

# Investigations of motion of the lifeboat lowered from ship's deck

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## ABSTRACT

*A calculation model as well as differential equations of motion of the lifeboat lowered from the deck of the ship under rolling in rough seas, are presented. The equations were numerically integrated. The computer software elaborated on their basis makes it possible to trace instantaneous positions of the lowered lifeboat, calculate velocities and accelerations of its elements, as well as determine forces acting on the boat and persons accommodated in it.*

**Keywords** : ship life-saving equipment, lifeboat launching from ship in waves, mathematical model

## INTRODUCTION

Human life protection and rescue is one of the most important problems with which designers of man-serving objects including first of all sea-going passenger ships must cope.

Introduction of mechanical propulsion to ships contributed to dynamic development of shipbuilding and shipping and various life-saving equipment, however till the tragedy of the Titanic there were no standards and legal regulations concerning the safety of crews and passengers. In 1914 in London the international convention on safety of life at sea was signed and then – as late as in 1929 – ratified by 40 states.

The gradually developing requirements for ship life-saving appliances were set in the successive SOLAS conventions adopted in 1948, 1960, 1974 and 1999.

Systems for evacuation of persons from large passenger ships have evolved from the simple open lifeboats and rafts dropped to water to unsinkable sheltered lifeboats of high strength and fire resistance, launched by means of more-and-more perfect side boat davits.

Fig.1 and 2 show examples of the contemporary davits with lifeboats accommodated in them [1].

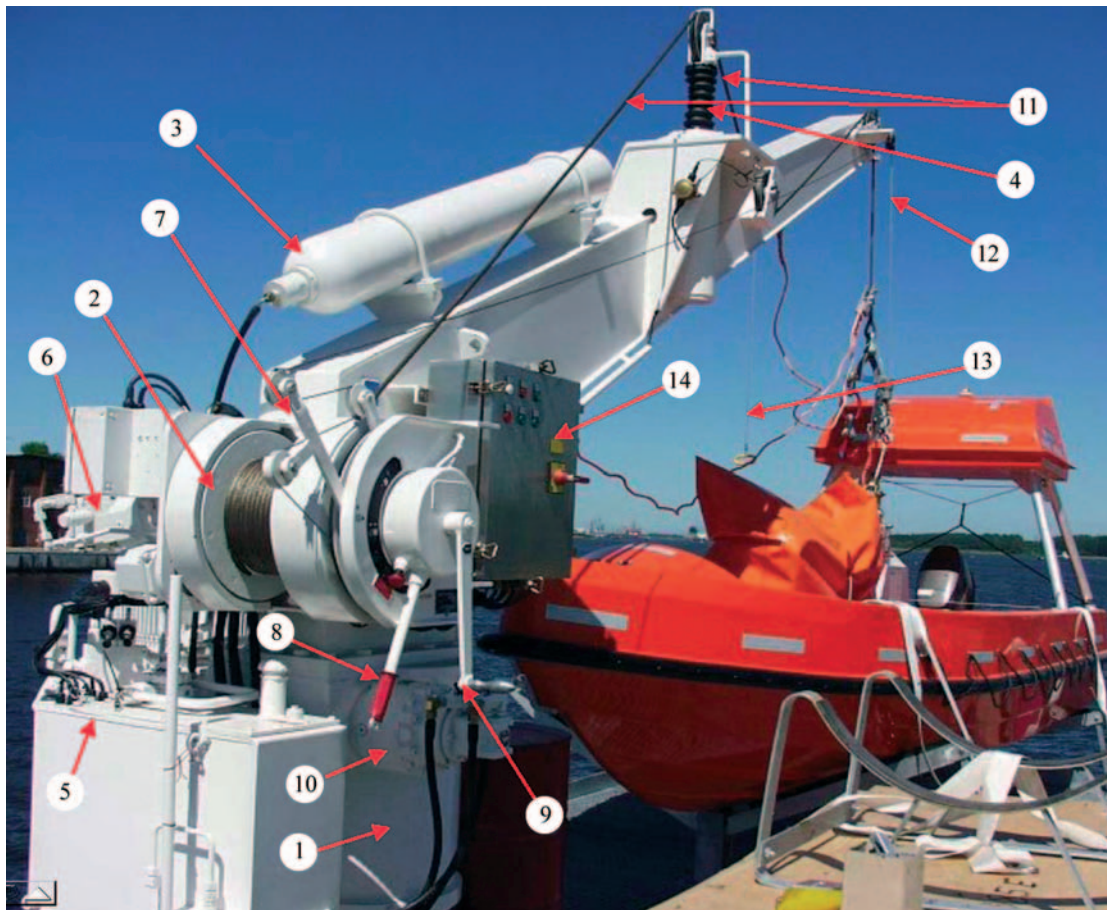
The lowering operation of the lifeboat with persons to water is the most dangerous phase of saving the lives from the ship in emergency at rough sea. The boat lowered from a large height (which is the case especially on large passenger ships) close to the side of rolling ship, often bumps against the ship side. The bumps generate relatively great overloads resulting from change of motion of the boat, that affects the persons located on it. The overloads may lead to a failure of the boat and injuries and even deceases of persons.

For this reason many recognized research centres and leading producers of ship life-saving and rescue equipment search for more and more perfect calculation methods of motion parameters of boat lowered from ship to water in rough

sea conditions. Having a reliable calculation program one can carry out investigations aimed at improving the present design solutions and testing novel devices of the kind.



**Fig. 1.** View of the lifeboat and the luffing jib - traveling davit during defection test .



**Fig. 2.** A slewing davit for fast rescue boat: 1. foundation, 2. boat winch, 3. hydraulic-gas accumulator, 4. shock absorber (damper), 5. electro-hydraulic power unit, 6. hydraulic motor of the winch, 7. lever of the remote control system of the brake, 8. lever of local control of the brake, 9. crank handle of the winch drive, 10. hydraulic motor of the slewing mechanism, 11. wire rope, 12. wire of remote control system of the brake, 13. wire of remote control system of the davit slewing, 14. switchboard.

The problem has been included in the scope of topics of the European SAFECRAFTS project under way in which also Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, has taken part [2]. In the frame of the project, the SIREHNA, French research centre, using the standard MSC. Adams software, performed a research on the motion of the model boat launched from ship's deck by means of two unextensible ropes hanging from the flexible jibs fixed to ship's deck [3]. Motion of the ship in irregular waves was treated as a random phenomenon.

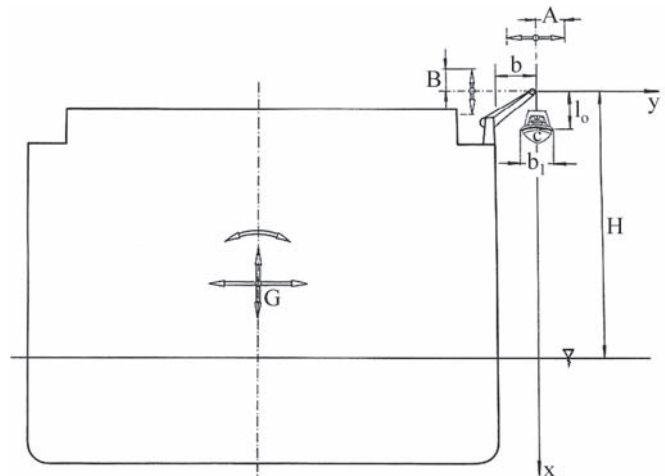
This paper presents some example results of the investigations, performed in Gdańsk University of Technology, of motion of a one-sling model boat launched from the ship in regular waves.

### ASSUMPTIONS OF THE THEORY

It was assumed that the rolling ship and the life boat launched from it is a system of two rigid bodies connected with a flexible davit and elastic rope of variable length. The davit can be fitted with a spring shock absorber with a damper automatically triggered in the case of exceeding the rated force value in the rope. The boat is lowered close to the flat part of ship side parallel to the ship's plane of symmetry. In the motion equations the influence of the boat motion on ship's rolling has been neglected because of the great difference of ship and boat masses.

☆ **The ship's rolling** was represented by a periodic planar motion described by harmonic functions [4]. For describing the boat motion, to know ship's roll period, displacement amplitudes of davit's nock and its coordinates at the initial

instant of boat launching, is necessary. The ship roll period can be determined by using the known methods of ship theory [5]. Values of displacement amplitudes of davit's nock depend on a place of location of the davit on ship's deck (Fig.3).



**Fig. 3.** Location of the boat davit on the ship deck.

☆ **Motion of lowered boat.** To describe the motion of the boat suspended on the rope, was used the model of pendulum consisting the linearly elastic line of variable length and having displacing point of suspension. The motion is realized due to influence of two forces :

- ◆ the gravity force  $mg$ , applied in the boat's mass centre and directed vertically down

- the longitudinal force in the rope,  $N$ , directed along the rope axis and proportional to the elastic displacement of the rope end,  $u$ , measured along the rope beginning from its location in the unloaded state :

$$N = K u$$

The proportionality coefficient  $K$ , called rigidity of elastic constraints of the lowered boat, is discussed below.

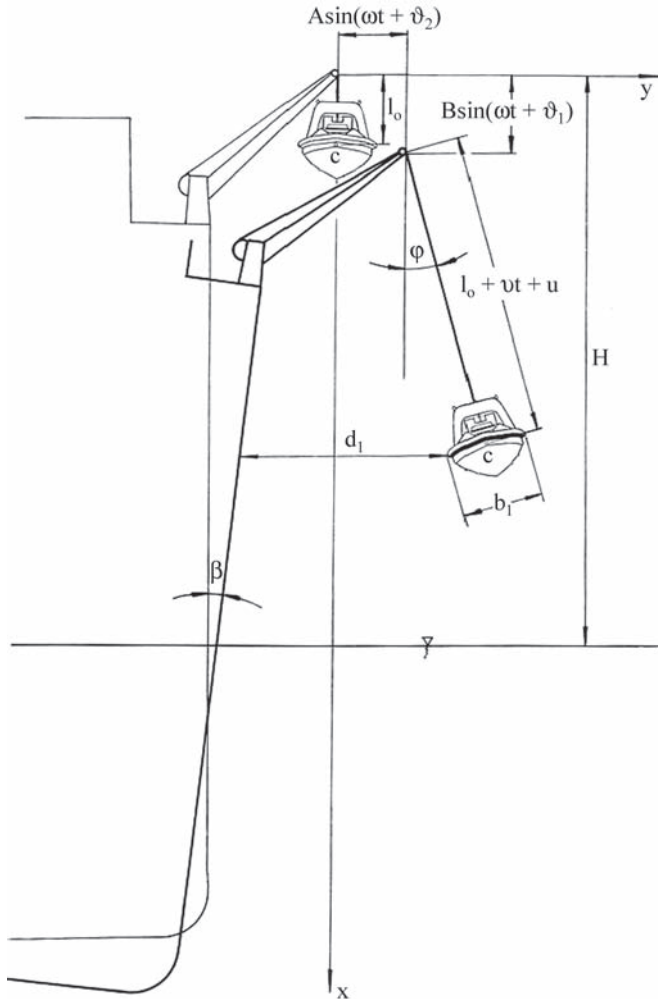


Fig. 4. Model of the motion of the boat launched from the ship rolling in waves.

After introduction of the following notation :

- $A$  – horizontal oscillation amplitude of davit nock [m]
- $b$  – davit nock distance from ship side [m]
- $b_1$  – life boat breadth [m]
- $B$  – vertical oscillation amplitude of davit nock [m]
- $g$  – acceleration of gravity  $9.81 \text{ [m/s}^2\text{]}$
- $l_0$  – davit nock distance from ship's mass centre at the beginning of motion [m]
- $m$  – mass of the boat with embarked persons [kg]
- $t$  – time [s]
- $T$  – period of ship's roll [s]
- $x_0$  – vertical coordinate of davit nock at the beginning of motion [m]
- $y_0$  – horizontal coordinate of davit nock at the beginning of motion [m]
- $v$  – rope running-out speed [m/s]
- $\varphi$  – rope inclination angle from vertical axis [rad]

the boat mass centre coordinates in the Earth - fixed frame of reference,  $Oxy$ , connected with the nock of the davit placed on the motionless ship in still water were described in the following form :

$$x = B \sin(\omega t + \vartheta_1) + (l_0 + vt + u) \cos \varphi$$

$$y = A \sin(\omega t + \vartheta_2) + (l_0 + vt + u) \sin \varphi$$

where :

$$\omega = \frac{2\pi}{T} \text{ [s}^{-1}\text{]}$$

Using Lagrange equations of 2<sup>nd</sup> kind one derived the differential equations of motion of boat mass centre :

$$\ddot{\varphi} = -\frac{1}{l_0 + vt + u} [2(v + \dot{u})\dot{\varphi} + g \sin \varphi + B\omega^2 \sin(\omega t + \vartheta_1) \sin \varphi - A\omega^2 \sin(\omega t + \vartheta_2) \cos \varphi]$$

$$\ddot{u} = g \cos \varphi - \frac{K}{m} u + [(l_0 + vt + u)\dot{\varphi}^2 + B\omega^2 \sin(\omega t + \vartheta_1) \cos \varphi + A\omega^2 \sin(\omega t + \vartheta_2) \sin \varphi]$$

in which the dots over symbols stand for derivatives respective to time. Numerical integration of the equations makes it possible to determine boat's position in an arbitrary instant  $t$ , as well as velocities, accelerations and forces acting on the boat.

- The rigidity of elastic constraints of launched boat** is expressed by the following formula :

$$\frac{1}{K} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3}$$

where :

- $k_1$  – tensile rigidity of rope
- $k_2$  – rigidity of davit
- $k_3$  – equivalent rigidity of shock absorber.

- The tensile rigidity of rope** can be estimated by means of the following formulae :

$$k_1 = \frac{EA}{1.6L}$$

where :

$L = l_z + vt$  – working length of rope in the instant  $t$   
 $E$  – Young modulus

$A = \frac{\pi d^2}{4}$  – rope cross-section area

$d$  – rope diameter,

or :

$$k_1 = \frac{EA_1}{1.3L}$$

where :

$A_1$  – sum of cross-section areas of particular wires of the rope.

- The rigidity of the davit of complex structure** is experimentally determined by measuring the static deflection of its jib's nock,  $f$ , under the load  $P$  :

$$k_2 = \frac{P}{f}$$

If the jib is of the form of uniform cross-section cantilever its rigidity is expressed by the formula :

$$k_2 = \frac{3EJ}{a^3}$$

in which :

- a – cantilever length
- E – modulus of elasticity
- J – moment of inertia of cantilever cross-section area respective to neutral bending axis.

★ **The equivalent rigidity of shock absorber.** If the davit's jib is fitted with a spring shock absorber its equivalent rigidity is defined as follows :

$$k_3 = \frac{k_0}{4 \cos^2 \alpha}$$

where :

- $k_0$  – rigidity of spring of shock absorber
- $\alpha$  – angle of rope inclination respective to spring axis (Fig.5).

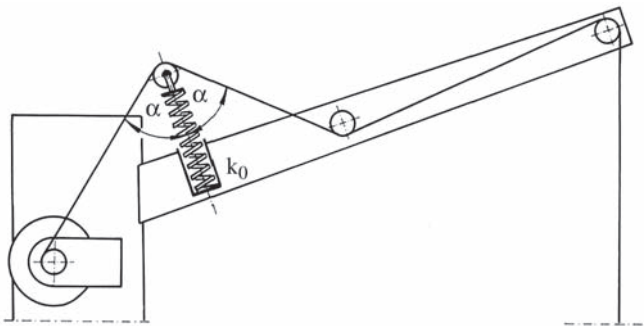


Fig. 5. Schematic diagram of work of shock absorber .

★ **The bump of the boat against ship's side.**

Lifeboat side is fitted with a fender made of an elastic damping material. To describe the bump of the fender against ship's side the linear- elastic collision model with the damping proportional to penetration velocity, was applied. In this case the motion equation is as follows :

$$\delta = \frac{v_0}{\lambda} e^{-ct} \sin(\lambda t)$$

where :

- $\delta$  – depth of penetration (depth of indentation of deformable fender in ship's side) [m]
- $v_0$  – initial velocity of collision [m/s].
- e – Napierian base
- c – damping coefficient of fender [s<sup>-1</sup>]
- t – time interval counted from the instant of contact of the fender and ship side

$$\lambda = \sqrt{\frac{K_0}{m} - c^2}$$

- $K_0$  – rigidity of elastic constraints [N/m]
- m – mass of the boat with accommodated persons .

The coefficients **c** and **K<sub>0</sub>** can be determined experimentally. The rigidity of elastic constraints can be determined by measuring the depth of fender indentation, **δ<sub>st</sub>**, caused by the horizontal force **F** which presses the boat into ship's side in the state of equilibrium :

$$K_0 = \frac{F}{\delta_{st}}$$

The boat which bumps, through its fender, against the ship's side with the initial relative velocity **v<sub>0</sub>**, will be bounced out the side with the relative velocity :

$$v_k = -v_0 e^{-c \frac{\pi}{\lambda}}$$

To determine the coefficient **c** it is hence necessary to know amplitudes of the angles of rope inclination from vertical, **φ<sub>0</sub>**, **φ<sub>k</sub>**, measured before and after the bump of the boat against the vertical motionless ship side, during the free letting-down of the pendulum consisted of the boat suspended on the rope of constant length :

$$c = \sqrt{\frac{\gamma^2 \frac{K_0}{m}}{\pi^2 + \gamma^2}}$$

where :

$$\gamma = \ln \sqrt{\frac{1 - \cos \phi_0}{1 - \cos \phi_k}}$$

The duration time of the contact of the boat fender and the ship side is as follows :

$$t_k = \frac{\pi}{\lambda}$$

★ **Example.** The collision run, calculated on the basis of the assumed model, for the lifeboat of the mass **m** = 3000 kg, fitted with the fender of the rigidity **K<sub>0</sub>** = 4.5 · 10<sup>6</sup> [N/m] and damping coefficient **c** = 22 [s<sup>-1</sup>], which bumps against the motionless ship side with the velocity **v<sub>0</sub>** = 1 [m/s], is presented in Fig.6.

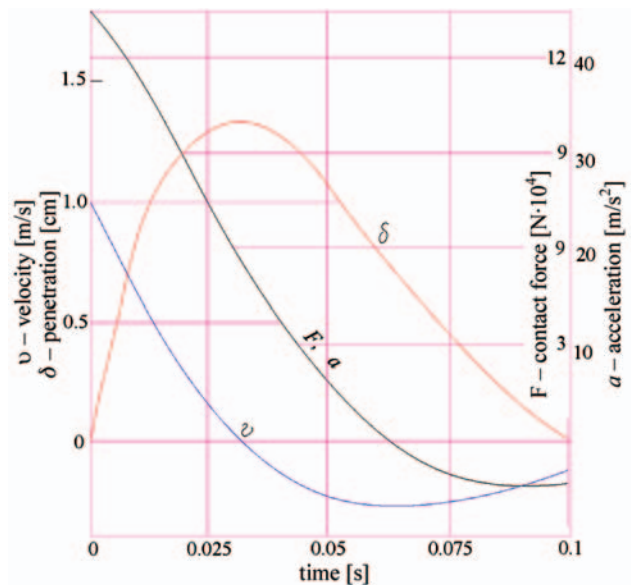


Fig. 6. Collision between the lifeboat and the hull of the vessel .

## COMPUTER SOFTWARE

The computer software based on the described calculation model serves for simulating the motion of the lifeboat suspended on the rope, lowered close to the side of rolling ship. It was written in C++ code and adjusted to operate under Windows system. Integration of the motion equations is performed by using the open step-by-step method [6] in which values of velocities and displacements calculated in the preceding step are the basis for computing the accelerations in the current time step.

The software user has to put in the following numerical data : ship roll period, amplitudes of horizontal and vertical oscillations of the davit nock , distance of the davit nock from the ship side and lifeboat mass centre at the beginning of the motion, davit nock coordinates at the beginning of the motion,

lifeboat mass and breadth, diameter and running-out velocity of the rope, rigidity of the davit and shock absorber spring, as well as lifeboat fender rigidity and damping coefficient.

In the software in question, the simulation of the motion amounts to displaying – on the monitor screen – successive positions of the lifeboat during its letting down. Values of boat mass centre velocity and acceleration, longitudinal force in the rope and deformation of the fender can be read out after termination of the software operation.

## EXAMPLE

### ★ Parameters of the lifeboat and davit

The lifeboat of the mass  $m = 3000$  kg and breadth  $b_1 = 2.3$  m, fitted with the fender of the rigidity  $K_0 = 4.5 \cdot 10^6$  [N/m] and damping coefficient  $c = 22$  [s<sup>-1</sup>], suspended on the steel line of the diameter  $d = 15$  mm, is let down by means of the deformable davit of the rigidity  $k_2 = 1.4$  [MN/m], fitted with the shock absorber with the spring of the rigidity  $k_0 = 500$  [kN/m], whose axis is inclined by the angle  $\alpha = 45^\circ$  respective to the line axis (Fig.5). The shock absorber is automatically triggered when the force in the line exceeds the value  $N = 1$  mg. The distance of the davit nock from the ship side plane :  $b = 4$  m, the line run-out velocity  $v = 0.5$  [m/s], the launching height  $H = 15$  m, the ship roll period  $T = 10$  s.

### ★ Variants of initial conditions for the calculated motion

The davit nock motion described by the equations :

$$x = 1.5 \sin \left( \omega t - \frac{\pi}{2} \right)$$

$$y = 3 \sin \left( \omega t - \frac{\pi}{2} \right)$$

has been caused by the dominating ship rolling motion, and the boat launching starts in the instant when the davit nock takes its highest position :  $x_0 = -1.5$  m ;  $y_0 = -3$  m, (Fig.7A). In the case when the motion is described by the equations :

$$x = 1.5 \sin \left( \omega t + \frac{\pi}{2} \right)$$

$$y = 3 \sin \left( \omega t + \frac{\pi}{2} \right)$$

the starting point of boat launching ( $x_0 = 1.5$  m ;  $y_0 = 3$  m) corresponds to the lowest position of the davit end (Fig.7B).

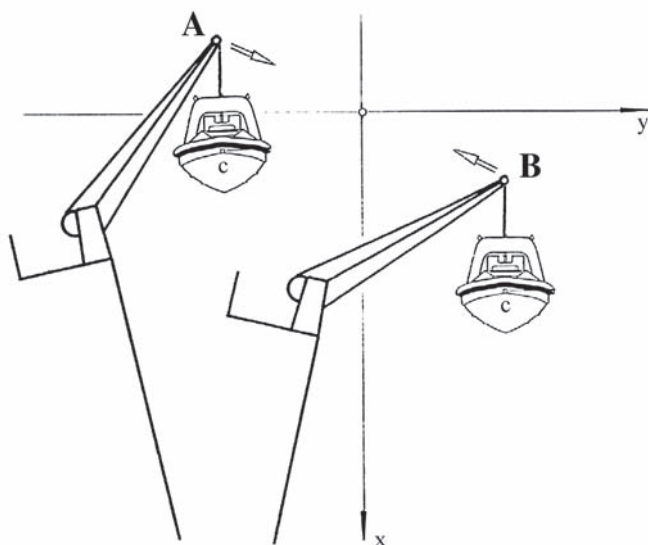


Fig. 7. Two extreme cases of the davit nock position in the instant of start of boat launching .

### ★ Selected results of the calculations

The diagrams shown in Fig.8A and B illustrate changes of the angle  $\phi$  and mass centre acceleration of the lifeboat with persons during its launching, for both the specified cases. In the first case the bump of the boat against the ship side occurred after 14 s, the maximum acceleration due to the bump achieved the value  $a = 14.6$  [m/s<sup>2</sup>] (Fig.8A).

In the case of start of the launching from the lowest position, the lifeboat bumped against the ship side after 18.6 s, and the maximum acceleration exceeded the value  $a = 60$  [m/s<sup>2</sup>] (Fig.8B).

## CONCLUSIONS AND FINAL REMARKS

- In this paper was presented a simple calculation model for describing the motion of the lifeboat launched with the use of one rope from the deck of rolling ship. The main simplifications of the model consist in the assumption that the boat motion is planar, ship rolling harmonic, and the rope axis does not incline from the boat plane of symmetry. However, elasticity of the rope, deformability of the davit as well as a contribution of the shock absorber in mitigating the dynamic overloads were taken into account. The simplicity of the assumed model makes it possible to reveal the most important features of the motion already in the preliminary phase of investigations.
- Despite the assumed simplifications, the results of the calculations based on the proposed model are close to those obtained by means of SAFECRAFTS software [3] in which description of the motion is three-dimensional, and ship rolling is assumed a random phenomenon.
- The computer software based on the assumed model is easy and fast in use. It can help the designer in monitoring the behaviour of the boat and the forces acting on it during its lowering or lifting from water. It makes it possible also to investigate the influence of both technical parameters of the appliance and environmental and initial conditions on the boat's motion and loading.
- The elaborated software may also serve for verification of calculations based on other models.
- The presented example results clearly show to which extent an instantaneous position from which the boat lowering starts as well as a place in which the davit is located on the ship's deck can influence course of the motion even in the same sea conditions. To influence number of bumps of the boat against ship side and to decrease bumping force even by several times is possible by an appropriate choice of the starting instant and position of the boat lowering.
- The applied theory may be easily generalized to arbitrary periodical davit nock motions as any periodical function can be developed into Fourier series fast convergent in the case of using it for description of ship motion. In the above presented differential equations of boat motion will then appear additional terms of the same structure as that of the already existing end terms representing the first harmonics.
- Influence of the inflection of rope axis from boat's plane of symmetry, longitudinal displacements of the boat as well as non-harmonic ship rolling – not accounted for in the presented model – on motion of the boat and magnitude of the forces inducing the motion, will be assessed in the course of further analyses.

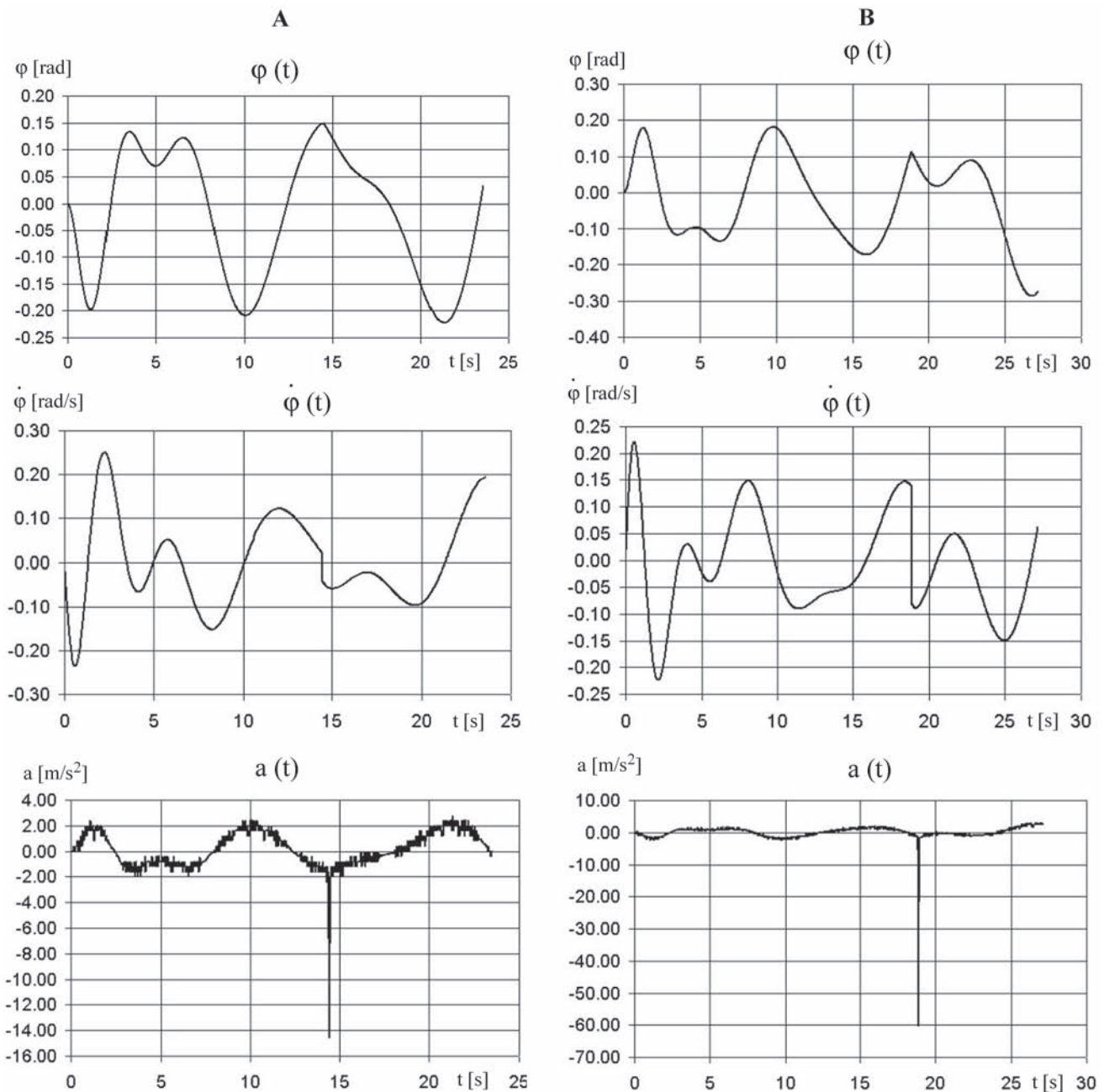


Fig. 8. The calculated runs of changes of the angle and angular velocity of inclination of the line with the lifeboat suspended on it, as well as the boat mass centre acceleration in the lateral direction respective to the ship side, in function of time, for two investigated cases of start of the launching process.

#### NOMENCLATURE

a	– cantilever length[m], maximum acceleration due to bump [m/s <sup>2</sup> ]	$k_0$	– rigidity of shock absorber spring [N/m]
A	– horizontal oscillation amplitude of davit nock [m], rope cross-section area [m <sup>2</sup> ]	$k_1$	– tensile rigidity of rope [N/m]
$A_1$	– sum of cross-section areas of particular wires of the rope [m <sup>2</sup> ]	$k_2$	– rigidity of davit [N/m]
b	– davit nock distance from ship side [m]	$k_3$	– equivalent rigidity of shock absorber [N/m]
$b_1$	– life boat breadth [m]	K	– rigidity of elastic constraints of the lowered boat [N/m]
B	– vertical oscillation amplitude of davit nock [m]	$K_0$	– rigidity of elastic constraints [N/m]
c	– damping coefficient of fender [s <sup>-1</sup> ]	$l_0$	– davit nock distance from ship's mass centre at the beginning of motion [m]
d	– rope diameter [m]	L	– total working length of rope [m]
e	– Napierian base	$L_z$	– initial working length of rope [m]
E	– Young modulus [N/m <sup>2</sup> ]	m	– mass of the boat with embarked persons [kg]
f	– static deflection of davit jib's nock under rated load [m]	N	– force in rope [N]
F	– force pressing the lifeboat into the ship side in state of equilibrium [N]	t	– time [s]
g	– acceleration of gravity 9.81 [m/s <sup>2</sup> ]	$t_k$	– duration time of contact of the boat fender and ship side [s]
H	– lifeboat launching height [m]	$T$	– period of ship's roll [s]
J	– moment of inertia of cantilever cross-section area respective to neutral bending axis [m <sup>4</sup> ]	u	– translation of the end of rope [m]
		x	– vertical coordinate [m]
		$x_0$	– vertical coordinate of davit nock at the beginning of motion [m]
		y	– horizontal coordinate perpendicular to ship plane of symmetry [m]

- $y_0$  – horizontal coordinate of davit nock at the beginning of motion [m]
- $\alpha$  – angle of rope inclination respective to spring axis [rad]
- $\delta$  – depth of indentation of deformable fender into ship's side [m]
- $\delta_{st}$  – depth of fender indentation caused by the horizontal force  $F$  [m]
- $\varphi$  – rope inclination angle from vertical axis [rad]
- $\varphi_0, \varphi_k$  – amplitude of the angles of rope inclination from the vertical, measured before and after the bump of the boat against the ship side, respectively [rad]
- $v$  – rope running-out speed [m/s]
- $v_0$  – initial velocity of collision (bump) [m/s]
- $v_k$  – relative velocity of the lifeboat bounced out of the ship side [m/s]
- $\omega$  – ship roll angular velocity [ $s^{-1}$ ]

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## Miscellanea

### ZTM Marine Technology Unit

On 19 January 2006 the first – in – the - year plenary meeting of the Marine Technology Unit (acting in the frame of the Transport Technical Means Section, Transport Committee, Polish Academy of Sciences), was held in Gdynia; it was hosted by Electrical and Mechanical Faculty of Polish Naval University of Gdynia.

During the scientific part of the meeting were presented two papers prepared by the scientific workers of the University :

- ❖ *Choice of repair technology for screw propellers, depending on distribution their strength (mechanical) properties* – by K. Rogowski
- ❖ *Choice of water content in water-fuel emulsion delivered to cylinders, depending on service process state of ship diesel engine* – by R. Zadrag

After the presentation an interesting discussion on both the papers developed.

In the organizational part of the meeting Prof. J. Girtler (Gdańsk University of Technology) presented the report on the Unit's activity in 2005 and submitted proposals to the program of the Unit's activities in 2006. Problems of organization of scientific conferences on design and operation of ship power plants were also discussed.

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On 12-14 September 2005

7th Domestic Scientific Technical Conference on :

### *Diagnostics of Industrial Processes*

was held in Rajgród, a small town  
in the lake region of north-eastern Poland.

It was arranged under the auspices of the Committee on Automation and Robotics, Polish Academy of Sciences, with the Institute of Automation and Robotics, Faculty of Mechatronics, Warsaw University of Technology as its main organizer. 3 plenary papers and 110 technical papers were prepared for the Conference, and presented during 15 topical sessions, as follows :

- ❖ **Diagnostic methods** (12 papers)
- ❖ **Analytical methods** (7 papers)
- ❖ **Artificial intelligence – fuzzy logics** (7 papers)
- ❖ **Artificial intelligence – neural networks** (6 papers)
- ❖ **Artificial intelligence – genetic algorithms** (6 papers)
- ❖ **Knowledge engineering** (6 papers)
- ❖ **Diagnostic systems** (12 papers)
- ❖ **Industrial applications** (12 papers)
- ❖ **Diagnostics of mechanical devices** (7 papers)
- ❖ **Diagnostics of automation systems** (6 papers)
- ❖ **Failure-proof systems** (7 papers)
- ❖ **Safety and diagnostics of computers** (5 papers)
- ❖ **Vision methods** (6 papers)
- ❖ **Selected problems** (7 papers)
- ❖ **Medical applications** (4 papers)

Authors of the papers represented 26 universities, scientific institutes and research centres including 4 from : Germany, Spain, Belarus and United Kingdom. Representatives of Warsaw University of Technology, Silesian University of Technology, Gdańsk University of Technology, University of Zielona Góra and Metallurgy and Mining Academy gave the greatest contribution to the Conference program, presenting 23, 15, 14, 13 and 11 papers, respectively.

The Conference's proceedings were published in the special issue (September 2005) of *POMIARY, AUTOMATYKA, KONTROLA*, a scientific technical monthly.