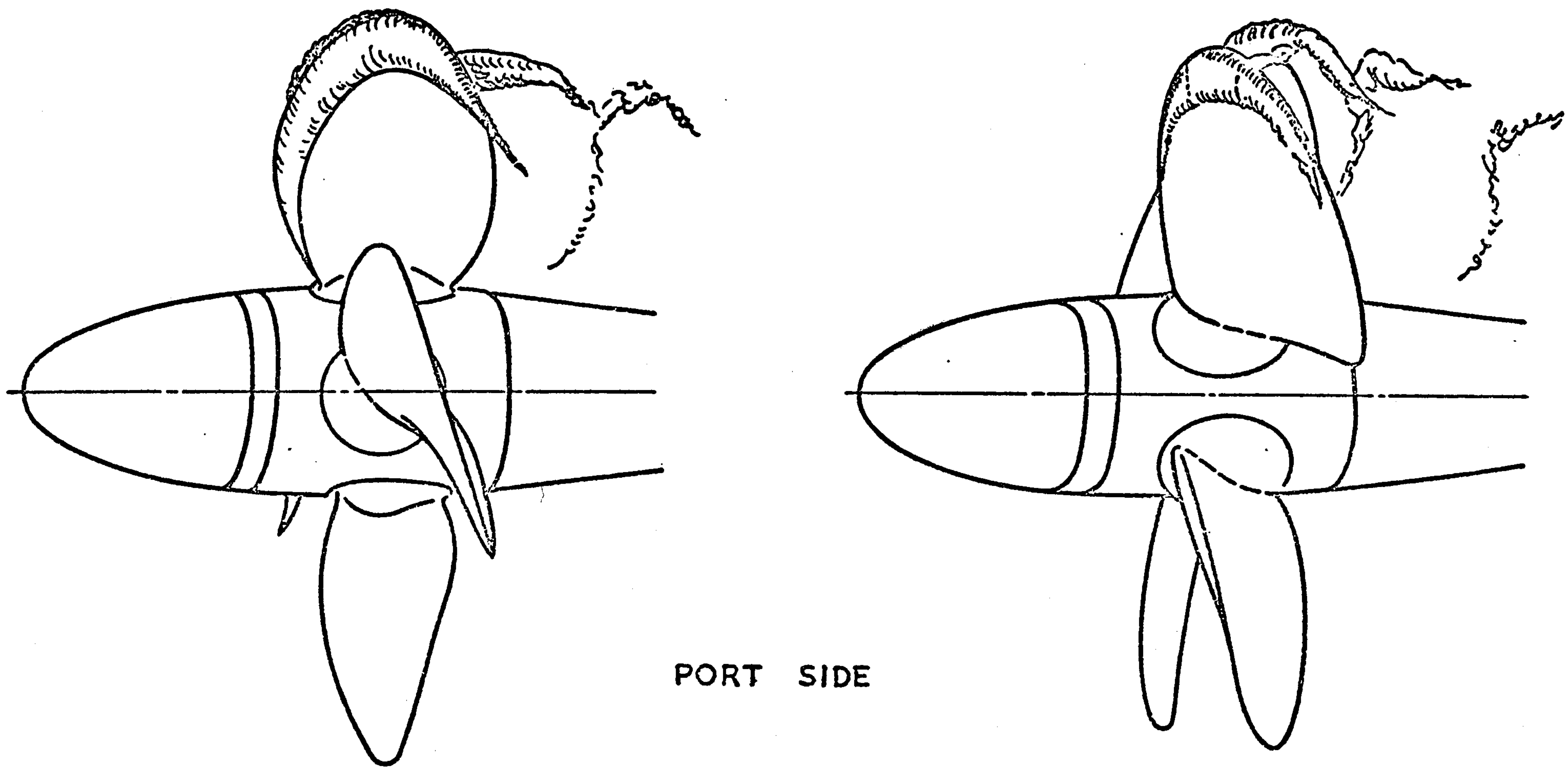
PORT SIDE

V = 18 KNOTS
N = 140 REV/MIN
P_D = 10,000 hp
$\theta_{0 \cdot 7R}$ = 23° 42′

K_Q = 0·0329
σ_n = 0·458
AIR CONTENT = 35% SATURATED

SCREW U 481 CAVITATION SKETCH

FIG. 2·4



PORT SIDE

$V = 18$ KNOTS	$K_Q = 0{\cdot}0395$
$N = 140$ REV/MIN	$\sigma_n = 0{\cdot}458$
$P_D = 12{,}000$ hp	AIR CONTENT = 25% SATURATED
$\theta_{0{\cdot}7R} = 25^\circ 24'$	

SCREW U 481 CAVITATION SKETCH

FIG. 2·5

Following on a minor grounding on trials the ship had to be docked again and this gave some of us a further opportunity to form a three-dimensional impression of the ship's lines. Assuming the vibrations to have been caused by wake/propeller induced cavitation, and also that the wake peaks were associated with unduly thick or even separated boundary layer flow over the aft end of the hull - and bearing in mind the model tests already made - the senior author again stated his conviction that a few vortex generators (placed a few stations ahead of the stern post) would probably control the boundary layer enough to improve the flow into what was after all an area of under 100 ft^2.

However it was thought best to commission a further overseas ship model testing laboratory to develop an effective and acceptable modification of the ship before she was due to dock after an initial period of operation under reduced speed. However the hull shape appears to have been very unkind to such empirical studies for none of the 24 combinations of flow directors which were tested showed promise of significant improvement in wake characteristics. The problem had now become very urgent, not only because of the operational limitation on "Lysaght Enterprise" but also in view of the advanced stage of construction of "Lysaght Endeavour" which was to make its trial voyage in September 1973.

In the meantime "Lysaght Enterprise" had become something of a "bateau célèbre", particularly among those members of the marine world who were unaware of the extent to which the same disease had afflicted world shipping and indeed some other Australian ships. One overseas consulting organisation then made a novel and generous proposal : it would provide drawings of an arrangement of vortex generators and sternpost fairing by return, on a no cure/no pay basis. This was accepted in time for fitting to "Lysaght Endeavour" before her first trial voyage, i.e. she sailed with two rows of vortex generator plates approx. 9 in. x 8 in. set at 3 ft. intervals at varying angles of incidence at each of stations 5/8 and 1, making 44 in all and with a fairing plate in the upper area of the aperture. It was presumably the urgency of the situation which prompted the organisation to recommend a suit of vortex generators without first satisfying themselves as to the state of the boundary layer that they were trying to control. In the upshot, the generators proved ineffective and it was then recommended that one row of 22 vortex generator plates be burned off in order to reduce possible interference. This was done but gave negligible improvement - presumably because the boundary layer at station 1 and aft was many times thicker than the height of the recommended generators (see Section III for explanation of operation of these devices). The recommended sternpost fairing was considered to be beneficial* and was retained on both ships. The Classification Society was obliged to put the same qualification on propeller pitch as for the sister ship.

By the end of September 1973, a small team from Det Norske Veritas was asked to make a more detailed vibration study of "Lysaght Endeavour" which was then equipped with 22 vortex generators and stern post fairing. Pressure transducers were fitted into the hull plating a little ahead between frames 3,4 and 4,5 and above the propeller just outside the sternframe casting in order to measure pressure impulses received by the hull and integrating accelerometers were placed in a number of locations on the ship in order to determine amplitudes of vibration. Typical results from the pressure transducers are shown in Fig. 2.6. They exhibit the characteristically rapid rise

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* It has been long known that splitter plates reduce the drag of bluff bodies in two-dimensional flow situations and that they inhibit fluctuating interactions of shed vorticity from the two sides of the body (see e.g. Ref. 2.2 for a more recent set of experiments).

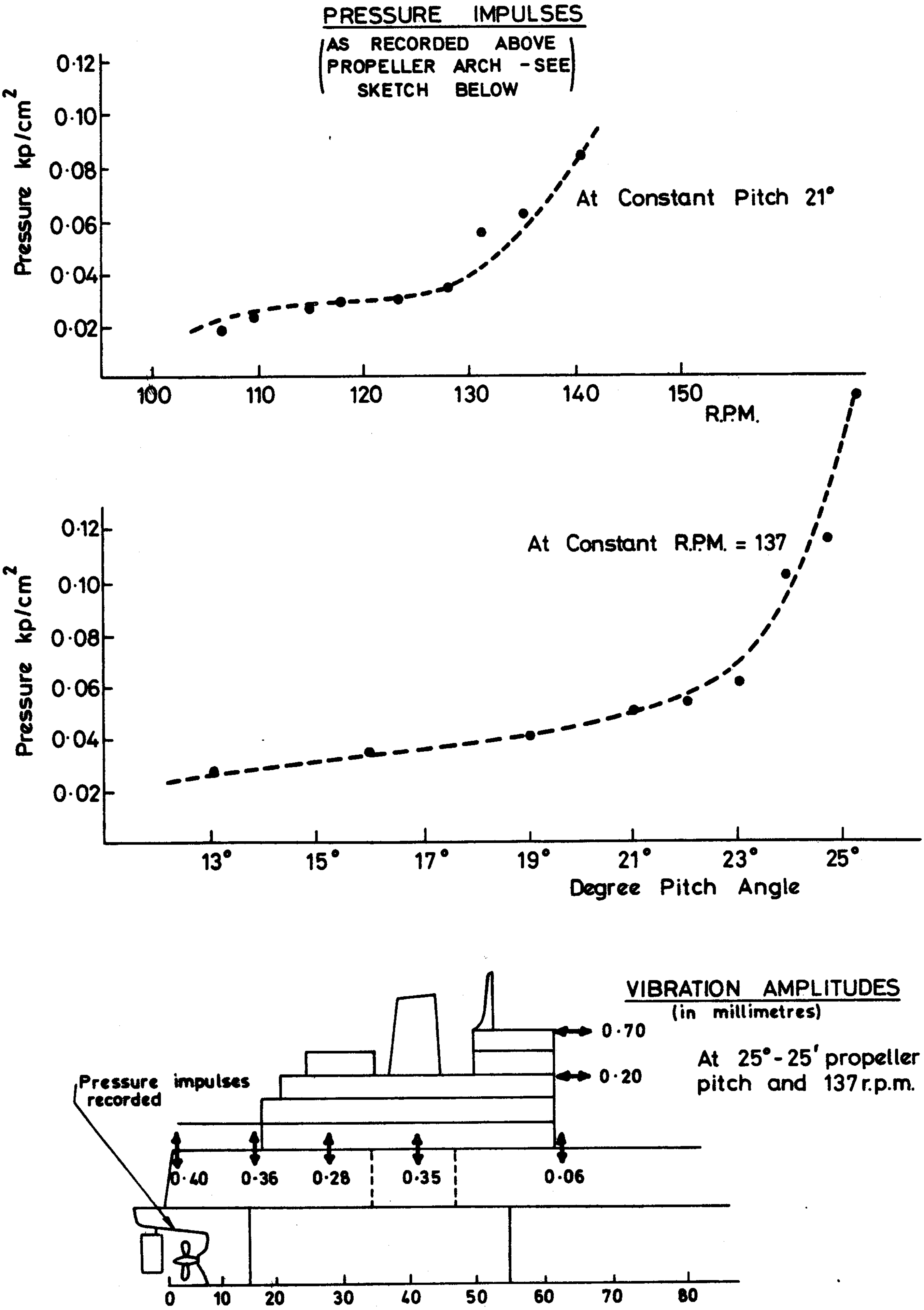
PRESSURE IMPULSES
(AS RECORDED ABOVE PROPELLER ARCH - SEE SKETCH BELOW)
Pressure kp/cm2
0·12
0·10
0·08
0·06
0·04
0·02
At Constant Pitch 21°
100
110
120
130
140
150
R.P.M.
At Constant R.P.M. = 137
13°
15°
17°
19°
21°
23°
25°
Degree Pitch Angle
VIBRATION AMPLITUDES
(in millimetres)
At 25°-25' propeller pitch and 137 r.p.m.
0·70
0·20
Pressure impulses recorded
0·40
0·36
0·28
0·35
0·06
0
10
20
30
40
50
60
70
80

FIG. 2·6

of impulse pressure with increasing power absorption. It was interesting to note that Norske Veritas recommend that propellers should be designed to generate amplitudes no greater than 0.08 - 0.10 kp/cm^2 and that this amount corresponds to propeller pitch of 21^o at normal engine r.p.m. in the present case; i.e. the pitch angle judged by Lloyds' surveyor to be the maximum that could be safely applied in either ship before modifications were made in 1974. The pressure amplitudes proved to be 13% greater than the theoretical estimate for the component at blade passing frequency using the previously mentioned "Hydrocalc" programme. However there was also significant energy content in theoretically predicted higher harmonics. Conclusions of the investigation were as follows.

Recorded vibration levels on upper deck and afterbody .... far exceeds acceptable values; vibrations are the forced type excited by hydrodynamic propeller forces becoming large due to heavy cavitation; "hydrocalc"s predict sheet cavitation over significant areas of propeller (see e.g. Fig. 2.7), pressure impulses estimated at least three times maximum tolerable value; upperdeck and other areas unsupported due to handling and storage requirements and need to be stiffened : a number of detailed recommendations were made such as fitting of triangular plate on each side between navigation bridge deck and upper bridge deck, radar mast etc. Since permanent pillars cannot be fitted to reduce unsupported spans it was suggested that a hydraulically removable pillar be fitted at frame 76 (see Fig. 2.6) if hydrodynamic input could not be reduced adequately. An unofficial estimate was given that vibration amplitudes might well be halved by adequate stiffening alone.

2.3 Steps to a Solution

When the principal author learned of the previously mentioned consulting organisation's generous no cure/no pay offer in respect of what might be called an "instant" vortex generator solution, he became seriously concerned that in the event of its failure, further development work to find a more appropriate suit of vortex generating fins might be prejudiced. He had not seen the recommended arrangement but he knew that it could not have been based on an appropriate boundary layer investigation. In any case, as explained earlier, "Lysaght Endeavour" was to have gone on trials very shortly after so that the consultants would have been unable to make the investigations which they would no doubt like to have made. As a precautionary measure Shipbuilding Division then agreed to sponsor a windtunnel study of the boundary layer on a double model of the A.N.L. ships. After the failure of the other solution this developed into the test series described in Section III, a last resort since all previous recommendations had also failed. However it must be said that progress towards the goal of lifting the Classification Society's qualification was not straight forward and the "home side" committed its share of oversights and confusions.

Windtunnel tests were conducted by the junior author at Aeronautical Research Laboratories in October 1973 and by the end of that month he had devised simple vortex generator arrangements which achieved wake velocity distributions of remarkable uniformity with far smaller nominal wake peaks than had appeared anywhere in the literature. Power losses in terms of E.H.P. appeared at that stage to be under 10%. A meeting was then held with representation of all parties, including two of the consulting organisations which had helped in the project. The above results were received with very

SUCTION SIDE

DISTRIBUTION OF CAVITATION, SCHEMATICALLY

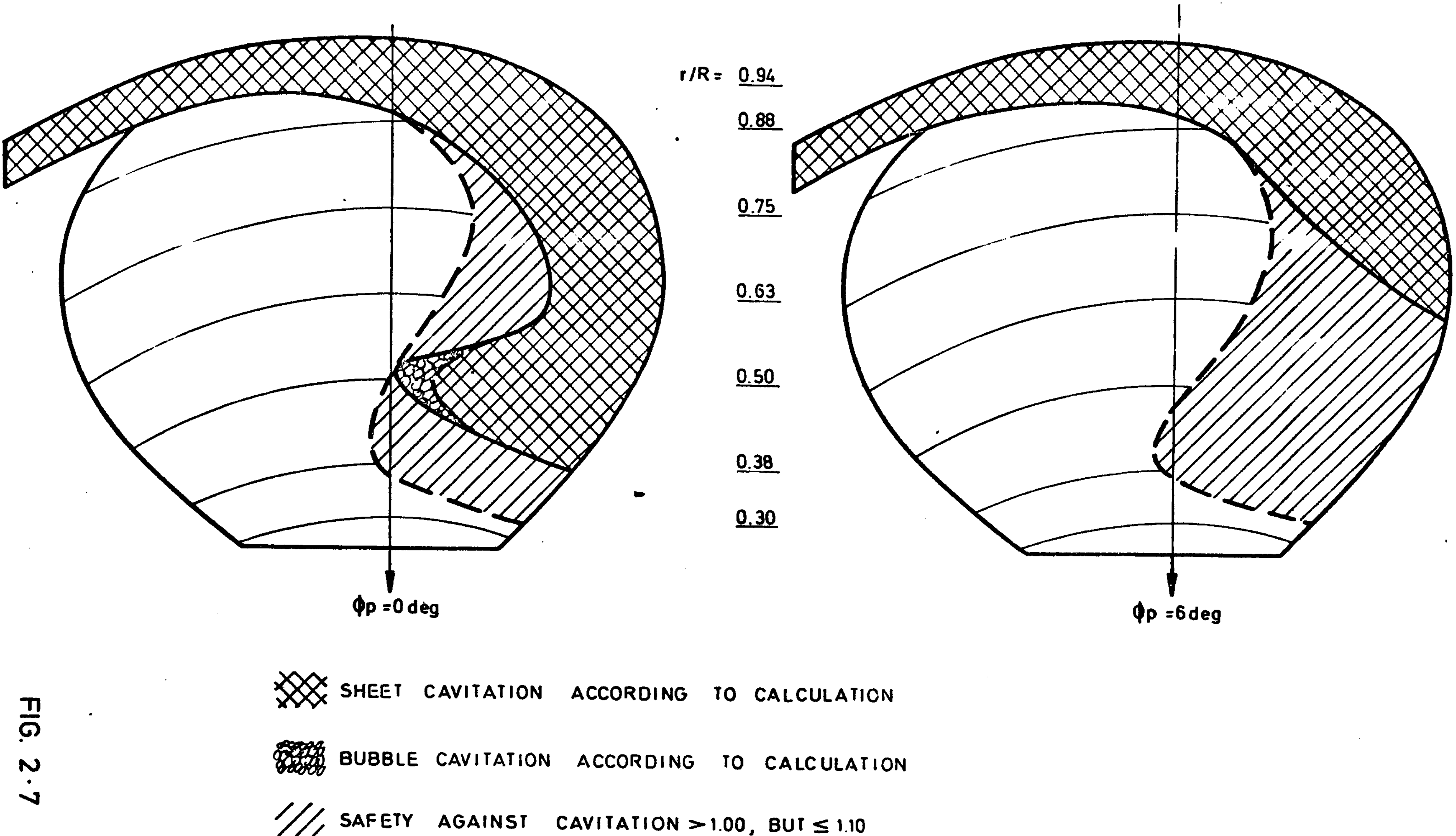

FIG. 2·7

considerable scepticism and it was said repeatedly that free surface effects would most likely intervene to produce rather different answers : the zero-Froude number approximation* was not acceptable and the rough argument that static pressure increase from the stern wave would be offset by pressure decrease due to propeller suction was brushed aside! The meeting did not discover an error in scaling - for which two of the present authors must take the blame - which produced a serious under-estimate in E.H.P. increment. It also failed to recognise the rise in thrust deduction factor associated with the vortex generator drag and hence the probability of an even greater increment in D.H.P. The scepticism did serve the very useful purpose of producing a commission for towing tank checks of vortex generator effectiveness in improving the wake and for resistance and propulsion tests to check the power loss due to fitting vortex generators.

These check tests were run at Vickers, St.Albans, during the coal strike and the short working week early in 1974. It is greatly to that laboratory's credit that results were obtained on time. Again, progress was not smooth. The initial tests were made with vortex generators mounted in incorrect positions due to failures in communication associated with the Australian summer holiday period. That set of results was to prove useful later when it became useful to know more about the sensitivity of the results to shifts of vortex generator position. When the correct positions were established it became possible to confound the critics of the $Fr \rightarrow 0$ approximation by demonstrating good agreement between wakes measured in both facilities - see e.g. Fig. 2.8. On learning that the St.Albans laboratory found an E.H.P. increment of 24% for the original optimum recommended by A.R.L., the "home side" checked its extrapolations and produced a new estimate : 26%. While this agreement was gratifying scientifically, it meant that the optimum configuration had to be abandoned in order to arrive at acceptable power levels.

At that stage Det Norske Veritas were presented with the modified wakefield produced by a less fuel hungry arrangement of vortex generators and the previously mentioned "hydrocalc" computer programme** was used to give estimates of corresponding hull pressure fluctuations. The remainder of the story of progress towards the solution, including the part played by Vickers, St.Albans in moving towards a four-generator arrangement is given in Section 3.5.

The senior author had the pleasure of being on board "Lysaght Enterprise" when she went on trial after her first docking - complete with a set of four "small" A.R.L. type vortex generating fins on 26th August, 1974. The aforementioned structural stiffening had also been made. It would therefore be incorrect to assert that the very greatly reduced vibration levels found on that trial were entirely due to hydrodynamic modifications. However there is a considerable body of evidence to suggest that the major part of the improvements found are due to reduction of hydrodynamic vibration input. Certainly, the Lloyds' surveyor had no difficulty in removing all pitch qualifications from the ship.

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* Unfortunately there were no representatives from the Swedish S.S.P.A. to justify *their* investment in a multi-million dollar zero-Froude number facility for the study of aft-end and propeller-wake interaction problems!

** That computer programme is the only quick method known to the authors for estimating the amount of wake improvement required to prevent excessive pressure impulse radiation.

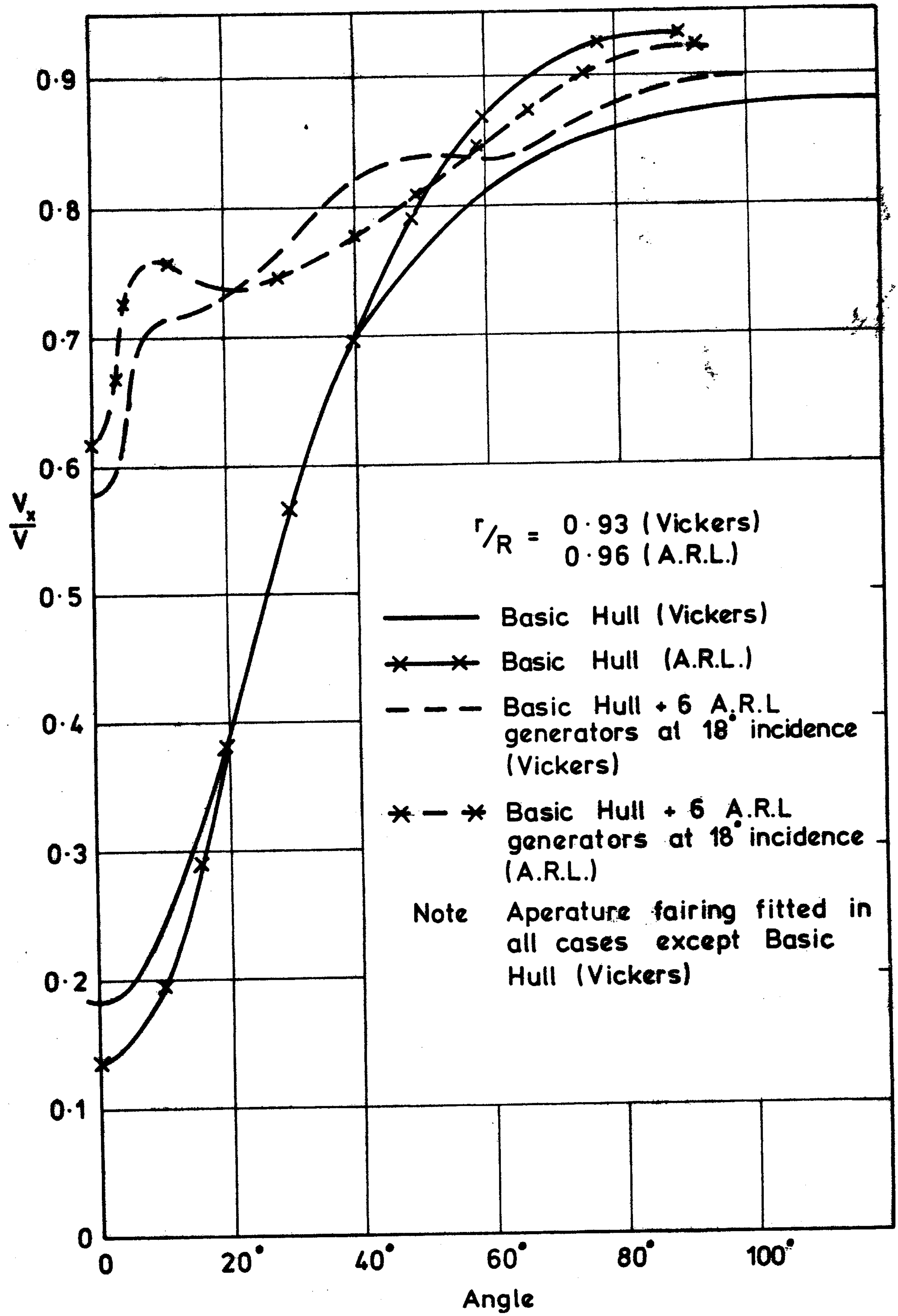


FIG. 2·8. AXIAL WAKE VELOCITY DISTRIBUTION: COMPARISON OF AERODYNAMIC AND HYDRODYNAMIC TESTS.

It should also be mentioned that the shipowner was kind enough to allow a traversing Pitot tube to be fitted for a limited boundary layer study on "Lysaght Enterprise". This was used to satisfy ourselves that the fitted vortex generators will almost certainly retain their effectiveness as the ship fouls.

"Lysaght Endeavour" completed her first docking on 20th October and went on trials with a similar set of four vortex generators but set to smaller incidence angle of 16^{o} as explained in Section 3.5. Once again, the ship was able to develop full power without vibrations and the senior author was happy to witness a small ceremony on the bridge in which the pitch restriction plaque was carefully removed from the control console with a knife after the ship had been cleared by the representative of the Classification Society.

REFERENCES :

2.1	-	Shipping World & Shipbuilder, August 1973, pp. 871-873
2.2	Bearman, P.W.	"Investigation of the Flow Behind a Two-Dimensional Model with a Blunt Trailing Edge and Fitted with Splitter Plates", J.Fluid Mech., 1965, 21, Part 2, pp.241-255

III. WIND TUNNEL INVESTIGATIONS OF VORTEX GENERATOR SOLUTION

3.1 Vortex Generator Design : First Test Series; October 1973.

Vortex generators were considered to be the most suitable devices for improving the velocity distribution over the propeller disc in the present case, as they are both efficient, and simple to make and fit.

Vortex generators operate by sweeping high momentum fluid particles along helical paths to mix with retarded fluid in the boundary layer near the body surface. The resultant increased energy of the fluid in the boundary layer counters the tendency of the adverse pressure gradient to cause separation.

A number of types of vortex generators exist, such as vane, wing, wedge and ramp types. In the majority of cases vane type generators have been found to give the best performance.

The broad criteria for good performance are that the generators produce a high level of effectiveness over a wide range of conditions coupled with low drag. Unfortunately, these requirements are mutually conflicting and some compromise is necessary. Usually many tests are made in order to refine the generator design to obtain the best solution for a particular problem.

Vane type vortex generators were chosen for tests with the model and were initially designed according to the rules given by Lachman (3.1). A triangular planform was selected because of its known efficiency in producing a strong vortex.

Initial tests were made with vortex generators fitted at various locations on the stern of the model. In all cases they were positioned so that they did not protrude beyond the maximum draught and beam of the hull. Surface flow visualisation, velocity wake survey and drag tests were used to evaluate the performance of each generator configuration.

3.2 Experimental Equipment

All tests were carried out on a 1/48 scale mirror image model (approximately 9' L.B.P.) without either the propeller or rudder fitted.

The model was mounted in the 9' x 7' working section low speed wind tunnel using a sting and shroud as shown in Fig.1. Provision was made for tests in both the full load and ballast conditions. Studs, 0.035" x 0.125"d spaced at 3 d, were fitted approximately 6" aft of the bow waterline endings of the model to promote turbulent flow over its surface similar to that over the full scale ship. The model was fitted with a propeller arch fairing which had been fitted to the ship after its initial trials. Reynolds number of the tests was approximately 1/50th full scale.

At selected positions velocities in the wake were measured in magnitude and direction in the plane of the propeller using five-hole pitot probes connected to a multitude manometer.

The surface flow pattern was made visible using french chalk mixed with kerosene.

FIG. 1. MODEL AT FULL LOAD DRAUGHT FITTED WITH THE "OPTIMIZED" VORTEX GENERATORS

3.3 Model Experiment Results

The main aim of the experiments was to obtain a vortex generator system which would create a more uniform distribution of the axial wake velocity over the propeller disc, and have acceptable resistance characteristics. Initial tests were run on the bare model. These confirmed the presence of some boundary layer separation over the aft portion of the hull.

Tests were made for some sixty systematic variations in vortex generator design parameters. Initial tests soon indicated the broad requirements for the generators and most of the later tests were aimed at optimizing the generator design.

The generator heights were changed from 1/4" to 3/4" and the aspect ratio from 1.0 to 1.8. These parameter changes were coupled with changes in the number of generators and variations in the generator angle of incidence to the local surface flow from 8° to 40°. Longitudinal positioning of the generators was varied from station three to station one, and transverse positioning varied from close to the keel centreline to close to the water-line. It was found preferable to use a relatively small number of large generators. If the generators were placed too far apart or too close together they became less effective. Likewise, if the generators were placed too close to the stern or too far upstream they were also less effective than in the positions finally chosen. The results of all these experiments are not discussed in detail. However, a vortex generator configuration emerged which gave a very good velocity distribution in the wake combined with what were thought to be acceptable drag increments for the model. These generators are shown mounted on the model in their optimised position in Fig.1.

Details of the set of vortex generators finally selected, and the relevant flow patterns, wake velocities and resistance characteristics for the model obtained both with and without these generators are presented and compared with one another in the following sections. A few results are also given for similar generators of different size and angle of attack.

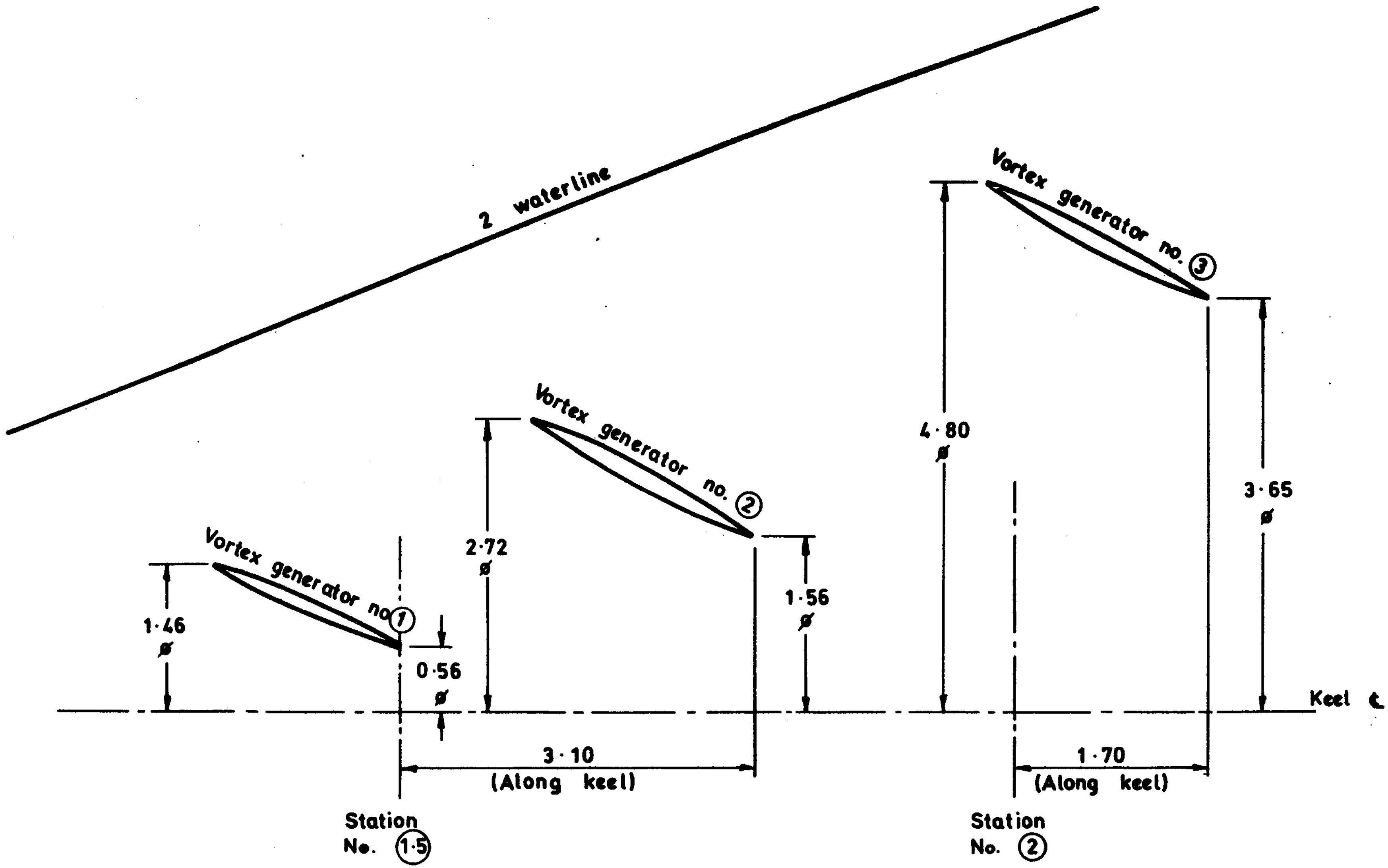
3.3.1 Vortex Generator Configuration

The positions of the set of vortex generators which was considered to give the best results are marked on a plan view of the starboard undersurface of the model hull in Fig.2(a). Similar generators were placed symmetrically about the keel centreline on the port side of the hull. All generators were attached normal to the hull surface at a point two-thirds of the generator length aft of the forward end. Dimensions of the generators for the model scale are given in Fig.2(b).

3.3.2 Surface Flow Patterns

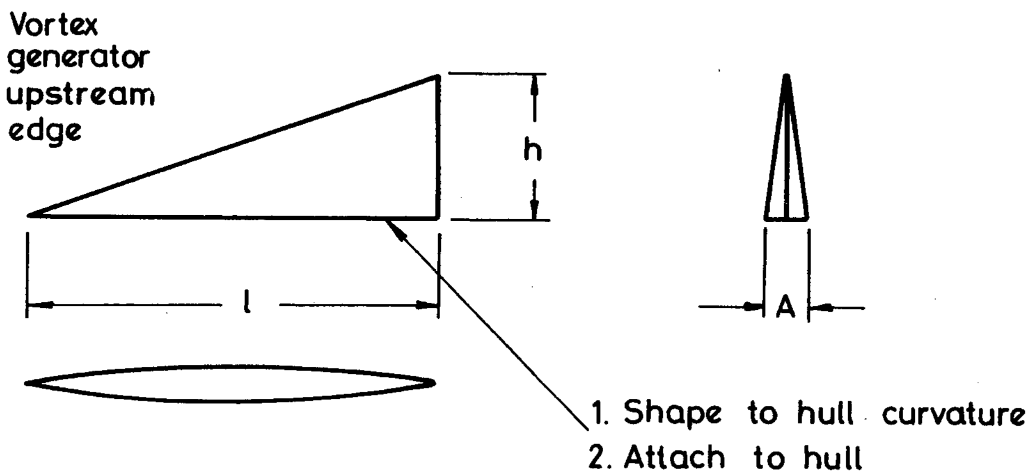
The surface flow pattern over the stern of the model without vortex generators is shown in Fig.3, for the full load condition at a Reynolds number of 1.2×10^7. A similar pattern was obtained for the ballast condition.

The flow patterns in both conditions show a region of separated flow extending from approximately the propeller shaft centreline to the top of the propeller arch, and from approximately station three-quarter to the waterline endings. These flow patterns, showing a region of separated flow, appear to be consistent with the original tests made at N.P.L.



(a) Vortex generator position

FIG. 2. VORTEX GENERATOR SHAPE AND POSITIONS AT MODEL SCALE



Generator No.	l	h	A
①	1·80"	0·500"	0·150"
②	2·25"	0·625"	0·175"
③	2·25"	0·625"	0·175"

(b) Vortex generator shape

FIG. 2. VORTEX GENERATOR SHAPE AND POSITIONS AT MODEL SCALE

FIG. 3. FLOW PATTERN OVER THE STERN OF THE MODEL AT FULL LOAD DRAUGHT WITHOUT VORTEX GENERATORS.

Fig.4 shows the surface flow pattern over the stern for the full load condition with the selected set of vortex generators fitted to the model and tested at the same Reynolds number. The surface flow effects of the vortices shed from the generators are clearly visible. The flow patterns also show how the generators sweep relatively high energy fluid from near the keel of the model up towards the top of the propeller disc.

In both load conditions the generators completely eliminated the region of separation forward of the propeller arch. As a result the vortex generators were expected to increase the axial wake velocity in the upper regions of the propeller disc where low wake velocities had been measured at N.P.L.

3.3.3 Wake Velocity Components

The axial velocity components were measured at a number of positions in the plane of the propeller disc for the full load and ballast conditions, both with and without the generators fitted to the model. The majority of results were taken over the starboard side of the propeller disc with only a few measurements over the port side to check for symmetry. The wake velocities were measured at a Reynolds number 1.2×10^7.

Axial wake velocity components for the full load and ballast condition without vortex generators are shown in Figs. 5 and 6 expressed as a ratio of the effective forward velocity of the model. In general, wake velocities were lower in full load than in ballast.

The axial wake velocity components for the model fitted with the selected set of vortex generators in both load conditions are shown in Figs. 5 and 6. Figs. 5 and 6 show the very large increase in axial wake velocity obtained when the generators were fitted, as was expected from the flow visualisation studies. If the vibration observed on the ship is, in fact, caused by low wake velocities over the upper segment of the propeller disc, then, provided similar improvements in wake velocity are obtained on the full scale ship, the vibration of the ship should be substantially reduced.

The axial wake velocities over the upper segment of the propeller disc for the generators set at angles of incidence of $\alpha = 8^\circ$, 15°, 20°, 30° and 40° in the full load condition are given in Table 1. At $\alpha = 8^\circ$ the axial wake velocities are quite low in the region $0^\circ < \Theta \leqslant 10^\circ$ and it was concluded that this low angle of incidence is inadequate. Angular position in the propeller disc plane at which wake velocities were measured is denoted by Θ measured from top dead centre, positive in a clockwise direction when viewed from aft. For α between 15° and 40° the results do not vary appreciably, indicating that the angle of incidence of the generators within this range is not critical. The angle of incidence, 25°, was chosen for the generators to provide for changes in the angle of incidence as the ship rolls in rough seas. This choice gives $\pm\, 10^\circ$ margin before the angle of incidence, on the one hand, falls below the level at which the effectiveness of the generators starts to decrease, and, on the other, rises above the level at which stalling of the generators is imminent.

The effects on the axial wake velocity for some changes in generator size at an angle of incidence of 25° are given in Table 2. There is little difference between the axial wake velocities for the two larger sets of generators. However, as the height decreases the axial wake velocities decrease, especially over the outer radii of the propeller disc. These results indicate that either of the two largest sets of generators would give suitable increases in axial wake velocity. The smaller of these two was finally chosen because of the lower drag penalty involved.

FIG. 4. FLOW PATTERN OVER THE STERN OF THE MODEL AT FULL LOAD DRAUGHT FITTED WITH THE "OPTIMIZED" VORTEX GENERATORS.

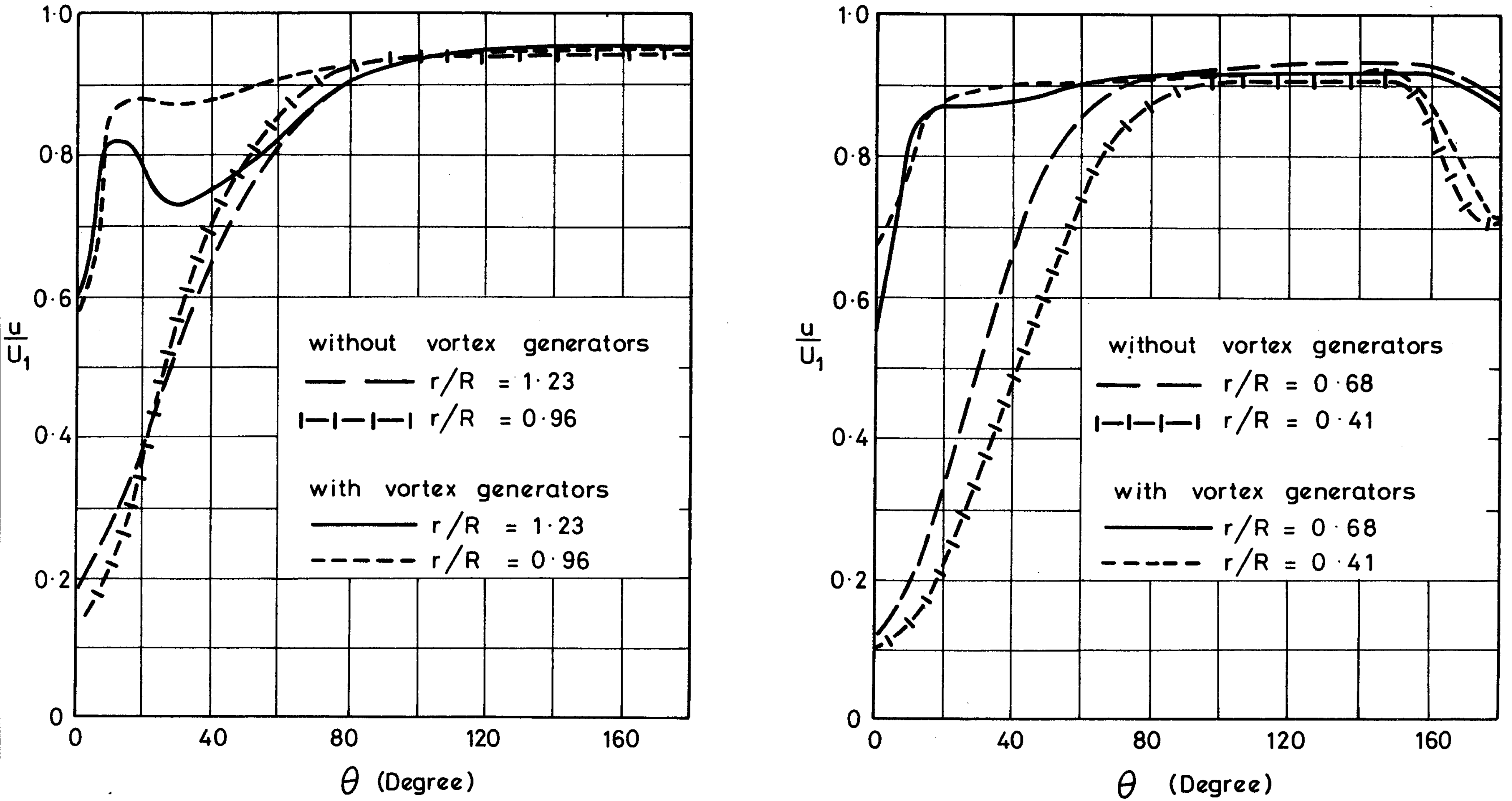


FIG. 5. AXIAL WAKE VELOCITY DISTRIBUTION IN THE PROPELLER PLANE OF THE MODEL AT FULL LOAD DRAUGHT WITHOUT VORTEX GENERATORS AND FITTED WITH THE OPTIMIZED VORTEX GENERATORS.

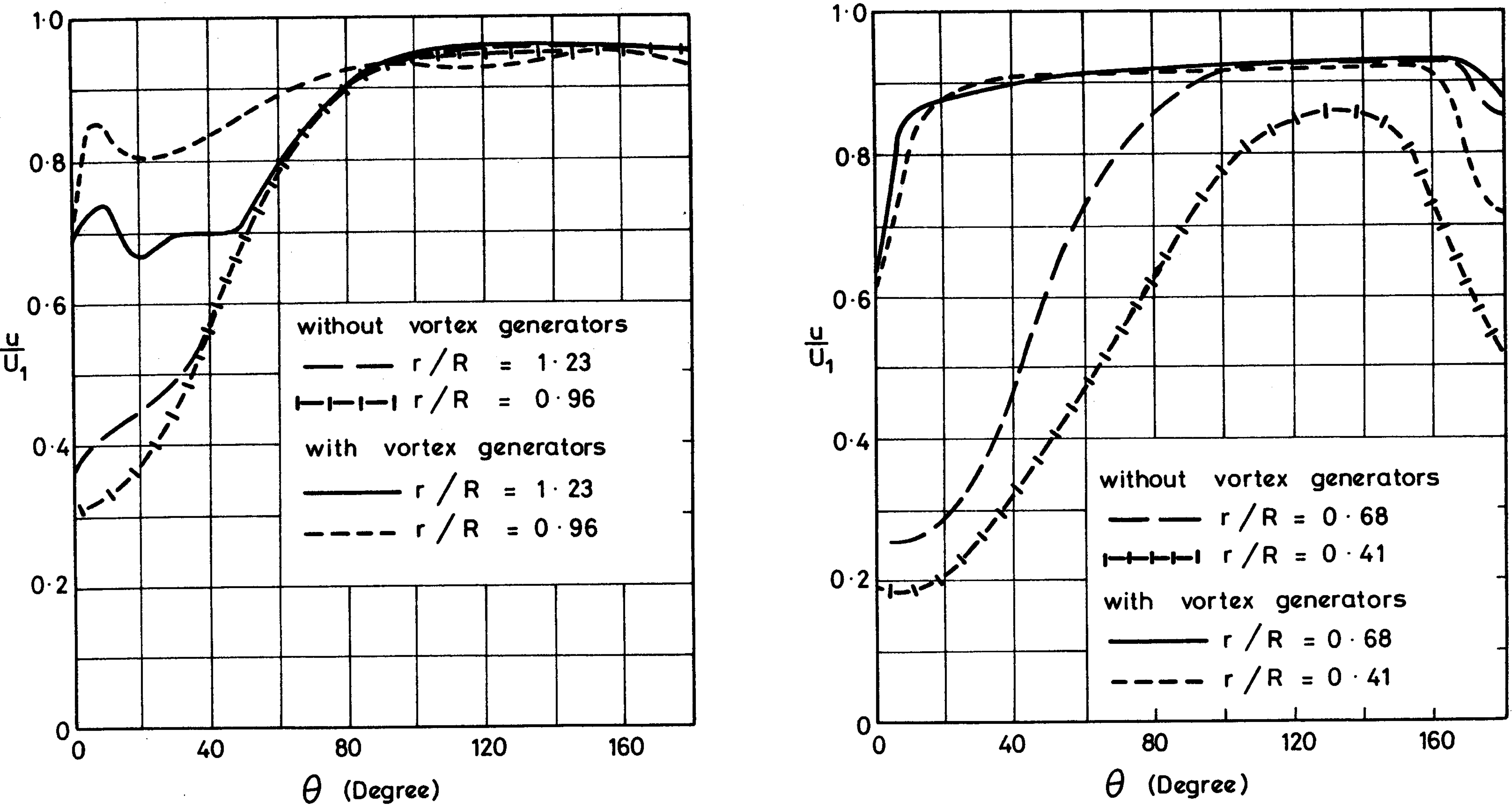


FIG. 6. AXIAL WAKE VELOCITY DISTRIBUTION IN THE PROPELLER PLANE OF THE MODEL AT BALLAST DRAUGHT WITHOUT VORTEX GENERATORS AND FITTED WITH THE OPTIMIZED VORTEX GENERATORS.

TABLE 1.

Axial wake velocity component ratios for the model in the full load condition with the vortex generators fitted at various angles of incidence.

r/R = 1.23					
Θ (Degree)	u/U				
	$\alpha = 8^\circ$	$\alpha = 15^\circ$	$\alpha = 20^\circ$	$\alpha = 30^\circ$	$\alpha = 40^\circ$
0	0.24	0.58	0.59	0.58	0.59
10	0.35	0.72	0.80	0.82	0.79
20	0.62	0.70	0.74	0.78	0.73
30	0.77	0.77	0.70	0.73	0.70
50	0.77	0.78	0.78	0.75	0.75
70	0.89	0.89	0.87	0.88	0.88
90	0.92	0.92	0.92	0.92	0.92

r/R = 0.96					
Θ (Degree)	u/U				
	$\alpha = 8^\circ$	$\alpha = 15^\circ$	$\alpha = 20^\circ$	$\alpha = 30^\circ$	$\alpha = 40^\circ$
0	0.22	0.58	0.59	0.57	0.58
10	0.53	0.84	0.86	0.86	0.84
20	0.79	0.83	0.85	0.86	0.86
30	0.82	0.83	0.85	0.86	0.84
50	0.83	0.86	0.89	0.89	0.88
70	0.92	0.92	0.92	0.92	0.92
90	0.93	0.93	0.92	0.93	0.93

r/R = 0.68					
Θ (Degree)	u/U				
	$\alpha = 8^\circ$	$\alpha = 15^\circ$	$\alpha = 20^\circ$	$\alpha = 30^\circ$	$\alpha = 40^\circ$
0	0.42	0.58	0.58	0.58	0.54
10	0.70	0.83	0.84	0.85	0.83
20	0.86	0.88	0.87	0.88	0.88
30	0.85	0.87	0.88	0.88	0.89
50	0.90	0.90	0.90	0.90	0.90
70	0.93	0.92	0.91	0.92	0.92
90	0.93	0.93	0.92	0.93	0.93

r/R = 0.41					
Θ (Degree)	u/U				
	$\alpha = 8^\circ$	$\alpha = 15^\circ$	$\alpha = 20^\circ$	$\alpha = 30^\circ$	$\alpha = 40^\circ$
0	0.52	0.65	0.67	0.68	0.63
10	0.69	0.80	0.82	0.84	0.81
20	0.85	0.88	0.88	0.89	0.88
30	0.90	0.88	0.90	0.90	0.90
50	0.91	0.91	0.91	0.91	0.91
70	0.91	0.91	0.91	0.91	0.91
90	0.91	0.91	0.91	0.91	0.91

TABLE 2.

Axial wake velocity component ratios for the model in the full load condition with various height vortex generators.

r/R = 1.23			
Θ (Degrees)	u/U: h_1 = 5/8", h_2 = 3/4", h_3 = 3/4"	u/U: h_1 = 3/8", h_2 = 1/2", h_3 = 1/2"	u/U: h_1 = 1/4", h_2 = 3/8", h_3 = 3/8"
0	0.58	0.58	0.49
10	0.84	0.65	0.44
20	0.82	0.59	0.55
30	0.77	0.62	0.58
50	0.79	0.74	0.67
70	0.88	0.88	0.86
90	0.92	0.92	0.92

r/R = 0.96			
Θ (Degrees)	u/U: h_1 = 5/8", h_2 = 3/4", h_3 = 3/4"	u/U: h_1 = 3/8", h_2 = 1/2", h_3 = 1/2"	u/U: h_1 = 1/4", h_2 = 3/8", h_3 = 3/8"
0	0.61	0.60	0.59
10	0.88	0.81	0.68
20	0.90	0.78	0.70
30	0.90	0.77	0.72
50	0.91	0.83	0.76
70	0.92	0.91	0.89
90	0.93	0.93	0.90

r/R = 0.68			
Θ (Degrees)	u/U: h_1 = 5/8", h_2 = 3/4", h_3 = 3/4"	u/U: h_1 = 3/8", h_2 = 1/2", h_3 = 1/2"	u/U: h_1 = 1/4", h_2 = 3/8", h_3 = 3/8"
0	0.59	0.57	0.53
10	0.85	0.84	0.68
20	0.88	0.87	0.81
30	0.89	0.87	0.84
50	0.90	0.90	0.88
70	0.92	0.92	0.91
90	0.92	0.92	0.91

r/R = 0.41			
Θ (Degrees)	u/U: h_1 = 5/8", h_2 = 3/4", h_3 = 3/4"	u/U: h_1 = 3/8", h_2 = 1/2", h_3 = 1/2"	u/U: h_1 = 1/4", h_2 = 3/8", h_3 = 3/8"
0	0.66	0.60	0.49
10	0.81	0.80	0.58
20	0.89	0.88	0.75
30	0.91	0.90	0.85
50	0.91	0.91	0.90
70	0.92	0.91	0.91
90	0.92	0.91	0.91

The radial and tangential velocity components were measured at selected positions in the plane of the propeller disc for the full load and ballast conditions with the chosen set of vortex generators fitted to the model.

In both load conditions the radial and tangential velocity components are not significantly different from those without vortex generators and are not considered large enough to significantly change the performance of the propeller.

3.3.4 Resistance of Model

The resistance coefficients for the model in a freestream in the full load and ballast conditions both with and without the selected set of generators are plotted in Fig.7. Results are also shown in Fig.7 for the generators set at angles of incidence of 8^{o}, 20^{o}, 30^{o} and 40^{o} for the model in the full load condition. In addition, resistance coefficients are plotted for the model in the full load condition with the generators set at an angle of incidence of 25^{o}, but with their height alternately reduced and increased by an eighth of an inch.

The resistance coefficients for the model fitted with the chosen set of vortex generators are approximately 25% and 17% greater than those without generators for the full load and ballast condition respectively.

As expected, variations in either the angle of incidence or height of the generators causes corresponding changes in the resistance coefficients. The selected generator heights, and the angle of incidence of 25^{o} represents a compromise between the resistance characteristics of the model and the wake velocity distribution produced by the generators.

3.4 Extrapolation to the Ship

The principal difficulty in interpreting the model experiments is to scale the vortex generators to the ship. It is necessary to determine both the physical size of the generators and the change in resistance of the ship caused by fitting them.

Scale effects will cause differences in the relative boundary layer thickness, velocity distribution and flow directions between the model, a "smooth" ship and the actual ship. Flat plate boundary layer theories indicate that the boundary layer thickness on the model is approximately two and a quarter times greater than required by geometrical similarity for the ship at 18 knots. Although these theories neglect the influences of surface curvature and pressure gradient they may be expected to give a reasonable estimate of the variation due to scale effect.

Hull roughness on the ship will offset scale effects to some extent. The increase in skin friction caused by structural and paint roughness effects was assumed to be 0.0004[2]. Fouling of the hull surface will cause an additional increase in thickness of the boundary layer on the ship. An increase in frictional resistance of 50% was allowed to account for fouling. Together these roughness allowances would cause the boundary layer on a "smooth" ship at 18 knots to be increased by approximately 60%. The boundary layer on the model will then only be about 1.4 times thicker than required for geometrical similarity with the ship at 18 knots.

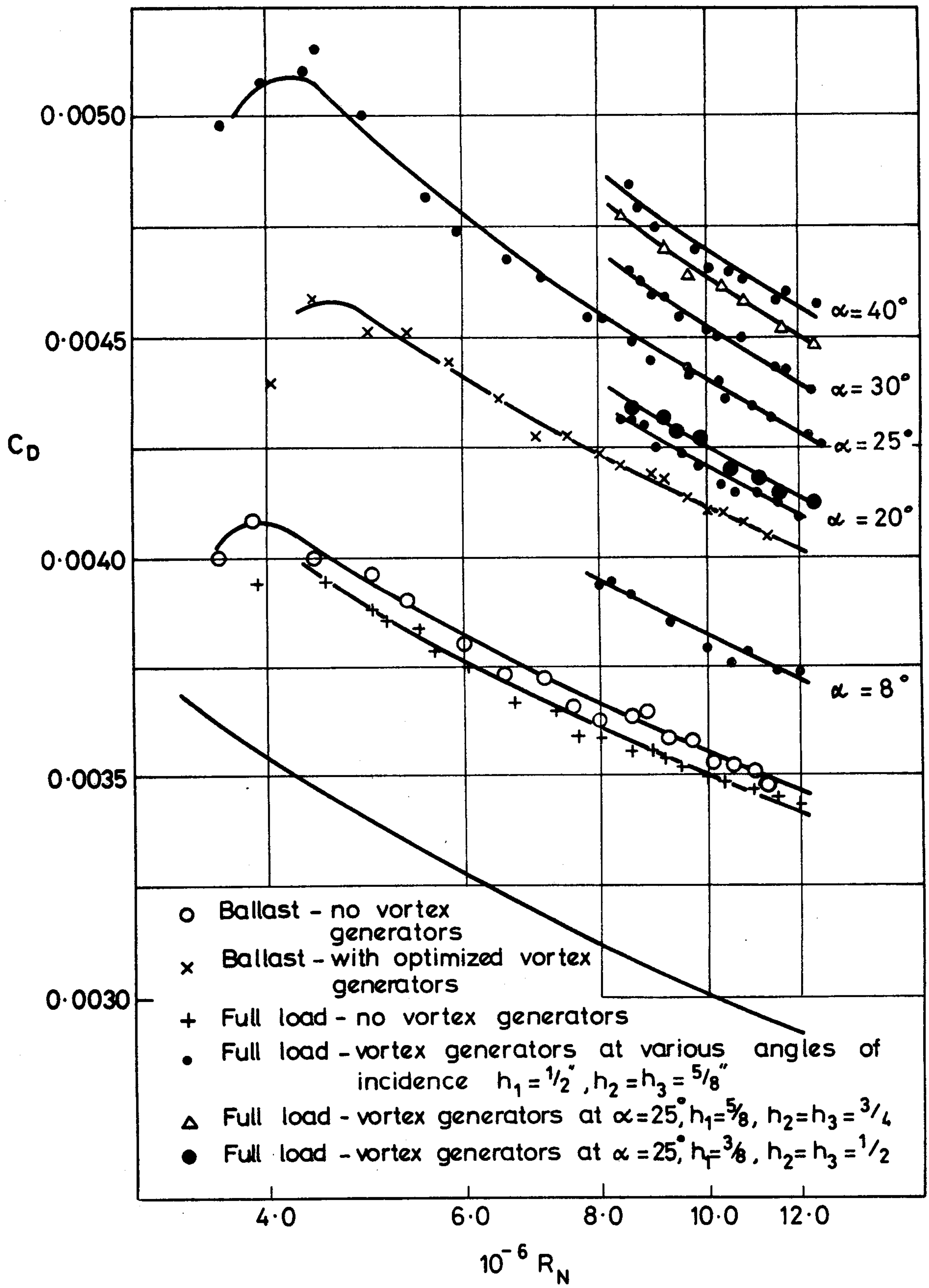


FIG. 7. RESISTANCE COEFFICIENTS FOR THE MODEL AT FULL LOAD AND BALLAST DRAUGHT, FITTED WITH AND WITHOUT THE "OPTIMIZED" VORTEX GENERATORS RESISTANCE COEFFICIENTS ARE ALSO SHOWN FOR THE MODEL IN THE FULL LOAD CONDITION WITH VORTEX GENERATORS OF VARIOUS SIZE AND AT VARIOUS ANGLES OF INCIDENCE

Using geometrical scaling the 5/8" high vortex generators on the model would become 30" high on the ship, whereas boundary layer thickness scaling for the rough hull would require the generators to be only 22" high.

In view of a lack of knowledge concerning the flow directions on the ship and because of the essentially unknown effects of wavemaking, propeller action, and asymmetric flow it was decided that the generators shown in Fig.2 should be geometrically scaled from the model and fitted to the ship.

The problem remaining is to determine the E.H.P. for the ship fitted with the vortex generators. To do this it is assumed that the increase in resistance coefficient of the model caused by fitting the generators remains constant and is applicable to the ship.

Using data for the resistance of a model without vortex generators tested in a towing tank, coupled with a roughness allowance of 0.0004, it is estimated that, at 18 knots, there would be an increase in E.H.P. of approximately 26% in the full load condition and 20% in the ballast condition. Since propulsion factors for the model with the vortex generators were not available to the wind tunnel staff, the corresponding increase in D.H.P. was not estimated. These increments in horsepower were later considered unacceptable as they would have reduced the power margin of the ship by an excessive amount.

3.5 Vortex Generator Design : Second Test Series, October 1974.

The recommendations arrived at after a month's wind tunnel testing late in 1973 were communicated to the Ship Model Experiment Tank, Vickers Ltd., St.Albans for confirmation and for the propulsion tests which had not been made in the wind tunnel. Agreement of the axial wake velocity distributions was considered satisfactory bearing in mind that towing tank Reynolds number was less than the wind tunnel value by a factor of about two and that deceleration effects due to sternwave were not cancelled by propeller suction in E.H.P. and wake tests.

Fig.8 shows some of the results obtained for (i) the original optimum recommendation from A.R.L. of six "large" vortex generators, (ii) the alternative recommendation from A.R.L. of six smaller generators set to a lower angle of incidence and (iii) the bare hull and earlier proposals for rectification of the wake. The Vickers propulsion tests revealed excessive increments of D.H.P. for the "large" generator proposal and even the alternative of six smaller generators set to reduced angles of incidence $\alpha = 18^{\circ}$ was predicted to increase D.H.P. by about 20 per cent. Vickers also advised that flow visualisation tests made with help of their underwater television camera indicated that removal of the foremost pair of vortex generators might not greatly reduce the effectiveness of the remaining four in maintaining ordered flow into the propeller disc.

At that time, i.e. March 1974, the docking of "Lysaght Enterprise" was thought to be imminent and a decision in respect of the fitting of vortex generators could not be delayed for further experiments. After re-examination of the considerable volume of model test data then in existence for the ship it was decided to instruct the builder to manufacture a set of four only reduced size vortex generating fins and to mount them at an angle of incidence

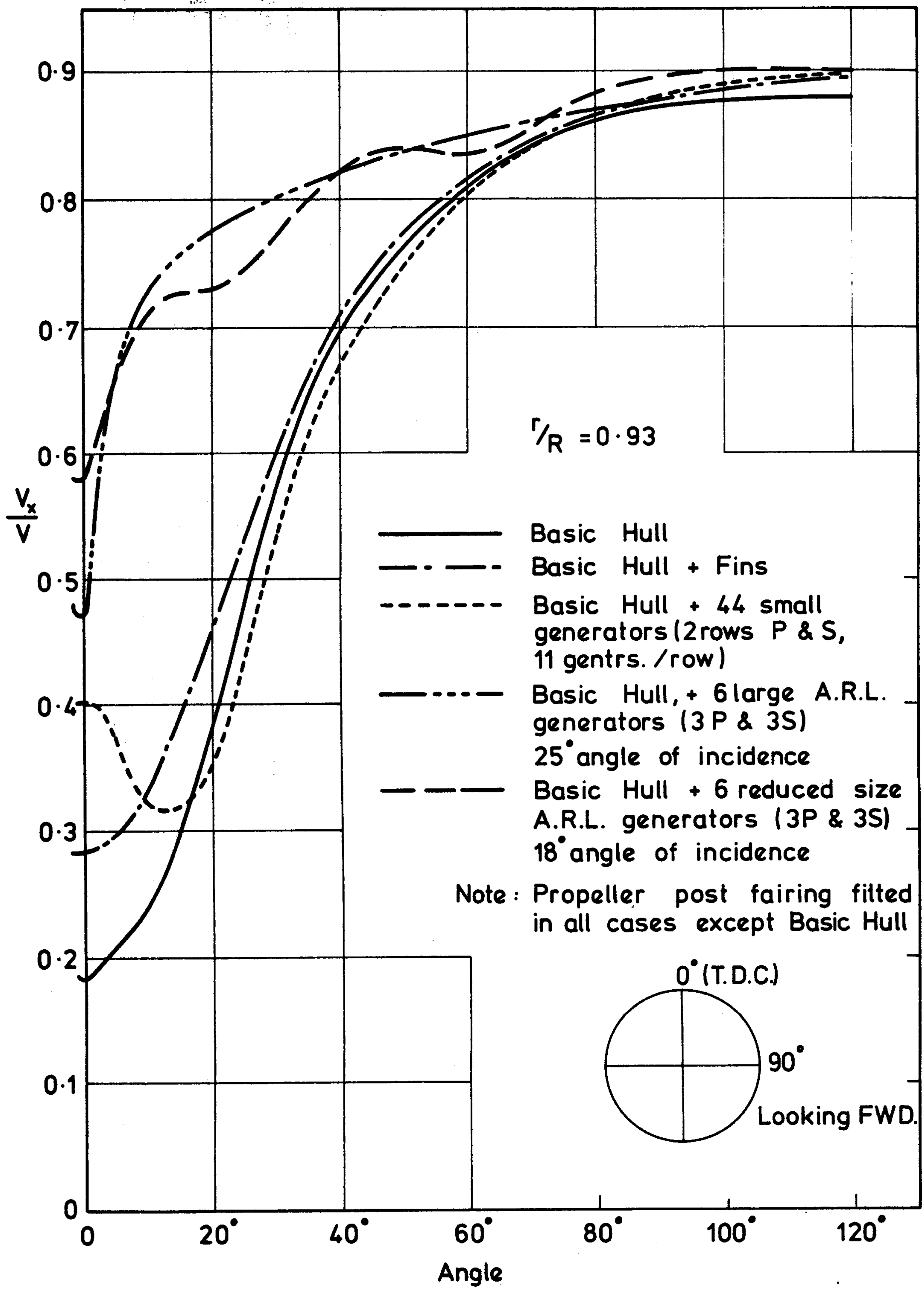


FIG. 8. AXIAL WAKE VELOCITY DISTRIBUTION TOWING TANK TESTS

of 18°. Subsequent sea trials proved this arrangement to be effective in respect of propeller wake induced vibration excitation and further wind tunnel tests were commissioned at A.R.L. in order to determine what margin of reserve there was to cater for fouling of the ship and also to optimise the suit of vortex generators within the constraint that only four generators in all were to be fitted and that power absorption was to be minimised. However, A.R.L. were unable to conduct these tests before mid-October 1974 due to other commitments on the wind tunnel. In the meantime the sister ship "Lysaght Endeavour" underwent its guarantee docking after one year in service and the question of further modification - again before appropriate test results were available - became pressing. It was thought from trials on "Lysaght Enterprise" that the absence of the characteristic cavitation noise even during rolling motion with maximum amplitude of 26° showed that her vortex generator arrangement was probably conservative. The decision for "Lysaght Endeavour" was therefore to proceed with identical vortex generators but set to incidence $\alpha = 16^{\circ}$ in order to reduce the associated power loss. Trials on that ship vindicated this decision also.

The last-mentioned wind tunnel test series is in progress at time of writing but some preliminary results are available for presentation here. Fig.9 shows the axial wake velocity distributions at radius ratio $r/R = 0.96$ for $\alpha = 13^{\circ}, 16^{\circ}, 18^{\circ}$. The uneven nature of the curves results from removal of the forward pair of vortex generators. Reduction of generator incidence from 18° to 16° appears to promote no significant change in axial velocity distribution while promising a worthwhile reduction in power loss of the order of 20 per cent since the resistance of delta-shaped fins is proportional to a power of incidence angle greater than square in the range of interest. Further reduction of incidence to $\alpha = 13^{\circ}$ appears to lead to a large increase in wake fraction near "top dead centre" and is not recommended. The effect of progressive reduction of number of vortex generators is also being examined. Fig.10 shows a preliminary comparison of 6, 4 and 2 generator arrangements at radius ratio $r/R = 0.96$ with $\alpha = 18^{\circ}$ for all the vortex generators used. It is hoped to have corresponding E.H.P. increments available for the presentation of the paper. In any case it appears that an improved version of a two-generator suit (1 port, 1 stbd.) may be evolved in the current experiments. The resulting wake velocity distribution would need to be fed into the Det Norske Veritas programmes for cavitation performance and hull pressure fluctuation intensity of the propeller and the results considered before making recommendations to the owner for the next docking of either ship.

References :

3.1 Lachmann, G.V. (ed) "Boundary Layer and Flow Control", Pergamon Press, Oxford, 1961

3.2 Comstock, J.P. (ed) "Principles of Naval Architecture", The Society of Naval Architects and Marine Engineers, New York, 1967

3.3 N. Matheson "Wind Tunnel Studies of a Ship Model using Vortex Generators to improve Wake Velocities", Aerodynamics Note 347, Aerodynamic Research Laboratories, Australian Defence Scientific Service, 1974.

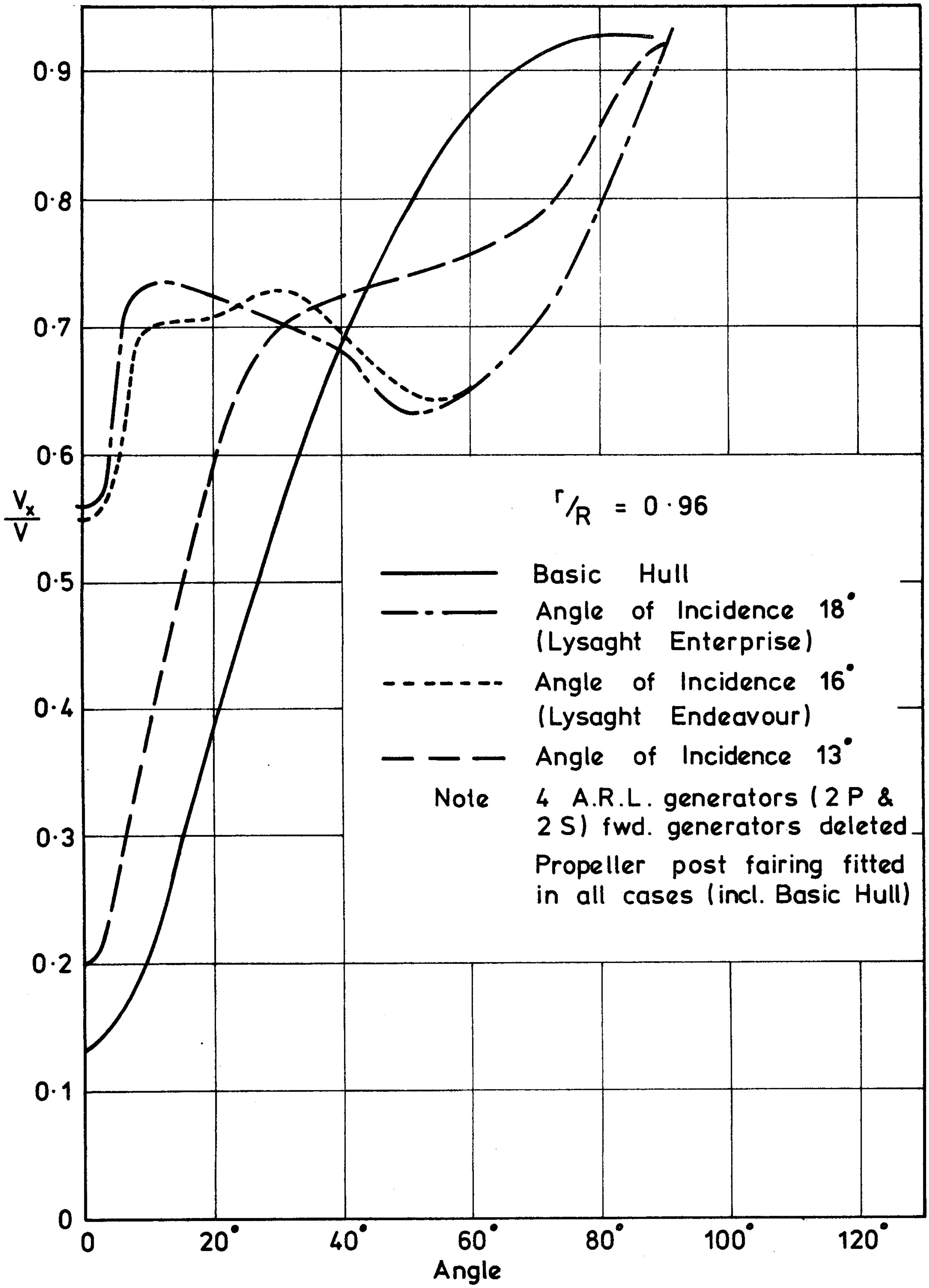


FIG. 9. AXIAL WAKE VELOCITY DISTRIBUTION WIND TUNNEL TESTS – 2nd SERIES

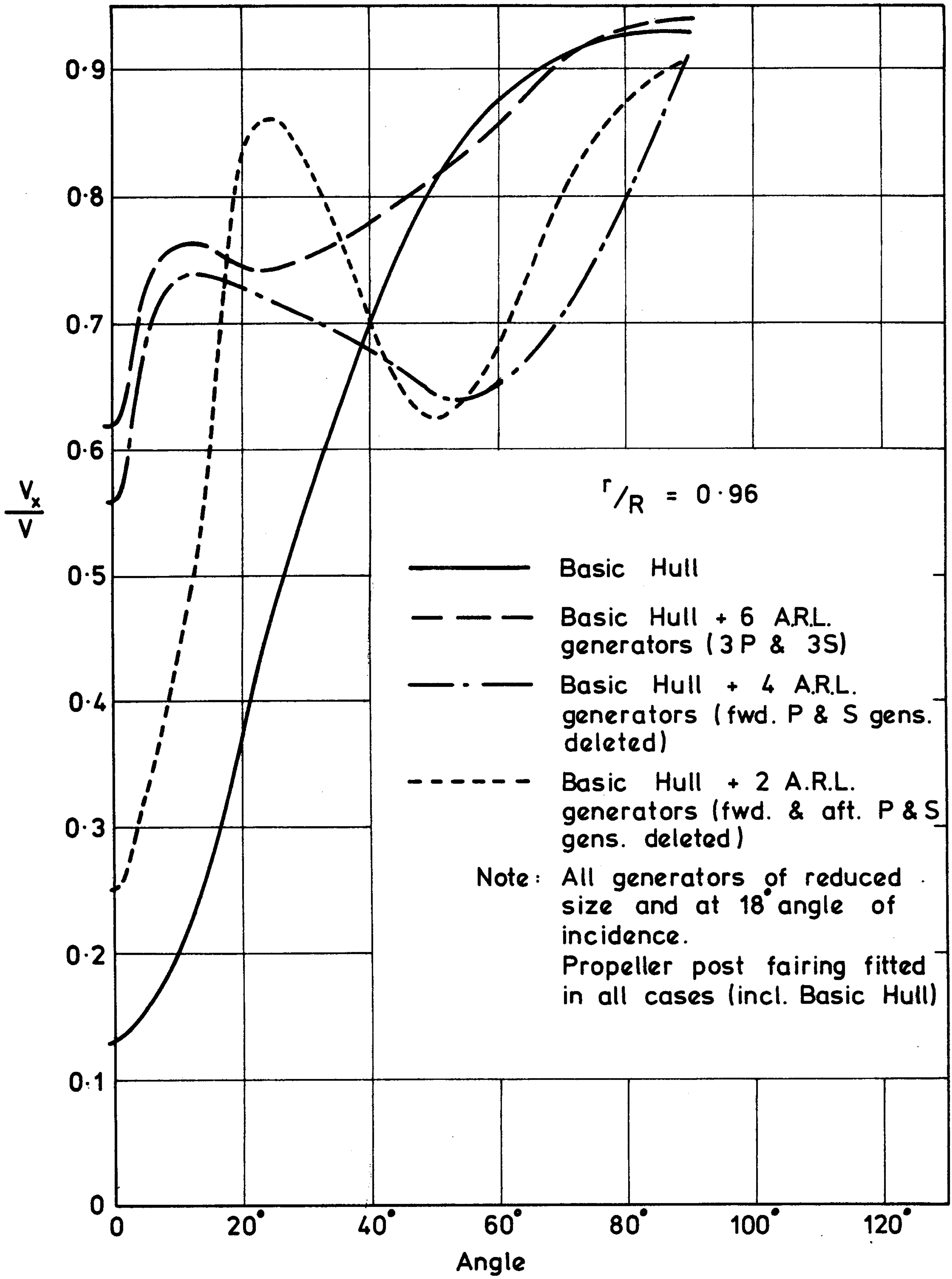


FIG. 10. AXIAL WAKE VELOCITY DISTRIBUTION WIND TUNNEL TEST - 2nd SERIES

IV., CONCLUDING REMARKS *

4.1 The Present Position and Further Work

In assessing the worth of the modifications now applied to the ships under discussion, two questions emerge, viz.

i. are the A.R.L. vortex generators effective in suppressing vibration input to the hull structure?

ii. is the accompanying power increment acceptable and can it be reduced?

The answer to the first of these questions is almost certainly affirmative. Complete confirmation could only come from a longer period of faultless operation than has taken place so far. One would need to be satisfied of the success of the "generators" under all conditions of weather, shallow water operation, unusual trim angles etc. - quite apart from the question of effectiveness in the presence of substantial fouling of the hull. However it could be said, from what voyage reports have been received to date that the indications are very favourable indeed.

The reliable determination of ships' power, or even fuel consumption is never a simple matter for reasons which will be obvious to mariners. It was not possible to run all the trials which would have been required to give thoroughly reliable estimates of the power increase associated purely with the fitting of vortex generators. What estimates there are - and these are still being analysed - include a component due to the increased roughness of the hull after one year of operation; i.e. the deteriorated surface of repainted plate-work is responsible for a residual power increment as distinct from that associated with fouling. There is much contradictory literature relating to this effect which is reported as varying between 5% and 20% after the first of an annual series of dockings.** The senior author's present interpretation of the data and of a few voyage reports is that the D.H.P. increments are approximately +15% for "Lysaght Enterprise" and +10% for "Lysaght Endeavour". This matter will become clearer as further reports are received.

The wind-tunnel evidence would appear to show that the power loss can be further reduced and it is hoped to have conclusive evidence for recommendations before the next docking by combining advice from the wind tunnel and the previously mentioned "hydrocalc" programme on the one hand and reports from ship voyages relating to vortex generator effectiveness in heavy weather on the other.

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* This section - and some others - had to be worded by the senior author without previous consultation with his co-authors due to the coincidence of a number of commitments towards the end of the period of preparation of the paper.

** See 13th I.T.T.C. Proceedings : Summary paper by M.C. Jourdain and J. J. Muntjewerf, pp.362-381.

4.2 Conclusions

It appears that the vortex generator can safely be added to the list of flow control devices available to the designer for cleaning up a faulty propeller inflow. It could be hoped that future applications of the idea might be implemented with rather less turmoil and expense than was the case here. Enough is not yet known about aft end flows to recommend the vortex generator for incorporation into the design of deliberately blunt sterns, although there may be special applications for this.

Development of a suit of vortex generators for a particular hull requires reasonably good knowledge of the boundary layer characteristics of that hull and also reasonable familiarity with basic concepts of modern fluid mechanics such as vorticity, recirculation, secondary flow etc.

The phenomena in question are well enough understood to enable some but not all judgments to be made on the basis of model tests. More research should be devoted to the problem of (Reynolds) scale effect of boundary layer development, boundary layer separation, thrust deduction etc.

The loss of confidence in single screw installations felt by some owners in various parts of the world may be misplaced in view of the combined talents, experimentally and theoretically which can now be brought to bear in order to ensure successful operation of a heavily loaded single screw.

When a problem of the magnitude described here occurs, a number of overseas consultants are likely to be involved - and properly so. Their advice will at times be conflicting. It is necessary - to put it at its lowest - to maintain a pool of expertise in Australia to enable us to make professional judgments in such situations. It is hoped that the frank descriptions in the present case study will help that cause.

V. ACKNOWLEDGMENTS

The writers are grateful to the following :

Australian National Line for permission to refer to some of their vessels

Australian Department of Supply, Australian Department of Transport, for permission for their officers to contribute to this paper

State Dockyard, Newcastle, for their cooperation

Mr. E. S. Clarke, Department of Transport, for his backing

Mr. L. G. Gibbons and staff of the Machinery Section, Shipbuilding Division, Department of Transport, for advice and help in trials

Mr. D. Howitt of the School of Mechanical & Industrial Engineering and Miss M.Pekarek who managed to assemble this report on the afternoon of its delivery

One or more of the writers have had interesting discussions about the project with the following persons and they have found these to be profitable even when they were not in agreement with some of them.

Mr. L. Brookes, Superintendent, Aerodynamics Division, Aero.Res.Labs., Melbourne

Dr. G. L. Brown, University of Adelaide

Dr. E. C. B. Corlett, Burness, Corlett & Partners Ltd., Basingstoke

Mr. R. C. Cree, Graham & Woolnough Ltd., Liverpool

Dr. A. Emerson, University of Newcastle, England

Dr. J. W. English, National Physical Laboratory

Mr. L. Hawdon, Stone Manganese Propellers, Ltd.

Mr. D. I. Moor, Vickers, St.Albans

Dr. W. B. Morgan, Naval Ship Research & Development Centre, U.S.A.

Professor J. N. Newman, Massachusetts Institute of Technology

Mr. M. Pearson, State Dockyard, Newcastle, N.S.W.

Mr. J. W. E. Pettersen, Det Norske Veritas

Mr. F. Righton, Australian National Line

Dr. H. Ritter, National Physical Laboratory, England

Mr. R. Rutherford, Swan Hunter Shipbuilders, Newcastle, England

Mr. T. Søntvedt, Det Norske Veritas

Mr. O. H. Solumsmoen, Det Norske Veritas

Mr. K. J. Wood, Australian National Line

APPENDIX I

CAVITATION - THE FUNDAMENTAL PROCESS*

It is difficult to give a concise definition of cavitation and at the same time convey much significant information about it; let us use instead the following brief description of the main features of the fundamental cavitation process.

When a body of liquid is heated under constant pressure, or when its pressure is reduced at constant temperature by static or dynamic means, a state is reached ultimately at which vapor or gas- and vapor-filled bubbles, or cavities, become visible and grow. The bubble growth may be at a nominal rate if it is by diffusion of dissolved gases into the cavity or merely by expansion of the gas content with temperature rise or pressure reduction. The bubble growth will be "explosive" if it is primarily the result of vaporization into the cavity. This condition is known as *boiling* if caused by temperature rise, and *cavitation* if caused by dynamic-pressure reduction at essentially constant temperature. Bubble growth by diffusion is termed *degassing* although it is also called *gaseous cavitation* (in contrast to *vaporous cavitation*) when induced by dynamic-pressure reduction.

In this treatise we will be concerned principally with the events concomitant to dynamic-pressure changes such as occur in hydrodynamic and acoustic pressure fields. Moreover, we will be concerned with both the rising and falling pressures in such cases, for if a growing bubble is subjected to a pressure increase its growth will be arrested and reversed. The bubble will collapse and possibly disappear by solution of gases and condensation of vapor. Collapse occurs "implosively" for a vapor-filled cavity with negligible gas content and less so if the gas content is high. Thus cavitation involves the entire sequence of events beginning with bubble formation and extending through cavity collapse. By contrast, in the usual boiling process vapor bubbles grow continuously. Rather than collapsing violently, growth and coalescence yield vapor masses that condense slowly.

Examined critically, the description is seen to contain a number of pertinent facts and ideas. For example :

1. Cavitation is a liquid phenomenon and does not occur under any normal circumstances either in a solid or a gas.

2. Cavitation is the result of pressure reductions in the liquid and thus presumably it can be controlled by controlling the amount of the reduction, or, strictly speaking, the minimum absolute pressure. If the pressure is reduced and maintained for sufficient duration below a certain critical pressure, determined by the physical properties and condition of the liquid, it will produce cavitation. If not, no cavitation will occur.

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* From : KNAPP R.T., DAILY J.W., HAMMIT F.G. "Cavitation", McGraw-Hill Inc. 1970.

3. Cavitation is concerned with the appearance and disappearance of cavities in a liquid. Note the term "cavity". In comparing the meanings of "hole" and "cavity", Webster's dictionary states in part, "Cavity is a more learned word ....; it connotes particularly hollowness or empty space". In many respects, the choice of the term cavitation for this phenomenon is a happy one because it emphasizes this concept of emptiness. It is simple to infer that, if cavities are truly empty, the contents can play no active part in the physical phenomenon. Therefore, all the observed effects of cavitation should be traceable to the behaviour of the liquid. As described already and as will be discussed later, when the details of cavitation are examined, this concept of the cavities as empty voids is not strictly true. However, over most of the life of a cavity, its contents play only a minor role. The important exceptions to this statement are found in the very beginning and the very end of the cavity cycle, i.e. when the cavity dimensions are microscopic or submicroscopic.

4. Cavitation is a dynamic phenomenon, as it is concerned with the growth and collapse of cavities.

Some pertinent information may also be obtained from examining what this description does not contain. Some of the important omissions are :

1. There is no indication whether the liquid is in motion or at rest. Thus it may be implied that cavitation can occur in either case.

2. There is no indication that the occurrence of cavitation is either restricted to or excluded from solid boundaries; therefore, it would seem that cavitation may occur either in the body of the liquid or on a solid boundary.

3. The description is concerned with dynamics of cavity behavior; a distinction is implied between the hydrodynamic phenomenon of cavity behavior and its effects such as cavitation erosion.

The previous description of the vaporization-collapse cycle is the basic characteristic of cavitation, and in many cases the phenomenon is manifested completely by the simple cycle dynamics of small bubbles. At advanced stages beyond inception, hydrodynamically produced cavitation may become more complex in ways described later. However, all the general conclusions reached above still apply.

The previous description has identified boiling, cavitation (vaporous), and gaseous cavitation as related phenomena even if not identical in all respects. Another related case is the large quasi-steady cavity maintained by *ventilation*. This important situation is obtained under some circumstances if a continuous flow of gas is naturally drawn, or artificially fed, into a hydrodynamically caused low-pressure region behind a body. Large ventilated cavities have many of the same general features as certain advanced-stage cavities resulting from vaporization except at the downstream end of the cavity where gas is removed by entrainment without condensing.