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ZAGREB

PAPER No 9

216/2411/814.

met appendix.

Dynamic Measurements on Propeller Models

by

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Paper to be presented at the Symposium on the Towing Tank Facilities, Instrumentation and Measuring Technique
Zagreb 22—25 September 1959.

DYNAMIC MEASUREMENTS ON PROPELLER MODELS

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Joint Communication of the Netherlands Ship Model Basin, Wageningen, and the Netherlands Research Centre T.N.O. for Shipbuilding and Navigation.

Summary:

The properties of the propeller, which have to be known for an investigation of the crankshaft - thrustblock - propeller-shaft - propeller system, are briefly discussed. The development of an apparatus for measuring dynamical thrust and torque on propeller models and the electronic part of the measuring system are described. Finally some measuring results are given.

Introduction:

Some recent cases of vibration troubles with crankshaft caused that interest was focussed into research on vibrations of the propeller - propellershaft - engine - system of a ship. This interest is also stimulated by the development of the diesel-engine.

For low speed single-stroke diesel engines of 12 cylinders an output of 15.000 shp must be considered as a maximum value for cylinder diameters of 740 - 780 mm. The demand for diesel engines with still higher outputs will lead to cylinder bores of 840 - 900 mm. in the near future and an increase of the output per cylinder by 40 per cent. With these high power diesel engines, having ten or more cylinders, a combination of axial and torsional vibrations of the crankshaft sometimes occurs which cannot be explained by the existing views and theories.

Under the auspices of the Netherlands Research Centre T.N.O. for Shipbuilding and Navigation a committee "Fluctuating forces in Shafts" has been formed. The research program of this committee includes three items, viz.:

1/ An experimental investigation on crankshaft models.
2/ An experimental investigation for the determination of the fluctuating forces in the shaft, excited by the propeller model.
3/ A theoretical investigation with the purpose to develop a method for the calculation of the vibrations of propeller-shaft and crankshaft, based on the results of the investigations mentioned under item 1 and 2.

The problems of measuring technique, connected with the experimental investigation - item 2 - to be carried out by the N.S.M.B. are dealt with in this paper. Item 1 and 3 will not be discussed in this paper.

I. The properties of the propeller to be known for the investigation of the crankshaft.

For an exact determination of the systematical load variations of the crankshaft model to be investigated, it is necessary to know the force-variations in the shaft caused by the propeller with respect to thrust, torsion and bending. These force variations can be considered to be periodical and a harmonic analysis into some components is possible. Besides the mutual difference in phase of these harmonics and the correlation with the position of the propeller model are important for the testing of the crankshaft model. Also the added mass and the damping coefficient of the propeller model, for an axial as well as for a torsional vibration, have to be determined.

Little is known about the added mass and the damping coefficients of ship propellers. [1] [2] .

Another committee of the Netherlands Research Centre T.N.O. for Shipbuilding and Navigation has been occupied during the last years with an experimental investigation into the added mass of the "Wageningen B-Series" propellers. A preliminary report of the results of this investigation has been completed. [3].

A method of calculation of the added mass and the damping coefficient of the ship propellers has been put forward by the N.S.M.B.[4] . Starting from the equation for a vibrating flat plate the added mass for the propeller could be calculated with the aid of a "strip-theory". The same equation gave a good definition of the damping coefficient of the propeller, but the three dimensional character handicapped the calculation of the damping coefficient according to a "strip-theory".

The recording and analysis of the force-vibrations in the shaft, excited by the propeller model, is a kind of experimental research which has been developed in the different naval research institutes during the last years. [5] [6] [7] [8] [9] .

II. Development of the dynamometer for measurements of the dynamical thrust and torque.

a/ Historical review:

The problems in measuring techniques to be solved in the design of a dynamometer for dynamical thrust and torque are numerous.

The highest frequency to be recorded is of much importance for the construction of the apparatus, in connection with the dynamical character of the phenomena. For this reason the highest possible stiffness of the shaft is required in order to obtain a high natural frequency of the device. The propeller models on which the measurements are made runs at about ten revolutions per second. For the five bladed propeller the blade frequency is 50 c.p.s., where as 100 c.p.s. occurs still as an important frequency. It is desirable to have the opportunity of recording frequencies up to 200 c.p.s. With a great stiffness of the mechanical system resonnance or overshoot phenomena can be shifted to a frequency range, which is of no importance for the dynamical thrust and torque measurements on propeller models.

Due to the great stiffness the construction of the pick-ups according to an inductive or capacitive principle is most obvious.

Schuster, who designed the first pick-up for this type of measurements preferred the inductive measuring principle, as no collector rings are required and background noise in the faint measuring signal can be kept small, such that the ratio of background noise and the desired signal is favourable.

Lerbs and Krohn [6] chose a combined capacitive and inductive measuring principle and applied a cylindrical condensor, resp magnetic transmission instead of collectior rings. They attained a natural frequency of 400 c.p.s. for the thrust as well as for the torque measuring device with a white metal propeller model in water. The transient thrust dynamometer of Tachmindji and Dickerson, operating on the principle of a differential transformer, had a natural frequency of 245 c.p.s. for a mass of four pounds [7] . By order of the N.S.M.B. the Institute T.N.O. for Mechanical Constructions developed a measuring device with strain gauges for recording the dynamic thrust and torque. This choice has been made due to the great experience of the manufacturer in this field. A natural frequency of 300 c.p.s. for axial vibrations and of 500 c.p.s. for torsional vibrations has been attained for aluminium propeller models in water.

b/ <u>Properties of the measuring device:</u>

The variations of the forces, created by the propeller are measured by pushing a spring. The tension variations of this spring are converted by means of strain gauges to an electrical signal.

The combination of the spring with the propeller as a mass has a transfer function of a second-order system. Schematically the device is given in fig. 1.

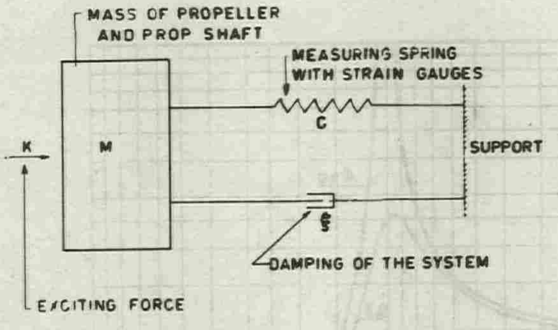

Fig. 1. - Schematic diagram of measuring system.

For this system holds:

$$\frac{Xu}{K} = \frac{1}{C_1} \quad \frac{1}{\left(\frac{j\omega}{\omega_o}\right)^2 + 2\beta\left(\frac{j\omega}{\omega_o}\right) + 1}$$

in which:

- Xu = output signal of the strain gauges.
- K = exciting force.
- C_1 = static calibration factor.
- ω_o = natural frequency $= \sqrt{\frac{c}{M}}$
- c = spring constant.
- M = mass of the propeller.
- β = damping coefficient defined as $\beta = \frac{1}{2} \frac{\xi}{\sqrt{cM}}$
- ξ = damping of the system.
- ω = frequency of the exciting force.

The well-known transfer functions of this second order system are given in figure 2. From this figure it will be seen that for properties below the natural frequency it is desirable for the transfer accuracy of the amplitudes to make $\beta = 0,6$. In order to record correctly the phase of the signal $\beta = 0$ is the desired

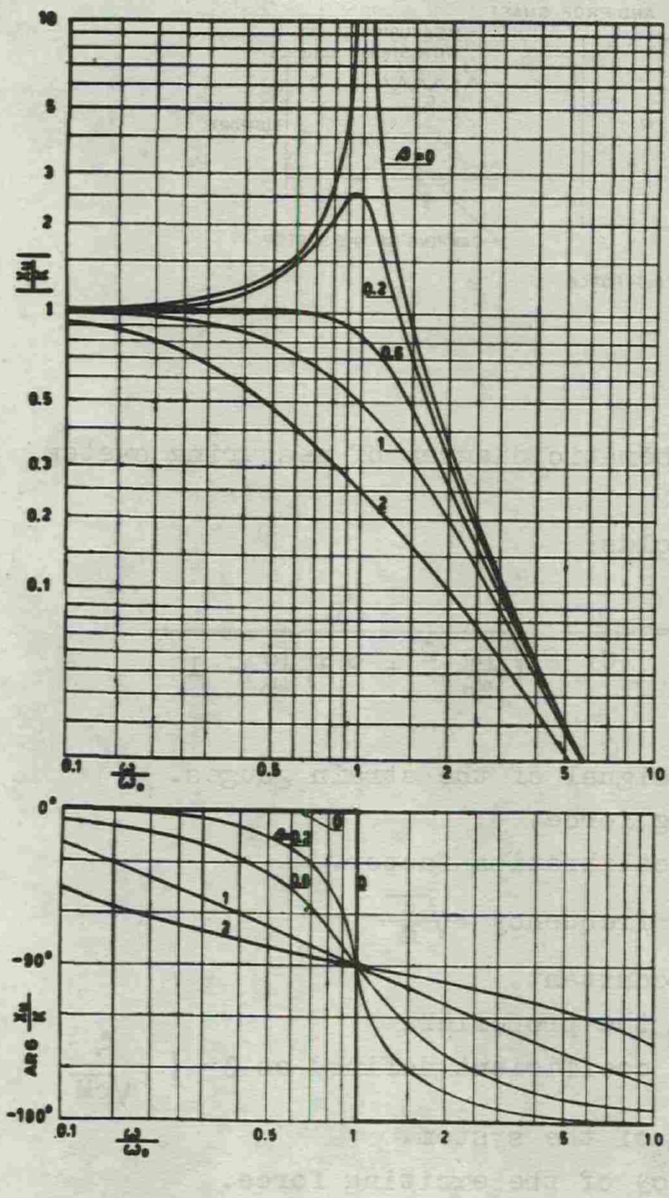

Fig. 2. - Transfer functions of second order system.

value. In the measuring system as applied here the damping is determined by the properties of the propeller model in water. From experiments this damping-factor appears to be smaller than 0,6 , sothat overshoot effects occur. For this reason the natural frequency has to be chosen as high as possible. The natural frequency increases with the spring stiffness. A more rigid spring results on the other hand into a less sensitive measuring element, sothat the ratio between signal and noise of the measurements becomes less favourable. The choice of the spring-stiffness is thus based on a compromise. However by reducing the mass of the propeller the natural frequency of the system can be increased without influencing the ratio between signal and noise. From this respect the propeller mass has to be kept as small as possible. For this reason the propeller models for these measurements are made in aluminium. The noise-level, which determines the sensitivity of the measuring element at given force-variations depends on mechanical and electrical disturbances. The mechanical construction must be carried out with the greatest care, sothat vibrations are avoided, in order to minimize these disturbances as much as possible.

c/ The design and manufacture of the apparatus for measuring the dynamical torque and thrust:

The design has been based on measuring with strain gauges and transmission by collector rings.

In the design of the instrument serious attempt has been made to obtain natural frequencies of 400 c.p.s., as a compromise, whereas the output of the torque- and thrust instrument was adjusted at 2 mV representing resp. 36 kgcm. and 25 kg.

The unavoidable misalignment of the propeller shaft and a minimum shaft length of approximately 40 cm. have also been strongly determining for the construction. The misalignment, which has been tolerated in the construction is 0,75 mm.

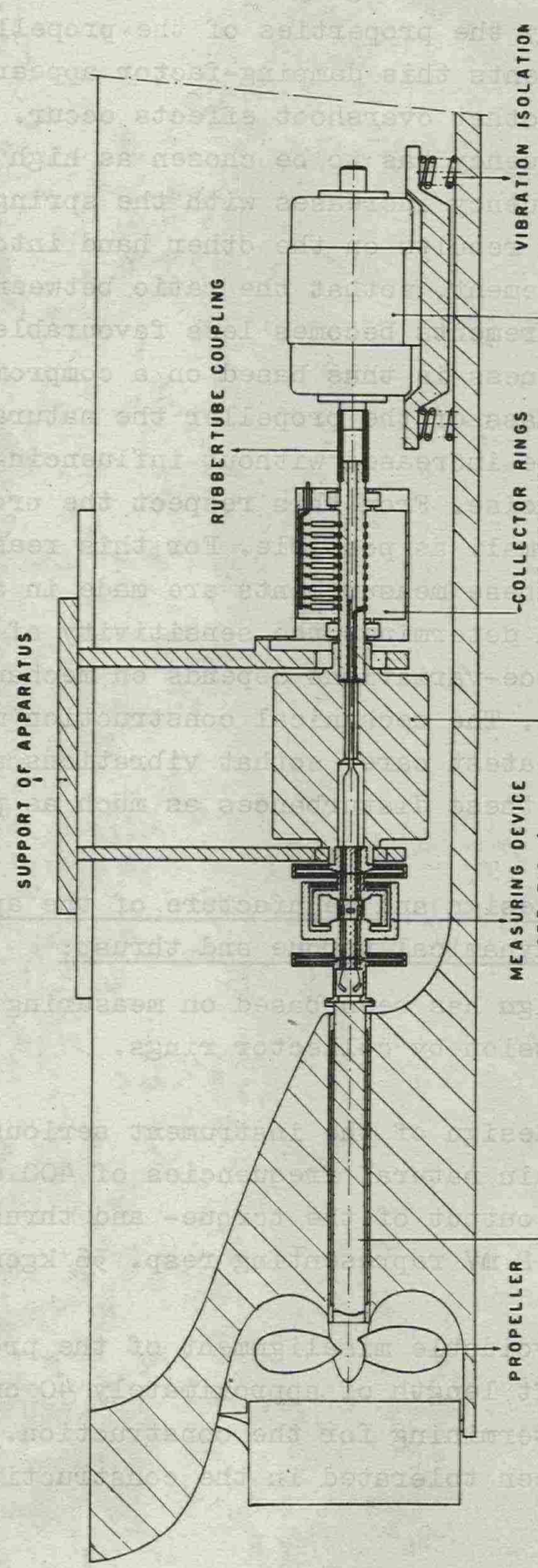

Fig. 3. - Dynamic thrust and torque measuring device.

From the schematic arrangement given in fig. 3 the great care taken with the isolation against mechanical vibration is clear. A flywheel, in combination with a non-rigid coupling /rubber tube/ between the driving electric motor and the flywheel, isolates eventual vibrations excited by the motor. In order to avoid also the transmission of these vibrations via the "ship's hull", the electric motor has been placed on foamrubber /indicated in the sketch as springs/. In the arrangement from propeller to flywheel the highest possible rigidity has been aimed at in order to obtain high natural frequencies.

The measuring strain gauge - bridges have separate feedings. The applied strain gauges are of the type AB7, manufactured by Baldwin. For the electronical connection between these bridges on the rotating shaft and the electronic equipment silver slip-rings each having 3 silver coal brushes have been applied. The bridge resistances are 360 Ω and 240 Ω , whereas the brush to slipring contact resistance amounts to approximately $2.10^{-3}\Omega$

The torsional natural frequency of the system is largely determined by the moment of inertia of the propeller model /aluminium/ and the shaft /stainless steel/ and further by the stiffness of the propeller shaft and the torque measuring element. The torsional stiffness of the thrust element is very high, because the protecting covers of the thrust element have been built as a system with only one degree of freedom, the axial displacement / see fig. 4/. A simplified calculation, carried out for a solid shaft of 4 cm. diameter, gives a natural frequency of 475 c.p.s. Actually this appears to be 500 c.p.s.

The axial natural frequency is largely determined by the masses of the propeller model and the shaft, and by the stiffnesses of the torque cylinder, the thrust cylinder and the special thrust bearings in the membrane couplings /see Fig. 4/. These membrane couplings form together with the pre-stressed thrust bearings a coupling, which is stiff against axial and torsional deformations, but not against bending.

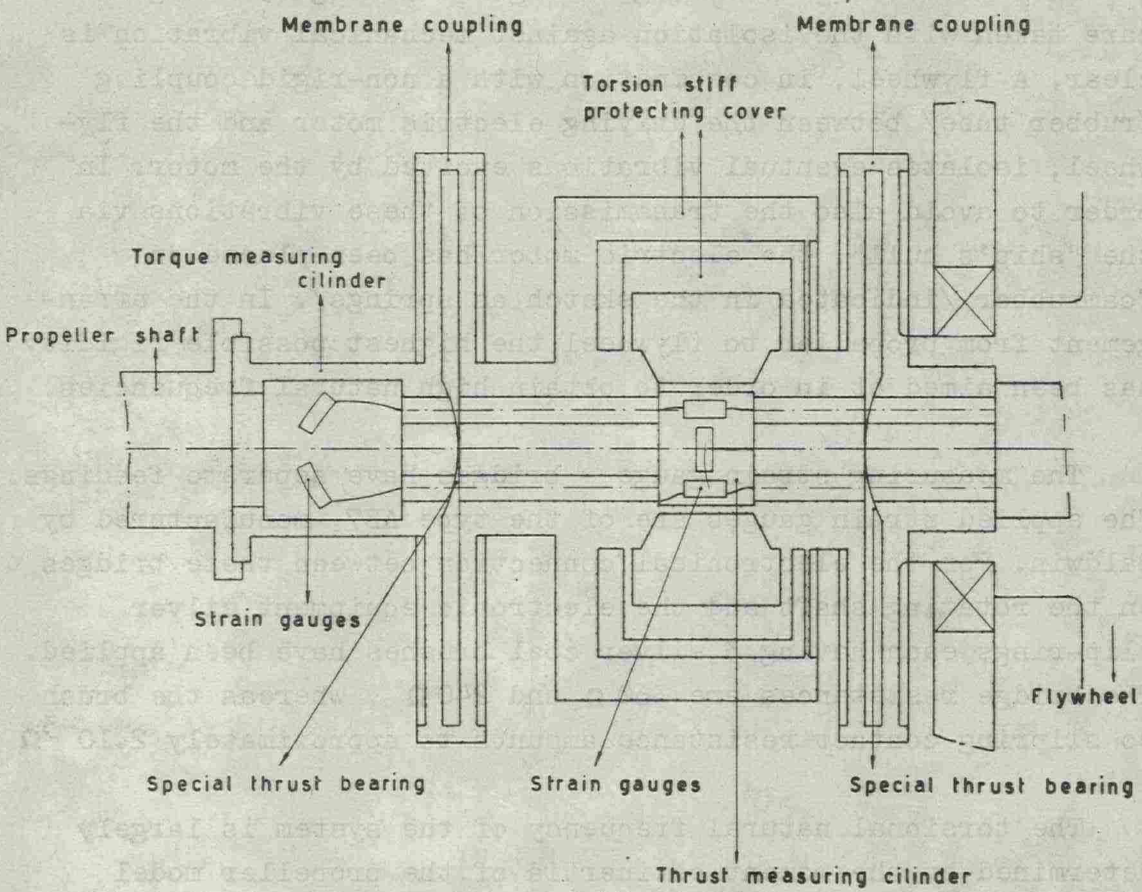

Fig. 4. - Detail of thrust and torque pick-up.

A certain inaccuracy in the alignment of the shaft is allowed for in this way. The special thrust bearings rolling off on each other when running in a not perfectly aligned condition, are an uncertain factor in the calculation. It appears that the natural frequency for axial vibrations is 300 c.p.s. and that for torsional vibrations about 500 c.p.s. By boring out the propeller shaft it is possible to increase the natural frequency in axial direction at the sacrifice of the torsional natural frequency. Both frequencies will then become about 450 c.p.s. Up till now the measurements are carried out with a solid shaft.

III. The electronic measuring system.

The electronic part of the measuring apparatus is shown in figure 5. The signal of the strain gauges is amplified by a carrier-amplifier with a bandwidth of 500 c.p.s. and made visible on an oscilloscope, /Tektronix type 545/ after having passed a filter and a marker. The signal is recorded by means of a Siemens high speed camera, with a film speed of 733 cm/sec., which is mounted on the oscilloscope.

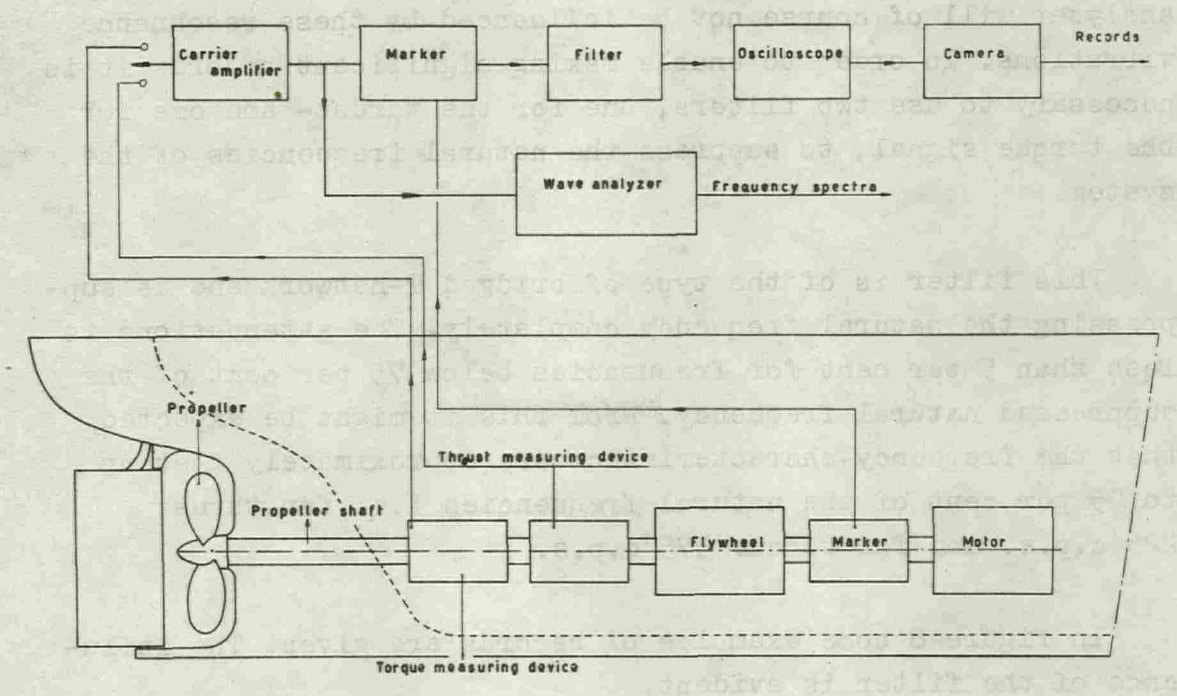

Fig. 5. - Block-diagram of electronic equipment.

It appeared to be possible to record sufficiently free from noise 1 kg. thrust and 2 kgcm. torque which are represented by approximately 1 cm. shift on the screen of the oscilloscope.

A selective measuring apparatus /wave - analyzer/ detects the farmonic components of the screw-excited force vibrations up to 200 c.p.s. From these measurements the frequency-line - spectra can be composed.

In addition to the periodical attack of the propeller by the medium also random phenomena are present in the flow around the propeller, due to turbulence. Therefore the measuring system is not only periodically excited up to frequencies of 200 c.p.s. but also up to the natural frequency of the device and higher by noise. Due to the magnification factor of the measuring system /sub critical damping/ at frequencies in the vicinity of the natural frequency the periodical signal of the thrust and torque variations is disturbed by the noise generated natural frequency.

The results of the measurements below 200 c.p.s. with the wave-analyzer will of course not be influenced by these resonnance vibrations. In order to enable making significant records it is necessary to use two filters, one for the thrust- and one for the torque signal, to suppress the natural frequencies of the system.

This filter is of the type of bridged T-network and is suppressing the natural frequency completely. The attenuations is less than 5 per cent for frequencies below 75 per cent of the suppressed natural frequency. From this it might be expected that the frequency-characteristics are approximately flat up to 75 per cent of the natural frequencies i.e. for thrust 225 c.p.s. and for torque 375 c.p.s.

In figure 6 some examples of records are given. The influence of the filter is evident.

The position of the propeller-blades relative to the screw aperture in correlation to the signal is also given in fig.6. When one of the propeller-blades passes the top of the screw-aperture a short impuls is superimposed on the recorded signal, sothat one revolution is clearly marked.

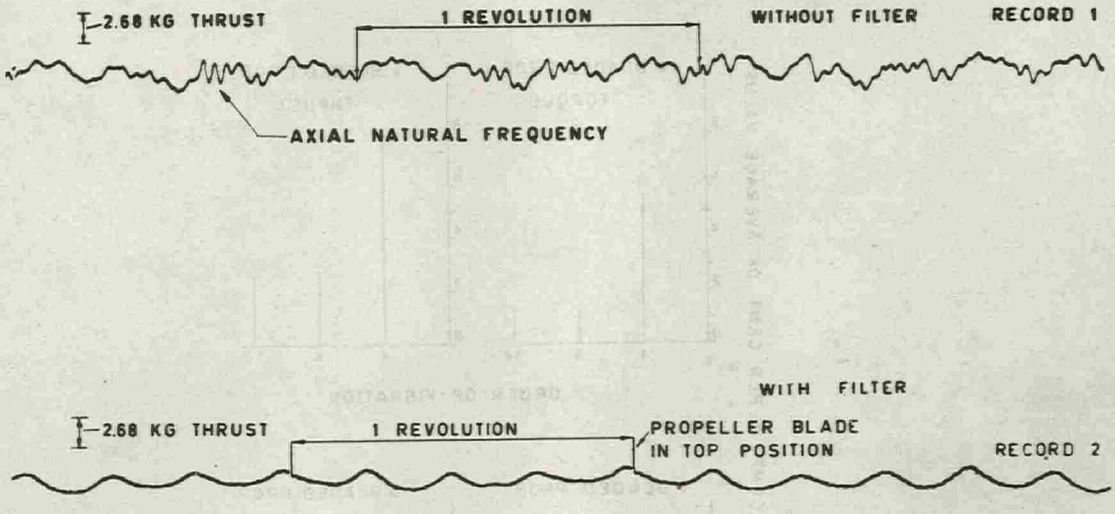

Fig. 6. - Records showing the influence of the filter.
Record I : without filter.
Record II : with filter

IV. Results and final remarks

With four bladed propellers, the first harmonic component of the excited thrust and torque variations is in general the most important one. This is caused by the fact that two blades are passing the screw aperture and thus a highwake region at the same time.

With five bladed propellers every tenth of a revolution a blade is passing the screw aperture, either in the vertical upper or the lower positions. From this fact it will be clear that both the second and the first harmonic of the thrust and torque variations are important.

The differences between four and five bladed propellers are clearly shown in the line spectra of fig. 7 and the records of fig. 8.

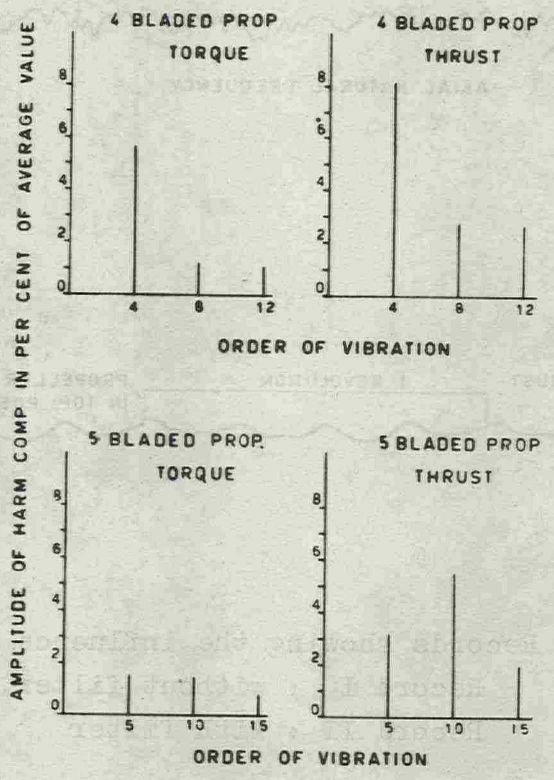

Fig. 7. - Line spectra of 4-and 5 bladed propeller.

Records no. 3 and 4 of fig. 8 show that the maximum thrust and torque values of the four bladed propeller occur before the blades pass the screw aperture. This phase shift is in agreement with results of flutter calculations [10].

The random character of the records for the five bladed propeller makes the determination of the relation between propeller position and peak values of thrust and torque impossible.

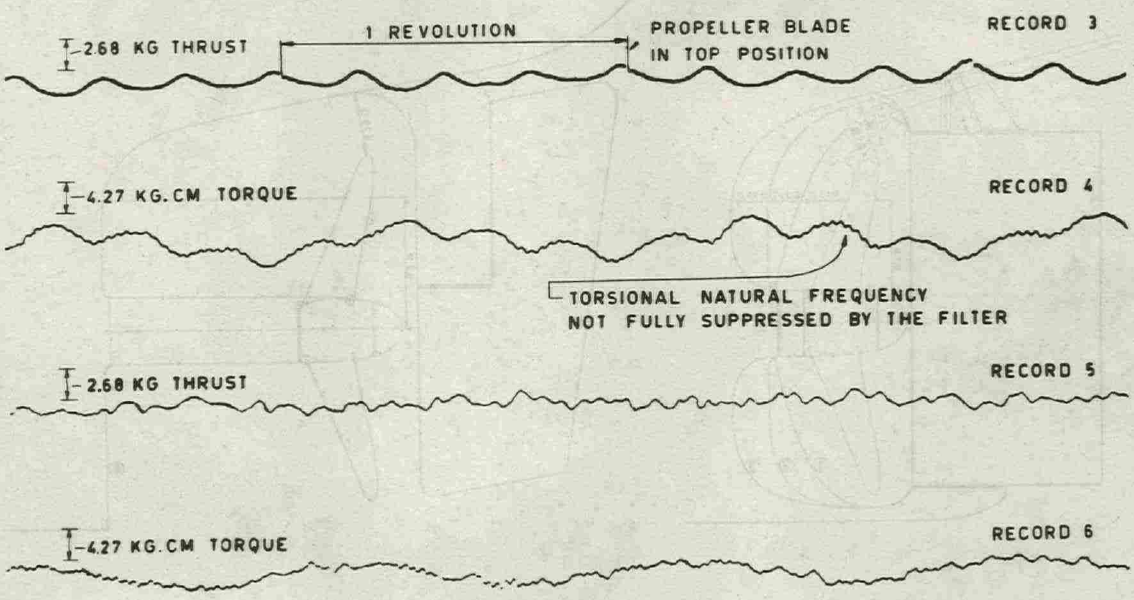

Fig. 8. - Records of a 4- and a 5 bladed propeller.

Record III: thrust variations of a 4 bladed propeller.
Record IV : torque variations of a 4 bladed propeller.
Record V : thrust variations of a 5 bladed propeller.
Record VI : torque variations of a 5 bladed propeller.

From the records of the torque signal it can be seen that a rather large variation occurs each revolution. This variation is caused by the bearing friction of the propellershaft.

In fig. 9, 10 and 11 some results are shown of a systematic research into the influence of clearances of the screw aperture on the amplitudes of the thrust and torque variations. The results will be published in full detail in a communication of the Netherlands Research Centre T.N.O. for Shipbuilding and Navigation

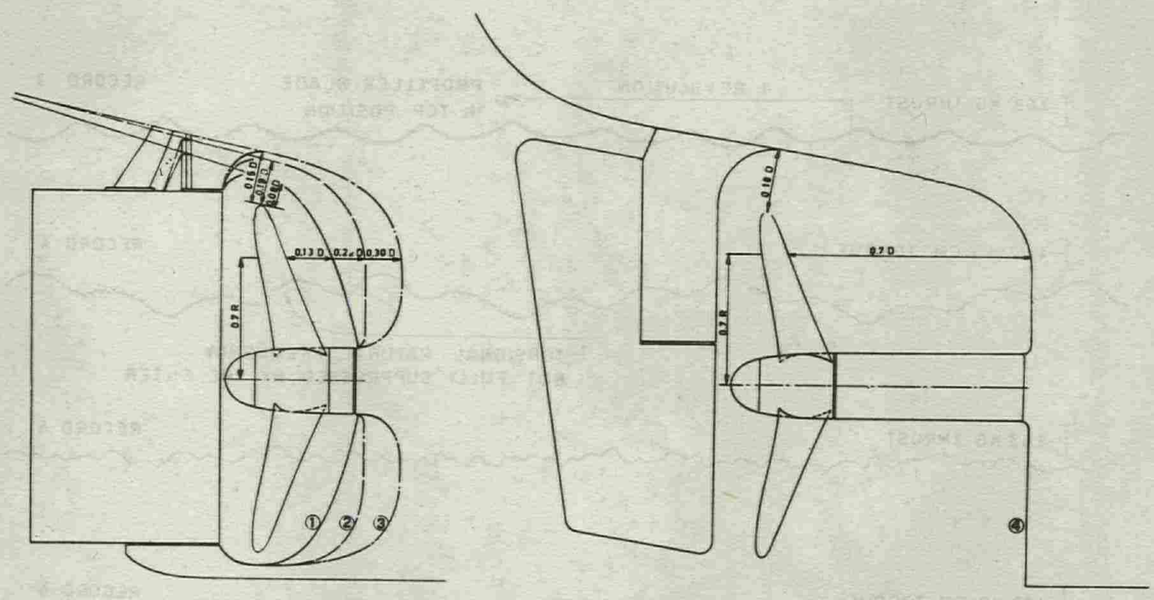

Fig. 9. - The examined screw apertures.

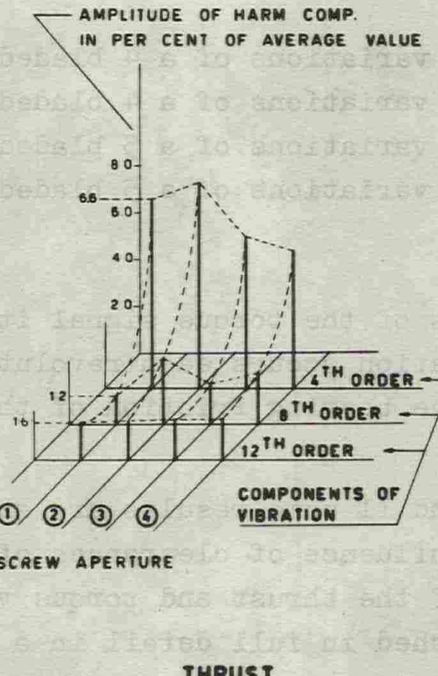

Fig. 10. - Line spectra of the 4 bladed propeller as a function of the type of screw aperture.

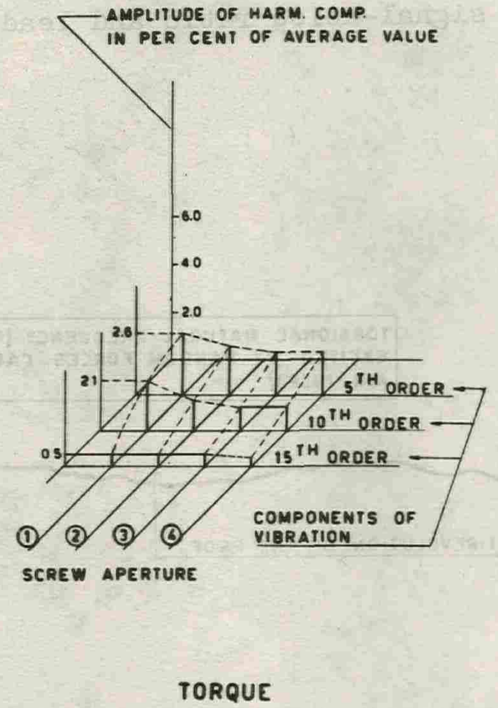

Fig. 11. - Line spectra of the 5 bladed propeller as a function of the type of screw aperture.

The excitation of the measuring system by air inlet in the screw disc is evident in the right part of the record of the torque signal of fig. 12. The natural frequency amounts approximately to 500 c.p.s.

Although the natural frequency of the N.S.M.B. system is relatively high, for a more specified analysis of the phase differences between the harmonics and propeller position and for a higher qualified analysis of the amplitudes some additional techniques are in development.

The signal to be measured can be correlated by a sinusiodal or other periodical noise-free reference, because the repeating frequency is known in advance. The reference signal has to be synchronized with the propeller shaft. This correlation gives an improvement of the signal-noise ratio and leads to high qualified records.

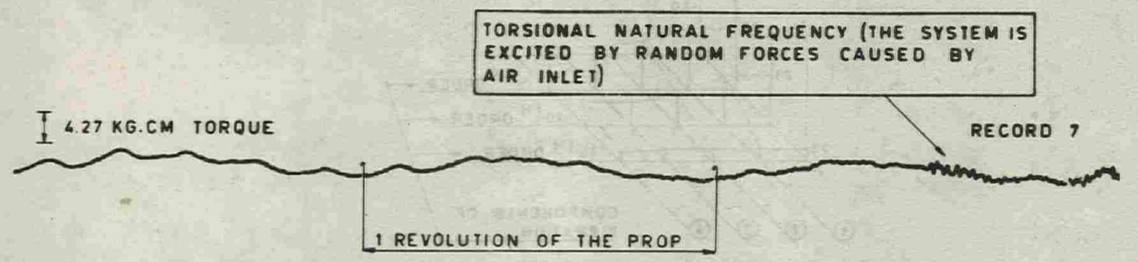

Fig. 12. - Random forces, caused by air-inlet, in the torque variations of a 5 bladed propeller.

The system under construction is working on the principle of "periodic sampling".

With the aid of this improved system a measuring device for the bending vibrations in the propellershaft could be developed.

Finally it may be mentioned that comparative tests with the different measuring systems **will** be carried out by V.W.S. - Berlin, H.S.V.A. - Hamburg and N.S.M.B. - Wageningen with a six meter single screw ship model of "araldit".

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Literature

[1] Kane, J.R. and Mc Goldrick, R.T. "Longitudinal Vibrations of Marine - Shafting systems", Trans. of the S.N.A.M.E. Vol 57 1949 p. 193-252.

[2] Lerbs, H. and Baumann, H. "Trägheitsmoment und Dämpfung belasteter Schiffsschrauben", Hydromechanische Probleme des Schiffsantriebs, Teil II, 32-45, 194o.

[3] Visser, N.J. "Onderzoek naar de vergroting van het massatraagheidsmoment van een scheepsschroef ten gevolge van het meetrillende water", Report of the Netherlands Research Centre for Shipbuilding and Navigation, November 1958.

[4] Manen, J.D. van and Vossers, G. "Calculations of the added mass of a screw propeller", N.S.M.B. report /to be published/.

[5] Schuster, S. "Schiffbauliche Modellversuchstechnik I und II", Archio für Technisches Messen, Nov. 1955; Dez.1955.

[6] Lerbs, H. and Krohn, J. "Beitrag zur Messung von Schub- und Drehmomentschwingungen am Modell", Schiff und Hafen 1957, H. 11.

[7] Tachmindji, A.J. and Dickerson M.C. "The measurement of thrust fluctuation and free space oscillating messures for a propeller",Report No. 11o7 of the NAVY Dep. of D.T.M.B.

[8] Christensen, H. "Experimental Determination of Propeller torque vibrations on ship models", Intern. Shipb. Progress 1956; Vol.3, No.20.

[9] Schuster, S. "Über die hydrodynamisch bedingsten Schub- und Drehmoment Schwankungen in Schiffsantrieb-Anlagen", V.D.I. Zeitschrift Band 98, No.32, Nov. 1956.

[10] Manen, J.D. van, "Invloed van de ongelijkmatigheid van het snelheidsveld op het ontwerp van scheepsschroeven",Report No. 100 of the N.S.M.B., see also Appendix I of: Lammeren, W.P.A. van, "Testing screw propellers in a cavitation tunnel with controllable velocity distribution over the screw disc",Trans. of the S.N.A.M.E., 1955.

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