

# IDENTIFICATION OF DAMAGES OF TRIBOLOGICAL ASSOCIATIONS IN CRANKSHAFT AND PISTON SYSTEMS OF TWO-STROKE INTERNAL COMBUSTION ENGINES USED AS MAIN PROPULSION IN SEA-GOING VESSELS AND PROPOSAL OF PROBABILISTIC DESCRIPTION OF LOADS AS CAUSES OF THESE DAMAGES

Jerzy Girtler, Prof.

Gdańsk University of Technology, Poland

## ABSTRACT

*The article discusses damages of essential tribological associations in crankshaft and piston systems of large power two-stroke engines used as main engines, which take place during transport tasks performed by those ships. Difficulties are named which make preventing those damages impossible, despite the fact that the technical state of engines of this type is identified with the aid of complex diagnostic systems making use of advanced computer technology. It is demonstrated that one of causes of the damages is the lack of research activities oriented on recognising random properties of the loads leading to those damages. A proposal is made for the loads acting at a given time  $t$  on tribological associations in crankshaft and piston systems of internal combustion engines used as main engines to be considered as random variables  $Q_t$ . At the same time the loads examined within a given time interval  $t_1 \leq t \leq t_2$  would be considered stochastic processes  $\{Q(t): t \geq 0\}$ . Essential properties of the loads of the abovementioned tribological associations are named and explained by formulating hypotheses which need empirical verification. Interval estimation is proposed for estimating the expected value  $E(Q_t)$  of the load  $Q_t$  acting at time  $t$ . A relation is indicated between the mechanical load and the thermal load acting on tribological associations in the ship main engine crankshaft and piston system. A suggestion is formulated that a stochastic form of the relation between these types of load is to be searched for, rather than statistic relation, and a proposal is made to measure the intensity (strength) of the stochastic relation using the Czuprow's convergence coefficient.*

**Keywords:** probability, friction associations, crankshaft and piston system, semi-Markov process, marine internal combustion engine, wear

## INTRODUCTION

In sea transport, ships are frequently driven using gearless propulsion systems with diesel engines as main engines (Fig. 1). Due to difficult conditions of their operation, especially in stormy weather, (Fig. 2) the cooperation of each main engine (1) and propeller (8) in a given propulsion system (Fig. 1) is executed at different points of the performance field (operating field) of this engine (Fig. 3). The main engines are frequently forced to operate in the overload field (Fig. 4), in which they are excessively loaded. As a result, their tribological associations are subject to fast wear and damages, which can sometimes be very widespread. Propulsion systems of numerous sea-going vessels such as container ships, bulk carriers, and tankers, make use of two-stroke, turbocharged, low-speed, crosshead, large-power, reversible, direct-power diesel engines (Fig. 5). Damages of tribological associations in this type of engines lead to

substantial financial losses of ship-owners, which are not only connected with necessary repair, frequently done in ship repair yards, but also with ship stoppages which make performing transport services impossible [21]. Moreover, damages of those engines taking place during navigation in stormy weather (Fig. 2) lead, as a rule, to ship's foundering. To prevent these damages, engine producers equip them with diagnosing systems (SDG), which make more and more frequent use of computer technology. A list of such computer-aided systems includes: CoCoS (Computer Controlled Surveillance System) worked out by MAN B&W Diesel Group, and CBM (Condition-Based Maintenance) worked out by Wartsila. These systems enable to identify technical states of ship main engines, being the diagnosed systems (SDN). Unfortunately, both

systems (SDN and SDG), altogether bearing the name of the diagnostic system (SD), are not adapted to generate a complete diagnosis which would include instantaneous diagnosis, genesis, and prognosis, as well as to make decisions based on the statistical decision-making theory. That is a reason why damages of ship main engines still take place during ship operation. These damages, especially those concerning tribological associations, are carefully documented in photos (Fig. 6, Fig. 12) and meticulously described by engine users and producers. Difficulties in preventing the abovementioned damages are connected with the use of deterministic, and not probabilistic description of wear of engine elements. This refers both to linear (surface) wear, mainly connected with sliding friction taking place in friction associations, and volumetric wear resulting from the appearance of microcracks and their further propagation inside the abovementioned elements. To change this situation, random nature of wear processes and the resultant damages is to be taken into account to make a basis for probabilistic description of both the course of wear of tribological associations of engine elements, and the loads being their sources.

Most heavily loaded main engine elements include tribological associations in the crankshaft and piston system and the cylinder liners (Fig. 5). That is why the analysis presented in this article mainly refers to probabilistic properties of thermal and mechanical loads of elements composing these associations.

### PROPERTIES OF LOADS AND SAMPLE DAMAGES OF TRIBOLOGICAL ASSOCIATIONS IN MAIN ENGINE CRANKSHAFT AND PISTON SYSTEMS

Properties of loads which lead to damages of tribological associations in crankshaft and piston systems are presented

here in the form of scientific hypotheses, complemented by relevant explanations. Each hypothesis is an empirically verifiable statement which includes syntactic implication. Part of this statement situated before the syntactic implication is a probable cause of the fact described in the part situated after the implication. The properties of the abovementioned loads are described using a probabilistic approach. This approach is necessary because the loads generated during the operation of friction associations are random variables and should be regarded as functions which attribute a numerical value to each random phenomenon having the form of load record. The random nature of these phenomena results from randomly changing atmospheric and sea conditions (WAM) in which ships perform their transport tasks [4, 5, 6, 26]. Depending on those conditions, (Fig. 1) the screw propeller (8) of each ship generates a randomly changing thrust force ( $T_N$ ) which leads to randomly changing load of the crankshaft and piston system (9) of the main engine (1).

Randomly changing atmospheric and sea conditions (WAM) can be affected by many factors, such as: temperatures, air pressure and humidity, height, natural precipitation, wave direction and speed, sea currents, wind direction and speed, hull trims and rolls, oscillatory hull motions (including heaving, pitching, swaying, rolling), etc. These conditions are a source of time dependent action of mechanical loads (pressures, forces, and moments) and thermal loads (heat fluxes, temperatures, temperature gradients) directed towards the main engine crankshaft system and, consequently, to its tribological associations. Similar loads are also generated due to: harbour manoeuvres, navigation in water regions (channels) of limited depth and width, hull fouling by algae and crustaceans, course change with respect to wave and wind directions, different hull load, etc. Moreover, large gas forces are randomly generated by the main engines, with the resultant inertia forces and torques of crankshaft and piston systems

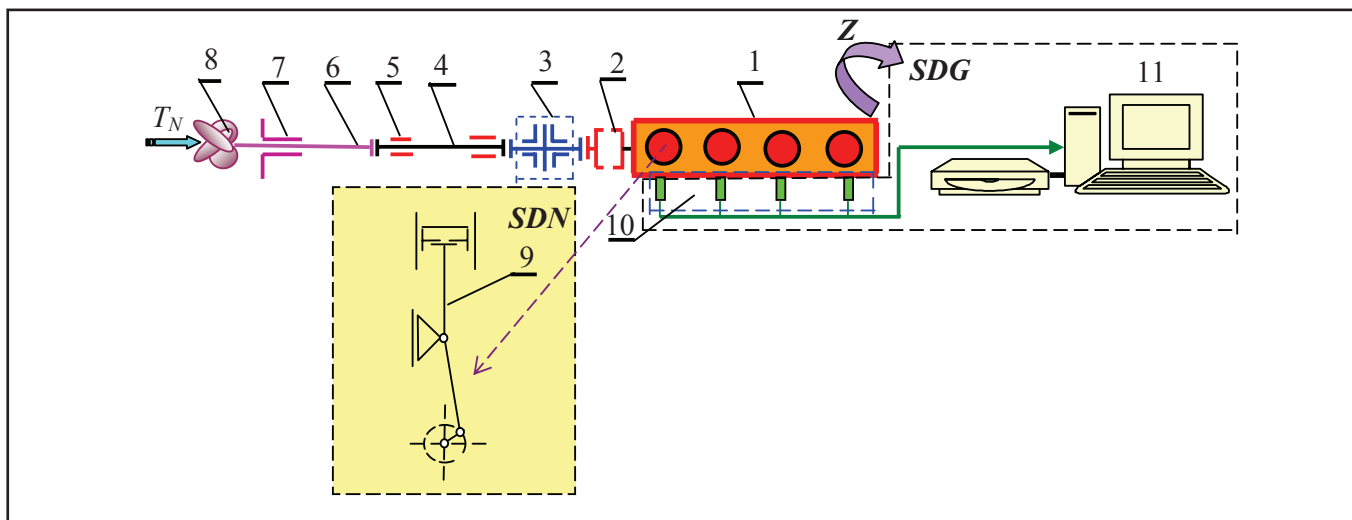


Fig. 1. Functional scheme of the propulsion system without gearbox (direct drive) and a diesel engine. Attention is focused on the crosshead crankshaft and piston system (9) as SDN (with SDG): SDN – diagnosed system (crankshaft and piston system), Z – disturbances, SDG – diagnosing system, 1 – main (diesel) engine, 2 – disengaging (elastic) coupling, 3 – thrust bearing with thrust shaft, 4 – intermediate shaft, 5 – shaft support bearings, 6 – propeller shaft, 7 – propeller shaft pipe, 8 – screw propeller, 9 – crosshead engine crankshaft and piston system, 10 – SDG measuring instruments, 11 – SDG computer unit

transmitted to ship screw propellers. The above loads can be large, especially in cases of cargo ships, such as container ships, tankers, and bulk carriers, which have high draught and screw propellers of over 9 m in diameter and mass exceeding 100 t [28, 29, 30]. These loads become remarkably larger when the main engines work during ship navigation in stormy weather (Fig. 2). In these conditions the randomly changing part of load of these engines increases significantly, which leads to similar increase of the loads of tribological associations in their crankshaft and piston systems, with their intensive wear and damages as a further consequence.

Stormy weather at sea is a reason why crankshaft and piston systems of main engines are frequently heavily loaded (overloaded) with maximal power generated in engine cylinders, and accompanied by extremely large inertia forces when the stern is lifted by the undulated sea and the propeller protrudes above the water surface.



Fig. 2. View of a ship performing transport task in stormy weather [33]

Another source of crankshaft and piston system load changes are unavoidable changes of the rate of fuel injected, depending on current needs, to the main engine combustion chambers to obtain the ship speed necessary to perform transport tasks in the conditions determined by propeller characteristic changes provoked by changing sea conditions. This situation makes the engine operate in different areas of its performance field. Figure 3 shows possible performance field areas (1) of the main engine (2) of a given type which drives directly the screw propeller (3).

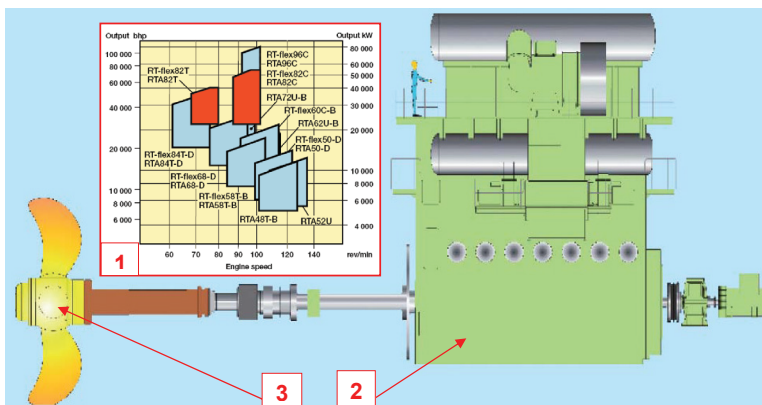


Fig. 3. Propulsion system often used on container ships, bulk carriers, and tankers [1]: 1 – performance fields of different engine families, delivered by their producer, 2 – two-stroke low-speed engine, 3 – screw propeller

If possible, the engine is loaded in such a way that it can operate in a contracted area, i.e. the area which allows the engine to reach its overall efficiency ( $\eta_o$ ) as high as possible, for instance  $\eta_o = 0,55$ . Randomly changing atmospheric and sea conditions (WAM) are a reason why randomly changing main engine loads and, consequently, crankshaft and piston system loads can be shifted to different areas of engine performance field, including the engine overload area (Fig. 4).

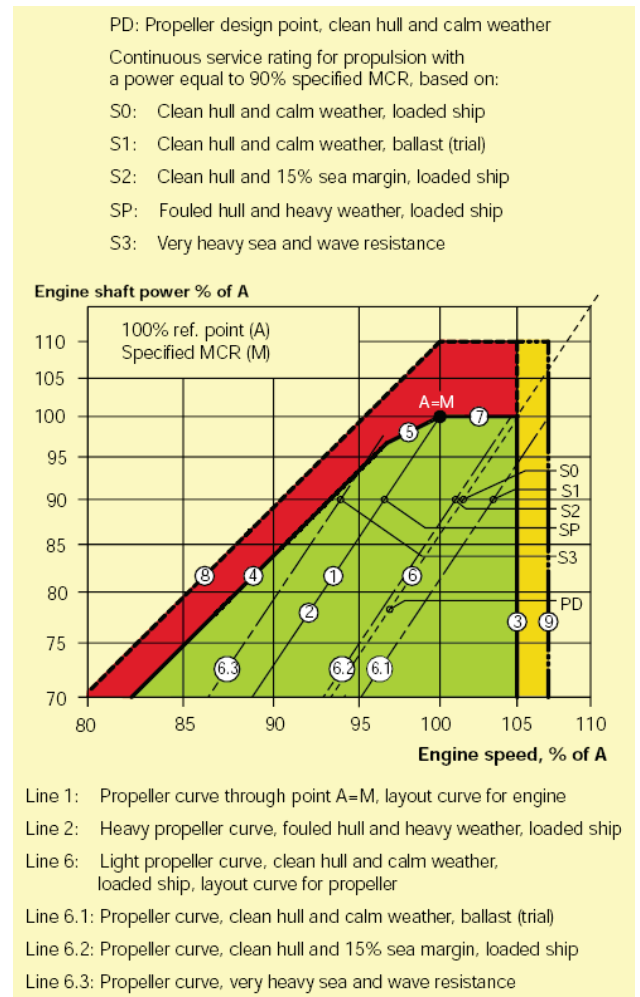


Fig. 4. Engine operation area with marked curves and points of different engine loads resulting from current WAM conditions [31]

The operation of the ship main engine crankshaft and piston system, defined as SDN, is controlled by SDG, which nowadays is more and more frequently equipped with a computer (Fig. 1). The use of these systems (SDG) aims at arriving at such control of engine operation that the engine load does not lead to excessive wear and resultant widespread damage of, most of all, cylinder liners, pistons and their rings, and journals and bushes of main, crankshaft, and crosshead bearings (Fig. 5) [33].

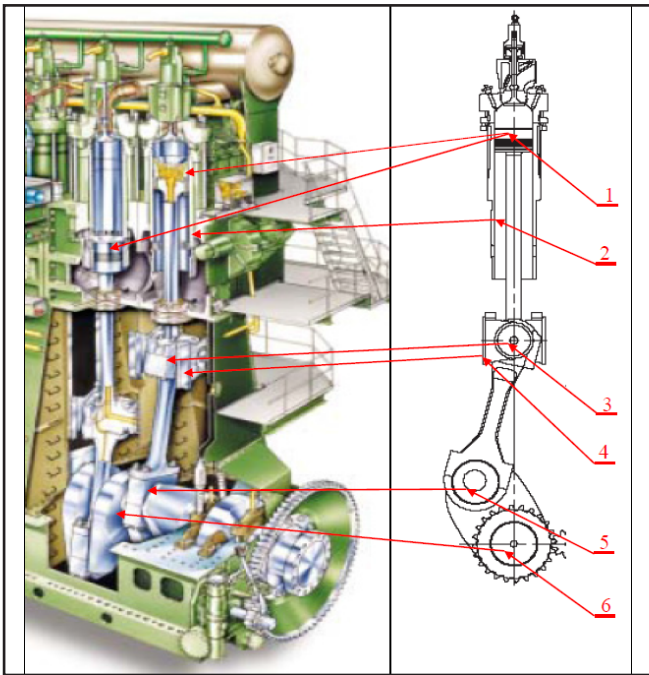


Fig. 5. Elements of main friction associations in SG: 1 – piston, 2 – cylinder liner, 3 – crosshead bearing (journal and bush), 4 – crosshead shoe sliding along the guideway, 5 – crankshaft bearing (journal and bush), 6 – main bearing (journal and bush)

Despite the use of a computerised SDG, the producer cannot predict the course of wear of engine elements and its scale at a given time of engine operation. Consequently, they cannot prevent damages of tribological associations illustrated in Figs. 6÷12. That is why damages, sometimes very widespread, of (Fig. 5): side surfaces of the piston 1 and its rings, inner (bearing) surfaces of the cylinder liner 2, crosshead bearings 3, crosshead shoes 4, crankshaft bearings 5, and main bearings 6 can be observed during main engine operation. Sample cases of excessive wear and resultant damages are shown in Figs. 6÷11 for: piston 1 and its rings (Fig. 6), cylinder bearing surface 2 (Fig. 7), journal of the crosshead bearing 3 (Fig. 8) and its lower bush 1 mounted in the connecting rod head (Fig. 10b), crosshead shoe 4 (Fig. 9), lower bush 2 of the crankshaft bearing 5 (Fig. 10c), and main bearing journal 6 (Fig. 11).

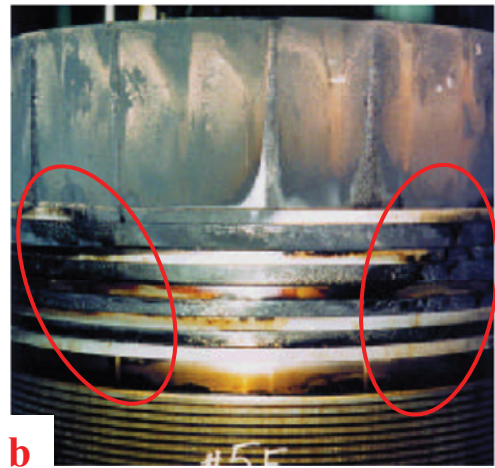


Fig. 6. View of: a) side surfaces of piston 1, excessively worn due to friction (Fig. 5), b) cracked piston rings with losses of material [15, 37]

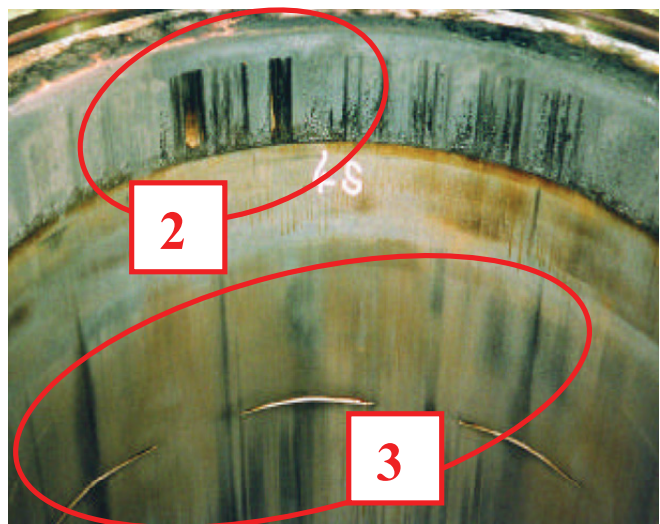


Fig. 7. Views of: indentation 1, corrosion wear 2, and traces of seizure 3 of the inner (bearing) surface of the cylinder liner 2 (Fig. 5) [15, 32]



a

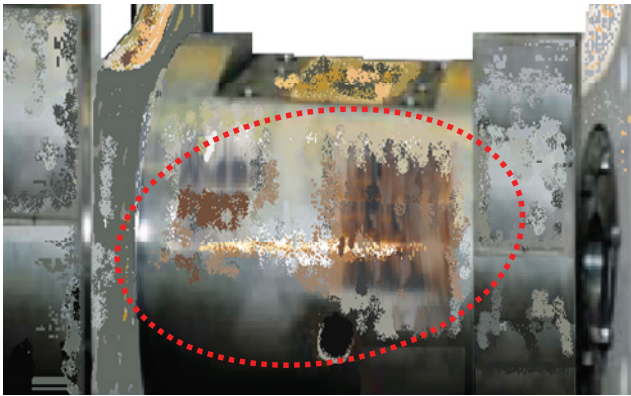


Fig. 8. View of traces of seizure and corrosion wear resulting from overheating of the crosshead bearing journal 3 (Fig. 5) [39]

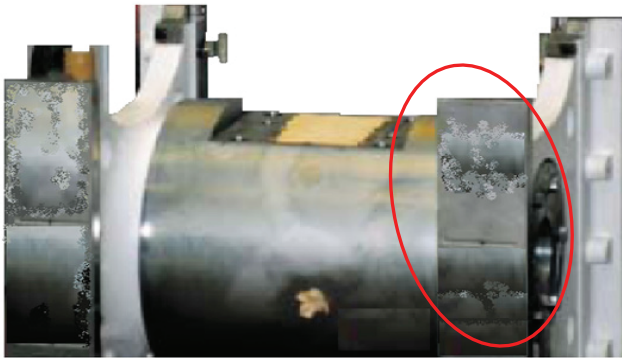


Fig. 9. View of excessive friction and corrosion wear, accompanied by local deep losses of material in the surface layer of the crosshead shoes 3 (Fig. 5) [36]

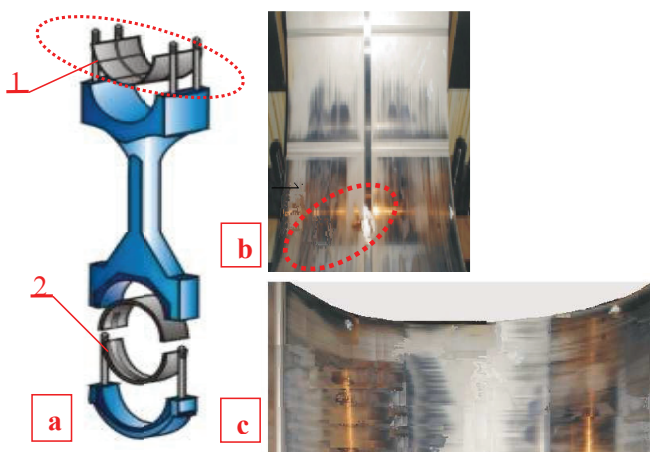


Fig. 10. Views of: a) connecting rod, b) excessive wear of the sliding surface of the lower bush 1 of the connecting rod head bearing; c) excessive wear of the sliding surface of the lower bush 2 of the crankshaft bearing 5 (Fig. 5) [35, 36]

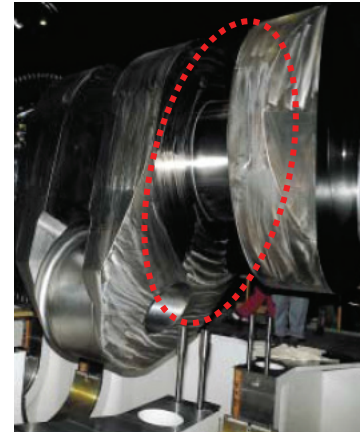


Fig. 11. View of excessive friction wear of the main bearing bush of the crankshaft 6 (Fig. 5) [15, 35]

The presented examples show that friction associations in ship main engines are exposed to widespread damages during engine operation, therefore reasons of difficulties in constructing SDGs which would be able to prevent such damages are to be recognised. The preformed research suggests that the main reason of the above difficulties are random properties of their loads, with large dispersion.

## PROPERTIES OF LOADS OF TRIBOLOGICAL ASSOCIATIONS IN MAIN ENGINE CRANKSHAFT AND PISTON SYSTEMS

Examining the time-history of load  $\{Q(t); t \geq 0\}$  of an arbitrary tribological association in any crankshaft and piston system enables to record real realisations  $q_i(t)$ ,  $i = 1, 2, \dots, n$ , of this process [3, 4, 13, 17]. Three sample realisations of the process  $\{Q(t); t \geq 0\}$  which have been recorded starting from time  $t_r$  are shown in Fig. 12.

This figure shows that at each time  $t$  the load  $Q_t$  as a random variable can take one of three possible values (with certain probability):  $q_1, q_2, q_3$ . The role of variables  $q(t)$  can be played, for instance, by: lubricating oil pressure, temperature and heat released by friction in the friction association, etc.

Consequently, the random variables  $Q_t$  analysed at an arbitrary time  $t_r \leq t \leq t_z$  (Fig. 12) of engine operation compose a family of random variables. Therefore the process of loading of friction associations in an arbitrary crankshaft and piston system is a stochastic process  $\{Q(t); t \geq 0\}$ , i.e. a function the values of which are random variables assigned to a given time  $t$  of their action. Here  $t$  as the process parameter is not a random variable [4, 6, 9, 13, 20].

In this situation the following hypothesis  $H_1$  can be

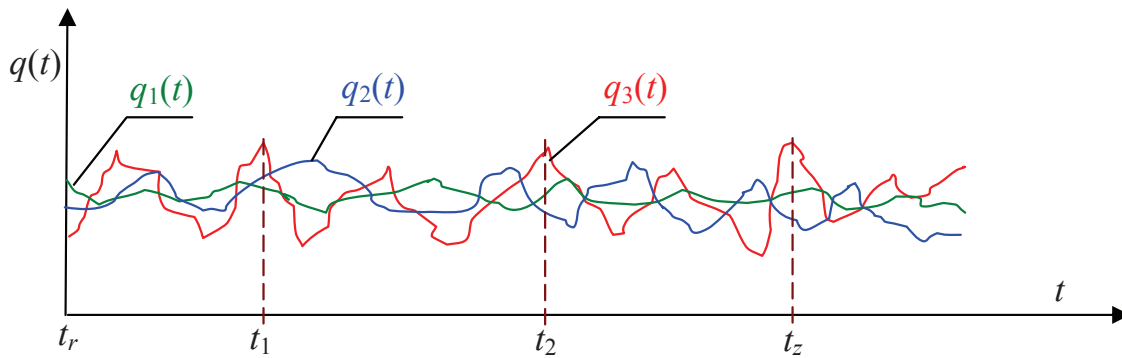


Fig. 12 Sample realisations of the process  $\{X(t): t \geq 0\}$  of load of an arbitrary internal combustion engine crankshaft and piston system;  $t$  – time of engine operation during which energy conversion takes place,  $t_r$  – examination start time,  $t_z$  – examination stop time

formulated: **the process of loading of an arbitrary friction association in each crankshaft and piston system of a arbitrary ship main engine is the stochastic process, because the association load values assigned to arbitrary times  $t$  are random variables.**

Hence the loads of tribological associations in the ship main engine crankshaft and piston systems analysed at an arbitrary time  $t$  of engine operation are to be considered as random variables  $Q(q, t)$ , where  $q$  is an event understood as the load value recorded at time  $t$  which should be considered the realisation of this random variable. In other words: at a given time  $t$  there exists  $Q(t) = q$ . That is why when examining the time dependent load of an arbitrary tribological association of each ship main engine crankshaft and piston system, this load should be considered a random (stochastic) process  $\{Q(t): t \geq 0\}$  [6, 5, 18, 19, 23].

Probabilistic description of each stochastic process requires working out limited-dimension distributions of this process. That means that working out the  $n$ -dimensional distribution of the load process  $\{Q(t): t \geq 0\}$  requires working out  $n$ -dimensional random vectors  $[Q(t_1), Q(t_2), Q(t_3), \dots, Q(t_n)]$  for all parameters  $t_1, t_2, t_3, \dots, t_n$ . This distribution, having the form of  $n$ -dimensional distribution function  $F$ , is given by the formula:

$$F(q_1, q_2, q_3, \dots, q_n, t_1, t_2, t_3, \dots, t_n) = P\{Q(t_1) < q_1, Q(t_2) < q_2, Q(t_3) < q_3, \dots, Q(t_n) < q_n\} \quad (1)$$

Continuous load changes can be only assessed and taken into account in the distribution (1) when the realisations of the process  $\{Q(t): t \geq 0\}$  are recorded continuously, otherwise the abovementioned real process of loading of friction associations of tribological systems should be considered the stochastic process which is discrete in states. In this situation the process should be described using an  $n$ -dimensional distribution having the following form:

$$P(q_1, q_2, q_3, \dots, q_n, t_1, t_2, t_3, \dots, t_n) = P\{Q(t_1) = q_1, Q(t_2) = q_2, Q(t_3) = q_3, \dots, Q(t_n) = q_n\} \quad (2)$$

When examining  $n$  realisations of the load  $Q_t$  as the random variable at an arbitrary time  $t$ , we arrive at the results in the form of  $q_{t(i)}$  ( $i = 1, 2, 3, \dots, n$ ), (Fig. 12). These results enable to calculate the value of the statistics  $\bar{Q}_t$ , which is the mean value  $\bar{q}_t$  calculated from [3, 9, 13, 17]:

$$\bar{q}_t = \frac{1}{n} \sum_{i=1}^n q_{t(i)} \quad (3)$$

where:

$q_{t(i)}$  – value of the load  $Q_t$  recorded during  $i$ -th examination

The statistics  $\bar{Q}_t$  has the asymptotically normal distribution  $N\left(E(Q_t); \frac{\sigma_t}{\sqrt{n}}\right)$  regardless of the functional form of the distribution of the random variable  $Q_t$ . Here:  $E(Q_t)$  and  $\sigma_t$  are, respectively, the expected value and the standard deviation of the random variable (load)  $Q_t$ . This statistics can be almost always used, as for  $n \geq 4$  the concurrence of distribution of the statistics  $\bar{Q}_t$  to normal distribution  $N\left(E(Q_t); \frac{\sigma_t}{\sqrt{n}}\right)$  is very fast. Adopting an assumption that the load  $Q_t$  examined at time  $t$  has normal distribution  $N(E(Q_t); \sigma)$  does not constitute a practical limitation, as the statistics  $\bar{Q}_t$  always has the asymptotically normal distribution [9, 13, 17]. The statistics  $\bar{Q}_t$  is consistent and unbiased, and hence it is the best estimator of the unknown expected value  $E(Q_t)$  of the load  $Q_t$  as the random variable. The results of examination of the load  $Q_t$  can be used for calculating, along with the mean value  $\bar{q}_t$ , also the standard deviation  $s_t$  from the formula [9, 13, 17]:

$$s_t = \sqrt{\frac{1}{n} \sum_{i=1}^n (q_{t(i)} - \bar{q}_t)^2} \quad (4)$$

The statistics  $s_t$  given by the formula (2) is consistent and asymptotically unbiased, and hence again, it is the best estimator of the standard deviation  $\sigma$  of the random variable  $Q_t$ . The values of  $\bar{q}_t$  and  $s_t$  obtained during the load examination

make it possible to assess the expected value  $E(Q_t)$  of the load in the interval form using the formula [9, 13, 17]:

$$P \left\{ \bar{q}_t - t_{\alpha, n-1} \frac{s_t}{\sqrt{n-1}} \leq E(Q_t) \leq \bar{q}_t + t_{\alpha, n-1} \frac{s_t}{\sqrt{n-1}} \right\} \quad (5)$$

where:

$t_{\alpha, n-1}$  – Student's  $t$ -distribution parameter.

The above approach to the examination of loads of tribological associations in ship main engine crankshaft and piston systems enables to determine the course of the stochastic process  $\{Q(t): t \geq 0\}$  along with its expected value  $E[Q(t)]$  and two margins, which are the upper margin  $E[Q(t) + \sigma[Q(t)]$  and the lower margin  $E[Q(t) - \sigma[Q(t)]$  (Fig. 13), i.e. the interval

$$E[Q(t) - \sigma[Q(t)]] \leq E[Q(t)] \leq E[Q(t) + \sigma[Q(t)]] \quad (6)$$

This results from the fact that at an arbitrary time of their action, the loads of tribological associations in ship main engine crankshaft and piston systems are random variables  $Q_t$  with the expected value  $E(Q_t)$  and the variance  $D^2(Q_t)$ . When examined within the time interval  $(t_r, t_z)$ , i.e. when  $t_r \leq t \leq t_z$  ( $t_r$  – examination start time,  $t_z$  – examination stop time), these loads compose a set of random variables  $Q_t$  which are the instantaneous states of the stochastic process  $Q(t)$  with the expected value  $E[Q(t)]$  and the variance  $D^2[Q(t)]$ . Indeed, the values of  $E[Q(t)]$  and  $D^2[Q(t)]$  of the stochastic process  $Q(t)$  depend on time  $t$ , as the values of  $E(Q_t)$  and  $D^2(Q_t)$  are different for different times  $t$ . Nevertheless, the values of  $E[Q(t)]$  and  $D^2[Q(t)]$  of the process  $Q(t)$  are not random functions, because the values of  $E(Q_t)$  and  $D^2(Q_t)$  are not random variables, but are constant for a given time  $t$  and the given set of random variables  $Q_t$  [4, 5, 16, 21, 27, 28]. A sample realisation of the stochastic process  $\{Q(t): t \geq 0\}$  which reveals the dependence of the expected value  $E[Q(t)]$  and the variance  $D^2[Q(t)]$  on time  $t$  is illustrated in Fig. 13.

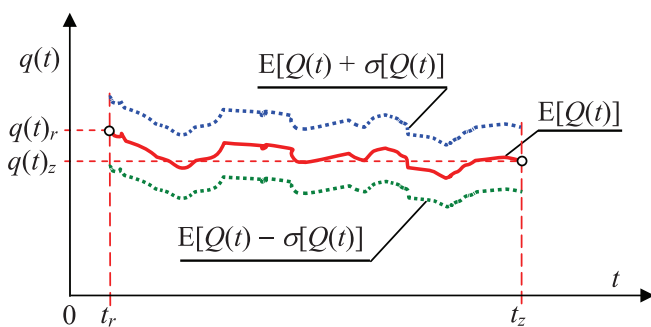


Fig. 13. Sample realisation of the stochastic process  $Q(t)$  illustrating changes of load  $Q$  at time  $t$ :  $Q(t)$  – load at time  $t$ ,  $q(t)$  – realisation of the process  $Q$  at time  $t$ ,  $q(t_r)$  – expected value of the process  $Q$  at time  $t_r$ ,  $q(t_z)$  – expected value of the process  $Q$  at time  $t_z$ ,  $t_r$  – examination start time,  $t_z$  – examination stop time,  $t$  – time as process parameter,  $E[Q(t)]$  – expected value of  $Q$ ,  $\sigma[Q(t)]$  – standard deviation of  $Q$

The load of friction associations in the ship main engine crankshaft and piston system can be considered a combination of mechanical load ( $Q_M$ ) and thermal load ( $Q_C$ ), expressed by the following formula:

$$Q(t) = f[Q_M(t), Q_C(t)] \quad (7)$$

where:

$Q$  – total load of the friction association in a given engine crankshaft and piston system,

$Q_M$  – mechanical load of the friction association,

$Q_C$  – thermal load of the friction association,

$t$  – time of engine action (operation).

Hence, at least two stochastic processes can be considered in empirical load examination, which are  $\{Q_M(t): t \geq 0\}$  and  $\{Q_C(t): t \geq 0\}$ . These processes are components of the vector process  $\{Q(t): t \geq 0\}$  [4, 7, 20, 28].

We can write for the loads  $Q_M$  and  $Q_C$  that at an arbitrary time  $t$  they are vectors with the following components:

$$Q_M : [p_{\max L}, P_L, P_N, P_{TL}, P_{TC}, P_B, N_{TL}, N_{TC}, \dots] \quad (8)$$

$$Q_C : [\dot{q}, \nabla T, t_L, t_C, t_{ol}, \dots] \quad (9)$$

where:

$p_{\max L}$  – maximal bearing pressure (in the oil wedge),

$P_L$  – force carried by the bearing,

$P_N$  – normal force of the piston pressure against the cylinder liner surface,

$P_{TL}$  – friction force in the (main, connecting rod, or crosshead) bearing,

$P_{TC}$  – friction force between the piston with rings and the cylinder liner,

$P_B$  – inertia force acting on the connecting rod bearing,

$N_{TL}$  – friction power loss caused by friction in the bearing,

$N_{TC}$  – friction power loss caused by friction between the piston and the cylinder liner surface,

$\dot{q}$  – density of the heat flux penetrating through the surface,

$\nabla T$  – temperature gradient,

$t_L$  – temperature of the (main, connecting rod, or crosshead) bearing,

$t_C$  – temperature of the cylinder liner,

$t_{ol}$  – temperature of the lubricating oil in the bearing, etc.

It results from relations (8) and (9) that the load imposed on friction associations in each ship main engine crankshaft and piston system depends on many quantities (parameters, indices), therefore it can be considered a stochastic process being a superposition of a number of individual processes (in the simplest case: processes  $Q_M$  and  $Q_C$ ). This interpretation of loads of friction associations in crankshaft and piston systems is convenient for general considerations, but can be of little applicability in practice.

In general, loading of friction associations in the internal combustion engine crankshaft and piston system can be understood as a process, the states of which can be interpreted

as random variables. As seen from relations (8) and (9), this process is a multidimensional process, therefore at each time  $t_k$  ( $k = 1, 2, \dots, n$ ) the following  $n$  random variables:  $Q_1, Q_2, \dots, Q_n$  are to be taken into account. These variables can represent  $n$  different quantities (parameters, indices) of the examined load (for instance,  $Q_1 = p_{\max L}$ ,  $Q_2 = P_L$ ,  $Q_3 = P_N$ ,  $Q_4 = Q_p$ , etc.).

When analysing simultaneously a number of individual properties of the load of friction associations in the crankshaft and piston system, a set on  $n$  random variables composing an  $n$ -dimensional random variable  $Q$  is obtained.

In order to simplify further considerations we can limit our analysis by assuming that the random variable  $Q$  is a two-dimensional variable ( $Q_M, Q_C$ ). Since the measurements are repeated periodically we can assume that the variables  $Q_M$  and  $Q_C$  are stepwise (discrete) random variables. The realisations of these variables are the quantities  $q_{Mi}$  and  $q_{Ci}$  respectively, therefore the realisation of the random variable ( $Q_M, Q_C$ ) is the pair ( $q_{Mi}, q_{Ci}$ ). This variable takes values ( $q_{Mi}, q_{Ci}$ ) with certain probability  $p(q_{Mi}, q_{Ci})$ , which is the probability of simultaneous occurrence of events  $Q_M = q_{Mi}$  and  $Q_C = q_{Ci}$ .

The set of the abovementioned probabilities  $p(q_{Mi}, q_{Ci})$  composes a two-dimensional distribution of the random variable ( $Q_M, Q_C$ ). The distribution  $p(q_{Mi}, q_{Ci})$  meets the following condition [9, 13, 17]:

$$\sum_{i=1}^k \sum_{j=1}^k p(q_{Mi}, q_{Cj}) = 1. \quad (10)$$

The marginal distribution of the random variable  $Q_M$  is the following

$$p(q_{Mi}) = \sum_{j=1}^k p(q_{Cj}, q_{Mi}), \quad (11)$$

while the distribution of the random variable  $Q_C$  has the following form:

$$p(q_{Cj}) = \sum_{i=1}^k p(q_{Mi}, q_{Cj}) \quad (12)$$

It results from these considerations that physical quantities which characterise the load, for instance  $p_{\max L}$ ,  $t_L$  [2, 19, 23], determine both mechanical and thermal load of each friction association in the internal combustion engine crankshaft and piston system. Therefore it is evident that there exists a relation between mechanical load and thermal load of this association. Since both loads are random processes, this relation is expected to be of stochastic nature. Consequently, the following hypothesis  $H_2$  can be formulated for this relation: **there is a stochastic relation between mechanical load  $Q_M(t)$  and thermal load  $Q_C(t)$  of an arbitrary tribological association in the internal combustion engine crankshaft and piston system, because certain variants of one of those variables are accompanied by different variants of the other variable.** Hence the conclusion

that the relation between the loads  $Q_M(t)$  and  $Q_C(t)$  cannot be described using ordinary algebraic equations. This conclusion is obvious, as the load depends on a large number of factors, including those which cannot be measured [18, 19, 23].

The level to which the load  $Q_C$  is determined by the load  $Q_M$ , or  $Q_{m_i}$  ( $i = 1, 2, \dots, n$ ), as independent variables, can differ much. A situation may occur in practice that one independent variable  $Q_M$  almost entirely determines the dependent variable  $Q_C$ . It may also happen that a number of independent variables  $Q_{M_i}$  only minimally affect the dependent variable  $Q_C$  and vice versa, the examined variable  $Q_C$ , as the independent variable affects slightly the dependent variables  $Q_{M_i}$ . For instance, the engine works at low rotational speed and low power output, and simultaneously the crack of the oil line decreases the amount of the lubricating oil delivered to the main and connecting rod bearings. In this situation, despite low load  $Q_M$ , technically dry friction which develops in friction associations leads to the appearance of very high thermal load  $Q_C$  and bearing seizure as a final effect. This example illustrates the need for taking into account the intensity (strength) of the stochastic relation between  $Q_M$  and  $Q_C$ .

The intensity (strength) of the stochastic relation between  $Q_M(t)$  and  $Q_C(t)$  can be assessed during empirical examination using the formula [13]:

$$T_{MC}^2 = T_{CM}^2 = \frac{\chi^2}{N\sqrt{(k_M - 1)(k_C - 1)}}, \quad (13)$$

where:

- $k_M$  – number of variants of the variable  $Q_M$ ;
- $k_C$  – number of variants of the variable  $Q_C$ ;
- $N$  – marginal number of variants of the variable  $Q_M$  or  $Q_C$ ;
- $\chi^2$  – value calculated from the chi-square formula;
- $T_{(\bullet)}^2$  – Czuprow's convergence coefficient.

It can be proved [13] that  $T_{MC}$  takes values from [0,1]. This coefficient is equal to zero ( $T_{MC} = 0$ ) when no relation exists between the values of processes  $Q_M$  and  $Q_C$ , while the value equal to one ( $T_{MC} = 1$ ) testifies to the existence of a functional relation. Generally, the existence of the dependence between random variables  $Q_M$  and  $Q_C$  can be checked by analysing probabilities given by formulas (10) ÷ (12).

Comparing loads of arbitrary tribological associations in crankshaft and piston systems which are recorded in successive working cycles, we can conclude that they are different. Consequently for a given time  $t$ , when examining the load of tribological associations of the engine crankshaft and piston system in particular engine operation cycles for instance at times  $t_1, t_2, \dots$  (Fig. 12), we get different values of this load. Hence, obtaining a certain (expected) value of the load is a random event, because in the same operating conditions existing during the empirical examination the expected value can be recorded or not. That means that the load increments become less and less dependent on each other when the time spread  $\Delta t = t_2 - t_1$  is increased [3, 19, 20, 23].

Based on the above considerations the following hypothesis



$H_3$  can be formulated: *the load of an arbitrary friction association in the internal combustion engine crankshaft and piston system is the process with asymptotically independent increments, because the increasing time spread between the times at which the load is examined (measured) makes its values less and less depended on each other.*

The next property which characterises changes of the load of tribological associations in the engine crankshaft and piston system is that the examined load does not reveal any oriented (monotonic) changes. Consequently we can assume that the peak values of the loads of the above associations appear accidentally. The lack of monotonicity of engine load changes allows us to formulate the following hypothesis  $H_4$ : *the load of friction associations in the engine crankshaft and pistons system is a stationary process, because for a long time interval no monotonicity is observed in the load changes* [4, 6, 12, 19, 23, 26].

Past examinations of diesel engines have revealed that the load of their crankshaft and piston systems is changing continuously in such a way that its values measured after very short time intervals are strongly correlated with each other. On the other hand, when the time spread between load measurements performed on each crankshaft and piston system increases, the correlation between these loads decreases. Hence the load values of an arbitrary crankshaft and piston system measured after time intervals (or times) which are very distant from each other can be considered independent. This property bears the name of asymptotic independence of load values [3, 20]. This asymptotic independence between the load values measured (or calculated) at times  $t_1$  and  $t_2$  (Fig. 12) reflects the fact that the increasing  $\Delta t = t_2 - t_1$  leads to the decrease of the dependence between these values.

The above opinions on properties of loads of friction associations in crankshaft and piston systems of an arbitrary ship main engine can lead to the appearance of new opportunities to define a relation between the wear of these associations and their load, once relevant empirical examinations are performed. This can positively stimulate research activities oriented on analysing loads and wear of tribological associations taking into account their random aspects. Studies in this area have not been undertaken so far, but publications can be found in the literature which suggest that there are grounds for studying this subject [3, 7, 16, 26]. The wear and loads of internal combustion engine crankshaft and piston systems are most often analysed based on a deterministic approach [2, 8, 10, 11, 19, 22, 23, 24, 25]. However, the loads of associations in these systems should be analysed at an arbitrary time  $t$  as random variables  $Q_t$  which compose the stochastic process  $\{Q(t); t \geq 0\}$ , hence there is a need to work out a research plan and methodology taking into account probabilistic aspect of these loads. This will enable to examine the wear of these associations as the effect of their load, taking into account the relation:

$$T_{MC}^2 = T_{CM}^2 = \frac{\chi^2}{N\sqrt{(k_M - 1)(k_C - 1)}}, \quad (14)$$

In formula (14) the wear  $Z_t$  is a random variable which at time  $t$  takes the value  $\bar{z}_t$ , provided that the load  $Q_t$  exists and has the value  $\bar{q}_t$ .

## REMARKS AND CONCLUSIONS

Sea-going vessels can perform transport tasks in conditions of heavy undulation of the seawater surface layer and strong wind counteracting their motion. All this creates unfavourable conditions for operation of tribological associations in ship main engine crankshaft and piston systems, which can lead to extremely high loads and resultant damages of these associations, with sea disasters as possible final consequences.

The load imposed to each tribological association in the abovementioned systems can be considered a random variable  $Q_t$ , with asymptotically normal distribution  $N\left(E(Q_t); \frac{\sigma_t}{\sqrt{n}}\right)$  regardless of the functional form of the load distribution.

When examined at an arbitrary time of its action (operation), the load of each tribological association in the crankshaft and piston system of an arbitrary main engine can be considered a multidimensional random variable. Consequently, the process of loading of each internal combustion engine crankshaft and piston system should be examined based on the assumption that the process is multidimensional and stochastic. Hypotheses are proposed to justify the assumptions that the process of loading of the crankshaft and piston system of an arbitrary diesel engine can be considered: stochastic with asymptotically independent increments, stationary, and ergodic, and that there exists a stochastic relation between its mechanical load and thermal load. The intensity of the stochastic relation between these loads can be assessed empirically using the Czuprow's convergence test.

## BIBLIOGRAPHY

1. Aeberli K.: New high-economy engines for panama containerships and large tankers, Wartsila Switzerland Ltd, Winterthur, September 2007
2. Brun R.: High-speed diesel engines (in Polish). WKiŁ, Warsaw 1973
3. Gercbach I.B., Kordonski Ch., B.: Reliability models of technical objects (in Polish). WNT, Warsaw 1968
4. Girtler J.: Controlling the process of operation of marine internal combustion engines based on the diagnostic decision-making model (in Polish). ZN AMW, nr 100A, Gdynia 1989
5. Girtler J., Kuzmider S., Plewiński L.: Selected issues of operation of sea-going vessels in the aspect of safety of navigation. Monograph (in Polish). WSM, Szczecin 2003
6. Girtler J.: Diagnostics as condition for controlling the operation of marine internal combustion engines (in Polish). Studia Nr 28, WSM, Szczecin 1997

7. Girtler J.: Issue of changes in technical states of a Diesel engine as the results of wear of its tribological systems. *Zeszyty Naukowe Akademii Morskiej w Szczecinie*, 31 (103), 2012
8. Girtler J., Korczewski Z., Mańczak J.: Conception of diagnosing self-ignition engines fed with biofuels in operation conditions. *Journal of Polish CIMAC*. Vol. 5, No 2, 2010
9. Greń J.: *Models and tasks of mathematical statistics* (in Polish). PWN, Warszawa 1968
10. Korczewski Z.: Analysing the potential for application of the phase shift method in endoscopic examination of marine engines. *Polish Maritime Research*. No 1(77), Vol. 20, 2013
11. Korczewski Z.: Endoscopic image processing and analysis of pistons' service failures of marine diesel engines. *Journal of Polish CIMAC*. Vol. 7, No 2, 2012
12. Kozłowiecki H.: *Bearings of piston internal combustion engines* (in Polish). WKiŁ, Warsaw 1982
13. Krzysztofiak M., Urbanek D.: *Statistical methods* (in Polish). PWN, Warszawa 1979
14. Krasowski P.: Pressure and velocity distribution in slide journal bearing lubricated micropolar oil. *Journal of Polish CIMAC*. Vol. 5, No 1, 2010
15. Łosiewicz Z.: Analysing operating conditions of large power two-stroke low-speed ship main engines, and causes and results of damages of their main tribological systems (in Polish). *Proceedings of the VII International Scientific and Technical Conference „Explo-Diesel & Gas Turbine 2014”*. Gdańsk – Gdynia – Karlskrona, 23 – 27 May, 2014 r., on CD
16. Niewczas A.: *Fundamentals of the stochastic model of friction caused wear in application to durability issues of machine elements* (in Polish). *Zeszyty Naukowe, Mechanika nr 19*, Politechnika Radomska 1989
17. Pawłowski Z.: *Mathematical statistics* (in Polish). PWN, Warsaw 1980
18. Piotrowski I., Witkowski K.: *Marine internal combustion engines* (in Polish). Ed. TRADEMAR, Gdynia 1996
19. Piotrkowski I., Witkowski K.: *Operation of marine internal combustion engines* (in Polish). AM, Gdynia 2002
20. Rozanov Ju.A.: *Stacionarnye sluczajnye processy*. Fitmazgiz, Moskva 1963
21. Rudnicki J.: The evaluation of the vibration measurement usability of electronic indicator LEMAG „PREMET C”. *Journal of Polish CIMAC*. Vol. 6, No 2, 2011
22. Serdecki W.: Determination of compression ring wall pressure distribution. *Journal of Polish CIMAC*. Vol. 5 No 1, 2010
23. Wajand J.A., Wajand J.T.: *Medium- and high-speed piston internal combustion engines* (in Polish). WNT, Warsaw 2005
24. Włodarski J. K.: *Operation of marine engines – friction and wear* (in Polish). WSM, Gdynia 1998
25. Włodarski J. K.: *Damages of bearings in marine internal combustion engines* (in Polish). *Wydawnictwo Akademii Morskiej w Gdyni*, Gdynia 2003
26. Girtler J.: Identifying conditions of operation of crankshaft and piston systems in ship main engines (in Polish). Report on completion of the own research project performed within the framework of the research project of the Ministry of Science and Information Technology (no. N N509 494638) entitled: “Decision making based control of the process of operation of crankshaft and piston systems in main propulsion engines of sea-going vessels with the aid of technical diagnostics and taking into account safety and environment protection”. Project manager: Prof. Jerzy Girtler, D.Sc. *Prace badawcze nr 1/10 /PB*. WOiO PG, Gdańsk, 2010
27. Girtler J.: Working out a set of technical and energy states of ship main engine crankshaft and piston systems which are essential for ship safety (in Polish). Report on completion of the own research project performed within the framework of the research project of the Ministry of Science and Information Technology (no. N N509 494638) entitled: “Decision making based control of the process of operation of crankshaft and piston systems in main propulsion engines of sea-going vessels with the aid of technical diagnostics and taking into account safety and environment protection”. Project manager: Prof. Jerzy Girtler, D.Sc. *Prace badawcze nr 2/10/PB*. WOiO PG, Gdańsk, 2010
28. Girtler J., Łosiewicz Z.: Mathematical models of the process of operation of power devices and systems, and verification of these models in the aspect of their applicability in the operating decision making process (in Polish). Report on completion of the own research project performed within the framework of the research project of the Ministry of Science and Information Technology (no. N N509 494638) entitled: “Decision making based control of the process of operation of crankshaft and piston systems in main propulsion engines of sea-going vessels with the aid of technical diagnostics and taking into account safety and environment protection”. Project manager: Prof. Jerzy Girtler, D.Sc. *Prace badawcze nr 3/10/PB*. WOiO PG, Gdańsk, 2010.

29. Joel Donnet News Worlds' Heaviest Ship propeller, built in the Netherlands. Joel Donnet News No 26, Sidebar 1996
30. Joel Donnet News, New World Record for a propeller at Lip's. Joel Donnet News No 58, Sidebar 1997
31. Propulsion Trends in Container Vessels. MAN B&W Diesel A/S, Copenhagen 2001
32. RTA Type Engines, Characteristics of Cylinder Liner and Piston Ring Wear. Service Information Bulletin No DUQ1-02E, Diesel United Sulzer, 1991
33. RTA-C. Technology review. Wartsila NSD Corporation Ltd, Helsinki 2004
34. Service Experience 2007, MAN B&W Engines The ME/ME-C and MC/MC-C Series. MAN B&W Diesel AS, Copenhagen 2007
35. Sulzer RT-flex 60C, Wartsila Corporation, Helsinki 2004
36. Transportation Safety Board of Canada. Marine Investigation Report M00L0034, Main Engine fires, 2001
37. Warsash Maritime Centre. The Learning Resource for Marine Engineers, Southampton Solent University, Southampton 2005

#### **CONTACT WITH THE AUTHOR**

Jerzy Girtler

Gdańsk University of Technology  
11/12 Narutowicza St.  
80 - 233 Gdańsk  
**POLAND**