CONTRIBUTIONS ON NEW HYDRODYNAMICS TEST FACILITIES (Nos I-6 are in Vol.I)

7. STATE OF THE ART OF SHIP VIBRATION ANALYSIS, HYDRO-ELASTICITY AND CONSEQUENCES FOR THE RELATED MEASURING TECHNIQUES IN TOWING TANK FACILITIES

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INTRODUCTION

The analysis and judgement of ship vibration is an interdisciplinary operation, ranging from hydrodynamics and cavitation, through hydro-elasticity and mass-elastic mechanics to ergonomics and human sensibility.
All involved disciplines have approached those parts of the problem in relation to their specialisms.
Ergonometrists have determined a maximum allowable harmonic vibration level, to which a human being may be exposed for shorter or longer times.
Practical ship-builders have analysed a large number of ships on their vibrational behaviour. As a result of this endeavour an "I.S.O." norm for maximum acceptable vibration level aboard ships is given in Fig. I /I/.
The structural engineer faces the problem how to analyse the vibration level to be expected, starting from the hull structure and the propeller excitation.
This effort is generally undertaken in close cooperation with Naval Architects and Hydrodynamicists /2/.

I) Critical consideration in ship vibration analysis.

In Fig.2 (from reference /2/) a review is given of the various steps involved in ship vibration analysis. From the review can be concluded that besides the pure hydrodynamical and mechanical problems a number of mixed hydro-elastic problems exists.
In particular the dynamics of the propeller and the hull are affected by hydro-elastic effects, normally indicated as added mass- and damping- coefficients (see par. 2).
In Fig. 3 a review is given of the existing sequence of evaluation of the vibratory operation of the ship to be built.
It may be noticed that a strong accent exists on the hydrodynamic side of the evaluation.
In particular the vibratory hull pressure distribution plays an important role. A criterium, including the hull elasticity e.g. a criterium on global hull vibration, also belonging to the I.S.S.C. domain is worthwhile to be incorporated in the existing diagram. The vibrating hull and its surrounding water is of equal importance for the ITTC and ISSC.
For the time being the ITTC work in the area of propeller vibrations can be presented by Fig. 4.
In this figure all hydro-elastic effects have been avoided by considering only the

solid propeller and the solid hull.
Although a large number of important and challenging problems do exist in the research of the "Solid Ship and propeller" hydrodynamics, essential for future developments, the restrictive condition that the structural flexibility is negligible needs to be eliminated for a successful development in the domain of vibration analysis. This does not only hold in the case of the analysis of full size ships, but also in the experimental techniques used in towing tanks.
Since the models in the test arrangement do have flexibility it might be questionable to assume rigid bodies.
Whether or not the rigid body assumption is acceptable depends on the type of measurement and the design of the measuring arrangement. In particular when time dependent phenomena are to be measured the body flexibility may play an important role.

2) Experiments and hydro-elastic effects.

Hydro-elastic effects can mainly be categorised into two effects i.e.
- Hydrodynamic reaction pressures, generated by vibrating boundaries, due to non-circular vortex free flow.
- Hydrodynamic reaction pressures, generated by vibrating boundaries , due to the non-circular flow and the effect of trailing vortices.

The latter type of reaction pressures are mostly encountered in vibrating profiles (propeller blades, hydrofoils, rudders) and may give rise to an instationary operation at the critical speed (flutter)/3/.

In this contribution the attention will be restricted to the first type of hydroelasticity. The effect is generally described as "added mass" and "damping". In fact these effects are distributed pressures , generated by the water due to the vibratory motion of the boundary and are partly in phase with the acceleration (mass effect) and partly in phase with the speed of vibration (damping).
In order to illustrate the implication of hydro-elasticity, met in experiments, an example as shown in Fig. 5, will be used. A fluctuating pressure is exerted on a body, supported by means of a structure and a force pick-up.
The body is assumed to be solid, the structure has elasticity.
In the body also a pressure pick-up is installed.
The dynamics of the entire arrangement are presented in the block diagram of Fig. 6.

P_p equals the excitation pressure to be measured
P_o equals the output of the pressure pick-up.
F_o equals the output of the force pick-up.
A equals the area of the body
M equals the mechanical mass of the body
μ equals the added mass related to the body acceleration $\ddot{w}$
D equals the damping related to the body speed $\dot{w}$
w equals the body displacement
S equals the stiffness of the supporting structure
ω equals the frequency of excitation in rad per sec.
ω_0 equals the natural frequency of body and structure

For harmonic signals and linear behaviour the following transfer functions can be derived.
The output of the pressure pick-up P_o is related to the interesting quantity P_p as follows:

$$\frac{P_o}{P_p} = \frac{1 - \frac{M}{S}\omega^2}{1 - \frac{M+\mu}{S}\omega^2 + i\omega\frac{D}{S}}$$

We want to have $P_o = P_p$. This can only be approached by either $\omega \to 0$ or

$$\frac{M}{S} < \frac{M+\mu}{S} \to 0,$$

which means that a qualified measurement can only be performed when the frequency tends to zero (a static measurement, which is not our interest), or by the condition S is very large, so that

$\frac{M+\mu}{S}\cdot\omega^2$ and as a consequence $i\omega\frac{D}{S}$

are very small compared to unity. In other words $\frac{\omega^2}{\omega_0^2} \Rightarrow 0$ or the natural frequency of the measuring arrangement needs to be very large as compared to the frequency of measurement.
Also the ratio $\frac{F_0}{P_p \cdot A}$, being the key for a qualified measurement of the integrated pressure excitation, needs to approach unity. This condition is also fulfilled when

$\frac{\omega^2}{\omega_0^2} \Rightarrow 0$ a similar requirement as for the measurement of P_p.

Referring now to towing tank experiments, it can be stated that for seakeeping tests the frequencies of measurement are so low that the mentioned requirement is fulfilled without special measures, although care should be taken when very high excitation frequencies are present (springing, slamming).
For the measurement of the fluctuating propeller forces the required mechanical condition of sufficient high natural frequency can only be obtained with an adequate design of the mechanics of the measuring arrangement.
For the measurement of the fluctuating propeller generated hull pressure distribution the required stiffness and mass-elastic properties of the arrangement (natural frequencies) are determined by the mechanical properties of the model /4/ and are not so easy to be influenced.
It can be stated that the lowest natural frequency of the model will certainly be much lower than the blade frequency, being the lowest frequency of interest when hull pressures are to be recorded. This condition makes it impossible to measure the hull pressure fluctuations directly, and comparisons of direct recordings of pressure phenomena on model and full scale are not meaningful.
Dynamic calibration of the model - vibration - introduced - pressure phenomena are a requisite for an improved method of experiments.
A detailed lay-out of the calibration procedure is given in ref. /4/.

3) Recommendations.

) It is recommended that uncertainties in the prediction of the propeller generated hull pressure fluctuations are eliminated. In the practical frequency range (i.g. 5-30 c.p.s. for model scale), model experiments in depressurized facilities suffer from an unknown dynamic response of the model structure.
The measured pressures are partly distorted by the hull vibration of the model. The measurement of the true pressure excitation requires a model having a lowest natural frequency well above (5-I0 times) the maximum frequency of interest (say 200 Hz). This leads to an impossibility and other ways of experiments are necessary. With a too weak model the response on the true excitation is measured, in this case the response of the model-structure. (The response consists of a mixture of vibratory motions, vibratory pressures due to the surrounding water, e.g. added mass, and a remainder of the true excitation pressures). A possible solution may be found in the inversion of the response of the model. The fortunate condition exists, that for a model a dynamic calibration is possible. From this calibration the response functions are known. An inversion of this calibration may lead to

the possibility to calculate the true propeller generated hull pressure fluctuations from the measured response of the model structure.

b) It is recommended that besides the existing criteria on the hydrodynamic excitations (hull and shaft forces) a criterion on global structural vibrations will be developed, to eliminate as far as possible the uncertainties the ship designer faces when a ship is designed and constructed. When the structure of the hull and the lay-out of the ship is known in an advanced design stage, an approximate calculation of the global vibration characteristics is possible. Together with the hull excitation from the propeller an analysis can be made of the main structural forced vibration level.

c) It is recommended to have the attention focussed in hydro-elastic problems in dynamic towing tank experiments. In this respect it is recommended to have one of the committees involved in the study of these problems and to have the subject considered formally by the I.T.T.C.

List of references.

/I/ I.S.O. Draft Proposal Document No. 6954, Revised Sept.1979 "Guidelines for maximum hull vibrations for ships longer than 100 metres"

/2/ Wereldsma, R.: "Ship vibration, state of the art 1979" Publication No.M38 of the Netherlands Maritime Institute, Rotterdam, 1980.

/3/ I.S.S.C , 1982, Report of committee II.4 (vibrations) (to be published)

/4/ Wereldsma, R.: "Experiments and interpretation of propeller vibratory hull pressures on an elastic structure of a ship model", Conference on "Advances in Propeller Research and Design" Gdansk, Poland, Febr. 1981.

List of figures.

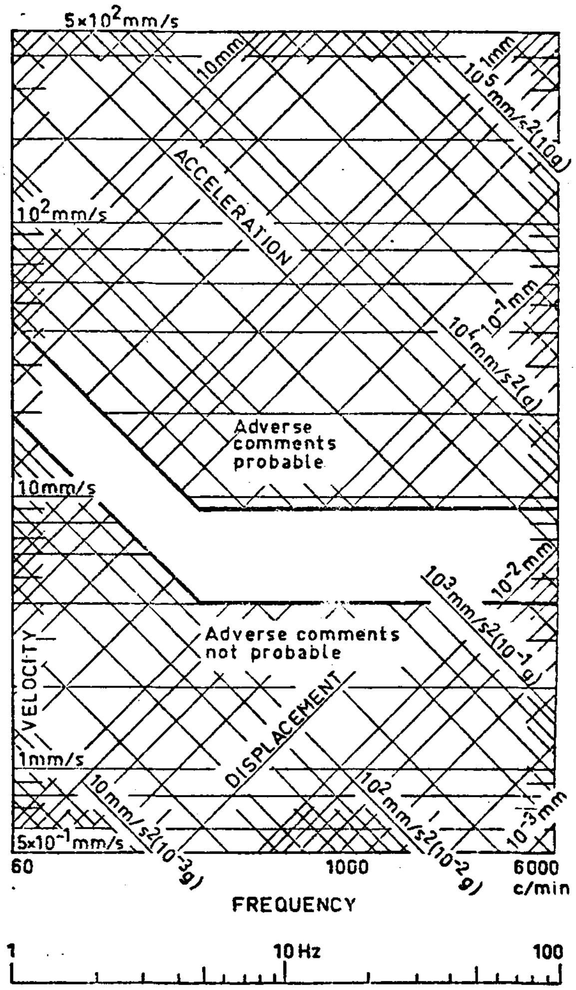


Fig.1 I.S.O. guide lines for maximum hull vibrations for ships longer than 100metres.

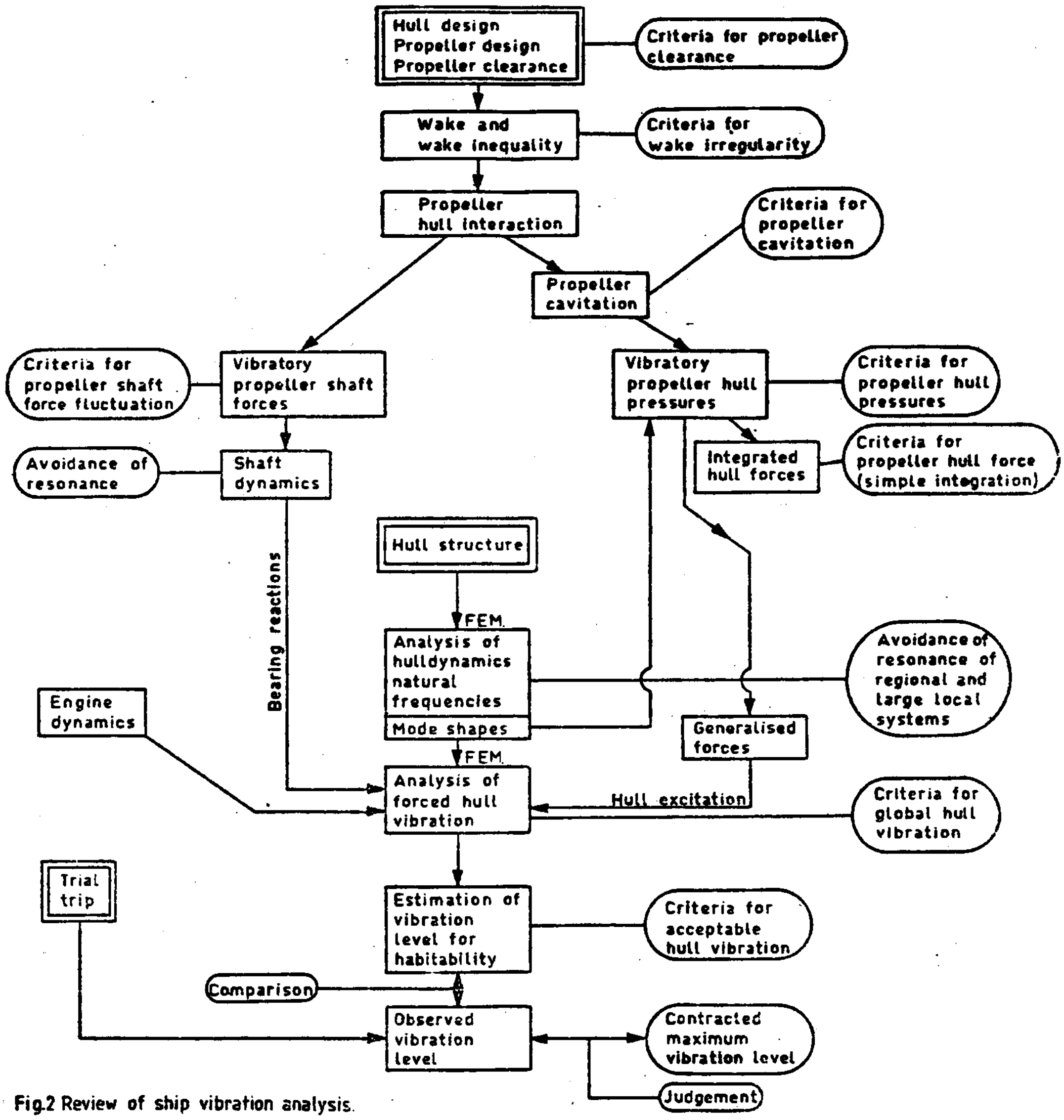


Fig.2 Review of ship vibration analysis.

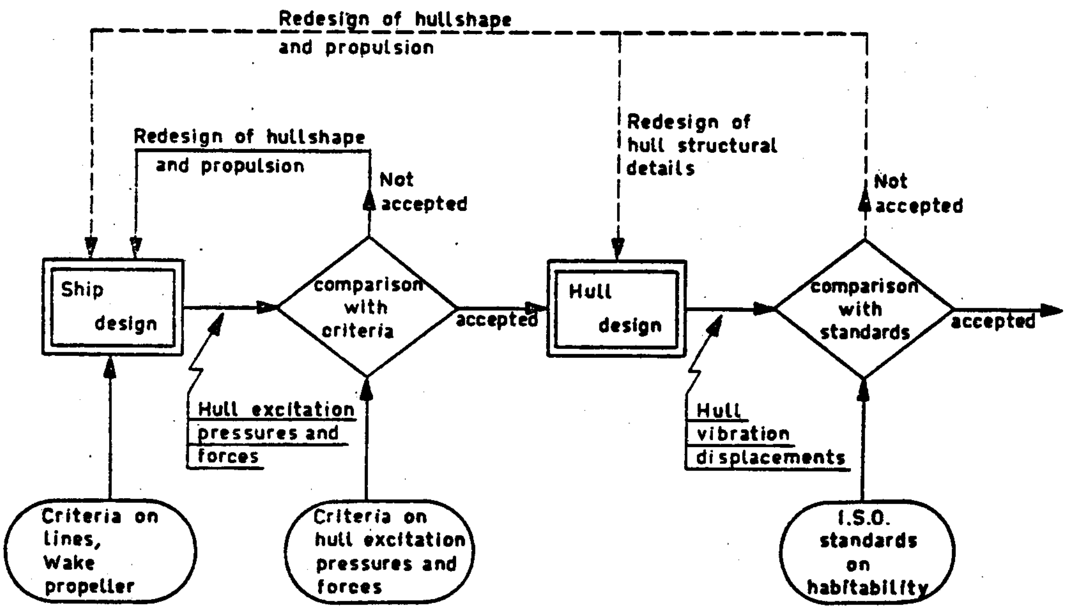
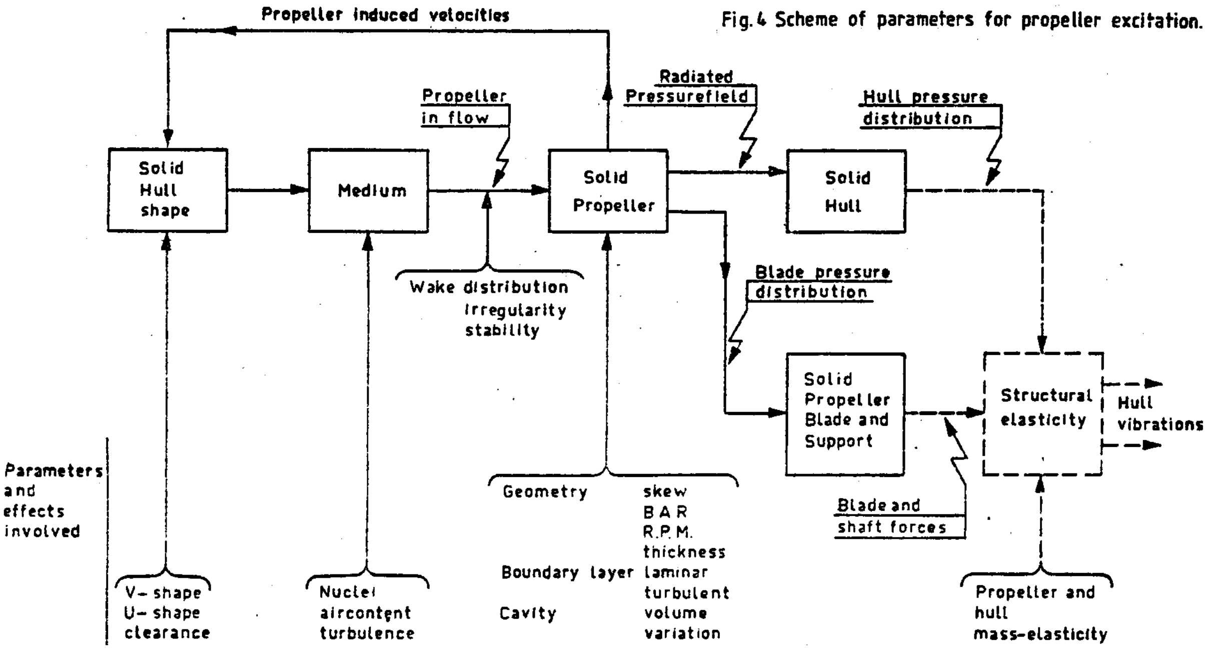


Fig.3 Existing design criteria for ship vibration evaluation.

Fig.4 Scheme of parameters for propeller excitation.



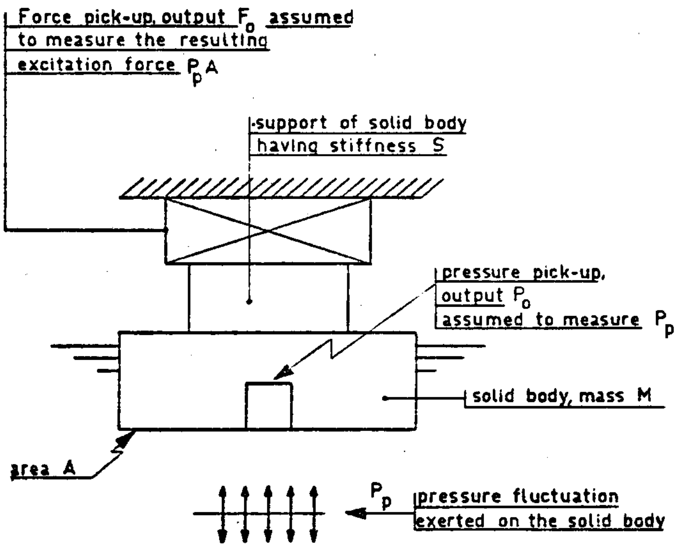


Fig.5 An example for the measurement of fluctuating pressures.

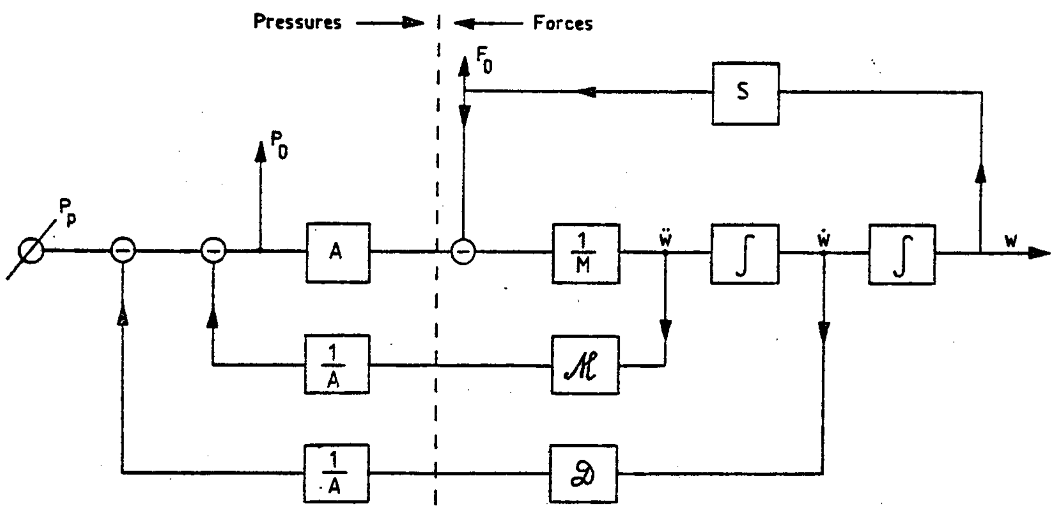


Fig. 6 Block diagram of the dynamics of the measuring arrangement of Fig.5