S/N 123639 10,00

THEORETICAL AND EXPERIMENTAL STUDY ON THE BEHAVIOUR OF SHIP'S ROLL TANK STABILIZER SYSTEMS

The university degree

doctor scientiae technicarum (Dr.sc.techn.)

D i s s e r t a t i o n (B)

HABILITACJA

presented to the Research Council of the

Wilhelm - Pieck - University ROSTOCK

by Dr. - Ing. Leonard Henryk ROZENBERG

born 21.03.1952, Szczecin/POLAND

Reviewers:

Rostock/Szczecin 1989

BIBLIOTEKA
GŁÓWNA
-24-
SIN.6156

Declaration:

I hereby declare that, I realized the presented study independantly, using literature contained in the references given at the end of the study and with the aid of experiments conducted by me.

Leonard H. Rozenberg

For my Parents, wife and close friends

CONTENTS page

NOMENCLATURE

(list of the main symbols)

A_H – cross sectional area of a water duct in the tank,

A_V – cross sectional area of awing section of a tank,

B – breadth moulded,

b_T – distance between surface center in a wing section and ship's center line (tank half – breadth),

D – gain coefficient in the channel of the roll's angular velocity,

D^2 – gain coefficient in the channel of the roll's angular acceleration,

$G(x_G, y_G, z_G)$ – ship's center of gravity,

GM_o – initial metacentric height ('frozen' water),

$h_{1/3}$ – significant wave amplitude,

I_T – static moment of inertia of free surface in the tank,

I_{xx} – M.I. of ship's mass relative to the roll axis,

k – wave number ($k = 2\pi/\lambda$),

l_T – length of the center line of the tank,

M_A – gravitational asymmetry moment generated by tank,

M_s – roll exciting moment, due to the waves,

M_{STAB} – stabilizing moment,

m – mass of ship,

$m_{\phi\phi}$ – moment of inertia of added mass relative to the roll axis,

N_ϕ^* – damping coefficient against rolling,

P – gain coefficient in the channel of the roll's angle,

$\hat{P}$ – vector representing the set of construction parameters of a tank,

S_h – spectral density of wave amplitude,

S_α – spectral density of wave slope,

S_ϕ – spectral density of rolling motion,

s – inertia coupling coefficient,

t_c – close blocking valves time,

t_o – open blocking valves time,

u(t) – control signal,

V - ship's speed,

w_ϕ - non - dimensional roll damping coefficient (quadratic part),

w_ϑ - non - dimensional stabilizing fluid motion damping coefficient (quadratic part),

α_A - wave slope angle amplitude,

α_E - effective wave slope angle amplitude,

β - course angle,

β_ϕ - non - dimensional roll damping coefficient (linear part),

β_ϑ - non - dimensional stabilizing fluid motion damping coefficient (linear part),

Γ - free surface factor (relative reduction GM_o),

Δ - weight of displacement,

δt - time gate,

ϑ - angular position of the fluid in the tank,

ϑ_N - angle of a tank saturation (maximum angular position of the stabilizing fluid in a tank),

$\varkappa_\phi$ - reduction roll factor,

ρ - mass density of sea water,

ρ_T - mass density of stabilizing fluid,

τ - transporting delay of stabilizing fluid,

ε_M - phase angle between tank moment M_A and rolling motion ϕ; positive if M_A leads ϕ,

ϕ - roll angle,

ϕ_A - roll amplitude,

$\phi_{A1/3}$ - significant roll amplitude due to irregular rolling motion,

$\ddot{\phi}_{A1/3}$ - significant athwartship's acceleration due to irregular rolling motion,

ω - circular frequency,

$\omega_{\vartheta o}$ - natural frequency of the motion of the stabilizing fluid,

$\omega_{\phi o}$ - natural ship roll frequency,

∇ - volume of displacement.

1. INTRODUCTION

New classes of sea-going units characterizes the contemporary stage of devolopement of ship building. Therefore problems connected with the functions and tasks set for the new ships arise.

More and more often highly specialized modern sea-going units are being built, which are designed to minimize the loss of speed as well as to maximize safety. Passenger ships always had to be very fast, but the quick development of air transport has caused the passenger ship to increase its role as a recreation object. More and more ships are being operated to explore and exploit the sea's natural resources. The seakeeping qualities of these ships play a primary role due to the need to increase their efficiency.

Due to the aforesaid we realize that the operational requirements of modern ships and offshore vessels have given a new dimension to the problems connected with the ships functions. The question of stabilization and compensation of oscillatory motion as well as the minimization of the mostly unfavorable influence of these motions take on a greater significance.

The ship's oscillatory motions stabilization, especially, its roll and yaw is one of the basic problems connected with sailing and operating ships.

These motions can cause a range of unfavorable phenomena [12, 14]:

1) the decrease of the ship's stability, in some cases causes the capsizing of the ship,
2) additional dynamic loading of the construction of the ship's hull,
3) renders impossible or deteriorates the operation of the deck equipment (especially on drilling ships, and exploration vessels),

4) deterioration of the ship's maneuverability and the decrease of its forward speed,
5) great deterioration of passenger and cargo comfort.

The oscillatory motion of the ship has a great influence on its seakeeping qualities and in some cases renders impossible the ships normal operation and threatens its safety. Therefore the reduction of this motion is of primary importance.

The ship's excessive oscillatory motion can be reduced by:

1) suitably choosing the dimensions and parameters of the hull,
2) suitably manoeuvering the ship relative to the wave front (the changing of the ship's speed and course),
3) using special equipment which reduce these motions.

The first two methods are known as natural stabilization of the oscillatory motion. They are of very limited importance and in principle do not solve the problem of the reduction of the ship's oscillatory motion. The third method is the most effective.

The equipment used in reducing the ship's oscillatory motion are called stabilizers.

The amplitude reduction of the ship's oscillatory motion is the main task of these stabilizers. Moreover we gain additional positive secondary effects:

1) increase of the ship's sailing safety,
2) possibility of realizing the ship's tasks, in a wide range of external conditions (sea states),
3) increase of the ship's manoeuverability,
4) betters the living and working conditions of the crew as well as operation of the technical equipment installed on the ship.

Chapter 2

2. CLASSIFICATION OF THE SHIP'S OSCILLATORY MOTION. LITERATURE AND THE AIM OF THE STUDY.

By simplification we can assume the ship to be a solid body with six degrees of freedom. It can perform three linear motions along the axis of a chosen cartesian coordinate system and three angular motions around these axes. Figure 2.1 shows the basic oscillatory movements of a vessel.

In examining the problem of the stabilization of the ship's roll it is sufficient to introduce two coordinate systems: fixed (terrestrial) and a system connected with the ship.

The system connected with the ship is chosen in such a way, so that the origin of the system lies on the ship's center of gravity (rarely on the center of buoyancy) and the axes lie along the ship's main axes of inertia.

We can differentiate the following components of the ship's motion (fig. 2.2) brought about by the ship's environment:

ϕ – ship's angular motion around the longitudinal axis – ROLL,

ψ – ship's angular motion around the vertical axis – YAW,

θ – ship's angular motion around the lateral axis – PITCH,

x – ship's oscillatory motion along the longitudinal axis – SURGE,

y – ship's oscillatory motion along the lateral axis – SWAY,

z –ship's oscillatory motion along the vertical axis – HEAVE.

The composition of the above motions, represent the oscillatory motion of the ship through space. In real conditions the disturbance forces and moments are random processes, therefore the ship's motion is a stochastic process.

Among the six types of motion described above, during the ship's operation the highest values are reached by the amplitudes of the ship's roll motion.

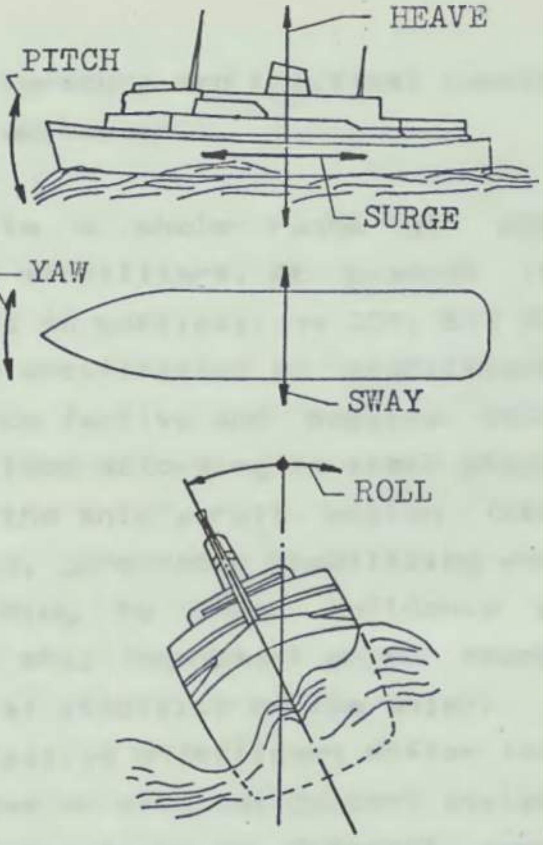

Fig.2.1. The ship's osscilatory motions.

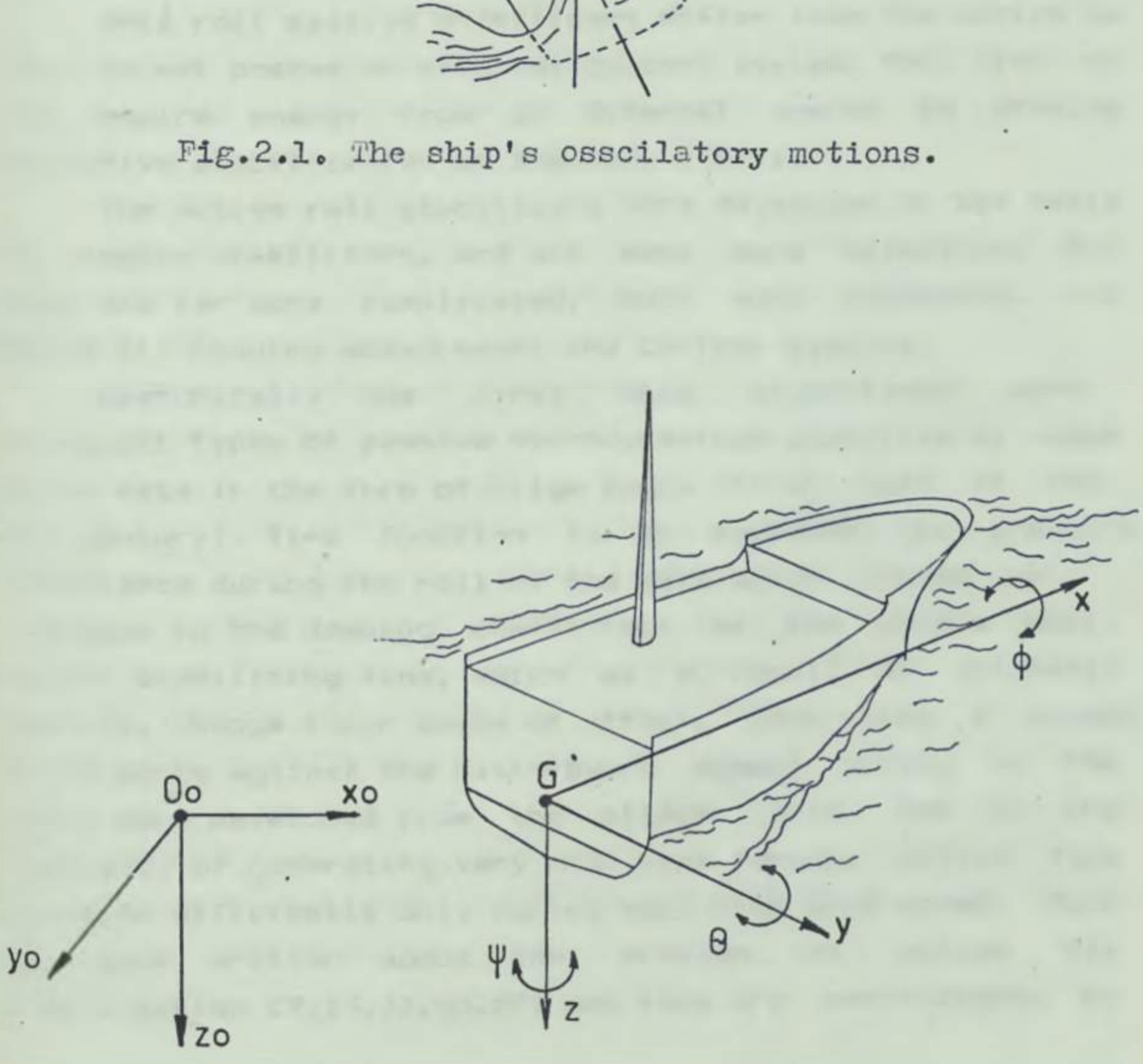

Fig.2.2. The basic coordinate systems.

2.1 Survey of literature and practical constructions of the ship roll stabilizers.

There exists a whole range of possibilities in classifying roll stabilizers. At present the most common methods described in publications [55, 61] dealing with this problem is the classification of stabilizers according to their power source (active and passive stabilizers). They can be duely divided according to their physical principles of compensating the ship's roll motion (stabilizing fins, stabilizing tanks, gyroscopic stabilizing equipment, etc.), as well as according to their influence on the lateral stability of the ship (equipment which change and do not change the initial stability of the ship).

Ship roll passive stabilizers differ from the active as they do not posses an external control system. They also do not require energy from an external source to provide effective stabilization of the ship's roll.

The active roll stabilizers were developed on the basis of passive stabilizers, and are much more effective. But they are far more complicated, much more expensive, and above all require measurement and control systems.

Historically the first ship stabilizers were different types of passive hydrodynamical stabilizers, used up to date in the form of bilge keels (first half of the XIX century). It-s function is to increase the pressure resistance during the roll of the ship which causes an increase in the damping coefficient of the ship's hull. Active stabilizing fins, which as a result of automatic control, change their angle of attack, generating a moment, which works against the disturbance moment acting on the hull, were developed from the bildge keels. Due to the necessity of generating very high lift forces, active fins function efficiently only during very high ship speed. Much has been written about the problem of active fin stabilization [9,13,37,55,57] and fins are manufactured by many firms (Siemens, Sperry, Brown-Brothers-Vickers, Muirhead, SNACH and others).

A whole group of tank stabilizers has been developed independent of fin stabilizers. They generate a gravitational stabilizing moment as a result of fluid motion. Up to the present moment, a whole range of tank types and subtypes has been developed (U-shaped tanks, Watt's type tanks, Flume type tanks). The most popular are passive tanks (especially Flume tanks). Although their effectiveness is less than that of fins, these tanks are the most common stabilizing equipment used to reduce the ship's roll today. This is due to the fact that the passive tank is the least costly and is relatively simple technologically.

The wide interest of ship owners in tank stabilizing systems is reflected in the large range of publications [3, 4,6,10,13,17,19,20,23,24,30,36,37,38,52,53,54,58,59,60,61, 65] on the theory, design and application of passive stabilizers. Authors of text books also devote much space to this problem [5, 55,61].

The lack of interest in active tank stabilizers can be explained by the fact that they need complicated control systems and high energy consumption to drive the pumps. Their theory is close to the theory of active fins. Basic information on active tank stabilizers is given in the book [55].

Passively-controlled tanks occupy a special place among ship roll stabilizers. They form a intermediate group between passive and active tanks, as they partly posses features of both groups. Various methods of influencing the natural motion of fluid in the tank are used in passively-controlled tanks. They eliminate certain negative features of active tanks (high energy consumption) and passive tanks.

The presented work deals with a certain solution of the passively-controlled tank.

There are very few publications devoted to the problem of passively-controlled tank stabilizers, other than a few positions [22, 37, 38, 52, 53, 60]. The case of the passively-controlled tank is described in the materials [37,38]. Due to the presented results of tests on the experimental ship, the cognitive value of these works is significant. The studies [52, 53] describe a ship roll U-type tank stabilizing system with the application of air valves, manufactured by the firm "Intering". The presented material is not sufficiently clear and leaves a lot of leeway in interpretation, to the reader (especially in the case of the control of the blocking valves and choice of the tank's parameters).
The publication [60] is a very interesting study of tank stabilizing systems, it also includes passively-controlled tanks. It widely presents the problems occurring during design and construction of such tanks.

Short information on passively-controlled tank is also to be found in works [13,55].

Due to the lack of publications on the above theme, a short survey of the passively-controlled tank market will be a necessary complement.
The world market in passively-controlled tanks is very much in the hands of two firms: Intering (FRG-USA) and Denny-Brown-Vickers (Great Britain). According to advertising brochures [18,51,52] of these firms, they have already sold a sufficiently big quantity of ship roll stabilizers with passively-controlled tanks. The passively-controlled tanks of both these firms are based on simple control algorithms, which have as their principle task the exclusion of the ship's roll increase in the range of low frequencies. An additional reason for the installation of this equipment is the possibility of its use in compensating the ship's static heel (anti-heeling). This quality is very useful in many cases (crane operations, quick loading and unloading of containers, etc.), and in

many cases is a deciding factor in its use, on board ship.

In the second half of the 1980's the Finnish firm HONKANEN [22] started offering its products. They are based on research (eg. [24]) conducted in the 1970's. Due to its short existence the firm has not produced many such stabilizers. However it is necessary to state the many achievements of this firm. Among them, the ample research material, as well as the publication, for the first time, of the passive control method of the Watt's type tank.

It was considered up to now that passively-controlled tanks could be built only as a U-shaped tank. Honkanen's proposition greatly widens the range of application of this type of tank and is a great achievement theoretically as well as constructionally.

Moreover it is necessary to point out the relatively new solution of the passively-controlled tank by the Flume firm (beginning of 1980's) [36], which is not a new tank, but an improvement of the old Flume tank. It is known that in the case of perpendicular walled tanks (Watt's) and flume tanks it is possible to change slightly the natural frequency of the tank by changing the quantity of stabilizing fluid in the tank. Flume therefore built a simple tri positional relay system, which could control the level of the fluid. This system uses as input the phase angle between the motion of the fluid and ship's roll angle.

At the end of this short survey of available publications we can confirm, that in the field of ship roll stabilization, the tendency is to use tank stabilizers. They are far cheaper than fin stabilizers and their operation does not depend on the ship's speed. Publications on the passively-controlled tank is scarce and mainly consist of producers' trade brochures. The absence of any sort of theoretical publication which considers the whole problem of the passively-controlled tank, inclined the author to take up this subject. The problem of the stabilization of the ship's roll during slow speeds and of the stopped ship, is of great importance due to the rapid development of drilling and mining techniques. In these conditions, tank stabilizers are the only sort of equipment that can be used on board ship and passively-controlled tanks belong to this group of stabilizers.

2.2. The aim of this thesis.

Although the passive tank stabilizer has many advantages (cheapness, simple construction), it has some defects too. The effectiveness of these tanks depends on the tuning of the dynamical characteristics of the tank, to the frequency characteristics of the ship's roll. The correctly tuned passive tank, is a tank whose natural period of the fluid motion is equal to, or close to the period of the ship's roll. Such a tuned tank stabilizes effectively the ship's roll in the resonance frequency range (in the range $0.6 \div 1.2$ of the ship's natural frequency) [13,61]. Out of this range the passive tank only increases the roll amplitudes. This quality is called destabilization and it greatly reduces the effectiveness of the passive tank in real (irregular) wave conditions. As this sometimes forces the closing down of the tank, we can call it the greatest defect of passive tank stabilizers.

In the presented work the author analyzes the operation of a tank stabilizing system that does not have this defect. He proposes the use of a specially designed U-shaped tank of high natural frequency. This tank is equipped with a set of valves which enables the blockage of the motion of fluid between the wing sections of the tank.

The aim of this work is to find out, whether there exists a physically realizable system for controlling the fluid motion, which would ensure better efficiency of roll stabilization in comparison with a correctly tuned passive tank. The improvement of the proposed stabilizing system should be looked at from the point of view of increasing its effectiveness as well as widening the frequency of its operation. Therefore it is necessary to:

1. propose the mathematical model of the dynamic system ship - passively-controlled tank as well as the algorithm of the blocking valves control,
2. work out the methodics of the synthesis of the proposed ship's roll stabilization system, which can be directly applied in ship's design,
3. realize the control algorithm in the real conditions and carry out tests of the tanks operation on scale models.

2.3. General characteristics of the ship's roll stabilization.

The stabilization of the ship's roll motion is a range of problems, which has as its aim the definition of the dynamic system which ensures the optimal filtration of random disturbances acting on an object [16,21].

In the ship's roll stabilization system, the ship as a control plant, is a dynamic system which can be described by the state equation in its general form:

$$\frac{dX}{dt} = F(X,U,Z), \tag{2.1}$$

where X is the state vector.
The state variables are the angle and linear coordinates and their derivatives. The number of state variables depend on the degree of simplification of the mathematical model.
Fig. 2.3 shows schematically the ship as a control plant in the roll stabilization system.

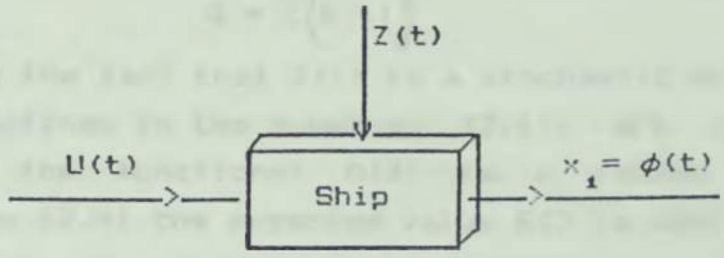

Fig. 2.3 The ship as a control plant in the roll stabilization system.

The output of the plant is the roll angle ϕ, which is the phase variable x_1. The control vector U(t) represents the stabilizing moment M_{STAB}, generated by the stabilizer installed on the ship. Due to certain construction limitations the maximum moment generated by the stabilizer is limited. Therefore:

$$|M_{STAB}| \leq M_{STABMAX} \tag{2.2}$$

The maximum value of the moment generated by the stabilizer is a function of the vector $\bar{P}$ which represents a set of construction parameters of the stabilizer:

$$M_{STABMAX} = F_1(\hat{P}) \tag{2.3}$$

Z(t) is the general vector of disturbances and represents the componants of the action of the environment, in other words the variable forces and moments acting on the hull due to the waves. They have a random character. Most often (in the case of the mathematical model of the ship's roll, which has one degree of freedom) vector Z(t) represents the random function $M_S(t)$, which is the external exciting moment. The values of M_S are non-measurable.

In evaluating the solutions of the concrete realization of the stabilizer there must be defined certain criteria which show the magnitude of the ship's roll, and often has the form:

$$Q = E\left\{G(x)\right\} \tag{2.4}$$

Due to the fact that Z(t) is a stochastic process, X(t) and U(t), defined in the equation (2.1), are also random functions, the functional G(X) is a random variable. Therefore in (2.4) the expected value E{} is used.

It is necessary to find such a way of control U(t), within the acceptable solution (which is physically realizable), so as to minimize the efficiency criteria of the ship's roll stabilization.

It seems advisable to look for optimal control in the form:

$$U = V(X) \tag{2.5}$$

determined in the state space.

One of the points of this study is the possibility of finding such a control method of the special U-type passively-controlled tank with air or fluid valves, which ensures a greater efficiency of the ship's roll stabilization, in comparison with the passive tank (optimally tuned).

The technical conception of the proposed solution is as follows. The solution will be found for a U-type stabilizing tank of high natural frequency, equipped with valves which enable the blocking of the fluid flow between the wing sections of the tank at any given moment. Thanks to the equipping of the tank with blocking valves (binary working) it is possible to tune the frequency of the moment generated by the tank to the exciting frequency, through out a wide range, limited only by the natural frequency of the tank. The case of such a tank installed on board ship is examined. The method of installation of such a tank on board ship is identical with the installation of a U-shaped passive tank. The idea of applying such a tank was proposed in 1930's by Agiejew. A short note on the application of such a tank type can be found in [3,14,52,53,55,61].

The general block diagramme of the ship roll stabilization system using the proposed tank is shown in fig. 2.4.

The ship roll stabilization system shown in fig. 2.4 is devoid of the comparison plant, characteristic for control systems operating in a closed feedback loop. This is connected with the zero value of the setpoint of the ship's roll angle.

A solution close to the presented one is published in [37,52]. In the 1970's the firm Brown-Brothers introduced passively-controlled tanks, which used air valves to interrupt the natural motion of fluid between the wing sections of the tank. Although there is a principle difference between the natural frequencies of the above mentioned tank and the one discussed in this work, it seems advisable, for better understanding to present [37] the schematic drawing of the ship roll tank stabilizing system installed on the American drilling ship "Glomar - Challenger" (fig. 2.5). Fig. 2.6 shows the Honkanen [22] passive control method of U-type and Watt's tanks.

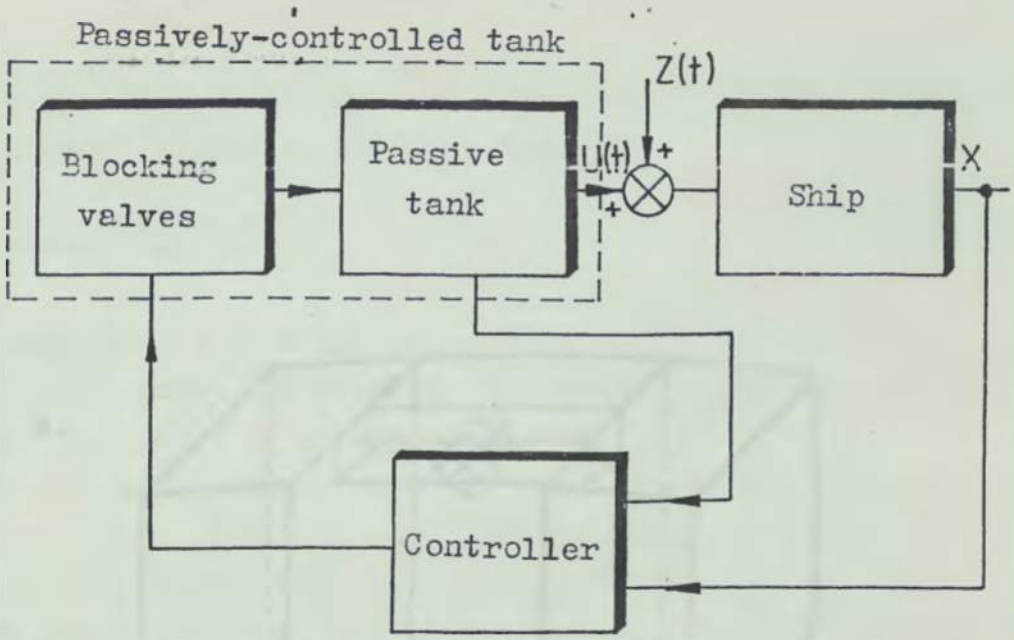

Fig.2.4. Block scheme of the ship roll stabilizing system with the passively-controlled tank.

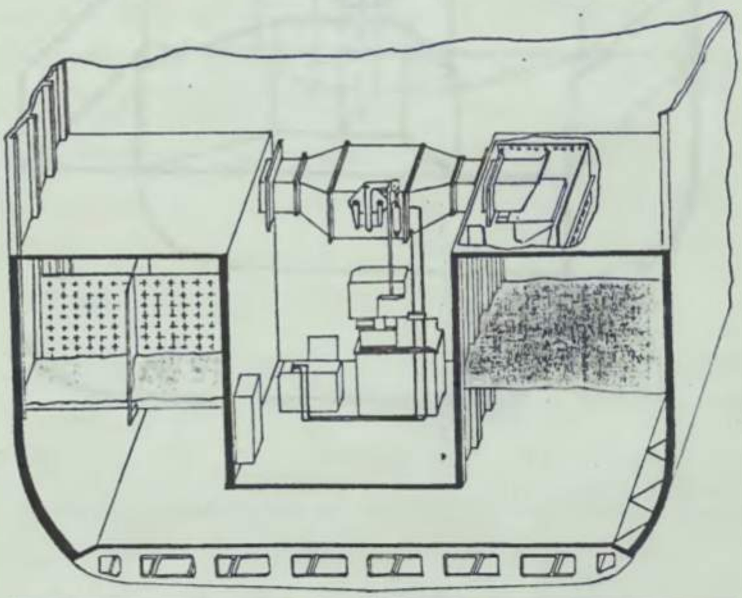

Fig.2.5. The passively-controlled tank installed onboard the ship 'Glomar Challenger'.

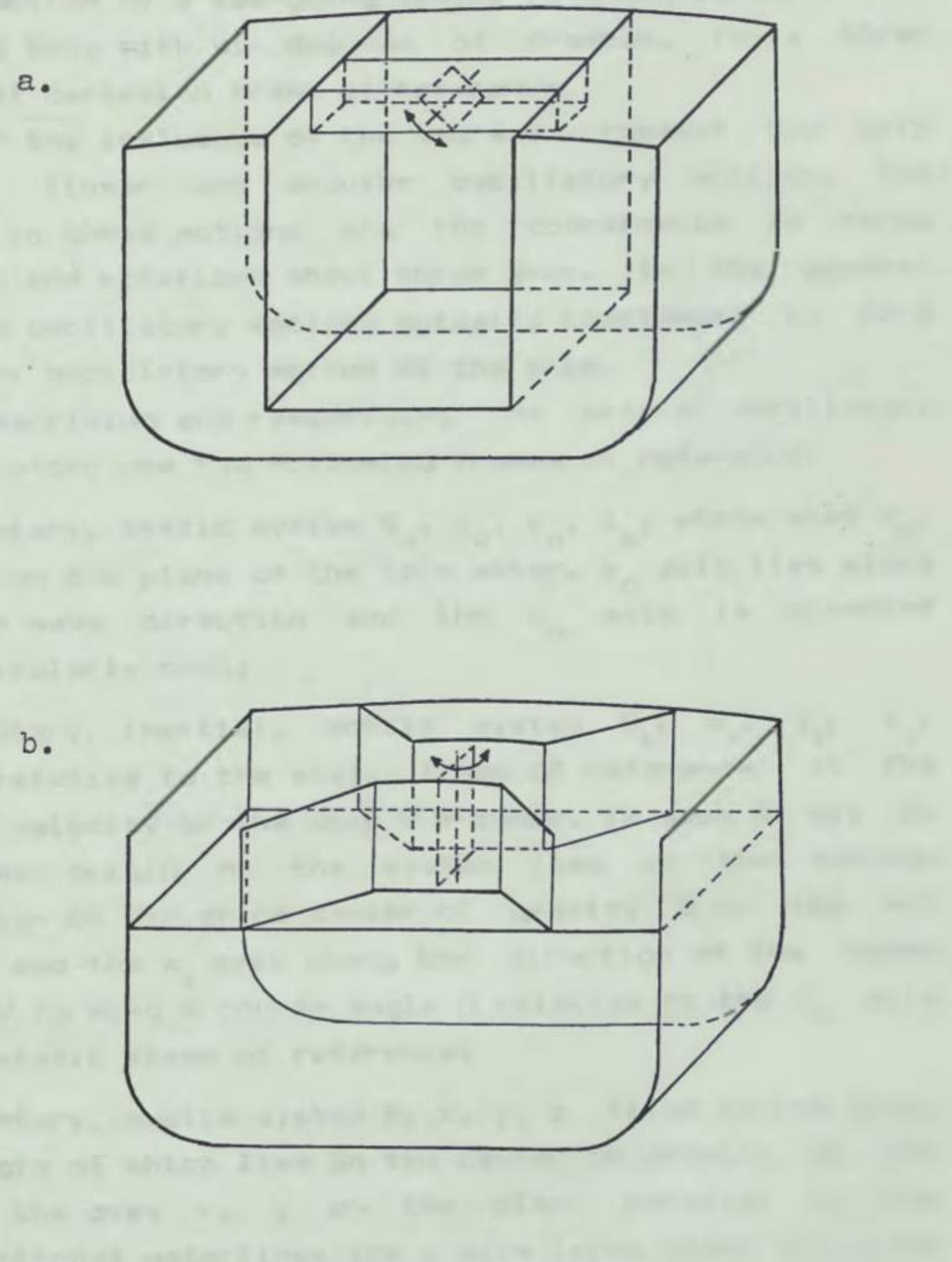

Fig.2.6. The passively-controlled tanks of Honkanen's firm:

a - U-tube tank,
b - free surface tank.

Chapter 3

3. THE MATHEMATICAL MODEL OF THE SHIP ROLL STABILIZATION SYSTEM: SHIP - PASSIVELY-CONTROLLED TANK.

3.1 The coordinate systems.

The motion of a sea-going vessel is shown as the motion of a solid body with six degrees of freedom, in a three dimensional cartesian frame of reference.

Under the influence of the sea's environment the ship performs linear and angular oscillatory motions. The variables in these motions are the coordinates in three directions and rotations about three axes. In the general case these oscillatory motions mutually complement to form the complex oscillatory motion of the ship.

In describing and researching the ship's oscillatory motion we often use the following frames of reference:

- dextrorotary, static system O_o, x_o, y_o, z_o, whose axes x_o, y_o lie on the plane of the calm water, x_o axis lies along the main wave direction and the z_o axis is directed perpendicularly down;
- dextrorotary, inertial, mobile system O_1, x_1, y_1, z_1, moving relative to the static frame of reference, at the average velocity of the ship V = const, in such a way so that the origin of the system lies on the average projection of the ships center of gravity G on the sea surface and the x_1 axis along the direction of the speed vector V forming a course angle β relative to the x_o axis of the static frame of reference;
- dextrorotary, mobile system G, x, y, z fixed to the ship, the origin of which lies on the center of gravity of the ship G, the axes x, y on the plane parallel to the constructional waterline; the x axis lying along the plane of symmetry of the ship directed towards the bow, the y axis to starboard and the z axis downwards (in hydrostatics, upwards).

The systems are shown in fig. 3.1.

The instantaneous position of the oscillating ship in the inertial system O_1, x_1, y_1, z_1 is specified by six coordinates: three coordinates of the pole, which is generally the ship's center of gravity G and three rotation angles relative to the axes going through this pole. The dependency of these six coordinates as a function of time fully describes the ship's oscillatory motion.

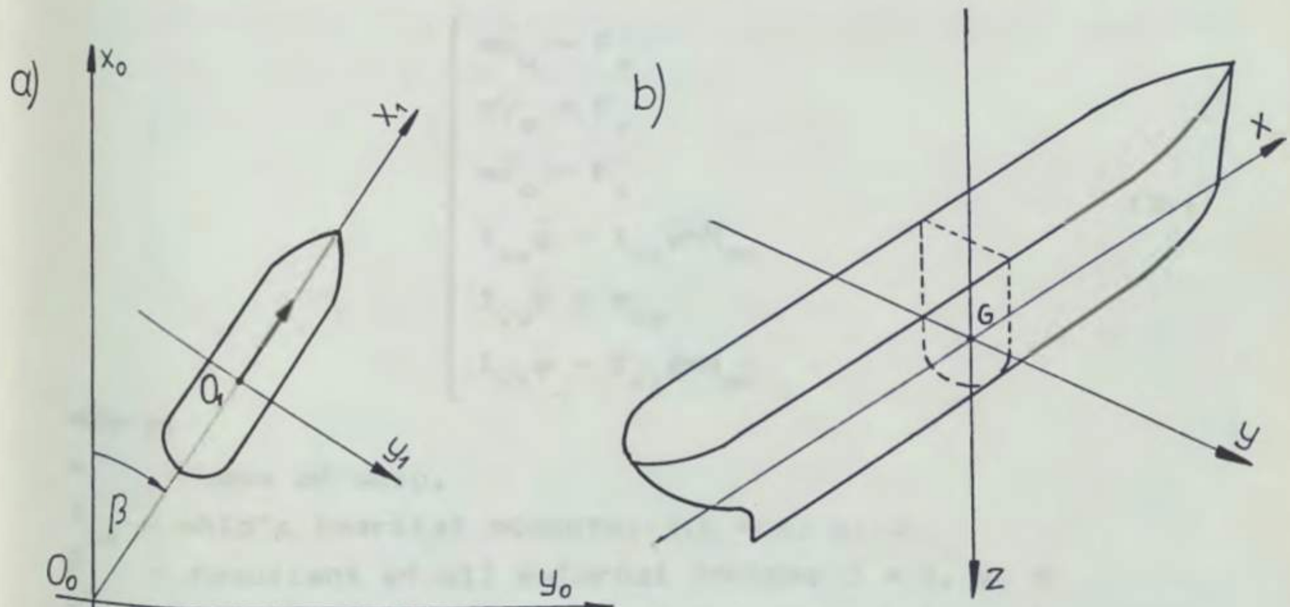

Fig. 3.1 The Cartesian coordinate systems:
a - inertial systems,
b - system fixed to the ship.

3.2 The mathematical model of the ship's oscillatory motion.

An equation of the ship's motion in the coordinate system G, x, y, z was formulated to define the position of the ship in space at any given moment of time. This system, fixed to the ship was chosen, as in this system the inertial moments of the ship's mass are constant.

It is also assumed that the ship's characteristics (as a solid body) are known, which means that known are:

BIBLIOTEKA GŁÓWNA -24-

- mass of ship,
- position of the ship's center of gravity G and the inertial moments of the ship's mass,
- deviation moments of the mass, relative to the axis of the coordinate system fixed to the ship.

The set of differential equations describing the ship's simple oscillatory motions takes on the following form [12]:

$$\begin{cases} m\ddot{x}_G = F_x \\ m\ddot{y}_G = F_y \\ m\ddot{z}_G = F_z \\ I_{xx}\ddot{\phi} - I_{zx}\ddot{\psi} = M_{Gx} \\ I_{yy}\ddot{\theta} = M_{Gy} \\ I_{zz}\ddot{\psi} - I_{zx}\ddot{\phi} = M_{Gz} \end{cases} \tag{3.1}$$

where:

m - mass of ship,
$I_{j,k}$ - ship's inertial moments; $j,k = x, y, z$
F_j - resultant of all external forces; $j = x, y, z$
M_{Gj} - resultant of all external moments relative to point G; $j = x,y,z$.

In solving the set of equations (3.1), it is necessary to determine the external forces and moments which cause the ship's oscillatory motions. These forces depend on the wave parameters, geometry of the hull and distribution of the ship's mass, velocity and course angle, relative to the main waves direction.

Generally, for a solid body of any given shape, a predetermined type of oscillatory motion causes the generation of certain forces which effect the ship's other oscillatory motions. This is the reason for the interaction between oscillatory motions.

Due to the lengthwise symmetry of the ship, it is necessary in forming the linear interdependencies of the exciting forces and the parameters of motion, to group the ship's oscillatory motions in threes: surge, heave and pitch form the so called symmetrical oscillatory motions and sway, roll and yaw form the so called asymmetrical oscillatory motions of the ship.

After taking into account the aforesaid assumptions, the ship's oscillatory motions can be described by a system of six second degree linear differential equations:

- equations of the ship's symmetrical oscillatory motions (in the ship's plane of symmetry):

$$\begin{cases} (m + m_{xx})\ddot{x}_a + N_{xx}\dot{x}_a + m_{xz}\ddot{z}_a + N_{xz}\dot{z}_a + \\ \qquad + m_{x\theta}\ddot{\theta} + N_{z\theta}\dot{\theta} = F_{A_x} e^{-i\omega_E t} \\ m_{zx}\ddot{x}_a + N_{zx}\dot{x}_a + (m + m_{zz})\ddot{z}_a + N_{zz}\dot{z} + \\ \qquad + B_{zz}z_a + m_{z\theta}\ddot{\theta} + N_{z\theta}\dot{\theta} + B_{z\theta}\theta = F_{A_z} e^{-i\omega_E t} \\ m_{\theta x}\ddot{x}_a + N_{\theta x}\dot{x}_a + m_{\theta z}\ddot{z}_a + N_{\theta z}\dot{z}_a + B_{\theta z}z + \\ \qquad + (I_{yy} + m_{\theta\theta})\ddot{\theta} + N_{\theta\theta}\dot{\theta} + B_{\theta\theta}\theta = F_{A_\theta} e^{-i\omega_E t} \end{cases} \tag{3.2}$$

- equations of the ship's asymmetrical oscillatory motions (in the plane of the midship section):

$$\begin{cases} (m + m_{yy})\ddot{y}_a + N_{yy}\dot{y}_a + m_{y\phi}\phi + N_{y\phi}\phi + \\ \qquad + m_{y\psi}\ddot{\psi} + N_{y\psi}\dot{\psi} = F_{A_y} e^{-i\omega_E t} \\ m_{\phi y}\ddot{y}_a + N_{\phi y}\dot{y}_a + (I_{xx} + m_{\phi\phi})\ddot{\phi} + N_{\phi\phi}\dot{\phi} + \\ \qquad + B_{\phi\phi}\phi + (m_{\phi\psi} - I_{zx})\ddot{\psi} + N_{\phi\psi}\dot{\psi} = F_{A_\phi} e^{-i\omega_E t} \\ m_{\psi y}\ddot{y}_a + N_{\psi y}\dot{y}_a + (m_{\psi\phi} - I_{zx})\ddot{\phi} + N_{\phi\psi}\dot{\phi} + \\ \qquad + (I_{zz} + m_{\psi\psi})\ddot{\psi} + N_{\psi\psi}\dot{\psi} = F_{A_\psi} e^{-i\omega_E t} \end{cases} \tag{3.3}$$

This set of equations is the mathematical model of the ship's oscillatory motion, on the regular wave of infinitesimal amplitude.

3.3 The ship's roll motion.

In the linear general model of the complex oscillatory motions , presented in section 3.2,by excluding the through interactions, the equation for the simple oscillatory motion (any roll with one degree of freedom) can be formulated.

In the equations (3.3), by excluding the interactions with the sway and the yaw, we obtain the equation of the ship's roll, in the form:

$$(I_{xx} + m_{\phi\phi})\ddot{\phi} + N^{*}_{\phi\phi}\dot{\phi} + B_{\phi\phi}\phi = F_{A_{\phi}} e^{-i\omega_E t} \tag{3.4}$$

In equation (3.4), $N^{*}_{\phi\phi}$ is the damping coefficient of the roll, due to the influence of the viscosity of water, the bilge keels and the ship's velocity. The behavior of the ship throughout the resonance range depends on the value of this coefficient [15, 26].

By determining the coefficient of the righting moment:

$$B_{\phi\phi} \cong \rho g GM_o \Delta \tag{3.5}$$

in which:

ρ – mass density of water,

g – acceleration of gravity,

GM_o – initial metacentric height of the ship,

Δ – displacement of the ship,

and by writing the amplitude of the moment exciting the ship's roll in the form:

$$F_A = \varkappa_{\phi} h_A k \left[\omega N^{*}_{\phi\phi} + i(gmGM_o - \omega\omega_E m_{\phi\phi})\right] \tag{3.6}$$

in which:

$\varkappa_{\phi}$ – reduction coefficient of the Froude-Krylow part of the exciting moment,

k – wave number,

h_A – wave height,

the ship roll equation takes following form:

$$(I_{xx} + m_{\phi\phi})\ddot{\phi} + N^*_{\phi\phi}\dot{\phi} + gmGM_o\phi =$$
$$= \varkappa_\phi h_A k\left[\omega N^*_{\phi\phi} + i(gmGM_o - \omega\omega_E m_{\phi\phi})\right]e^{-i\omega_E t} \tag{3.7}$$

After introducing the values:

$$\omega^2_{\phi o} = \frac{gmGM_o}{I_{xx} + m_{\phi\phi}}$$

$$2\beta^*_\phi = \frac{N^*_{\phi\phi}}{I_{xx} + m_{\phi\phi}} \cdot \frac{1}{\omega_{\phi o}}$$

$$q = \frac{m_{\phi\phi}}{I_{xx} + m_{\phi\phi}}$$

the equation (3.7) takes the following form:

$$\ddot{\phi} + 2\beta^*_\phi\omega_{\phi o}\dot{\phi} + \omega^2_{\phi o}\phi =$$
$$= \varkappa_\phi\alpha_A\left[2\beta^*_\phi\omega_{\phi o}\omega + i(\omega_{\phi o} - q\omega\omega_E)\right]e^{-i\omega_E t} \tag{3.8}$$

where

ω_{ϕ_o} - natural frequency of the free roll,

$\varkappa_\phi\alpha_A$ - amplitude of the effective wave slope angle,

β^*_ϕ - nondimensional roll damping coefficient.

By ignoring the diffraction part of the exciting forces, we can further simplify (3.8) to obtain [12, 14]:

$$\ddot{\phi} + 2\beta^*_\phi\omega_{\phi o}\dot{\phi} + \omega^2_{\phi o}\phi = \varkappa_\phi\omega^2_{\phi o}\alpha_E(t) \tag{3.9}$$

which describes the ship's roll, caused by a sinusoidal of ω_E frequency.

The equation (3.9), describes the roll on the regular wave, in which the wave slope angle $\alpha_E(t)$ is a harmonic function of time:

$$\alpha_E(t) = \alpha_A \sin\omega_E t \tag{3.10}$$

In analyzing the roll, the equation (3.9) is supplemented by introducing linear-quadratic damping, using ordinary stochastic linearization [14, 61]:

$$2\beta^*_\phi\omega_{\phi o}\dot{\phi} \cong 2\beta_\phi\omega_{\phi o}\dot{\phi} + w_\phi\dot{\phi}|\dot{\phi}| \tag{3.11}$$

in which:

β_ϕ - linear part of the damping coefficient,

w_ϕ - quadratic part of the damping coefficient.

The one dimensional equation describing the ship's roll takes the form:

$$\ddot{\phi} + 2\beta_\phi\omega_{\phi o}\dot{\phi} + w_\phi\dot{\phi}|\dot{\phi}| + \omega^2_{\phi o}\phi = \omega^2_{\phi o}\varkappa_\phi\alpha_A\sin\omega_E t \tag{3.12}$$

in which the coefficients β_ϕ and w_ϕ are practically independent of the angular roll velocity.

3.4 Dynamics of the system ship - passively-controlled tank.

3.4.1 The principle of the passively-controlled tank's operation.

This study takes into consideration the possibilities of using a U-shaped passive tank of high natural frequency, controlled by air valves,placed in the air duct connecting wing sections, to stabilize the ship's roll.

The proposed stabilization system can exist in two dynamic states, connected with the closing and opening of the air valves. Due to the use of air valves in the system, it is possible to control the stabilizing effect of the tank by influencing the natural motion of the fluid in the tank during the roll (fig. 3.2)

In the case of the open air valves (fig. 3.2a) the tank can be thought of as a U-tube with open ends. The natural motion of the fluid in the tank due to gravity is not disturbed. This state of the passively-controlled tank is called the <u>passive state</u>.

When the air valves are closed (fig. 3.2b), then the tank can be assumed to be aU-tube with closed ends. Ignoring certain effects connected with the compressibility of the air in the wing sections, we can assume that the fluid motion between wing sections is impossible . The stabilizing fluid in the tank with closed air valves is "frozen" in the angular position which it had when the valves were closed. This passively-controlled tank state is called the frozen state. In the frozen state, the tank generates a moment about the roll axis, proportional to the difference in the levels of the fluid ΔH (the angular position of the fluid θ), in the wing sections.

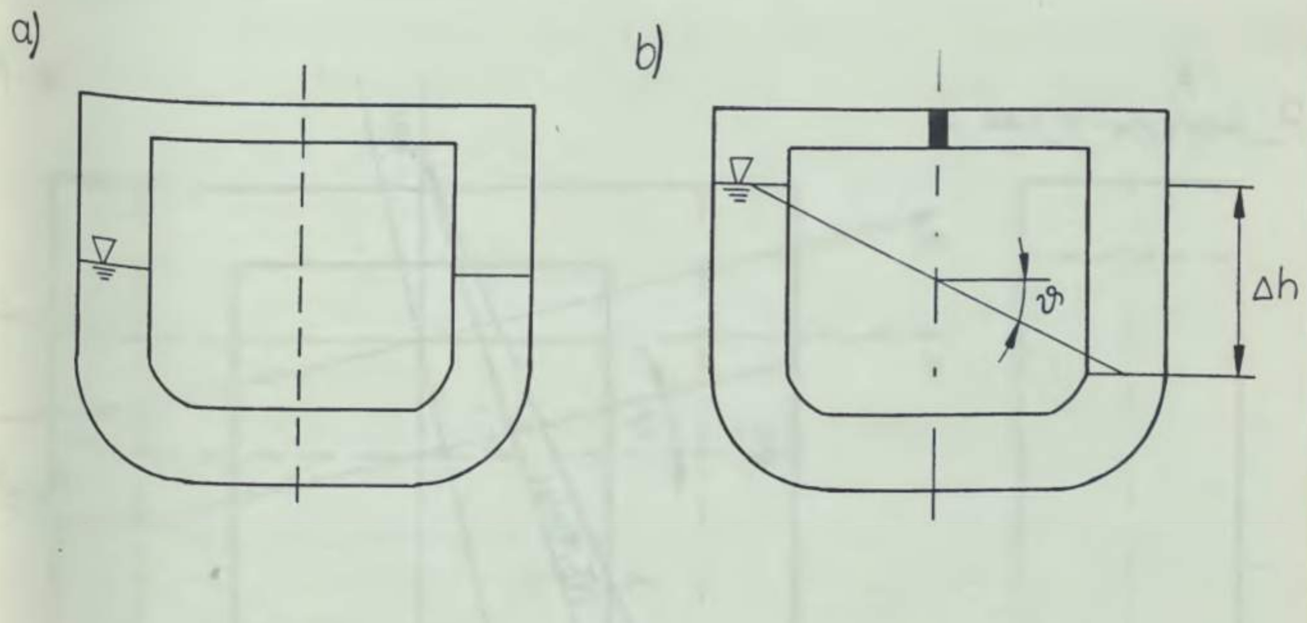

Fig. 3.2 The dynamic states of the passively-controlled tank:
a - air valve open,
b - air valve closed.

3.4.2 The mathematical description of the passively-controlled tank's operating states.

If the ship's roll is a simple oscillatory motion around the longitudinal axis, passed through the ship's center of gravity, then the dynamic system ship - passively-controlled tank can be assumed to be a system with two degrees of freedom.

The mathematical description of the system ship - passively-controlled tank was formulated, assuming that the tank operates in two states. Additionally it was assumed that the closing and opening of the air valves is an instantaneous process in comparison with the processes taking place within the system.

The angle ϑ was accepted as the coordinate which describes the position of the fluid flow. ϑ is the momentary angle which the surface of the fluid forms with the reference plane of the ship (fig. 3.3).

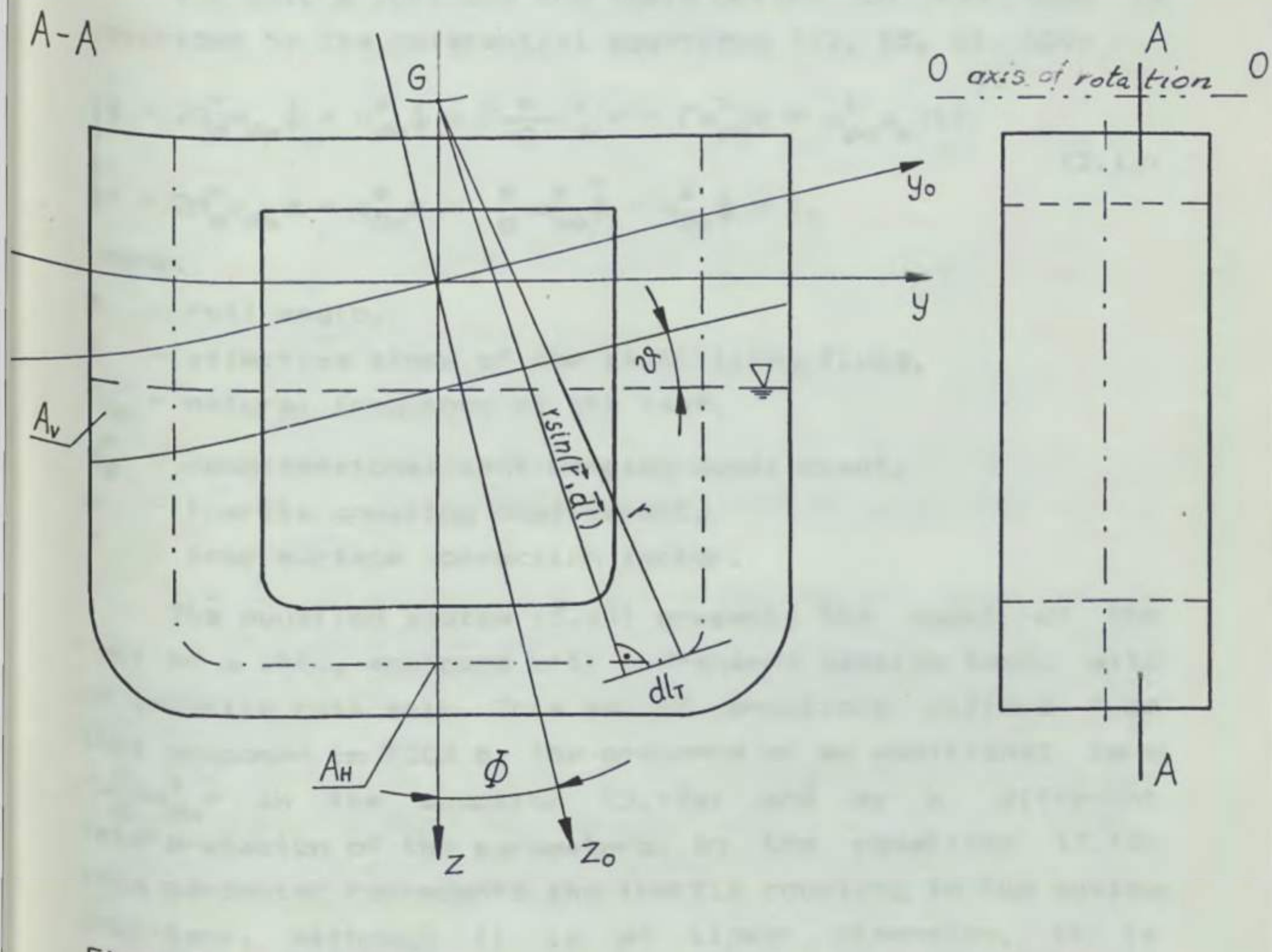

Fig 3.3 The coordinates and symbols used in describing the system ship - tank.

The actual state of the system is determined, through the open and closed states of the air valves:

- passive state - for the time $t \in (t_o^i, t_c^{i+1})$,
- frozen state - for the time $t \in (t_c^i, t_o^{i+1})$,

where t_o symbolizes the moment of the opening of the valves, t_c - the moment of the closing of the valves and the index "i" determines the discrete time.

The passive state of the system

The dynamic system ship - passively-controlled tank in the passive state, is identical to the dynamic system ship - U-shaped passive tank.

The ship's roll and the fluid motion in the tank is described by the deferential equations [13, 55, 61, 65]:

$$\begin{cases} \ddot{\phi} + 2\beta^*_\phi \omega_{\phi o}\dot{\phi} + \omega^2_{\phi o}\phi + \Gamma\frac{s}{g}\omega^2_{\phi o}\ddot{\vartheta} - \Gamma\omega^2_{\phi o}\vartheta = \omega^2_{\phi o}\alpha_E(t) \\ \ddot{\vartheta} + 2\beta^*_\vartheta \omega_{\vartheta o}\dot{\vartheta} + \omega^2_{\vartheta o}\vartheta + \frac{s}{g}\omega^2_{\vartheta o}\ddot{\phi} - \omega^2_{\vartheta o}\phi = 0, \end{cases} \tag{3.13}$$

where:

ϕ - roll angle,

ϑ - effective slope of the stabilizing fluid,

$\omega_{\vartheta o}$ - natural frequency of the tank,

β^*_ϑ - nondimensional tank damping coefficient,

s - inertia coupling coefficient,

Γ - free surface correction factor.

The equation system (3.13) presents the model of the roll of a ship, equipped with a U-shaped passive tank, with an immobile roll axis. This set of equations differs from that proposed in [20] by the presence of an additional term $\Gamma\frac{s}{g}\omega^2_{\phi o}\ddot{\vartheta}$ in the equation (3.13a) and by a different interpretation of the parameters. In the equations (3.13) this parameter represents the inertia coupling in the system ship-tank. Although it is of linear dimension, it is generally not equal to the vertical position of the stabilizing fluid's static level with respect to the ship's axis of rotation, as was assumed in [20].

Only the pure rolling motion of a ship is considered in equations (3.13). It is well known, however, that the ship's sway motion has to be taken into account when calculating the stabilized ship's roll motion. According to [23], this can be approximately achieved in a simple way by introducing the expression $-\omega_{\vartheta}^{2}\alpha_{E}(t)$, instead of zero, into the right side of equation (3.13b) (Appendix 1); then the following set of equations is obtained:

$$\begin{cases} \ddot{\phi} + 2\beta_{\phi}^{*}\,\omega_{\phi o}\dot{\phi} + \omega_{\phi o}^{2}\phi + \Gamma\frac{s}{g}\omega_{\phi o}^{2}\ddot{\vartheta} - \Gamma\omega_{\phi o}^{2}\vartheta = \omega_{\phi o}^{2}\alpha_{E}(t) \\ \ddot{\vartheta} + 2\beta_{\vartheta}^{*}\,\omega_{\vartheta o}\dot{\vartheta} + \omega_{\vartheta o}^{2}\vartheta + \frac{s}{g}\omega_{\vartheta o}^{2}\ddot{\phi} - \omega_{\vartheta o}^{2}\phi = -\,\omega_{\vartheta o}^{2}\alpha_{E}(t) \end{cases} \tag{3.14}$$

After introducing linear-quadratic damping in equations (3.14) (as in equation 3.11) they take the following form:

$$\begin{cases} \ddot{\phi} + 2\beta_{\phi}\,\omega_{\phi o}\dot{\phi} + w_{\phi}\,\dot{\phi}|\dot{\phi}| + \omega_{\phi o}^{2}\phi + \Gamma\frac{s}{g}\omega_{\phi o}^{2}\ddot{\vartheta} - \Gamma\omega_{\phi o}^{2}\vartheta = \omega_{\phi o}^{2}\alpha_{E}(t) \\ \ddot{\vartheta} + 2\beta_{\vartheta}\,\omega_{\vartheta o}\dot{\vartheta} + w_{\vartheta}\,\dot{\vartheta}|\dot{\vartheta}| + \omega_{\vartheta o}^{2}\vartheta + \frac{s}{g}\omega_{\vartheta o}^{2}\ddot{\phi} - \omega_{\vartheta o}^{2}\phi = -\,\omega_{\vartheta o}^{2}\alpha_{E}(t) \end{cases} \tag{3.15}$$

where:

β_{ϕ}, w_{ϕ} – linear and quadratic parts of the ship's roll damping coefficient,

β_{ϑ}, w_{ϑ} – linear and quadratic parts of the damping of the fluid in the tank when the system is in the passive state.

The set of equations (3.15) describes the dynamics of the system ship – tank for any U-shaped passive tank, which is therefore true, for the passively-controlled tank in the passive state.

This model can be used for testing the behavior of the system ship – tank on regular and irregular waves.

The frozen state of the system

In the frozen state, the fluid in the tank is practically immobile. The derivatives of the fluid motion take the form:

$$\dot{\vartheta}(t) \cong \ddot{\vartheta}(t) \cong 0$$

The closing of the air valves and blocking of the fluid motion between the wing sections of the tank causes a change in the natural frequency of fluidoscillation. The assumption that the fluid is completely frozen leads to the following inference:

$$\omega_{\vartheta z} = 0$$

where:

$\omega_{\vartheta z}$ – frequency of oscillation of the fluid in the passively-controlled tank in the frozen state.

Therefore we obtain the description of the dynamics of the system ship – passively-controlled tank in the form of a single differential equation:

$$\ddot{\phi} + 2\beta_{\phi}\omega_{\phi o}\dot{\phi} + w_{\phi}\dot{\phi}|\dot{\phi}| + \omega^2_{\phi o}\phi = \omega^2_{\phi o}[\alpha_E(t) + \Gamma\vartheta(t_c)] \qquad (3.16)$$

Due to the fact that the angular fluid position does not change in this case, the value of the expression:

$$\Gamma\omega^2_{\phi o}\vartheta(t_c) = \text{const}$$

which causes the passively-controlled tank in the frozen state togenerate the moment, known as the <u>asymmetrical moment</u>:

$$M_A = (I_{xx} + m_{\phi\phi})\Gamma\omega^2_{\phi o}\vartheta(t_c) \qquad (3.17))$$

where:

I_{xx} – ship's inertial moment about the longitudinal axis,

$\vartheta(t_c)$ – angular position of the fluid at the instant t_z (the point of closing of the blocking valves).

The analysis of the equation (3.16) leads us to the conclusion that in the frozen state, the sum of the disturbance moment and the asymmetrical moment M_A (generated by the tank) acts on the ship. The asymmetrical moment of the passively-controlled tank can be looked upon as an external additional moment acting on the ship's hull (superposition principle).

The mathematical description of the dynamics of the system ship - passively-controlled tank in both states of operation, taking into account the changes in the natural frequency of the fluid's oscillation, in the case of the frozen state, can be formulated using parametrical differential equations:

$$\begin{cases} \ddot{\phi} + 2\beta_{\phi}\omega_{\phi o}\dot{\phi} + w_{\phi}\, \dot{\phi}|\dot{\phi}| + \omega^2_{\phi o}\phi + \Gamma\frac{s}{g}\omega^2_{\phi o}\ddot{\vartheta} - \Gamma\omega^2_{\phi o}\vartheta = \omega^2_{\phi o}\alpha_E(t) \\ \ddot{\vartheta} + 2\beta_{\vartheta}\omega_{\vartheta}(u,t)\dot{\vartheta} + w_{\vartheta}|\dot{\vartheta}|\dot{\vartheta} + \omega^2_{\vartheta}(u,t)\vartheta + \\ \qquad + \frac{s}{g}\omega^2_{\vartheta}(u,t)\dot{\phi} - \omega^2_{\vartheta}(u,t)\phi = -\omega_{\vartheta}(u,t)\alpha_E(t) \end{cases} \tag{3.18}$$

The system's variable parameter is the frequency of the fluid's oscillation $\omega_{\vartheta}(u,t)$, which is a function of the blocking valves control and time.
The control of the air valves is an integral part of the mathematical model of the system ship - passively-controlled tank. Chapter 5 is devoted to this problem.

3.4.3 The compressibility of the air in the passively-controlled tank.

The use of the air valves, which would block the natural fluid motion requires taking into account, in the mathematical model of the system, the compressibility of the air, above the fluid surfaces as well as its influence on the stabilizing moment of the passively-controlled tank in the frozen state.

During the operation of the system, the tank's air space undergoes periodic polytropic processes of compression and decompression. As it is not possible to determine the exponent of these changes, it is assumed that the polytropic curves are between the isothermic and adiabatic curves.

In the frozen state, the asymmetrical moment generated by the tank is proportional to the angular positions of the fluid's surfaces in the tank ϑ, during the ship's roll (fig.3.4).

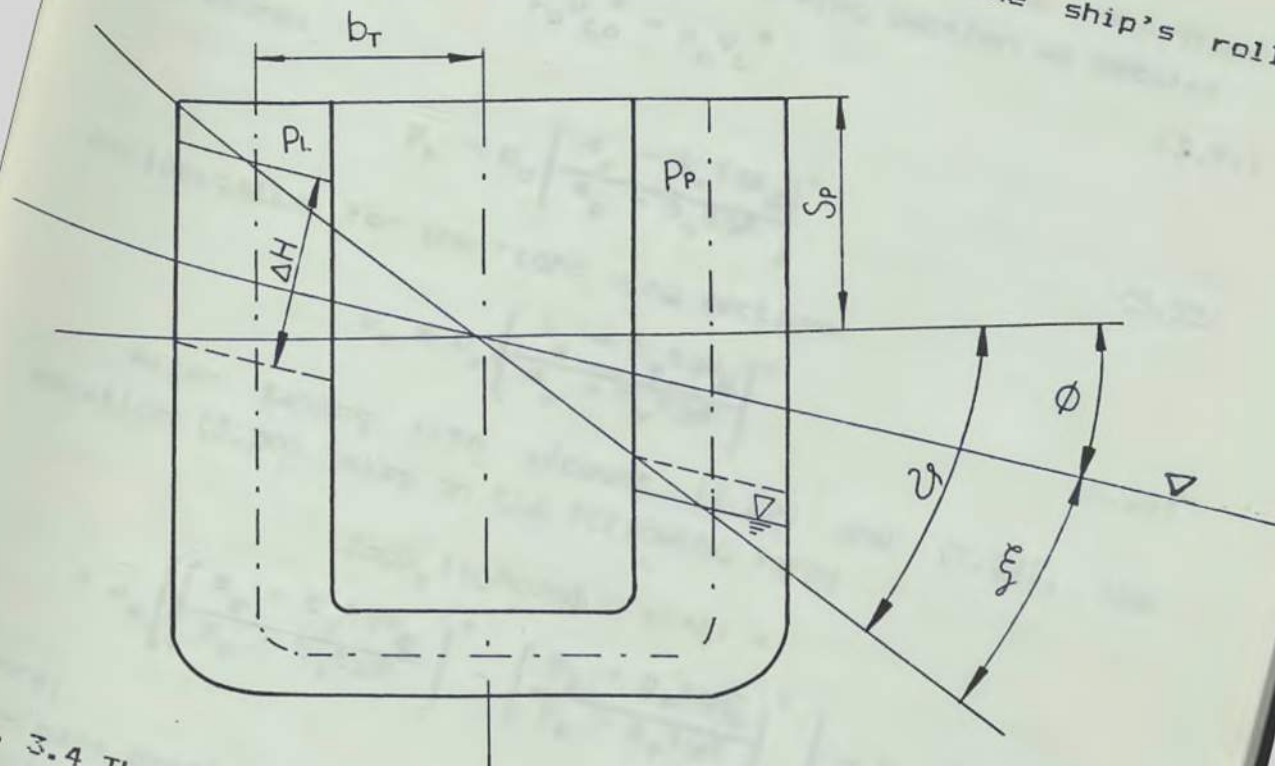

Fig. 3.4 The passively-controlled tank in the frozen state.

Generally the angle ϑ is a function:

$$\vartheta(t) = f\left[\vartheta(t_c), \phi, \hat{P}, t\right]$$

where:

$\hat{P}$ – vector representing the constructional parameters of the tank stabilizer.

Generally, for the tank in the frozen state, the following is true:

$$\begin{cases} \xi=\vartheta - \phi \\ \Delta H=2b_T \dfrac{\sin\xi}{\cos\vartheta} \end{cases} \qquad (3.19)$$

The pressure equation for the fluid's surfaces can be described as:

$$\Delta H \rho g + p_L - p_P = 0 \qquad (3.20)$$

Assuming that the exponent of the polytropic transformation is z, for the left wing section we obtain:

$$p_o V_{Lo}^{\ z} = p_L V_L^{\ z} \tag{3.21}$$

therefore:

$$p_L = p_o \left(\frac{s_p - b_T \mathrm{tg}\vartheta_C}{s_p - b_T \mathrm{tg}\vartheta} \right)^z \tag{3.22}$$

Analogically, for the right wing section:

$$p_P = p_o \left(\frac{s_p + b_T \mathrm{tg}\vartheta_C}{s_p + b_T \mathrm{tg}\vartheta} \right)^z \tag{3.23}$$

After taking into account (3.22) and (3.23), the equation (3.20) takes on the following form:

$$2\rho g b_T (\mathrm{tg}\vartheta \cos\phi - \sin\phi) + \\ + p_o \left[\left(\frac{s_p - b_T \mathrm{tg}\vartheta_C}{s_p - b_T \mathrm{tg}\vartheta} \right)^z - \left(\frac{s_p + b_T \mathrm{tg}\vartheta_C}{s_p + b_T \mathrm{tg}\vartheta} \right)^z \right] = 0 \tag{3.24}$$

where:

ρ - mass density of the stabilizing fluid,

b_T - half the width of the tank,

p_o - normal atmospheric pressure,

s_p - length of the air-tube above the fluid surface in the wing section, when $\phi = 0$.

The compressibility of the air in mathematical model of the system ship - passively-controlled tank can be taken into account by solving the equation (3.24) for the variable angles ϑ_c and ϕ.

3.5 The mathematical description of the wave motion.

The real sea wave motion has an irregular character and is a random process, which can be described using the probability and the random theories. A description of the qualities of the wave motion was presented in [14].

The sea wave motion is assumed to be a stationary, homogeneous and ergodic space-time random process. Its values, which describe the wave's profile is of normal distribution.

Approximately the wave motion can be presented as a two-dimensional irregular wave. It is in fact the finite sum of elementary sinusoidal waves (called harmonic waves), of different angular frequencies. It is also assumed that the phase angle of each of these waves is a random variable of uniform distribution, in the range <0, 2Π>.

The greatest amount of information about the irregular wave, is contained in the characteristic of the wave power spectrum. The wave energy spectrum shows the way in which the total energy of the waves is distributed between the harmonic components (frequencies) of the wave motion.

In the case of insufficient data about the ship's operating region, International Towing Tank Conference (ITTC) recommends the following form of the standard wave spectrum [14]:

$$S_h(\omega) = A\omega^{-5}\exp(-B\omega^{-4}) \ [m^2 s] \tag{3.25}$$

where:

ω - circular frequency of the wave in [rad/s],

If the significant wave height $\bar{h}_{1/3}$ and characteristic period $\bar{T}$ are known, then the constants A and B can be determined from:

$$A = 173 h_{1/3}/\bar{T}^4 \qquad B = 691/\bar{T}^4 \tag{3.26}$$

where:

$\bar{h}_{1/3}$ - significant wave height, determined as the average value of 1/3 of the highest waves,

$\bar{T}$ - characteristic period.

Alongside the wave height density spectrum $S_h(\omega)$, the wave slope density spectrum function $S_\alpha(\omega)$ can be used. The following relationship is reached between these functions:

$$S_\alpha(\omega) = k^2 S_h(\omega) \qquad (3.27)$$

where:

k - wave number: $k = \frac{\omega^2}{g}$

The wave power spectrum shown as in (3.25) does not take into account the motion of the ship relative to the wave motion, which can be described, giving the ship's velocity and the angle between the vectors: ship's velocity and the wave propagation velocity. The frequency of the wave exciting the ship's roll is called the relative circular frequency and is expressed by:

$$\omega_E = \left|\omega - \frac{\omega^2}{g} V \cos\beta\right| \qquad (3.28)$$

where:

ω_E - relative frequency, ie.: the wave frequency in the mobile inertial system O_1, x_1, y_1, z_1,

ω - absolute wave frequency in the static coordinate system,

V - ship's velocity,

β - angle between the ship's course and the dominant direction of wave propagation.

3.6 Remarks on the identification of the coefficients in the mathematical model of the system ship - passively-controlled tank.

In this chapter, the complete synthesis of the model of the dynamics of the system ship - passively-controlled tank, as a system of differential equations with a variable parameter has been formulated. In this model we can differentiate the following component models:

- the mathematical model of the ship's free roll as a second degree oscillator with linear-quadratic damping,
- the mathematical model of the stabilizing fluid's free oscillatory motion, which is also a second degree oscillator with linear-quadratic damping,

- the model of the coupling elements between the stabilizing fluid's and the ship's roll motions,
- the mathematical model of the polytropic gas transformations in the frozen state of the passively-controlled tank,
- the mathematical model of the sea wave motion.

The purpose of the model (3.18) has a deciding influence on the structure and form of the accepted model as well as on the range of simplifications done during its synthesis.

The basic problem in the synthesis of this mathematical model is the acceptance of the form of the model in the passive state. The model of the system in the frozen state results directly from the model in the passive state. Therefore it is burdened with all the simplifications used in the construction of this model.

The dynamic model of the passive state was formulated in the general form. In literature we find articles which discuss the correlations between the dynamic models and the real systems [23, 61]. The main problem is the impossibility of referring all the results obtained in model tests to the real conditions. The realization of such a comparison leads to great difficulties. For example it is impossible to limit the degrees of freedom of a real ship.

An interesting discussion on the usefulness of mathematical models during the design of ship roll stabilizing systems can be found in [3].

The particular usefulness of models of two degrees of freedom (roll - sway), is pointed out in the above mentioned publication [3]. This is the kind of model proposed and used in this work.

Chapter 4

4. THE EFFECTIVENESS CRITERIA OF THE SHIP'S ROLL STABILIZATION.

The necessity to formulate the objective criteria defining the effectiveness of the ship's roll stabilization arises from the following premises:

- possibilities of mutual objective comparison between several realizations of stabilizers,
- necessity to use these criteria as elements of the target function, during the optimization of the construction and control of the stabilizer,
- determination of the costs (profit and loss), resulting from the installation of the roll stabilizer on board ship.

The ship's roll motion is generallya non-stationary stochastic process and depends on a range of factors, of which the most important are: sea state, velocity and course angle of the ship, relative to the main direction of the sea wave motion, and load conditions.

Generally, we accept the following division of the effectiveness criteria, for the functioning of the roll stabilizers:

1. criteria which determine the effectiveness of roll stabilization on the regular wave,
2. criteria of the effectiveness of roll stabilization on the irregular wave, which can be divided into two groups:
   - short time prognosis,
   - long time prognosis.

The evaluation of the effectiveness of roll stabilization is connected quite frequently with a whole range of misunderstandings and incomplete information, especially in prospects and advertising brochures.
The frequent reasons for these misunderstandings are:

1. omition of the criteria used in evaluating the effectiveness of the stabilizer,
2. absence of the conditions in which the concrete criteria values were obtained.

4.1 Methods of determining the effectiveness of roll stabilization on regular waves.

The regular wave conception ($\alpha_A(t) = \alpha_A \sin \omega_E t$) is in someways an ideal. In reality such a wave does not exist. It can only be a certain approximation of the so called dead wave. The testing of roll stabilizers on the regular wave should be connected with tradition and practice in the automatic control theory. The roll stabilizers are generally automatic control systems or systems defined by a set of differential equations.

For such stabilizers (as long as they are defined by a set of linear differential equations), it is possible to assign amplitude and frequency characteristics:

$$\begin{cases} |K(j\omega_E)| = f_1(\omega_E) \\ \varepsilon(\omega_E) = f_2(\omega_E) \end{cases} \tag{4.1}$$

in which:

$|K(j\omega_E)|$ - module of the frequency characteristic of the system ship - stabilizer,

ε - phase angle (phase delay) of the response, corresponding to the system ship - stabilizer, relative to external disturbance.

The characteristics (4.1) allow us to draw concrete conclusions as to the behavior of the stabilized ship in the function ω_E. These conclusions although drawn on the basis of an idealized form, can be used in evaluating the behavior of the stabilizing system in real conditions.

The amplitude and phase characteristics of the form (4.1) are the most often used criteria, in evaluating the functioning of roll stabilizers.

If the system ship - stabilizer is described by a set of non-linear differential equations, then it is not possible to directly assign its characteristics (4.1). In such a case we use the frequency characteristics in the form:

$$\begin{cases} Y_{\phi\alpha}(\omega_E) = \left|\dfrac{\phi_A}{\alpha_A}\right|_{\alpha_A = \text{const}} \\ \varepsilon(\omega_E) = \varepsilon_{\phi\alpha}(\omega_E)\Big|_{\alpha_A = \text{const}} \end{cases} \tag{4.2}$$

in which:

ϕ_A - maximal or effective amplitude of the ship's roll,

α_A - amplitude of the wave slope angle,

ε - phase angle of the response, corresponding to the system ship - stabilizer, relative to external disturbance.

An additional difficulty is the control of the stabilizer, due to the presence of two types of characteristics (4.2):

- characteristics for the optimal (different for each frequency) settings of the controller,
- characteristics for the constant settings of the controller (usually these are settings, optimal for the resonance frequency of the ship).

Other than information on the behavior of the system ship - stabilizer, the first group of characteristics ordinarily constitute the premise for the necessity for self-tuning of the stabilizer controller. The second group of characteristics shows the frequency range through out which the stabilizer functions properly. The evaluation of the characteristic (4.2), in the case of the non-linear model of stabilizer dynamics, as well as in the case of model tests, is done by the point by point method, for fifteen or sixteen points, within the chosen frequency range.

On the basis of the characteristic (4.1) or (4.2) it is possible to evaluate the effectiveness factor of stabilization [13] according to the formula:

$$E = \frac{\phi_A - \phi_{A\ STAB}}{\phi_A} \tag{4.3}$$

or

$$\tilde{E} = \frac{\phi_{A\ STAB}}{\phi_A} \qquad (4.4)$$

in which:

ϕ_A, $\phi_{A\ STAB}$ - respectively, the roll amplitudes of the unstabilized and stabilized ship, determined for the same input frequency.

Complete information on the stabilizer's operation can be obtained by representing the relationships (4.3) and (4.4) as functions of the input frequency ω_E, and the control u. Therefore:

$$E(\tilde{E}) = f(\omega_E, u) \qquad (4.5)$$

There exist other criteria, common for the evaluation of the stabilizer operation, on the regular and irregular wave. These criteria will be discussed later.

4.2 Evaluation methods of the effectiveness of the roll stabilization on the irregular wave.

In formulating the criteria determining the effectiveness of the roll stabilizers, it is necessary to define the influence of the stabilizer (construction and control), on the behavior of the ship equipped with such a stabilizer.

It is also necessary to take into account that, the majority of values occurring during the ship's roll motion are random ones.

Generally, in evaluating the effectiveness of the roll stabilizer, the variance criteria is accepted in the following form:

$$J_1 = \frac{1}{T} \int_0^T \left[k_1\phi^2 + k_2\dot{\phi}^2 + k_3\ddot{\phi}^2\right] dt \qquad (4.6)$$

in which:

T - process time,

k_1, k_2, k_3 – respective weight coefficients of the angle, angular velocity and the angular acceleration of the ship's roll motion.

The criterion (4.6) is a simple expression to determine the value of the ship's roll and its derivatives. This criterion is often named "criterion of comfort", as it directly determines sailing comfort.

The energy consumption of the stabilizer, plays an important role in most of the controlled stabilizers. Therefore into the expression (4.6) we introduce a term proportional to the consumed energy,

$$J_2 = \frac{1}{T}\int_0^T \left[k_1\phi^2 + k_2\dot{\phi}^2 + k_3\ddot{\phi}^2 + q\xi^2\right]dt \qquad (4.7)$$

where:

ξ – coordinate determining the motion of the actuator of the stabilizer (ex. angle of attack of the stabilizing fins).

The criteria (4.6) and (4.7) are general ones used in evaluating the efficiency of the stabilizer, on regular and irregular waves. The only inconvenience connected with the use of these criteria is the difficulty in defining the weight coefficients k_i and q.

If we assume, that $k_1 = 1$ and $k_2 = k_3 = q = 0$, then the value of the criterion (4.6) is equal to the variance of the ship's roll motion. It is worthwhile to add, that the coefficient q (representing generalized control costs) is not significant in the case of the passively-controlled tank. The motion of the stabilizing fluid is not an energy consuming process.

The effectiveness factor of the stabilizer on the irregular wave can be determinated on the basis of the criteria (4.6) and (4.7). This factor gives information on the advisability of using a stabilizer, especially when it can be evaluated for the wave characteristics of the ship's future operation regions. If the ship is designed for free sailing then the criteria of effectiveness are determinated for the North Atlantic.

The effectiveness factor of roll stabilization on irregular waves is evaluated analogically to (4.3) and (4.4) as:

$$E_N = \frac{\sqrt{D^2_{\phi_N}} - \sqrt{D^2_{\phi_{STAB}}}}{\sqrt{D^2_{\phi_N}}} \tag{4.8}$$

or

$$\tilde{E}_N = \sqrt{\frac{D^2_{\phi\,STAB}}{D^2_{\phi_N}}} \tag{4.9}$$

in which:

$D^2_{\phi_N}$, $D^2_{\phi_{STAB}}$ - the respective roll variances of the unstabilized and stabilized ship.

Other than the criteria given by the expressions (4.6) to (4.9), a whole range of auxiliary criteria determining the average values of the roll angles and accelerations are in use. These criteria have been defined and discussed in detail in [14, 61].

Criteria often applied in describing the ship's oscillatory motion on irregular waves are the power spectra.

If the dynamic system ship - stabilizer is described by linear differential equations with constant coefficients, then the form of the energy spectrum can be analytically obtained from Parseval's formula:

$$S_\phi(\omega_E) = \left|K(j\omega_E)\right|^2 S_\alpha(\omega_E) \tag{4.10}$$

in which:

$S_\phi(\omega_E)$ - roll spectrum,

$S_\alpha(\omega_E)$ - spectrum of the sea wave motion.

It is relatively difficult to determine the energy spectrum (4.10) in model tests, but having suitable apparatus it can be determined through the special procedure (ex. fast Fourier transform). On the basis of the stabilized and unstabilized ship roll spectra it is possible to draw wider conclusions than from the expressions (4.8) and (4.9).

The evaluation criteria for the effectiveness of the ship's installations, especially useful for the roll stabilizers, are the probability functions of exceeding the roll amplitudes and the angular roll accelerations. These functions in principle, complement the random distribution function to unity:

$$P\left\{|\phi| \geq \phi^*\right\} = f(\phi^*) \tag{4.11}$$

in which:

$P\left\{|\phi| \geq \phi^*\right\}$ - probability of the rolling ship exceeding the fixed roll angle ϕ^*.

The properties of the criterion (4.11) result directly from the definition of a distribution function:

$$P\left\{|\phi| \geq 0\right\} = 1 \quad \text{and} \quad P\left\{|\phi| \geq \infty\right\} = 0$$

The criterion (4.11) is best determined from experimental tests, by special data processing programmes.

The criterion (4.11) is especially useful for ship designers, because in a simple way, it characterizes the ship's roll motion on the irregular wave. With the aid of criterion (4.11) it is simple to define the limits of the angles (angular accelerations) of the ship's roll motion. It allows us to determine the influence of the roll motion on deck equipment installed on board ship, ex. drilling systems, handling systems, etc.

4.3 Analysis of the efficiency of roll stabilization.

The necessity for using the roll stabilizer on board ship, in practice depends on the tasks to be realized by the ship, in the future. Accordingly two groups of tasks can be distinguished [49]:

1. transportation - understood as displacement of cargo, passengers and equipment with personel and stores, between two points on the sea,
2. sojourn - understood as the stay or the slow motion on the sea, of the ship with the aim of completing tasks such as loading - unloading, hydrographical, drilling, geological, trawling, etc.

The necessity of the ship's roll stabilization, in the first group of tasks, is economically justified by the reduction of fuel consumption by the main engines at a given ship's speed.

The necessity of the ship's roll stabilization in the second group of tasks, is considered justified, if the unstabilized ship cannot, due to bad weather conditions, perform its assigned tasks (due to the ship exceeding the limits of the roll angles and angular accelerations).

To evaluate the necessity and effectiveness of roll stabilization (according to the aforementioned aims), it is necessary to have at one's disposal long time prognosis of the ship's roll, which as a base, uses the statistical model of the ship's operation [62], characteristics of the main engines and information on the sailing economics.

The criteria discussed in the previous sections of this chapter, are the only ones used in this study. These criteria are appropriate in evaluating the effectiveness of the stabilizer, whose installation on board ship has already been preordained, on the basis of economical analysis.

Chapter 5

5. CONTROL ALGORITHMS OF THE VALVES BLOCKING THE FLUID MOTION IN THE PASSIVELY-CONTROLLED TANK.

The control of the ship roll stabilizer can be considered a problem of optimal filtering. A stabilizer is necessary to compensate the effects of external disturbances (regular and irregular), which act on the ship. The ship is the object of control in the roll stabilization system.

Practically, the problem of control of the ship roll stabilizer is not discussed in scientific literature. The general theories presented in [16, 21] can be adapted only to the synthesis of control of active roll stabilizers, especially for the stabilizing fins. Similar general theories on this subject can be found in [55, 57]. In literature published by various producers [22, 37, 51, 52], the problem of the passively-controlled tank's control is not discussed.

In the case of the passively-controlled tank, when the controlled parameter is the natural frequency of the tank, it is impossible to practically apply the general theories for the control synthesis. For such objects, it's difficult to find the proper control through pure empirical methods. Due to certain interesting aspects of this problem, this chapter tries to find the blocking valves control algorithms, as well to explain phenomena connected with the control of the fluid motion, in the passively-controlled tank.

This algorithm can be most easily derived from the observation of phenomena occurring in the system ship - tank. The aim of the algorithm is to determine the moment of closing (t_c) and opening (t_o) of the valves blocking the fluid motion in the tank.

A properly operating control system should ensure that the stabilizing moment, generated by the tank, counteracts (in the opposite direction) the disturbance moment, created by the sea-surface. In the case of active stabilizers this condition is relatively easy to realize. However, in the case of the passively-controlled tank, in which the fluid motion occurs only under the influence of gravity, it is not always possible to ensure the above mentioned conditions.

5.1 Certain phenomena occurring during the fluid motion in the tank.

In the case of the fluid blocking valves being open, the fluid moves in such a way so as to achieve a horizontal position. This tendency does not depend on the position of the ship's hull. A few such cases are shown in fig. 5.1.

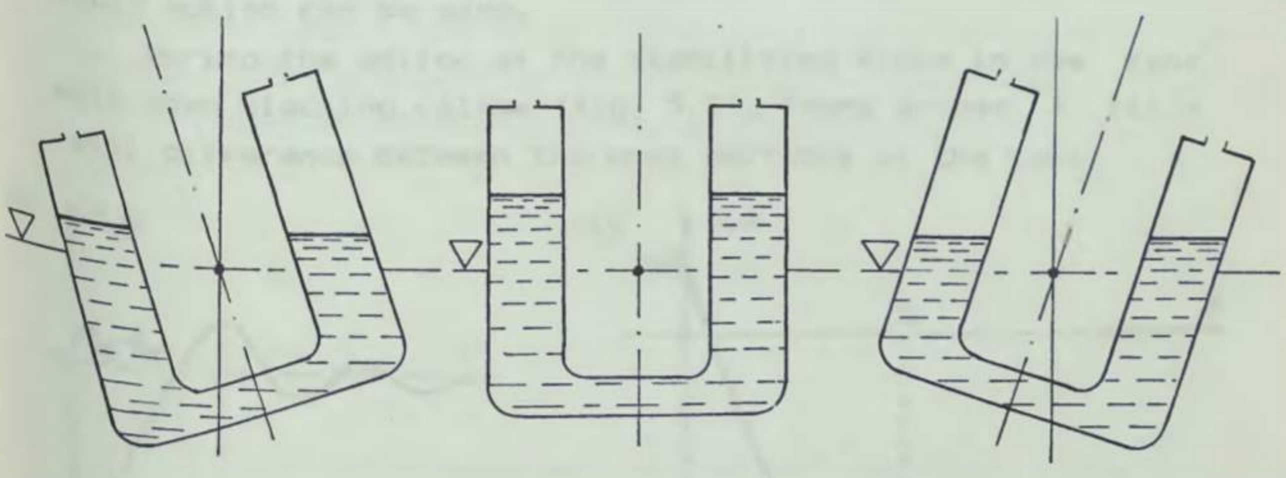

Fig.5.1. Different positions of the ship's hull on the sea and their corresponding fluid positions.

If we examine the fluid motion in the tank from any given moment to a stable state, depending on the actual position of the ship's hull relative to the terrestrial coordinate system, we will note that the fluid motion is characterized by a periodical tendency to achieve a new state of stability. In real conditions, the ship's hull rolls, therefore, the stable state of the stabilizing fluid changes continuously. This can be illustrated by the fig.5.2.

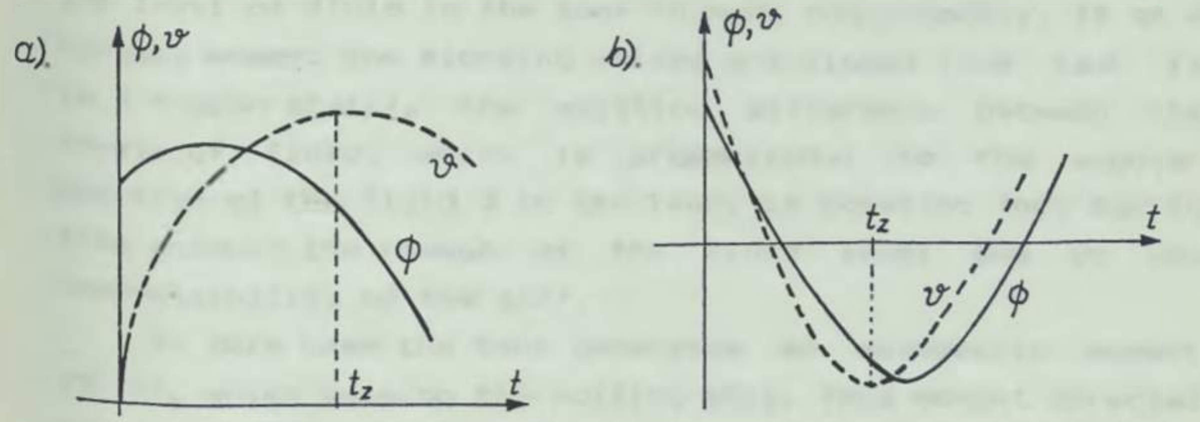

Fig. 5.2 The time diagrammes of the fluid motion in the tank.

Assuming that the ship's hull is heeled at a certain angle and remains static in this position, concrete inferences regarding the control of the valves blocking the fluid motion can be made.

During the motion of the stabilizing fluid in the tank with open blocking valves (fig. 5.3), there arises a fluid level difference between the wing sections of the tank.

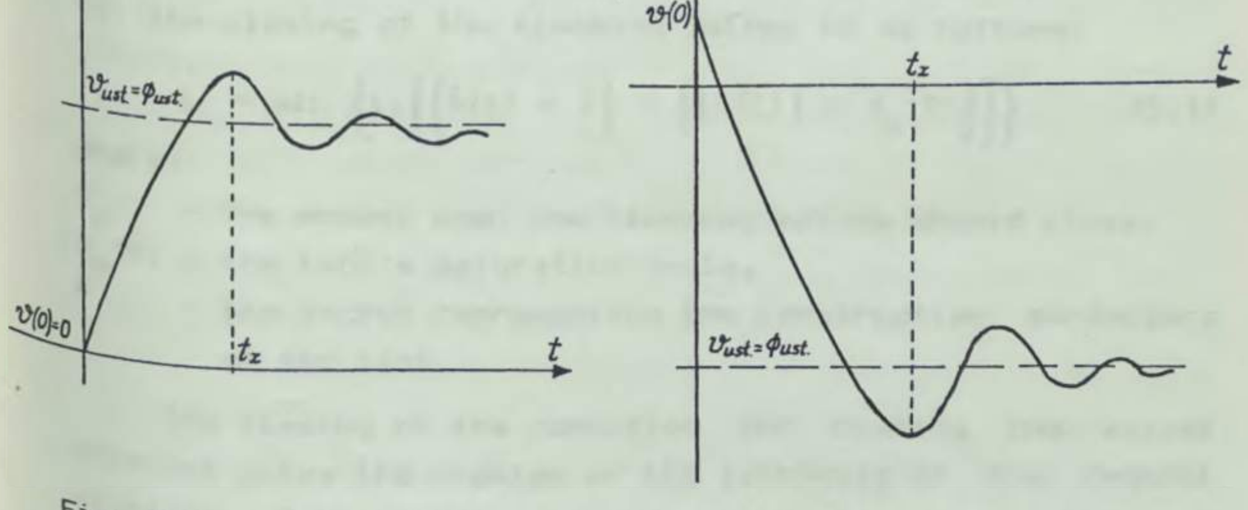

Fig. 5.3 The time diagramme of the transient processes of the stabilizing fluid motion for the static ship's heel angle:

a) for zero initial conditions of the fluid,

b) for non-zero initial conditions of the fluid.

The level of fluid in the tank changes continuously. If at a certain moment the blocking valves are closed (the tank is in a frozen state), the existing difference between the levels of fluid, which is proportional to the angular position of the fluid ϑ in the tank, is constant (not taking into account the change of the fluid level due to the compressibility of the air).

In this case the tank generates an asymmetric moment (3.17), which acts on the rolling ship. This moment directed along (in phase), the action of the exciting moment, increases the amplitudes of the ship's roll. In the opposite case, when the tank's moment is directed against (in counter phase) the ship's roll,its amplitudes decrease.

We can prove that the maximum moment of the tank is generated when the difference of the fluid levels between the wing sections achieves the greatest value. This happens when $\vartheta(t) \rightarrow max$, therefore $\dot{\vartheta}(t) \rightarrow 0$. Independent of the aforesaid it is necessary to interrupt (block) the fluid motion, when the fluid level in one wing section reaches its maximum, when $|\vartheta(t)| \geq \vartheta_N$.

Therefore the condition for freezing the fluid flow, ie. the closing of the blocking valves is as follows:

$$t_c = \min \left\{ t: \left[\left[\dot{\vartheta}(t) = 0 \right] \vee \left[|\vartheta(t)| = \vartheta_N(\hat{P}) \right] \right] \right\} \qquad (5.1)$$

where:

t_c - the moment when the blocking valves should close,

$\vartheta_N(\hat{P})$ - the tank's saturation angle,

$\hat{P}$ - the vector representing the construction parameters of the tank.

The finding of the condition for closing the valves does not solve the problem of the synthesis of the control algorithm. In particular, it does not solve the problem of generating the proper stabilizing moment by the tank. The proper stabilizing moment should be directed against (in counter phase, if it is possible) the ship's roll exciting moment.

To satisfy the above mentioned condition, it is necessary to find a physically realizable algorithm, which defines the conditions for the opening of the blocking valves. It is also notable that the algorithm (5.1) determining the conditions closing the blocking valves which ensures only the maximization of the tank's moment, does not ensure the proper action of the tank.

The algorithm which determines the opening of the blocking valves is the control algorithm of the passively-controlled tank, looked for by us. The closing of the blocking valves is in fact only a consequence of their earlier opening.

The control of the opening of the blocking valves (called hereafter the control of the tank) should take into account two conditions arising from the phenomena of the proposed system. They are as follows:

- determination of the possibilities of fluid flow between the wing sections of the tank; due to the properties of the tank, the fluid flow should take place only under the force of gravity, therefore it is necessary to determine the conditions of the fluid motion in the required direction,
- determination of the necessity for opening the valves, thereby restoring the fluid flow between the wing sections of the tank, which determines the effect of the change of the tanks asymmetrical moment on the ship's roll.

The control algorithm of passively-controlled tank should consist of two parts:

- the precondition which determines the possibility of the fluid motion,
- the adequate condition which determines necessity of this motion.

5.2 The evaluation of the precondition of the opening of the fluid motion in the passively-controlled tank.

It is necessary to find the relationships, which determine the relative positions of the ship's hull and the frozen fluid in the tank, which after the opening of the blocking valves make possible the relocation of the fluid to the opposite wing section, only under the influence of gravity.

Let's examine four relative positions of the ship's hull and the frozen fluid in the tank (fig. 5.4). The situations shown in fig. 5.4 take into account all possible relative positions of the ship's hull and the frozen fluid, except for the stable position determined by the conditions: $\phi = 0$ and $\vartheta = 0$.
The above mentioned statement is correct, because the running values of the angles ϕ and ϑ are not important. Important are their signs only (positive or negative).
According to the fig. 5.4, we see that in the case of the situations described in fig. 5.4a and 5.4b, the opening of the valves does not cause the looked for fluid motion. This is due to the fact that the ship is heeled to the side weighted down by the column of fluid.

Therefore the disadvantageous relative position of the hull and the frozen fluid in the tank is determined by the expression:

$$\delta_t = \left\{ t\colon [\phi(t)\cdot\vartheta(t) > 0] \right\} \tag{5.2}$$

where $\delta_t(R)$ defines the time gate (time delay) for which the relation R is true.

The advantageous influence of gravity is shown in fig. 5.4c and 5.4d. These situations are characterized by the time gate:

$$\delta_E t = \left\{ t\colon [\phi(t)\cdot\vartheta(t) < 0] \right\} \tag{5.3}$$

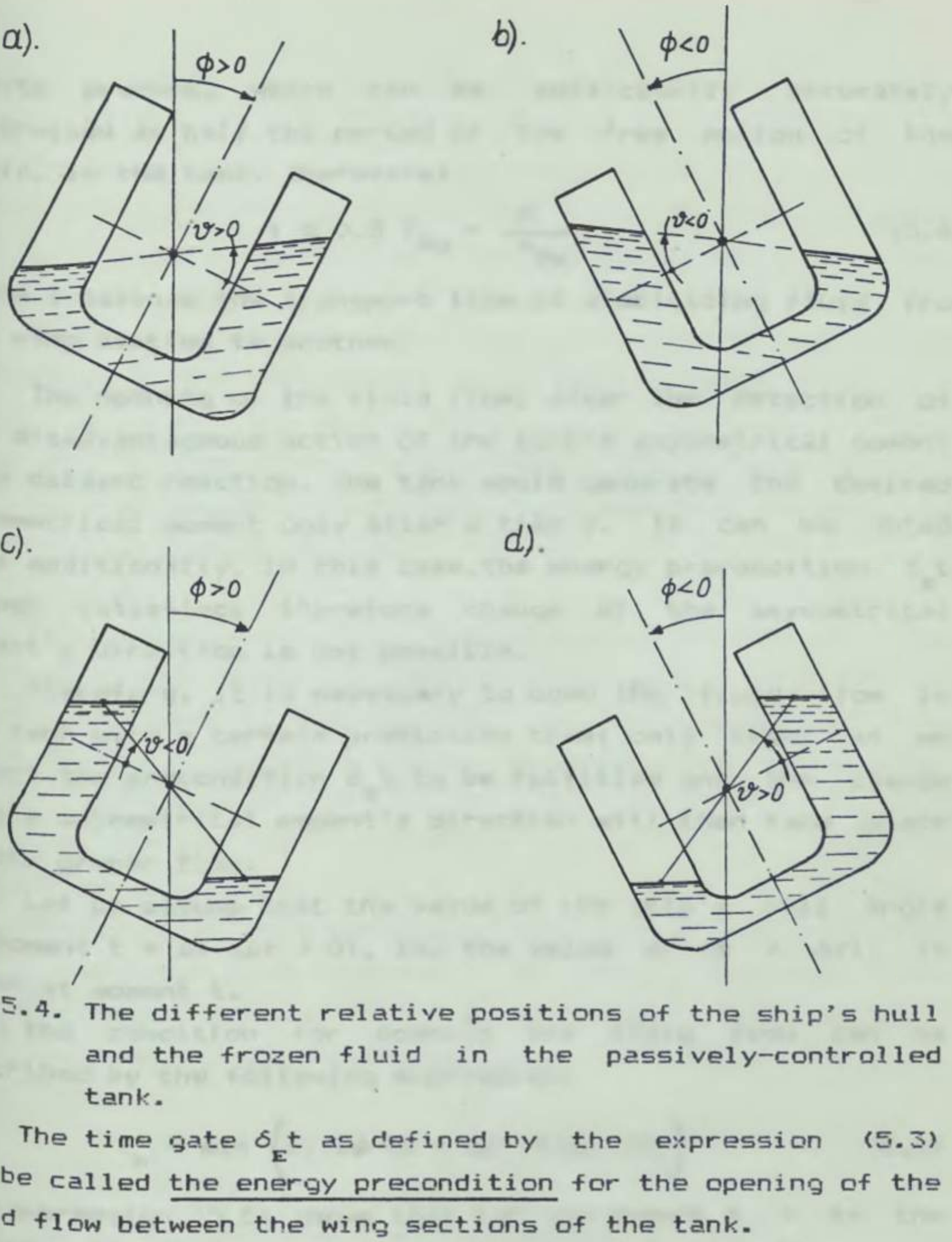

Fig.5.4. The different relative positions of the ship's hull and the frozen fluid in the passively-controlled tank.

The time gate $\delta_E t$ as defined by the expression (5.3) can be called <u>the energy precondition</u> for the opening of the fluid flow between the wing sections of the tank.

5.3 The evaluation of the adequate condition for the opening of the fluid motion in the passively-controlled tank.

The passively-controlled tank with valves blocking the natural fluid motion is an example of the plant with transporting delay. The flow of fluid from one wing section to another is not a timeless (infinite) process. It is a finite process, which can be sufficiently accurately determined as half the period of the free motion of the fluid, in the tank. Therefore:

$$\tau \approx 0.5\ T_{\vartheta o} = \frac{\Pi}{\omega_{\vartheta o}} \qquad (5.4)$$

where τ defines the transport time of stabilizing fluid from one wing section to another.

The opening of the fluid flow, after the detection of the disadvantageous action of the tank's asymmetrical moment is a delayed reaction. The tank would generate the desired asymmetrical moment only after a time τ. It can be noted that additionally, in this case, the energy precondition $\delta_E t$ is not satisfied; therefore change of the asymmetrical moment's direction is not possible.

Therefore, it is necessary to open the fluid flow in the tank with a certain prediction time; only then can we expect the precondition $\delta_E t$ to be fulfilled and the change of the asymmetrical moment's direction will then take place at the proper time.

Let us assume that the value of the ship's roll angle at moment $t + \Delta\tau$ $(\Delta\tau > 0)$, ie. the value ϕ $(t + \Delta\tau)$, is known at moment t.

Then the condition for opening the fluid flow can be described by the following expression:

$$t_o = \min \left\{ t: [\phi\ (t + \Delta\tau)\cdot\dot{\vartheta}(t) > 0] \right\} \qquad (5.5)$$

The expression (5.5) shows that for the moment $t + \Delta\tau$ the tank's action be disadvantageous. If $\Delta\tau > 0$, then the energy precondition $\delta_E t$ is satisfied at the same time.

To ensure the optimal prediction time for fluid relocation, it is necessary to predict the ship's behavior for $\Delta\tau = \tau$. Therefore the condition for opening the fluid flow in the passively-controlled tank can be finally written as follows:

$$t_o = \min \left\{ t: [\phi\ (t + \tau)\cdot\dot{\vartheta}(t) > 0] \right\} \qquad (5.6)$$

5.4 The possibilities of predicting the ship's roll angle.

The prediction of the ship's behavior at sea is not a simple problem and is difficult to realize. For example, paper [5] presents a complex theory for such a prediction, based on the use of the Wiener-Kolmogorov filter.

Several realizable algorithms which make possible the prediction of the ship's roll angle are presented below.

5.4.1 The prediction of the ship's roll angle using Taylor's extrapolation method and Butterworth's filter in filtering the signal.

The determination of the future values of the processing variables is one of the basic problems of engineering today. The prediction of the above variables makes possible in many cases the realization of optimal or sub-optimal control.

The problem of prediction, in our case, can be formulated as follows:

- on the basis of information on the course variable, in the previous time interval, we should evaluate the corresponding Butterworth's approximation (determination of the prefiltered magnitude); this algorithm is described in [35],
- this approximation is extrapolated, expanding it into the Taylor's series.

This can be illustrated by the fig. 5.5.

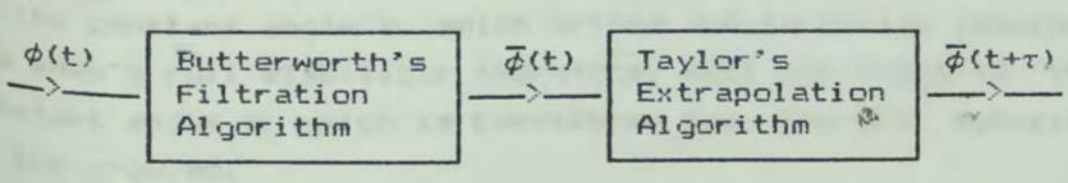

Fig. 5.5 Obtaining the value of the predicted ship's roll angle.

To obtain the value $\overline{\phi}(t+\tau)$ we expand the approximation into the Taylor's series in the following form:

$$\overline{\phi}(t + \tau) = \sum_{i=0}^{n} \overline{\phi}^{(i)}(t)\frac{\tau^{i}}{i!} \tag{5.7}$$

Of course the accuracy and quality of the prediction depends on the number of terms in the series. A larger number of terms makes the prediction more accurate. Practice shows that it is possible to measure easily the angle or the angular velocity of the ship's roll, rarely is it possible to measure the angular acceleration. For each of the above measurements ,it is easy to measure or calculate the following magnitudes: angle, angular velocity and angular acceleration of the ship's roll. Therefore we must limit the number of terms of the series (5.7) to three and we obtain:

$$\overline{\phi}(t + \tau) \cong \overline{\phi}(t) + \tau\frac{d(\overline{\phi}(t))}{dt} + \frac{\tau^{2}}{2}\frac{d^{2}(\overline{\phi}(t))}{dt^{2}} \tag{5.8}$$

The formula (5.8) concerns the measurement of the angle $\phi(t)$. In case of the measurement of the angular velocity the series (5.7) can be written in the following form:

$$\overline{\phi}(t +\tau) \cong \int_{t}^{t+\tau} \overline{\dot{\phi}}(t)dt + \tau\, \overline{\dot{\phi}}(t) + \frac{\tau^{2}}{2}\frac{d(\overline{\dot{\phi}}(t))}{dt} \tag{5.9}$$

Determining the prediction $\overline{\phi}(t+\tau)$, from the formula (5.9) we avoid the necessity to take into account the value of the constant angle ϕ, which arises due to faulty loading. The ship's roll stabilizer therefore, will not react to the constant angle ϕ, which is considered the correct solution of the problem.

The structure of the predictors (5.8) and (5.9) is in principle consistent with the structure of the controllers: PDD^2 (for the form (5.8)) or PID (for the form (5.9)).

Therefore the predictor, in this case can be found in the structural class $PIDD^2$. Such a structure is not difficult to realize.

Algorithm of the real regulator $PIDD^2$

The real controller $PIDD^2$ can be realized through the direct digital control technique in the form of the following algorithm:

1. *position algorithm, in which the actual signal from the regulator is the actual control signal,*
2. *speed algorithm, in which the regulator calculates the actual increment of the control signal relative to the previous signal.*

It was decided that we design a speed algorithm, which is derived directly from it's basic structure, therefore, from the position algorithm. The position structure has the following form:

$$\phi(t + \tau) = P\phi(t) + I\sum_{i=0}^{n} \phi\Big[t - (n - i)\Delta t\Big] + \tag{5.10}$$

$$D\Big[11\phi(t) - 18\phi(t - \Delta t) + 9\phi(t - 2\Delta t) - 2\phi(t - 3\Delta t)\Big]/6\Delta t +$$

$$D^2\Big[2\phi(t) - 5\phi(t - \Delta t) + 4\phi(t - 2\Delta t) - \phi(t - 3\Delta t)\Big]/\Delta t^2,$$

where:

$t = n\Delta t$; Δt – *period of signal sampling $\phi(t)$,*

P, I, D, D^2 – *gain coefficients for the channels: proportinal, integrational, first and second derivatives.*

Due to practical calculations, as well as for the purpose of reduction of the sensitivity of the algorithm, the calculation of the derivatives was done on the basis of the left differences approximation with four point smoothing [21]:

$$\begin{cases} \dot{\phi}(t) = \left[11\phi(t)-18\phi(t-\Delta t)+9\phi(t-2\Delta t)-2\phi(t-3\Delta t)\right] / \, 6\Delta t = \\ \quad = \frac{1}{\Delta t}\left[\nabla\phi(t) + \frac{1}{2}\nabla^2\phi(t) + \frac{1}{3}\nabla^3\phi(t)\right] \\ \ddot{\phi}(t) = \left[2\phi(t)-5\phi(t-\Delta t)+4\phi(t-2\Delta t)-\phi(t-3\Delta t)\right] / \Delta t^2 = \\ \quad = \frac{1}{\Delta t^2}\left[\nabla^2\phi(t) + \nabla^3\phi(t)\right] \end{cases} \tag{5.11}$$

Therefore the speed algorithm of the structure $PIDD^2$ takes the following form:

$$\begin{aligned} \phi(t + \tau) &= \phi(t + \tau) - \phi(t + \tau - \Delta\,) = P\left[\phi(t) - \phi(t - \Delta t)\right] + \\ &+ I\Delta t\phi(t) + \frac{D}{\Delta t}\left[\left[\nabla\phi(t) - \nabla\phi(t - \Delta t)\right] + \frac{1}{2}\left[\nabla^2\phi(t) - \nabla^2\phi(t - \right.\right. \\ &\left. - \Delta t)\right] + \frac{1}{3}\left[\nabla^3\phi(t - \nabla^3\phi\,(t - \Delta t)\right]\Big] + \frac{D^2}{\Delta t^2}\left[\nabla^2\phi(t) - \right. \\ &- \nabla^2\phi(t - \Delta t)\Big] + \left[\nabla^3\phi(t) - \nabla^3(t - \Delta t)\right]\Big], \end{aligned} \tag{5.12}$$

where the integrating element is best realized on the basis of the Adam's method [21]:

$$\int^{t} \phi(t)dt \cong I\Delta t\left[\phi(t)+\frac{1}{2}\nabla\phi(t)+\frac{5}{12}\nabla^2\phi(t)+\frac{3}{8}\nabla^3\phi(t)+\frac{251}{720}\nabla^4\phi)\right] \tag{5.13}$$

The Butterworth sine low pass filter

The Butterworth sine low pass filter - as described in literature [35] - is not only one of the most simple high order filters available but also one of the most effective. The features of this filter as well as the dynamic characteristics of the ship as an object which effectively damps high frequencies, decided the choice of this filter.

The square of the module of the sine transfer function of the Butterworth filter is described as follows:

$$|H(f)|^2 = \frac{1}{1 + \left[\frac{\sin(\pi f \Delta t)}{\sin(\pi f_o \Delta t)}\right]^{2M}}, \qquad (5.14)$$

where:

f_o - *the cut off frequency of the filter,*

M - *order of the filter (the number of poles).*

The practical realization of this filter is relatively simple. It is based on the well known principle, that any high order filter can be realized as a cascade filter, which is made up of first and second order filters connected in chain. From this the equation of the m-th constituent filter has the following form:

$$\overline{y}^{(m)}_{tm}(t) = b_o \overline{y}^{(m-1)}_{tm}(t) - a_{1m} \overline{y}^{(m)}_{tm}(t-\Delta t) - a_{2m} \overline{y}^{(m)}_{tm}(t-2\Delta t) \qquad (5.15)$$

where:

$\overline{y}^{(0)}_{tm} = y_{tm}(t)$, $\overline{y}^{(p)}_{tm}(t) = \overline{y}_{tm}(t)$; P = M/2 *when* M *is even*

P = (M+I)/2 *when* M *is odd,*

b_o - *the constant coefficient, the same for all elements of the cascade filter,*

a_{1m}, a_{2m} (m = 1,...,P) - *the coefficients of the sine filter, where, if* M *is odd then the last* P-*th coefficient is* $a_{2p} = 0$.

The coefficients a_{1m} *and* a_{2m} *as well as* b_o *are calculated according to the Otnes algorithm* [35].

It is necessary to prove that for such a complex structure of the prediction device, it is possible to obtain a prediction time according to the expression (5.4) and to use PID or PDD^2 structures (dependant on the measurement possibilities of the process variables) for the control of the passively-controlled tank.

We will prove the case of the PDD^2 structure (during the measurement of the ship roll angle ϕ) for the regular exciting moment.

With sufficient accuracy we can approximate the roll of the ship stabilized by the passively-controlled tank on regular wave to a harmonic one (a more precise explanation is given in chapter 7.3.3), which can be expressed by the equation:

$$\phi(t) = \phi_A \sin(\omega_E t - \varepsilon) \tag{5.16}$$

where:

$$\phi_A \cong \frac{\alpha_A \varkappa_\phi(\omega)\omega^2_{\phi_o}}{\sqrt{4\beta^{*2}_\phi \omega^2_{\phi_o}\omega^2_E + (\omega^2_{\phi_o} - \omega^2_E)^2}}$$

$$\varepsilon = \mathrm{arctg}\,\frac{2\beta^*_\phi\, \omega_{\phi_o}\omega_E}{\omega^2_{\phi_o} - \omega^2_E}$$

According to the expression (5.8) and using (5.16) the prediction of angle $\phi(t + \tau)$ can be formulated:

$$\begin{cases} \phi(t+\tau) = P\sin(\omega_E t-\delta) + D\omega_E\cos(\omega_E t-\delta) + D^2\omega^2_E\sin(\omega_E t-\delta) \\ \delta = \mathrm{arctg}\left(\dfrac{2\beta^*_\phi\, \omega_{\phi_o}\omega_E}{\omega^2_{\phi_o} - \omega^2_E}\right), \end{cases} \tag{5.17}$$

where:

P, D, D^2 – *static gain coefficients in the channels of angle, angular velocity and angular acceleration of the ship's roll motion.*

After compounding the harmonics we get:

$$\begin{cases} \phi(t+\tau) = \sqrt{(P + D^2\omega^2_E)^2 + (D\omega_E)^2}\,\sin(\omega_E t-\delta+\sigma) \\ \sigma = \mathrm{arctg}\,\dfrac{D\omega_E}{P + D^2\omega^2_E}, \end{cases} \tag{5.18}$$

where δ is defined as in expression (5.17).

Converting the prediction time τ given by the expression (5.4) to the corresponding prediction angle σ:

$$\sigma = \omega_E \tau = \frac{\omega_E \cdot \pi}{\omega_{\vartheta_0}} \tag{5.19}$$

Therefore to obtain the searched for prediction time it is necessary to choose the static gain values P,D,D^2 in such a way so that they satisfy the relation:

$$\frac{\omega_E \pi}{\omega_{\vartheta_0}} = \text{arctg} \left[\frac{D\omega_E}{P + D^2\omega_E^2}\right] \tag{5.20}$$

Let's examine the expression (5.20) in resonance ($\omega_E = \omega_{\phi_0}$). The frequency of the passively-controlled tank is chosen, ordinarily, as the highest frequency occuring during the ships operation. This usually brings us to the dependence $\omega_{\vartheta_0} \in (1.5,\ 2)\omega_{\phi_0}$. Let's accept the higher value: $\omega_{\vartheta_0} = 2\omega_{\phi_0}$

and we receive from (5.20):

$$\frac{\pi}{2} = \text{arctg} \frac{D\omega_{\phi_0}}{P + D^2\omega_{\phi_0}^2} \tag{5.20a}$$

If the equation (5.20a) is to be fulfilled then the term $P + D^2\omega_{\phi_0}^2$ should be zero and D should not be equal to zero. Analysing (5.20) and (5.20a) the following conclusions can be made:

(i) *the achievement of searched for prediction time τ, determined by the expression (5.6) is possible for the frequencies of the ship's roll; this is true for the resonance, directly from the expression (5.20a); for the whole range of frequencies, this will be done in the digital and physical simulation process (chapter 8),*

(ii) *the static gain coefficient in channel $\dot{\phi}$ (angular velocity) must be different from zero.*

(iii) *it is possible to have an infinite combination of settings of P,D,D^2 in the respective channels of the prediction structure, which ensure different prediction times.*

(iv) *in the case of D^2 having a positive value, then the coefficient P should be negative and vice versa.*

The above mentioned hypothesis of the prediction of the ship's roll angle as well as the short proof which shows the possibilities of its applications were developed using certain simplifications.

Therefore it is necessary to look at this analysis critically, in the sense that it is a starting point for simulation experiments. During these experiments, using static optimization procedures it is necessary to choose the gain coefficients P, D and D^2 for concrete conditions.

5.4.2 Prediction of the ship's roll angle using the Kalman filter and the k-step Kalman's predictor.

In the last few years, we have seen the development of control methods based on the theory, developed by a group of American scientists in the 1960's. These methods based on optimal control with minimal variance, k-step prediction and on-line variable identification. These methods make possible the realization of complicated control algorithms. Today such algorithms can be realized due to the use of computers.

Presented below is an application proposition of an original method developed to control the work of the passively-controlled tank's blocking valves.

The starting point is the development of the model of the system's dynamics to synthesize the Kalman filter. The model of the system ship - tank is derived directly from the set of equations (3.13) and is as follows:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_{1f} \\ \dot{x}_{2f} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -K\left[1+\Gamma\frac{s}{g}\omega_\vartheta^2\right]\omega_{\phi_o}^2; & -2K\beta_\phi\omega_{\phi_o}; & K\left[1+\frac{s}{g}\omega_\vartheta^2\right]\Gamma\omega_{\phi_o}^2; & K\Gamma\frac{s}{g}\omega_{\phi_o}^2\,\omega_\vartheta^2 2\beta; & 0; & Kk_f\left[\omega_{\phi_o}^2+\Gamma\frac{s}{g}\omega_{\phi_o}^2\,\omega_\vartheta^2\right] \\ 0 & 0 & 0 & 1 & 0 & 0 \\ K\left[1+\frac{s}{g}\omega_{\phi_o}^2\right]\omega_\vartheta^2; & 2K\beta_\phi\omega_{\phi_o}\frac{s}{g}\omega_\vartheta^2; & -K\left[\Gamma\frac{s}{g}\omega_{\phi_o}^2+1\right]\omega_\vartheta^2; & -2K\beta_\vartheta\omega_\vartheta; & 0; & -Kk_f\left[\omega_\vartheta^2+\frac{s}{g}\omega_\vartheta^2\omega_{\phi_o}^2\right] \\ 0 & 0 & 0 & 0 & 1 & 1 \\ 0 & 0 & 0 & 0 & -\left[\alpha^2+\beta^2\right] & -2\alpha \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_{1f} \\ x_{2f} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 1 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \eta_1 \\ \eta_2 \end{bmatrix} \qquad (5.21)$$

where:

$$K = \left[1 - \Gamma\frac{s^2}{g^2}\omega_{\phi_o}^2\,\omega_\vartheta^2\right]^{-1}$$

The model (5.21) is a aggregated model of the ship's roll dynamics, the motion of the fluid in the tank as well as dynamic disturbances. The structure of this model can be more simply written as:

$$[\dot{x}] = \left[\begin{array}{c|c} A_{11}(4,4) & K_{21}(4,2) \\ \hline 0\ (2,4) & A_{22}(2,2) \end{array}\right][x] + D[\eta], \qquad (5.22)$$

where:

A_{11} - block sub matrix describing the associated dynamics of the ship and tank,

A_{22} - block sub matrix describing the disturbance formulating filter from the Gaussian white noise,

K_{21} - sub matrix scaling the colored noise, inputting into the object,

0 - zero sub matrix,

$\dot{x}$, x - the respective vectors: the derivatives of the state variables and the state variables defined as follows:

$x_1 \longrightarrow \phi$

$x_2 \longrightarrow \dot{\phi}$

$x_3 \longrightarrow \vartheta$

$x_4 \longrightarrow \dot{\vartheta}$

$x_{1f} \longrightarrow$ first coordinate of the formulating filter,

$x_{2f} \longrightarrow$ second coordinate of the formulating filter.

The formulating filter was chosen [57] in such a way so as to map, with the help of rational functions, the ITTC spectrum. The form of the transfer function of the filter is as follows:

$$H_{\alpha}(s) = \frac{\alpha_E(s)}{\eta_2(s)} = \frac{\sqrt{2\alpha D_r}}{g} \frac{s}{s^2 + 2\alpha s + (\alpha^2 + \beta^2)}, \qquad (5.23)$$

where:

D_r - variation of the sea wave, as a stationary and ergodic random process with a average zero value,

α, β - coefficients of the filter,

$\frac{\sqrt{2\alpha D_r}}{g} = k_f$ - gain coefficient of the formulating filter,

η_1, η_2 - Gaussian white noises of zero average value, and respective variations of λ_1^2 and λ_2^2.

The model (5.21) describes the dynamics of the ship's roll, the oscillations of the fluid in the tank and external disturbances in the form of a set of six first order differential equations. The description of the states: passive and frozen can be obtained from the following:

- $\omega_\vartheta = \omega_{\vartheta_o}$ for the passive state
- $\omega_\vartheta = 0$ for the frozen state.

Due to the control of the passively-controlled tank, it is important to estimate the object's state vector, in the frozen state of the system. Therefore the model (5.21) is greatly simplified. Further simplification can be derived from the fact that the value of the inertia coupling coefficient s, influences the dynamics of the system ship - passively-controlled tank in a very small way [20,61].

Therefore assumption $s = 0$ is justified especially because the filter structure is constant. After such simplification the model (5.21) can be formulated as:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_{1f} \\ \dot{x}_{2f} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ -\omega^2_{\phi_o} & -2\beta_\phi\omega_\phi & \Gamma\omega^2_{\phi_o} & 0 & k_f\omega_{\phi_o} \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & -\left(\alpha^2+\beta^2\right) & -2\alpha \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_{1f} \\ x_{2f} \end{bmatrix} +$$

$$
+\begin{bmatrix} 000 \\ 100 \\ 010 \\ 000 \\ 001 \end{bmatrix} \begin{bmatrix} \eta_1 \\ \eta_2 \\ \eta_3 \end{bmatrix} \tag{5.24}
$$

The white noise in the third equation of (5.20), was introduced so as to avoid zero values of the Kalman filter's corrector, from the state variable x_3. The model (5.24) is a continuous model, which contains a few inconveniences in the construction of the Kalman filter. Due to this it is most convenient to transform the continuous model (5.24) into a discrete model using the method given in [21]. The discrete model is obtained as follows:

$$
\begin{cases} x_{k+1} = Fx_k + b\eta_k \\ y_k \ \ = Cx_k + \nu_k \end{cases} \tag{5.25}
$$

The second equation of the model (5.25) is an equation of outputs (measurements) and ν_k is the vector of measurement noises. In the system, we measure two outputs; ie. the roll angle and angular position of the fluid in the tank or (the second possibility) the angular velocities of the roll and fluid motion in the tank.

In the first case the matrix C takes the following form:

$$
C_1 = \begin{bmatrix} 10000 \\ 00100 \end{bmatrix}
$$

and in the second one:

$$
C_2 = \begin{bmatrix} 01000 \\ 00100 \end{bmatrix}
$$

Now it is easy to determine the value of the gain matrix, ofthe stationary Kalman filter's corrector. This matrix is obtained through the digital solution of the Riccati's discrete equation [7, 16], where the input data are the elements of the matrix F,b and C_1 and C_2.

The estimated values of the state vector $\hat{x}_k$, can be used now for k-step prediction. The algorithm of the predictor can be easily determined [7, 16], and in the case of the model (5.25) has the form:

$$x_{k+n} = F^n x_k \tag{5.26}$$

Due to the impossibility to measure the vector x_k (as well as due to measurement noises) it is best to replace the vector x_k with the vector $\hat{x}_k$. Then we finally obtain:

$$\hat{x}_{k+n} = F^n \hat{x}_k \tag{5.26a}$$

Fig 5.6 shows the block diagramme of the control system in the case of the measurement of the ship's roll angle and the position of the fluid in the tank.

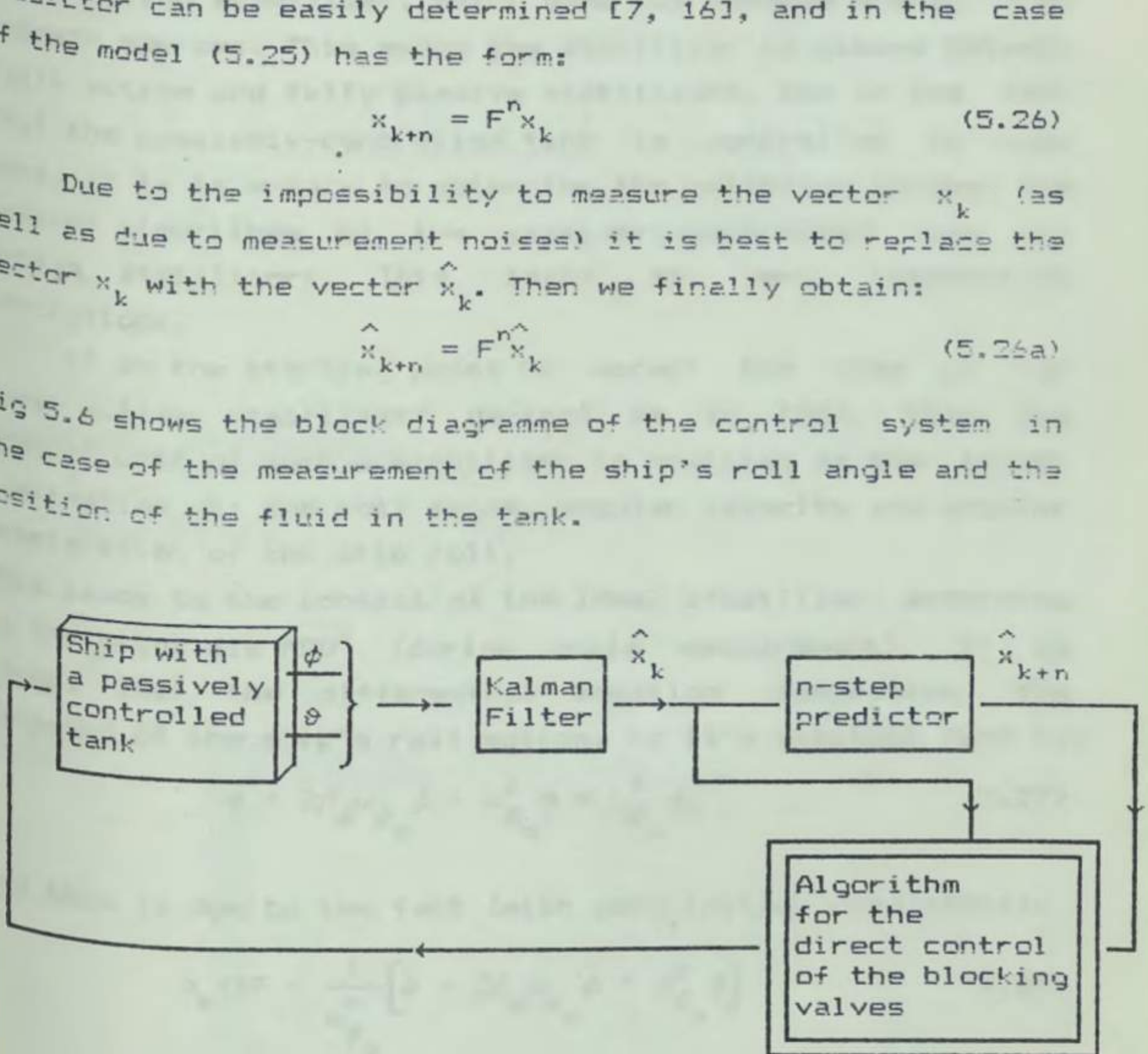

Fig. 5.6 General block diagramme for control of the ship's roll stabilizing system using the passively-controlled tank.

5.5 Comments on the control of the passively-controlled tank.

From the above assumptions we can see that there are many ways to solve the control algorithm of the passively-controlled tank. Fig. 5.7 shows the general block diagramme for control of the passively-controlled tank.

The passively-controlled tank is a special type of anti-rolling stabilizer, which does not consume energy from onboard sources. This means the stabilizer is placed between fully active and fully passive stabilizers. Due to the fact that the passively-controlled tank is controlled in some ways, it is necessary to determine the relations between the control algorithms of the passively-controlled tank and active stabilizers. This leads to very interesting conclusions.

If as the starting point we accept the idea of "an ideal active stabilizer" defined as in [55], then the control unit of such a stabilizer is realized as the linear combination of: the roll angle, angular velocity and angular acceleration of the ship roll.
This leads to the control of the ideal stabilizer, according to the principle PDD^2 (during angle measurement). If we accept that the differential equation describing the dynamics of the ship's roll motion, in it's simplest form is:

$$\ddot{\phi} + 2\beta_{\phi}\omega_{\phi_0}\dot{\phi} + \omega_{\phi_0}^2\phi = \omega_{\phi_0}^2\alpha_E \qquad (5.27)$$

and this is due to the fact (with zero initial conditions):

$$\alpha_E(t) = \frac{1}{\omega_{\phi_0}^2}\left[\ddot{\phi} + 2\beta_{\phi}\omega_{\phi_0}\dot{\phi} + \omega_{\phi_0}^2\phi\right] \qquad (5.28)$$

From the equation (5.28) we can conclude that the structure PDD^2 is in principle an observer (estimation plant) of the running values of the effective wave slope. Therefore the control of the ideal ship's roll stabilizer is

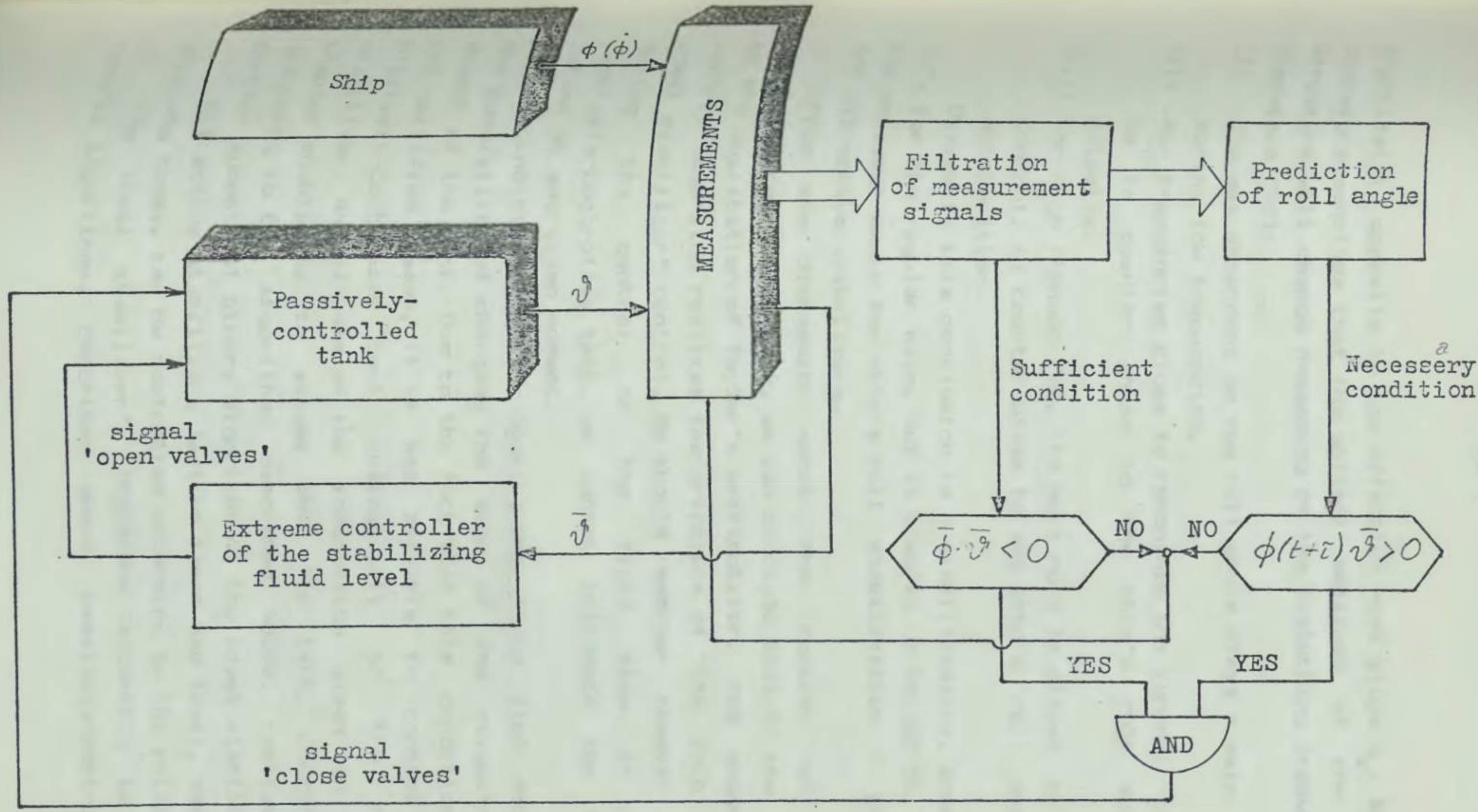

Fig.5.7. The block diagramme for control the blocking valves in the passively-controlled tank.

practically opposite to the effective wave slope α_E. We can therefore conclude that the optimal settings of the PDD^2 structure will change depending on the disturbing frequency. Therefore [55]:

(i) the set dependent on the roll angle plays a main role during low frequencies,

(ii) for frequencies close to resonance the control should be in counter phase to the ship's roll angular velocity,

(iii) for high frequencies, the main role is played by the control, in counter phase to the ship's roll angular acceleration.

Obviously this conclusion is a simplification, possible only for the regular wave, but it enables us to get to know the physical basis for ship's roll stabilization - common for all active stabilizers.

From the discussion about the control of the passively-controlled tank, we can conclude that in the case of the application of Taylor's extrapolator, the suggested control algorithm realizes the principle of "the ship roll ideal stabilizer" control. We should remember however that during the control of the fluid flow in the passively-controlled tank, we cannot influence the fluid column at any given moment.
The precondition (5.3) for opening the fluid flow decides the possibility of changing the sign of the asymmetrical moment of the tank. Due to the fact that this condition is not satisfied always, it is not possible to control the passively-controlled tank analogically to the ideal stabilizer. An analysis of the information given in this chapter enables as to assume that the tank, controlled according to the algorithms described above, realizes a certain suboptimal binary algorithm of the ideal stabilizer.

All active stabilizers (active tanks and fins), used in shipping today, can be controlled according to the principle of "the ideal stabilizer". Therefore according to the control algorithms, described above, passively-controlled tanks realize principle of "the binary ideal stabilizer" control.

If the control algorithms of the passively-controlled tank using filtration and ship's roll angle prediction, are equivalent to the algorithm of control in counter phase to the disturbance, then it is possible to use the optimal state feed-back controller algorithm, for control of the passively-controlled tank. This algorithm is based on the adaptation of the principle, described in [46], where this algorithm was used in active stabilizing fins control. The adaptation of the optimal control algorithm as described among others in [16, 21]:

$$u(t) = k^T \hat{x}(t), \qquad (5.29)$$

for the passively-controlled tank is simple. If the signal $\dot{u}(t)$ determines the optimal control of the continuously operating ship roll stabilizer, then its use, in the control of the passively-controlled tank can be realized as:

$$t_o = \min\left\{t: \left[\left[\phi(t)\cdot\theta(t)<0\right]\wedge\left[\dot{u}(t)\cdot\theta(t)>0\right]\right]\right\} \qquad (5.30)$$

In the other words, the replacement of the continuous algorithm by the bistate algorithm leads to the replacement of the value $u(t)$ by the value $\phi(t+\tau)$ (vide fig. 5.7).
The control hypothesis (5.30) of the passively-controlled tank will be checked experimentally during simulation tests (chapter 8).

The synthesis of the regulator type. (5.29) is based on the system model given by the expression (5.21) or the more simple formula (5.24). It is easier to use the formula (5.24) in the synthesis of the regulator, due to the changes in the dynamics of the tank during valves operation. Therefore it is best to estimate only the ship's movement component and excitement, and to measure the component θ *(state variable* x_3*).*
In this way we can avoid the neccesity for continuosly tuning the Kalman filter.

6. THE PROBLEMS OF THE STABILITY OF THE SYSTEM: SHIP - PASSIVELY-CONTROLLED TANK.

The problem of the stable operation of the ship roll passively-controlled tank stabilizer has to be examined from two points:

- theoretical - which has as its aim the definition of the general stability conditions, for example : the determination of the range of settings, of the prediction structure PDD^2(PID) and the tank dimensions (parameters) which would ensured the stable operation of system,
- technical - which leads to the selection of the stable prediction structures as well as tank parameters, through practical tests and suitably directed simulation experiments.

6.1 Formulation of the problem of the system's theoretical stability.

The formulation of the theoretical stability of the system is not difficult. In the linear systems stability theory, there exists the method of root position ((modes) investigation of the characteristic equation, in other words the investigation of the stability ex definition. There also exist mnemo-technical methods (for example Ruth - Hurwitz criterion, Mihailov criterion, Nyquist's criterion, etc.). They give a explicit answer to the question of the stability conditions. This answer is independent of the external disturbance values (it is the characteristic of linear systems).

In the case of non-linear systems, momentarily neglecting the many types and subtypes, the explicit determination of stability conditions, is much more difficult. The non-linear system stability, is dependent on the values of the external disturbances, therefore,the determination of the solution for a certain class of inputs does not solve the problem generally.

The general-purpose method for all cases (in literature we find about 30 different definitions of the stability of non-linear systems [16, 21]), of the evaluation of non-linear system stability, is to prove that in the case of the limited input the system, output is also limited. This limitation is understood in such a way ,that, if $\xi(t)$ symbolizes the i-th coordinate of the state vector, or the vector of system output, then $\xi(t)$ is limited, when for the time $t \in (0, \infty)$ the following condition is satisfied:

$$\left|\xi_i(t)\right| \le R, \tag{6.1}$$

where R - finite constant.

There exists a whole range of methods to solve this problem (Lapunov's methods, phase space method, etc.).
The analytical solution of this problem comes across a whole range of difficulties, connected with the peculiarities of the passively-controlled tank system.This is a system with a controlled parameter. Therefore the control of this system does not have the typical character of control through an energy flow.

Parametric systems, the control of which, is connected with a minimal consumption of energy does not yet have a sufficiently developed theoretical base. Attempts are being made to formulate such a theory [28].
As yet only the problem of local stability of the harmonic oscillations in periodic changing parameter systems has been solved. This stability can be determined for such a system, only during operation in non-zero initial conditions. The heeling test of the ship is such a situation, but even in this case the following relationship is not satisfied:

$$\omega_\vartheta(t) = \omega_\vartheta\left(t + \frac{2\pi}{\Omega}\right) \tag{6.2}$$

therefore the control parameter is not a periodic one.

The method presented above is connected with the investigation of the limitation of the system output, during the limitation of the system input, and in our case is also

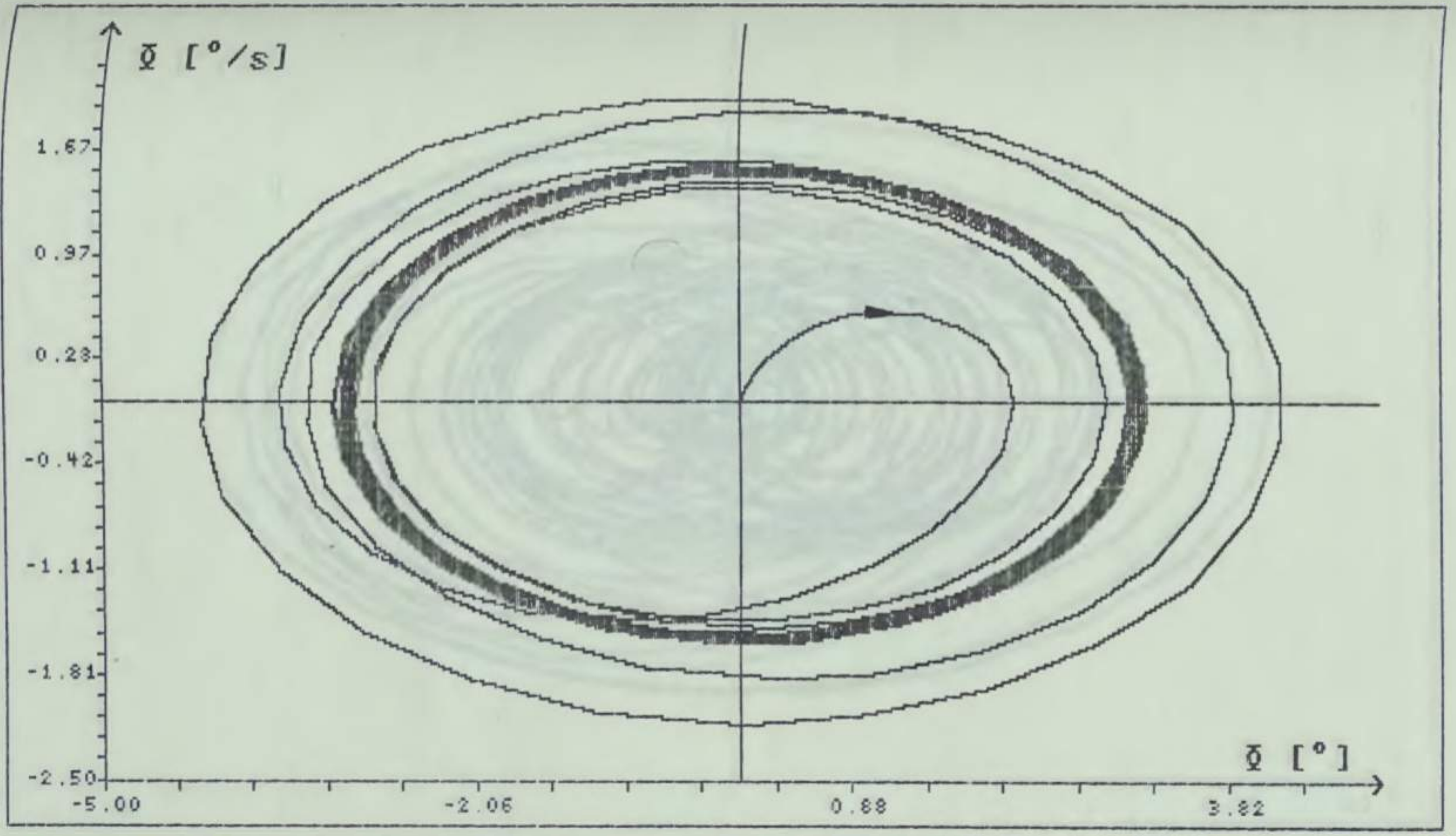

Fig.6.1. The phase trajectory of the ship stabilized by passively-controlled tank; the results of numerical simulation in regular waves.
Wave slope angle - 1.5°, wave frequency - 0.499 rad/s

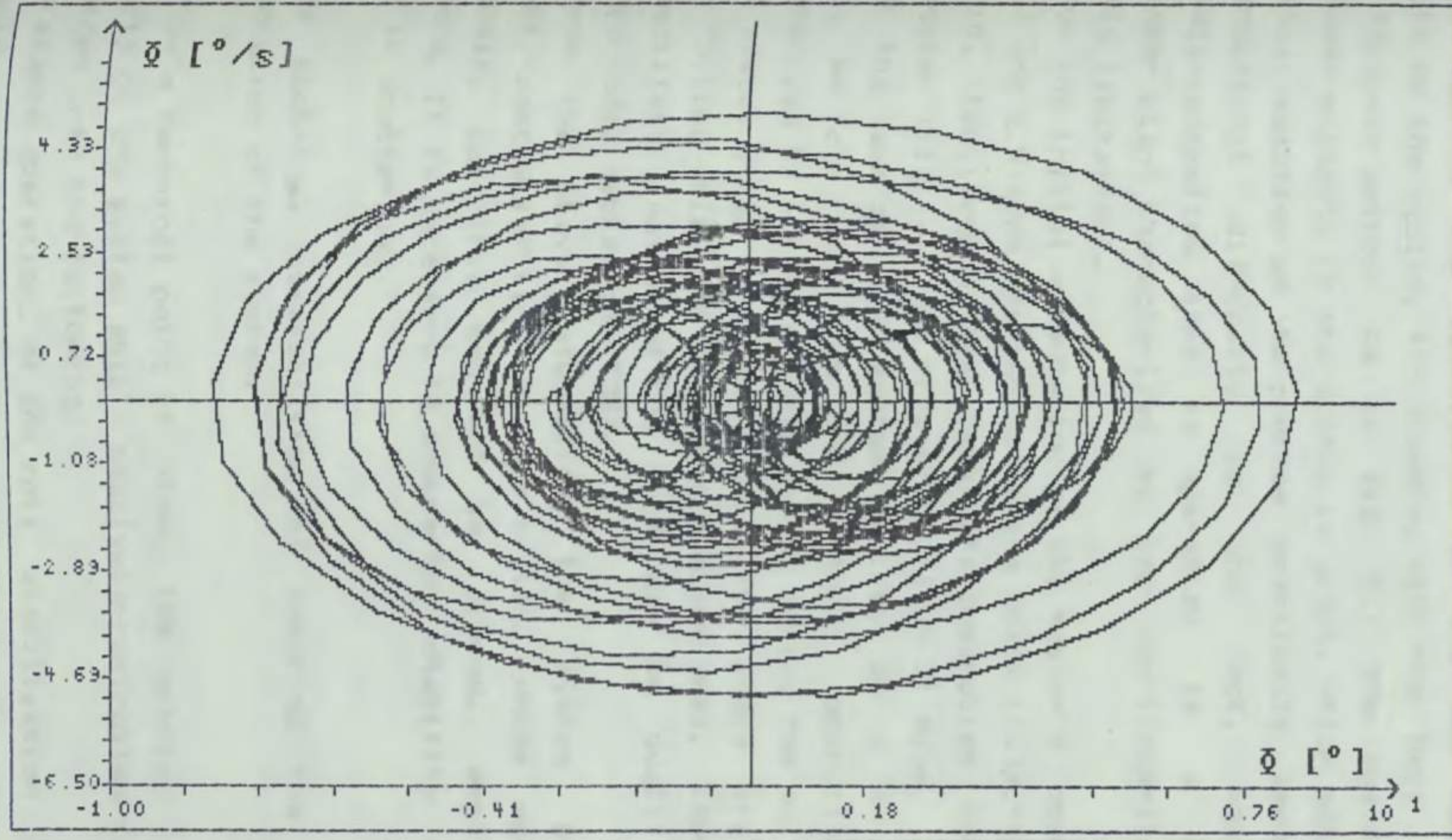

Fig.6.2. The phase trajectory of the ship stabilized by passively-controlled tank; the results of numerical simulation in irregular beam seas.
Sea state: $h_{1/3}$ = 4.5 m, T = 9.35 s

difficult to realize. The analytical solution of this problem, demands the description of all the non-linear elements of the system, for example, with the help of the first harmonic method. As in fig. 5.7 the quantity of non-linear elements in the system is great, which makes the analytical solution of the problem practically impossible. An additional difficulty is the fact, that the passively-controlled tank in operation is a dynamic non-linear plant characterized by the non-linearity type "dynamic limitation".

For the initial orientation of the system's operation, fig.6.1 and 6.2 show the course of the phase trajectories of the ship, stabilized by the passively-controlled tank, on the regular (fig 6.1) and irregular (fig.6.2) wave.

In the case of the system excited by a sinusoidal signal, we can see that the system's operation, is characterized by a stable limit cycle. During the excitation of the system by an irregular signal, the phase trajectory of the rolling ship runs through a limited area, therefore, this oscillatory motion of the ship can be qualified as quasi-periodic vibrations [28].

From the above information, the system ship - passively-controlled tank can be a system, whose operation is stable, but this can not be proved, explicitly. Therefore, it is necessary to ensure the stability of the system in another way.

6.2 The technical possibilities of ensuring the stable operation of the system.

From a technical point of view, the problem of the stability of the system ship - passively-controlled tank can be divided into the following:

a) the stable operation, of the roll stabilization system, should ensure a great reduction of the roll amplitudes; the scale of the roll reduction can be observed on the basis of the efficiency criteria (especially the big decrease in the quadratic criterion (4.6)),

b) it is necessary to ensure the minimal reduction of the ship's natural righting moment, caused by the installation of the stabilizing tank; it secures the minimal reduction of the transverse stability of the ship, which is a deciding factor in avoiding of the capsize of the ship,

c) it is necessary to ensure stable operation of the valves blocking the fluid flow in the tank.

This means, that the problem of the stability, leads to the maximal reduction of the ship's roll motion. The chosen method of synthesis of the control system, based on the computer simulation of the roll process, with the simultaneous use of the static optimization procedures, allows us to solve this problem. The unstability of the system, causes the increase of the roll amplitudes of the stabilized ship (sometimes to a greater level, than the unstabilized ship's roll). The quality of stabilization, then greatly deteriorates. Due to the fact, that the optimization procedures used in the control synthesis, automatically eliminates the excessive increase of the variation criterion. The cases of unstability of the system are also eliminated automatically.

The limitations, due to the classification rules, have to be taken into account in designing the tank stabilizer, as this has a great influence on the seakeeping qualities of ship. These rules, enforce a strict limit on the maximum static moment of the stabilizer. This ensures, in the case of the defective operation of the stabilizer, required stability of the ship. These rules, limit the geometrical dimensions of the tank (passive and passively-controlled), as well as the quantity of fluid in the tank. The stabilizer moment is generally given in the form of its static characteristic: the stabilizer wave slope capacity. According to the rules of classification, the wave slope capacity of a roll stabilizer cannot exceed 5°.

Although this limitation is strictly a static one, in the case of tank stabilizers it is closely connected to the dynamics of the ship's roll motion.
It is known that the installation on the ship, of a free surface tank, decreases the initial metacentric height of the ship. This feature takes the following form:

$$\frac{\Delta(GM)}{GM_o} = \frac{I_T \rho_T g}{\Delta GM_o} \tag{6.3}$$

The formula (6.3), expresses the relation of the tank heeling moment to the ship's righting moment.

The typical passive tanks causes a reduction of 20÷30%of the ship's initial metacentric height. The passively-controlled tank generates the stabilizing moment, mainly due to high amplitudes of the stabilizing fluid. Therefore, it is designed, so as to reduce the initial metacentric height by 15÷20%. We can see from this, that the passively-controlled tank causes a lesser decrease in the stabilizing moment of the ship, in comparison to the passive tank. Therefore, the installation of a passively-controlled tank does not greatly deteriorate the ship's stability.

We should now analyze extreme cases of failure of the fluid flow control in the passively-controlled tank, as this could be a source of danger to the ship. Two extreme cases of failure are:

1. when it is not possible to close the blocking valves; in this case the passively-controlled tank behaves like a ordinary passive tank of high natural frequency; the complete solution of the problem of stability of the system ship - passive tank can be found in [55]. From the analysis contained in the book, it seems that for all typical shapes of the U-tube passively-controlled tank the stability conditions are fulfilled,
2. when it is not possible to open the blocking valves, while the difference between the levels of fluid in the wing sections is at its greatest. In this case it is

necessary to treat the tank as a source of continuous heeling moment; this does not however endanger the ship.

Due to the system not being hermetically tight, there exists a certain loss, of the difference between the levels in the wing sections of the tank. Moreover it can be observed that, under the condition that the tank moment is limited to the values recommended by the above mentioned classification rules, this break-down does not endanger the ship.

In [29], we find an original method to determine the conditions for the ship to capsize. This could be applied in a wider analysis of the results of the fluid blockage in the maximal position.

The analysis of the problem of the stable operation of the blocking valves, can be considered as follows. The analysis of the control algorithm shows that the unstable operation of the valves could take place,

1. directly after the blocking of the fluid flow, if this blockage happened due to the fluid reaching the maximum position (saturation angle ϑ_N) but not due to the action of the extreme regulator. In this case, the condition for the opening of the valves will be fulfilled, soon after their closing, etc.,
2. directly after the release of the natural fluid flow in the tank (the opening of the valves); in this situation the condition for the closing of the valves will be fulfilled at once, etc.

In both these cases the stable operation of the valves is ensured by suitably delaying their next action. In the first case, the opening of the valves can start only after the sign change of the ship's roll angle value $\phi(t)$.

In the second case, it is necessary to enforce delay, dependant on the non-zero value of the angular velocity of the fluid motion $\dot{\vartheta}(t)$.

We can see that the general problem of the stability of the passively-controlled tank is difficult to solve. Leaving this question open, we note from the facts mentioned above, that the installation of the passively-controlled tank does not introduce into the system, elements of the ship's roll unstability.

Chapter 7

7. MODEL EXPERIMENTS.

The actual state of knowledge of passive tanks, allows us to determine the amplitude frequency characteristic of the ship stabilized by this tank, when we know the tank's parameters. Normally these characteristics are shown in the form:

$$Y_{\phi\alpha}(\omega) = \left.\frac{\Phi_A(\omega)}{\alpha_A(\omega)}\right|_{\alpha=\mathrm{const}} \tag{7.1}$$

where:

$\Phi_A(\omega)$ - the ship's roll amplitude,

$\alpha_A(\omega)$ - the wave slope amplitude.

Most of the tank parameters are geometrical ones, ordinarily defined by the ship's designer, therefore, their determination is not difficult. However some of these parameters (damping coefficient of the fluid motion, tank's natural frequency in non-typical constructions and - in a smaller way - the inertia coupling coefficient) cannot be defined sufficiently accurately, theoretically or by calculation. They can be determined however experimentally. In this case we can also investigate the influence of the "saturation" of the tank, during high amplitude values of the stabilizing fluid.

Experiments conducted on ship's models, equipped with thestabilizing tank on regular and irregular waves give us the possibility of directly determining, the effects of the tank's installation in the ship's hull. The experiments on the regular wave, directly determines the characteristic (7.1), and the experiments on the irregular wave, give information about the ship's roll motion, in concrete weather and loading conditions.

The above mentioned steps are however troublesome and costly, and the obtained results are difficult to generalize.

Therefore most of the experiments are conducted only for the separated tank with the aid of a special dynamometrical cradle (the bench test mechanism) [64].

The basis of these experiments, is that the model of the tank is placed on the self-aligning table of the bench mechanism, which is in harmonic motion of constant amplitude and frequency. During this motion, the total moment due to hydrodynamic forces caused by the fluid flow relative to the roll axis and the phase angle (between this moment and the bench mechanism motion) are measured.

If the tank's motion is in the form:

$$\phi(t) = \phi_A \sin \omega t \Big|_{\phi_A = \text{const}} \tag{7.2}$$

then the hydrodynamical moment of the fluid reaction, relative to the roll axis can be written as:

$$M_H = M_A(\omega) \sin(\omega t - \varepsilon_M) \tag{7.3}$$

The relationships:

$$\begin{cases} M_A = M_A(\omega) \Big| \; \phi_A = \text{const} \\ \varepsilon_M = \varepsilon_M(\omega) \Big| \; \phi_A = \text{const} \end{cases} \tag{7.4}$$

give us the searched for characteristics. These characteristics, make possible the determination of the tank characteristics against the stabilizing fluid volume, roll amplitudes, dimensions and shape of the tank, construction of damping elements of the fluid motion in the tank, etc. It is obvious that in recalculating the characteristics from model scale to full scale, Froude and Strouhal numbers should be equal for the model and the real tank [66].

Due to the fact that the bench mechanism measures the total moment (the hydrodynamic forces moment and reaction moment of the "frozen" fluid, the tank casing and table), it is necessary to calibrate the test station before experiments.

The influence of the "frozen" fluid and tank's casing is determined through special tests, most often through simulation of the fluid by a solid body.

If we designate the reaction of the tank's model and its phase angle as M'_A and ε'_M, and reaction and phase angle of the tank casing and the "frozen" fluid as M''_A and ε''_M, then the real hydrodynamic force reactions and their phase angle are given by:

$$\begin{cases} M_A = \sqrt{(M'_A\cos\varepsilon'_M - M''_A\cos\varepsilon''_M)^2 + (M'_A\sin\varepsilon'_M - M''_A\sin\varepsilon''_M)^2} \\ \varepsilon_M = \mathrm{arctg}\dfrac{M'_A\sin\varepsilon'_M - M''_A\sin\varepsilon''_M}{M'_A\cos\varepsilon'_M - M''_A\cos\varepsilon''_M} \end{cases} \tag{7.5}$$

The analysis, with the help of (7.5) must be conducted for a few values of ϕ_A and for a wide frequency range. From these results characteristics (7.4) can be obtained.

In the case of the passive tank, this leads to the determination of optimal tankage, as well as the placing of the damping elements, in such a way so as to attain the maximal tank moment and the correct course of characteristics (7.4).

For the passive tank, this stage usually ends the tests, but for the passively-controlled tank this is only the initial stage of the tests.

7.1 Tests of the passively-controlled tank on the bench test mechanism.

The basic difficulty during the testing of the passively-controlled tank on the bench test mechanism is the absence of a preferred standard for such tests by ITTC. The above mentioned methods are standard only for passive tanks and are applied by most of the ship model basins in the world.

Difficulties during the testing of the passively-controlled tank on the bench test mechanism arise due to the following:

1. the behavior of the passively-controlled tank mainly depends on the actual control algorithm; this is the basic difference between the passive and passively-controlled tank,
2. very quick changes of the moment generated by the passively-controlled tank.

The technical equipment necessary for testing the passively-controlled tank on the bench mechanism consists of:

- bench test mechanism designed by Polish Ship Research Center (CTO) in Gdańsk [64],
- tensometer bridge produced by Hottinger (W. Germany) type KWS 3071, with an accuracy of 1%,
- linear motion sensor type KWS 3070 with an accuracy of 1% produced by Hottinger,
- thyristor controller of the DC electric motor used for stabilization of motor revolutions (with the aid of a tachometer),
- set of measurement equipment produced and designed by the firm Solatron (Great Britain) consisting of:
  (i) - mechanical reference resolver type JX 1606,
  (ii)- digital transfer function analyzer type JX 1600, with an accuracy of up to 0.5%.

The model of the passively-controlled tank, made of steel, was built in 16,3 scale and was additionally equipped with a capacitance meter type S70. This is a part of the measurement system Silometer S7/S70, produced by Endress & Hauser with an accuracy of up to 1%.

The direct control of the blocking valves, the realization of the control algorithm and data acquisition was done by a IBM PC/AT computer. This computer was equipped with a A/D 12-bit converter card and a digital isolated outputs card (TTL standard). Due to the fact that 10 elder bits was used in the converter, its accuracy was assessed as 0.1%.

7.2 The programme of experiments on the bench test mechanism.

The programme of tests for stabilizer tanks on the bench test mechanism consists of:

a. <u>for the passive tank:</u> tests on the bench mechanism in the frequency range: from 0.81 rad/s to 3.4 rad/s which corresponds to the frequency range: from 0.2 rad/s to 0.85 rad/s in the full scale. The bench mechanism amplitudes were set at values $\phi_A = 1.5$; 3 and 6 degrees. At $\phi_A = 12^{\circ}$ the air channel of the tank was flooded; therefore this amplitude was neglected.

b. <u>for the passively-controlled tank:</u> tests on the bench mechanism in the frequency range: from 0.2 rad/s to 0.85 rad/s in the full scale. The bench mechanism amplitudes were set at values $\phi_A = 1.5$; 3 and 6 degrees. At $\phi_A = 12^{\circ}$ the dynamic loads of the bench mechanism were so large that they threatned the destruction of the mechanism; therefore this value was neglected.

The tests were carried out as follows:

1. tests on the separated passive tank, to find the optimal positions for the damping elements and to avoid noisy operation [34],
2. tests on the separated passively-controlled tank as a passive tank (when the blocking valves are open) to define its dynamic parameters in the passive state,
3. tests on the separated passively-controlled tank with its control and measuring systems being in operation to determine its characteristics (7.4).

The block diagramme of the measuring system used in testing the passive tank is shown in fig.7.1.

The use of the apparatus as in fig.7.1 in testing the characteristics of the passively-controlled tank with its operating valves was not possible. Moreover, the phase angle between the signals $\phi(t)$ and $M_A(t)$, is impossible to define; this is due to the fact that this idea does not exist for the pair of signals: sinusoidal (the bench mechanism motion) and trapezoidal (the passively-controlled tank moment). Therefore the system for testing the passively-controlled tank used the bench mechanism to force the harmonic motion (see fig.7.2).

7.3 Description of the test methods and result processing.

7.3.1 The handling of data for the bench tests of the passive tank.

For the bench tests of the passive tank and the passively-controlled tank in the passive state the method described in [65] was applied. This method is being applied in many countries (for ex. FRG, Italy, Japan and Poland).

In short, this method can be described as follows. If we assume that the tank's motion on the bench mechanism can be described by:

$$\phi = \phi_A \sin \omega_E t \tag{7.6}$$

then, taking into account the expression (7.6) and its derivatives, and placing the above in the expression (3.13b), we get the following differential equation:

$$\ddot{\vartheta} + 2\beta^*_\vartheta \, \omega_{\vartheta_0} \dot{\vartheta} + \omega^2_{\vartheta_0} \vartheta = (\frac{s}{g}\omega^2_E + 1)\omega^2_{\vartheta_0} \phi_A \sin\omega_E t \tag{7.7}$$

The solution of the equation (7.7) is:

$$\vartheta = \vartheta_A \sin\,(\omega_E t - \varepsilon_M), \tag{7.8a}$$

where:

$$\vartheta_A = \frac{(\frac{s}{g}\,\omega^2_E + 1)\omega^2_{\vartheta_0}\phi_A}{\sqrt{(\omega^2_\vartheta - \omega^2_E)^2 + 4\beta^{*2}_\vartheta\,\omega^2_{\vartheta_0}\,\omega^2_E}} \tag{7.8b}$$

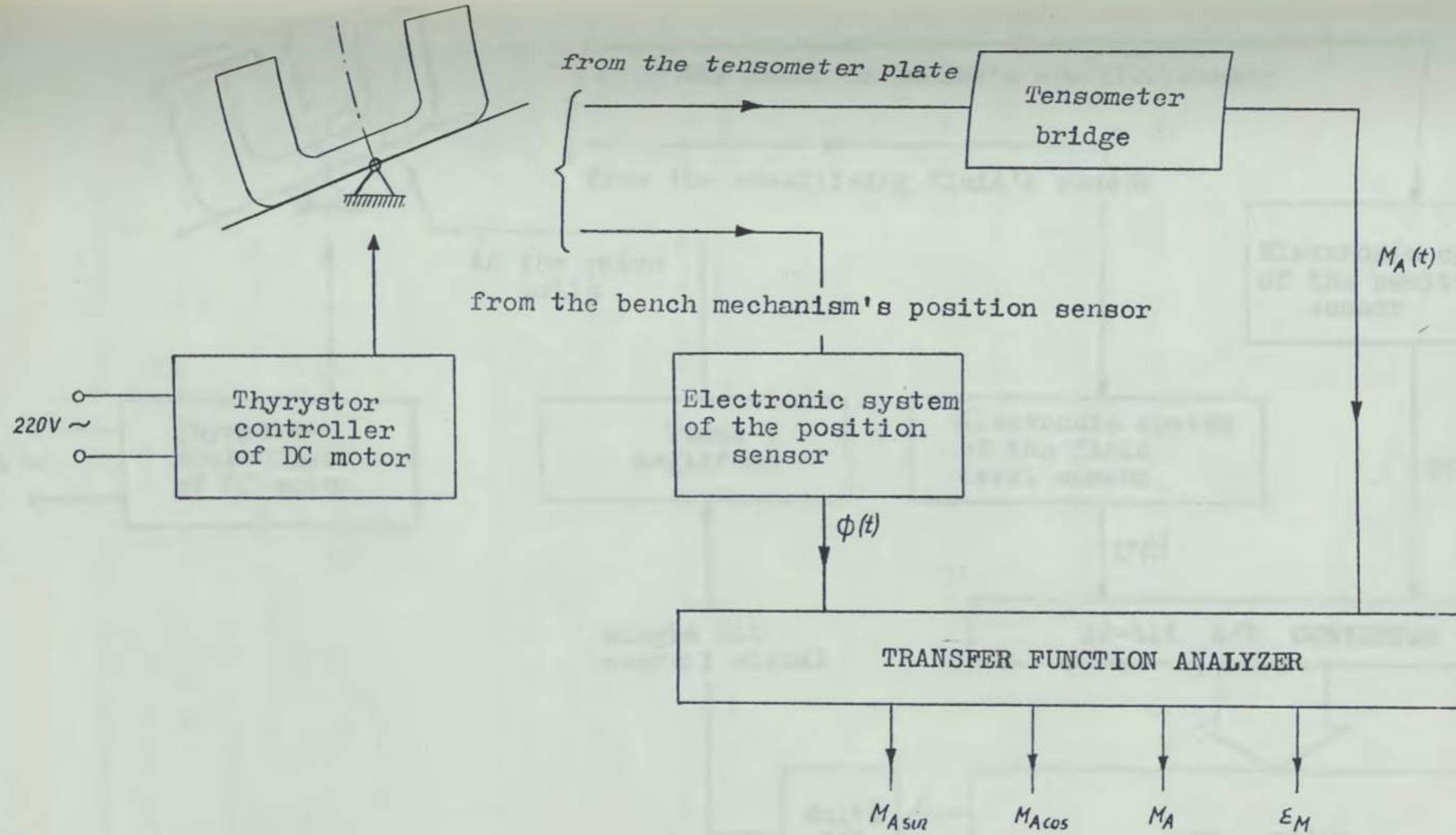

Fig.7.1. The block diagramme of the measuring system for bench tests of the passive tank.

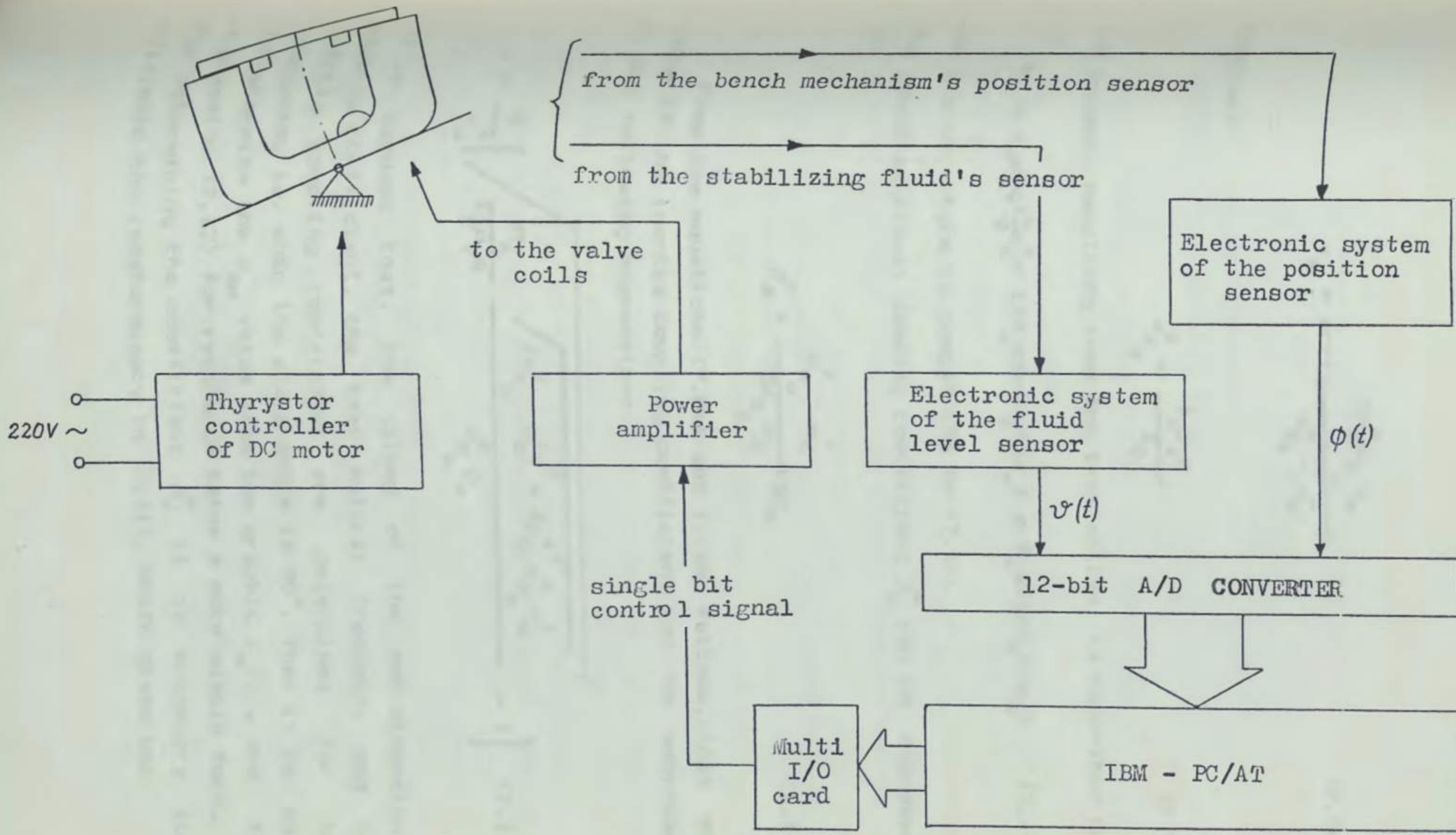

Fig.7.2. The block diagramme of the measuring system for bench tests of the passively-controlled tank.

$$\varepsilon_M = \mathrm{arctg}\frac{2\beta_\vartheta^* \omega_{\vartheta_o} \omega_E}{\omega_{\vartheta_o}^2 - \omega_E^2} \qquad (7.8c)$$

Because:

$$\omega_{\vartheta_o}^2 = \frac{I_T \rho_T g}{I_\vartheta} \qquad (7.9)$$

the moment, resulting from the tank motion, is described by:

$$M_H = I_\vartheta \rho_T g(\frac{s}{g}\omega_E^2 + 1)\vartheta_A \sin(\omega_E t - \varepsilon_M) = M_A \sin(\omega_E t - \varepsilon_M) \qquad (7.10)$$

and its structure is compatible to (7.3).

The non-dimensional damping coefficient β_ϑ^* can be expressed as:

$$\beta_\vartheta^* = \frac{\omega_{\vartheta_o}^2 - \omega_E^2}{2\omega_{\vartheta_o} \omega_E} \mathrm{tg}\varepsilon_M \qquad (7.11)$$

From the equations (7.8b) and (7.10) follows, that the value of the inertia coupling coefficient can be expressed by the following expression:

$$s = \frac{g}{\omega_E^2}\left[\sqrt{\frac{M_A}{I_T \rho_T g} \frac{\sqrt{(\omega_{\vartheta_o}^2 - \omega_E^2)^2 + 4\beta_\vartheta^{*2} \omega_{\vartheta_o}^2 \omega_E^2}}{\omega_{\vartheta_o}^2 \phi_A}} - 1\right] \qquad (7.12)$$

It is assumed that, the values of the non-dimensional damping coefficient, the tank natural frequency and the inertia coupling coefficient are determined for the resonance, ie. when the phase angle is 90°. Then it is easy to determine the $\omega_{\vartheta o}$ value from the graphic $\varepsilon_M(\omega)$, and the expression (7.12) for resonance takes a more simple form.

For determining the coefficient β_ϑ^* it is necessary to eliminate the indeterminacy in (7.11), which gives us:

$$\beta^*_\vartheta = \frac{1}{\omega_{\vartheta_o}\left(\frac{d\varepsilon_M}{d\omega}\right)_{\omega=\omega_{\vartheta_o}}} \tag{7.13}$$

Therefore we have relatively easily obtained all the parameters of the passive tank.

7.3.2 The handling of data for the bench tests of the passively-controlled tank.

For the bench tests of the passively-controlled tank, we have only the values: $\phi(t)$ and $\vartheta(t)$. From these two signals and from the results of the tests of the passively-controlled tank in the passive state, the characteristics of the moment of stabilizing fluid reaction were determined. The method used in obtaining these results try to connect the passive tank theory to passively-controlled tanks. Fig.7.3 shows an example of the run of the signals $\phi(t)$ and $\vartheta(t)$. This run is an idealistic one but in a simple way explains the accepted method of the result processing.

From the expression (7.10), we see that the fluid reaction moment M_H is proportional to the angular position of the stabilizing fluid $\vartheta(t)$. Moreover, all the expressions from the section 7.3.1 are based on the assumption that the fluid reaction moment is a harmonic curve with an amplitude of M_A and phase angle ε_M. This assumption is not true for the passively-controlled tank, whose reaction moment is similar to a trapezoidal curve.

If we determine the area under the curve of the angle ϑ, ie: the area S_{bcde} and compare it to the area under the sinusoidal curve then we get:

$$S_{bcde} = 2\vartheta^*_A, \tag{7.14}$$

where:

ϑ^*_A - amplitude of the substitute sinusoid, whose area under half the period is equal to the trapezoid bcde.

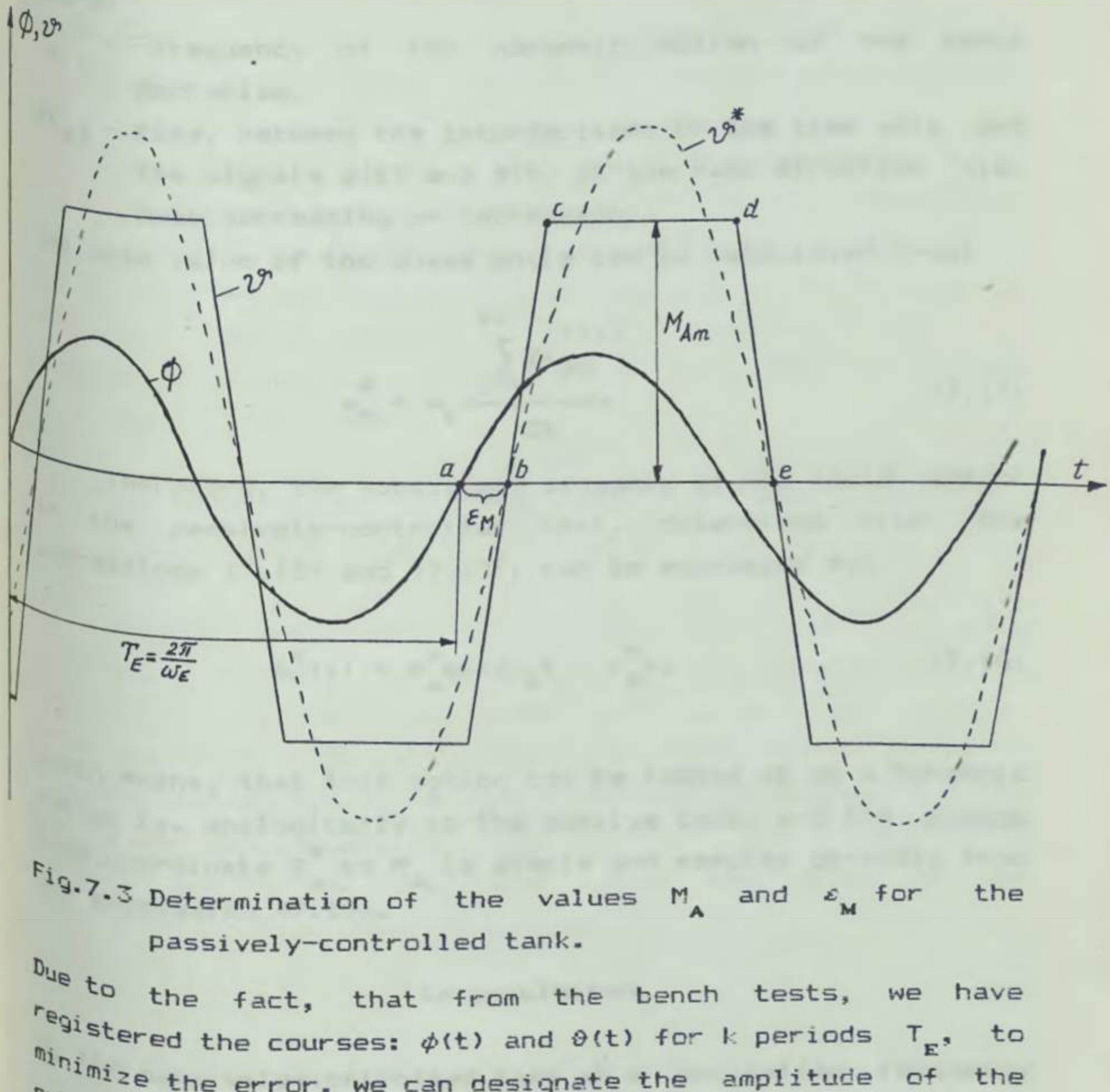

Fig.7.3 Determination of the values M_A and ε_M for the passively-controlled tank.

Due to the fact, that from the bench tests, we have registered the courses: $\phi(t)$ and $\vartheta(t)$ for k periods T_E, to minimize the error, we can designate the amplitude of the substitute sinusoid from the expression:

$$\vartheta^*_A = \frac{\int_0^{kT_E} |\vartheta(t)| dt}{4k} \tag{7.15}$$

The phase angle ε_M, has physical sense only as a time period between points a and b (fig.7.5). Therefore the phase angle can be expressed by:

$$\varepsilon_M = \omega_E \Delta t_{\phi\vartheta} \tag{7.16}$$

where:

ω_E - frequency of the harmonic motion of the bench mechanism,

$\Delta t_{\phi\vartheta}$ - time, between the intersections of the time axis and the signals $\phi(t)$ and $\vartheta(t)$ in the same direction (ie: both decreasing or increasing).

The mean value of the phase angle can be calculated from:

$$\varepsilon_M^* = \omega_E \frac{\sum_{i=1}^{2k} \Delta t_{\phi\vartheta}^{(i)}}{2k} \tag{7.17}$$

Therefore, the substitute sinusoid of the fluid motion in the passively-controlled tank, determined from the expressions (7.15) and (7.17), can be expressed by:

$$\vartheta^*(t) = \vartheta_A^* \sin(\omega_E t - \varepsilon_M^*), \tag{7.18}$$

which means, that this motion can be looked at as a harmonic motion ie. analogically to the passive tank, and the change from coordinate ϑ_A^* to M_A is simple and results directly from the expression (7.10).

In conclusion,

1. the passively-controlled tank at a excitation frequency ω_E = const is equal, in an energetic sense, to the passive tank with a natural frequency of ω_E;
2. the passively-controlled tank, in a wide range of excitation frequencies, can be substituted by an infinite set of passive tanks of natural frequencies, placed within the defined range of frequencies and with different saturation angles ϑ_N.

A special computer programme was written to designate the values in the expression (7.18). In connection with the above information, we see that the basic problem of the bench tests of the passively-controlled tank, is the formulation of the data processing method. This method, ensures the possibility of the transformation of the stabilizing fluid motion in the passively-controlled tank, into the corresponding fluid motion in the substitute passive tank. The influence of the substitute passive tank on the ship's roll motion is identical, to the influence of the fluid motion in the passively-controlled tank. The geometrical and dynamic parameters of the substitute passive tank are not important.

7.3.3 The determination of the ship's roll response from the bench tests.

The roll amplitude characteristic (7.1) for the ship equipped with the stabilizing tank, on the condition that the ship is rolling around a static axis can be obtained from the bench tests of the separated tank. The influence of the stabilizing fluid on the ship is expressed by the additional moment, which can be treated as an additional external excitation. Then the linear expression of the ship's roll motion takes the form:

$$(I_{xx} + m_{\phi\phi})\ddot{\phi} + 2N_{\phi}\dot{\phi} + \Delta GM\phi = \Delta GM\alpha_{E}\sin\omega_{E}t + M_{H} \qquad (7.19)$$

In the case of the forced roll of the stabilized ship the moment M_H is a harmonic function with an angular frequency equal to the excitation frequency ω_E. Assuming that the moment M_H is a sinusoidal curve with an amplitude M_A and phase angle ε_M, relative to the ship's roll, the solution of the expression (7.19) is the relationship:

$$\phi(t) = \phi_A\sin(\omega_E t - \varepsilon) \qquad (7.20)$$

where:

$$\phi_A = \frac{\omega^2_{\phi_0}\alpha_E}{\sqrt{(\omega_1^2 - \omega_E^2)^2 + 4\beta^{*2}_{\phi}\omega^2_{\phi_0}\omega_E^2}} \tag{7.21}$$

$$\varepsilon = \mathrm{arctg}\,\frac{2\beta^*_\phi\,\omega_E\,\omega_1}{\omega_1^2 - \omega_E^2} \tag{7.22}$$

and

$$\beta^*_\phi = \beta_\phi + \frac{\omega_{\phi_0} M_A \sin\varepsilon_M}{2\omega_E \Delta GM\phi_A} \tag{7.23}$$

$$\omega_1^2 = \omega^2_{\phi_0}\left(1 - \frac{M_A\cos\varepsilon_M}{\Delta GM\phi_A}\right) \tag{7.24}$$

From the expressions (7.23) and (7.24) we see that the tank has an influence on the damping of the ship, (the component of the tank reaction proportional to the angular velocity of the ship's roll) and on the stiffness of the ship (the component proportional to roll angle).

The method described above, presented in [61] was worked out for passive tanks. However, by means of the relationships described in 7.3.2, this method can be used in passively-controlled tanks. The solution (7.21) in this case is a non linear one, and the roll response characteristic should be determinated point by point, using the iterative method.

7.3.4 The analysis of measurement errors during the testing of passively-controlled tanks on the bench test mechanism.

The measurement process during tests on the bench mechanism consists of measuring the components of the tank's reaction. This process characterizes the two stages of tests of the separated tank model (measurement of the fluid reaction and determination of the influence of the tanks casing).

The main role in this case, is played by the systematic errors. All the main measurements on the bench mechanism were done in the following way. Using the set of instruments shown in fig.7.1, only results repeated five times were written in the measurement protocol. This is a situation when the measurement results are practically identical. For determining the systematic errors, we should take into account, that the measurements of the reaction components, are their indirect measurements. Generally, it means that the measurement result, depends on the measurement of many values of direct measurements:

$$z = f(x_1, x_2, \ldots) \qquad (7.25)$$

We must determine the measurement error of the value z, assuming that we know the measurement errors of x_i. In our case, we do not have any additional premises, therefore, we used the maximal errors method described in [56]. The maximal measurement error is expressed by:

$$\Delta z = \sum_{i=1}^{k} \left| \frac{\partial f(x_1, x_2, \ldots\ldots)}{\partial x_i} \right| \Delta x_i \qquad (7.26)$$

where:

Δx_i - error of the i-th direct measurement.

The expression (7.26) is the total differential of the expression (7.25). Due to the fact, that the errors are designated as maximal, the value in the expression (7.26) is absolute (module).

In determining the maximal measurement error of the fluid reaction moment, and its phase angle, it is necessary to use the total differential of the formula (7.5). We must first note that:

$$M'_A=\sqrt{M'^2_{A_{cos}}+M'^2_{A_{sin}}}\ ;\ \varepsilon'_M=\mathrm{arctg}\frac{M'_{A_{sin}}}{M'_{A_{cos}}}$$
$$M''_A=\sqrt{M''^2_{A_{cos}}+M''^2_{A_{sin}}}\ ;\ \varepsilon''_M=\mathrm{arctg}\frac{M''_{A_{sin}}}{M''_{A_{cos}}} \tag{7.27}$$

from which:

$$\Delta M'_A=\frac{1}{2\sqrt{M'^2_{A_{cos}}+M'^2_{A_{sin}}}}\left[\left|2M'_{A_{cos}}\right|\Delta M'_{A_{cos}}+\left|2M'_{A_{sin}}\right|\Delta M'_{A_{sin}}\right] \tag{7.28a}$$

$$\Delta M''_A=\frac{1}{2\sqrt{M''^2_{A_{cos}}+M''^2_{A_{sin}}}}\left[\left|2M''_{A_{cos}}\right|\Delta M''_{A_{cos}}+\left|2M''_{A_{sin}}\right|\Delta M''_{A_{sin}}\right] \tag{7.28b}$$

$$\Delta\varepsilon'_M=\frac{1}{M'^2_{A_{cos}}+M'^2_{A_{sin}}}\left[\left|M'_{A_{cos}}\right|\Delta M'_{A_{sin}}+\left|M'_{A_{sin}}\right|\Delta M'_{A_{cos}}\right] \tag{7.28c}$$

$$\Delta\varepsilon''_M=\frac{1}{M''^2_{A_{cos}}+M''^2_{A_{sin}}}\left[\left|M''_{A_{cos}}\right|\Delta M''_{A_{sin}}+\left|M''_{A_{sin}}\right|\Delta M''_{A_{cos}}\right] \tag{7.28d}$$

The terms $\Delta M'_{A_{cos}}$, $\Delta M''_{A_{cos}}$, $\Delta M'_{A_{sin}}$ and $\Delta M''_{A_{sin}}$, in the formulae (7.28) determine the measurement errors of the various components of the tank reaction moment, with the moving fluid (index '), and (index ") of the frozen state. They are not direct measurements, but the maximal error of the measurements can be easily presented as the sum of the relative errors. The measurement error, consists of the following errors:

- angle of the angular position of bench mechanism $\delta\phi = 1\%$,
- reaction moment on the bench mechanism axis $\delta M = 1\%$,
- phase sensitive apparatus $\delta\varepsilon = 0.5\%$.

The measurement error, of any given component of the moment is the sum of all measurement errors, and its value is 2.5%. The absolute error in the expressions (7.29), is calculated by multiplying the total relative error by the actual value being measured.
Substituting the expressions (7.28), to the total differential (7.5), we find the necessary values of the maximal measurement errors, in the following form:

$$\Delta M_A = \frac{1}{2\sqrt{\left(M_A'\cos\varepsilon_M' - M_A''\sin\varepsilon_M''\right)^2 + \left(M_A'\sin\varepsilon_M' - M_A''\sin\varepsilon_M''\right)^2}}$$

$$\cdot\left[2\left|M_A'\cos\varepsilon_M' - M_A''\cos\varepsilon_M''\right|\left(\left|\cos\varepsilon_M'\right|\Delta M_A' + M_A'\left|\sin\varepsilon_M'\right|\Delta\varepsilon_M' -\right.\right.$$

$$\left.- \left|\cos\varepsilon_M''\right|\Delta M_A'' - M_A''\left|\sin\varepsilon_M''\right|\Delta\varepsilon_M''\right) + 2\left|M_A'\sin\varepsilon_M' - M_A''\sin\varepsilon_M''\right|\cdot$$

$$\cdot\left(\left|\sin\varepsilon_M'\right|\Delta M_A' + M_A'\left|\cos\varepsilon_M'\right|\Delta\varepsilon_M' - \left|\sin\varepsilon_M''\right|\Delta M_A'' -\right.$$

$$\left.\left.- M_A''\left|\cos\varepsilon_M''\right|\Delta\varepsilon_M'\right)\right] \qquad (7.29)$$

and

$$\Delta\varepsilon_M = \frac{1}{A^2 + B^2}\left[\Delta M'_A\left(A\left|\sin\varepsilon'_M\right| - B\left|\cos\varepsilon'_M\right|\right) + M'_A\Delta\varepsilon'_M\left(A\left|\cos\varepsilon'_M\right| - B\left|\sin\varepsilon'_M\right|\right) + \Delta M''_A\left(B\left|\cos\varepsilon''_M\right| - A\left|\sin\varepsilon''_M\right|\right) + M''_A\Delta\varepsilon''_M\left(B\left|\sin\varepsilon''_M\right| - A\left|\cos\varepsilon''_M\right|\right)\right] \tag{7.30}$$

where:

$$A = \left|M'_A\cos\varepsilon'_M - M''_A\cos\varepsilon''_M\right|$$

$$B = \left|M'_A\sin\varepsilon'_M - M''_A\sin\varepsilon''_M\right|$$

Assuming, the equality of relative errors δM_A and $\delta\varepsilon_M$, we can calculate in full scale the error values ΔM_A and $\Delta\varepsilon_M$, derived from the formulae (7.29) and (7.30). Such a calculation is necessary only for the values of the tank moment M_A. In calculating the values of the errors, according to (7.29) and (7.30), we used a computer programme written in GW BASIC.

Another way of determining the measurement errors, was applied to the analysis of errors, in the bench tests of the passively-controlled tank. Due to the fact, that in this case, the bench mechanism angle $\phi(t)$ and the angular position of the fluid motion in the tank $\theta(t)$ were measured and their values was inputed into the computer, the only method available for use, is the statistical processing of the errors.

The evaluation of the statistical maximum errors was based on the following formulae [56]:

$$\left(\delta M_A\right)_{MAX} = \delta\phi + \delta\theta + \frac{3\sigma_\theta}{\overline{\theta}_A} \tag{7.31a}$$

$$\left(\delta\varepsilon_M\right)_{MAX} = \delta\phi + \delta\theta + \frac{3\sigma_{\varepsilon_M}}{\overline{\varepsilon}_M} \tag{7.31b}$$

In the expressions (7.31), $\delta\phi$ and $\delta\vartheta$ denote the relative errors corresponding to the bench mechanism angle ϕ and angular position of the stabilizing fluid ϑ , while the other terms, represent the standard deviations calculated relative to their mean values. The fluid reaction moment in the tank was determined from the expression (3.17). Wherefore, in (7.31a) the appearance of σ_{ϑ} as a component of the measurement error of the moment. In calculating the measurement error values for the various bench mechanism frequencies, the computer programme was written in language Fortran 77.

7.4 Tests of the passively-controlled tank on irregular waves.

The tests of the ship roll tank stabilizers on irregular waves are carried out very rarely. According to authors [36,61,65], the bench tests are sufficient and give complete information on the operation of such a stabilizer installed onboard ship. This is true, however, mainly for the passive tank. In our case, the passively-controlled tank is an object, which has been insufficiently tested, therefore it was decided to test the tank, in the model basin, using a physical model of the ship's hull.

Due to high costs, tests in the model basin were limited to:

1. tests of the behaviour of the ship,equipped with the passively-controlled tank, for two different realizations of the irregular wave,
2. tests of the unstabilized ship, on the above mentioned wave realizations.

The methodics of these tests, as well as the presentation of their results, are simpler than in the case of the bench tests.
It is not necessary to convert the obtained results, in such a complicated way (as in the case of the bench tests) to

obtain finished results, which can be used in discussion.

The ship's model, equipped with a model of the passively-controlled tank (the models being constructed in 28 scale) were placed in the model basin, in such a way that the ship's plane of symmetry is perpendicular to the basin's main axis (fig.7.4).

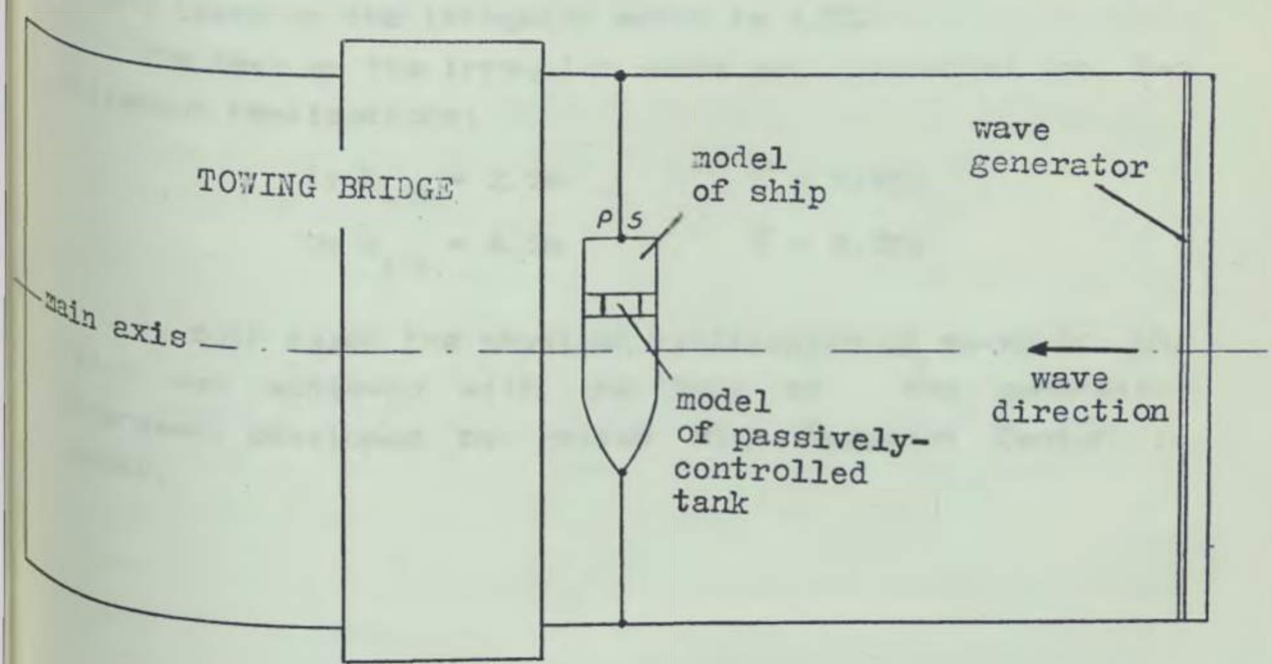

Fig. 7.4 The placing of the ship's model in the model basin for testing on the irregular wave.

The irregular waves were generated by Hottinger's hydraulic wave generator. The model was equipped with a gyroscope which meassured the ship's roll angle. This gyroscope, an CGW4 (made in USSR) is an aviation model. It measures the roll angle in the range ±15° with an accuracy of 2%. Also the tank model was equipped with a capacity sensor, identical to the one in the model, tested on the bench mechanism. Due to the small scale of the model, the accuracy, in the measurement of the fluid angular position is 2.5%.

Due to the character of the excitation, the only possible method to determine the error is the method of maximum statistical errors (therefore expressions identical to (7.31)).

The value of the standard deviation is difficult to determine. Therefore, we can assume that the maximum error, during tests on the irregular waves is 4.5%.

The test on the irregular waves were conducted for two different realizations:

1: $h_{1/3} = 2.5m$ $\quad\quad \bar{T} = 7.45s$

2: $h_{1/3} = 4.5m$ $\quad\quad \bar{T} = 9.35s$

In both cases the physical realization of waves in the basin was achieved with the help of the generating programme, developed by Polish Ship Research Center in Gdańsk.

Chapter 8

8. RESULTS OF EXPERIMENTS.

The type of problem, which needs to be solved, has a decided influence on the choice of the experiments conducted. The analysis of the ship - passively-controlled tank system's behaviour experimentally has as its aim:

1. investigation of thesuitability of the proposed mathematical model (equations (3.13) or (3.14), for describing the system's behaviour;
2. prooving of the hypothesis connected with the control of the passively-controlled tank in conditions close to the reality.

These experiments were divided into several groups. Each of these groups solved a specific component problem. These groups are as follows:

a. bench tests conducted for the passively-controlled tank in the passive state, to determine the tank's parameters in this state of action,
b. simulation experiments on the computer, of the ship - passively-controlled tank system's operation, for parameters obtained from the above tests,
c. experiments on the bench test mechanism, conducted on the passively-controlled tank, with the blocking valves in operation; the aim of these tests, is to compare their results with results obtained from the simulation tests (point b),
d. testing the operation of the passively-controlled tank on irregular waves and comparison of the model test results with the results of the computer simulations.

In testing the passively-controlled tank on the bench test mechanism, the results of these experiments were shown, using the tests conducted on the passive tank, as a background. [34].

The scale models used in these experiments are shown in photographs 8.1 and 8.2. They were built to 1:16.3 and 1:28 scale.

The test results presented in this chapter constitute the final stage of the analysis of the passively controlled tank.

Although the sequence of tests as shown above should force us to present their results in the same order, the author delibrately presents them differantly in the hope that this would influence the clarity of the conclusions.

As the test object, the author chose a fishing vessel. Her particulars are presented in table 8.1.

Based on the ship's body lines, the shape of the passive and passively-controlled tanks were designed as shown in fig. 8.1 and 8.2. The tanks were designed in such a way that their connecting ducts run above the ship's double bottom.

These tanks and their models were equipped with tuning - damping devices in the form of a revolving grid of blades, placed in the water connection duct. In this way the duct was divided into four independant channels, which can be separately closed by revolving the blades ,causing the desired damping of the stabiling fluid's motion. The chosen parameters of the passive tank were based on the principles presented in [55,56]. The parameters of the passively-controlled tank were based on the above principles, as well as on the author's propositions.

Table 8.2 shows the basic parameters of the two tanks.

Photo.8.1. Model of passively-controlled tank
in 16.3 scale for the bench tests.

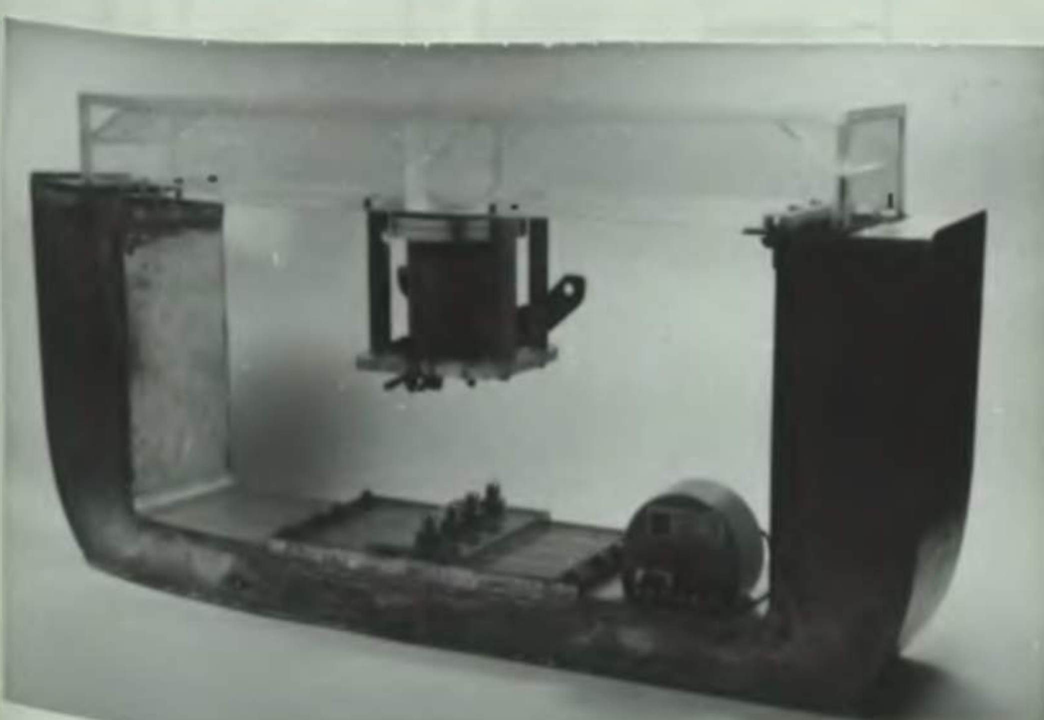

Photo.8.2. Model of passively-controlled tank
in 28 scale for basin tests.

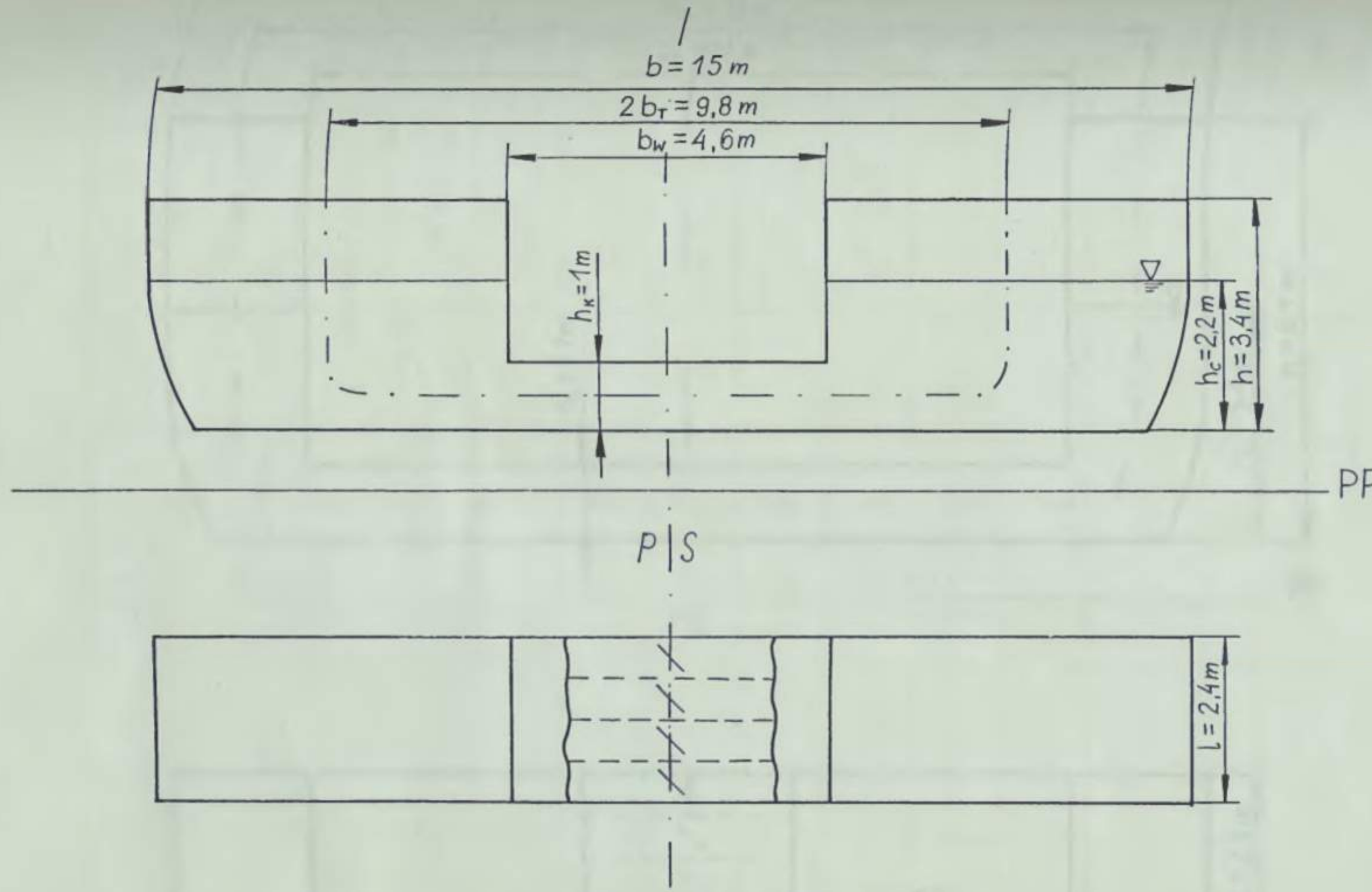

Fig.8.1. The basic dimensions of passive tank designed for tested ship (in full scale).

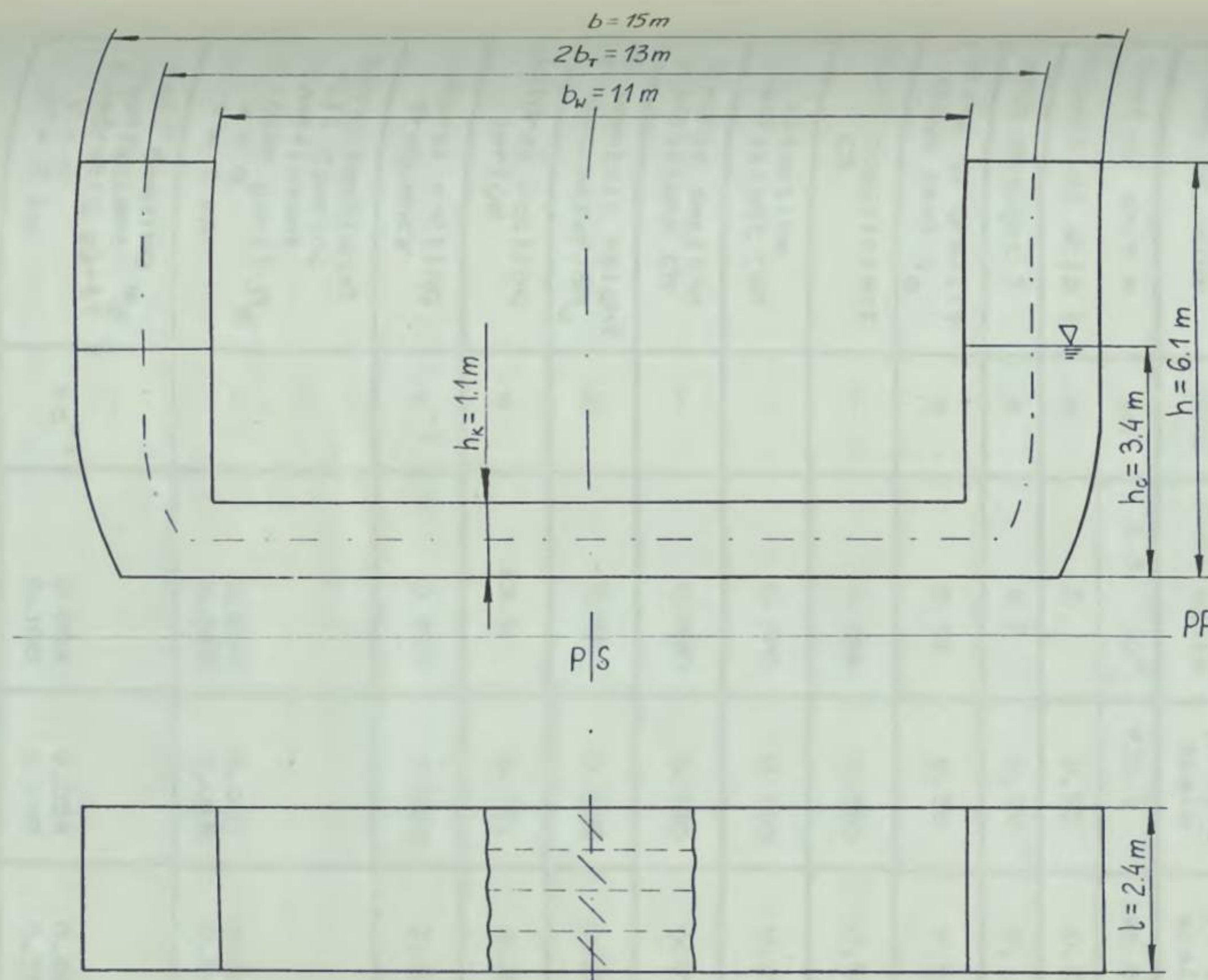

Fig.8.2. The basic dimensions of passive-controlled tank designed for tested ship (in full scale).

Table 8.1 Ship particulars without roll stabilization tanks

Name of value	Unit	Full scale	In 16.3 scale	In 28 scale
mass of ship m	kg	$3.2 \cdot 10^6$	738.9	145.8
breadth of ship B	m	15	0.92	0.536
mean draught T	m	4.7	0.29	0.168
centre of gravity above keel z_G	m	5.94	0.36	0.212
block coefficient CB	–	0.586	0.586	0.586
waterline coefficient CWP	–	0.770	0.770	0.770
midship section coefficient CM	–	0.920	0.920	0.920
metacentric height (frozen water) GM_0	m	0.84	0.052	0.03
natural rolling period	s	12.6	3.121	2.381
natural rolling frequency	s^{-1}	0.499	2.013	2.639
non-dimensional roll damping coefficient (linear part) β_ϕ V = 0 V = 7 kn	–	0.007 0.018	0.007 0.018	0.007 0.018
roll damping coefficient w_ϕ (quadratic part) V = 0 V = 7 kn	rd^{-1}	0.054 0.108	0.054 0.108	0.054 0.108

Table 8.2 The settings of the basic parameters of the stabilizing tanks (calculated and assumed values).

Γ	ϕ_{st}	δGM	$\omega_{\vartheta 0}$
-	o	m	
PASSIVE TANK			
0.23	3.44	0.21	0.601
PASSIVELY-CONTROLLED TANK			
0.155	3.52	0.13	0.85

The data included in table 8.2 needs comment, because the wave slope capacity of the passively-controlled tank is practically equal to the wave slope capacity of the passive tank (a difference of 2%). This is true, although the coefficient Γ of the passively-controlled tank has a lower value. This is connected with the different shape of the two tanks and their different methods of operation.

From fig. 8.1 and 8.2 we note, that the saturation angles for the passive and passively-controlled tanks are respectively 12° and 21°. Therefore, the gravitational moment generated through the passive tank, is due to the displacement of a mass of great cross-section A_v through small angles ϑ. In the passively-controlled tank the angles of displacement are greater, therefore, so as to obtain a similar moment, as in the passive tank, it is necessary to design the tank's wing section with a much lesser cross-sectional area A_v.

This principle was taken into account when constructing both tanks. From fig. 8.1 and 8.2 we can note, that the wing sections of the passively-controlled tank are much higher than their counterparts in the passive tank. They also have a much lesser cross-section, which causes the tank to have lesser influence on the ship's initial metacentric height.

8.1 Tests of the passively-controlled tank during regular excitation.

All the tests of the passively-controlled tank on the regular wave were conducted with the help of the FDD^2 regulator, equipped with the Butterworth's filter. Comparative testing of the other controllers (described in chapter 5) were conducted on irregular waves. The presentation of the operation of the other regulators on the regular wave, would complicate the results and make them unreadable.

The presentation of the regular wave test results starts by indicating the coefficient of the ship's roll damping (obtained from testing the free roll of the hull's model), as well as the damping coefficients of the stabilizing fluid in both tanks. Fig. 8.3 presents the course of the ship's roll damping coefficient when the ship is stationary, fig. 8.4 - the course of the stabilizing fluid's damping coefficient in the passive tank and fig. 8.5 - in the passively-controlled tank. The values of these coefficients are shown in table 8.3.

Table 8.3 The damping coefficients of both stabilizing tanks

	β_ϑ	w_ϑ	ω_ϑ
	-	1/rd	1/o
Passive tank	$3.63 \cdot 10^{-3}$	12.60	0.22
Passively-controlled tank	$1.08 \cdot 10^{-1}$	0.215	$3.75 \cdot 10^{-3}$

The damping coefficient of the ship was shown earlier in table 8.1. Therefore tables 8.1, 8.2 and 8.3 fully present the parameters of both tank models used in the tests. It should be noted, that in choosing the damping coefficients the passive and the passively-controlled tank were treated differently. The basic component of the passive tank's damping is the quadratic component, where as the

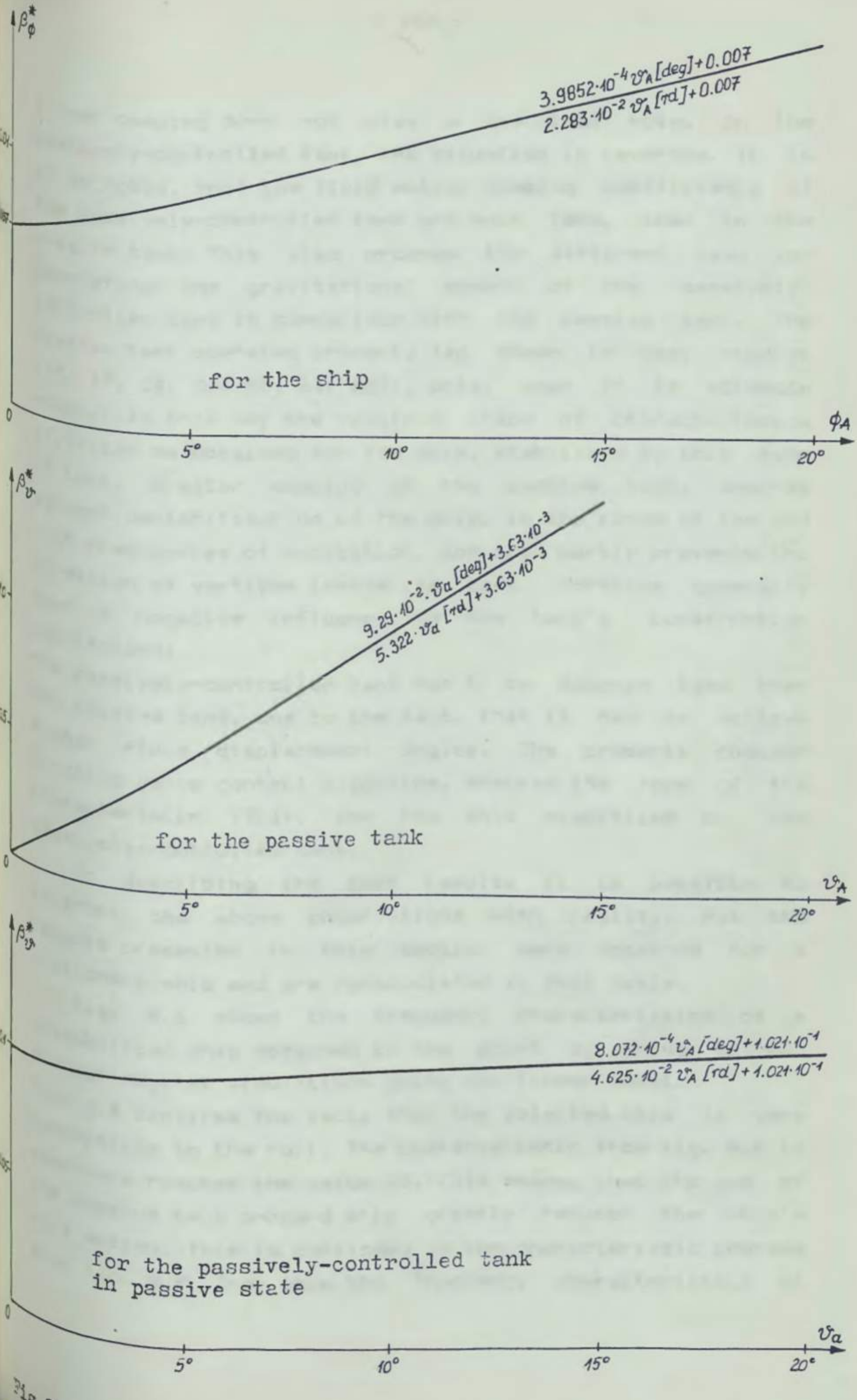

Fig.8.3,8.4,8.5. Non-dimensional damping coefficients.

linear damping does not play a practical role. In the passively-controlled tank, the situation is reversed. It is to be noted, that the fluid motion damping coefficients of the passively-controlled tank are much less, than in the passive tank. This also prooves the different base for generating the gravitational moment of the passively-controlled tank in comparison with the passive tank. The passive tank operates properly (as shown in many studies [13, 17, 24, 30, 55, 61, 65]), only, when it is strongly damped. In this way the required shape of characteristics (7.1),can be obtained for the ship, stabilized by this type of tank. Greater damping of the passive tank, ensures against destabilization of the ship, in the range of low and high frequencies of excitation, and also partly prevents the formation of vortices inside the tank. Vortices generally have a negative influence on the tank's construction (cavitation).

The passively-controlled tank has to be dampned less than the passive tank, due to the fact, that it has to achieve higher fluid displacement angles. The properly choosen blocking valve control algorithm, ensures the form of the characteristic (7.1), for the ship stabilized by the passively-controlled tank.

In describing the test results it is possible to confront the above observations with reality. All the results presented in this section were obtained for a stationary ship and are recalculated in full scale.

Fig. 8.6 shows the frequency characteristics of a unstabilized ship obtained by the point by point method, through digital simulations using non linear model.

Fig. 8.6 confirms the fact, that the selected ship is very susceptible to the roll. The characteristic from fig. 8.6 in resonance reaches the value 20. This means, that the use of the passive tank onboard ship greatly reduces the ship's roll motion. This is confirmed in the characteristic courses from fig. 8.7. They show the frequency characteristics of

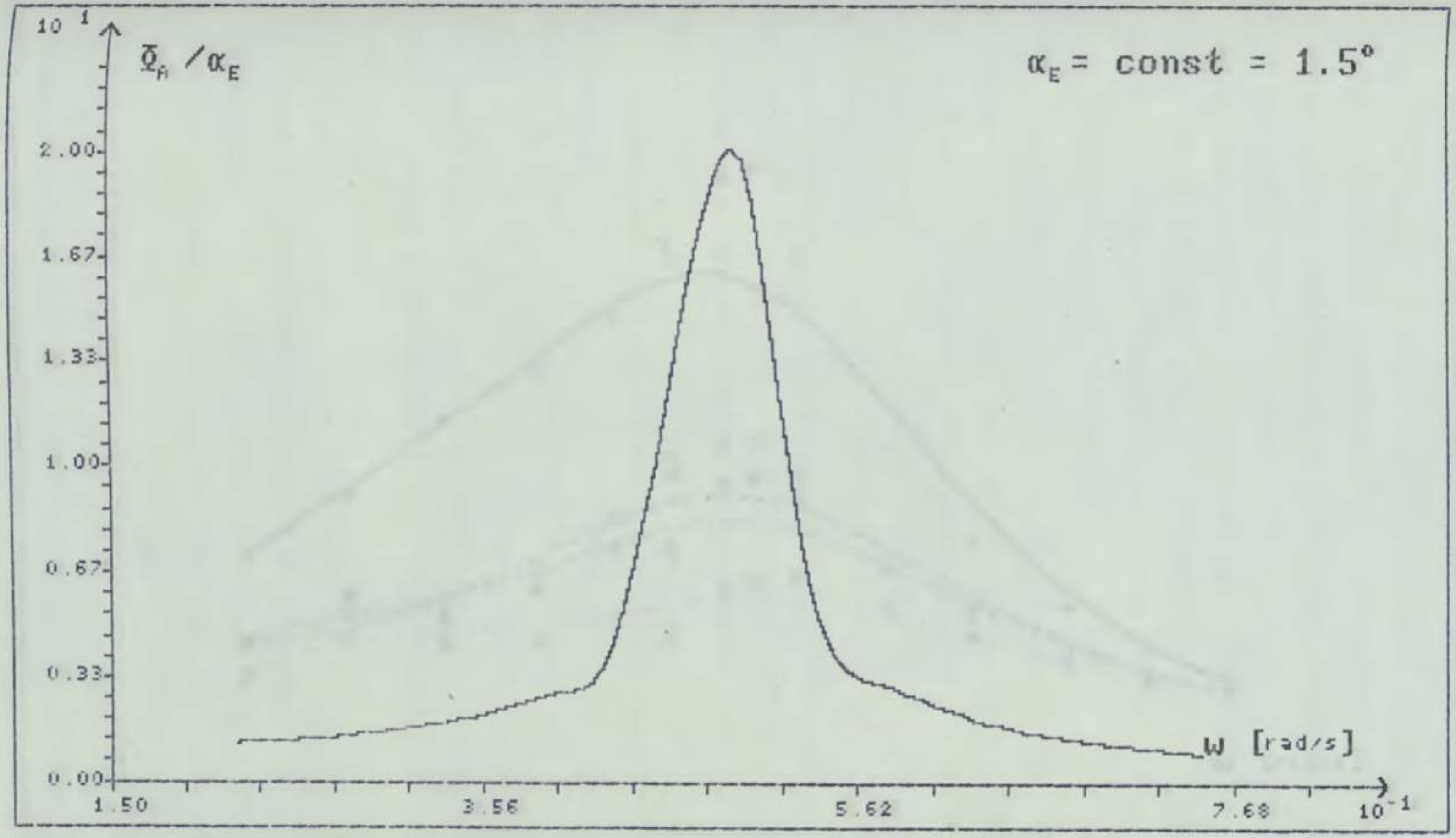

Fig.8.6. The roll response characteristic for unstabilized ship, calculated from numerical simulation experiments.

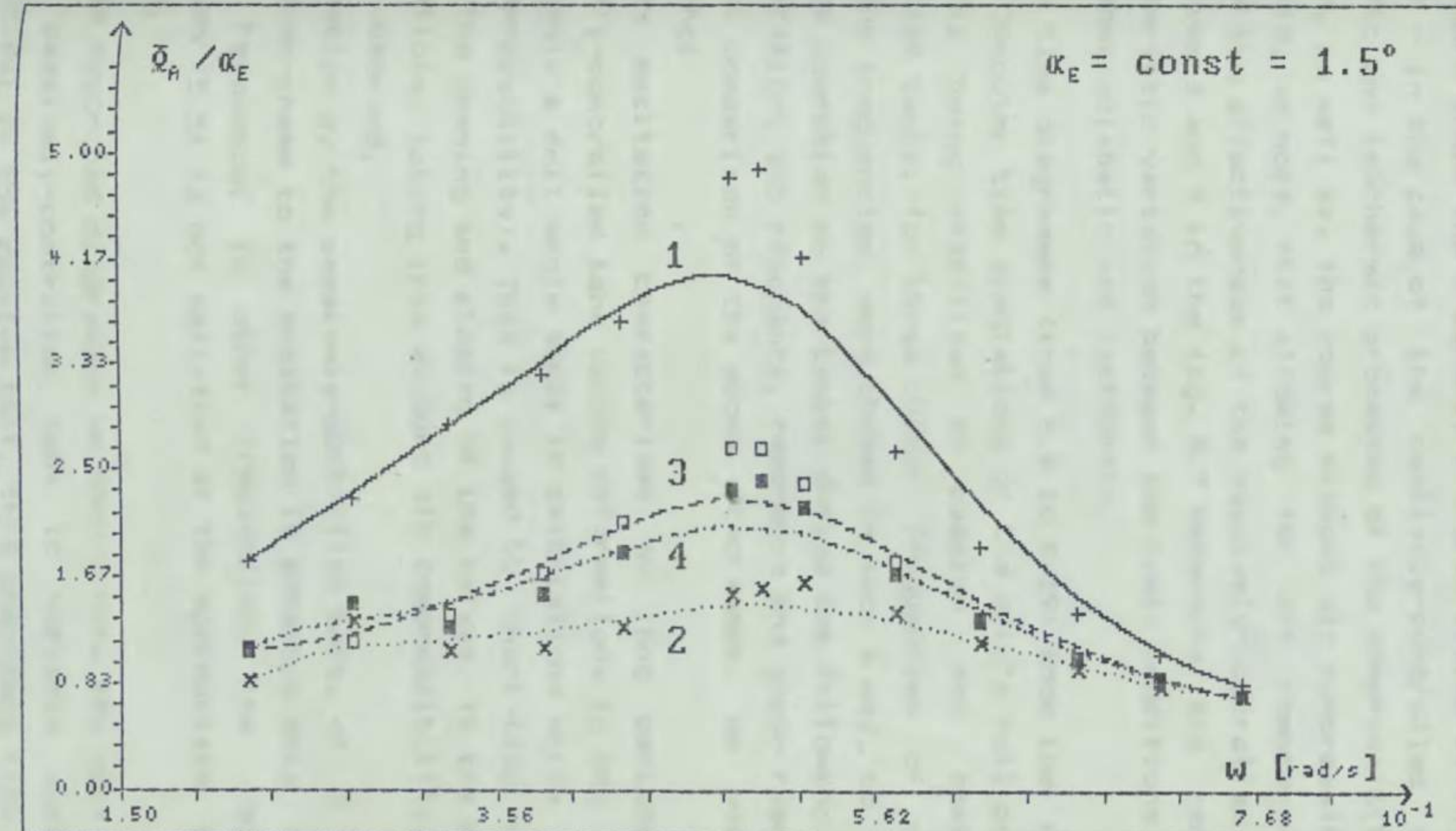

Fig.8.7. Results of numerical experiments with the ship's roll motion; the roll response curves for ship stabilized by:

1 - the passive tank,
2 - the passively-controlled tank (without compressibility of the air),
3 - the passively-controlled tank (compressibility of the air - isothermic process),
4 - the passively-controlled tank (compressibility of the air - adiabatic process).

the ship stabilized by the passive and passively-controlled tank.

This figure shows the frequency characteristics taking into account - in the case of the passively-controlled tank - adiabatic and isothermic processes of the compressibility of the air, as well as, the course without air compressibility. From this, we note, that allowing for air compressibility reduces the effectiveness of the passively-controlled tank. The curves 3 and 4 in the fig. 8.7 determine the range of characteristic variation between the limit conditions of the processes: adiabatic and isothermic.

The time diagrammes (from 8.8 to 8.19) show the results of the computer time simulations of the ship's roll process. The ship being stabilized by passive and passively-controlled tanks, for three chosen frequencies of regular wave. The frequencies were chosen in such a way, so as to show the operation of the tanks, during the following forms of excitation: sub resonance, resonance and super resonance. From the comparison of the above diagrammes, we note the following:

- during excitation characterised by long periods, the passively-controlled tank causes deformations in the course of the ship's roll angle (only in calculations which ignore air compressibility). This is caused by short-time changes during the opening and closing of the valves. In the case of calculations, taking into account air compressibility, this is not observed,
- generation by the passively-controlled tank, of a moment in counter-phase to the excitation is possible only in the ship's resonance; in other frequencies, the necessary condition (5.5) is not satisfied at the appropriate instant of time,
- in the described diagrammes we should note the self tuning of the passively-controlled tank to variable excitation frequencies; in the passive tank, this phenomena takes place

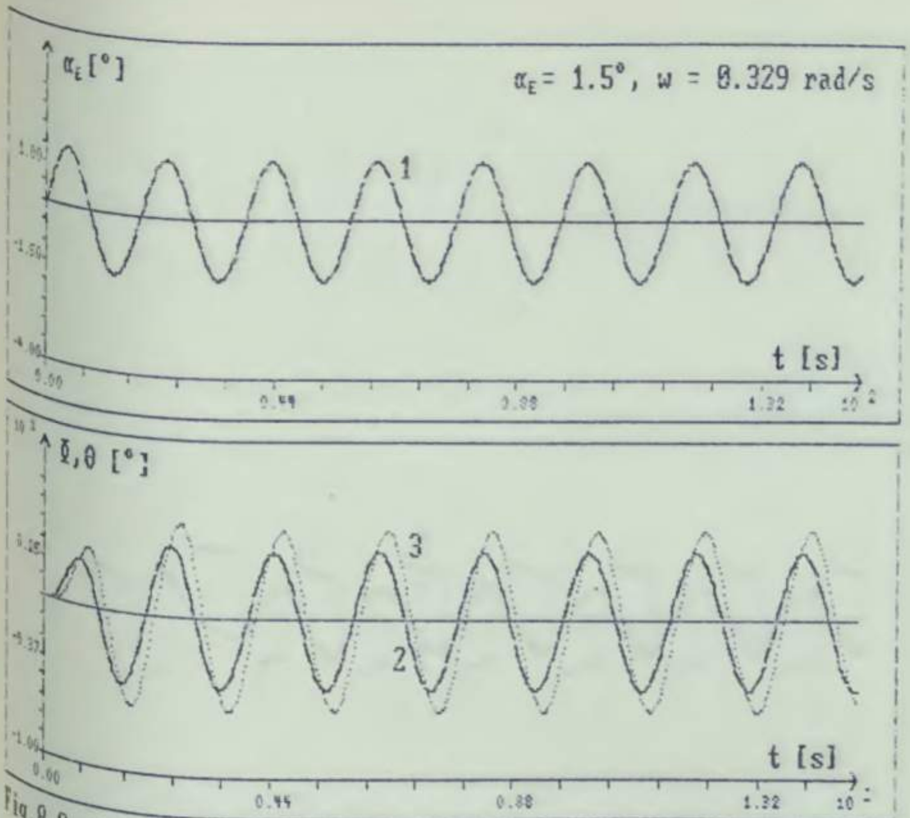

Fig.8.8. Time diagramme of the ship's roll simulation for the ship stabilized by passive tank (numerical simulation).

1 - the wave slope angle,
2 - the ship roll angle,
3 - the angular position of the fluid in the tank.

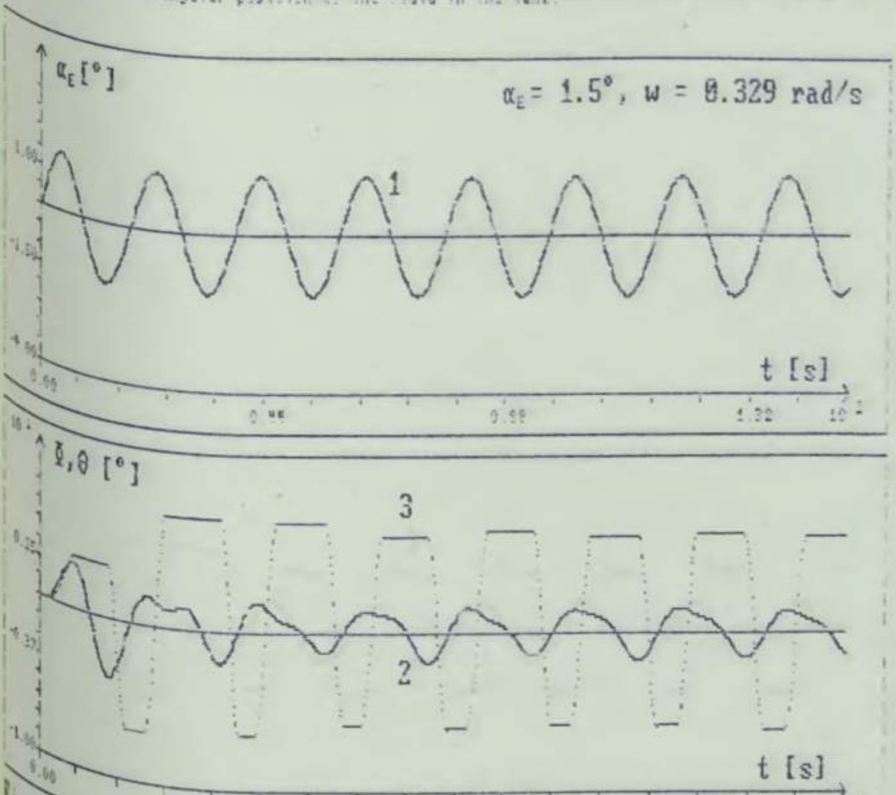

Fig.8.9. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (without compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

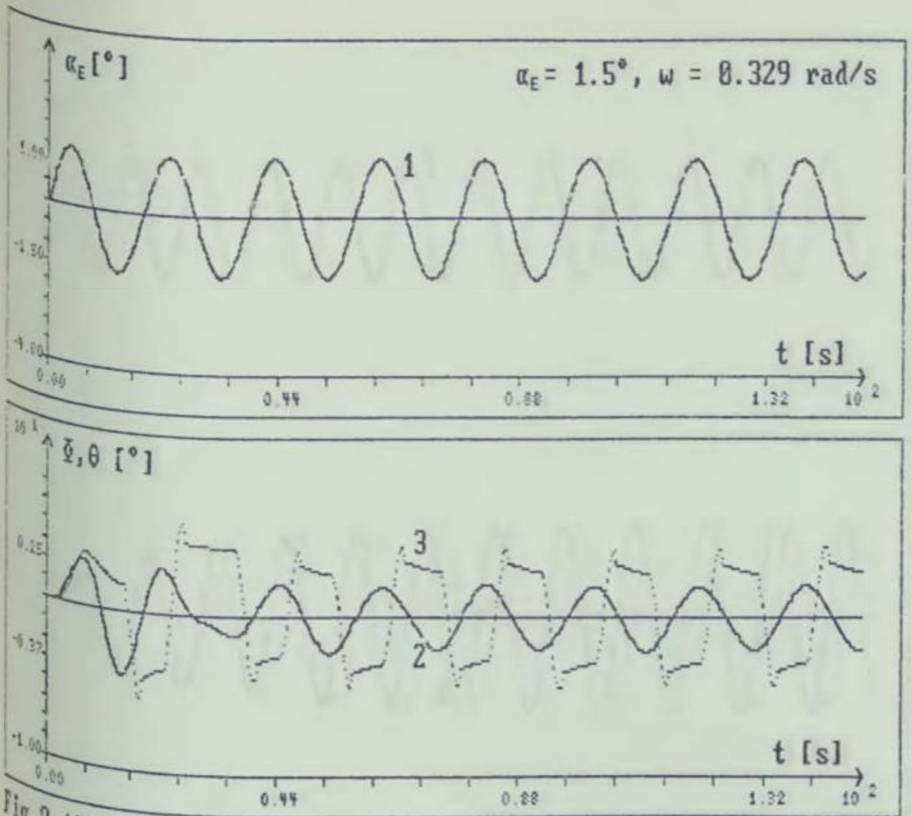

Fig.8.10. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (isothermic process for compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

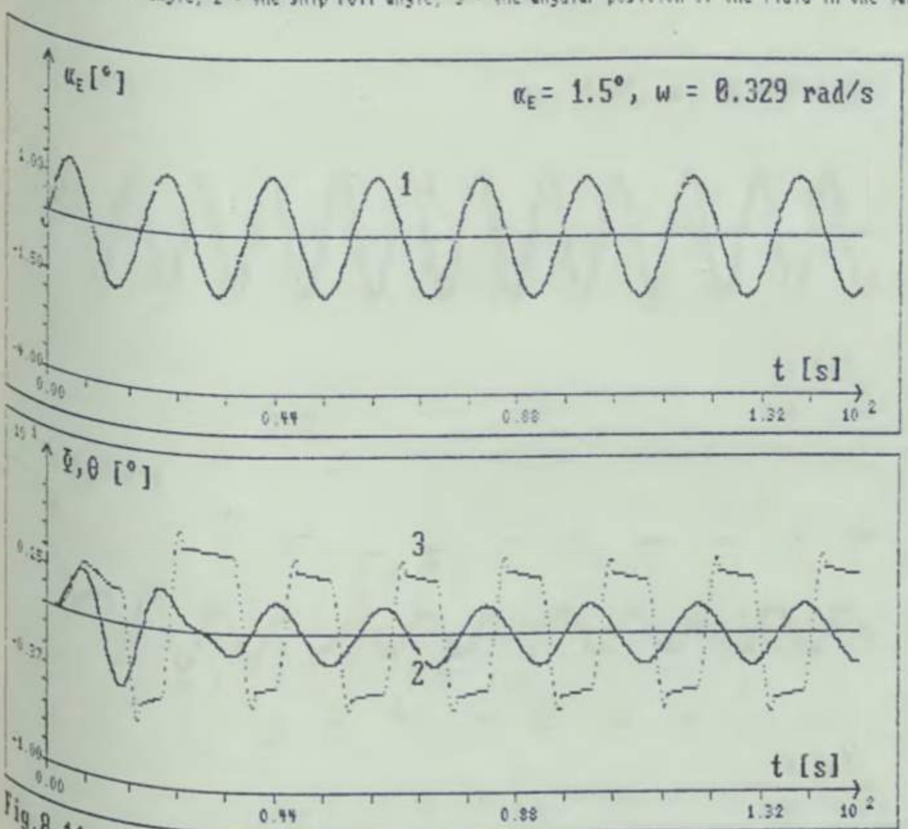

Fig.8.11. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (adiabatic process for compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

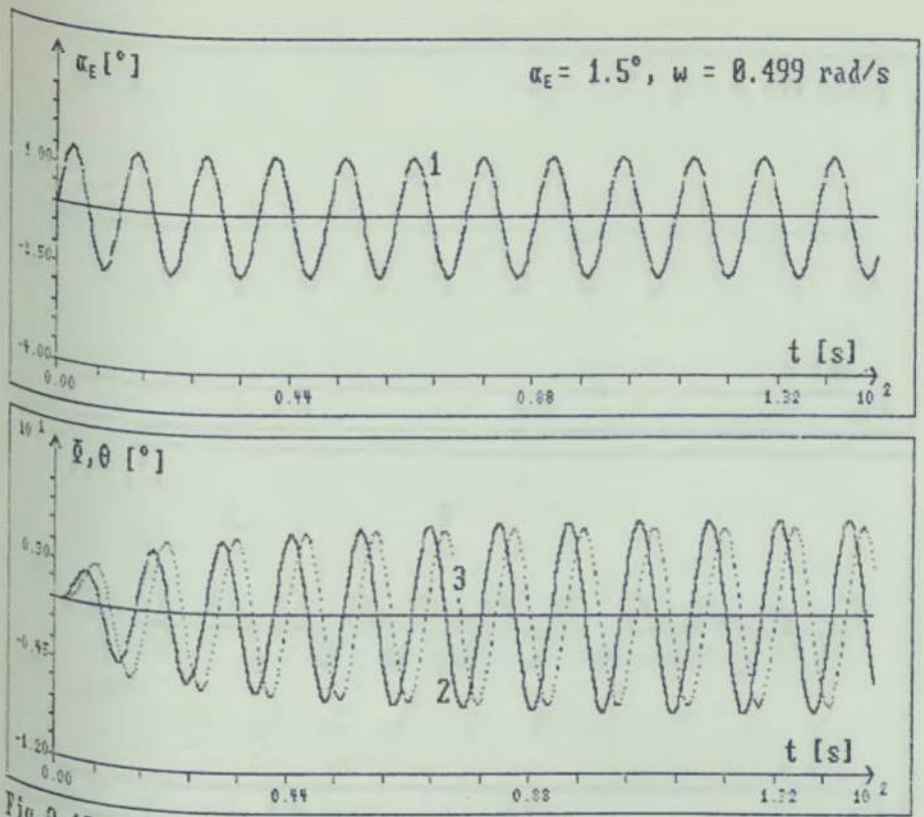

Fig.8.12. Time diagramme of the ship's roll simulation for the ship stabilized by passive tank (numerical simulation).

1 - the wave slope angle,
2 - the ship roll angle,
3 - the angular position of the fluid in the tank.

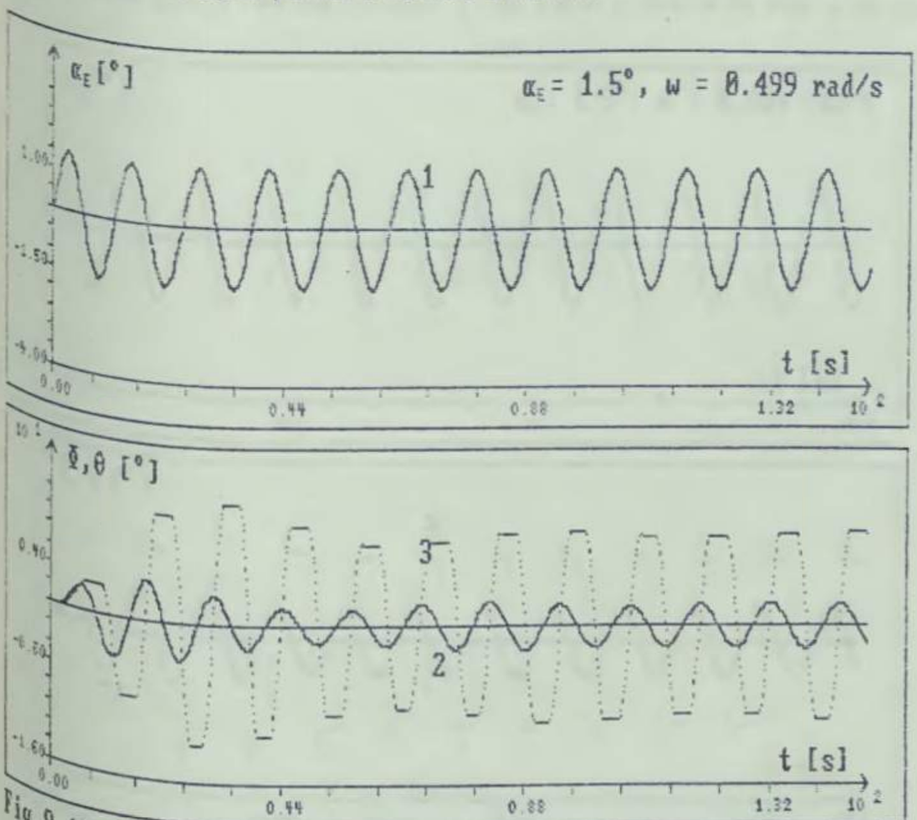

Fig.8.13. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (without compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

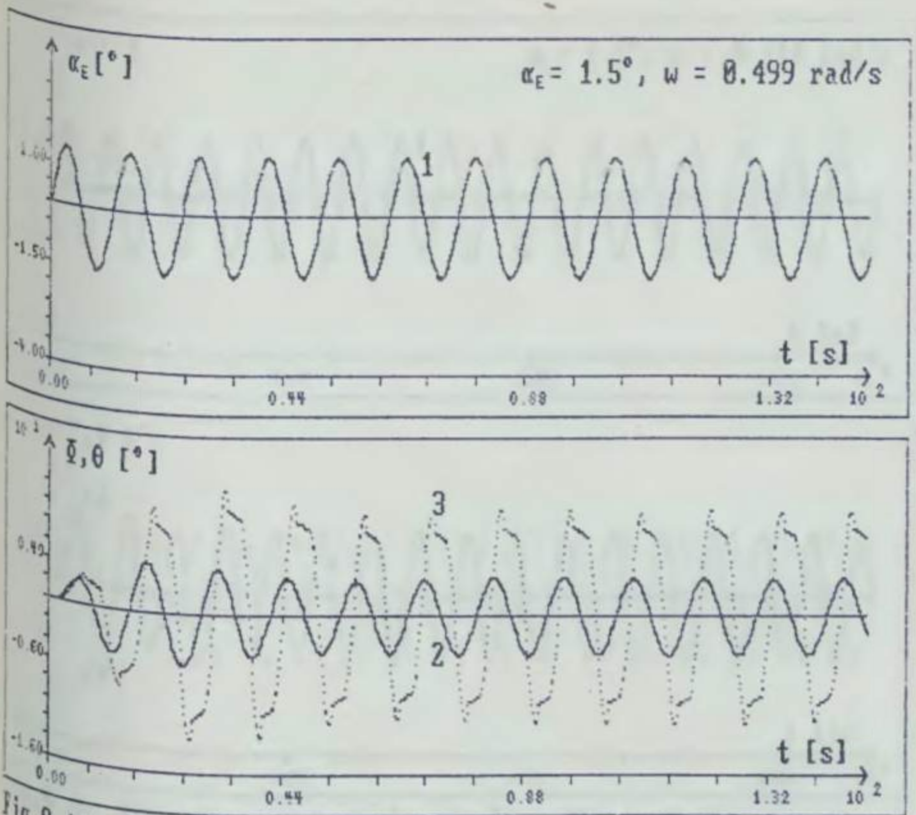

Fig.8.14. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (isothermic process for compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

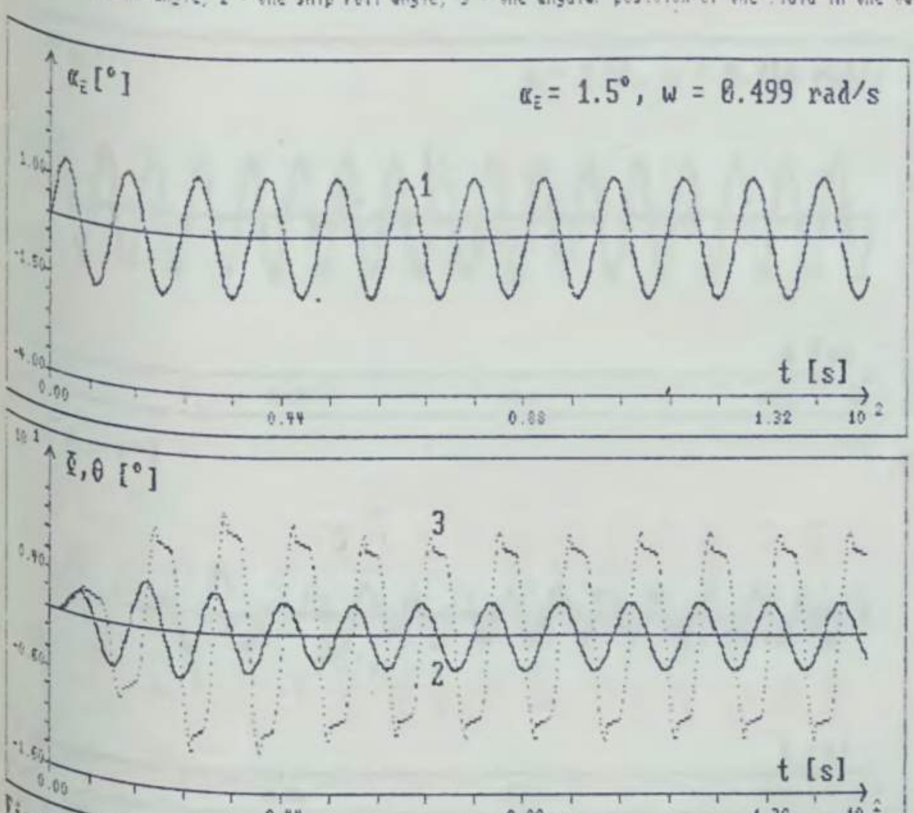

Fig.8.15. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (adiabatic process for compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

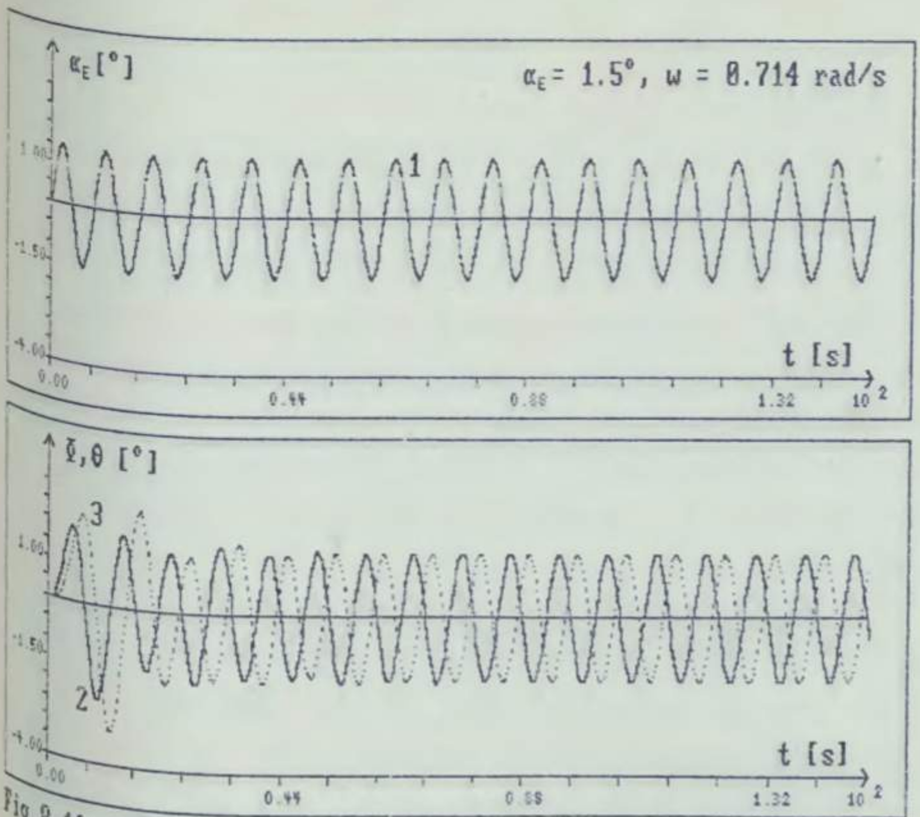

Fig.8.16. Time diagramme of the ship's roll simulation for the ship stabilized by passive tank (numerical simulation).

1 - the wave slope angle,
2 - the ship roll angle,
3 - the angular position of the fluid in the tank.

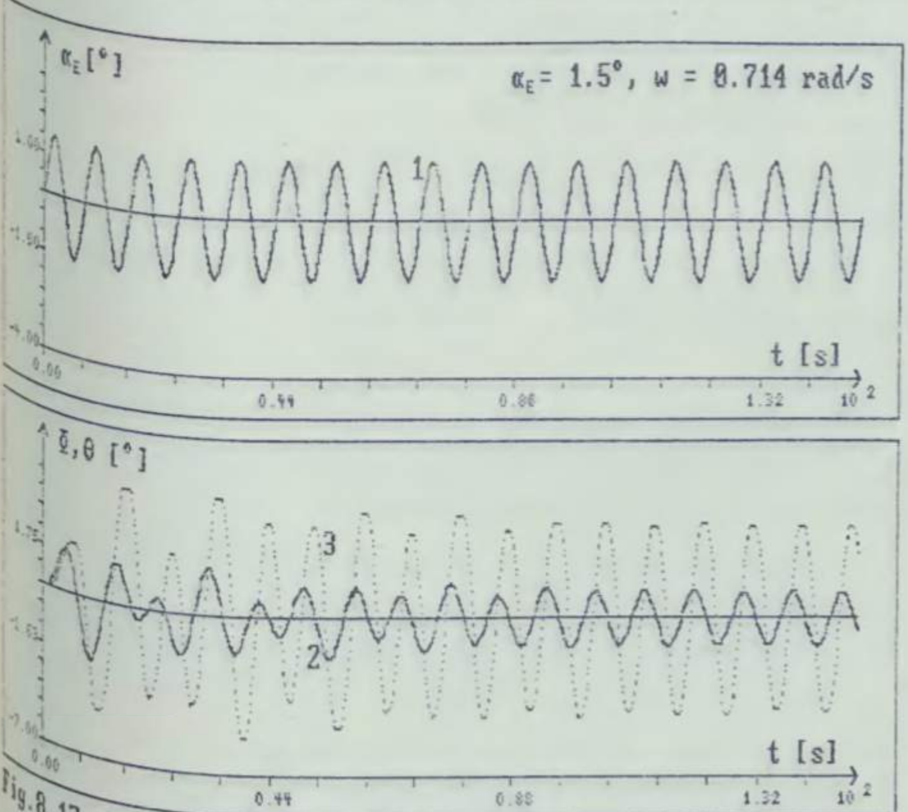

Fig.8.17. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (without compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

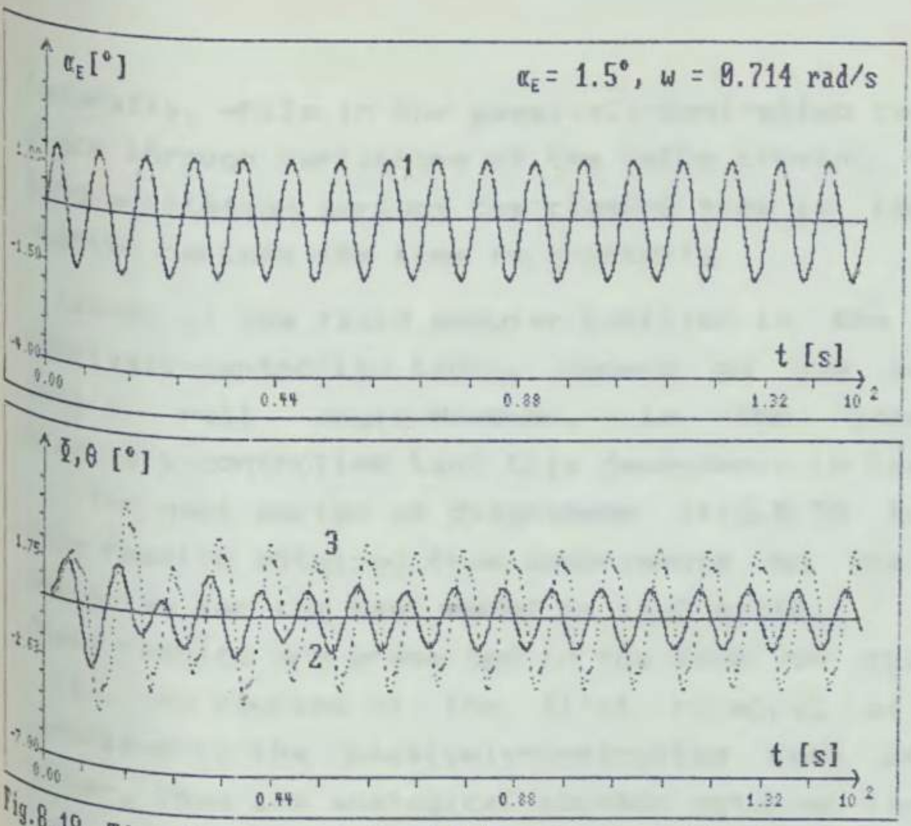

Fig.8.18. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (isothermic process for compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

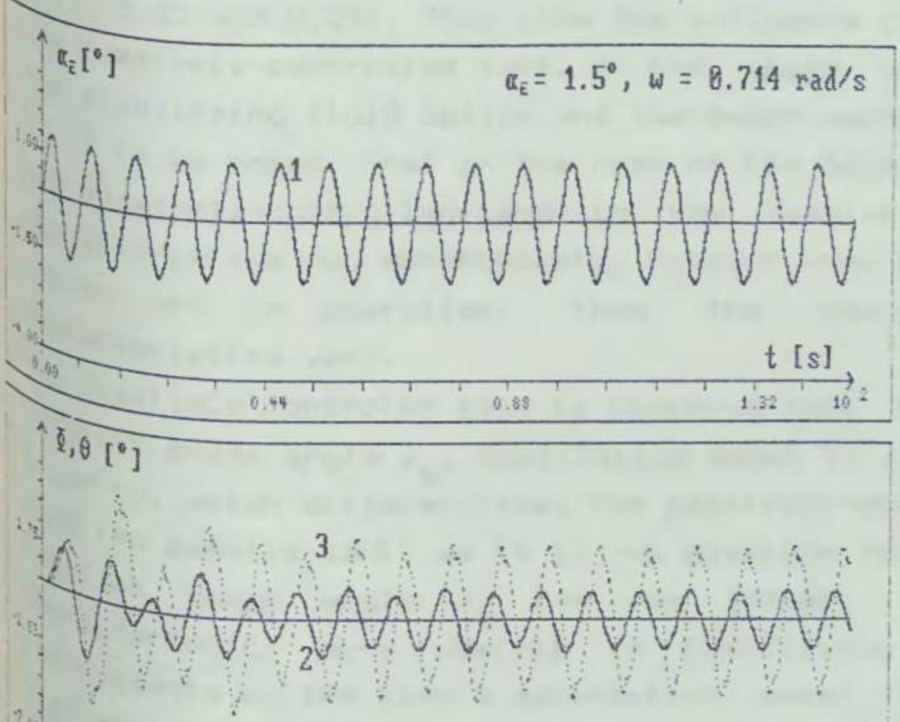

Fig.8.19. Time diagramme of the ship's roll simulation for the ship stabilized by passively-controlled tank (adiabatic process for compressibility of the air).

1 - the wave slope angle, 2 - the ship roll angle, 3 - the angular position of the fluid in the tank.

naturally, while in the passively-controlled tank this takes place through variations of the valve closing time (during long excitation periods the closing time is longer, during shorter periods the time is shorter),

- values of the fluid angular position in the passive and passively-controlled tanks, depend on the value of the ship's roll angle. However, in the case of the passively-controlled tank this dependency is less.

The next series of diagrammes (fig.8.20 to fig 8.25) show results obtained from experiments on the bench test mechanism for the tank model in 16.3 scale.
These results are presented in the form of characteristics (7.4). The courses of the first harmonic of the moment generated by the passively-controlled tank are decidedly better, than the analogical courses obtained for the passive tank. These characteristics proove the higher efficiency of the passively-controlled tank's action. They have to be examined together with the phase angle characteristics (fig. 8.21, 8.23 and 8.25). They show the influence of control of the passively-controlled tank, on the phase angle between the stabilizing fluid motion and the bench mechanism motion.
It is to be noted, that in the case of the passive tank and the passively-controlled tank in the passive state, the characteristics run monotonously, however when the blocking valves are in operation, then the course of the characteristics vary.
The passively-controled tank is characterised by a nearly constant phase angle ε_M, oscillating about 90°. This is a property, which differentiates the passively-controlled tank from the passive tank; as it is not possible to determine a constant phase angle ε_M for the latter. The phase characteristics were usefull in identifying two basic coefficients of the tank's mathematical model.
They are: the passively-controlled tank's natural frequency and the damping coefficient in the passive state. The results obtained from these characteristics were given

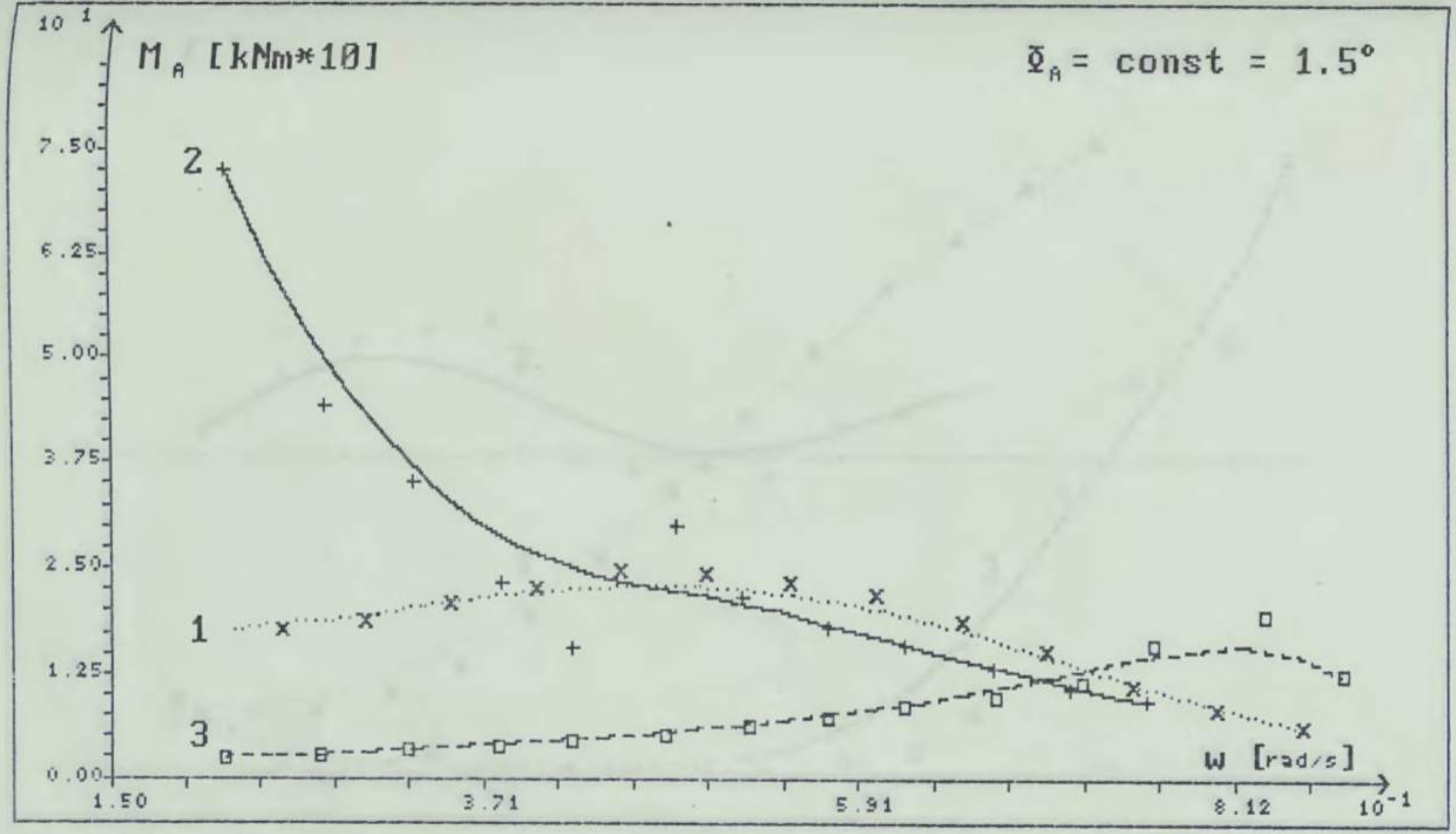

Fig.8.20. Characteristics of the moment amplitude (first harmonic) of the fluid reaction as a function of frequency, calculated from the bench tests for:

1 - the passive tank, 2 - the passively-controlled tank,
3 - the passively-controlled tank in the passive state.

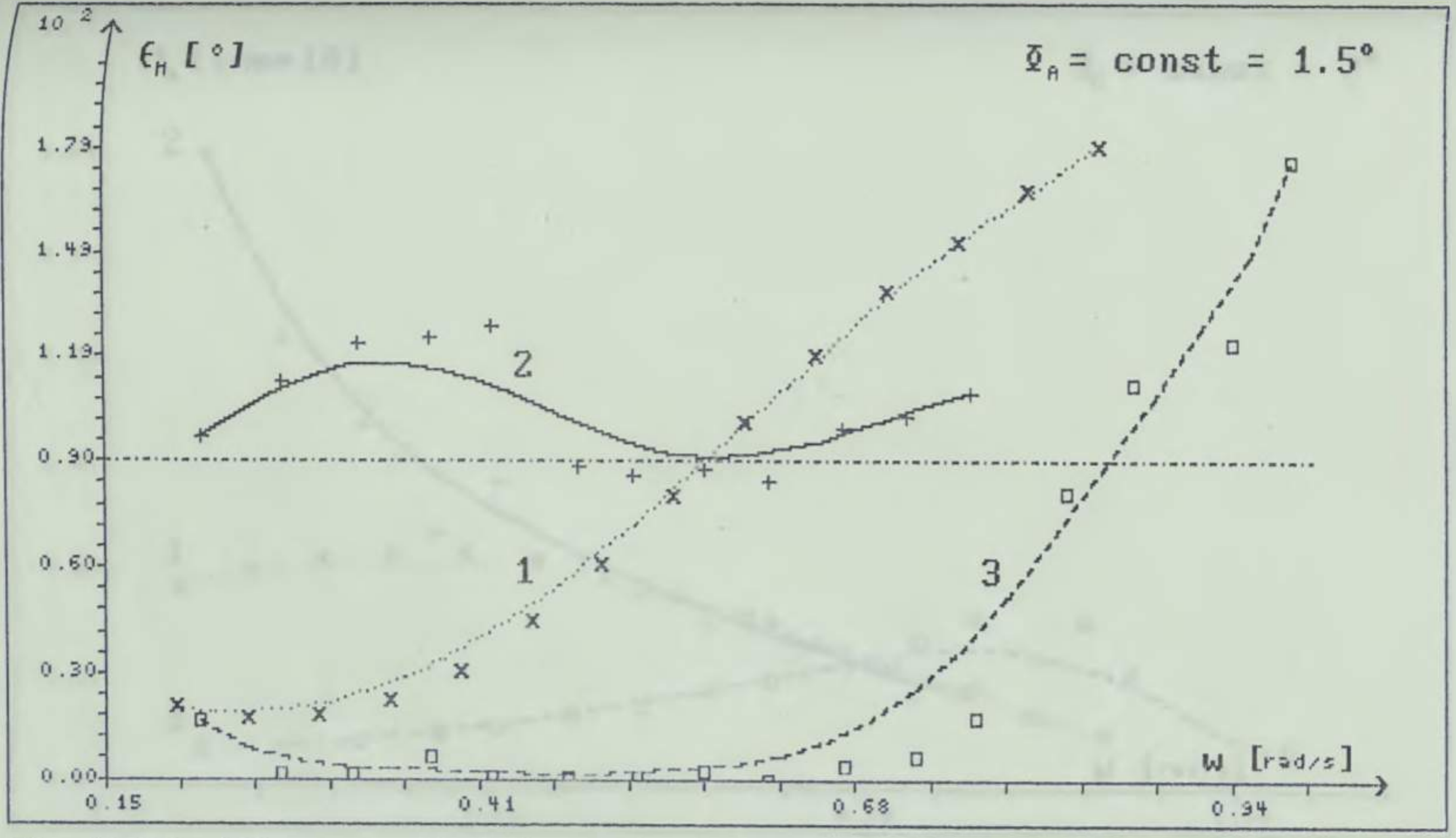

Fig.8.21. Phase angle characteristics of the fluid reaction as a function of frequency, calculated from the bench tests for:

1 - the passive tank, 2 - the passively-controlled tank,
3 - the passively-controlled tank in the passive state.

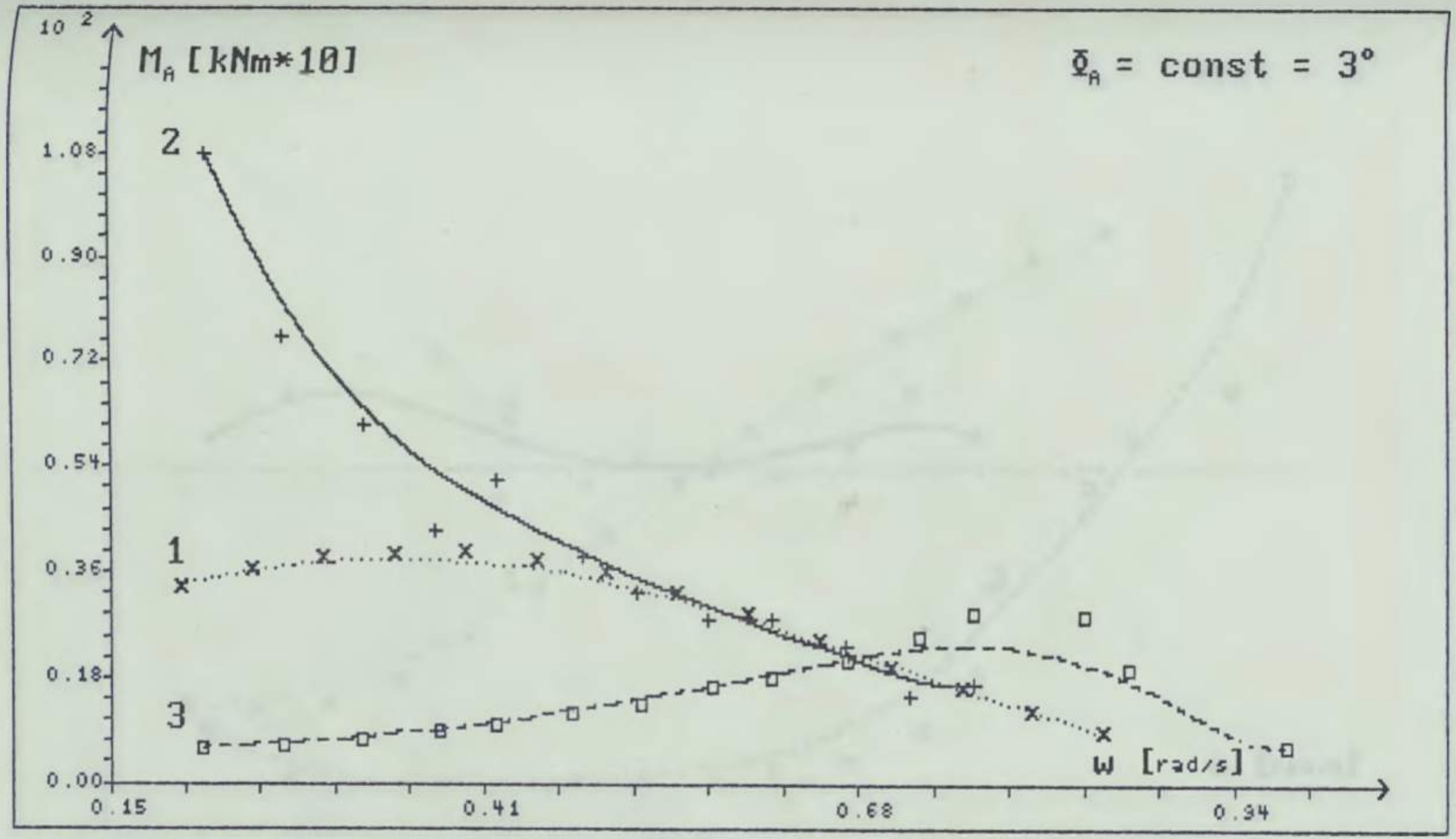

Fig.8.22. Characteristics of the moment amplitude (first harmonic) of the fluid reaction as a function of frequency, calculated from the bench tests for:

1 - the passive tank, 2 - the passively-controlled tank,
3 - the passively-controlled tank in the passive state.

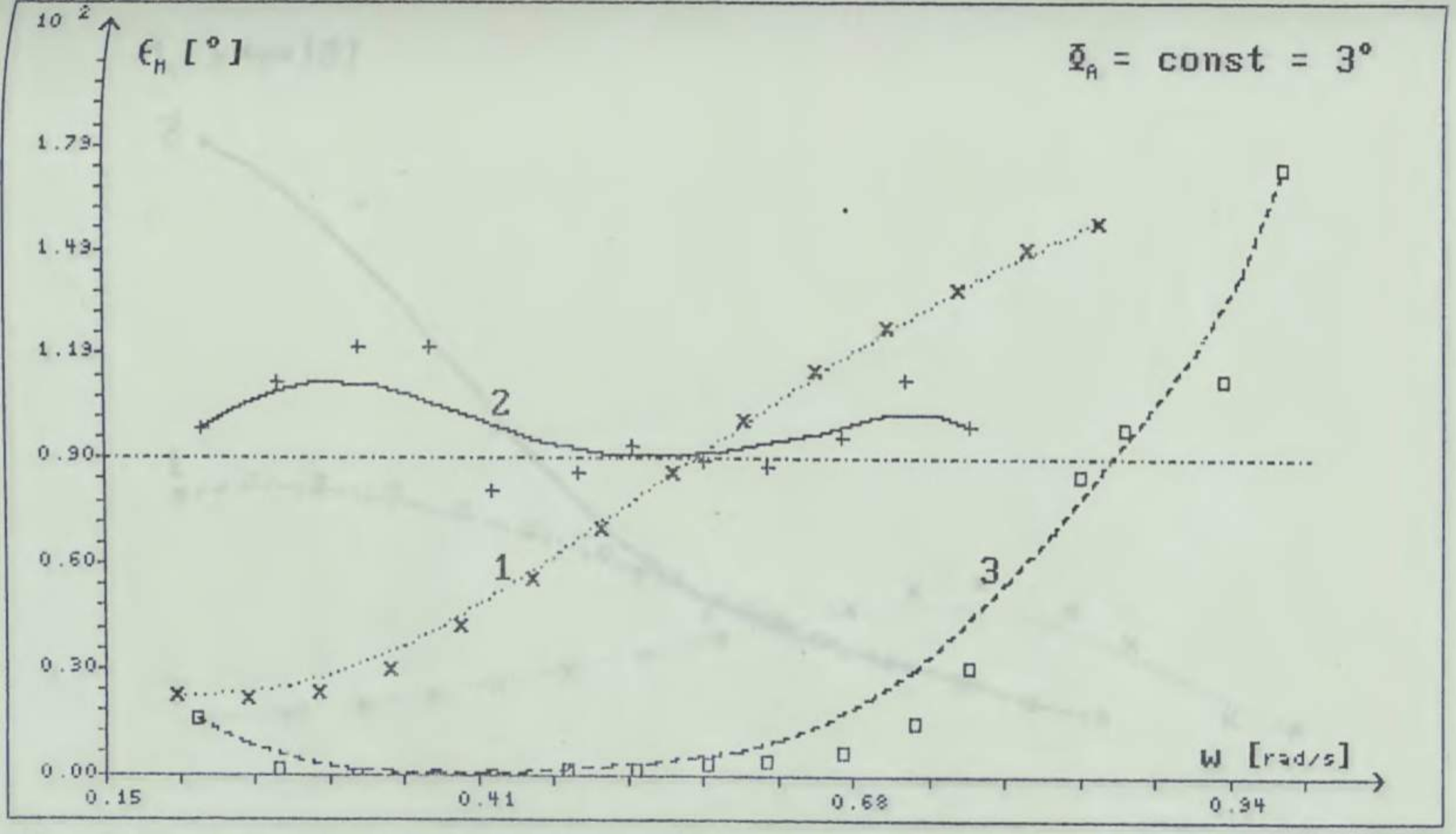

Fig.8.23. Phase angle characteristics of the fluid reaction as a function of frequency, calculated from the bench tests for:

1 - the passive tank, 2 - the passively-controlled tank,
3 - the passively-controlled tank in the passive state.

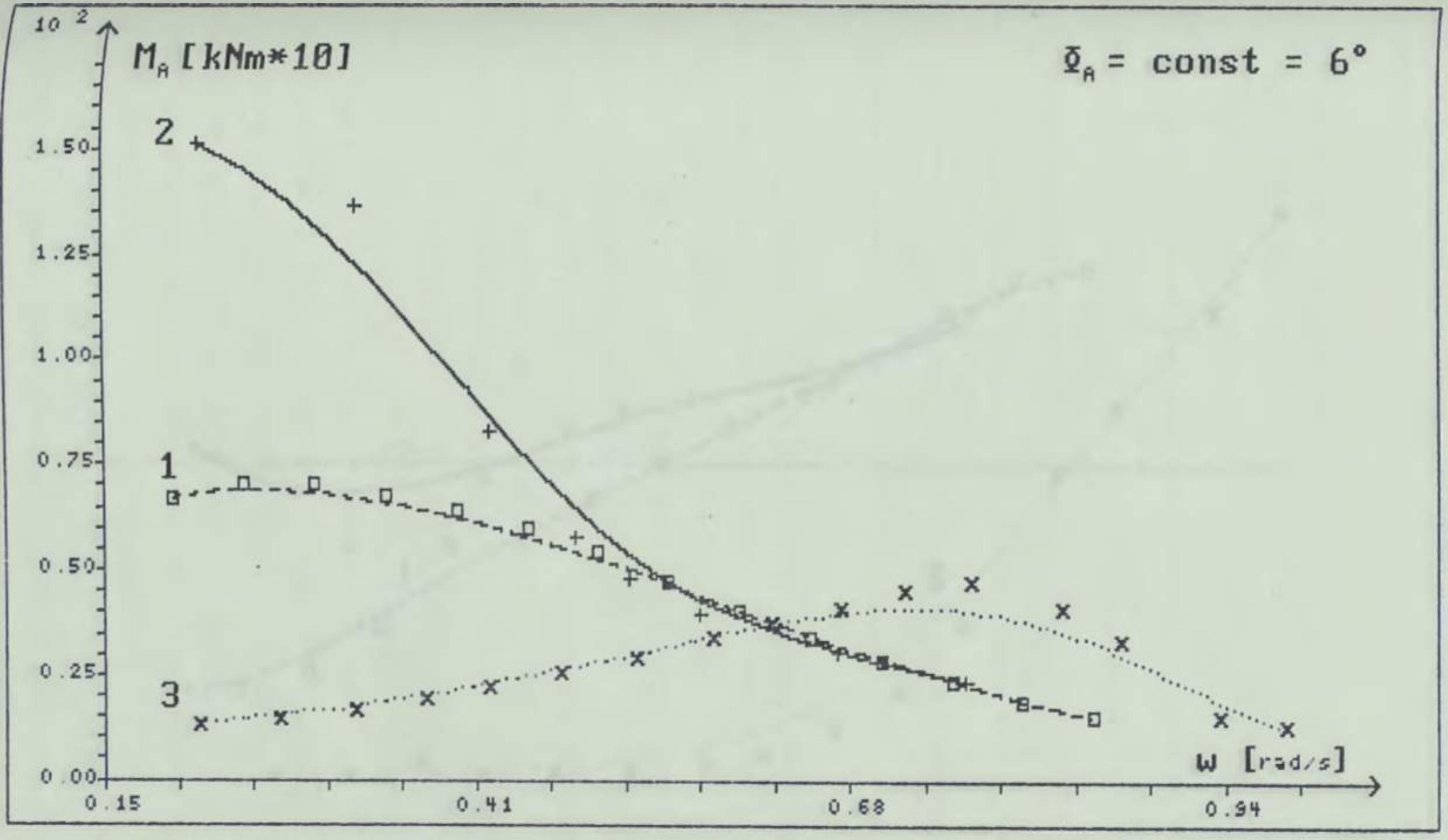

Fig.8.24. Characteristics of the moment amplitude (first harmonic) of the fluid reaction as a function of frequency, calculated from the bench tests for:

1 - the passive tank, 2 - the passively-controlled tank,
3 - the passively-controlled tank in the passive state.

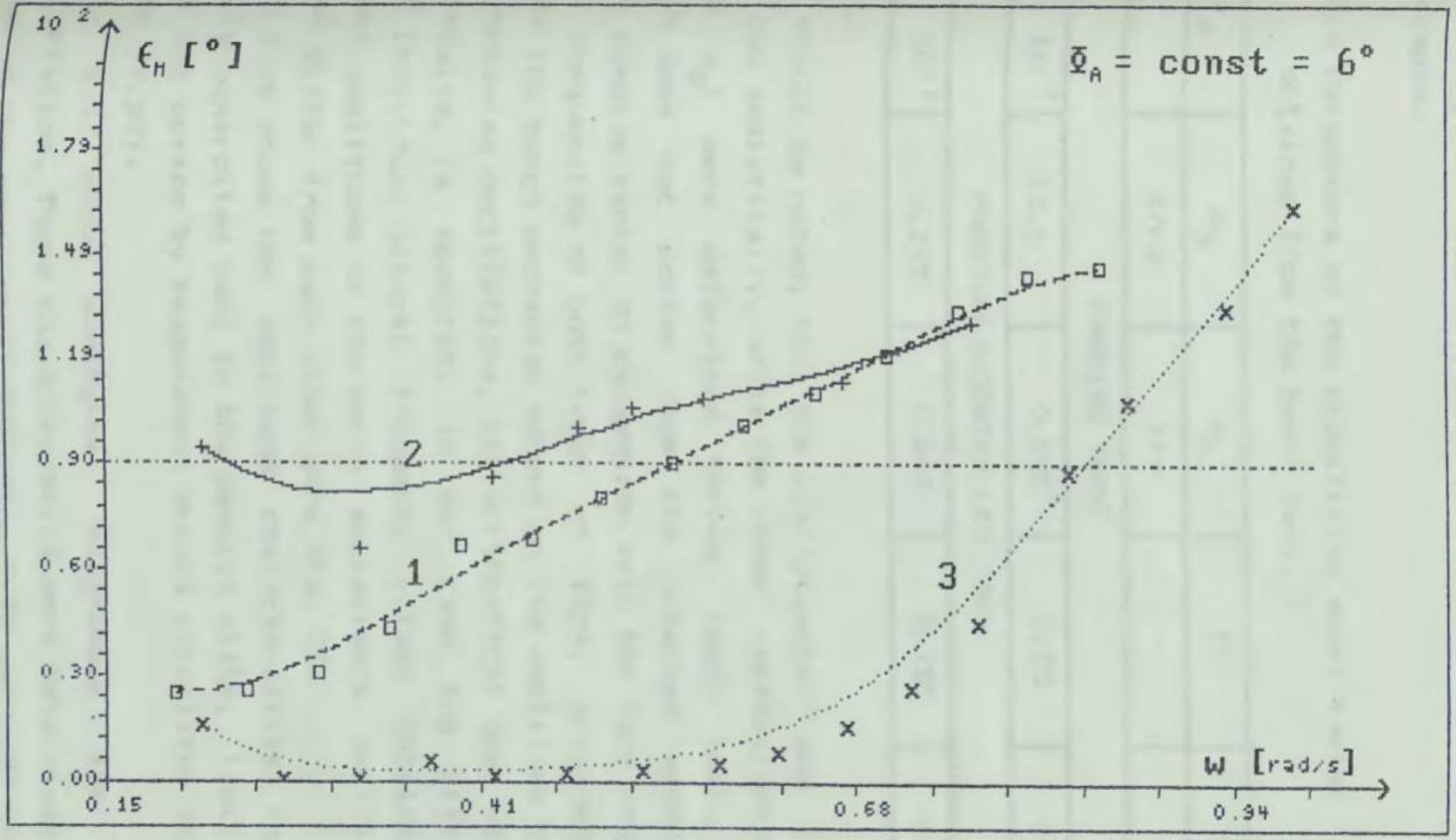

Fig.8.25. Phase angle characteristics of the fluid reaction as a function of frequency, calculated from the bench tests for:

1 - the passive tank, 2 - the passively-controlled tank,

3 - the passively-controlled tank in the passive state.

earlier (fig. 8.4 and 8.5 and in table 8.3), as they were used in computer simulations. Table 8.4 shows all the coefficients.

Table 8.4 Parameters of the stabilizing model tanks obtained from the bench test.

β_ϑ	w_ϑ	ω_ϑ	Γ	s
–	1/rd	1/s	–	m
PASSIVE TANK				
$3.63 \cdot 10^{-9}$	12.6	0.565	0.23	-4.5
PASSIVELY-CONTROLLED TANK				
$1.08 \cdot 10^{-1}$	0.215	0.842	0.155	-4.0

It should be noted, that the coefficients Γ and s were calculated analytically, while the other coefficients (β_ϑ, w_ϑ and ω_ϑ) were determined through bench tests. This approach does not doffer from the accepted methods of testing passive tanks. In connection with the fact that the natural frequencies of both types of tank, obtained from tests on the bench mechanism depend on the amplitude of the bench mechanism oscillations, the arithmetical mean value of these results, is accepted. In our case the difference between individual natural frequency values obtained for different amplitudes of the bench mechanism's oscillations does not differ from each other more than 3%.

Fig 8.26 shows the amplitude characteristics for the passively-controlled tank in the passive state, limited at the top and bottom by measurement errors calculated from the equation (7.29).

Fig. 8.27 shows analogical diagrammes for phase characteristics. These characteristics were determined with the help of the equation (7.30). Fig. 8.28 and 8.29 show the amplitude and phase characteristics of the passively-

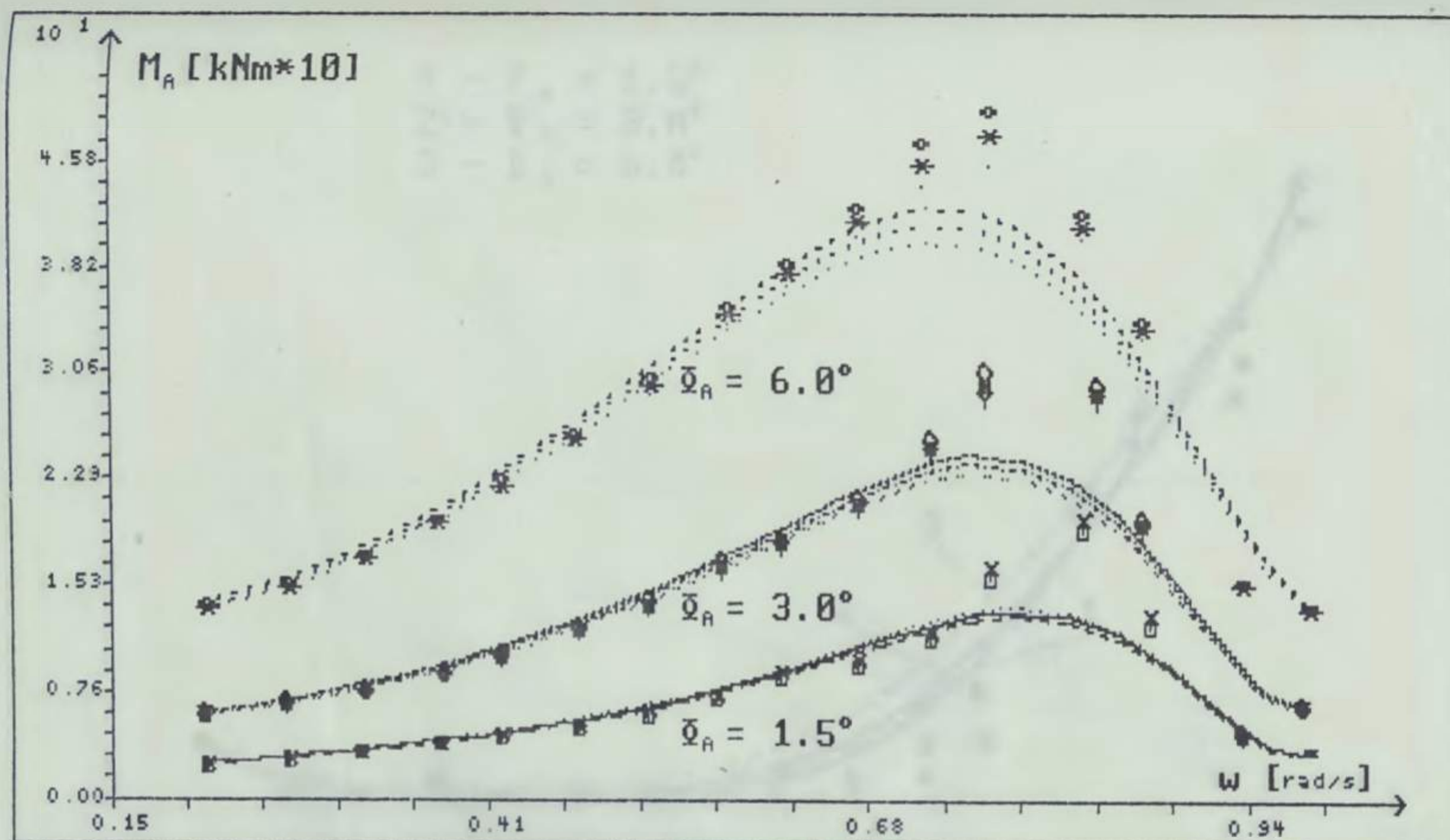

Fig.8.26. Characteristics of the moment amplitude of the fluid reaction as a function of frequency for the passively-controlled tank in the passive state; the measurement curves bordered by the values of the maximum errors.

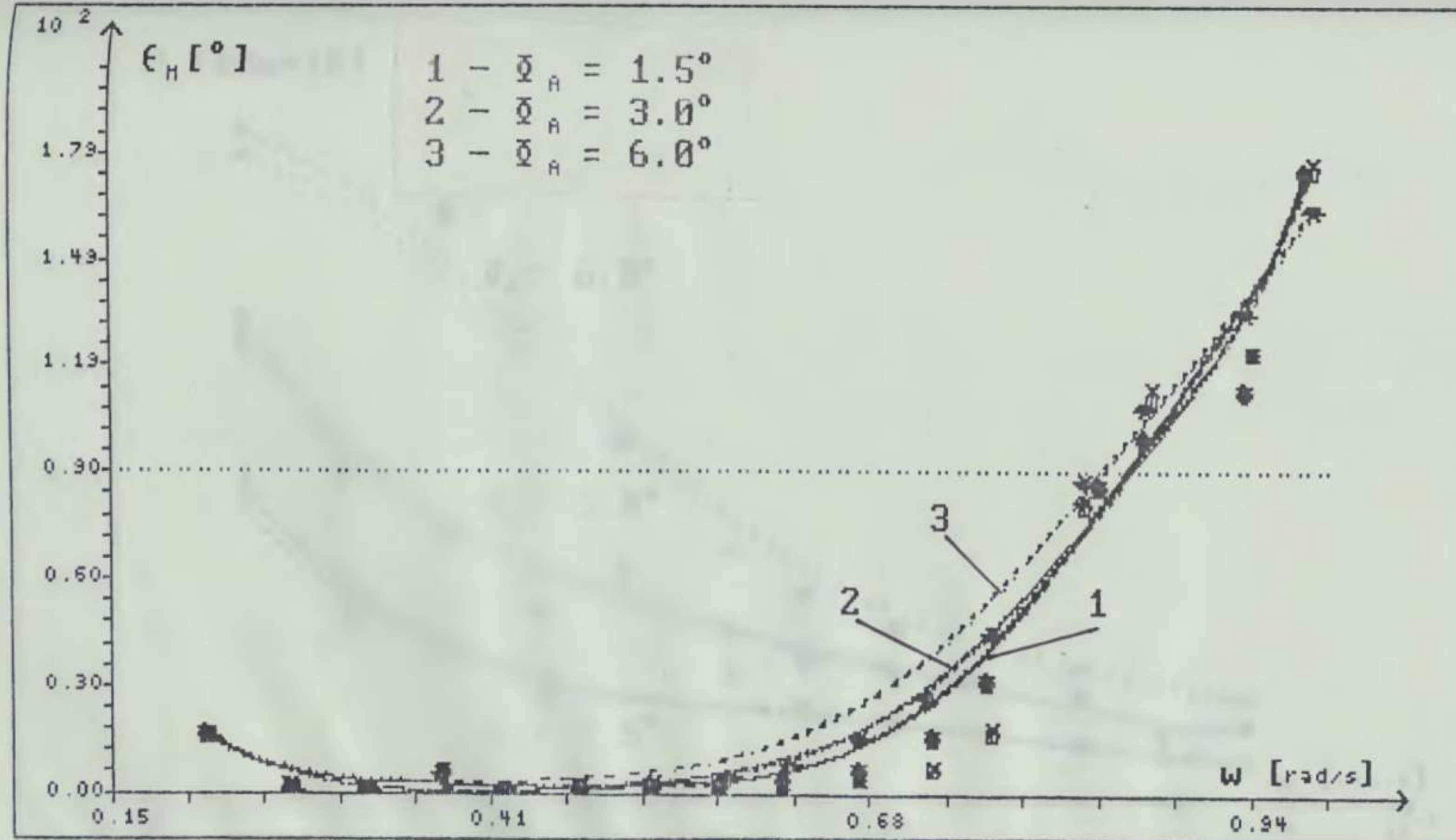

Fig.8.27. The phase angle characteristics of the fluid reaction as a function of frequency for the passively-controlled tank in the passive state; the measurement curves bordered by the values of maximum errors.

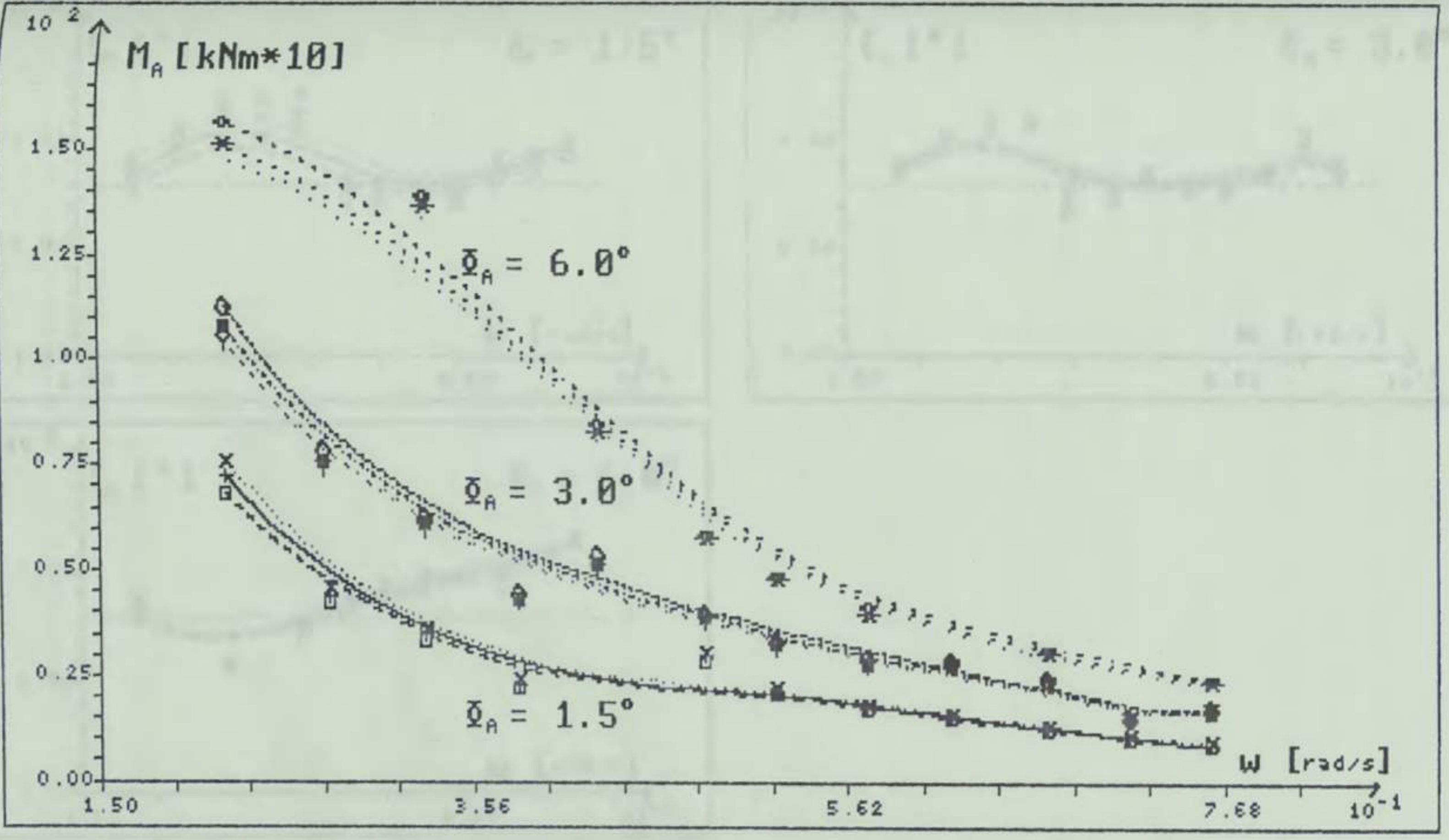

Fig.8.28. Characteristics of the moment amplitude of the fluid reaction as a function of frequency for the passively-controlled tank; the measurement curves bordered by the values of the maximum errors.

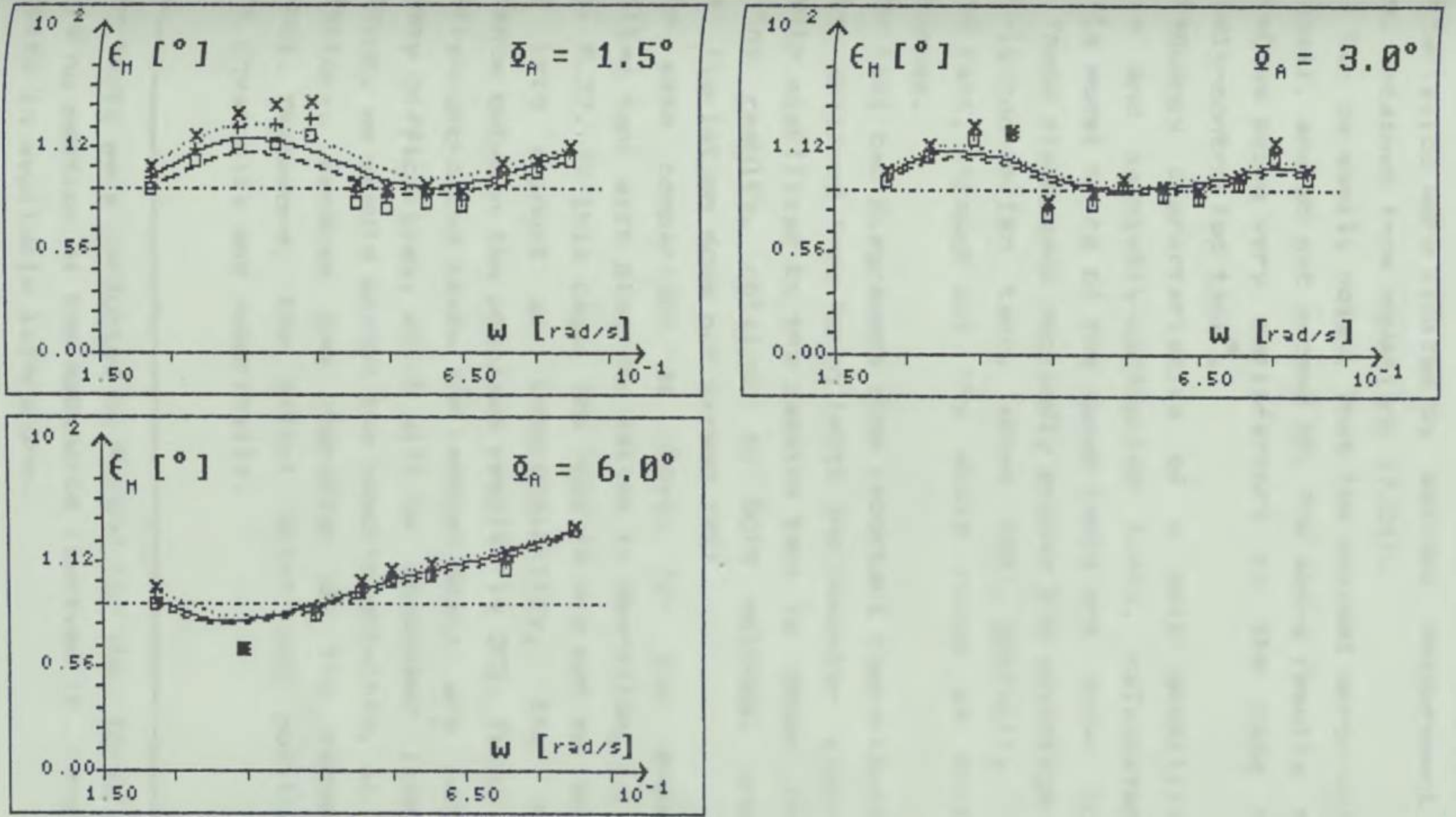

Fig.8.29. The phase angle characteristics of the fluid reaction as a function of frequency for the passively-controlled tank; the measurement curves bordered by the values of the maximum errors.

controlled tank with the air valves in operation. The characteristics were limited by maximal measurement error values, obtained from equations (7.31).

It can be easily noted, that the maximal error values are very small, and do not exceed 8%. The above results can be accepted as being very satisfactory in the case of the passively-controlled tank*.

Frequency characteristics of a ship stabilized by passive and passively-controlled tanks, calculated from separate model tests of the above tanks are shown in fig. 8.30. These diagrammes decidedly proove the advantage of the passively-controlled tank, above the optimally chosen passive tank, through out the whole range of excitation frequencies.

The last two diagrammes show important comparisons. The results, obtained from bench tests and computer simulations of a ship stabilized by the passive tank is shown in fig. 8.31. The results, obtained by both methods, are very similar (variation does not exceed 10%).

The same comparison as above, for the passively-controlled tank with blocking valves in operation, is shown in fig. 8.32. In this case, the results are not so similar. Taking into account air compressibility, the maximum difference between the obtained results is 23%. Tests of the passively-controlled tanks (in reduced scale) are connected with many difficulties, which will be discussed later. At this point, we should accept the results obtained, as being satisfactory, because the character of the curves are identical. Therefore, the effect which was modelled is correct physically and numerically.

---

* These tests were conducted in Poland for the first time. There is no mention of the methodics involved in conducting such tests in available literature.

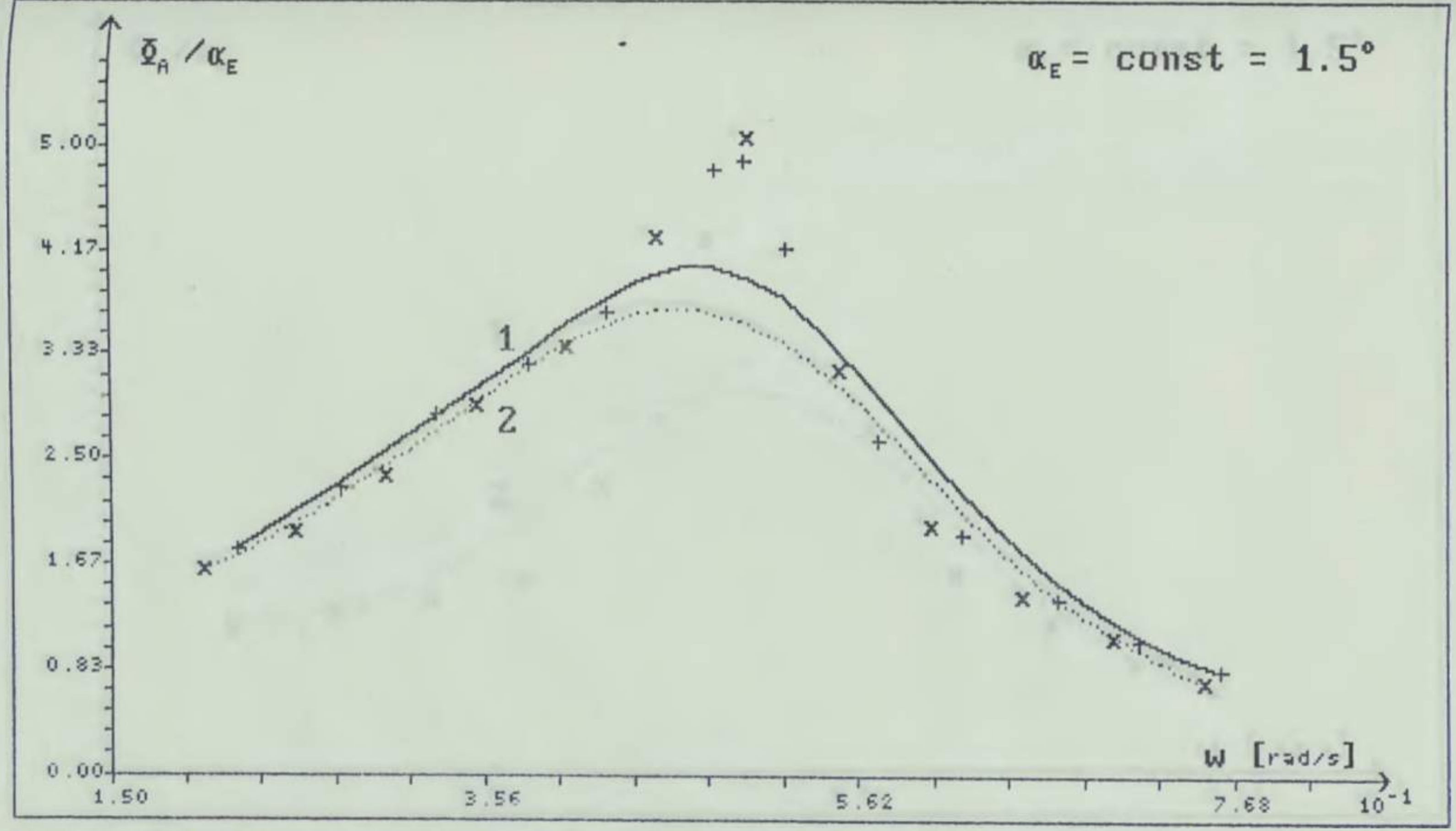

Fig.8.31. Comparison between the model bench tests and the numerical simulation in regular beam waves of constant height for the passive tank;

1 - results from the numerical simulation,
2 - results from the bench tests.

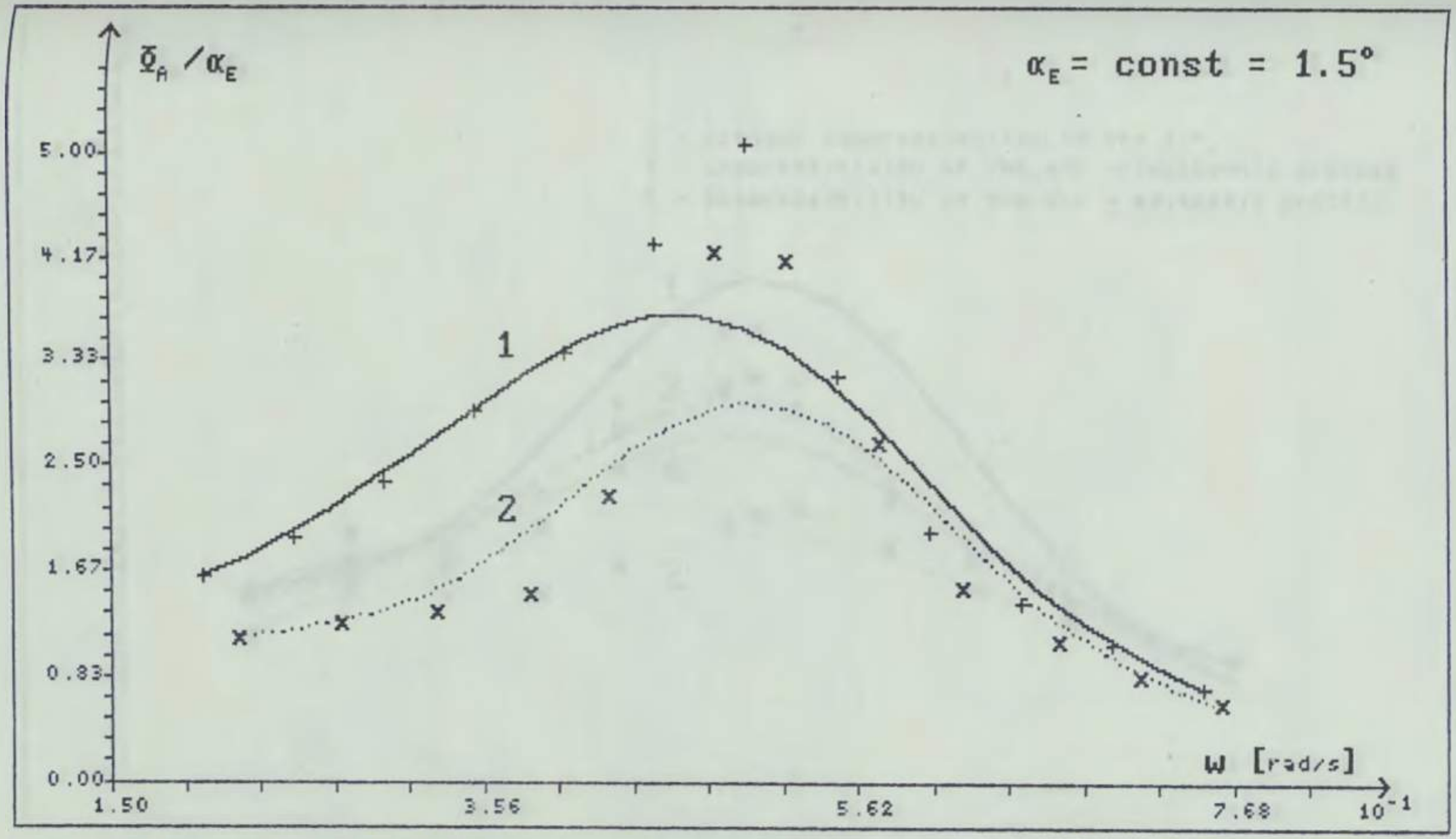

Fig.8.30. The roll response characteristics of the ship with the passive tank (1) and ship with the passively-controlled tank, calculated (in full scale) from the bench tests.
The dots correpond to the direct application of the tank moment characteristics.

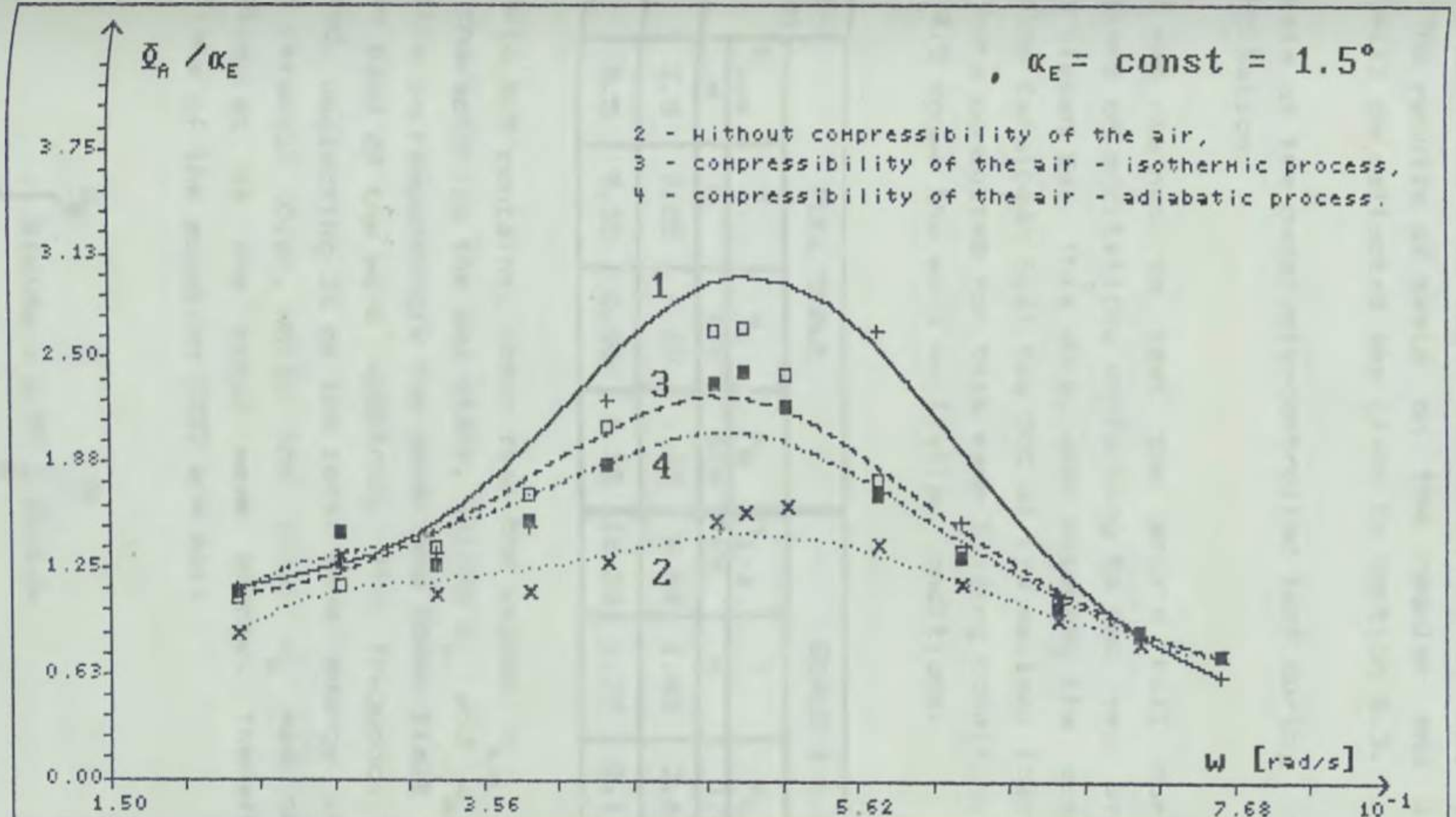

Fig.8.32. Comparison between the model bench tests and the numerical simulation in regular beam waves of constant height for the passively-controlled tank;

1 - results from the bench tests,
2,3,4 - results from the numerical simulations.

The next step in this study, is the discussion of results obtained from tests (experiments) on the irregular wave. The results of tests on the regular and irregular waves will be collected and given in section 8.3.

8.2 Tests of the passively-controlled tank during irregular excitation.

It was planned to test the ship's roll during two conditions of excitation, conforming to two sea states on the Caraibean Sea. This ship, when entering its operational area (the Caraibean Sea) has 50% of its maximal load. Bench tests were conducted for this same loading condition. Table 8.5 shows the wave excitation conditions.

Table 8.5

Realization	FULL SCALE				SCALE 1 : 28			
	$h_{1/3}$	T	ω_L	ω_H	$h_{1/3}$	T	ω_L	ω_H
	m	s	1/s	1/s	cm	s	1/s	1/s
1	2.5	7.55	0.50	1.43	8.93	1.43	2.64	7.57
2	4.5	9.35	0.40	1.15	16.10	1.77	2.12	6.09

Table 8.5 contains, other than the values $h_{1/3}$ and T, which characterize the sea state, values ω_L and ω_H, which determine correspondingly the upper and lower limit of the ergodic band of the wave spectrum. The frequency ω_L was obtained, neglecting 3% of the total wave energy contained in the interval $(0,\infty)$, while the value ω_H was obtained, neglecting 4% of the total wave energy. Therefore the conditions of the equation [25] are met:

$$\int_{\omega_L}^{\omega_H} S(\omega)\,d\omega = 0.93 \int_{0}^{\infty} S(\omega)\,d\omega \qquad (8.1)$$

where:

$$S(\omega) = \frac{173h^2_{1/3}}{T^4}\omega^{-5}\exp\left[-\frac{691}{T^4}\omega^{-4}\right].$$

During initial experiments in the basin, it was found, that in wave conditions of $h_{1/3} = 4.5m$, the heel angles of the ship's model (unstabilized) was so great that, there was a possibility of the hull capsizing and the destruction of the equipment built into the hull. Therefore, it was decided to reduce the amplitude motion of the wave generator's plates, ie. the reduction of the wave amplitudes, preserving, however, the proper frequency ranges of the harmonic components. This does not in any way deviate from the aims of the experiments, because the aim was not to test the ship's behaviour on a concrete predetermined sea, but to check the effectiveness of the passively-controlled tank in irregular, near resonance wave conditions.

The width of the tank model, built into the hull is 522mm. The main problem in all model tests are the scale effects and errors, due to conducting tests in reduced scale. This was discussed at 14-th International Towing Tank Conference in 1975 and is also discussed in [8]. From these materials, we can conclude, that the required minimum width for tank models is 380 mm. This width is much less than the width of the tank used in our model experiments.

The diagramme 8.33 presents the energy spectra of both wave types, determined directly from measurements of the basin's wave elevation using resistance sensors. The energy spectrum 1 corresponds to $h_{1/3} = 2.5m$ (in the scale 1:28 $h_{1/3} = 8.93cm$). The curve 2 corresponds to the second type of wave, however, of lesser amplitude. This can be clearly seen in spectrum 2. The value $\omega_{\phi o}$ of the ship (in scale 1:28 $\omega_{\phi o} = 2.64s^{-1}$), is presented in fig. 8.33. This diagramme prooves, that the tests were conducted for near resonance waves. However, wave conditions 2 should be accepted as fully

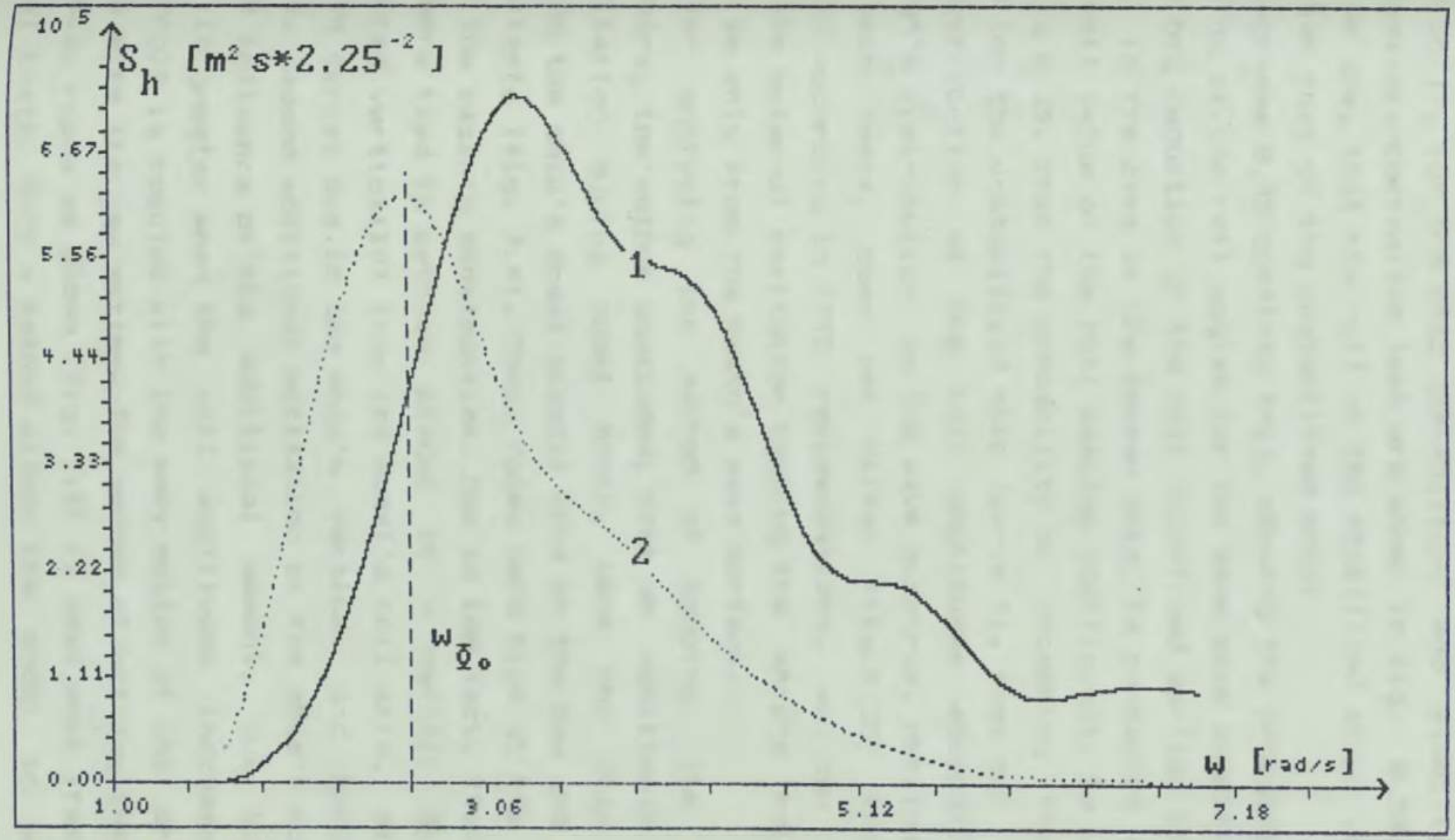

Fig.8.33. The wave height spectra of irregular beam waves generated as an external disturbance during model experiments with the ship roll tests in the towing tank.

corresponding to conditions of resonance waves.
The ship's roll energy spectra (obtained for the wave condition 1), for the ship unstabilized and stabilized by the passively-controlled tank are shown in fig. 8.34. From this, we see, that the roll of the stabilized ship is much less than that of the unstabilized ship.

Diagramme 8.35 confirms this, showing the probability of exceeding of the roll angles for the same wave condition.
Such a big reduction of the roll (confirmed earlier by bench tests), in the case of the tested ship, is connected with a very small value of the roll damping coefficient. We can see from fig 8.35, that the probability of exceeding the roll angles for the unstabilized ship (curve 1), does not define the distribution of the roll amplitudes according to Rayleigh's distribution. As the wave spectrum, obtained from the basin tests, does not differ (fig.8.33) from the spectrum according to ITTC recomendations, we can assume that, the external excitation causing the ship's roll did not arise only from the basin's wave surface.

After analysing the method of keeping the model stationary, the author concluded, that an additional cause of excitation during model tests, were the stay ropes (holding the ship's model steady) tied at the bow and stern during tests (fig. 7.4). These ropes were tied at the other end to the basin's construction. Due to the fact, that these ropes were tied to catches placed at a certain distance (measurred vertically) from the model's roll axis, all the reaction forces due to the ship's vertical and horizontal motions, caused additional excitation of the ship's roll.

The influence of the additional moment, due to stay ropes, is greater when the roll amplitudes increase. The ship's roll is coupled with the sway motion of the ship as well as with the yaw motion. The method of holding the ship using stay ropes as shown (fig. 7.4) is used most frequently in model tests. Such a method allows the model to execute

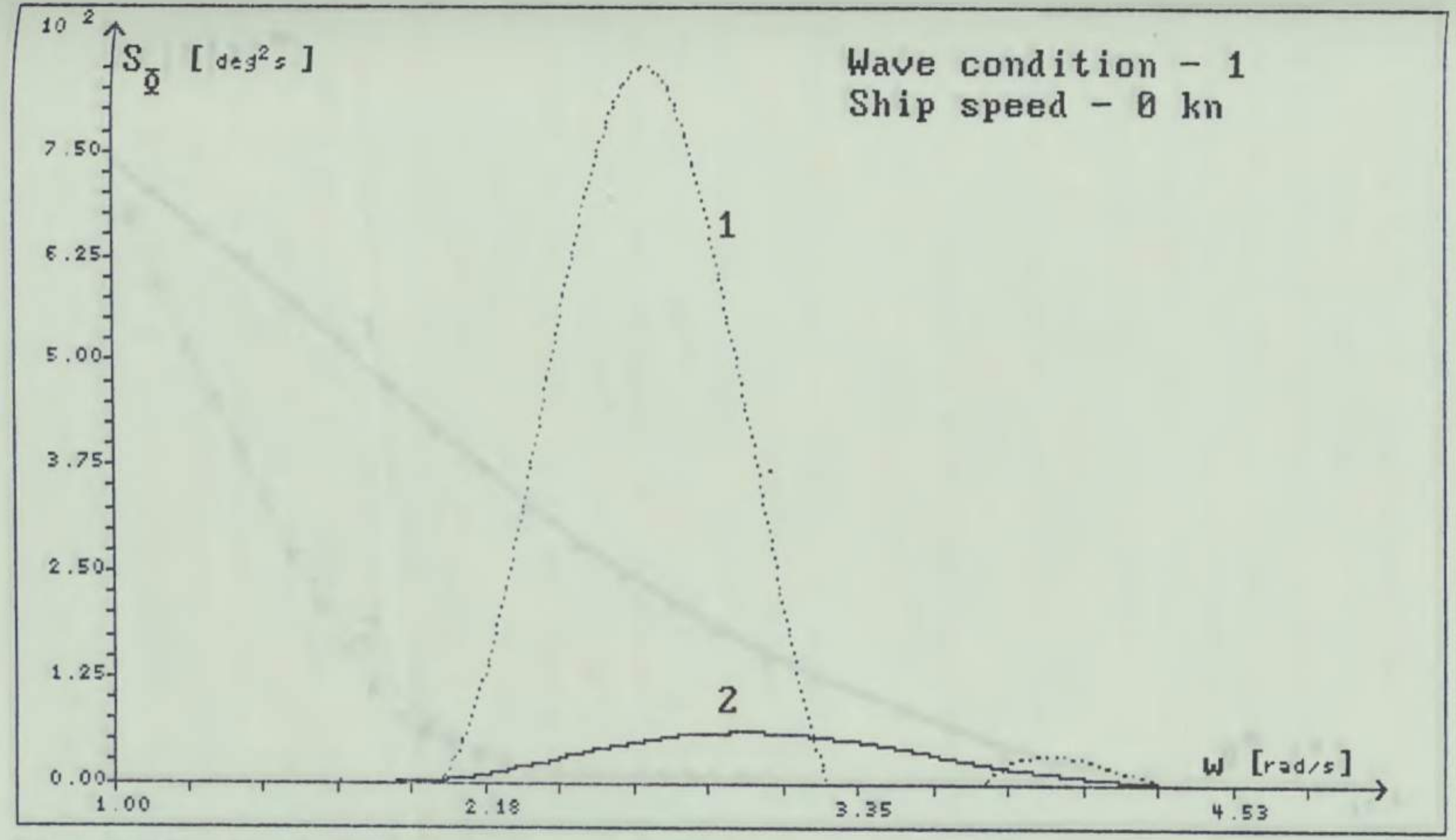

Fig.8.34. The ship roll spectra of irregular beam waves in model scale for the unstabilized ship (1) and ship with the passively-controlled tank (2).

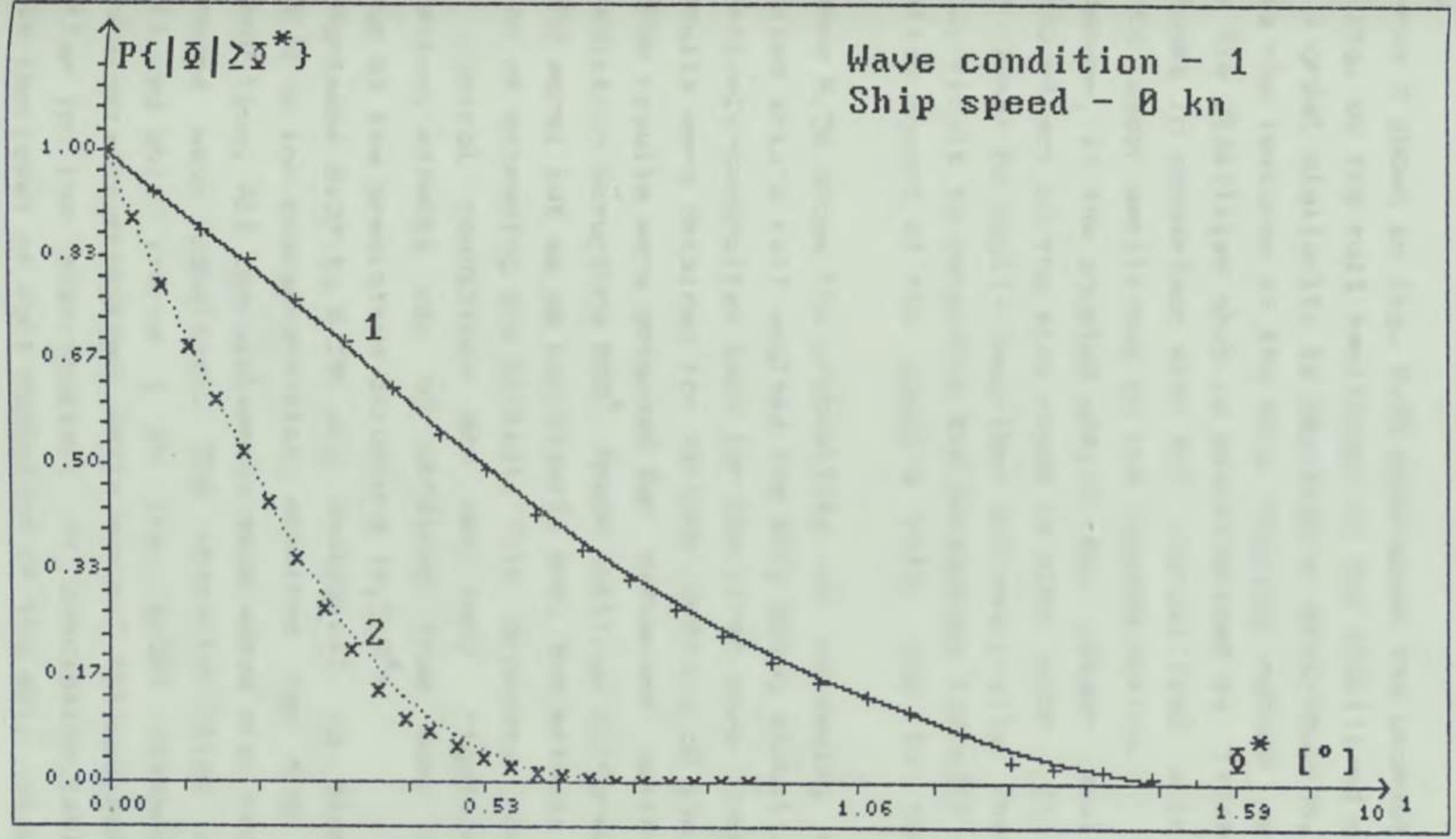

Fig.8.35. Probability of exceeding an angle Φ^* in irregular beam waves for the unstabilized ship (1) and ship with the passively-controlled tank (2).

complex motion, similar to motion at sea in real conditions, but causes, however, such undesired effects as described above.

The curve 2 shown in fig. 8.35 determines the probability of exceeding, of the roll amplitudes of the stabilized ship. It shows a great similarity to Rayleigh's distribution, which prooves the features of the ship holding method described above. The stabilized ship is characterised by lesser roll amplitudes in comparison with the unstabilized ship. This leads to lesser amplitudes of the coupled motion.

However, if the coupled motion has lesser amplitudes, then the effect of the stay ropes is also much less. This effect cannot be easily described mathematically. Therefore, it is difficult to determine the percentage increase of the excitation moment of the model's roll, due to the stay ropes.

Diagramme 8.36 shows the probability of exceeding of the stabilized ship's roll angles; the ship being stabilized by the passively-controlled tank for the first wave condition. The results were obtained for various controls of the tank.

These results were obtained for different settings of the prediction structure PDD^2. These settings differed up to 100% and more, but as we can clearly see, the effects on the function of exceeding are minimal. This prooves, that the tank's control conditions are not very rigorous and satisfactory effects can be obtained from many control settings of the prediction structure (P,D,D^2).

Diagrammes 8.37 to 8.39 are analogical to diagrammes 8.34 to 8.36 for characteristics, obtained for the second wave condition. All the statements made above are true for the second wave condition. The characteristics of the unstabilized ship (curve 1 in fig. 8.38) differs from the Rayleigh's distribution, while curve 2 (stabilized ship) is similar to the distribution. In conclusion, table 8.6 presents the level of roll reduction of the ship stabilized by the passively-controlled tank.

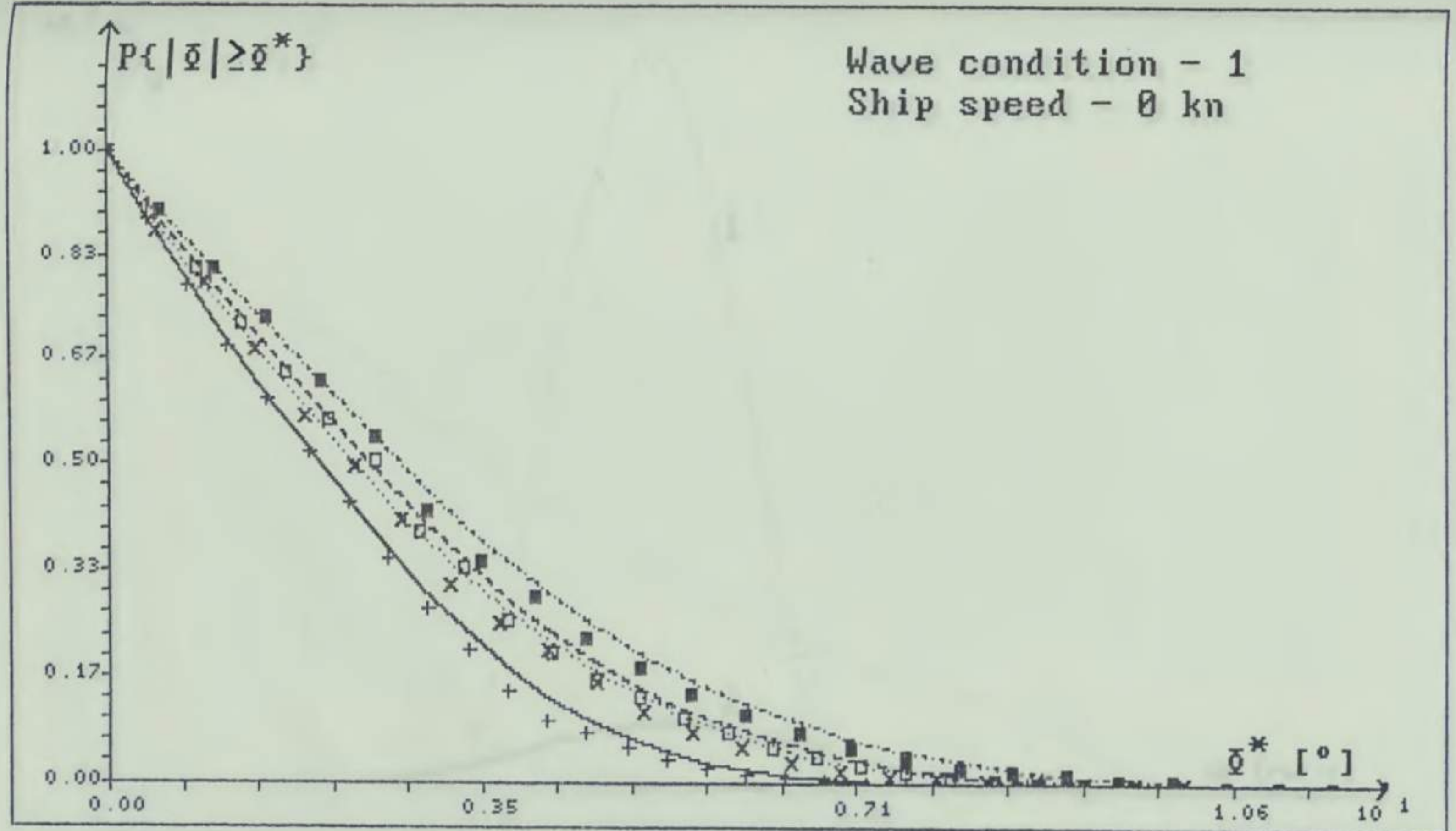

Fig.8.36. Probability of exceeding an angle Φ^* in irregular beam waves for the ship stabilized by passively-controlled tank, obtained from rolling model tests for different variants of tank control.

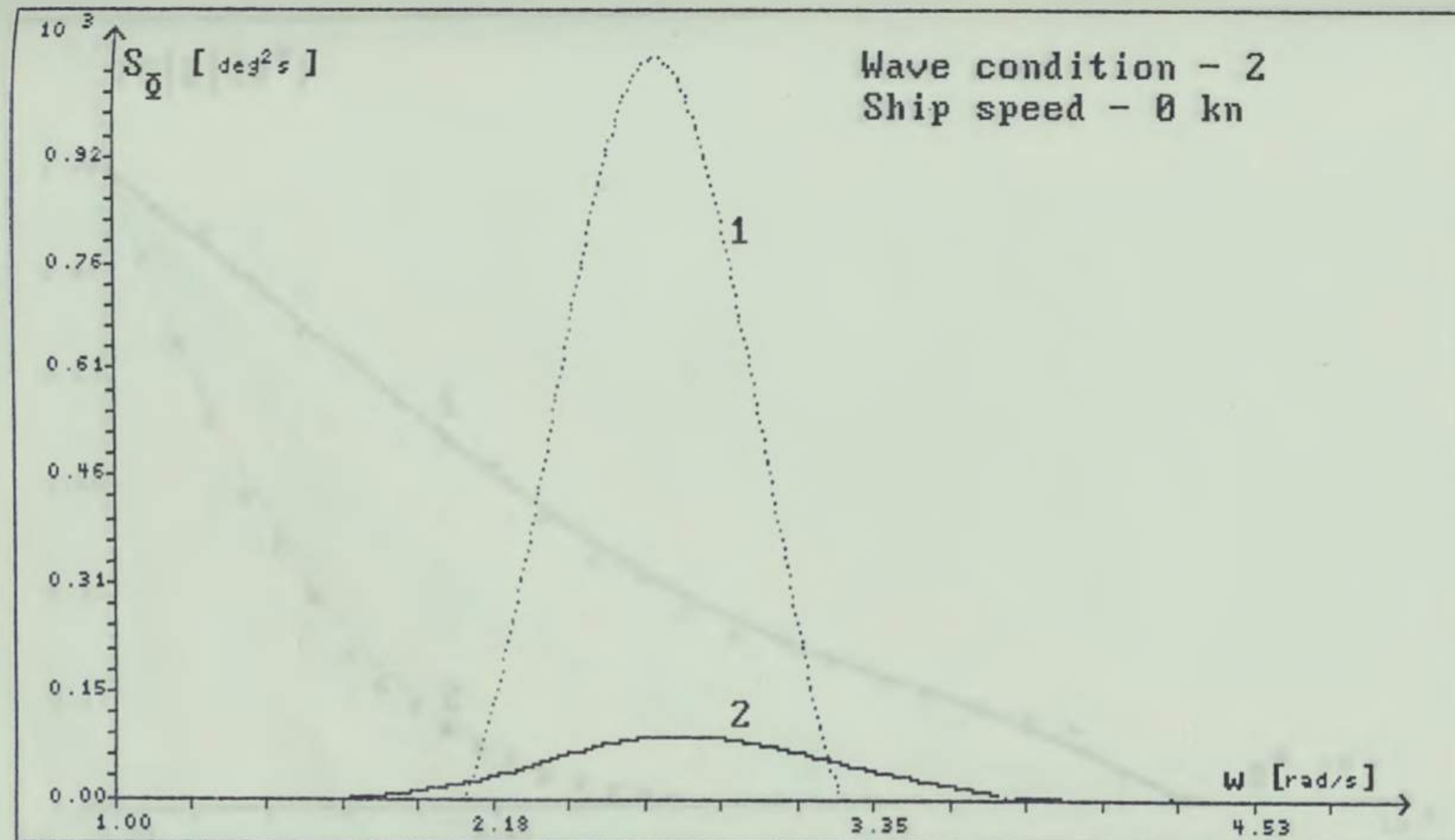

Fig.8.37. The ship roll spectra of irregular beam waves in model scale for the unstabilized ship (1) and ship with the passively-controlled tank (2).

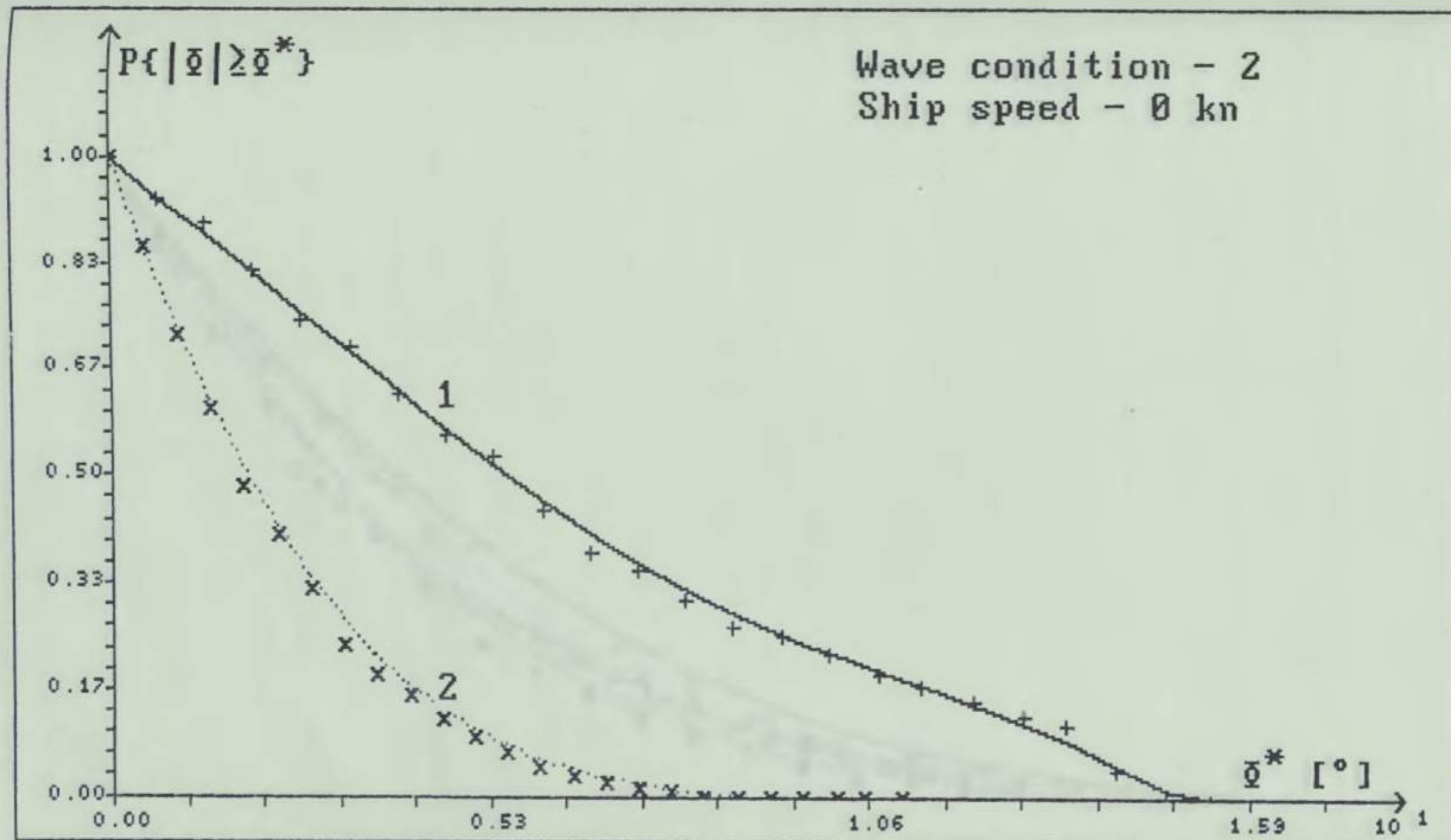

Fig.8.38. Probability of exceeding an angle Φ^* in irregular beam waves for the unstabilized ship (1) and ship with the passively-controlled tank (2).

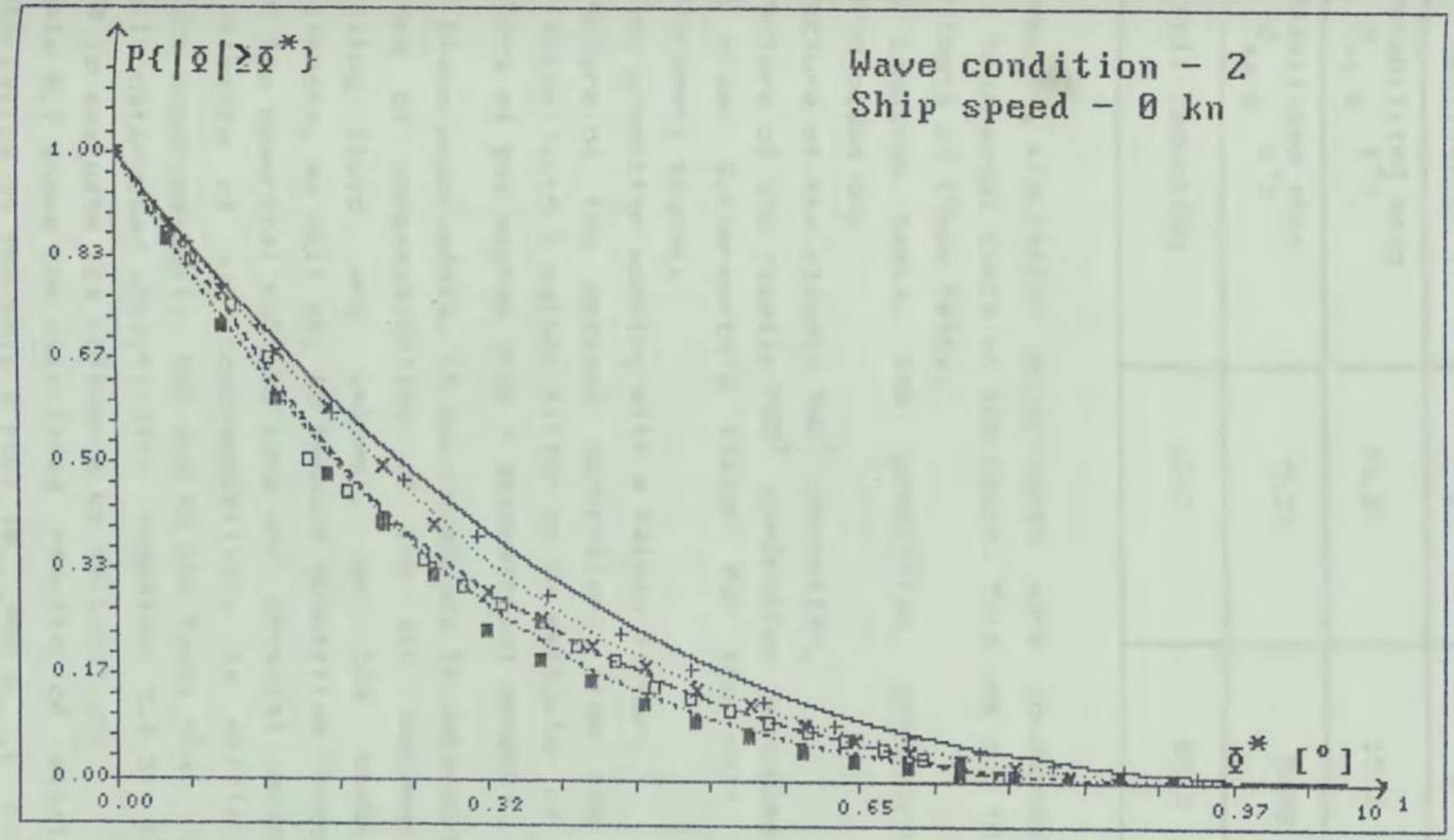

Fig.8.39. Probability of exceeding an angle Φ^* in irregular beam waves for the ship stabilized by passively-controlled tank, obtained from rolling model tests for different variants of tank control.

Table 8.6

Realization	1	2
Unstabilized ship $\phi_{A1/3}$ [°]	13.8	15.2
Stabilized ship $\phi_{A1/3}$ [°]	5.21	5.58
% roll reduction	62.2	63.3

Computer simulation experiments were conducted more widely, than model tests in the basin. This was due to much lesser costs of these tests.

In computer tests, the prediction structure was simulated based on:

1. structure of the classic PDD^2 controller,
2. structure of the classic PDD^2 controller equipped with 3rd order Butterworth's filter for the roll angle measurement signal,
3. k-step predictor working with a Kalman filter,
4. structure of the optimal controller, from the state variables (with a Kalman filter as an estimator of state vectors of the system ship - asymmetrical moment).

In these experiments, it was necessary to determine the influence of compressibility of the air (between the stabilizing fluid and valves), on the stabilizing effectiveness, as well as, to conduct quantitive comparison between the numerical calculations and physical modelling. The influence of air compressibility, is difficult to determine experimantally, but due to the fact, that it can be easily determined analytically (section 3.4.3) it was decided to evaluate its influence by calculation.

Table 8.7 shows the calculated results of statistical characteristics of the ship's roll ($\phi_{A1/3}$ and $\ddot{\phi}_{A1/3}$) for two sea states, as well as, for all possible variations of the mathematical model. By analysing this table we conclude, that allowing for air compressibility causes a small percentage reduction in the roll stabilization. It is important that allowing for air compressibility, we do not make any essential changes in the transition processes.

Table 8.7 Parameters characterising the ship's roll process of the unstabilized ship and the ship stabilized by the passive and passively-controlled tanks, determined on the basis of computer simulation (control of the tank using PDD^2 structure with a Butterworth filter).

Ship's speed [knots]		WAVE PARAMETERS									
		h = 2.5 m, T = 7.55 s					h = 4.5 m, T = 9.35 s				
		COURSE ANGLE of WAVE					COURSE ANGLE of WAVE				
		30°	60°	90°	120°	150°	30°	60°	90°	120°	150°
UNSTABILIZED SHIP											
7	$\bar{\phi}_{1/3}$	6.07	8.02	3.63	2.09	1.19	10.7	15.1	15.5	12.1	3.26
	$\bar{\ddot{\phi}}_{1/3}$	1.59	2.29	1.70	1.41	0.87	2.72	4.05	4.45	3.71	1.50
0	$\bar{\phi}_{1/3}$	X	X	8.64	X	X	X	X	23.7	X	X
	$\bar{\ddot{\phi}}_{1/3}$	X	X	2.48	X	X	X	X	5.74	X	X
SHIP STABILIZED by PASSIVE TANK											
7	$\bar{\phi}_{1/3}$	2.11	3.54	3.21	2.07	1.24	3.80	7.63	8.32	6.90	2.80
	$\bar{\ddot{\phi}}_{1/3}$	0.74	1.41	1.71	1.49	0.93	1.14	2.38	2.99	2.76	1.49
0	$\bar{\phi}_{1/3}$	X	X	3.71	X	X	X	X	9.75	X	X
	$\bar{\ddot{\phi}}_{1/3}$	X	X	1.70	X	X	X	X	3.18	X	X

Ship's speed [knots]		WAVE PARAMETERS									
		h = 2.5 m, T = 7.55 s					h = 4.5 m, T = 9.35 s				
		COURSE ANGLE of WAVE					COURSE ANGLE of WAVE				
		30°	60°	90°	120°	150°	30°	60°	90°	120°	150°
SHIP STABILIZED by PASSIVELY-CONTROLLED TANK											
WITHOUT AIR COMPRESSIBILITY											
7	$\bar{\phi}_{1/3}$	1.72	2.86	2.36	1.83	1.05	3.00	4.47	5.48	4.81	2.01
	$\bar{\ddot{\phi}}_{1/3}$	0.51	1.11	1.54	1.44	0.90	0.79	1.53	2.24	2.26	1.33
0	$\bar{\phi}_{1/3}$	X	X	2.92	X	X	X	X	5.84	X	X
	$\bar{\ddot{\phi}}_{1/3}$	X	X	1.49	X	X	X	X	2.14	X	X
ISOTHERMIC PROCESS											
7	$\bar{\phi}_{1/3}$	1.69	2.45	2.28	1.69	1.00	3.27	4.96	5.35	4.38	1.96
	$\bar{\ddot{\phi}}_{1/3}$	0.54	1.08	1.47	1.38	0.85	0.89	1.64	2.21	2.16	1.27
0	$\bar{\phi}_{1/3}$	X	X	2.53	X	X	X	X	6.13	X	X
	$\bar{\ddot{\phi}}_{1/3}$	X	X	1.41	X	X	X	X	2.23	X	X
ADIABATIC PROCESS											
7	$\bar{\phi}_{1/3}$	1.60	2.48	2.28	1.73	1.01	3.09	4.92	4.98	4.41	1.98
	$\bar{\ddot{\phi}}_{1/3}$	0.51	1.04	1.47	1.39	0.86	0.85	1.63	2.22	2.17	1.28
0	$\bar{\phi}_{1/3}$	X	X	2.53	X	X	X	X	5.69	X	X
	$\bar{\ddot{\phi}}_{1/3}$	X	X	1.41	X	X	X	X	2.12	X	X

The course angle of the wave in table 8.7, is determined in such a way, that the angle 0° denotes following seas and the angle 180° denotes head seas. Beam seas are denoted by the wave uprush of 90°.

Through well chosen computer simulation experiments, the problem of determing the type of the prediction structure, (optimal for the control of the passively-controlled tank) can be easily solved. The value $\overline{\Phi}_{A1/3}$ was accepted as being the optimal criterion, in these experiments.

Table 8.8 shows the criteria for all the types of predictor structures determined for the stationary ship.

Table 8.8 Stabilization quality criteria of the ship stabilized by the passively-controlled tank for different types of prediction structures (stationary ship, adiabatic process)

	WAVE PARAMETERS	
	h = 2.5 m, T = 7.55 s	h = 4.5 m, T = 9.35 s
	PDD^2 CONTROLLER	
$\phi_{A1/3}$ [°]	2.71	5.80
	PDD^2 CONTROLLER with Butterworth's filter	
	2.53	5.69
	KALMAN PREDICTOR	
	2.62	6.00
	OPTIMAL CONTROLLER	
	2.50	5.81

Results achieved through simulation of the ship's roll, taking into account adiabatic changes in the air compressibility process are presented. No special reasons were taken into account, and table 8.8 is representative for other conditions. For the model, without taking into account air compressibility, as well as, for modelling isothermic changes, the change tendencies of the criterion value

$\bar{\phi}_{A1/3}$ are identical.

The effective wave slope spectra of irregular beam waves used as external excitation during numerical simulation tests are presented in fig. 8.40. The wave height spectra, were not included here (as in fig. 8.33), because the simulation programmes did not output the wave height, but only the wave slope.

Diagrammes 8.41 to 8.46 present the results of simulation experiments for wave heights of $h_{1/3}$ = 2.5m. Diagramme 8.41 shows the exceeding function of the ship's roll,obtained for the stationary ship. This prooves the excellent efficiency of the passive tank, although in this case too, it is to be seen, that the passively-controlled tank is superior.

The other diagrammes show the roll spectrum functions for the ustabilized ship and the ship stabilized by the passive and passively-controlled tank.

The presentation of spectrum functions, not exceeding characteristics was, due to the fact, that in practice the exceeding characteristics would not clearly differ from each other.

Diagrammes 8.47 to 8.52 show the results of simulation tests identical to the ones above, touching however higher seas, characterised by wave heights $h_{1/3}$ = 4.5m. The conclusions drawn from these tests are identical to the ones presented above.

8.3 Conclusions drawn from experiments.

The methodics and results of many years of theoretical and experimental investigations on the passively-controlled tank stabilizing the ship's roll, as presented above, allows us to answer the following two basic questions:

(i) - whether the control algorithm of the valves, blocking the natural fluid flow in the passively-controlled tank is physically realisable, in other words, whether it has the ability to operate in real sea conditions,

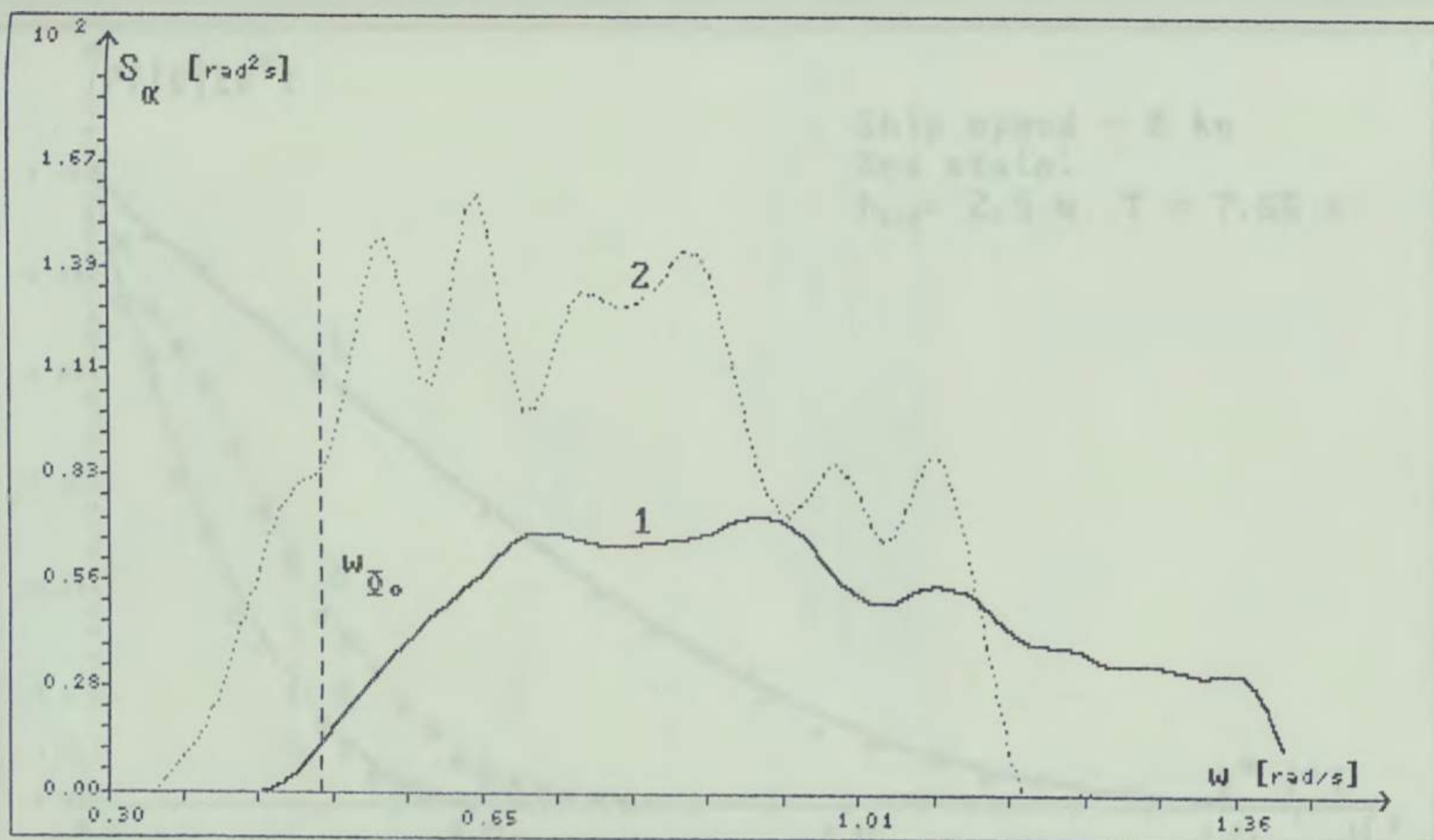

Fig.8.40. The wave slope spectra of irregular beam waves used as an external disturbance during numerical simulation tests with the ship roll motion.

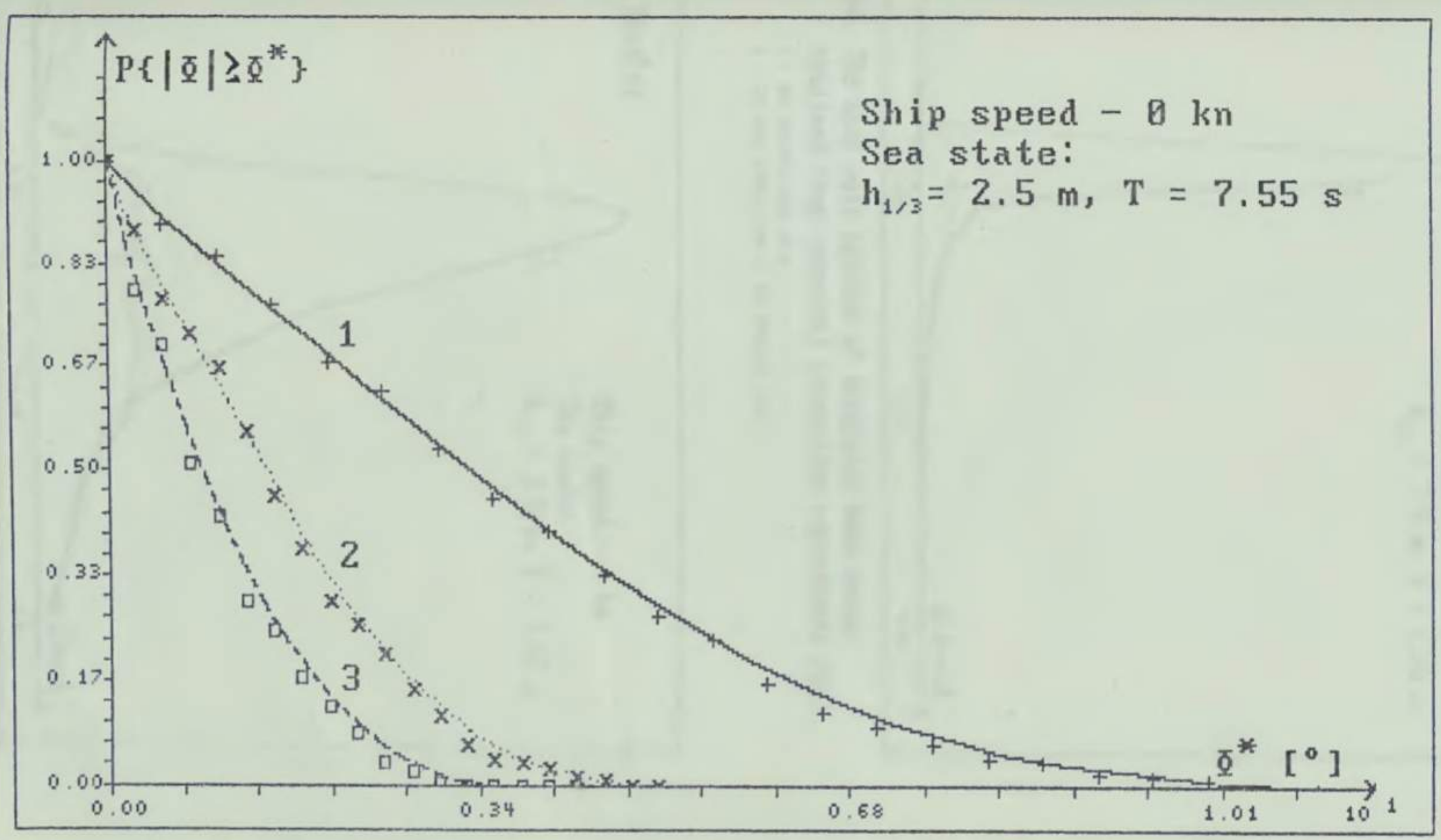

Fig.8.41. Probability of exceeding an angle Φ^* in irregular beam waves, obtained from numerical simulation experiments for:

1 - the unstabilized ship,

2 - the ship stabilized by the passive tank,

3 - the ship stabilized by the passively-controlled tank.

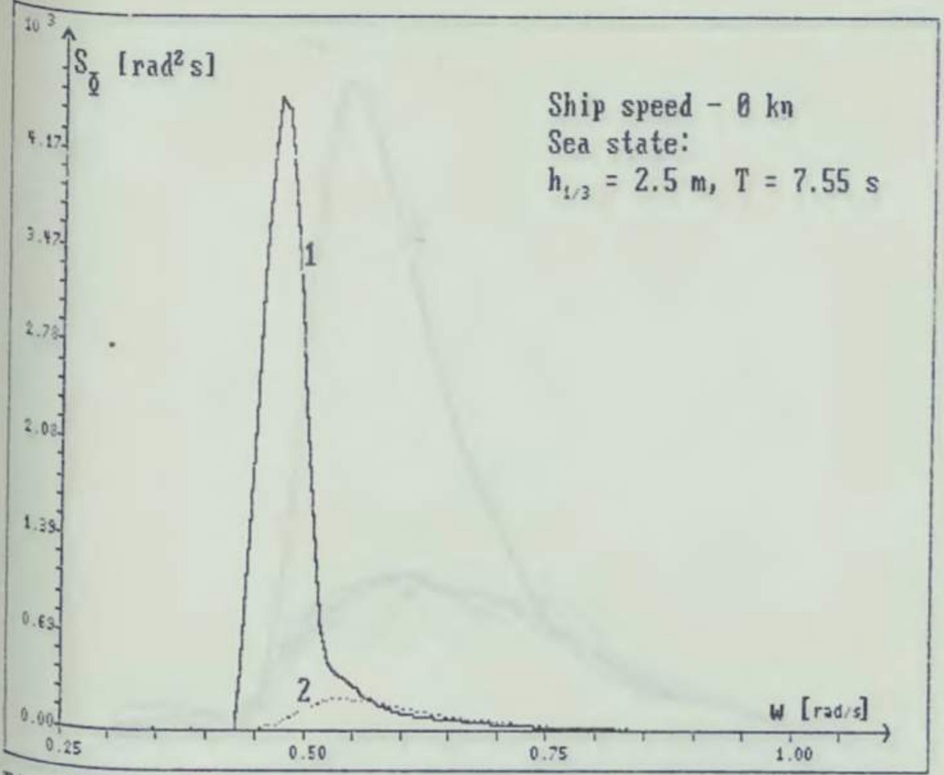

Fig.8.42a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the unstabilized ship,
2 - the ship stabilized by the passive tank.

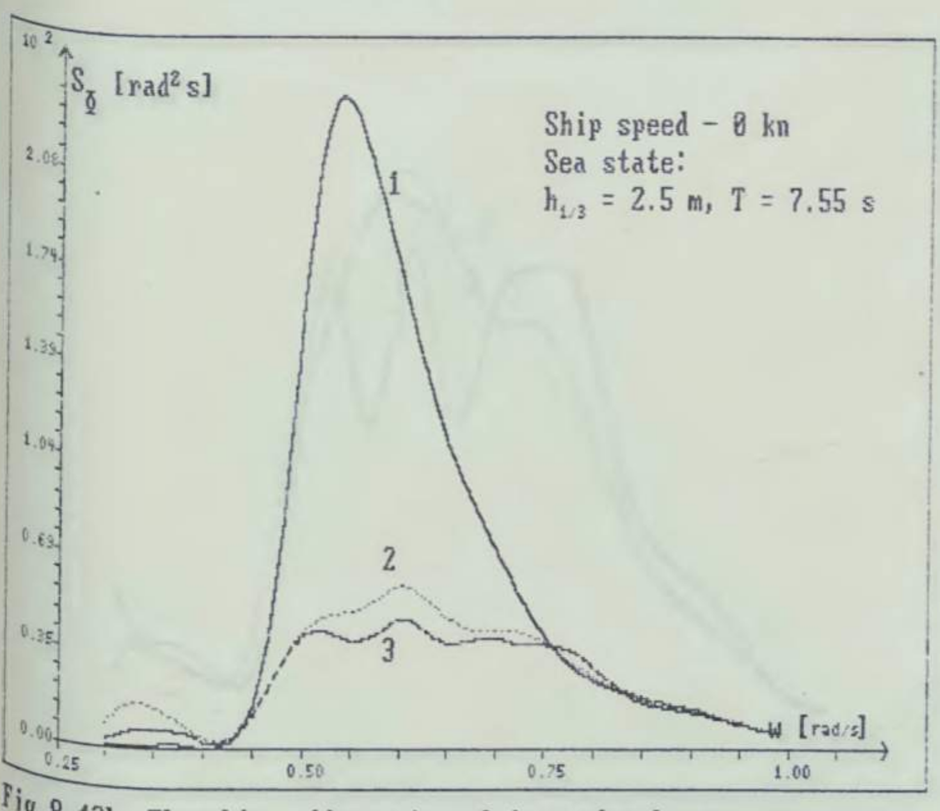

Fig.8.42b. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank (isothermic process),
3 - the ship stabilized by the passively-controlled tank (adiabatic process).

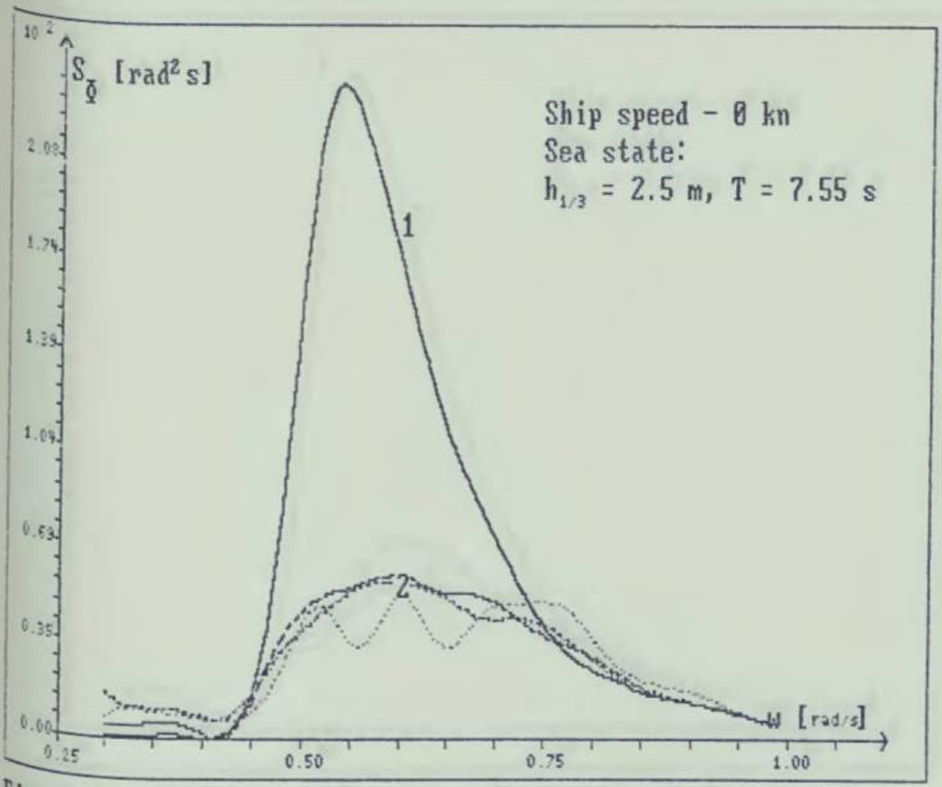

Fig.8.43a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank with the PDD² controller.

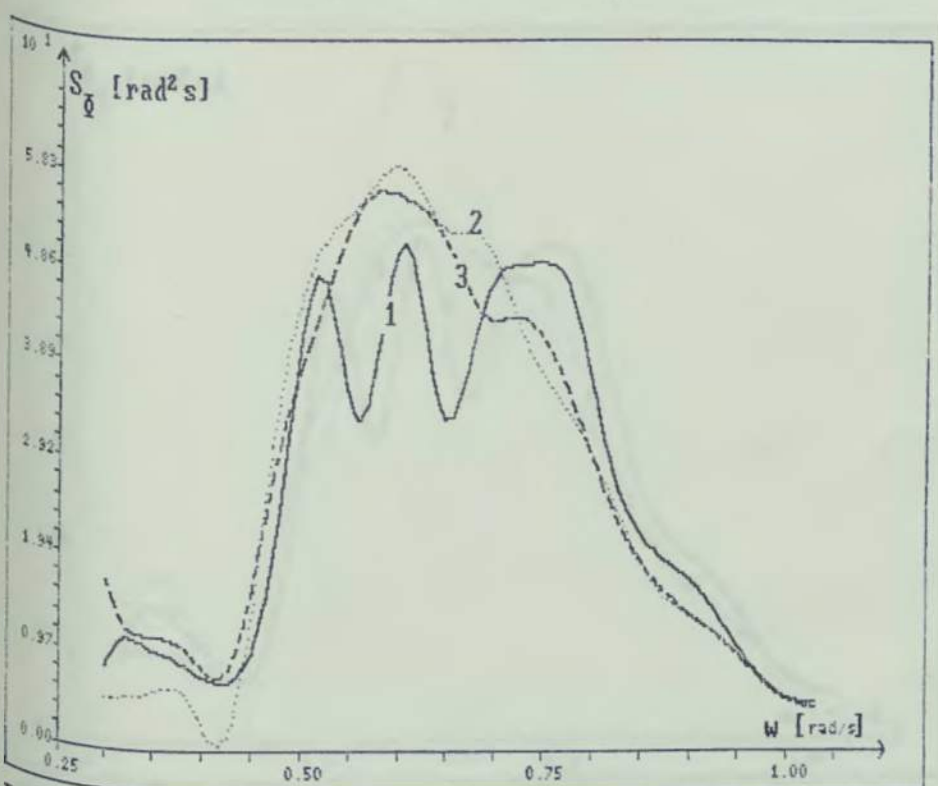

Fig.8.43b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.43a:

1 - without compressibility of the air, 2 - isothermic process,
3 - adiabatic process.

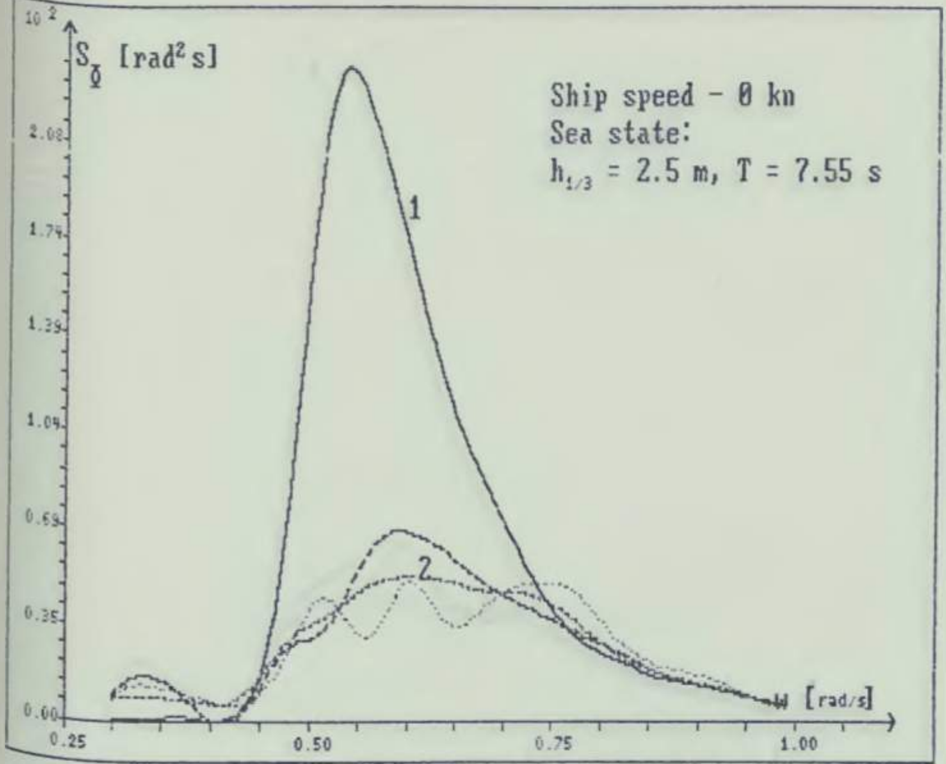

Fig.8.44a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,

2 - the ship stabilized by the passively-controlled tank with the PDD^2 controller and the Butterworth's filter.

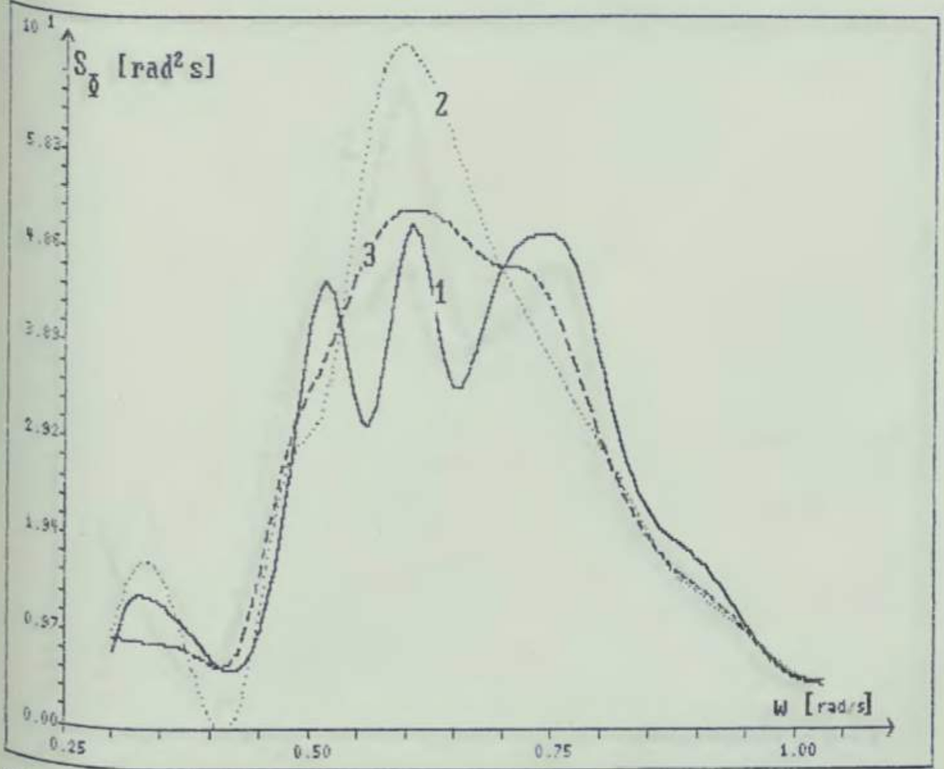

Fig.8.44b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.44a:

1 - without compressibility of the air, 2 - isothermic process,

3 - adiabatic process.

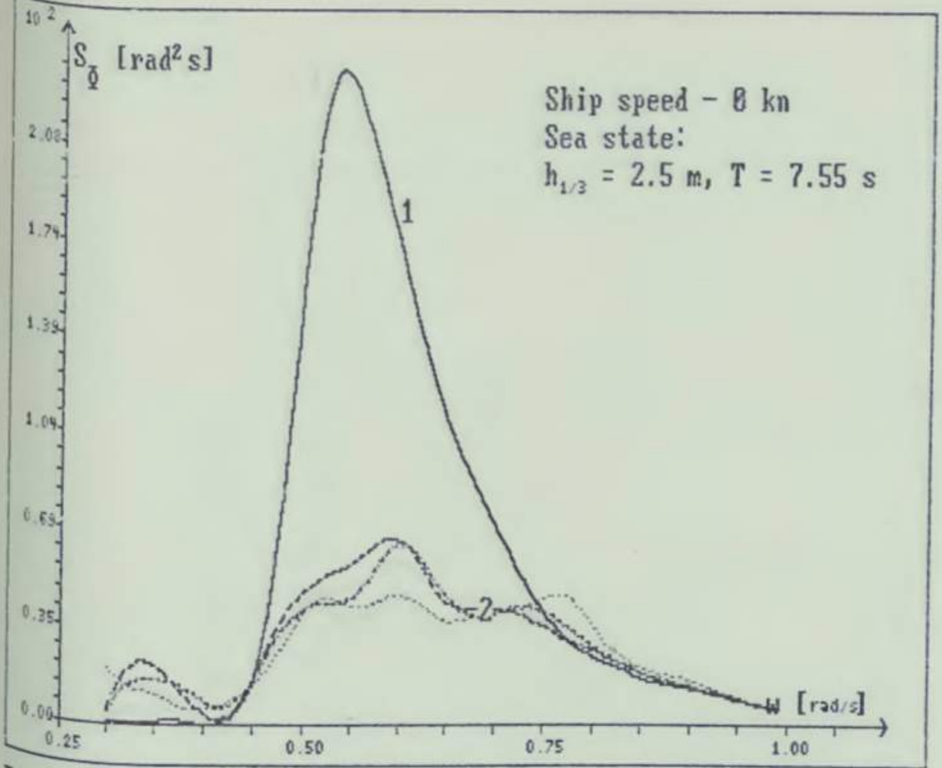

Fig.8.45a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank with the n-step Kalman's predictor.

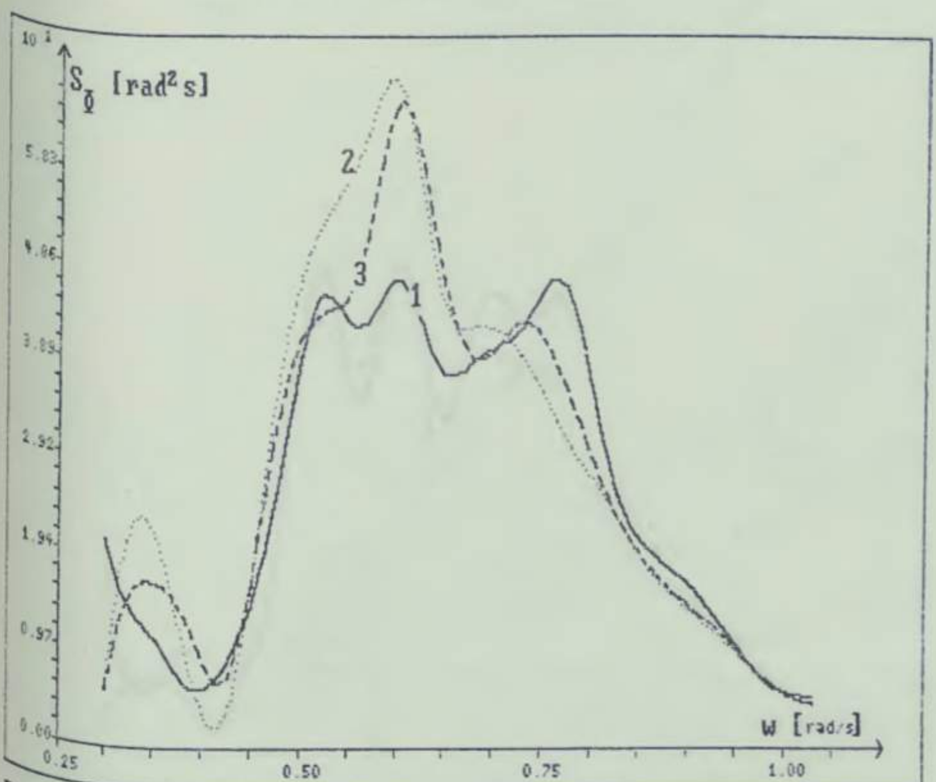

Fig.8.45b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.45a:

1 - without compressibility of the air, 2 - isothermic process,
3 - adiabatic process.

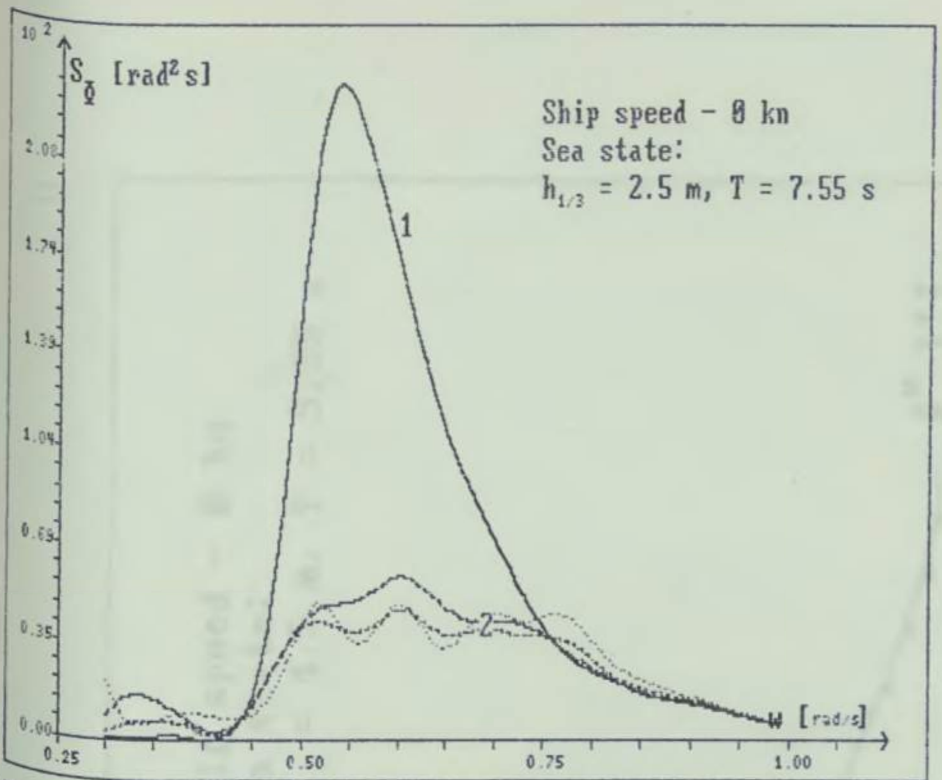

Fig.8.46a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank with the optimal controller and Kalman filter.

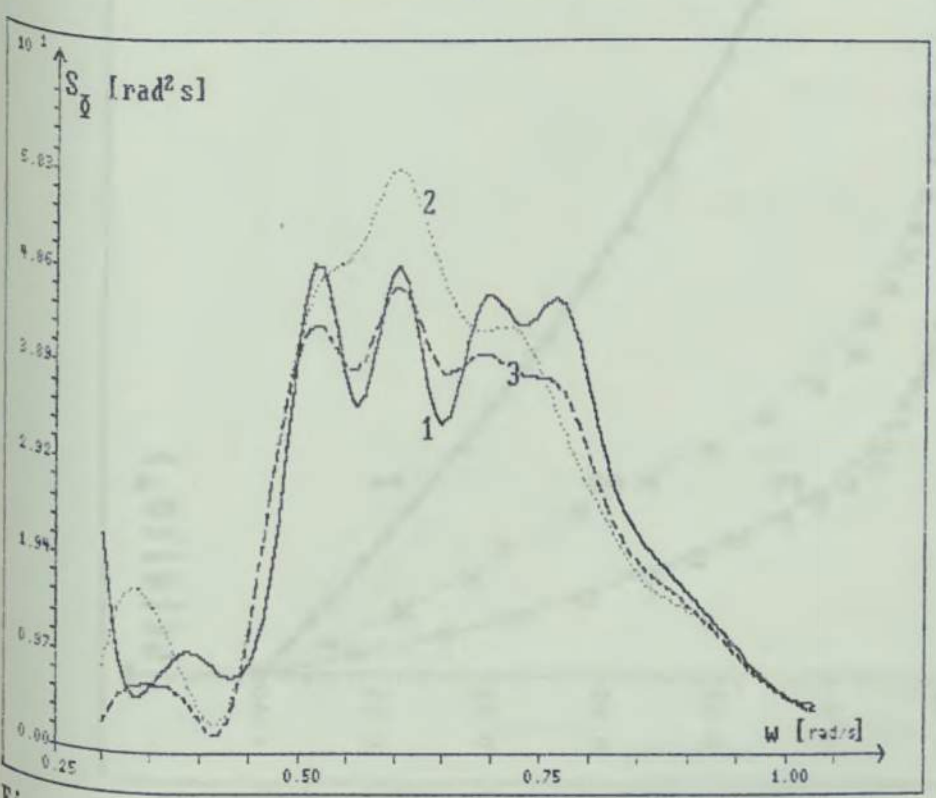

Fig.8.46b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.46a:

1 - without compressibility of the air, 2 - isothermic process,
3 - adiabatic process.

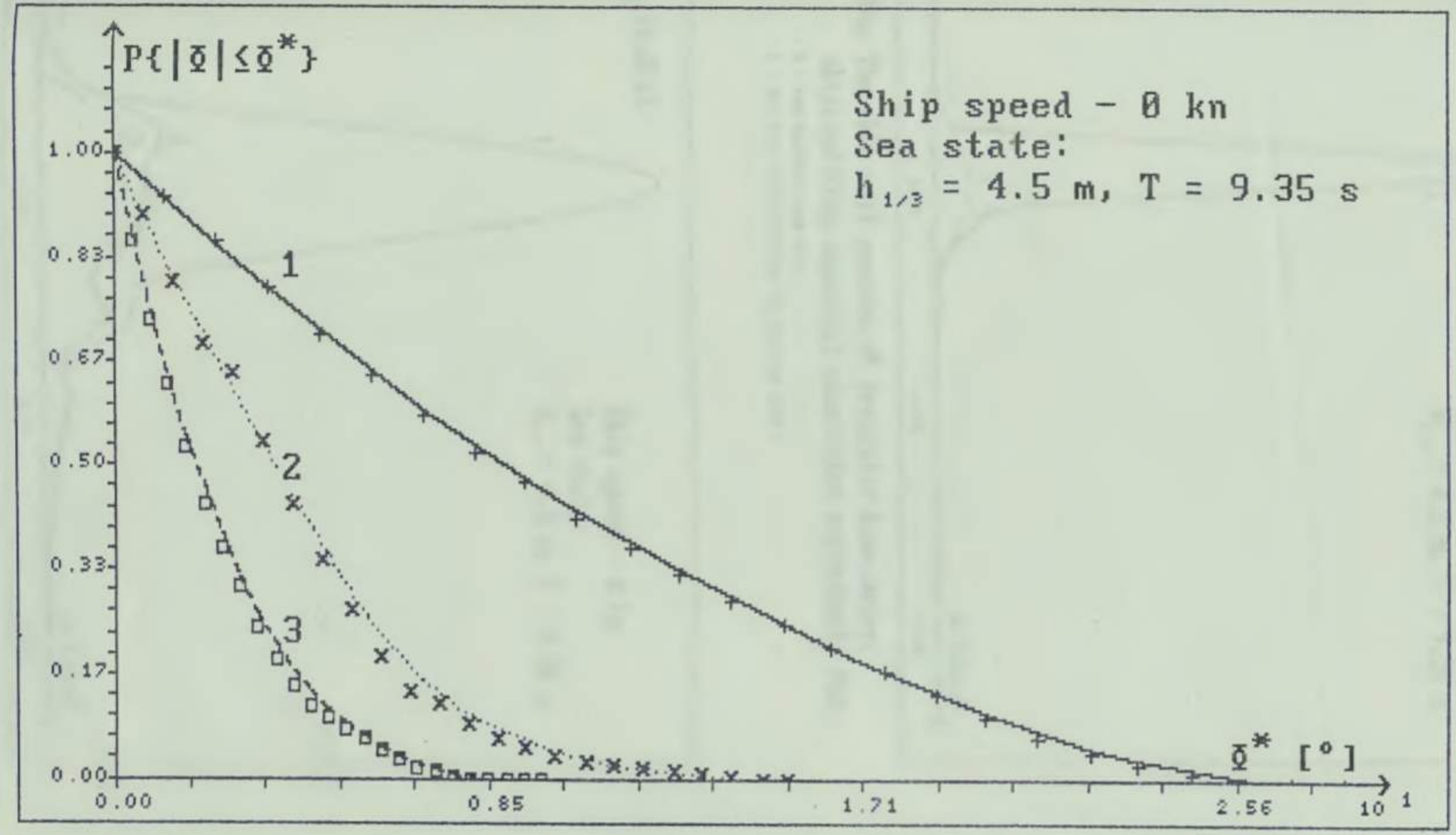

Fig.8.47. Probability of exceeding an angle Φ in irregular beam waves, obtained from numerical simulation experiments for:

1 - the unstabilized ship,

2 - the ship stabilized by the passive tank,

3 - the ship stabilized by the passively-controlled tank.

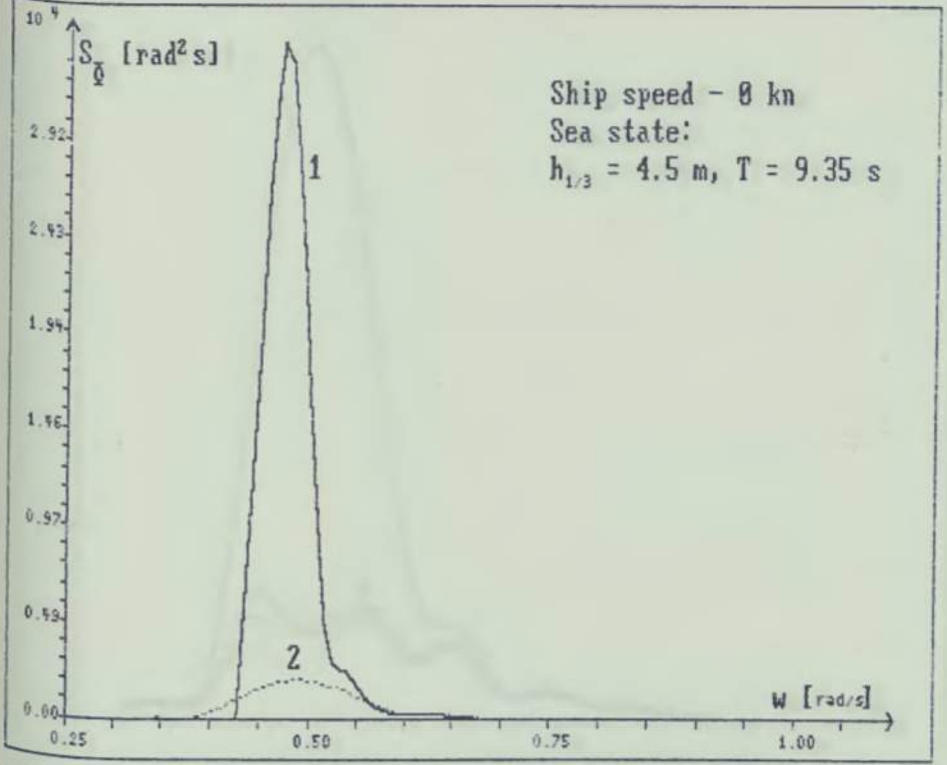

Fig.8.48a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the unstabilized ship,
2 - the ship stabilized by the passive tank.

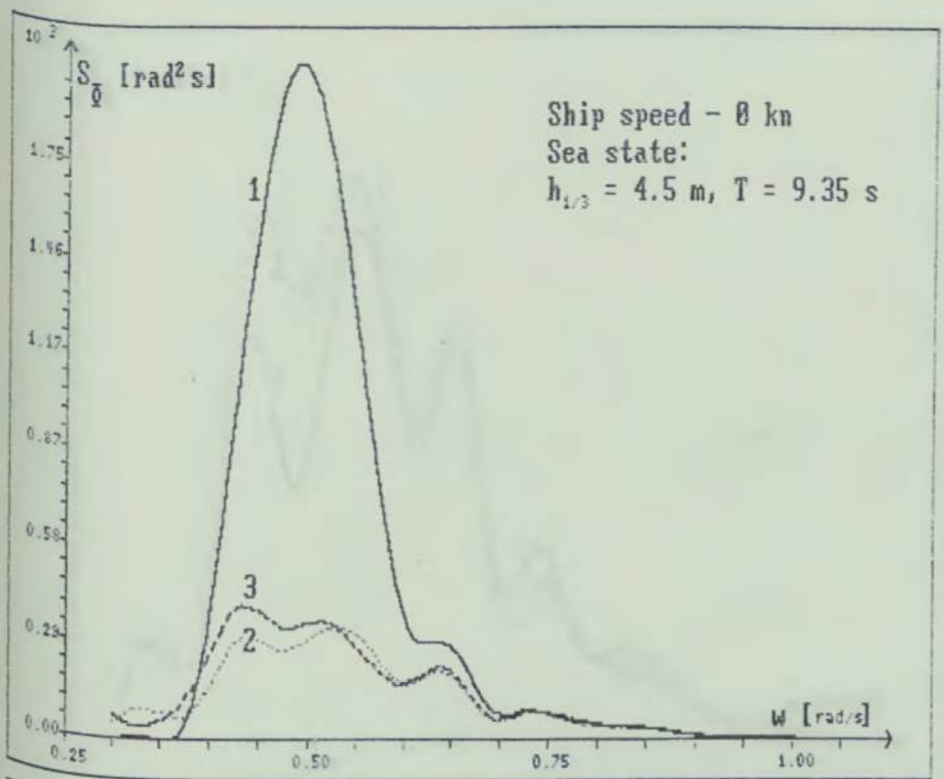

Fig.8.48b. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank (isothermic process),
3 - the ship stabilized by the passively-controlled tank (adiabatic process).

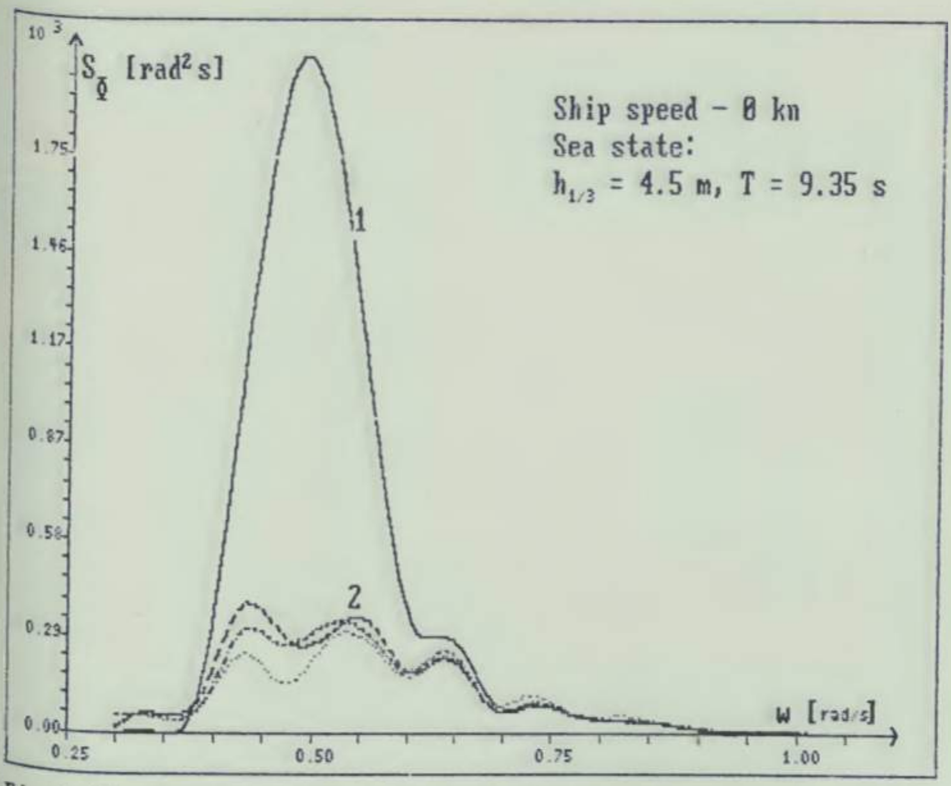

Fig.8.49a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank with the PDD^2 controller.

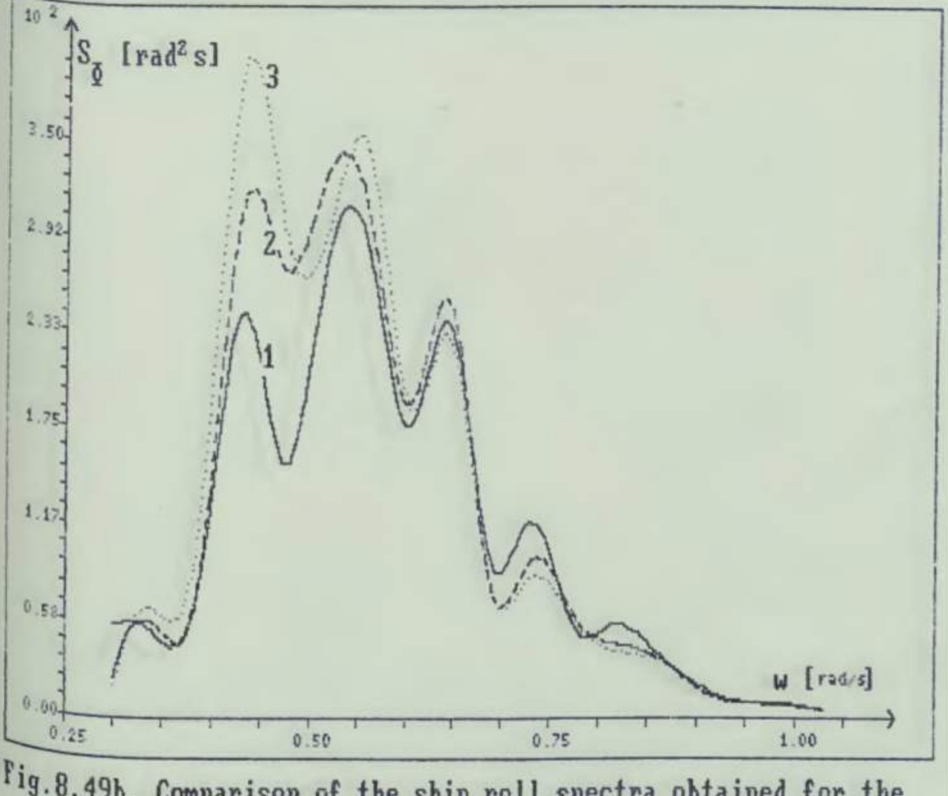

Fig.8.49b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.49a:

1 - without compressibility of the air, 2 - isothermic process,
3 - adiabatic process.

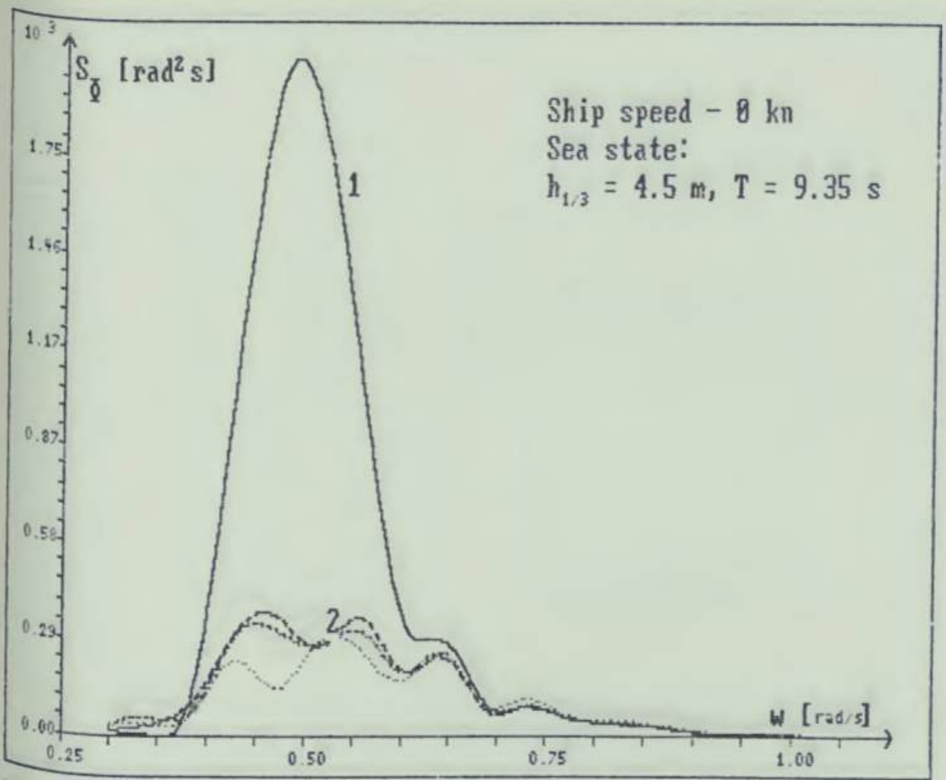

Fig.8.50a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,

2 - the ship stabilized by the passively-controlled tank with the PDD^2 controller and the Butterworth's filter.

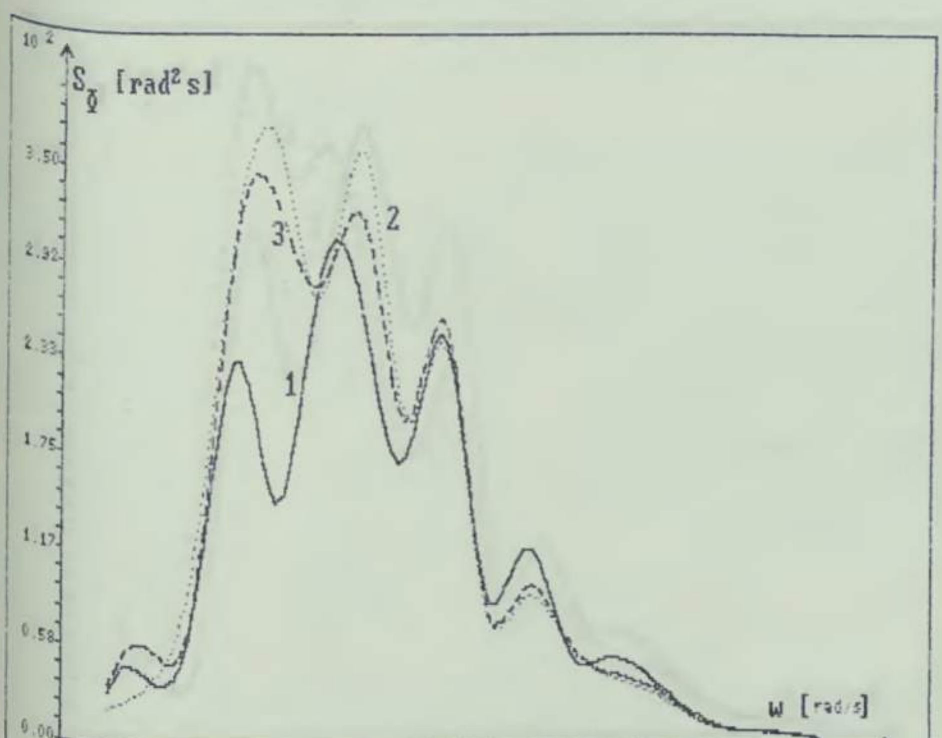

Fig.8.50b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.50a:

1 - without compressibility of the air, 2 - isothermic process,
3 - adiabatic process.

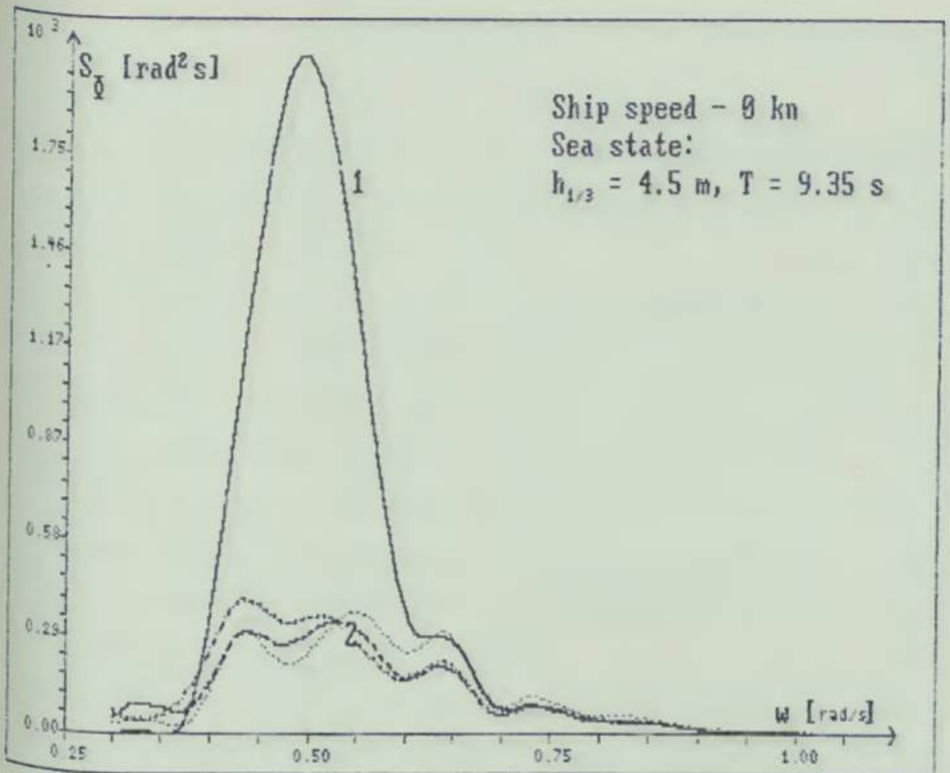

Fig.8.51a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank with the n-step Kalman's predictor.

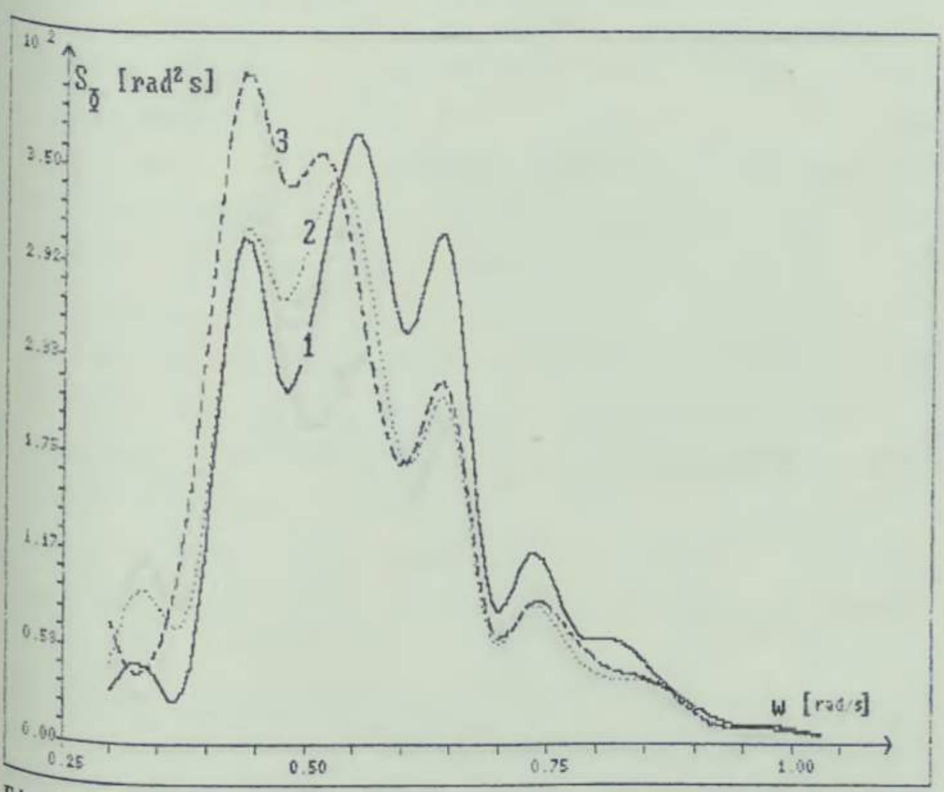

Fig.8.51b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.51a:

1 - without compressibility of the air, 2 - isothermic process,
3 - adiabatic process.

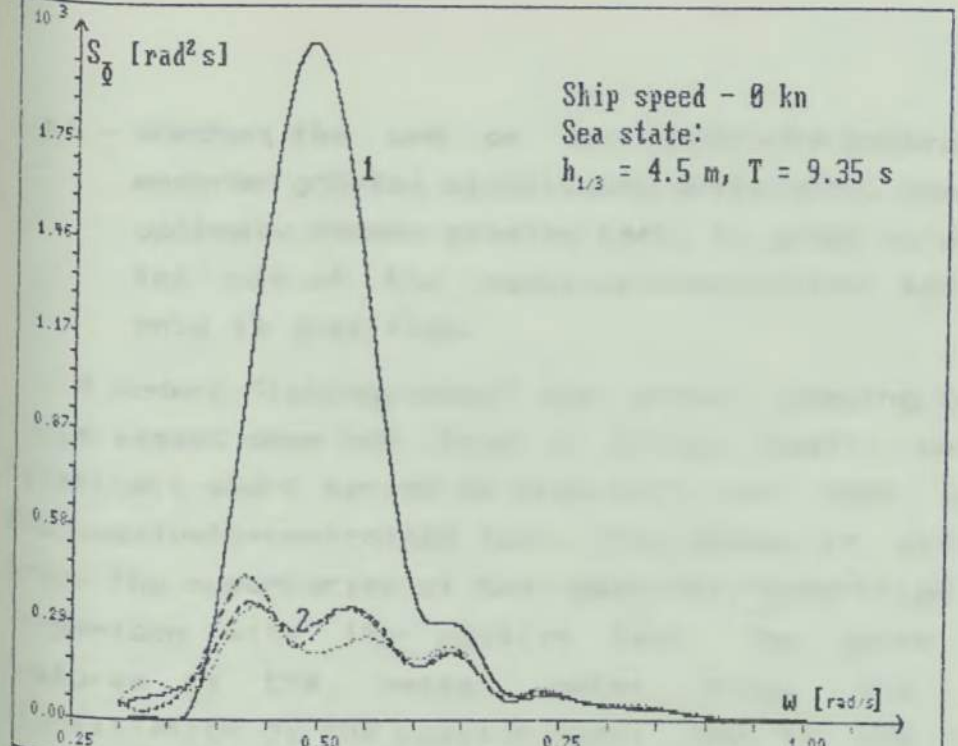

Fig.8.52a. The ship roll spectra of irregular beam waves, obtained from numerical simulation experiments for:

1 - the ship stabilized by the passive tank,
2 - the ship stabilized by the passively-controlled tank with the optimal controller and Kalman filter.

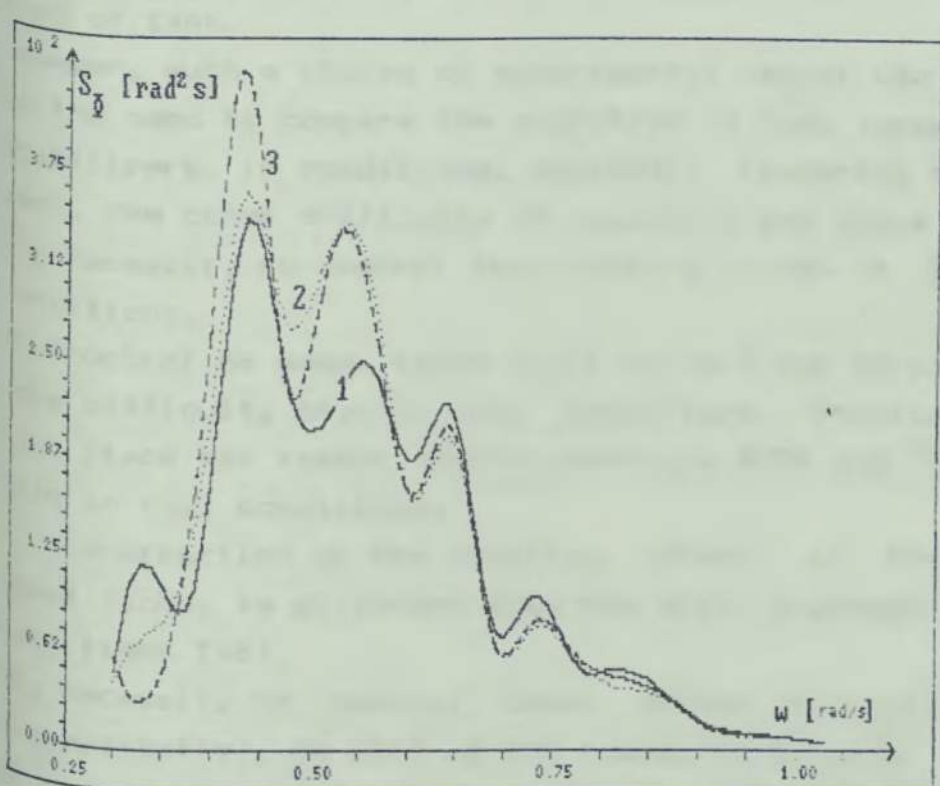

Fig.8.52b. Comparison of the ship roll spectra obtained for the ship stabilized by the passively-controlled tank for the conditions given in Fig.8.52a:

1 - without compressibility of the air, 2 - isothermic process,
3 - adiabatic process.

(ii) - whether the use of the passively-controlled tank ensures greater stabilizing efficiency, compared to a optimaly chosen passive tank, in other words, whether the use of the passively-controlled tank onboard ship is justified.

A modern fishing vessel with a small damping coefficient (this vessel does not have a bildge keel), and with a relatively short period of free roll, was used in testing the passively-controlled tank. This makes it difficult to prove the superiority of the passively-controlled tank in comparison with the passive tank. The above mentioned features of the vessel makes easy, its effective stabilization by the passive tank, due to the relatively high amplitudes of stabilizing fluid motion, in the passive tank. This effect does not greately influence the operative efficiency of the passively-controlled tank, as the control of the blocking valves decides the proper operation of this type of tank.

However, such a choice of experimental vessel was made due to the need to compare the operation of both types of tank stabilizers, in conditions, decidedly favouring the passive tank. The other difficulty in realizing the above tests was the necessity to control the blocking valves in model test conditions.

The control of model tanks built in 16.3 and 28 scale is far more difficult, than in real conditions. Physical process take place far faster (correspondingly 4.04 and 5.3 times) than in real conditions.

Construction of the blocking valves, in the case of model tanks, is different from the ones proposed for real conditions [46].

The necessity to control these valves electrically (not pneumatically), as well as the necessity to have very much shorter opening and closing times of the blocking valves, caused inaccuracy in the stabilizing fluid's blockage

control.
The above mentioned problems caused the decrease in the efficiency of the passively-controlled tank's operation during model tests, and it's not possible to correctly determine the decrease analytically.

In real conditions (full scale), the measurement - control algorithm loop time is 0.25 seconds. During bench tests, the loop time was 0.06 seconds and during tests on physical model, this time was 0.04 seconds. This prooves the great difficulties in controlling and building the passively-controlled tank models.

Computer and physical simulations of the stabilization system using the passively-controlled tank, are the last stages of this research study and verification of the operative possibilities of the proposed solution.

The passively-controlled tank is far more effective in stabilizing the ship's roll in regular and irregular waves, in comparison with the optimally tuned passive tank. This was confirmed in this study, presenting results of experiments. A great adventage of the passively-controlled tank is the fact, that it stabilizes the ship's roll in the full range of possible frequencies. Therefore, this tank does not cause destabilization, which is a great defect of the passive tank.

The influence of air compressibility was examined only during computer simulations. The stabilizing moment, generated by the tank, is decreased when air compressibility is taken into account, especially during regular excitation. During irregular excitation, this does not cause any essential changes in the value of $\bar{\Phi}_{A1/3}$, as well as in the energy spectra of the stabilized ship. The greatest loss of the stabilizing moment, due to air compressibility, was observed in the near resonance band. In the range of low and high frequencies of regular excitation, the air compressibility has lesser influence.

This can be explained by analysing the characteristics shown in fig. 8.8 to 8.19. During low and high frequencies, the passively-controlled tank, operates with relatively small angles of the stabilizing fluid's motion, and the loss in the tank's moment due to compression (decompression) is small. In regular excitation, by near resonance waves, the passively-controlled tank operates with greater angles of the stabilizing fluid's motion . Therefore, the decrease of the tank's moment is much larger.

An attempt at finding the most effective control algorithm for the passively-controlled tank, was made during simulation experiments on the irregular wave. The results of these tests shown in table 8.8 proove, that the problem of choosing the regulator (prediction structure) is not easily solved. It appears, that all the proposed algorithms practically give the same value of $\overline{\phi}_{A1/3}$ (for lower waves the maximum difference is 8%, for higher waves - 5.5%).

This is due to proper prediction structures and their coefficients, as well as, due to the specifics of the system ship - passively-controlled tank. The algorithm, simple in realisation (algorithm PDD^2 and PDD^2 with Butterworth's filter) ensures proper stabilization, but is, however, very sensitive to changes in natural frequencies, and in the ship's roll damping, during the ship's operation. Complicated algorithms, based on optimal filtration are more difficult in realisation, but are less sensitive to measurement noises and are self-tuning.

The next important problem is the inter-relation between results, obtained from model tests and results obtained from computer simulations.

Fig. 8.31 and 8.32 show the comparison of the above mentioned results. The results obtained for the passive tank are very compatible (better than 10%). Such comparison for the passively-controlled tank, does not show such compatibility, but due to difficulties in conducting these experiments on scale models the compatibility achieved (better than 22%) is satisfactory.

In the case of irregular waves, it is difficult to talk about result comparison, because the excitation (as described earlier) during model tests was not similar to excitation during computer simulation. The analysis of ship's roll reduction in both the above mentioned cases gives a certain possibility of comparison.

According to tables 8.6 and 8.7, the efficiency of the passively-controlled tank for the stationary ship, when $h_{1/3} = 2.5m$ is:

- for model tests - 62.2%,
- for computer simulations - 69.2% (the average from table 8.7).

Therefore, the results obtained from tests conducted according to the two test methods, are practically the same. This allows us to assume, that the numerical simulation and physical modelling were carried out properly.

The discussion above, and the presented results, allow us to certify, that the testing of the system ship - passively-controlled tank was carried out properly and the achieved results are correct.

9. CONCLUSIONS AND FINAL REMARKS.

The presented thesis is an attempt to completely solve the problems connected with the application of passively-controlled tanks in stabilizing the ship's roll motion. World literature, gives us only fragments of information, published by producers of such equipment. There is no complete study of this theme as yet.

This study examines a proposed mathematical model which describes the dynamic system ship - passively-controlled tank (Chapter 3). The system's mathematical model, other than the dynamic relations, analyzes phenomena of the air compressibility in the tank, when the valves are closed. The control description of the tank is an integral part of the mathematical model. This problem is described in chapter 5, in which we analyzed a range of control algorithms of the passively-controlled tank, which are physically realizable.

Chapter 7 contains a proposition of new methodics for the bench tests of passively-controlled tanks, as well as a method for processing the results obtained in this way. The material in this chapter is especially useful, because the proposed methodics ensures possibility of directly comparing the test results of this tank with the results obtained from traditional testing of the passive tank.

Chapter 8 includes the results of all tests (computer simulation and model tests). Results obtained from both tests methods were found to be comparable.

This study demanded the solution of a range of problems in the field of the automatic control theory (for fluid flow blockage), and certain hydrodynamical problems which are connected with the operation of the passively-controlled tank.

The material presented in this study, solves the synthesis of the system ship - passively-controlled tank in a satisfactory way. Certain difficulties were found in analytically determining the conditions of system stability. This problem was solved in chapter 6, through analysis of the technical possibilities in ensuring the system's stable operation.

The analysis of the material contained in the study (theory and results of tests), allows us to formulate the following conclusions about the operation of the passively-controlled tank.

The main advantages of the passively-controlled tank, as a ship's roll stabilizer, in comparison with the passive tank (optimally designed) are:

- greater effectiveness of roll stabilization for the same wave slope capacity,
- lesser reduction of the ship's initial metacentric height; therefore, the passively-controlled tank can be used in wide range of different loading conditions of the ship,
- possibility of eliminating the free surface effect in the case of any dangerous situation,
- possibility of the control algorithm to self tune to changing external disturbances and to different loading conditions,
- possibility of using the passively-controlled tank in anti-heeling operations.

These conclusions show us that the passively-controlled tank is a particularly useful ship's roll stabilizer for ships with changing (through a wide range) loading conditions.

It is necessary to emphasize the value of the presented methodics of system synthesis, due to the possibility of its practical application. A whole family of digital simulators which make possible the full investigation of the effects of the installation of the passively-controlled and passive tank on any given hull was designed. A wide range of computer programmes used in analyzing the results, of model tests of the separated tanks (passive and passively-controlled) were also designed.

The methodics described in this study was used by the author practically in the shipyard. Comparison projects, which had as their aim the determination of the possibilities of ship's roll stabilization, using passively-controlled or passive tanks, were carried out for three specialized ships built by the Szczecin Shipyard. In two cases (a passenger-car ferry and geophysical ship) it was decided to use passive stabilizers with control elements (blocking values so as to quickly eliminate, the free surface effect in case of any dangerous situation). In the third case, inspite of the decidedly better efficiency of the passively-controlled tank (achieved through computer aided analysis), it was decided not to use this type of tank only due to the fact, that the stabilizing fluid was to be fuel. In the case of the application of fuel as a stabilizing fluid, the electrical measurement system of the fluid motion could not be used.

Several problems connected with the theme of this study was only signalized or ignored (as being out of the study's competancy).

The following important problems were omitted.

1. Synthesis of the special algorithms of control with self-tuning to changing external disturbances, as well as to changing dynamic parameters of the ship; during the realization of this study tests were conducted for one chosen loading condition of ship; the change of external conditions caused the long range search for optimal (suboptimal) settings of the regulator. This shows the necessity for controlling the adaptation.

2. Elaboration of the monitoring algorithms and accident blockade algorithms and analysis of all dangerous situations.

REFERENCES

[1] A Computer Program System for Theoretical Calculation of Sea-keeping Quality of Ships. In: Japan Shipbuilding and Marine Engineering. 3/1975.

[2] Abkowitz M.A., Vassilopoulos L.A., Sellars F.H.: Recent Developments in Sea-keeping Research and Its Application to Design. In: Transactions SNAME. Vol. 74/1966.

[3] Barr R.A., Ankudinov V.: Ship Rolling, Its Prediction and Reduction Using Roll Stabilization. In: Marine Technology. 1/1977.

[4] Bell J., Walker W.: Activited and Passive Controlled Fluid Tank System for Ship Stabilization. Part I i II. In: Transactions SNAME. Vol. 74/1966.

[5] Borodaj I.K., Necvetaev J.A.: Kačka sudov na morskom volnenii. Leningrad, Sudostroenie, 1969.

[6] Bosh J.J., van den Vugts J.H.: On Roll Damping by Free - surface Tanks. In: Transactions RINA. Vol. 108/1966.

[7] Brammer K., Siffling G.: Kalman-Bucy-Filter. Deterministische Beobachtung und Stochasische Filterung. München, R. Oldenburg Vlg, 1982. Translation into Russian. Nauka. Moscow, 1982.

[8] Brunsell P.A., Field S.B.: Correlation of Full Scale and Model Basin Stabilization Test Results for the Canadian Weatherships. In: Transactions RINA. Vol. 111/1969.

[9] Conolly J.E.: Rolling and Stabilization by Active Fins. In: Transactions RINA. Vol. 111/1969.

[10] Cox G.G., Lloyd A.R.: Hydrodynamic Design Basis for Navy Ship Roll Motion Stabilization. In: Transactions SNAME. Vol. 85/1977.

[11] Dalzell J.F.: A Note on the Form of Ship Roll Damping. In: Journal of Ship Research. Vol.22/1978.

[12] Dudziak J.: Okręt na fali. Gdańsk, Wydawnictwo Morskie, 1980.

[13] Dudziak J.: Teoria i projektowanie urządzeń stabilizujących kołysania boczne statku. Report of A. Warski's Szczecin Shipyard. Szczecin, 1978.

[14] Dudziak J.: Teoria okrętu. Gdańsk, Wydawnictwo Morskie, 1988.

[15] Dudziak J., Pawlik A.: Porównanie współczynników tłumienia kołysań bocznych wyznaczonych doświadczalnie i metodami przybliżonymi. Report of Ship Design and Research Centre No RK-80/R-059. Gdańsk, 1980.

[16] Elbert T.F.: Estimation and Control of Systems. New York, Van Nostrand Reinhold Co., 1984.

[17] Field S.B., Martin J.P.: Comparative Effects of U-tube and Free Surface Type Passive Roll Stabilization Systems. In: Transactions RINA. Vol. 118/1976.

[18] Folding Fin Stabilization. Glasgow, Brown-Brothers - Vickers, [1987]. Product information.

[19] Free Area Anti - Rolling Passive Stabilizer. Le Havre, SNACH, 1974. Product information.

[20] Goodrich G.J.: Development and Design of Passive Roll Stabilizers. In: Transactions RINA. Vol. 111/1969.

[21] Gutenbaum J.: Problemy teorii regulatorów. Warszawa, WNT, 1975.

[22] HEELMASTER. Helsinki, H.G. Honkanen, [1987]. Product information.

[23] Hirano M.: On the Equations of Motion of Ships equipped with an Anti-rolling Tank in Beam Seas. In: Japan Shipbuilding and Marine Engineering. Vol. 7/1973.

[24] Honkanen M.: Tests and Design Data of Tank Stabilizers of the Free-surface type. Report of Structural Laboratory. Helsinki, 1971.

[25] Hueckel S.: Zarys hydrotechniki morskiej. Gdańsk, Wydawnictwo Morskie, 1976.

[26] Ikeda Y., Himeno Y., Tanaka N.: A Prediction Method for Ship Roll Damping. Report of the University of Osaka No 00405, 1978.

[27] Jarosz A.: Okrętowe baseny modelowe. Gdańsk, Wydawnictwo Morskie, 1977.

[28] Kudrewicz J.: Częstotliwościowe metody w teorii nieliniowych układów dynamicznych. Warszawa, WNT, 1970.

[29] Kure K., Katelsen H., Jensen V.: The ultimate half roll before Capsise on the analog Computer. Paper from XI Symposion Naval Hydrodynamics. London, 1976.

[30] Lewison G.R.: Optimum Design of Passive Roll Stabilizers Tanks. In: Transactions RINA. Vol. 118/1976.

[31] Mazurek P., Rozenberg L.: Stabilizacja kołysań bocznych jednostek oceanotechnicznych. Report of Ship Research Institute Technical University of Szczecin, No 81/A/80.

[32] Mazurek P., Rozenberg L.: Stabilizacja kołysań bocznych statku wyposażonego w zbiorniki aktywizowane. In: Budownictwo Okrętowe. Gdańsk, 3/1981.

[33] Melsa J.L., Jones S.K.: Computer Programs for Computational Assistance in the Study of Linear Control Theory. New York, McGraw Hill, 1973. Translation into Russian, Mašinostroenie, Moscow 1981.

[34] Nowak J., Zdybek T.: Próby modelowe odosobnionego zbiornika stabilizacyjnego. Report of Ship Design and Research Centre No RH-86/Z-150. Gdańsk, 1986. (The Paper in Service Secret)

[35] Otnes R.K., Enochson L.: Analiza numeryczna szeregów czasowych. Warszawa, WNT, 1978.

[36] Phase Sensing System. [Hamburg], Flume Stabilization System, 1979. Product information.

[37] Rorke J.: Ship Stabilizers - Fins and Tanks. Edinburgh, Brown - Brothers Ltd. Co., 1973. Product information.

[38] Rorke J., Volpich H.: Selection of a Ship's Stabilizer. In: Transactions IME. Vol. 82/1970.

[39] Rozenberg L.: Sposób i urządzenie do stabilizowania kołysań zwłaszcza bocznych statku. Patent of Polish People's Republic No 132925. Warszawa 1987.

[40] Rozenberg L.: Stabilizacja kołysań bocznych jednostek oceanotechnicznych. Report of Ship Research Institute Technical University of Szczecin, No 91/A/81.

[41] Rozenberg L.: Stabilizacja kołysań bocznych statku ze zbiornikiem aktywizowanym. In: Budownictwo Okrętowe. Gdańsk, 4/1983.

[42] Rozenberg L.: Stabilizator kołysań statku. Patent of Polish People's Republic No 130893. Warszawa 1986.

[43] Rozenberg L.: Synteza systemu stabilizacji kołysań bocznych statku przy wykorzystaniu zbiornika aktywizowanego. Paper read in IV Conference "Cybernetics in the Marine Industry". Sopot, September 1983.

[44] Rozenberg L., Dumański J., Macuk D.: Metoda oceny celowości stosowania zbiorników kontrolowanych do tłumienia kołysań bocznych oraz model układu statek - zbiornik. Report of Ship Research Institute Technical University of Szczecin, No 259/A/88.

[45] Rozenberg L., Dumański J., Macuk D.: Metoda oceny celowości stosowania zbiorników kontrolowanych do tłumienia kołysań bocznych oraz model układu statek - zbiornik. Report of Ship Research Institute Technical University of Szczecin, No 262/A/89.

[46] Rozenberg L., Parczewski S., Pejaś J.: Performance analysis for ship roll stabilizers; simulation experiments. Paper read in Internationales Rostocker Schiffstechnisches Symposium, Rostock, W. Pieck - Universität, 1987.

[47] Rozenberg L., Pejaś J.: Zasada sterowania zbiornika kontrolowanego. Obliczenie krótkoterminowych prognoz kołysań bocznych statku niestabilizowanego i stabilizowanego. Report of Ship Research Institute Technical University of Szczecin, No 204/A/87.

[48] Rozenberg L., Wandrey P.: Zawór bezpośredniego działania. Patent of Polish People's Republic No 135270. Warszawa 1987.

[49] Rozenberg L., Zdybek T.: Stabilizacja kołysań bocznych statku. Stan obecny i perspektywy. Paper read in XIII Scientific Session of Ship Designers, Gdańsk, May 1988. Book 1.

[50] Schmitke R.T.: Ship Sway, Roll, and Yaw Motions in Oblique Seas. In: Transactions SNAME. Vol. 86/1978.

[51] Ship Roll Damping and Anti - Heeling. Glasgow, Brown - Brothers - Vickers, [1987]. Product information.

[52] Stabilizer and Anti - Heeling System. Hamburg, INTERING GmbH, 1984. Product information.

[53] Stabilizer and Anti - Heeling Systems Combined. In: The Naval Architect. 3/1979.

[54] Stigter C.: The Performance of U - tanks as a Passive Anti - Rolling Device. In: International Shipbuilding Progress. Vol. 13/1966.

[55] Šmyrev A.N., Morenšildt W.A., Ilina S.G.: Uspokoiteli kački sudov. Leningrad, Sudostroenie, 1972.

[56] Teoria pomiarów. Praca zbiorowa pod red. Henryka Szydłowskiego. Warszawa, PWN, 1978.

[57] Lerner D.M., Łukomski J.A.: Upravlenie morskimi podvižnymi obiektami. Leningrad, Sudostroenie, 1979.

[58] Vugts J.H.: A comparative Study on four Different Passive Roll Damping Tanks. In: International Shipbuilding Progress. Vol. 16/1969.

[59] Watanabe Y.: About the Frahm's Anti - rolling Tank with Auxiliary Tanks. In: JSNA. 31/1966.

[60] Webster W.: Analysis of the Control of Activated Antiroll Tanks. In: Transactions SNAME. Vol. 75/1967.

[61] Zdybek T.: Bierne zbiorniki stabilizacyjne. Gdańsk, Wydawnictwo Morskie, 1982.

[62] Zdybek T.: Charakterystyki eksploatacyjne kołysań bocznych i ich wyznaczanie dla danego modelu eksploatacji statku. In: Budownictwo Okrętowe. Gdańsk, 4/1972.

[63] Zdybek T.: Charakterystyki kołysania bocznego i ich zastosowanie do oceny prawdopodobnego zachowania się statku w rzeczywistych warunkach morskich. In: Budownictwo Okrętowe. Gdańsk, 10/1979.

[64] Zdybek T.: Stabilization Tank Model Tests. In: Budownictwo Okrętowe. Gdańsk, 4/1979.

[65] Zdybek T.: The Use of Bench Test Results for Calculating Roll Response of the Tank Stabilized Ships. In: International Shipbuilding Progress. April 1980.

[66] Zierep J.: Kryteria podobieństwa i zasady modelowania w mechanice płynów. Warszawa, PWN, 1978.

Appendix 1

Equations of roll about the fixed axis, for a ship with a U-tube tank [65].

The set of differential equation (3.13) was derided by using a method almost identical to that given in [55]. Only the angle of the stabilizing liquid effective slope ϑ was introduced instead of the coordinate of the (instantaneous) water level position in the wing section of a tank z.

The viscosity coupling between stabilizing liquid and the ship was neglected.

The natural frequency of the tank is expressed as:

$$\vartheta_o = \sqrt{\frac{g}{\overline{l}_t}} \qquad (A1.1)$$

where l_t is the reduced tank length:

$$l_t = \frac{1}{2} \int_{l_T} \frac{A_v}{A(l_T)}\, dl_T \qquad (A1.2)$$

where:

l_T – the length of the center line of the tank,

A_v – the cross sectional area of a wing section of the tank,

$A(l_T)$ – the local sectional area, measured perpendicular to the center line.

The inertia coupling coefficient is expressed by the formula:

$$s = -\frac{1}{2b_T}\cdot\beta^2 \qquad (A1.3)$$

in which $2b_T$ denotes the distance between free surface centers in wing sections of the tank (Fig.3.3), and

$$\beta^2 = \int_{l_T} r \sin(\vec{r}, \vec{dl})\, dl_T \qquad (A1.4)$$

where:

r – the distance of the center line element dl_T from the axis of rotation of the ship,

$\sphericalangle(\vec{r}, \vec{dl})$ – the angle between the radius r and the corresponding tangent to the center line.

The free surface correction factor Γ is expressed by the formula:

$$\Gamma = \frac{I_T g \rho_T}{\Delta GM} = \frac{2b_T^2 A_v g \rho_T}{\Delta GM} \qquad (A1.5)$$

where:

I_T – static surface moment relative to the axis of rotation of the ship,

Δ – ship displacement including weight of the stabilizing liquid in the tank,

GM – metacentric height of the ship with "frozen" stabilizing liquid in the tank.

The particular solutions of the equations (3.13) are:

$$\phi = \phi_A \sin(\omega_E t - \varepsilon_\phi) \qquad (A1.6)$$

$$\vartheta = \vartheta_A \sin(\omega_E t - \varepsilon_\vartheta) \qquad (A1.7)$$

where:

$$\phi_A = \alpha_A \omega^2_{\phi o} \sqrt{\frac{A_o^2 + B_o^2}{C_o^2 + D_o^2}} \qquad (A1.8)$$

$$\varepsilon_\phi = \mathrm{arctg}\, \frac{A_o D_o + B_o C_o}{A_o C_o - B_o D_o} \qquad (A1.9)$$

$$\vartheta_A = \alpha_A \omega_{\vartheta o} \frac{A_1}{\sqrt{C_o^2 + D_o^2}} \qquad (A1.10)$$

$$\varepsilon_{\vartheta} = \operatorname{arctg} \frac{D_o}{C_o} \qquad \text{(A1.11)}$$

and

$$A_o = \omega_{\vartheta_o}^2 - \omega_E^2$$

$$A_1 = \left(\frac{s}{g}\omega_E^2 + 1\right)\omega_{\phi_o}^2$$

$$B_o = -2\beta_{\vartheta}\omega_{\vartheta_o}\omega_E$$

$$C_o = \left(\omega_{\vartheta_o}^2 - \omega_E^2\right)\left(\omega_{\phi_o}^2 - \omega_E^2\right) - 4\beta_{\phi}\omega_{\phi_o}\beta_{\vartheta}\omega_{\vartheta_o}\omega_E^2 - \Gamma\left(\frac{s}{g}\omega_E^2 + 1\right)^2\omega_{\phi_o}^2\omega_{\vartheta_o}^2$$

$$D_o = 2\left[\left(\omega_{\phi_o}^2 - \omega_E^2\right)\beta_{\vartheta}\omega_{\vartheta_o} + \left(\omega_{\vartheta_o}^2 - \omega_E^2\right)\beta_{\phi}\omega_{\phi_o}\right]\omega_E$$

The stabilized ship roll equation taking the sway effect into account.

The particular solutions of the set of equations (3.14) are:

$$\phi = \phi_A' \sin\left[\omega_E t - \varepsilon_{\phi}'\right] \qquad \text{(A1.12)}$$

$$\vartheta = \vartheta_A' \sin\left[\omega_E t - \varepsilon_{\vartheta}'\right] \qquad \text{(A1.13)}$$

where:

$$\phi_A' = \alpha_A\omega_{\phi_o}\sqrt{\frac{A_o'^2 + B_o^2}{C_o^2 + D_o^2}} \qquad \text{(A1.14)}$$

$$\varepsilon_{\phi}' = \frac{\pi}{2} - \operatorname{arctg} \frac{A_o'C_o - B_oD_o}{B_oC_o + A_o'D_o} \qquad \text{(A1.15)}$$

$$\vartheta'_A = \alpha_A \omega_{\vartheta_0 \gamma} \sqrt{\frac{A_1'^2 + B_1^2}{C_0^2 + D_0^2}} \tag{A1.16}$$

$$\varepsilon'_\vartheta = \frac{\pi}{2} - \mathrm{arctg}\, \frac{A_1' C_0 - B_1 D_0}{B_1 C_0 + A_1' D_0} \tag{A1.17}$$

and the differences of the above are:

$$A_0' = -\Gamma\left(\frac{5}{3}\omega_E^2 + 1\right)\omega_{\vartheta_0}^2 + \left(\omega_{\vartheta_0}^2 - \omega_E^2\right)$$

$$A_1' = \left(\omega_{\phi_0}^2 - \omega_E^2\right) - \left(\frac{5}{3}\omega_E^2 + 1\right)\omega_{\phi_0}^2$$

$$B_1 = -2\beta_\phi \omega_{\phi_0} \omega_E$$

Appendix 2

The approximate estimation of certain parameters of the mathematical model of the system ship - passively-controlled tank.

The mathematical model of the system ship - passively-controlled tank was presented in chapter 3. The problem of identifying the coefficients of the models (3.13) and (3.14) was solved in chapters 3 and 7. But sometimes in the initial design stage of the tank it is helpfull to use approximate values. It is known that the non-dimensional coefficient of damping of the fluid motion in the tank can be obtained by model tests. The free surface correction factor Γ can be calculated from the expression (A1.5). The other two coefficients (s and $\omega_{\vartheta 0}$) can be estimated quite accurately (about 20%) from the empirical formulas.

In the author's opinion a more accurate value can be obtained using the method described in [1].

This method is based on the following notations (fig. A2.1).

The natural frequency of the tank is determined from the expression (A1.1). The parameter l_T can be approximately calculated from the formula:

$$l_T \cong d_T + \frac{A_V}{A_H} b_T \qquad \text{(A2.1)}$$

or accurately using (A1.2).

The inertia coupling coefficient s is determined from the formulas (A1.3) and (A1.4). Coefficient s can also be determined approximately from the expression:

$$s \cong -(a_T + d_T) \qquad \text{(A2.2)}$$

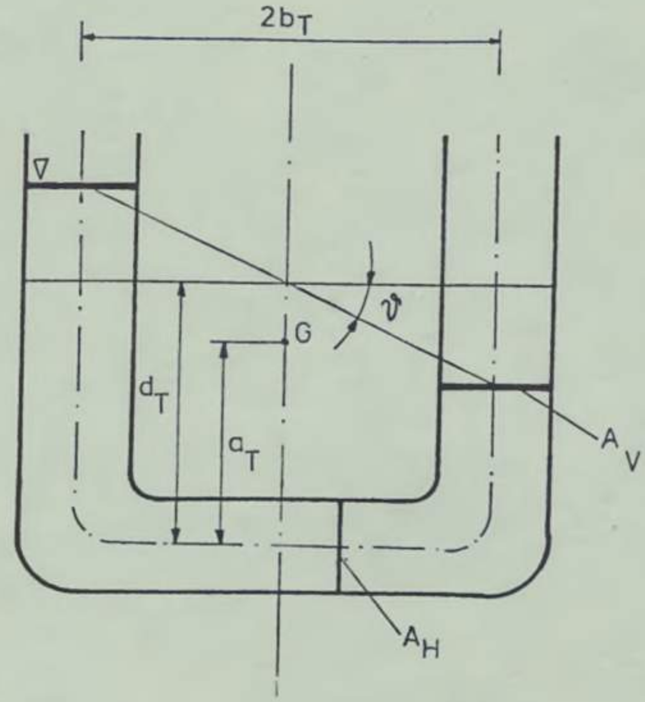

Fig. A2.1. The notations used for evaluating parameters s and $\omega_{\vartheta o}$ of the tank.

BIBLIOTEKA GŁÓWNA