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AN ANALYSIS OF CHOSEN ENGINE FAILURES OF THE "SEISMIC RESEARCH" TYPE SHIPS

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Abstract

This article presents chosen failures, which occurred on one of the ships with the Diesel Electric type propulsion operating as the "Seismic Research" type ship. The specifics of maritime units requires high reliability of both the engine as well as other machines of the ship power system, which leads to their practically continuous uninterrupted operation. This study deals with the occurring failures of timing gear and piston-connecting rod systems of engines of chosen ships of the "Seismic Research" type.

Keywords: *timing gear system, failure, pitting, diesel engine, seismic research*

1. Introduction

The increasingly rising large scale global power crisis brings about a constant increase of oil prices. This has lead, among others, to intensified searching and mining of this recourse from the sea bed. For the searching purposes, specialized ships of the "Seismic Research" type are applied. Usually, oil searching is currently carried out using the acoustic echo reflected after a "shot" of compressed air. These units belong to the "off-shore" type ships. Each day when the ship is out of service generates loses that no ship owner can afford. However, uninterrupted ship operation is connected with practically uninterrupted operation of its machines.

Specifics of such ship operation and high costs of maintenance make the units stay at sea for up to six months at a time without entering a port.

On most ships, electrical motors of high power, which are driven by electricity generated in the ship electric plant, are used as propulsion. It is the so called "Diesel-Electric" propulsion. This article presents a few non-typical failures noticed on one of the "off-shore" type ships.

Fig. 1. A view of "Seismic Research" type ships [1,2]

2. An analysis of a gear system failure

Four engines of the Bergen Diesel BRG-8 type (with total power of 13 420 KW) were mounted on the discussed unit. Each of the engines drove a separate high voltage current generator of 4 266 kVA.

A cracked cam of the fuel pump was found on one of the cylinders. (Fig. 2) Thus, a decision was made to replace the faulty part of the camshaft. It was feasible thanks to a segment structure of the shaft, which consisted of four parts. The replacement of one of the four segment parts was performed without the help of a specialized service by dismounting the faulty part and mounting a new segment in the same place. Very similar failures occurred on a few sister ships, therefore, the manufacturer concluded that the failure was due to the inappropriately chosen material out of which the cam was made. This lead to a change of record made in the preventive maintenance schedule referring to the frequency of the whole timing gear system surveys of all engines of this type.

Fig. 2. The crack on the cam of the fuel pump (marked with an arrow)

Despite replacing the cams with new ones, the main problem remained. Most probably it was not caused by faulty material but by the miscalculation of the interference size at cam mounting. When cams were being mounted on the shaft, the shrinkage method with an induction heater was applied (Fig.3). An inappropriate construction size of assembly clearance (negative allowance) caused that after the mounting , the stress was too high. Additional stresses on the cam at operation lead to exceeding the acceptable stresses and resulted in piling up the stress especially at the notch which in this case was the splineway.

Fig. 3. Warming up of the shaft gear cam with the use of an induction heater

The increased frequency of surveys of the timing gear system resulted in finding another type of failure, i.e. a very pronounced wear of rollers (Fig.4) and cams of the injection pumps (Fig. 5). The service of Rolls-Royce, which is the manufacturer of Bergen-Diesel engines, did not pinpoint the cause of the roller and injection pump cam wear. It replaced them with new ones, which after a relatively short period of operation got identically worn as those before. The author of this study analysed the changes of the surfaces of the studied rollers and cams. The types of faults pointed to classical pitting which is fatigue wear caused by cyclic interaction of contact stress occurring at superficial layers of elements that are rolling or rolling with a slide. at lubricated contact within the Hertzian contact stress limits. Thus, it is the fatigue wear occurring in the presence of oil.

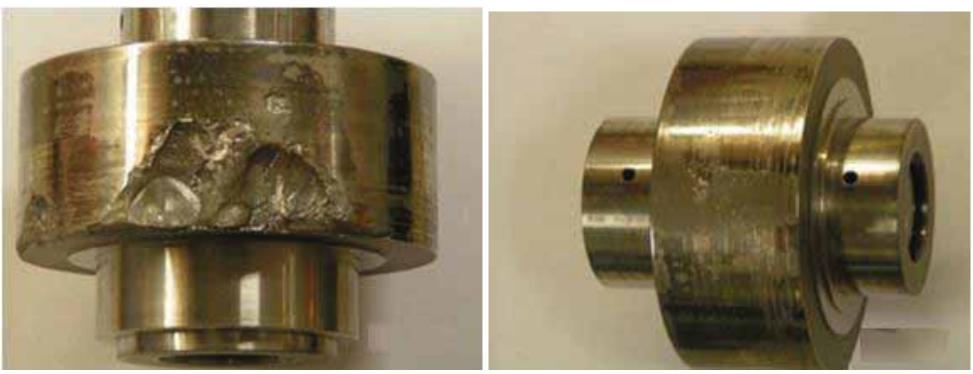

Fig 4. Damaged rollers of a fuel pump

In the first stage, the fatigue was due to the cyclic presence of stress. Initiation of cracks and the appearance of the first micro-crack is usually connected with the place of the highest material stress, i.e. the moveable Bielaiev point [3]. Apart from the possibility of crack initiation at the Bielaiev point, they can also appear on the surface of the material even when the ratio of the tangent force to the axial one is not equal to 1/3. Any surface fault or structure discontinuity may be the source initiating fatigue cracks. During the first stage, lubricating oil slows down the progress of fatigue processes, as it relieves unitary stress in the contact area, which has a beneficial influence on material surface durability. Thus, the effect of wear can only be seen after the system has operated for a period of time. Therefore, the fatigue wear in the presence of oil occurs much later than it is the case for the contact without lubrication [3].

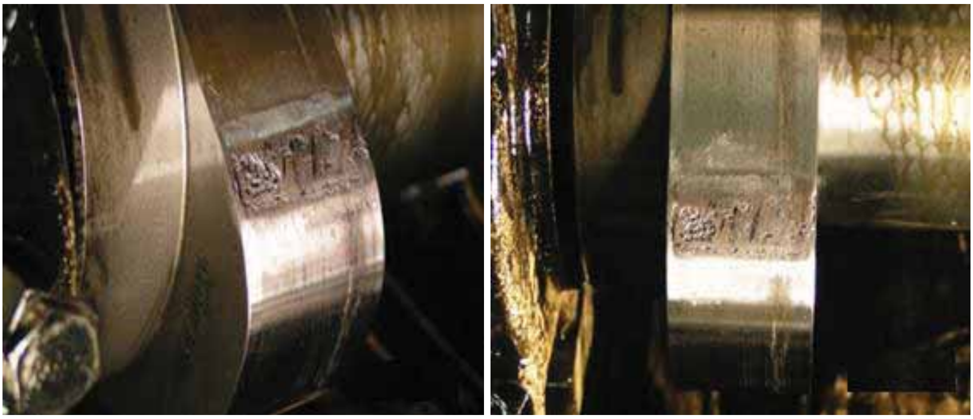

Fig. 5. Damage of a cam

In the second stage, fatigue cracks get bigger and spread to larger parts of the material. Oil plays a pronounced negative part in the process of crack expansion. The oil, which is present in the contact area within the Hertzian contact stress limits is a subject of high unitary pressures and gets into existing surface crevices. **The ability of oil to penetrate them is higher when its viscosity and surface tension are lower. Thus, highly processed oils, containing dispersing additives and detergents are more penetrable** and such oils are used in the lubricating systems of ship engines. Oil as a non-compressible liquid is pumped under high pressure into a crack (at pressure-circulation lubrication) and it acts as a wedge and enlarges the crack. Improvement additives of high physical and chemical adsorption properties, get adsorbed on the crack surfaces and lower the surface energy and decrease the continuity of material inside the crack that was due to adhesion. . Thus, oils with surface active additives are characterized by strong wedge-like activity and because of this play a part in crack spreading [3].
When a roller is rolling on a cam, “surface areas” of both elements are alternately compressed or stretched . The stretched particles, strongly connected to the base, tear some parts of the material, which as a result of fatigue cracking lost or diminished its cohesion with its own specimen.

Analysing the problems connected with wear of elements on the discussed unit and also on twin ships, a conclusion can be drawn that calculations and construction of the whole gear system should be checked again.

3. The failure of the piston – connecting rod system of the Bergen Diesel engine

During seismic measurements of oil resources, a burst of a cylinder block of one of the engines took place. One of the engine systems got almost completely torn apart. It also became the cause of fire in the engine room. Additionally, the so–called secondary damages appeared – not only the elements of the engine but also other machines in the engine room suffered. The base of the connecting-rod got completely severed. (Fig. 6).

Fig 6. The photographs show the severed connecting-rod bases of the Bergen- Diesel engines on one of the "Seismic Research" type ships.

An analysis of the condition of particular elements was carried out in order to find the source and cause of the damage. It was noticed, among others, that the nut of the connecting-rod bolt at the time of the damage was not in its proper position. Attention was focused on the appropriate tightening of the connecting-rod bolts [4]. The area where the nut contacts the connecting-rod showed pronounced deformations, which rather did not appear at the time of the failure. After a complete analysis the following conclusions were reached:

- the main cause of the failure was the insufficient (at the proper time) tightening of the nut of the connecting-rod bolt,.

The subsequent stages of the damage were as follows:

- insufficient initial tightening of the nut caused its further loosening and gradual shearing of the securing nut;
- when the nut was completely loose, additional shearing and stretching forces were acting on the bolt at engine operation, and at the same time the lubricating oil valve was getting opened or closed depending on which direction the piston (TDC-BDC "*Top Dead Centre – Bottom Dead Centre"*) was moving;
- the above were the main reasons of disappearance of the lubricating oil in the cylinder barrel and of the cooling oil at the piston head;
- because of lack of lubrication and piston head cooling, the piston got stuck in its barrel and as a result the whole connecting-rod base was severed, the piston pin cracked and the engine cylinder block burst;
- until the moment when the engine was fully stopped, the spinning crankshaft "threw" parts of the connecting-rod outside.

Conclusions

This article presents atypical kinds of wear (or failures) in a Diesel- Electric engine on one of the "Seismic Research" type ships. The general conclusion which comes to mind when analysing damages occurring on the discussed ship confirms that neither the ship type nor the extensive operation of the engine or systems is the main cause of failures. The basic problem lies in construction errors or the quality of spare parts, or as it was the case in the most expensive failure of the piston-connecting rod system – a mistake at mounting dependent only on human omission (or ignorance). The last one rarely happens in the case of Polish crews. Engineering officers – graduates of Polish maritime academies are equipped in knowledge much more extensive than that of the so called "cheap crews", and the savings that a ship owner may make employing them, might by insignificant in comparison to the consequences of failures and subsequent costs connected with them.

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THE POSSIBILITIES OF DIAGNOSING COMMON-RAIL INJECTION SYSTEMS OF MARINE DIESEL ENGINES

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Abstract

This article presents the application of acoustic stress wave emission for diagnosing Common-Rail injection systems of engines. Failures typical to such systems, as well as problems connected with faulty operation of injectors, have been presented. One of the methods of acoustic emission signal analysis has been discussed. It enables a non-invasive way of diagnosing the injection system of an engine with self-ignition and the Common-Rail system.

Keywords: *acoustic emission, technical diagnostics, Common-Rail injection system, marine diesel engines, injector*

1. Introduction

On the basis of the author's own studies, it can be claimed that failures of marine diesel engines caused by faults connected with the system of fuel supply comprise about 70 % of all failures among the functional systems of the studied engines with self-ignition. Faulty operation of the injection system causes, apart from the engine being out of service, higher pollution of the environment with exhaust gases.

Industrially developed countries have undertaken collective activities whose aim is to diminish the amount of harmful gases coming from combustion of fuel in the studied engines. Introduction of stricter and stricter limits on the content of NO_x , SO_x, solid particles, CO is now, from the practical point of view, the main reason for construction changes of industrial engines.. Present regulations require constant decrease of toxicity of engine exhaust gases. It is still a challenge for many manufacturers of ship engines to meet the *Tier Stage II* and *Tier Stage III* requirements concerning exhaust emissions.

In marine engines with self-ignition the most possible and at the same time an effective method of fulfilling the above regulations is the application of the Common- Rail system (C-R), but also more and more attention is paid to for example dual fuel systems: oil – gas LPG[2] or the systems with the so-called "recirculation of exhaust" EGR (Exhaust Gas Recirculation).

Analysing the solutions of the injection systems suggested by different engine manufacturers offering industrial engines with self-ignition, a tendency confirming the efficiency of the Common- Rail system in reaching the required regulations on exhaust emissions can be observed. Ship engines with the Common- Rail system are introduced (for example the medium-speed engines: Wartsila 46F (Fig. 1),Wartsila 32, Wartsila 38 or the low-speed ones: Wartsila RT-flex50-B, RT-flex58T-B, RT-flex68-D, Rt-flex82C, Rt-flex82T, RT-flex84T-D) with the increased pressure up to 160-180 MPa.

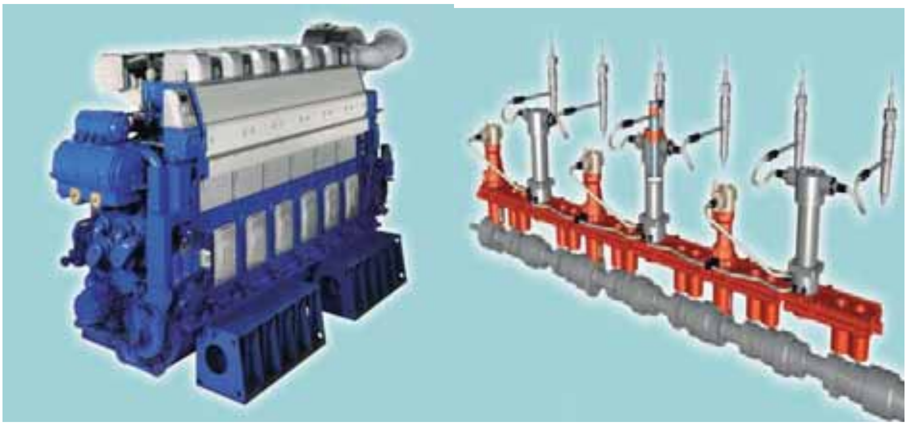

Fig. 1 An overview of the Wartsila 46 F engine with the injection system of the Common- Rail type [1]

2. Typical failures of the Common- Rail injection system

Analysing the formerly presented issues, it is therefore important to study univocally both the wear/damage of the injection system and its direct influence on the quantity and kind of toxic components comprised in the exhaust. Sometimes damage that is fairly inconspicuous to the user and may be of no consequence to the proper functioning of the engine, may be of great importance for environmental protection. Thus, it is an additional reason for the constant search of newer and better diagnostic methods. Typical failures connected with the C-R system and practical methods of dealing with them can be classified in the following way:

- Injector failures. They are divided into two types:
  - electrical damage – relatively easy to detect with the use of a diagnostic programme and performing the injector coil test
  - mechanical damage – possible to detect throughout studying the amount of fuel leaving the injectors. The amount of fuel flowing through the injector is precisely determined by the manufacturer for a certain period of time and a certain rotational speed of the engine(both at operation and as is the case with smaller engines at rotating the starter). The volume of fuel exceeding the ranges stated by the manufacturer indicates a mechanical failure in one or all injectors. Practically the removal of the fault requires a replacement of the whole set of injectors in the engine. Because of high pressures which occur there, in such a case high pressure conduits are replaced, as they cannot be reused after loosening their connector pipes (most manufacturers' recommendation)

- Leakages of the high pressure system – easy to locate throughout visual inpection, as the fuel under high pressure is instantly visible as leaks in the system

- Faults of electrical elements which take part in controlling the fuel pressure such as the fuel pressure detector in the fuel rail, electrical pressure regulator, electrical "stop" valve. Electrical faults are detected using a diagnostic programme of the engine. Mechanical faults are much more difficult to detect and in practice the problem can be solved by replacing the faulty element with a new one.

- Pressure pump failure caused by damage in mechanical elements for pumping, regulating, driving and electrical purposes. Diagnostics of the fault is performed after determining the pressure values in the Common-Rail by using signals obtained from installed detectors and later from the engine diagnostic programme after accepting the assumption that the detectors and engine regulators work properly. Comparing the reference pressure for a given speed or engine load it is possible to determine (for example in the form of a graph) the values of the real pressure (or the lack of it).

3. Diagnosing the Common- Rail injection systems

Obtaining a correct diagnosis is one of the most vital problems that is expected to be dealt with by diagnostic personnel. The user would like to obtain information about the occurring problem quickly and preferably with a 100% guarantee of certainty. Obviously it would be ideal to make a prognosis of a given system state with a very high probability.
When diagnosing very "sensitive" injection systems such as the Common-Rail, the most well-known and generally used by servicing companies are the so-called "on site" methods which focus mainly on "electrical" and "hydraulic" inspection of the injectors. Faults would be relatively easy to detect with the use of an oscilloscope monitoring the correctness of the signals controlling the injector. It is also possible to study the voltage in the piezoelectric injector or the current in the electromagnetic injector. Similarly, using the hydraulic method the "overflow" from particular injectors (or collectively from all of them) can be studied, but unfortunately only after dismounting them. In the case of marine diesel engines the second method is unfortunately without a practical application.

There are, however, the so-called "interim states" of real faults, in which the above mentioned diagnostic methods do not work, i.e. sticking together of the piezoelectric layers or overheating of coils in electromagnetic injectors. As a result a faulty signal is generated (inappropriate regulation) from control channels and in effect the injector operates incorrectly. It may happen that in spite of applying the C-R system which should meet the requirements of the regulations on toxicity of exhaust, due to such faults, the engine emits unacceptable quantities of toxins, not to mention the problems connected with the reliability of the propulsion system itself.

Obviously, other measurements can be performed, like for example capacity measurements in particular piezoelectric injectors (using a typical RCL bridge) or the inductance in electromagnetic injectors, which would enable determination of the values exceeding the given limiting signal and indicating to a faulty operation of the injector. However, application of such methods in not very realistic in practice. It can be claimed that they do not work at all in the case of marine engines. In fact, "laboratory measurement" methods are too expensive and first of all they consume too much time.

4. Elastic waves of Acoustic Emission

In practice, acoustic emission AE is understood [4] as a physical phenomenon and a measurement method. Acoustic emission is defined as “elastic waves caused by energy emission in a material or by a process” [5]. It is based on creation of elastic waves in a material, at the same time this phenomenon is caused by a dynamic local reconstruction of the material. Acoustic emission is also generated as a result of energy emission coming from intermolecular bonding due to deformations, cracking or phase transformations. Energy of acoustic emission wave (AE) also occurs when there is an external cause changing the existing state.

5. Application of Acoustic Emission waves for diagnosing the C-R injection system

Optimization of the fuel system repairs can be reached (taking into account both its time and costs) in the case of injector failures, by applying a diagnostic method detecting typical mechanical faults of injectors of the Common-Rail system. Due to a multi-phase injection and thus the speed of needle opening and the period of injection in one cycle of combustion process, the diagnostic method has to be very accurate and at the same time available and feasible to be used for an engine operating in a machine “on site”. The Common-Rail fuel system requires “ideal” servicing conditions as far as the purity of air filtering in the workshop or maintaining proper humidity and so on are concerned. Therefore, any servicing activities on the engine in its operation site are limited – it especially refers to ship engines. Thus it additionally implies requirements on verification or diagnosing injection elements in such a way that the need for their dismounting would be minimized. The author of this paper used acoustic stress wave emission for studying engines with the Common-Rail injection system. It is usually a high frequency stress wave caused by different external factors [4,5] (thus also by such as fuel flow, opening and closing of injector needle, tribological processes, chemical processes, material structure dislocations and so on).

Figures 2 and 3 show a comparison of spectral density of an acoustic emission signal for a faultlessly operating injector and for a faulty injector, respectively. The faulty injector had de-calibrated injection holes, incorrect period of injection and a scuffing injector needle. In the case of the faultless operation one can distinguish two distinctly enlarged amplitude values connected with the opening and later closing of the injector needle. The frequency of the emitted stress wave important for the process of faultlessly operating injector was about 9 to 12 kHZ (Fig. 2), whereas for the faulty one (Fig.3) – there is a distinctive disturbance of the emitted acoustic stress wave and its shift towards lower frequencies characteristic for the so-called “flowing injector” and a clearly lower value of signal amplitude connected with the pressure of fuel injection.

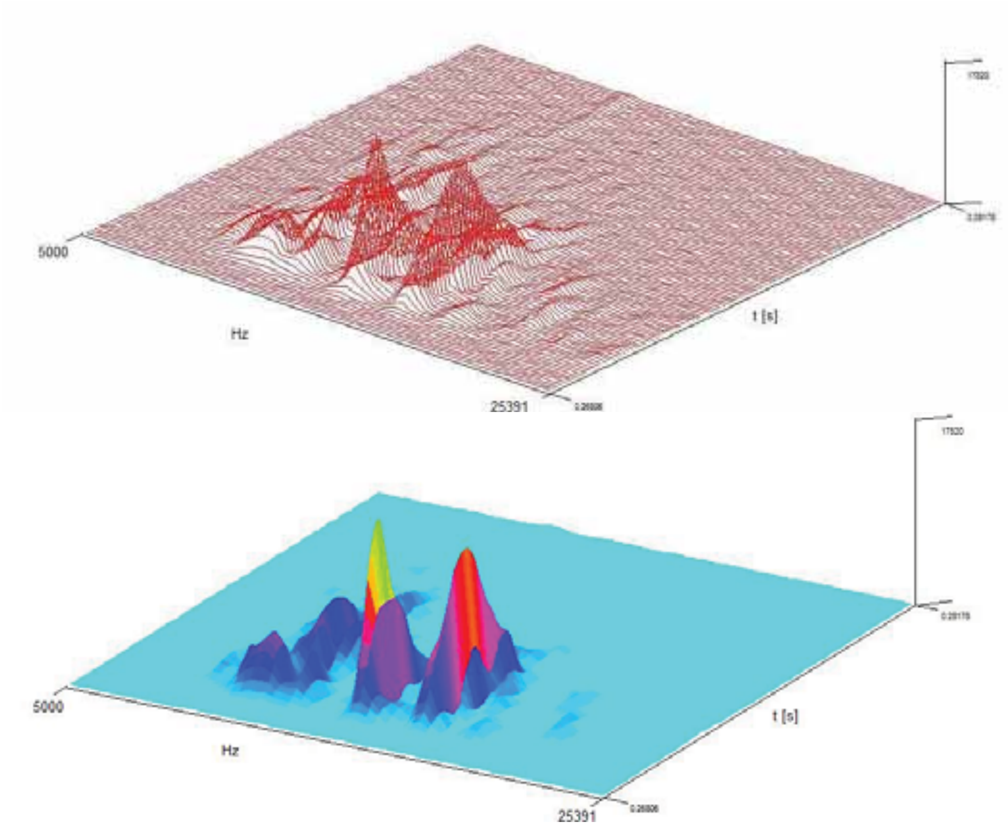

Figure 2. Spectral density of an injection process in a C-R engine for a new faultlessly operating injector (in the function of amplitude – frequency – time)

The flow of fuel through the injection holes, operation of injector needle, generates stress waves of acoustic emission. At a given moment during the injection process, the energy of the AE signal is strictly dependent on the condition of the injector. After a thorough analysis of the signal, one can spot all the changes connected with a particular fault (or wear). A diagnostic method for injection systems in ship engines has been developed for both the classical injection system and for the one with the Common-Rail. At the moment an application form for a patent, which accurately describes the way and the method of analysis for a non- invasive diagnosing of injecting systems, is being prepared. The term “a non - invasive method” means a study with no interference into the injection system. The detector is mounted on the outside available part of the system.

Spectral density presented in fig. 2 and fig. 3 is one of the simple tools which visualizes the difference in injector operation. The occurring change of frequency of the emitted wave is dependent on the condition of the injector.

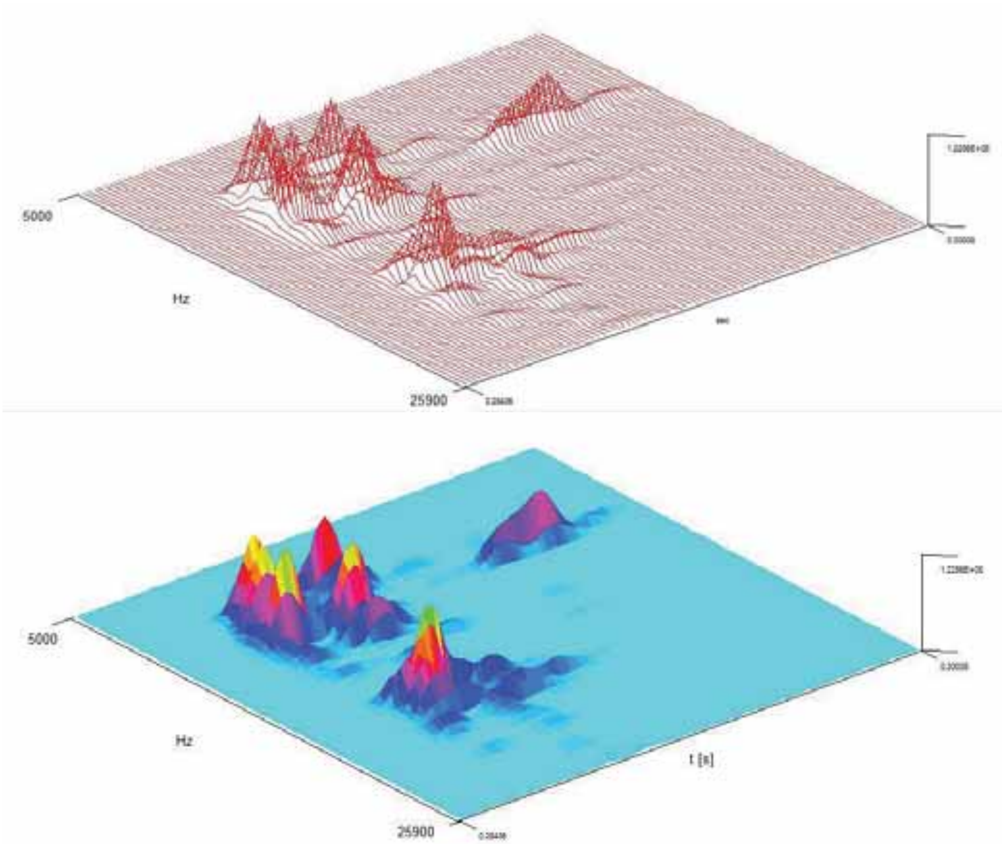

Figure 3. Spectral density of an injection process in a C-R engine for a faulty injector (in the function of amplitude – frequency – time)

Summary

An important value which should be taken into account from the diagnostic point of view in the considered analysed signals connected with the operation of a C-R engine injection system is an accurate extraction of a band of the signal frequency, its amplitude and shape distortion of the stress wave. Using a respective analysis of the acoustic emission signal of the injection process, one can distinguish the previously mentioned "electric" and "hydraulic" system faults. At present, the author of this study is preparing an application form for a patent referring to the application of an acoustic stress wave emission signal. It will also contain a thorough description of particular analyses important for particular failures like for example disturbances connected with the so-called multi-phase injection in engines with the Common-Rail injection system or injection hole blockage with coke.

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ANALYSIS OF DAMAGE OF SELECTED ELEMENTS OF THE INJECTION SYSTEM OF MARINE DIESEL ENGINES

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Abstract

The injection system of a marine diesel engine is one of the most important functional systems, but it also belongs to the most failure prone. This article presents chosen damage of main elements of the injection equipment. The causes of their failures and characteristic effects connected with their wear have been discussed. Problems encountered by ship crews in reference to injection system failures have been considered.

Keywords: *fuel injection system, injector, fuel injection pump, marine diesel engine*

1. Introduction

In the case of marine diesel engines there are tendencies to turning to residual fuels of worse and worse quality, with a high content of impurities. Impurities are one of the main factors influencing states of decreased reliability of injection equipment, causing the wear of its elements and in consequence, they decide on its durability and longevity.

Studies show, that failures of injection equipment are also to a great extent caused by worse quality atomizers and "non-original" spare parts for injecting pumps or injectors (longevity of these elements is almost by half shorter than it is the case for the original ones). However, the most important is bad quality fuel, which is the main cause of maintenance problems at operation of injection pumps, injectors and fuel filters.

2. Chosen failures of the injection system

During the studies carried out by the author of this paper, faulty operation of brand new atomizers occurred (after the ship owner had changed its spare part supplier) which was caused by a badly made atomizer well. It can consequently lead to a change of pressure under the needle and damage the atomizer itself (Fig.1).

Another example of atomizer damage, very dangerous to the engine, which the author of this paper encountered, is breakaway of a whole part of the atomizer body – the needle guide (Fig. 2). It is an example of using very bad quality fuel.

a) b)

Fig. 1. Partial (fig. a) and total (fig. b) breakaway of the atomizer nipple

Change of thermal conditions of atomizer operation, caused damage in the structure of the atomizer body material and as a result its cracking. At first the effect of this failure is usually damage of piston crown (Fig. 3), and later the damage of the whole piston – rod - crank system together with the engine body. Using improper quality of fuel (with too high viscosity) together with the wear of the seat - atomizer needle cone system causing a change of pressure under the needle lead to a failure of an atomizer of one of the engines manufactured by Caterpillar (Fig. 4)

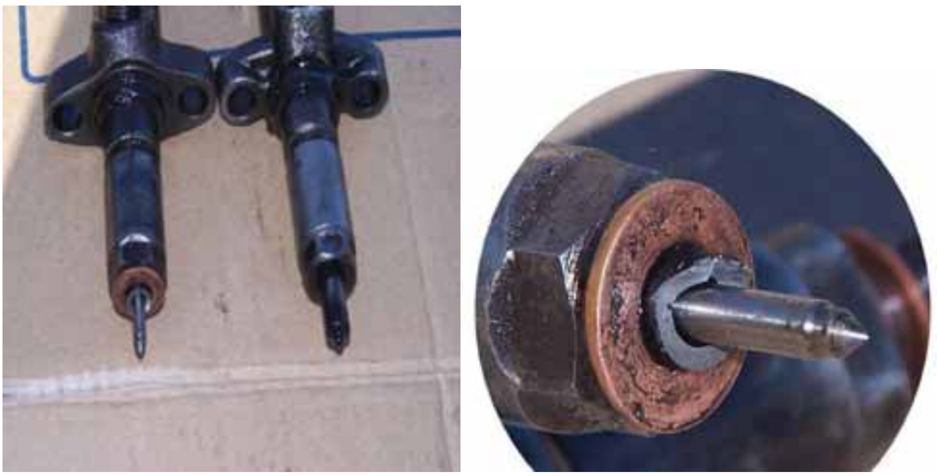

Fig. 2. An example of damage (breakaway) of a part of the atomizer body caused by the improper quality of fuel

Fig. 3. Damage of the piston and engine body being the result of the breakaway of the lower part of the atomizer body (from Fig. 2)

Fig. 4. A view of the damaged atomizer in a an injector pump of the Caterpillar engine

Interesting wear which the author had a chance to consider was the damage of directional multi-hole fuel atomizers of the Sulzer RTA 58 engine. After a relatively short period of injector operation, atomizers of one of the systems suffered from wear visible in Fig. 6. The ship owner decided to replace the atomizer with new ones, made by renowned manufacturers, which did not, however, bring expected results.

Fig. 6. Erosion-cavity formation damage together with local overheating (scorching) of the atomizer in a Sulzer RTA 58 engine

Analysing the kind of wear and the engine operation and maintenance manual the author of this paper came to a conclusion that when fuel flows from the injector, gas turbulence appears in the combustion chamber causing erosion- cavity wear on the opposite site of the holes. Also, local overheating of the atomizer takes place. The cause of this lies in the too pronounced wear of a part of the head (fig. 7). In practice a similar effect could occur due to the crew error in the case of a too deep milling of the injector seat in the head during cleaning. It causes deeper seating of the injector and its more pronounced protruding form the combustion chamber. It is the main cause of turbulence appearance which is responsible for the cavity erosion on the surface of the atomizer.

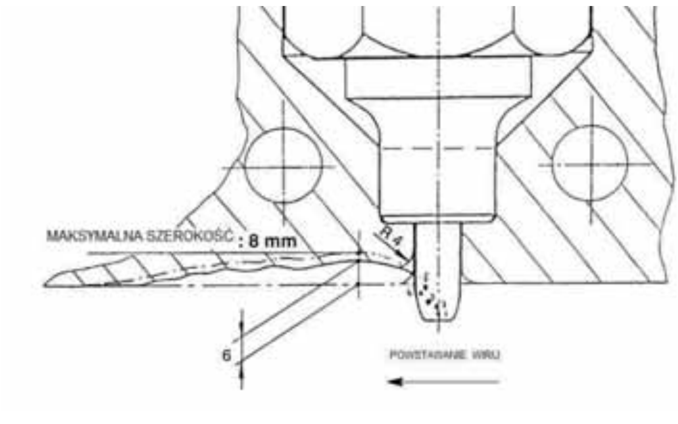

Fig. 7. A view showing the wear (scorching) of the inside part of the head of the Sultzer RTA 58 engine which causes the "cavity turbulelence" and atomizer damage as shown in Fig. 6

In the Sulzer RTA engines, very close to the lower part of the head (from the combustion chamber side) there is also a cooling duct. A decreased distance (because of the needle material scorching) between the cooling area and the combustion chamber causes much lower heat reception also from the atomizer tip itself.. In the studied case two phenomena interfered and they both lead to the studied kind of wear; cavity erosion and local overheating.

Similar overheating may also appear in the case of incorrect setting of the fuel injection commencement timing. When fuel injection comes too early (it is most often caused by an inappropriate setting of the injection pump for example after shipyard overhauls), it leads to the so-called knock combustion. Its effect is also a higher than normal maximum pressure of the combustion process.

In the case of new atomizers, the quality of fuel jet mainly depends on compression pressure (counter pressure in the combustion chamber) and the pressure under the needle in the injector. In one of the engines (Wartsila SW 280) a problem of incorrect combustion with high emissions of smoke occurred. For the crew the problem was so important that earlier they had replaced precision pairs of injection pumps with new ones, also they replaced atomizers together with the spring and they checked the supercharging pressure and all other parameters which could lead to increased emission of smoke. After preparing a developed indicator graph, it was observed that the maximum pressure had a “dual character” – there was an increase, a decrease and then another increase of pressure, this time main in character, until its maximum value was reached (Fig. 8) It is the case when an incorrectly sprayed fuel jet reached the hot surface of the cylinder barrel, initial fuel combustion took place, after which the main combustion process occurred. As it was mentioned before this effect is influenced by compression pressure and by the fuel pressure under the needle (with faultless atomizer holes). Accordingly, it was suggested that if the compression pressure was correct, just as other parameters, then probably “smoke emission” and “dual effect of the maximum pressure of the combustion process” could probably be ascribed to a faultily manufactured injector well. In fact, it turned out that the company supplying atomizers had changed its sub-supplier and thus a batch of **new** atomizers delivered onto the ship was faulty. The problem of brand new faultily manufactured parts is unfortunately becoming more frequent.

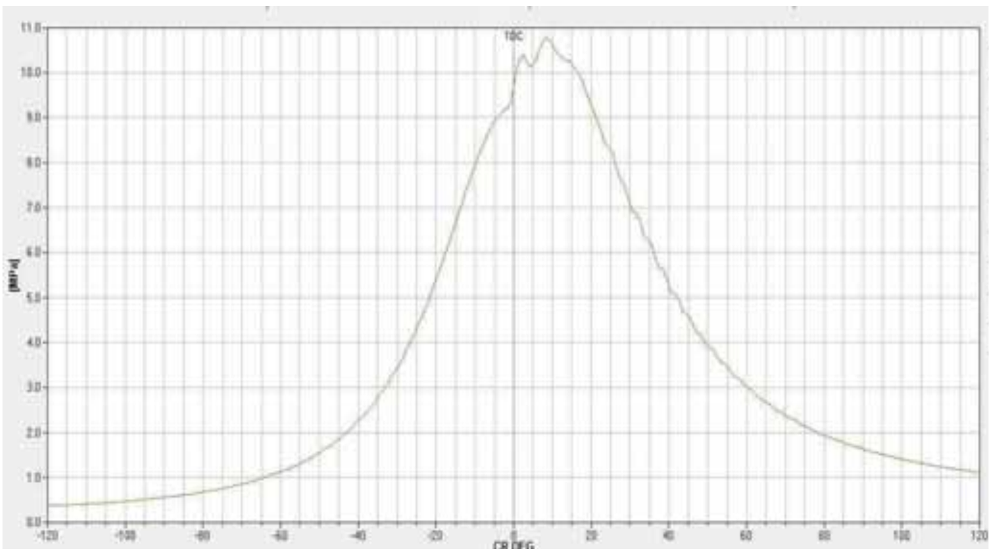

Fig. 8. A developed graph of an injection process run for a faultily manufactured atomizer well which changes the quality of fuel jet (description in the text).

Getting a correct atomization in the whole volume of the combustion chamber is possible when all the atomizer holes have equal ability of fuel flow capacity. Foul atomizer holes are the cause of far worse fuel atomization, decrease in engine power, and increase in unitary fuel consumption. The unburned fuel and carbon deposit increase the friction effects of the piston/ring in the barrel, as particles of fuel sediment on the piston and cylinder barrel, they also "wash out" the cylinder oil layer, while the coked particles of fuel directly increase the wear between the piston and the cylinder barrel. It can be concluded from the author's own studies that the problem of equal "flow capacity" of all atomizer holes is not only the result of atomizer wear. The problem is often valid for brand new atomizers delivered onto the ship (not to mention the repaired or regenerated atomizers).

Engine smoke emission caused by the change of fuel jet was observed when performing diagnostic measurements of a marine diesel engine of the MAN D2848 type. This engine belongs to the group of the so-called "difficult to diagnose" engines. It operates in the fork-like system of cylinders and it does not have indicator taps. The problem was due to the change of pressure under the needle caused by scorching of the atomizer needle cone (Fig. 9). It was important that the scorching took place during the guarantee period (just after a short period of operation time). Again it was caused by bad quality of atomizer parts.

Figure 9 Scorched atomizer needle cones.

Typical wear of injection pump leads to erosion and cavity formation action of fuel on the overflow hole of the pump (Fig. 10).

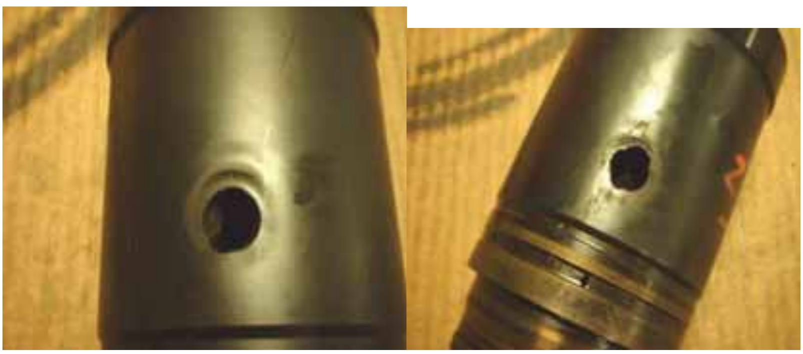

Figure10 Erosion - cavity formation of overflow hole of the injection pump cylinder

At the moment the failures caused by low quality spare parts are more common. Figure 11 shows seizing of a precision pair of an injection pump after about only four hours of operation.

Fig. 11. Seizing of a precision pair of a (new) injection pump after four hours of operation due to very low quality of manufacturing of parts (Chinese manufacturer)

Summary

In the case of elements of the marine engine injection systems, two groups of failures can be observed: the first ones are caused by bad quality fuel, the second ones- by inadequate quality of system parts. The cases analysed by the author show that renowned ship engine manufacturers, considering the economic side of purchasing new parts, more often replace their sub-suppliers with cheaper ones, endorsing the supplied parts. This is for example the case for both MAN as well as Wartsilla.

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DIAGNOSTICS OF MARINE PROPELLER SHAFTS

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Abstract

This author provides a description of the construction and function of a ship's propeller shaft, and states that the methods and resources for diagnosing a marine propeller shaft are insufficient. It is underlined that the diagnostics of machines mounted in plain bearings successfully makes use of measurements of the shaft journal centre trajectory. An attempt has been made to transfer this kind of diagnostics to the field of marine propeller shafts. A physical model of a propeller shaft was built in a test stand ROTOR KIT OIL WHIRL/WHIP OPTION made by Bently Nevada. The trajectory of a shaft journal centre and its maximum radius-vector were examined. The need to develop this method of diagnosing propeller shafts has been confirmed.

Keywords: *vibration diagnostics, marine propeller shaft, trajectory of the shaft journal centre*

1. Introduction

The propeller shaft is an important element of the marine propulsion system. What characterizes the propeller shaft is that it is supported in the stern tube bearing, it transfers the torque from the engine to the propeller and the axial forces from the propeller to the thrust bearing, and it runs through the hull. Set at the end of the propeller shaft is a screw, or propeller. As a rule, the stern tube bearing is a slide bearing lubricated with a liquid, mostly oil. The propeller shaft has to be sealed to prevent water from mixing with the lubricating oil, or oil mixing with sea water, finally, oil and sea water from getting into the ship.

The operation of propeller shaft components leads to their wear. The shaft undergoes tribological wear at places it co-acts with seals and sleeves. Corrosion and abrasive wear affects the shaft on the surface of its contact with the propeller, so that it can even suddenly break at this place. Seals mostly undergo abrasive wear and aging. When a seal is damaged, it results in oil leaks. When water leaks into the oil, tribological wear of shaft journals and bearing sleeves is accelerated. Tribological wear results in damage of the propeller shaft and the propulsion system failure, which in bad weather may lead in extreme cases to the sinking of a ship. Other causes of propeller shaft failures include ship's hull deformation, and changes in the position of sleeves in relation to the journals of the propeller shaft.

Classification societies consider propeller shafts as very high-risk devices. To enhance ship's safety, propeller shafts are periodically surveyed. Condition monitoring of these shafts is even recommended, which yields a bonus consisting in every second complete survey being replaced by a simplified one.

2. Required and recommended methods of propeller shaft condition assessment

The propeller shaft is a smooth shaft with a flange or journal on one end, which is used for mounting a coupling connecting the propeller shaft with the drive. On the other end the shaft has a collar or conical profile facilitating the mounting of a propeller. As a rule, the propeller shaft is supported by two or three hydrodynamic slide bearings. One bearing block (additional bearing) is fixed to the foundation bed in the ship's double bottom, while the other two bearing blocks (main bearing of propeller shaft) – aft stern tube bearing and forward stern tube bearing - are placed at the ship's stern frame or post. As standard, the stern tube is provided with forward and aft stern tube seals of the lip ring type having three lip rings in the aft seal and two lip rings in the forward seal.

For slide bearing important is proportion between bearing sleeve length *l* and shaft journal diameter *d*.

In compliance with classification societies rules:

- for aft stern tube bearing l/d=1,5,
- for forward stern tube bearing l/d=0,5.

Date of the example propeller shaft:

- distance between the stern tube bearings – 9057mm,
- diameter of the stern bearing journal – 589mm,
- clearance in aft stern tube bearing – from 0,8 to 1,08mm,
- reaction in aft stern tube bearing in operating condition – 246,4kN,
- kinematic viscosity of lubricating oil at 100°C – 11,3 $\left[\frac{mm^2}{s}\right]$ (Marine Oil Gulfmar AC 307).

The value of the relative clearance in slide bearings is chosen depending on the material of the bearing sleeve, load and revolutions per minute. The literature [6] includes reports that the relative clearance in bearings with sleeves made of white metal should range from 0.4 to 1‰, while for sleeves made of plastic 1.5 – 10.0‰.

Overhauls make up the basic methodology of propeller shaft condition assessment. Overhauls, however, require that all elements of the propeller shaft assembly be accessible for the evaluation of their structure and geometry. To this end, non-destructive tests are carried out (mostly visual, penetrating, magnetic-powder and ultrasound) and measurements of geometrical dimensions. An unlimited access to the propeller shaft assembly components requires that the propulsion is stopped, the ship is docked, the propeller is removed, its seals and stern bearing are dismantled, and the shaft is taken out.

Besides, classification societies recommend certain assessment methods that go beyond technical diagnostics. According to the PRS [1] diagnostics consist in:

- measurements of sleeve temperatures at points regarded as the most loaded,
- sampling the oil lubricating the stern tube bearing for analysis.

On the basis of energy conservation law one can explain that the temperature gradient – a measure of change in the sleeve internal energy – is connected with the intensity of tribological processes taking place in a slide bearing: it indicates that energy is accumulated and that there is a danger of converting this energy into the work of destructive processes that result in bearing damage. The diagnostics using this method are called thermal diagnostics.

Based on convective diagnostics, analysis of an oil sample can deliver a range of information about:

- technical condition of the oil – the third element of a tribological node;
- condition of the other components of the propeller shaft: journal, sleeve and seal.

A disadvantage of thermal diagnostics is that they indicate the intensity of destructive processes as they happen at the moment of observation, even with some delay resulting from the inertia of the measurement path. Convective off-line diagnostics provide information with delay needed for taking samples, delivering them to a laboratory, analysis and sending the results back.

3. Developments in methods and resources of propeller shaft diagnostics

Condition monitoring of propeller shafts consists of diagnosing three critical units: propeller – shaft, seal – shaft journal, and bearing sleeve – shaft journal.

Diagnosing of seals comes down to detection and measurements of oil leaks [2]. The existing methods and resources used for diagnosing propeller shaft seals are insufficient. There are reports on attempts to detect water leaks into the oil, where detectors are placed in the sealing.

However, available publications on the subject do not include any methods for diagnosing the propeller – shaft assembly. To assess its condition, the assembly has to be dismantled for the examination of the structure and geometry of the shaft journal and the propeller.

For the inspection of 'land-based' machines where shafts are supported by plain bearings diagnosing is successfully performed based on the trajectory (position) of rotor shaft journal centre. The measurement principles are laid down in standard [3] and numerous publications, e.g. [4, 5]. Both fixed and portable systems for measurements and monitoring of shaft vibrations are available. The position of the shaft centre within the measurement plane depends on the position of supports, bearing capacity and the load acting on the rotor. The capacity of a plain bearing for a given rotation speed (rpm) depends on the technical condition of the bearing (journal and sleeve) and of the lubricant. The rotor load consists of the working load, its own weight, load due to rotor unbalance and load due to misalignment of the motor and power receiver shafts. The shaft centre trajectory is also affected by dynamic properties of the rotor and bearing, including the lubricant [4, 5, and 6]. We may draw conclusions on changes in machine loads and technical state from changes in the centre trajectory, its shape, dimensions and direction of displacement.

There are grounds to think that this type of diagnostics can be also used for examining propeller shafts. In this case condition monitoring would consist of measurements and analysis of the journal centre trajectory, position of the journal centre inside the clearance circle and the changes of clearance circle. We may consider conclusions that condition of stern tube bearings, its seals and propeller will have an important influence on centre trajectory of aft stern tube journal. While the seals influence can have double impact [7]:

- reaction force between shaft and its seals have an effect on trajectory, the bigger eccentricity the bigger the reaction force,
- at too big radius vector (too big eccentricity), the seals can loose the expected sealing effect it means tail shaft can reach non-operational or unserviceable state.

4. Examination of the propeller shaft journal centre trajectory

Transverse vibrations of a propeller shaft model were examined. A propulsion system model including a propeller shaft is shown in Fig. 1. The physical model was built in a test stand ROTOR KIT OIL WHIRL/WHIP OPTION from Bently Nevada. The propeller shaft, 10mm in diameter, connected through a flexible coupling with the shaft of an electric motor, was supported by two journal bearing blocks, one hydrodynamic (aft stern tube bearing) and the other self-lubricating. The journal diameter in the hydrodynamic bearing was 25mm. At four points distributed on the hydrodynamic bearing circumference a lubricating liquid Chevron GST Oil 32, was supplied from an autonomous lubrication system (without cooling radiator). Two eddy-current sensors were mounted in the sleeve, perpendicular to the bearing axis, according to standard [3]. At the shaft end, apart from bearings, was a disk with a mass of 800g (heavier than the shaft mass) simulating a propeller. As the sealing is an integral part of the propeller shaft assembly, action of the aft seal was simulated by an elastically supported rolling bearing. The rolling bearing was fixed with an inner ring on the shaft, while the outer ring was stretched with four springs placed in a frame, which in turn was mounted on a common foundation bed of the test stand. The rolling bearing axis overlaps the axis of journal bearings of the propeller shaft, and the spring tension is approximately the same as the tension of the sealing rings. Another function of the elastic support was to prevent the direct contact between the shaft and the sleeve. This is due to the fact that in the test stand used, the hydrodynamic bearing sleeve, on account of the working principle of eddy-current sensors, is not made of antifriction metal.

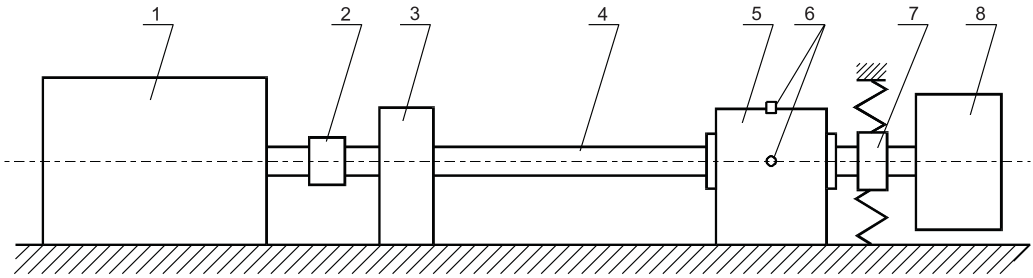


Fig. 1. Physical model of a propeller shaft: 1 – driving motor, 2 – coupling, 3 – plain bearing (self-lubricating), 4 – shaft, 5 – hydrodynamic plain bearing, 6 – eddy-current sensors, 7 – rolling bearing with elastic support (sealing), 8 – rotor disk (propeller)

The test parameters: absolute clearance of 0.35mm and the shaft journal diameter of 25mm gave a relative clearance of 14‰. The distance between bearing blocks was 340mm.

The Sommerfeld number was chosen to match rotational speed - revolution of actual shaft model. The results of calculation are presented in table no. 1.

Table 1. Determination of Sommerfeld number for actual tail shaft bearing and model tail shaft bearing

Quantity	Example of actual tail shaft	Model of actual tail shaft
b/d (bearing length / journal diameter)	1.5	25/25 = 1
Load of stern tube bearing	246.4kN	(0.800 + 0.338kg) x 9.81= 0.011kN
Relative clearance ‰	min = 0.8/589 = 1.36 max = 1.08/589 = 1.83	14.0
Square clearance	Min=1.8496 Max=3.3489	196
Bearing length x journal diameter *bd* $[m^2]$	0.883x0.589=0.520	0.025x0.025=0.000625
kinematic viscosity η at 100°C $\left[\frac{mm^2}{s}\right]$	11.3	5.2
Bdη	0.52x11.3=5.876	0.000625x5.2=0.00325
Sommerfeld number for min. clearance Sommerfeld number for max. clearance	$\frac{1}{\omega}$ 77.6 $\frac{1}{\omega}$140.5	$\frac{1}{\omega}$663.4

According to wide literature sources on this subject for example [6], slide bearings have the same position of journal in sleeve and the same friction coefficient when they are similar it means they have the same relative length *b/d* and the same angle of contact of bearing journal and sleeve and the same Sommerfeld number.

Bearing sleeves of the actual tail shaft and the model tail shaft are closed type and therefore, they have the same angle of contact of bearing journal and sleeve - for closed sleeves angle of contact is equal to 2Π (360°).

Sommerfeld number S_o is given by the following formula:

$$S_o = \frac{P\psi^2}{bd\eta\omega}$$

where:

P – load,

ψ – relative clearance = bearing clearance *s* / bearing diameter *d,*

b – sleeve length,

η – kinematics viscosity,

ω – journal angular velocity in relation to sleeve.

According to [6] for *b/d*>1 increase of *b/d* value is not creating, for the same Sommerfeld number significant changes in eccentricity. We may draw conclusions that the bearing in model shaft will hold the same eccentricity as in the actual tail shaft if Sommerfeld numbers hold the same values.

Permissible clearance changes in bearing (bearing wear) can cause changes in Sommerfeld number in range of 1/ω(77,6 – 140,5). Sommerfeld number in bearing in model will be equal to Sommerfeld number in actual bearing if the model shaft revolution is *k* times bigger than rotational speed of actual shaft. The substitution of data from table 1 *k* is vary from 663,4/77,6 to 663,4/140,5 (from 8,5 to 4,72). Because rotational speed of actual shafts is vary from 60 to 170 rev/min, it gives after calculation that shaft revolution during researches should be in range of 283 till 1445 rev/min:

- in range of 283 to 802 rev/min model bearing remains in the same way as actual bearing with max clearance,
- In range of 510 to 1445 rev/min model bearing remains in the same way as actual bearing with min clearance.

For the purpose of wider view, the tests were made in the range of 0 – 2500 rev/min. The displacement, i.e. trajectory of the journal centre was examined by measuring the maximum radius-vector. The trajectory was examined using a system of eddy-current sensors and a digital real-time oscilloscope TDS 210. The radius was measured by the eddy-current sensors combined with a WIBROPORT 41 system. The TDS 210 oscilloscope allows to filter the signal by using the in-menu 'coupling' function and setting DC or AC: DC passes both AC and DC components of the input signal; AC blocks the DC component of the input signal.

The following issues were examined:

- effect of sealing and rotation speed on the position of the trajectory inside the clearance circle, Figs. 2, 3, 4, 5;
- maximum radius-vector depending on the rotation speed, Fig. 6;
- maximum radius-vector $\mathbf{S_{max}}$ for a selected speed *n* depending on additional mass *m* of the disk, Table 2. The mass was added on the disk circumference at two opposite points φ = 0° and φ = 180° (position φ = 0° was chosen in random way).
- effect of the 'sealing' on shaft centre trajectory image for a speed close to the resonance speed of the propeller shaft, Fig.6;
- effect of the intensity of the lubricant flowing through the bearing for a selected rotation speed, Fig. 8.

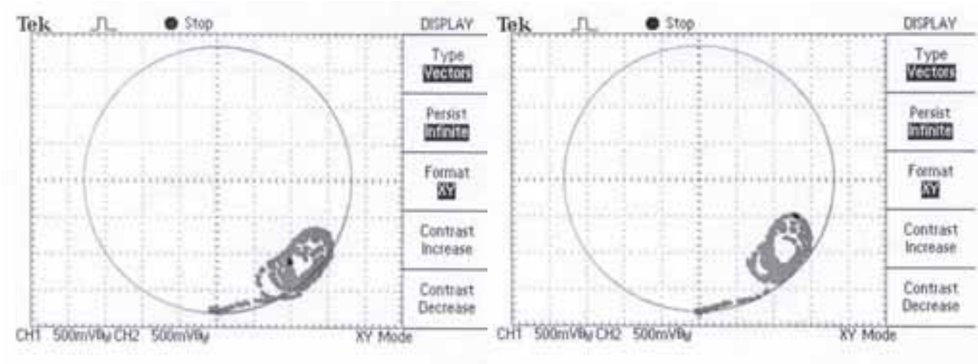


Fig.2. Trajectory of the journal centre for a shaft for $\dot{m}_{\text{oil}} = \max$ *at transition from 0 to 265 rev/min without 'sealing'(left)and with 'sealing'(right)*

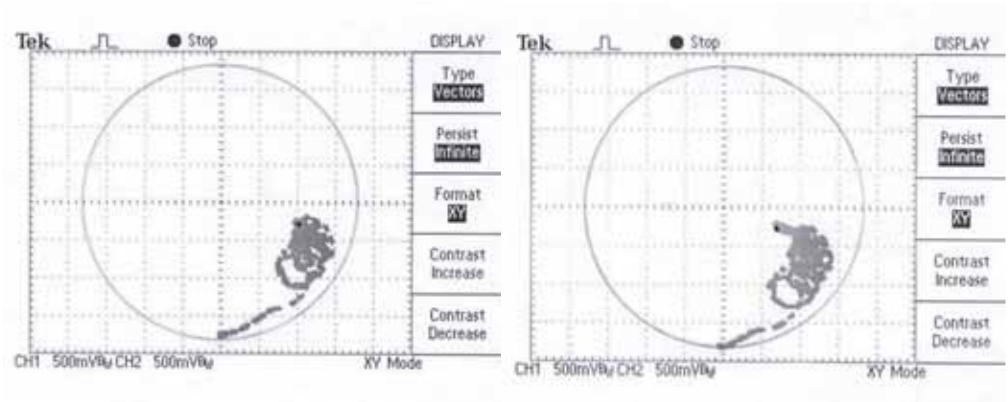


Fig. 3.Trajectory of the journal centre for a shaft with 'sealing' for $\dot{m}_{\text{oil}} = \max$ *at transition from 0 to 720 rev/min (left) and at transition from 0 to 1300 rev/min (right)*

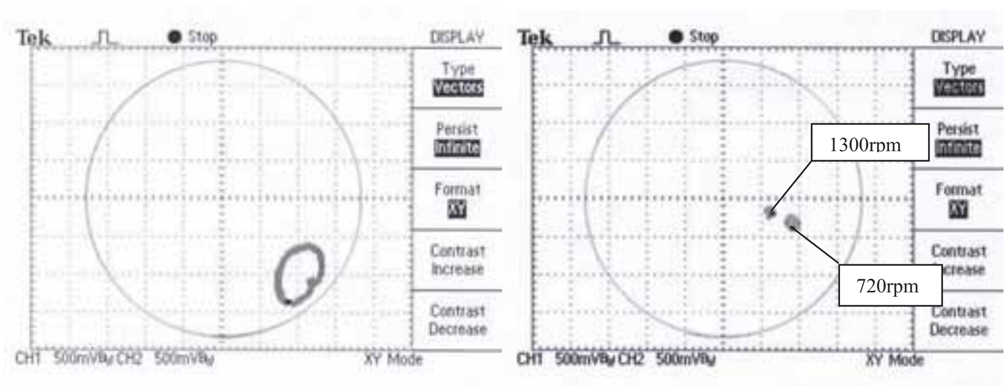


Fig.4. Trajectory of the journal centre for a shaft with 'sealing' for $\dot{m}_{\text{oil}} = \max$ *at: 260 (left), 720, 1300 (right) rev/min*

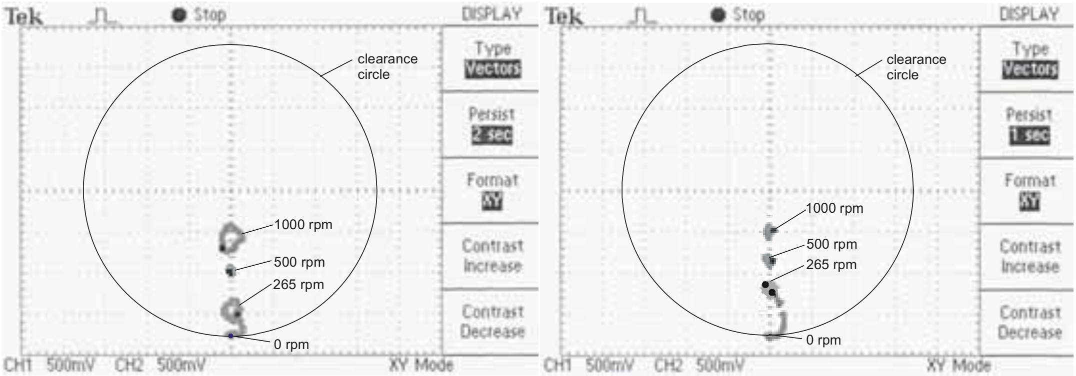


Fig.5. Trajectory of the journal centre for a shaft without 'sealing' for $\dot{m}_{\text{oil}}$ = max *at: 0, 265, 500, 1000 rev/min (left) and trajectory of the journal centre for a shaft with 'sealing' for* $\dot{m}_{\text{oil}}$ = max *at: 0, 265, 500, 1000 rev/min (right) (the DC component of the input signal in the horizontal direction is blocked)*

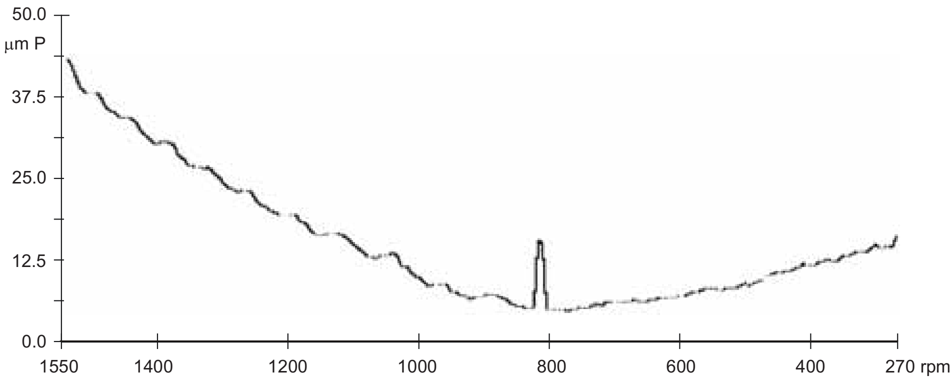


Fig. 6. $s_{\max} = f(n)$ *for n from 1550 to 270 rev/min*

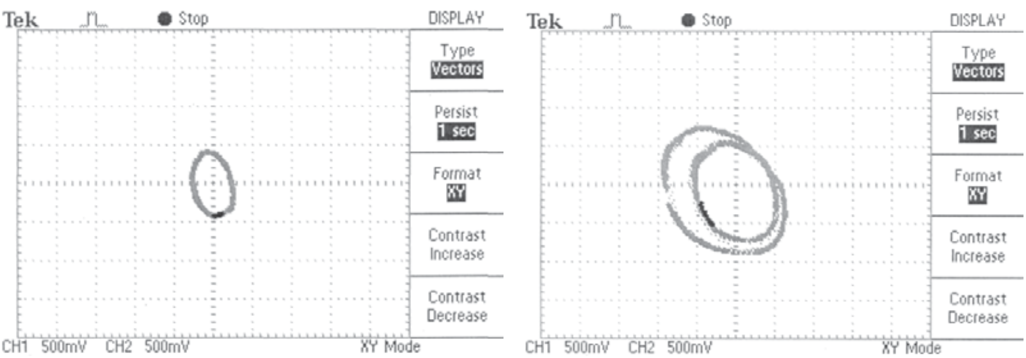
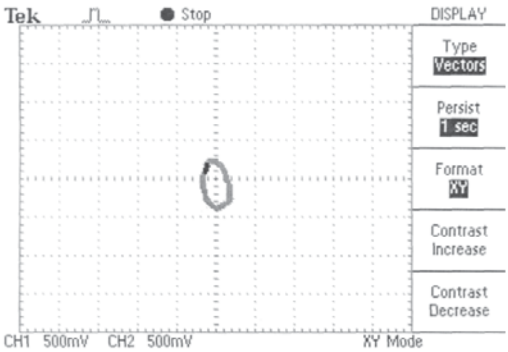
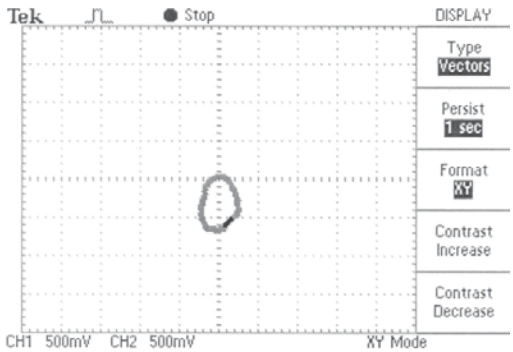


Fig.7. Trajectory of the journal centre for a shaft with 'sealing' (left) and without sealing (right) for $\dot{m}_{\text{oleju}}$ = max *at n = 1610 rev/min (the DC component of the input signal in the horizontal direction is blocked)*

a)



b)



c)

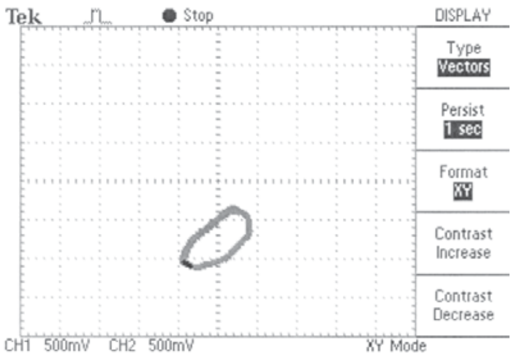


Fig.8. Trajectory of the journal centre for n = 1500 rev/min at various oil flow intensities: a) maximum, b) reduced by one revolution of the reduction valve, c) reduced by two revolutions of the reduction valve (the DC component of the input signal in the horizontal direction is blocked)

Tab. 2. Maximum radius-vector $\boldsymbol{S_{max}}$ *for a selected speed* ***n*** *in depending on additional mass* ***m*** *of the disk*

φ = 0 [°]			φ = 180 [°]		
n [obr/min]	m[g]	S_{max} [μm]	n [obr/min]	m [g]	S_{max} [μm]
1001	0	8,34	1001	0	8,34
1009	0,1	8,73	1003	0,4	7,50
1004	0,2	8,66	1006	1,0	5,50
996	0,4	9,51	1004	2,0	6,89
1005	2,0	15,0			

5. Analysis of the results

The following conclusions can be drawn from the journal centre displacements:

– for the rotation speed 0 rev/min, when the journal generatrix contacts the sleeve generatrix, the journal centre lies on the clearance circle line. As the rotation speed increases, initially the shaft journal rolls sliding on the sleeve, then lifts and remains in contact with oil film formed between the journal and the sleeve. The journal centre first travels on the clearance circle itself, then moves towards the circle inside. The higher the speed is, the thicker oil film forms and the closer the journal centre gets to the clearance circle centre, Fig. 2, 3, 4;

– similar effects are observed when the DC component of the signal in the horizontal direction is blocked, Fig. 5;
– for a preset rotation speed, the journal centre trajectory in a general case is a figure resembling an ellipse, with dimensions (e.g. maximum radius-vector) that get smaller as the bearing work is more stable (fixed load and better working conditions), see Figs. 4, 5;
– the position of the trajectory shows a sufficient difference between the minimum and maximum clearance (Fig. 4: 720 and 1300 rev/min., similarly in Fig.5: 500 and 1000 rev/min);
– additional elasticity, other than the one resulting from oil film elasticity, has a stabilizing effect on the trajectory of shaft journal, see Figs. 2, 5, 7. At the given revolution the average value of trajectory is not undergoing the significant changes, but the instantaneous value of trajectory is undergoing the significant changes, Fig. 7. During the start-up additional elasticity facilitates the formation of oil film and reduces friction, Figs. 2, 5;
– dimensions of the journal centre trajectory depend on the journal rotation speed, Fig. 6. Each rotor with one high mass has at least one significant frequency of free vibration. When the rotation speed is equal to the free vibration the bearing loses its stability, and the journal centre trajectory reaches the dimensions of the clearance circle. It was found during measurements that in the examined model of propeller shaft the free vibration frequency (with additional stabilizing elasticity and at maximum intensity of lubricant flow) corresponds to 1680 rev/min. One characteristic of plain bearings is that at 1/2 resonance speed the so called oil whirl appears, causing an essential increase in the rotor vibration amplitude, which translates into increased values of journal centre trajectory dimensions. In Figure 6 the effect of oil whirl is visible at a speed over 800 rev/min;
– amount of oil flowing through the bearing significantly affects the journal centre position and trajectory dimensions in stable working conditions. When the oil flow intensity is reduced, the trajectory, falling towards the clearance circle, increases its dimensions, Fig. 8.

It follows from the obtained measurements of the maximum radius-vector values that additional mass put on the rotor disk causes the radius value to change, Table 2. The value by which the radius-vector changes depends on additional mass as well as the place at which this extra mass is added relative to the residual unbalance of the rotor disk.

6. Conclusions

1. For ship's safety, a marine propeller shaft is a very important element of the propulsion system. The applied and recommended methods of its condition monitoring are insufficient. This author proposes diagnosing these shafts using relative vibrations measured in at least two planes perpendicular to the shaft axis. The measurement planes should be located in the plain stern and bow bearings of the shaft or close to them. In operational practice the journal centre trajectory can be visualized at the blocked DC component of the horizontal signal. In particular, this refers to propulsion systems operating at a constant rotation speed – systems with a controllable pitch propeller. At the constant rpm rate of the propeller shaft the vertical sensor axis should overlap the straight line connecting the point of thinnest oil film with the sleeve centre.
2. There is an assumption that existing relation between tightness and eccentricity and between eccentricity and reaction force (and sealing wear) in the sealing can be used to draw

conclusion on sealing condition. In this way the position and trajectory value of the shaft journal centre might provide data for conclusions concerning:

a) position of ship's propeller shaft,
b) wear of journals and bearing sleeves,
c) technical state of lubricating oil,
d) condition of shaft seals,
e) condition of the propeller (whether balanced or not).

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RELIABILITY MODEL OF SLIDE BEARINGS WITH PARTICULAR ATTENTION GIVEN TO LUBRICATING OIL

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ABSTRACT

The paper presents the slide bearing with circulating lubrication as a system of series three-element structure, where lubricating oil is the weakest link. In accordance with the Pierce statement that "strength of chain is the strength of its weakest link", a bearing reliability model has been developed. It allows to use the lubricating oil to evaluate the probability of correct working of the whole slide bearing, i.e. the reliability. The lubricating oil was taken from the dynamically loaded radial slide bearing fatigue strength testing stand "Smok" ("Dragon"). Lubricity tests were carried out in the friction node of the "T-02" four-ball extreme pressure tester, with balls dipped in the oil samples. Then, from the analysis and evaluation of test results, probabilities of correct bearing operation were determined.

Key words: cumulative stimuli, slide bearings, reliability, lubricating oil

1. Introduction

An important place among the causes of the slide bearing unserviceability has the wear (ageing) of all the bearing elements (i.e. journal, liner and the separating substance). Regardless of the initial constructional perfection of the bearing elements, they undergo irreversible changes in course of time. The wear of journal and liner is caused, among other reasons, by corrosion, deformation and material fatigue. The most harmful to a slide bearing is the loss of lubricity, which leads to technically dry friction and diffusion of one bearing material into the other.

The processes of corrosion, deformation etc. taking place e.g. in the combustion engine crankshaft bearing cause increased clearances. Due to the material fatigue, during the crankshaft rotation particles break off and adhere. In effect of those processes, the clearance gradually increases and the boundary layers get broken (i.e. the bond of lubricating oil with the journal or liner surface is broken).

The paper presents a wear model allowing to evaluate the slide bearing reliability from the analysis of oil lubricity.

2. Unserviceability of slide bearings

The criterion of slide bearing unserviceability classification is connected with the place of defect occurrence. This way the weakest "link" of the bearing can be detected and the minimum time of bearing correct operation determined.

With such attitude, the bearing will be treated as a system with series structure consisting of three elements [3]: journal, lubricating oil, liner.

The bearing reliability structure is shown in Fig.1.

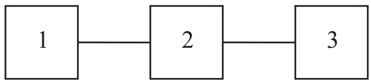

Fig. 1. Diagram of bearing as a system with series structure: 1 – journal, 2 – lubricating oil, 3 - liner

Durabilities of the system individual elements are random variables T_1, T_2, T_3 and their realizations are t_1, t_2, t_3, respectively. The system durability is T with realization t. The series structure system definition indicates that:

$$t = \min (t_1, t_2, t_3) \tag{1}$$

and that the system reliability $R(t) = P(T>t)$ is a product of probabilities: $P(T_1> t)$, $P(T_2>t)$, $P(T_3 > t)$:

$$R(t) = P(T_1 > t) \cdot P(T_2 > t) \cdot P(T_3 > t) \tag{2}$$

Additionally, the classification criterion allows to detect the reasons of defect occurrence at a given place.

The wear of slide bearings caused by action of cumulative stimuli [4] is only an indirect reason of defects. The bearing wear leading to an excessive clearance, mentioned in the introduction above, may cause a seizure. In such case no admissible values of clearance as a structural parameter can be determined for the journal and liner. The wear process only increases the bearing defect occurrence probability. For this type of defect the gradual bearing wear and the sudden change of its state to unserviceability occur jointly. As the changes of bearing technical condition occur suddenly and are a result of cumulative stimuli, the bearing unserviceability is caused by relaxation stimuli [4].

One of the three mentioned bearing elements is lubricating oil, which is also subjected to the action of cumulative stimuli in the form of loads. In effect of those stimuli the lubricity decreases, which leads to dry friction and in the worst case to seizure.

Therefore, in order to allow estimation of the parameters indicating the slide bearing inefficiency due to cumulative stimuli, a model of the lubricating oil wear has been developed.

3. Model of the lubricating oil wear

In order to maintain the bearing as a whole in the fitness for use condition, the respective characteristics of its components must be kept within strictly defined ranges determined during the bearing tests. Therefore, the bearing clearance should allow correct operation and prevent the dry friction occurrence. When one of the working characteristics (e.g. the working characteristic of the lubricating oil) exceeds the admissible limit and the bearing begins to function defectively, this is treated as bearing unserviceability.

The lubricating oil working characteristic (Fig. 2) deteriorates constantly as an effect of its ageing. It may be assumed that the bearing unserviceability will occur when the lubricating oil lubricity exceeds the admissible limit. The bearing correct operation time "T" is counted until the moment when the oil working characteristic exceeds the established limit value. Fig. 2 presents the situation when at randomly chosen moments single stimuli with determined value occur. After "r" such stimuli the bearing as a whole becomes unserviceable. Action of a single stimulus demonstrates itself as a stepwise decrease of the oil lubricity by a certain value "y" [4].

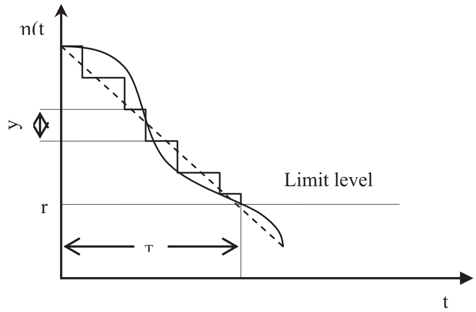

Fig. 2. Realization of the oil characteristic deterioration process as an effect of cumulation of the impairing stimuli: η(t) – working characteristic of the lubricaing oil; t – time; T – correct operation time, oil durability; r – number of stimuli necessary to make the oil useless; y – oil wear step

The wear model may be considered useful when it pertains to the stabilized wear period (i.e. normal wear). Then the probability of wear increase occurring in the time interval from "t" to "t+Δt" does not depend on the number of such increases in the time interval from 0 to t. Therefore, it is assumed that probability of each subsequent stimulus action does not depend on the total effect of all the preceding stimulus actions.

Opinion regarding the usefulness of investigation results (e.g. slide bearing wear determined from the lubricating oil properties) for confirming the suitability of the described model can be derived from observation of the realization of wear process. Fig. 3 and 4 present a realization of the bearing wear process (which may be used in these considerations) with "short" wearing-in period compared with the normal wear period, when mean value of the wear speed is the same for all bearings [4,5,6,9].

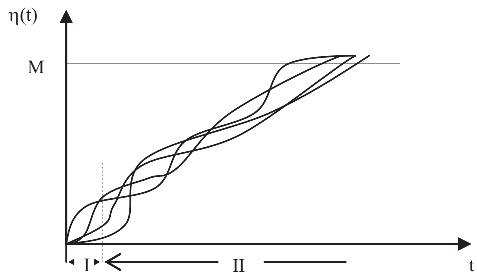

Fig.3. Realization of the bearing wear process: M – wear admissible limit; I – wearing-in period; II – normal wear period

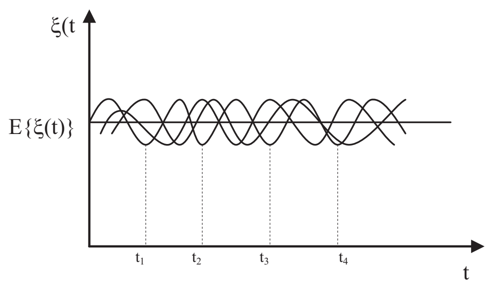

Fig.4. Variable speed of bearing wear during the normal wear period (mean value of the wear speed is the same for all bearings); E{ξ(t)} – expected value of the wear speed

In the presented realization of the bearing wear process (Fig.3) the individual wear realizations exceed the admissible wear level M before a disastrous effect occurs. After the end of wearing-in the wear realizations alternate. After elapsing of the wearing-in period the mean value of the wear speed is constant (Fig.4).

When on the basis of the investigation results a bearing wear model may be considered suitable, then it is assumed that the bearing durability (time to failure) has the gamma distribution. Then the value of the probability of correct work for the time t can be determined in a simple way from the nomogram presented in Fig.5.

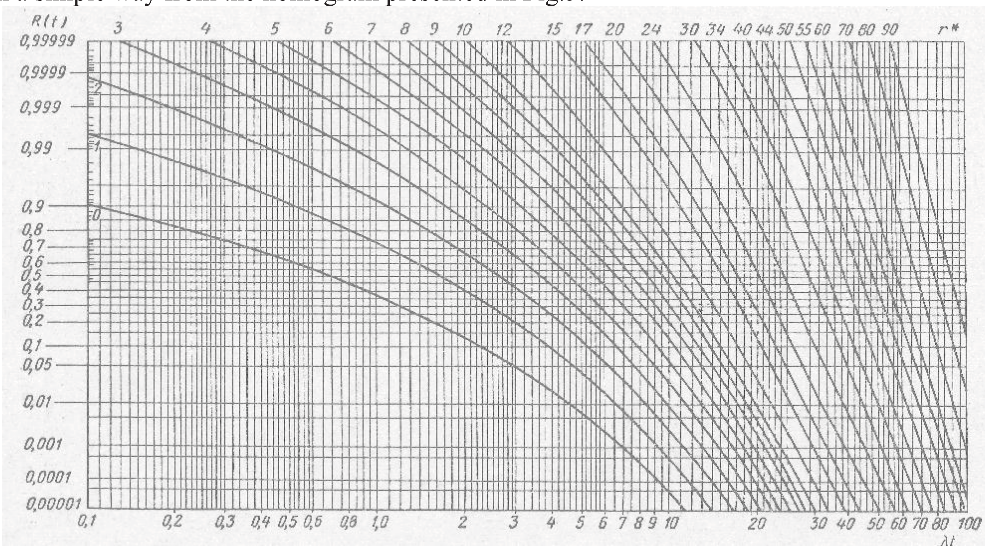

Fig.5. Nomogram for determining reliability R(t)=P(T>t) according to gamma distribution: r=r-1*

After completion of the empirical investigations and following the above presented considerations, the probability was estimated of the slide bearing correct operation. The basis for the probability estimation were results of the mentioned tests carried out with the T-02 four-ball extreme pressure tester.

4. Experimental investigations

For determination of the lubricating oil working characteristics the T-02 four-ball tester was used as a simple physical model of a friction node (Fig.6). The rubbing system in the tester consists of four 12.7 mm diameter balls made from the ŁH15 steel in the accuracy class 16 in accordance with the PN-83/M-86452 standard. Three balls are in the cup-shaped lower holder where the lubricating oil is poured to. The fourth ball is placed in the upper holder. Balls in the cup are pressed against the ball in the upper holder by means of a special lever. During testing the values of journal load, journal rotational speed and lubricating oil temperature can be controlled.

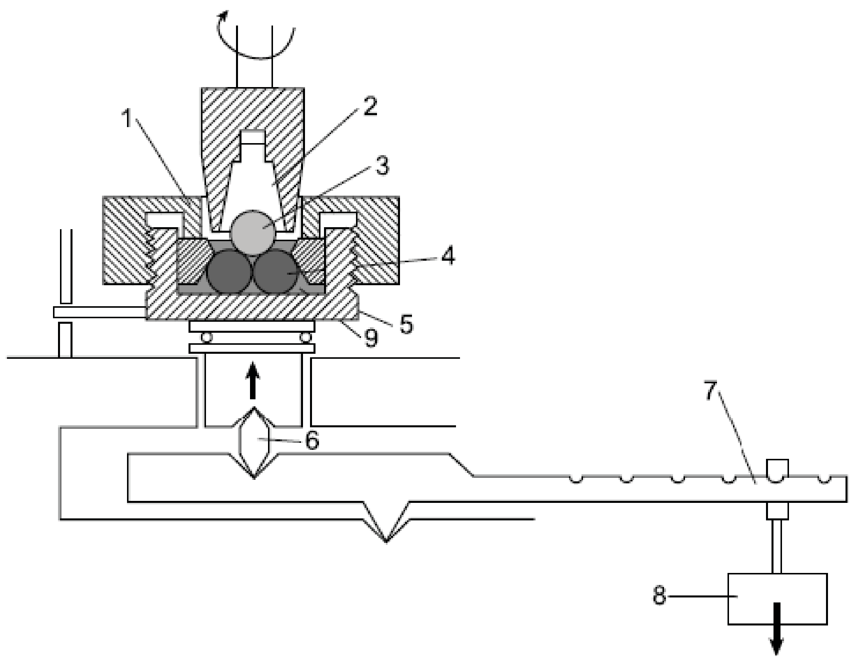

Fig. 6. Kinematic diagram of the four-ball tester: 1 – cover fixing the lower balls, 2 – upper ball holder, 3 – upper ball (rotating), 4 – lower balls (fixed), 5 – tested grease vessel, 6 – prism, 7 – lever, 8 – loads, 9 – tested grease [8].

Measurements were carried out in the T-02 four-ball tester friction node with balls dipped in the Selektor Specjal 20W40 oil sampled from the slide bearing of a "Smok" diagnostic stand [10].

Experimental investigations were carried out with:

1. Oil sampled on 19.01.2011 at 12.15 hrs. – symbol 1.
2. Oil sampled on 19.01.2011 at 14.15 hrs. – symbol 2.
3. Oil sampled on 19.01.2011 at 16.15 hrs. – symbol 3.
4. Oil sampled on 19.01.2011 at 18.15 hrs. – symbol 4.
5. Oil sampled on 20.01.2011 at 10.00 hrs. – symbol 5.
6. Oil sampled on 20.01.2011 at 13.30 hrs. – symbol 6.
7. Oil sampled on 20.01.2011 at 17.00 hrs. – symbol 7.

Slide bearings of the "Smok" diagnostic stand worked with different loads and in a wide range of rotational speed. Therefore, the T-02 four-ball tester measurements were carried out with different loads and different rotational speeds. There were 350 measurements performed altogether, in accordance with the following procedures:

– designation of the time "t" of breaking the boundary layer with constant increase of load at the spindle rotational speed of 500 rpm,
– designation of the time "t" of breaking the boundary layer with constant increase of load at the spindle rotational speed of 525 rpm,
– …………………….
– designation of the time "t" of breaking the boundary layer with constant increase of load at the spindle rotational speed of 1725 rpm.

After completion of the measurements (Table 1) a visual analysis was carried out of the realization of the lubricating oil wear process (Fig.7).

Table 1. Numerical data on breaking the boundary layer

Object no.	Spindle rotational speed n [rpm]																																																	
	500	525	550	575	600	625	650	675	700	725	750	775	800	825	850	875	900	925	950	975	1000	1025	1050	1075	1100	1125	1150	1175	1200	1225	1250	1275	1300	1325	1350	1375	1400	1425	1450	1475	1500	1525	1550	1575	1600	1625	1650	1675	1700	1725
i	Boundary layer breaking time τ [s]																																																	
1	5,15	4,97	4,91	4,85	4,79	4,73	4,67	4,61	4,55	4,50	4,44	4,39	4,33	4,28	4,23	4,17	4,12	4,07	4,02	3,97	3,92	3,87	3,82	3,78	3,73	3,68	3,64	3,59	3,55	3,51	3,46	3,42	3,38	3,33	3,29	3,25	3,21	3,17	3,13	3,09	3,06	3,02	2,98	2,94	2,91	2,87	2,84	2,80	2,77	2,73
2	4,87	4,60	4,55	4,51	4,46	4,42	4,37	4,33	4,29	4,25	4,20	4,16	4,12	4,08	4,04	4,00	3,96	3,92	3,88	3,84	3,80	3,77	3,73	3,69	3,66	3,62	3,58	3,55	3,51	3,48	3,44	3,41	3,38	3,34	3,31	3,28	3,24	3,21	3,18	3,15	3,12	3,09	3,06	3,02	2,99	2,97	2,94	2,91	2,88	2,85
3	4,62	4,38	4,34	4,30	4,25	4,21	4,17	4,13	4,09	4,05	4,01	3,97	3,93	3,89	3,85	3,81	3,77	3,74	3,70	3,66	3,63	3,59	3,55	3,52	3,48	3,45	3,42	3,38	3,35	3,32	3,28	3,25	3,22	3,19	3,15	3,12	3,09	3,06	3,03	3,00	2,97	2,94	2,91	2,88	2,85	2,83	2,80	2,77	2,74	2,72
4	5,39	5,30	5,23	5,17	5,10	5,04	4,98	4,92	4,85	4,79	4,74	4,68	4,62	4,56	4,51	4,45	4,39	4,34	4,29	4,23	4,18	4,13	4,08	4,03	3,98	3,93	3,88	3,83	3,78	3,74	3,69	3,65	3,60	3,56	3,51	3,47	3,42	3,38	3,34	3,30	3,26	3,22	3,18	3,14	3,10	3,06	3,02	2,99	2,95	2,91
5	5,47	5,23	5,14	5,05	4,97	4,88	4,80	4,71	4,63	4,55	4,47	4,40	4,32	4,25	4,17	4,10	4,03	3,96	3,89	3,82	3,76	3,69	3,63	3,57	3,50	3,44	3,38	3,33	3,27	3,21	3,16	3,10	3,05	2,99	2,94	2,89	2,84	2,79	2,74	2,70	2,65	2,60	2,56	2,52	2,47	2,43	2,39	2,35	2,31	2,27
6	4,67	4,62	4,57	4,53	4,48	4,44	4,40	4,35	4,31	4,27	4,22	4,18	4,14	4,10	4,06	4,02	3,98	3,94	3,90	3,86	3,82	3,78	3,75	3,71	3,67	3,64	3,60	3,57	3,53	3,49	3,46	3,43	3,39	3,36	3,32	3,29	3,26	3,23	3,19	3,16	3,13	3,10	3,07	3,04	3,01	2,98	2,95	2,92	2,89	2,86
7	4,64	4,67	4,59	4,51	4,43	4,35	4,28	4,20	4,13	4,06	3,99	3,92	3,85	3,79	3,72	3,66	3,59	3,53	3,47	3,41	3,35	3,29	3,24	3,18	3,12	3,07	3,02	2,97	2,91	2,86	2,81	2,77	2,72	2,67	2,62	2,58	2,53	2,49	2,45	2,41	2,36	2,32	2,28	2,24	2,20	2,17	2,13	2,09	2,06	2,02

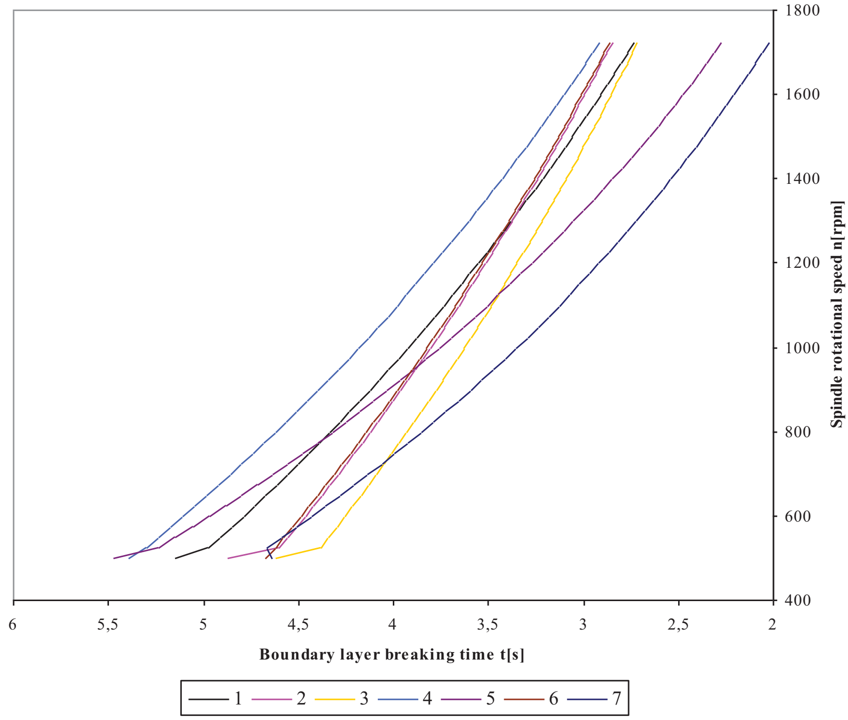

Fig.7. Realization of the lubricating oil wear process

Fig.7 indicates that the individual wear realizations alternate and the average wear speed is constant, therefore the presented lubrication oil wear model may be recognized as useful.

It may also be assumed that the reliability model of the tested oil corresponds with the gamma distribution [4]. Data given in Table 1 allow to estimate the λ (i.e. intensity of damage) and r (i.e. the number of stimuli necessary to cause unserviceability) parameters as well as probability of correct operation during t_1=2.5 s and t_2= 5 s time.

Table 2. Estimation of reliability parameters

Object no.	$\bar{t} = E(t)$	$s_t^2 = D(t)$	$\lambda^* = \frac{\bar{t}}{s_t^2}$	$r^* = \frac{(\bar{t})^2}{s_t^2}$
1	3,64	0,48	7,65	27,81
2				
3				
4				
5				
6				
7				

As the coefficient of variation is greater than 0.34, the gamma distribution will be applied [4]. Using the nomogram (Fig.5), from the boundary layer breaking time the probability of the tested lubricating oil correct operation can be found: P(T > 2.5)=0.9 and P(T > 5)=0.02.

5. Final remarks and conclusions

Rational operation of a slide bearing [5,7], consisting of three elements, is possible when the operating characteristics are known, mainly the reliability including durability of the weakest link. In practice, the weakest link of a bearing with circular lubrication is the lubricating oil. Therefore, reliability of a slide bearing as a series structure system depends first of all on the oil durability. Main parameter defining the oil durability is its lubricity [1,2]. It is then necessary to be able to estimate continuously the lubricity realization time in real conditions.

The presented reliability model of the slide bearing lubricating oil is used for analysing and evaluation of the oil lubricity when loads increase. Investigations were carried out in laboratory and the reliability model was a four-ball tester. The lubricating oil was sampled from the "Smok" diagnostic stand slide bearings.

The paper presents a method of determining the bearing lubricating oil reliability by measurements of the oil lubricity. Such tests should be carried out in real conditions, using the presented reliability model, which may be considered suitable for such tests.

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TRIAL APPLICATION OF THE FORMAL FIRE RISK ASSESSMENT TO FIRE PREVENTION IN SHIP ENGINE ROOMS

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Abstract

This paper contains an interpretation of the measures to prevent fire in ship engine rooms, which have been developed by the International Marine Organization as the Guidelines MSC.1/Circ 1321. Attention is drawn to the risk control measures in the design, construction, testing, installation and inspection of systems containing flammable oils.

The measures to prevent fire in flammable oils were verified using theoretical technique i.e. the formal fire risk assessment (FFRA). The FFRA is based on the probabilistic model for fire frequency calculations. The model has been described and results of trial application of the FFRA has been presented.

It can be said the effectiveness of risk control measures can be examined quantitatively by applying the analytical technique. However, making generic model for detailed examination with adequate accuracy poses number of challenges, the most significant being lack of adequate statistic data on past fire accidents and consideration of the human factor.

1.Introduction

Statistics on world fleet fire safety show that the engine room fire hazard in ships expressed in terms of relative frequency amounts about 1/1000 ship years, which is 30-50% of the overal ship fire accident rate [1]. The most likely cause of an engine room fire is the leakage or spray flammable oil. The past data on fires shows that fire due to leakage and dispersion of fuel oil, lubricating oil and waste oil account for about 60% of all engine room fires [6].

In responding to the needs for an improvement of the fire safety record in engine-rooms, the IMO developed the Guidelines for Measures to Prevent Fires in Engine-Rooms and Cargo Pump Rooms. Guidelines related to flammable oil fires were verified using an analytic technique named Formal Fire Risk Assessment (FFRA).

2. Interpretation of the guidelines for measures to prevent fire in engine-rooms

Mostly, the guidelines are aimed at the prevention of fires caused by self ignition of flammable oils. They are oriented towards elimination of malfunctions and leakages in flammable oil systems, insulating hot surfaces and ensuring safe atmosphere in engine-room spaces.

2.1. Elimination of malfunctions and leakages of flammable oil systems

The basic requirement enabling elimination of leakages of flammable oil systems is to ensure tightness of pipelines under pressure pulsations generated by injection pumps. Contractors of systems are obliged to inform users about maximum (peak) values of pressure pulsations, which should not exceed 1.6 MPa at the outlet of injection pump sets.

In order to eliminate pipeline damages (leakages) caused by vibrations, apart from meeting ISO and classifications societies' requirements, it is recommended to install pressure vibration dampers in injection pipes, and to use injection pumps ensuring a fixed injection pressure.

Operating practice has shown that currently used mechanical pressure accumulators and gas filled bellows are subject to fatigue damages, and their responsiveness is prolonged [1].

Fuel oil system sets should be selected and assembled in consideration of a possibility for vibrations in injection pipes to appear. It is highly recommended to use flanged joints to ensure a proper preliminary tension of joint bolts.

In order to limit fuel oil and diesel oil dispersion caused by lack of tightness, at pressure exceeding 0.18 N/mm^2, pipeline joints should be shielded in the vicinity of 'hot surfaces' (it also concerns centrifuge and fuel oil and diesel oil treatment systems). In engine supply systems, jacketed injection pipes with drain space and leakage signaling system are recommended.

In order to prevent leakages in elastic joints and elastic hoses, the following elements are recommended:

- constructions approval and pressure test certificates (every 5 years),
- adaptation of constructions for working conditions, i.e. temperature, pressure, mechanical load, properties of liquids, etc.,
- conditions for pipe installation, i.e. maximum length and radiuses of bending, curve angles, direction deviation, supporting structures,
- frequency and verification criteria for elastic hoses subject to replacement.

In tanks with flammable liquids (fuel oil, lubricant oil, heating oil, and hydraulic oil), it is recommended to use equipment preventing excessive pressure and temperature increase, and spill caused by lack of tightness or overflow. Such systems include, inter alia:

systems signaling critical levels,
overflow and ventilation pipes,
systems signaling critical temperature (220^{o}C),
level gauges and testing devices, constructed according to the designs included in "The Guidelines and requirements of classification societies.

In order to ensure tightness of other fuel system elements, such as filters, expansion compensators, measuring instruments, and pipelines fittings, construction approval by supervising institutions, pressure tests certificates, spray shields and drainage of leakages, adaptation for working conditions (vibrations and high temperature), appropriate assembly stresses, etc., are recommended.

For repair and service systems of fuel installations, it is recommended to include the following operations:

- execution of procedures according to 'checklists', especially in case of replacement of parts and assembly,
- coordination of installation (assembly) by responsible person responsible for it' who should ensure full implementation of project goals according to detailed design documentation,
- identification of vibrations, fatigue stresses of welded and hardened joints, and damages of elements, on the basis of recognized diagnostic procedures,
- periodic verification of preliminary tension of injection pipes' joint bolts (every 3 months),
- periodic verification of low pressure pipelines fittings (every 6 months),
- verification of threaded joints during every assembly.

2.2. Hot spots insulation

In accordance with SOLAS Convention, ‘hot surfaces’ in an engine room (exceeding 220^{o}C) should be thermally isolated. The isolation is approved by a supervising (classifying) institution. In particular, isolation of external surfaces is recommended for outlet exhaust pipes, boiler burners housing and units, turbo compressors exhaust pipes and frames, bare metal friction parts, inert gas generators, incinerating plants and highly loaded electrical control panels.

Apart from isolation, ventilation of the surrounding space is applied as well as water spray systems (Hi-Fog) for such machines as centrifuge boilers, fuel oil and diesel oil boilers, electrical boards and systems (wires).

The temperature in fuel oil and diesel oil tanks placed in the vicinity of boilers should not exceed 10^o below the flash point of flammable oil. Boiler control systems should ensure automatic fuel cut-off in case of flame failure, and burner system interlock in case if fuel supply is turned on.

The temperature of electric supply elements and drive transmission elements (glands, bearings) of flammable liquid pumps, should be constantly controlled through permanent electrical systems with sensors responding between 60 – 80^o (with automatic pump turn-off).

In steam (electrical) boilers for flammable liquids, temperature and fuel failure control and signaling systems, and automatic supply switches (at 220^{o}C).

In heating oil systems with combustion boilers, protection against an ignition includes temperature sensors in tanks, emergency tank drainage, fire and explosive mixtures detection, water spray systems, pressurized circuits of oil, safety temperature switch and pipe tightness inspections and tests.

It is recommended to conduct hot surfaces identification and isolation checks periodically, with the use of Thermo Vision cameras or laser thermometers with accordance to approved procedures.

2.3. Safe atmosphere

In order to ensure safe atmosphere of engine rooms (as well as inside ‘hot machines’) it is recommended to apply ventilation systems of capacity sufficient for safe dilution of flammable mixtures of gas and oil mist below 5% of the lower explosive limit (LEL). The atmosphere should be controlled by fixed elements of flammable gas, oil mist and smoke detection systems. Sensors layout should take into consideration the characteristics of space ventilation. Detailed design guidelines in this scope have not been prepared yet because of the lack of data which can be drawn up on the basis of already advanced computer simulations [2].

Recommendations for ventilation and atmosphere control also concern rooms with power hydraulics and heating oil systems, and centrifuges.

3. Formal fire risk assessment

The formal fire risk assessment involves the following steps:

- identification and classification of potential fire sources (liquid leakage and ignition),
- adoption of fire event tree,
- calculations if relative frequencies (probability) of various types of fires.

3.1. Identification and classification of fire sources

Fire sources identification involves selection of definite number of system elements, the damages of which may result in flammable liquid leakage and self-ignition. Such elements as, inter alia, pipelines joints and sections, valves and system equipment, are taken into consideration.

Sources of leakages are regarded as fire sources when they are at a specific (established) distance from the source of ignition, such as exhaust pipes, boilers, turbo compressors, electrical equipment etc.

Potential fire sources are classified and marked according to factors (construction and operational parameters) that influence occurrence and development of fire (hazard). Basic hazard factors are as follows:

- type of a system and its operation outline,
- localizing fire source in a system,
- parameters and properties of flammable liquids,
- localizing fire sources in an engine room,
- geometry of system elements (inter alia, flow section),
- characteristics of typical ignition sources (inter alia, temperature, distance to the leakage source and its shape),
- characteristics of fire protection system (inter alia, liquid dispersion reduction, a number of proper fire detectors, equipment and nozzles of fire-extinguishing system).

Identification of fire sources is based on statistical data concerning real fires, i.e. frequency and causes of their occurrence. To determine the overall number of potential fire sources, a condition that the estimated and real relative fire frequencies are equal (or similar) is taken into consideration.

1. Adoption of a fire event tree

To adopt a fire event tree, the following original scenario (sequence of events is assumed:

1. leakage of flammable liquid
2. flammable mixture and ignition source interaction
3. fire break out (ignition)
4. fire detection (detector activation)
5. preliminary fire fighting (without cutting off liquid supply)
6. power cut-off
7. full scale fire fighting (use of fixed fire-extinguishing system).

Depending on which possibility of an event has been chosen (yes or no), the original sequence is expanded as secondary events branches, the time limit of which is a fire of various scales (or no fire).

Fig. 1 presents the event tree for engine-room fires.

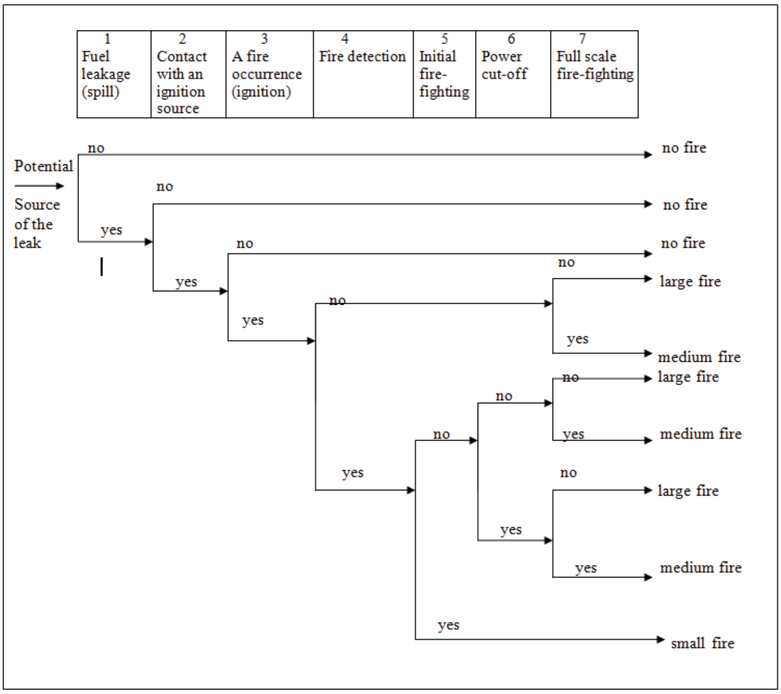

Fig. 1. Event tree for engine-room fires

3.3. Calculation of relative frequency

Frequency of flammable liquid leakages is determined on the basis of damage probability in oil systems using various data such as "Process Equipment Reliability Data" and "Ship Reliability Survey" [6]. The values used are:

Joints	4.10×10^{-3} 1/year
Valves	1.09×10^{-3} 1/year
Accessories	1.80×10^{-4} 1/year

Probability of sprayed flammable liquid (leakage) interaction with an ignition source is calculated with the following formula:

$$Pit = \frac{R\max - \lambda}{R\max} \cdot \frac{2\phi + \omega}{360} \cdot 0{,}5 \cdot f,$$

where:

Rmax – spray reach,

λ - reduced distance between a leakage point and a source of ignition,

ϕ – liquid spray cone angle in horizontal plane,

f – coefficient of splash protection,

ω - coefficient of shape (2D) projection of ignition source onto a plane perpendicular to the leakage direction,

The spray reach has been calculated with the following formula:

$$R\max = \frac{Vo^2}{2g} \cdot \sin 2\theta$$

where:

Vo – liquid leakage velocity,

2θ – spray cone angle in horizontal plane,

g – gravitational acceleration.

Probability of ignition during an interaction between liquid splashed during leaking and the source of ignition, is calculated with the following formula:

$$P_i = p_{it} \cdot C_t,$$

where:

C_t – coefficient taking into consideration a temperature of the liquid,

p_{it} – coefficient taking into consideration the distance between a source of leakage and a source of ignition,

Probability of fire detection is calculated with the following formulas:

$P_d = 1 - (1 - p_{det})$, for $n_d \geq 1$

$P_d = p_{det} \cdot 0.5$, dla $n_d = 0$,

where:

p_{det} – coefficient of fire detection,

n_d – a number of fire detectors.

Probability of fire extinguishment with the use of proper equipment (fire-extinguisher) is calculated with the following formulas:

$$P_g = P_{gi} \cdot k$$

$$P_{gi} = 1 - [1 - \exp\{-(\frac{E4}{b})^a\}]^{ng}$$

where:

$E4$ – radiant heat at a distance of 4 m from the centre of a flame [kJ/m^2 · h.],

$a = 3,4 \quad b = 16.33 \times 10^3$ – fixed coefficients,

n_g – a number of fire-extinguishers,

k – operator's safety coefficient.

Effective range of fire-extinguisher (4 m) and operator's safety coefficient (k = 0.99 – for $E4$ < 1000, k = 0.1 – for $E4$ < 5000) are taken into consideration for calculations.

Probability of fire extinguishment with the use of fire foam from fixed foam system nozzles is calculated with the following formula:

$$P_f = 1 - [1 - \exp\{-(\frac{E10}{b})^a\}]^{nd}$$

where:

$E10$ – radiant heat at a distance of 10 m from the centre of a flame,

n_d – a number of nozzles.

Fig. 2 presents a relation between radiant heat and a radius of a flame expressed with an empirical formula.

The radius of the flame 's base is calculated with the following ratio:

$$\Pi R^2 = \frac{A(0,2)^2 \cdot V_o}{V_B},$$

where:

R – radius of a flame (cylindrical),

A – section of a pipeline supplying flammable liquid,

V_o – liquid (leakage) outflow velocity,

V_B – burning velocity ($0.28 \cdot 10^{-4}$ m/s).

In a geometric model of the flame's shape, it is assumed that the leakage was a result of a crack or loose joint of a pipeline. It has been assumed that a fire of an overflow area corresponds to 20% of damage and that the amounts of leaking liquid and burning liquid are equal.

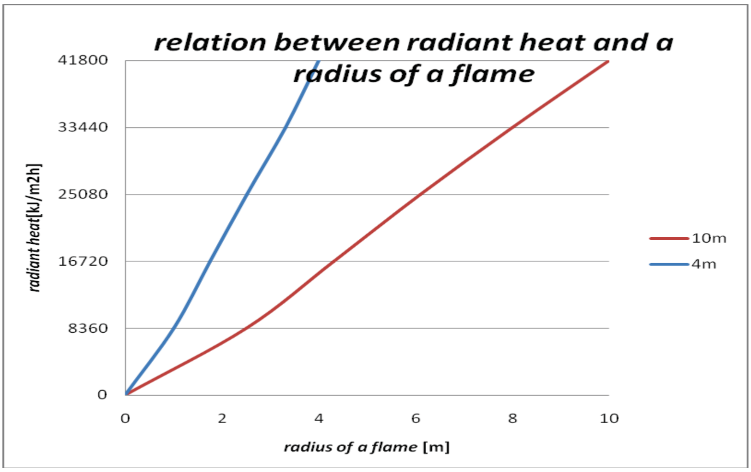

Fig. 2. The relation between radiant heat and a radius of a flame (for leakages caused by cracks or lack of tightness in joints of pipelines)

The probability of efficient initial fire extinguishing (small fire) is calculated with the following formula:

$$P_e = 1 - (1 - P_g) \cdot (1 - P_f)$$

The following values of probability of efficient large fire extinguishing (full scale) with the use of a fixed fire-extinguishing system are adopted:

$P_s = 0.99$ – after fuel cut-off,

$P_s = 0.9$ – without fuel cut-off.

The value of 0.99 for the probability of efficient fuel cut-off is adopted.

4. Fire risk control

In order to prevent or control fire accidents different countermeasures can be considered. These are: installing additional fittings, accesories and apparatuses for preventing oil leakage and ignition, for the fire detection and for fire fighting.

In the trial application numbers of safety equipment were considered as risk control measures. These were:

- number of smoke detectors (n_w),
- number of fire extinguishers (n_g),
- number of nozzles of foam extinguishing systems (n_d).

The effectivenes of the changes in numbers of safety equipment on fire hazard has been expressed by the risk sensivity factor using the following formula:

$$C_f = \frac{F_+ - F_-}{F_o} \cdot \frac{1}{n}$$

where:

F_o – relative fire frequency with the safety equipment according to the initial classification [1/ship year],

F_+ – relative fire frequency with more safety equipment,

F_- – relative fire frequency with less safety equipment,

$n = 2$,

$n = 1$ – for $F_- = 0$.

The effects of number of safety equipment to scale of fire are laid down in table 1.

Table 1 Effect of number of safety equipment to scale of fire.

Risk area	**Number of Safety Equipment**			**Small Fire**		**Medium Fire**		**Large Fire**	
	n_w	n_g	n_d	F_o 1/ship year	C_f	F_o 1/ship year	C_f	F_o 1/ship year	C_f
Power generat	3	3	2	$2.81 x 10^{-4}$	-	$2.26 x 10^{-5}$	-	$4.70 \text{ x } 10^{-2}$	-
Sets platform	3	3	2+1		$2.43 x 10^{-2}$		$-3.67 x 10^{-2}$		$2.43 x 10^{-2}$
	3	3+1	2		$8.35 x 10^{-5}$		$-1.23 x 10^{-4}$		$8.02 x 10^{-5}$
	3+1	3	2		$2.53 x 10^{-2}$		$-3.72 x 10^{-2}$		$2.39 x 10^{-2}$
Boiler platforn	1	2	3	$8.75 x 10^{-6}$	-	$1.98 x 10^{-6}$	-	$2{,}19 x 10^{-7}$	-
	1	2	3+1		$5.88 x 10^{-5}$		$-8.65 x 10^{-5}$		$-5.65 x 10^{-5}$
	1	2+1	3		$1.16 x 10^{-2}$		$-1.71 x 10^{-7}$		$-1.12 x 10^{-7}$
	1+1	2	3		$5.88 x 10^{-5}$		$-8.65 x 10^{-5}$		$-5.65 x 10^{-5}$

Note:

F_o – relative fire frequency for the initial amount of safety equipment (n_w, n_g, n_d),

C_f – risk sensitivity factor (for increase by 1 in n_w, n_g and n_d).

According to the table, a greater number of fire-extinguishers, detectors and nozzles causes an increase of probability of small fires and at the same time decreases probability of medium and large fires. The impact of the number of fire-extinguishers on risk sensitivity is much lower than that of the number of detectors and nozzles. Absolute value of the sensitivity coefficient is greater for areas of higher risk.

5. Conclusions

The formal fire risk assessment described in this paper is based on a probabilistic model. Its preliminary assumptions and initial parameters for fire frequency calculations are adopted on the basis of statistical data for real fires. With the use of the model, it is possible to calculate relative frequencies of fires of a various scale for individual fire sources in fuel oil and diesel oil systems as well as to calculate total fire hazard for machines and spaces, and their contribution to general hazard concerning engine rooms.

The accuracy of fire frequencies calculations is approximate and when applied to quantitative assessment of hazard the credibility of results is limited. The fundamental simplification of the model is that the influence of human factor is not taken into consideration.

For the purposes of searching for possibilities of construction influence on fire hazard, qualitative assessment is sufficient. It is so-called sensitivity analysis that is based on relative fire frequency calculations for various types of protection (e.g. with the use of liquid splash protection, various number of fire detectors and fire extinguishing nozzles).

In this respect, the model described in this paper has been used to verify requirements for flammable oil systems in ship engine-rooms. It may be assumed that creating an adequate calculation model for the purpose of a general fire hazard analysis of an engine-room is a significant challenge.

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A NOTE ON MODIFICATIONS TO THE METHODOLOGY FOR COMPONENTS IN THE COMPLEX TECHNICAL SYSTEMS RELIABILITY STRUCTURE IMPORTANCE EVALUATION

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Abstract

The paper presents State-of-the-Art in topic of reliability importance analysis for components and groups of components in complex technical systems (CTS). Basic importance evaluation measures for finding so called "weak links" in system structure have been discussed. Quantitative and qualitative importance analysis methods have been presented. Range of applicability of these methods have been pointed out. Some results of importance evaluation with use of Birnbaum's and Vesseley-Fussell's measures based on the example of 2-phased complex marine system have been shown. Critically conclusions on currently known importance analysis methods have been presented.

Keywords: *importance analysis, sensitivity analysis, importance measure, weak links, reliability structure*

1. Introduction

Dependability theory both in the statistical approach as well as physics of damage is mostly being concentrated on the functioning of systems and effectively allow to calculate of different measures of reliability, availability and safety of the systems. Basic reliability measures for the system are providing important information on a given system but for the evaluation of system components reliabilities this measures are providing only general information in the aspect correct operation of the system. Apart from the case of the serial reliability structure this measures are not giving information on influence of component reliability (component state) onto system reliability (system state). The tolerance of components failures by system depends on the reliability of components and the system reliability structure, in which the given component is located.

2. State-of-the-Art

From the dependability point of view, the importance of given component in the system depends on two factors:

- reliability characteristics of the component,
- reliability structure in which the component is located.

The influence of first factor is obvious. In relation to the component location in reliability structure, the component is the more important, if its location is more similar to the single component inserted to serial reliability structure of system. Component importance (influence on the change of the system reliability) is decreasing together with the increase in the level of redundancy of this component.

2.1. Quantitative importance evaluation

In 1969 Z.W. Birnbaum in the work [3] published the first quantitative measure of the importance of components in the reliability structure of the system, which he defined as the difference between the system reliability, when the *i*-th component is in up state at the time *t* and with the system reliability, when the *i*-th component is in down state at the time *t*. Birnbaum measure depends only on the structure in which the component is localized and from the reliability characteristics of all other (remaining) components. Birnbaum's measure do not depends on the reliability of considered the component. The process of quantitative analysis of the importance of components and groups of components was described schematically in the Fig.1.

In 1975 H.E. Lambert implemented term of the critical component of the system, which one if is in down state will cause system down state. He introduced the measure of the criticality which it is possible to define as, the probability, that the *i*-th component is critical for the system and is failed at time *t*.

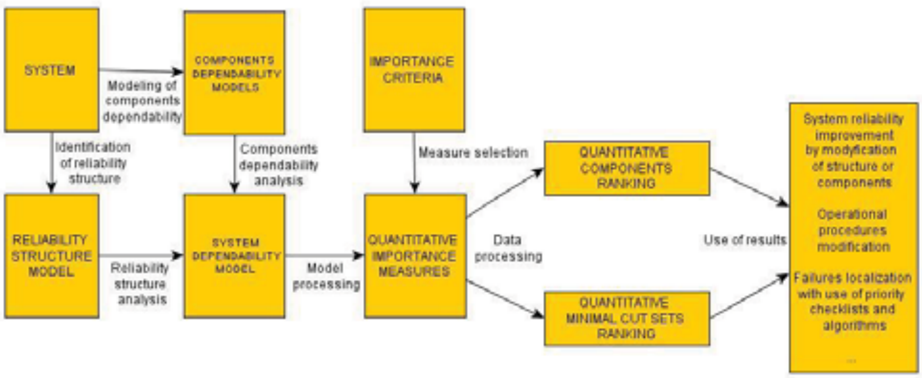

Fig. 1. Quantitative importance analysis of components and groups of components in the system reliability structure

In subsequent years new measures of the importance have been introduced, including measures drawn up by R.E. Barlow and F. Proschan [2]. Barlow-Proschan's measure is equal to the probability that the system failure is an effect if the *i*-th component failure. This measure can be treated as the average Birnbaum measure in reference to the unreliability of the *i*-th component.

At the end of 70's B. Natvig drew up a new reliability measure of the importance [15], in which he made the importance of the component conditional on loss of the remaining time to failure of the system caused by the transition of the considered component into down state. B. Bergman in 1987 suggested the next more widely known measure.

Issues of the evaluation of the importance were elaborated in subsequent years, importance measures have been developed for renewable systems by replacing the reliability and unreliability functions with functions of availability and unavailability [11,16]. A Vesely-Fussell's measure $I_i^{VF}(t)$ is one of commonly used quantitative measures of the importance for renewable systems and for the *i*-th component, and is defined as the conditional probability that an at least one minimal cut set containing the *i*-th component will occur at time *t*, assuming that the system is down at time t.

One should emphasize the fact, that different reliability importance measures are leading to different importance rankings because of different definitions of measures, therefore one should take characteristics of the given measure into consideration making interpretations during the analysis of the results. For finding components of the system, of which reliability characteristics should be corrected in order to increase the system reliability the most useful are reliability Birnbaum's measure and Barlow-Proschan's measure. However at seeking components which failures with the greatest probability will cause the breakdown of the system is recommended to use of the Vesely-Fussell's measure and the criticality measure. Example rankings of component importance [11] for vessel propulsion plant sea water cooling system for two different phases of operation based by two different measures have been shown in Fig. 2.

With reference to the evaluation of groups of components it is possible to consider the importance of minimal cut sets (minimal sets of components which simultaneous down state will cause the system down). The importance of the minimal cut sets is being interpreted as the conditional probability, that the k-th minimal cut set will appear at time t, assuming that the system is down at time t.

Fig. 2. Example CTS components importance rankings based on Birnbaum's and Vesseley-Fussell's measures [11]

2.2. Qualitative importance evaluation

The applicability of mentioned measures is very limited, because it requires the accurate knowledge of reliability characteristics of individual components and the entire system [11,15]. With reference to CTS an information about the probability density of time to component failure, reliability, time to failure functions etc. is usually unknown.

Partly solution in case of lack of full information on system is an application of qualitative importance analysis. In this method the importance of the given component is evaluated only with reference to the location of this component in the system reliability structure. However not considering by qualitative methods the information about components reliabilities causes that these measures have a limited applicability. The process of qualitative analysis of the importance was described in the Fig. 3.

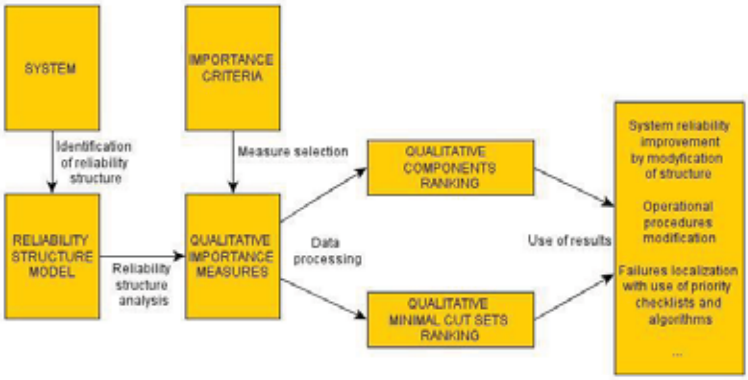

Fig. 3. Qualitative importance analysis of components and groups of components in the system reliability structure

As example of the qualitative measure of the component importance (dependent only on the system reliability structure, which the component is located in) is a structural measure of Birnbaum, which for the *i*-th component is being defined as the relative number of *n*-component system states for which the *i*-th component is critical for the system (states in which component down state will cause the system down).

L. Chybowski and Z. Matuszak suggested the normalised measure of streams [6,11] based on M. Kołodziejski and Z. Matuszak stream measure, which takes into consideration the participation of the *i*-th component in the reliability structure of a few subsystems at time *t*.

Qualitative importance measure for system cut sets is number of set components (cut set order). The cut set is usually the more important if it contains fewer components.

Creating rankings of components and events leading to components failures in the system is one of issues of the evaluation of the importance of components. The ranking of events can work on the assumption that human errors appear much more often than failure of active components, and failures of active components are more frequent than failures of passive components. For example the working pump will fail more likely than the standby pump.

3. Final conclusions

In 90's and in the last decade many theoretical works concerning analysis of the importance and connected issues came into existence, including works of: E. Zio, G. E. Apostolakis, E. Borgonovo, A. Brandowski, T. Aven, J. Jaźwiński, Z. Smalko, K. Kołowrocki, F. C. Meng, J. Z. Czajgucki, W. E. Vesely, J. Vatn, Z. Matuszak, P. J. Boland. Amongst the tendency in analysis it is possible to distinguish papers on: developing new measures importance of components [4,5,11, 18], describing specific applications of importance measures of the given class of systems in the assessment of the safety [5,10,11,12, 15.18] and far-reaching analyses of connections between different importance measures and getting of importance rankings [1,11,17] among others.

In the 21st century new importance analysis issue appeared taken up by researchers (among others S. Beeson, J. D. Andrews), which is the importance analysis of noncoherent systems components (systems, in which component fault can cause change system state from down into up state). Analysis of the importance for such systems was connected with a significant complication of mathematical models, because for every component one should appoint two measures associated with being of the component appropriately the condition of the up and down state. Such an approach makes difficulties of an interpretation of the achieved analysis results. However in spite of the developed models these methods are good for theoretical analysis, due to the fact that apart from few coincidences, CTS are coherent systems.

During last over 40 years from introducing the first measure of the importance by Z.W. Birnbaum many importance measures of the components and minimal cut sets have been introduced. However in spite of the theoretical models of the evaluation of the importance, they are creating quoted appliqué problems. CTS including vessel propulsion plant are systems difficult in the description (analysis) [7,8,9,13] due to CTS are systems:

- renewable or partial-renewable;
- with time dependent functional and reliability structure of the system;
- with hierarchical structure and multidimensional often not known feedbacks;
- with components failures partly or entirely dependent;
- for which replies only to the defined scope and character of inputs and disturbances are known;
- with unknown redundancy kinds and connections [14];
- for which the reliability structure in spite of known allocation of basic functional components is not often known entirely or in considerable parts.

In view of the described CTS features application of known measures of the importance is often limited or impossible because of the lack of complete information on relations in the system and components reliabilities, what makes that “classical importance measures” have limited application for CTS analysis.

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A NEW APPROACH TO RELIABILITY IMPORTANCE ANALYSIS OF COMPLEX TECHNICAL SYSTEMS

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Abstract

The paper presents new approach to the reliability importance analysis which is main topic of the author's last research. The concept of comprehensive qualitative-quantitative method for the importance analysis for complex technical systems components has been shown. Extension of importance criteria of components to: safety, maintainability, spare parts waiting time and other factors has been proposed. Research conception, plan and final conclusions have been shown. The application of expert knowledge and reliability databases in case of limited knowledge about the system have been proposed.

Keywords: *importance analysis, subjective probability, qualitative-quantitative method, expert knowledge, system structure*

1. Introduction

Due to problems with application of known reliability importance measures for complex technical systems (CTS) [5,7,10,15], the research work to create the new qualitative-quantitative methodology of importance analysis have been introduced. New methodology is supplied with expert knowledge, different importance criteria and application of reliability databases. Comprehensive methodology can be applied to CTS with different range of knowledge lack (unknown reliability structure, partly known components reliabilities etc.). The main aim of the paper is to show of the concept of developing of the current state-of-the-art of reliability theory mainly reliability importance analysis. It will be carried out by developing theoretical expressions and methods which enable quantitative and qualitative assessment of CTS reliability importance of components and group of components. As part of the work new tools will be developed and popularized (algorithms and the dedicated software) for the evaluation of the importance of the components in CTS. New developed tools will be used in the evaluation of the components importance of vessel propulsion plant as the appliqué example.

2. Significance of the new approach to importance analysis

High reliability of CTS is a basic condition of the safe and effective operation of the system. During operation the requirement of increasing of the system reliability by the alteration of the system structure or increasing the reliability of chosen components often occurs. During analysis of the reliability of the technical system the analyst usually cares about finding the most sensitive components (*importance measures of components*). Reliability of these components should be increased to optimal raise the reliability of the entire system [1,2,3,12,13,14]. Similarly an importance of the minimal cut sets of the system is being considered (*local measures of the importance*). These issues are connected with seeking so-called *weak links in the system*, which are of the most unreliable components and groups of components in the system (*importance analysis*).

CTS are objects about which we never have complete information on components reliabilities and on the system reliability structure. This problem makes known measures limited in use in the utilitarian aspect.

Criteria of the importance for known importance measures concern basically the reliability, availability or time to failure function, however in many cases in the analysis of the importance an assessment of the impact of the given component failure for the operation of the entire system in the aspects of maintainability, life cycle costs, availabilities of the spare parts and operational safety is supposed to be taken under consideration.

In relation to the limited applicability or the lack of applying many known importance measures [5,10,13,15], a requirement of draw up the modern qualitative-quantitative methodology giving the importance of components and groups of components for determining importance rankings in CTS for given criteria of the importance including the aspects: cost-effectivenesses, availabilities of the spare parts, maintainabilities and operational safeties appeared, what should be developed.

New methodology will develop the technical sciences as well as find application in many branches of industry. It will affect the development of the civilization and the society using complex objects of the exploitation in many aspects of the daily living.

3. Conception and research plan

Literature analysis made by the author shows, that publications devoted to the utilitarian application of importance measures of CTS components occasionally appeared in relation to presented problems with application.

According to the author, one of the ways of acquiring the knowledge about the system using expert methods and application of the subjective probability [4]. Current research papers describing scientific works on the importance evaluation as the object of analysis use simple theoretical systems with statistically independent components failures and implemented basic interactions with surroundings.

Creating comprehensive quantitative-qualitative methodology will allow to much more effective analysis compares to known methods.

New importance measures should use of the number of criteria of the importance which were not taken into account in so far known in theory reliability measures of the importance e.g. Birnbaum, Vesely-Fussell, Barlow-Proschan, Natvig etc.

The initial importance of components based on the unreliability of the given component $F_i(t)$ taken from operational research and number of path sets x_i, in which the i-th component is participating out of x of all path sets of the system will be determined:

$$I(t) = f[F_i(t), x_i, x] \quad (1)$$

The author proposes introducing measures describing the importance of the component including appropriate criteria. It will be determined by implementing appropriate rates of weight coefficients for various criteria, so as the criterion of the time required for maintenance, of circulation of the participation of the staff, maintainability, time required for spare parts delivery, the economics of the service and the operational safety. It is possible then to describe the importance of the component:

$$I^{KRYT}(t) = f[F_i(t), x_i, x, c_t(t), c_p(t), c_m(t), c_s(t), c_e(t), c_b(t)], \quad (2)$$

where:

$F_i(t)$ – unreliability of the i-th component of the system,
$c_t(t)$ – weight coefficient of circulation of the working time for performing the restoration of the component,
$c_p(t)$ – weight coefficient of participation of the staff for performing the restoration of the component,
$c_o(t)$ – weight coefficient of maintainability (service susceptibility) of the component as part of the restoration;
$c_s(t)$ – weight coefficient of waiting time to the spare parts for performing the restoration (repairs) of the component,
$c_e(t)$ – weight coefficient of costs of the maintenance of the component; $c_b(t)$ – operational safety change due to component failure.

Finding suitable functional forms new measures will be one of the tasks carried out as part of the work. Lack of complete information on system reliability structure and reliabilities of the system components will be solved with using of qualitative models (comprehensive – qualitative-quantitative) and with application of the experts knowledge described using rules of concluding and the subjective probability [4].

Results of the author's preliminary research [5,10] show the rightness of adopted assumptions and lack of possibility of the evaluation of the importance of components and groups of components of CTS using the known methodology.

4. Research methodology

Measures drawn up will be applied for model CTS which the vessel propulsion plant installed onboard the transport maritime vessel. Conception of comprehensive attempt at the evaluation of the importance of components and groups of components in CTS with the application experts knowledge is shown in the Fig 1.

The author put forward the theses, that: "full evaluation of the importance of components in the CTS reliability structure is possible only with the use of the comprehensive attempt by applying qualitative-quantitative methods" and that "for the global assessment of the importance of system components one should widen evaluation criteria introducing measures describing consequences of corrective (maintenance) action after fail of given system component".

Analysis results will be assessed by comparing gained importance rankings of components for different initial information about the examined system, in particular:

- coarse information about functional components and relations in the system;
- reliability structure well-known and system components reliabilities unknown;
- system components reliabilities and system reliability structure generally known;
- system reliability structure well-known and selected system components reliability known.

Research methodology has been schematically presented in Fig. 2. So far the author carried out preliminary, essential tasks (**already finished**) for the purposes of the new methodology implementation, such as:

- the identification of factors justifying the need of modifying evaluation tools for the reliability importance analysis of components and groups of components in CTS;

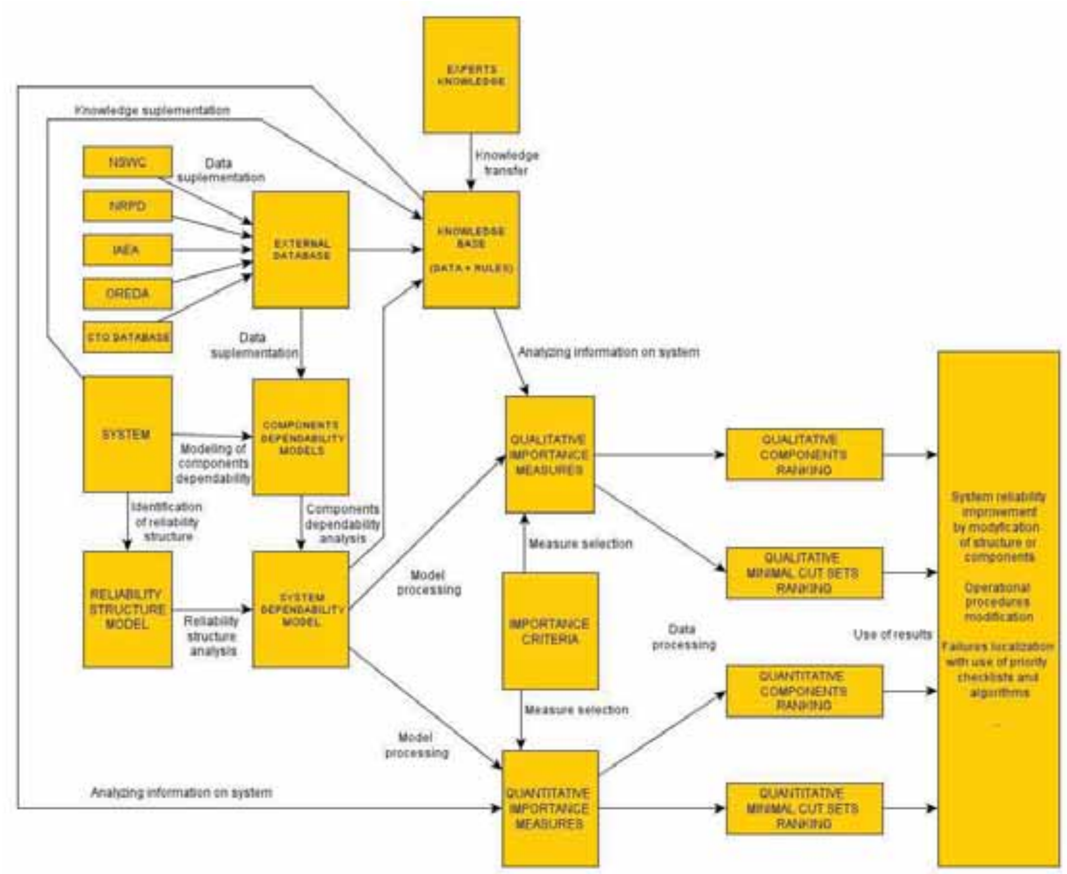

Fig. 1. Proposed comprehensive qualitative-quantitative importance analysis

- the review of known measures of the importance and models of the structure for CTS and the literature critical analysis of achievements of other researchers of the subject matter of the evaluation of the importance of components in CTS;
- tidy the terminology up in the analysis of the importance of components as well as the description of the system reliability structure;
- collecting statistical data about failures of components of systems of vessel propulsion plants installed onboard many transport vessels.

The main goal of the future work will be fulfilled by performing individual detailed tasks (**to be done**), which are:

- developing new measures of the importance of system components for various criteria of the importance (maintainability and maintenance costs, operational safety etc.) which will be the alternative to measures presently applied;
- developing new models (of way of the description) of reliability structure and redundancy (reserving in CTS), e.g. proposed by the author the application of the complex numbers plane [6], proposed by the author applying the external events vector [8,9] and applying of the modified measure of reservation by Jaźwiński and Smalko [11];
- developing the comprehensive attempt at the evaluation of the importance of system components by using quantitative-quality models;

- evaluation of the importance of components with proposed methods for chosen technical systems (exemplification based on vessel propulsion plant);
- assessment of the impact of applied methodology and the assortment of measures of the importance to gained rankings of the importance of components and groups of components by mutual comparing of rankings taking under consideration measures definitions;
- discussion on usefulness of known and newly developed importance measures with exemplificative examples;

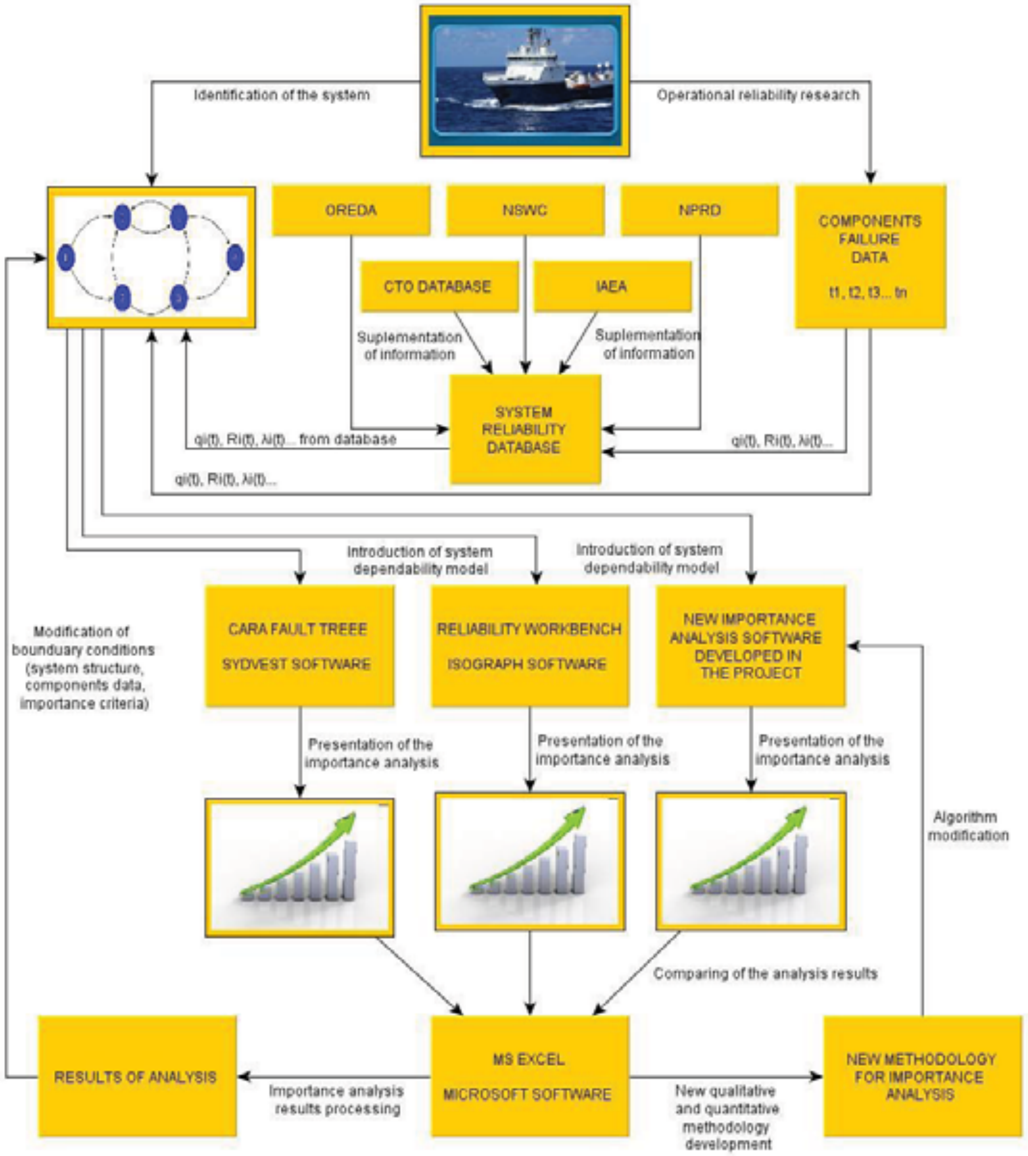

Fig. 2. Basic steps of research methodology for drawing up the modern comprehensive methodology of the evaluation of components importance and groups of components importance in CTS

- developing the general algorithm of the evaluation of the importance of components, in it of assortment of methodology and measures for the evaluation of the reliability importance depending on available information on the system;
- publication of new methodology results in scientific papers.

As part of examinations a vessel propulsion plant of a cargo ship will be an object of analysis as the example CTS. Literature analysis of the subject matter was the first research stage, next action will consist in drawing up new measures of importance and algorithms of acting in the evaluation importance of components and groups of components into CTS. It will be done by synthesis of current importance measures and quantitative and qualitative characteristics of the reliability of the analyzed system. Obtained measures will be used for estimating the value of the importance and creating rankings of the components importance. The lack of complete information about the system will be compensated by supplementing this information using the knowledge base (information gathered systematically from the system and the knowledge of experts). The knowledge of experts will be entered into analysis with using of subjective probability. As comparative results analogous rankings will be made based on different measures.

In the analysis comparing rankings gained from individual analyses of the importance for different initial details about the given CTS will be done. Rankings will allow to quantify through statistical processing of results get based on individual measures of the importance.

Calculations can be performed with use of professional software by known worldwide companies: Isograph Software (Reliability Workbench, Isolib libraries) and Sydvest Software (Fault Tree Analysis Academic Version). Comparative analyses will be carried out in the Microsoft Excel spreadsheet.

As part of the new methodology implementation a modern software will be drawn up being used for analysis of CTS components importance (in which a proposed comprehensive methodology for the evaluation of the reliability importance will be implemented).

5. Final conclusions

The new approach to the importance analysis will contribute to draw up modern methods, models and measures which allow to the evaluation of the system component importance in the reliability CTS structure. Additionally, implementation of new measures will contribute to: criticism of the current importance analysis methods; comparing both the evaluation of the usefulness of different importance analysis methods and measures; developing the algorithm of the rational evaluation of the importance of components and groups of components in CTS depending on information about the system; and of unification of the terminology in the area of importance analysis.

As a result of the proposed methodology implementation a series of statistical results will be obtained, which will be useful in further research, so as: details about importance of analyzed CTS components, estimating the reliability and availability of analyzed CTS, rankings of the importance of components in analyzed CTS and comparing the evaluation results from calculations made with various methods and for various criteria of the importance.

Due to interdisciplinary natures of issues being a subject of the plan, execution of particular tasks will affect the development of the civilization and the society using complex objects of the operation in many aspects of the daily living. Results can be applied in designing new systems and in operation of existing systems (the evaluation of the technical condition of the system, developing exploitation procedures and the alteration of the system in the destination of increasing the reliability).

With reference to CTS (e.g. vessel propulsion plants) the new methodology implementation will give utilitarian results, due to useful methodology drawn up for:

- aiding the CTS managing staff as the tool for assessment and developing maintenance schedules and programs;
- aiding CTS operators by means of diagrams, graphs, priority checklists of inspections and exploitation procedures created with use of the importance analysis.

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POSSIBILITIES OF USING THE FREE-END OF CRANKSHAFT IN DIAGNOSIS OF SLOW SPEED MARINE DIESEL ENGINES

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Abstract

In this article there are discussed possibilities in diagnosis of large marine diesel engines by utilization the free-end of crankshaft in gaining information about engine load in particular cylinders on the basis of course of crankshaft's transient rotational speed overlapped by torsional vibration. There is defined and calculated crankshaft's transfer function for torsional vibration being crankshaft's response to exciting force caused by crank pin efforts. There is presented methodology of reconstruction of crank pin efforts' course on the basis of calculated transfer function and measured in any given condition course of crankshaft's speed. There are indicated that similarities between courses of torsional vibration in time and frequency domain for slow and medium speed marine diesel engines will allow to apply the presented methodology of reconstruction of crank pin efforts' course in large marine diesel engines.

Keywords: *diagnostics, marine diesel engine, crankshaft, torsional vibration, transfer function*

1. Introduction

In searching new solutions and improvements in actually offered diagnostic equipment and systems for diagnosis of diesel engines more and more frequently elements of vibration diagnosis have been utilised. The reasons are given for that the measured at the free end of crankshaft: axial vibrations include information about crankshaft's alignment and crankpin bearings' condition [2], and torsional vibrations about main bearings' condition and the load in particular engine's cylinders [1, 2, 5]. Well known negative feature of vibration signals' measures is their random characteristic resulted from that their values are dependent not only on widely understandable condition which they are focused at but on load as well that is comprehensible as the load of engine and his particular piston – connecting rod sets. Such defined load is estimated by combustion force that value corresponds to transient values of combustion process in function of °ACR (angle of crankshaft rotation). Because to permanent and/ or effective purposes of diagnosis, usefulness of direct measurement of combustion pressures' courses is still not sufficient, and moreover limitation and problems, that characterise these methods, cause that they have not brought expected results yet, torsional vibrations as source of information about load have been adopted. The free end of crankshaft has began the place of interest for application angle encoders as °ACR indicators for combustion analysers – fig.1 [4] and cylinder liner lubricators – fig. 2 [3]. They are primary sources of information in relation to eddy current sensors (proximitors) applied close to engine‘s fly-wheel that use teeth or special marks as °ACR indicators. Adaptation of free-end of crankshaft with it extension outside of the engine housing was implemented on medium (4-stroke, medium speed) [2] and large (2-stroke, slow speed) diesel engines even with presence of axial damper [3, 4].

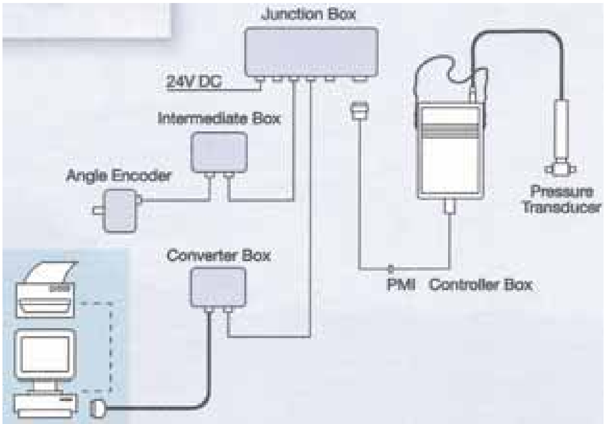

Fig. 1. PMI System, cylinder pressure analyser with angle encoder assembled on free-end of crankshaft designated for slow speed diesel engines by MAN B&W Diesel A/S [4]

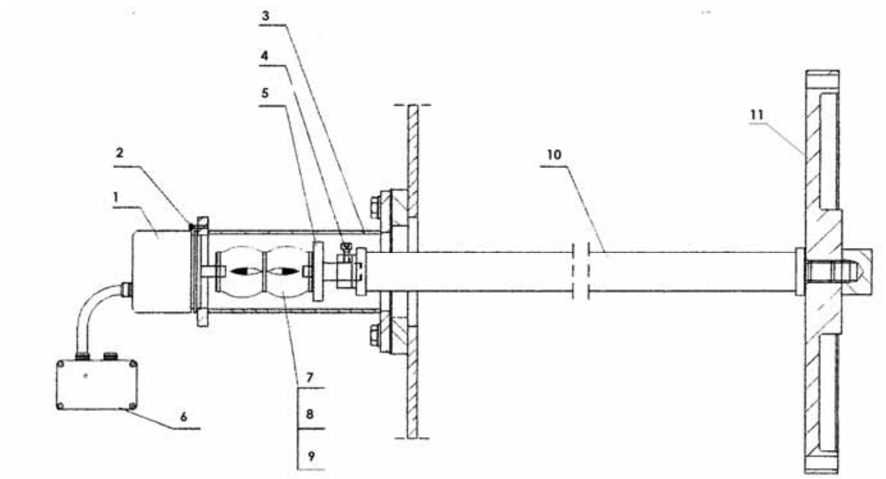

Fig. 2. Alpha Lubricator System, angle encoder installation diagram [3]:
1 –angle encoder, 2 – assembling screw, 3 – housing, 4 – securing screw, 5 – connecting piece, 6 – intermediate box, 7 – coupling, 8 – damper plate, 9 – cable ties, 10 – shaft, 11 – free-end of crankshaft (axial vibration damper)

For the purpose of measurement and analyse of axial and torsional vibration of crankshaft, non-uniformity of engine running and engine load in function of °ACR the following system to experimental researches on medium diesel engine (Sulzer 6AL24) [2] was implemented – fig. 3:

- Integrated Transducer of Axial and Torsional Vibrations (5-10) attached to free end of crankshaft (1) and body of engine (2) through blinding cover (3) and seal (5) together with measuring plate and shaft (4),
- Opto-electric Transducer rev/impulse with coupling type OLDHAM – Megatron (10),
- Eddy Current Sensor and Relative Vibration Measurement Instrument ZPW-2 by Sensor (8-9),

- System to Indicated Power Measurement (12) including: opto-electric key phasor by Sensor (13), 6 tensometric sensors of combustion pressure PT - 5101T and adaptation – antiknock heads by Unitest, 6 amplifiers AT – 5230 with autozero and stabilised BZ 5205 by Spais.
- Computer Diagnostic Analyser KSD – 400 by Sensor (11) including: notebook with software, 16 – channels PC CARD PCMCIA type DAS16S/330, input voltage cards and amplifiers.

a)

b)

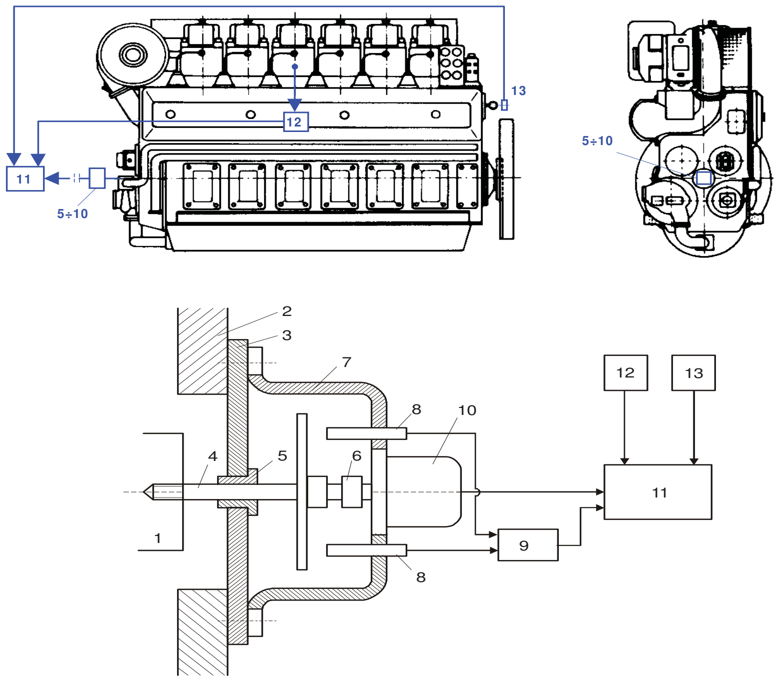

Fig. 3. Means of measurement of vibration signal and engine load in function of °ACR
a – measuring points at the engine, b – scheme of integrated vibration transducer
1 – free end of crankshaft , 2 – body of engine, 3 – blinding cover, 4 – shaft and measuring plate, 5 – seal, 6 – coupling , 7 – housing, 8 – eddy current sensors , 9 – signal adder , 10 – opto-electric transducer , 11 – computer diagnostic analyser, 12 – indicated power measuring unit, 13 – key phasor

For torsional vibration measurement on large diesel engines (Mitsubishi 7UEC72LII) angle encoder was used with same mounting as on fig. 2 (coupling and damping plates).

2. Crankshaft's transfer function for torsional vibrations

Crankshaft as a mechanical element of diesel engine possesses given mass, elasticity and damping. These magnitudes determine so called transfer function that means decide about crankshaft's response to exciting forces. Depending on place and direction of acting of exciting forces, crankshaft can execute torsional vibration being crankshaft's response to force caused by crank pin efforts and axial vibration being crankshaft's response to force caused by crankthrow efforts. Transfer function $H(j\omega)$ between these dependencies can be defined as a transmittance of researched object (crankshaft) – fig. 4.

$$F(j\omega) \longrightarrow \boxed{H(j\omega)} \longrightarrow X(j\omega)$$

Transfer function $H(j\omega)$ as a transmittance of researched object [1, 2]:

$$H(j\omega) = \frac{X(j\omega)}{F(j\omega)}$$

Fig. 4. Definition of transfer function H(jω)
X(jω) – response as crankshaft's axial or torsional vibrations, caused by force being of resultant force F(jω), correspondingly to crankthrow and crank pin efforts

In researches at utilisation of transfer function in diagnosis of diesel engines, crankshaft's transfer function for axial vibration has been utilised as a source of information about crank pin bearings' condition and for torsional vibrations as a source of information about the course of load in particular engine's cylinders. Taking into consideration dependencies between force in form of crank pin efforts and response in form of torsional vibration of crankshaft's free end, and treating crankshaft and dependencies existed in it and his bearings as so called "black box", crankshaft's transfer function for tosional vibration can be defined as inversion of the following transmittance:

$$H(TV(i)) = \sum_{i=0}^{n} \frac{T_{gas+m}(i)}{TV(i)} \tag{1}$$

where:

$T_{gas+m}(i)$	–	harmonic components of crank pin efforts,
$TV(i)$	–	harmonic components of torsional vibration,
i	–	harmonic order,
n	–	top harmonic order.

3. Calculation of crankshaft's transfer function

Transfer function $H(TV(i))$, has been calculated according to formula (1), possessing data in form of amplitude and phase spectrums calculated from course of crankshaft's transient rotational speed v (fig. 7 and 8), being response to force in form of course of crank pin efforts t (fig. 5 and 6). Amplitude components of transfer function were obtained from quotient of average values of amplitude spectrums of crank pin efforts and crankshaft's speed:

$$\Phi\ (\mathrm{A}) = \frac{t_{\mathrm{aver}}(\mathrm{A})}{v_{\mathrm{aver}}(\mathrm{A})} \tag{2}$$

where:

Φ (A)	–	amplitude components of transfer function,
t_{aver}(A)	–	average values of amplitude spectrums of crank pin efforts [MPa],
v_{aver}(A)	–	average values of amplitude spectrums of crankshaft's speed [rev/min].

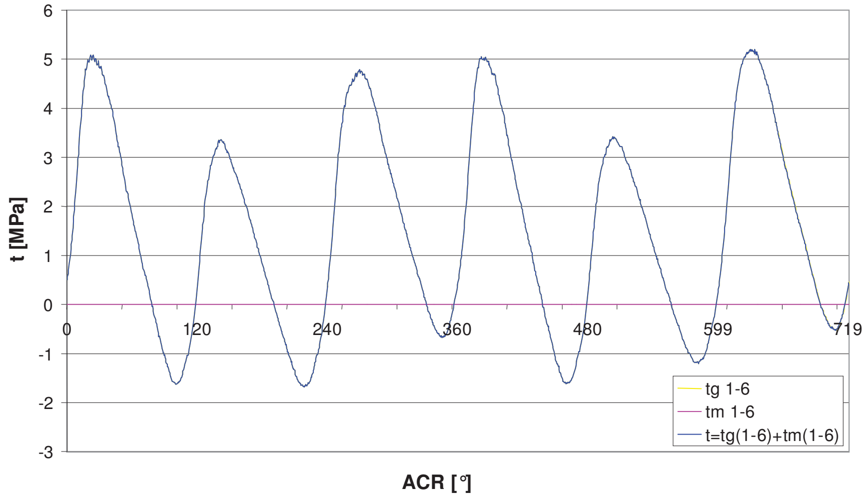

Fig. 5. Course of the resultant from gas t_g and mass t_m components total crank pin efforts t in particular cylinders at 720 rev/min and load index 100% for medium marine diesel engine Sulzer 6AL24

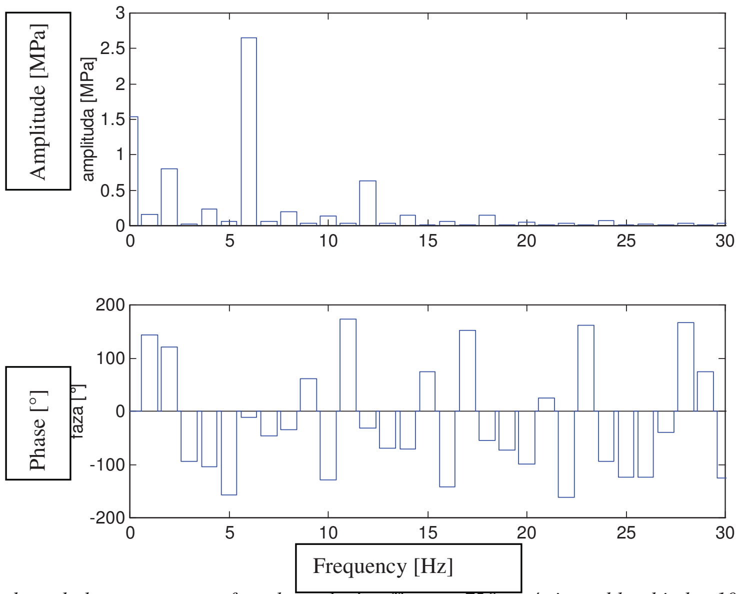

Fig. 6. Amplitude and phase spectrum of total crank pin efforts at 720 rev/min and load index 100% medium marine diesel engine Sulzer 6AL24

Phase components of transfer function were obtained from difference of average values of phase spectrums of crank pin efforts and crankshaft's speed:

$$\Phi(\mathrm{F}) = t_{aver}(\mathrm{F}) - v_{aver}(\mathrm{F}) \tag{3}$$

where:

Φ (F) – phase components of transfer function,
t_{aver}(F) – average values of phase spectrums of crank pin efforts [MPa],
v_{aver}(F) – average values of phase spectrums of crankshaft's speed [rev/min].

On the basis of quotient of harmonic components of amplitude spectrums (2) and difference of harmonic components of phase spectrums of crank pin efforts and crankshaft's speed (3) there was determined set of amplitudes and phases values of transfer function:

$$H(TV(i)) = \{\Phi(A),\Phi(F)\} \tag{4}$$

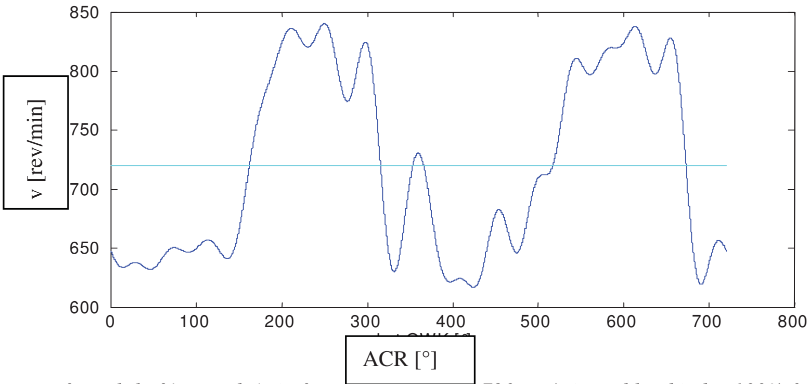

Fig. 7. Course of crankshaft's speed Δv in function of °ACR at 720 rev/min and load index 100% for medium marine diesel engine Sulzer 6AL24

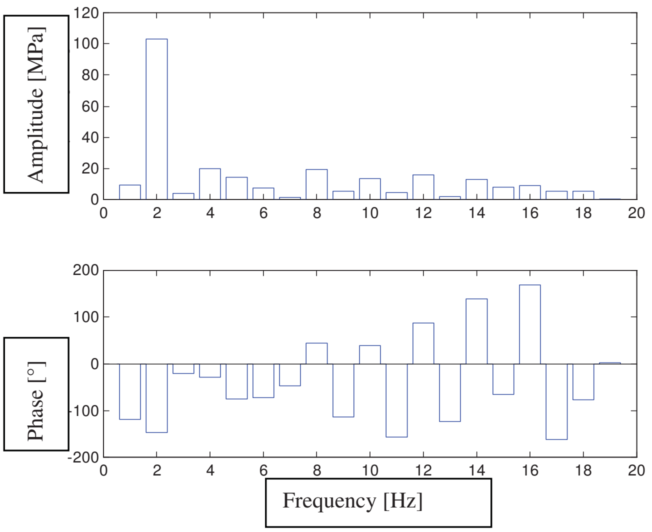

Fig. 8. Amplitude and phase spectrum of crankshaft's speed Δv at 720 rev/min and load index 100% for medium marine diesel engine Sulzer 6AL24

As it can be observed from fig. 9 and 10 the presented courses of torque fluctuation and its harmonics for large marine diesel engine have same character as for medium marine diesel engine depending on number of cylinders, fire order, crank-web's angles between particular cylinders and taking into consideration the number of cycles coinciding with engine's revolutions. Due to lack of technical possibilities (6 sensors) of simultaneous measurement of combustion pressure changes in particular cylinders and calculation of the resultant from gas t_g and mass t_m components total crank pin efforts t in particular cylinders, the contained in chapter 4 application of transfer function to reconstruction of course of crank pin efforts was made therefore only for measuring results obtained for medium marine diesel engine, however same methodology can be utilized for large marine engines.

Fig. 9. Course of torque fluctuation t in particular cylinders at 72 rev/min and load index 80% for large marine diesel engine Mitsubishi UEC 7L85II

Fig. 10. FFT of torque fluctuation t in particular cylinders at 72 rev/min and load index 80% for large marine diesel engine Mitsubishi UEC 7L85II

4. Application of transfer function to reconstruction of course of crank pin efforts

On the basis of calculated transfer function *H(TV(i))* there was made an attempt to reconstruct course of total crank pin efforts from measured at crankshaft's free end, in any given condition course of crankshaft's transient rotational speed.

In this purpose calculated amplitude spectrums of measured course of crankshaft's speed have been multiplied by amplitude components of transfer function Φ(A):

$$t_{tv}(\mathrm{A}) = \Phi(\mathrm{A}) \cdot v_{meas}(\mathrm{A}) \tag{5}$$

where:

$v_{meas}(\mathrm{A})$ – harmonics values of amplitude spectrums of measured course of crankshaft's speed [rev/min],
$t_{tv}(\mathrm{A})$ – harmonics values of amplitude spectrums of reconstructed course of crank pin efforts [MPa].

and to phase spectrums have been added phase components of transfer function Φ(F):

$$t_{tv}(F) = \Phi(F) + v_{meas}(F) \tag{6}$$

where:

$v_{meas}(F)$ – harmonics values of phase spectrums of measured course of crankshaft's speed [˚],
$t_{tv}(F)$ – harmonics values of phase spectrums of reconstructed course of crank pin efforts [˚].

On the basis of formulas 5 and 6 there have been calculated values of amplitude and phase spectrums of crank pin efforts t_{tv}, moreover utilising synthesis – sum from definition of trigonometric series of particular harmonics of both spectrums (formula 7) there have been obtained course of total crank pin efforts t_{tv} in function of °ACR (fig. 11).

$$t = t_o + t_k \cdot \sin[(k-1)\omega t + \varphi_k] \tag{7}$$

where:

t_o – zero harmonic of amplitude spectrum of total crank pin efforts [MPa],
t_k – next harmonics of amplitude spectrum of total crank pin efforts [MPa],
φ_k – next harmonics of phase spectrum of total crank pin efforts [˚].

Attempt of utilisation of transfer function confirmed correctness of reconstruction of course of total crank pin efforts from courses of crankshaft's speed in range of load 50-100%. In lower range of load irregularity and stochastic character of combustion process and connected with it non-uniformity of engine's speed caused oscillation of revolution. These oscillations have not important influence at harmonics of amplitude spectrums but caused significant differences for harmonics of phase spectrums. Next these differences caused important inaccuracy in calculation and reconstruction courses of crank pin efforts.

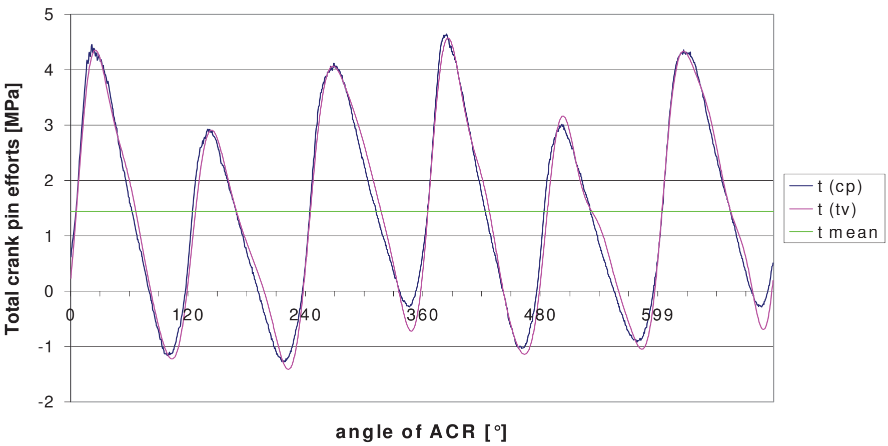

Fig. 11. Comparison of course of total crank pin efforts calculated on the basis of combustion pressures t_{cp} with course of total crank pin efforts obtained by multiplication of measured crankshaft's speed by transfer function t_{tv}

Conclusions

As it results from the above there is possibility to use the free end of crankshaft in diagnosis of marine diesel engines including large two stroke slow speed engines and to define crankshaft's transfer function in form of quotient of crankshaft's speed and total crank pin efforts spectrums.
Besides, there is possibility to reconstruct from course of crankshaft's speed and determined transfer function at defined load (but above 50 % of nominal engine load) the course of total crank pin efforts and to determine the load in particular engine cylinders. Moreover, there is possibility to apply the presented methodology of application of transfer function for large marine diesel

engine to gain information about load in particular cylinders but it will demand to create separate characteristic between exciting data in form of courses of crank pin efforts and response in form of courses of torsional vibration/ crankshaft's transient rotational speed in purpose of calculation of transfer function for particular engine.

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DYNAMICS OF THE COMPUTER-AIDED SYSTEM FOR COMPLEX TECHNICAL OBJECT MAINTENANCE IN THE UML LANGUAGE

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This paper is an attempt to use modeling methods employing the UML language for computer modeling of the complex object maintenance systems. A particular emphasis was put on the description of the system dynamics understood as changeable behavior resulting from the interaction with environment. The application case which has been selected here is a computer-aided system for railway track vehicle maintenance.

Keywords*: UML application, maintenance system design, track vehicle*

1. Introduction

Economy of any modern country possesses many objects of great value and a relatively long life-cycle within which the phases of utilization and maintenance occur interchangeably in accordance with the rules of object utilization (the means of transport can serve here as an example, e.g. road vehicles, boats or airplanes). The cost of an object purchase is frequently much lower than later outlays on its operation and maintenance in the course of its life-cycle. At a certain moment the rational approach to the problem of maintenance can lead to a decision about a withdrawal of technically still worthy vehicles but outdated because new objects appear on the market whose technical parameters are much better or the costs of their maintenance are much lower [4].

A synthetic analysis of the phenomena and making the right decisions related to them is impossible without appropriate computer systems providing support for business processes. The need to create the Computer Aided Maintenance Systems (CAMS) results also from changes in the service systems and replacing traditional maintenance-repair routines by servicing based on the actual state which in turn is based on current diagnostics. Thus, it is vital to gather data on reliability and maintenance parameters for particular objects which undergo maintenance. In this case assistance is provided by diagnostic systems on board. However, in order to collect information referring to the entire number of objects and to draw more general conclusions, its aggregation is necessary.

Whereas representation of economic events is fairly well developed, the technical events are often insufficiently illustrated and in practice many attempts of developing such systems have failed [8]. The major reason for such a state of affairs is the immense complexity of systems [7]. Apart from other causes mentioned in this paper [7], the said failure results also from:

- a great number of events of extreme variety which must be taken into account,

- communication problems within designers' teams as well as among their members and users in the phase of formulating assumptions.

Consequently, the final product requires multiple changes, corrections and supplements which lead to a serious extension of the time scheduled for work and significant exceeding of the preplanned budget [8]. As a result, the systems created – if they are created at all – are in a version significantly reduced in relation to original plans and their costs are relatively high.

Another serious difficulty is the fact that each time the system is created from scratch without taking advantage of the existing partial solutions. Partial results are most often inaccessible as they are the property of the companies which ordered them or software manufacturers.

Among many [2] possible ways of reducing complexity, one of the most promising is system modeling before its execution is started, or, consequently, creating design models. This paper is an attempt to apply the modeling methods with the use of the UML language for a design of a complex object maintenance system. The application cases were developed on the basis of the computer aided rail vehicle maintenance system.

2. System modeling

Creating model systems has this advantage that a model is not only a fragment of the documentation of the created system but it is possible to verify the correctness and completeness of the system being developed before it is actually created. In this way the number of indispensable modifications required for the preparation of a final product is limited.

Together with the growing complexity of systems, the number of people involved in the design grows. Consequently, significant communication barriers arise and then models become a basic tool of documenting mutually agreed solutions. Naturally, to this end the programming tools from the CASE group are necessary.

A model system consists of:

- ❖ Information components – static description – database architecture
- ❖ Result information and the way of its presentation – user's interface
- ❖ Data processing method – application logic

This is one of the most frequently used system divisions – a division into layers. The essence of the layer architecture is to construct the application structure in such a way as to make the functions of particular layers independent of one another.

Some authors distinguish more layers. For example, a 5-layer architecture is outlined in work [6].

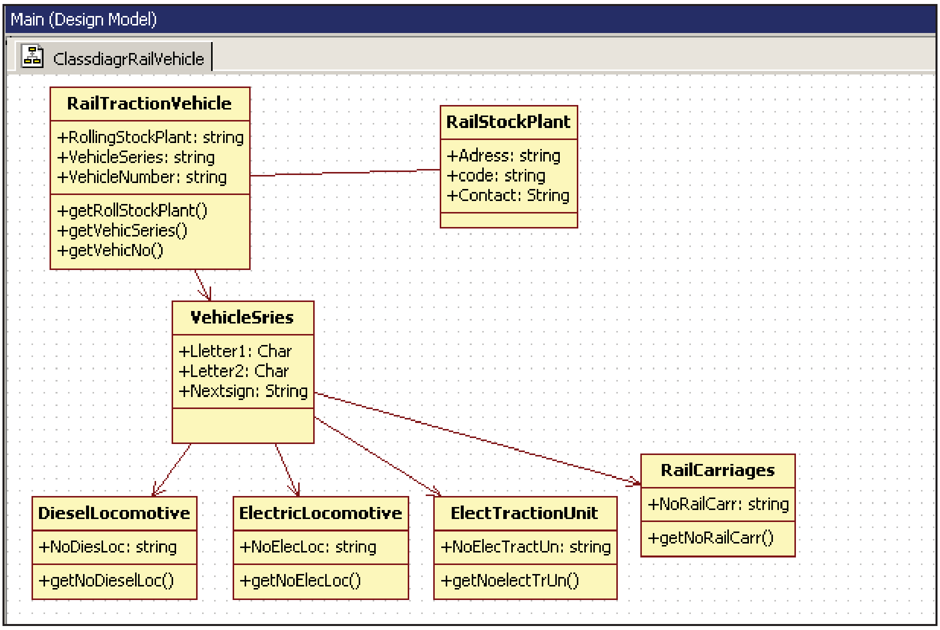

Fig. 1 Simplified model of the rail traction unit description class

A significant part of the static description is possible to achieve by means of class diagrams. They show the attributes constituting a class as well as methods operating on them. Figure 1 shows a simple model of classes included in a description of a rail traction vehicle which is the subject of the rail traction vehicle CAMS.

This model enables grasping the interactions among descendant class attributes contained in a description and the methods operating on them. It also allows us to distinguish between private and public methods.

The advantage of class diagrams and models achieved with their help is a possibility of automatic generation of programme codes on their basis.

```
Katalog: D:\_PR\Publikacyjne\Cimac11\railvehicle
10:55                254 DieselLocomotive.cpp
 10:55                354 DieselLocomotive.h
 10:55                258 ElectricLocomotive.cpp
 10:55                362 ElectricLocomotive.h
 10:55                257 ElectTractionUnit.cpp
 10:55                363 ElectTractionUnit.h
 10:55                244 RailCarriages.cpp
 10:55                339 RailCarriages.h
 10:55                349 RAilStockPlant.h
 10:55                368 RailTractionVehicle.cpp
 10:55                474 RailTractionVehicle.h
  10:55                280 TractVehicle.h
  10:55                341 VehicleSries.h
```

Fig. 2 Files generated on the basis of a class diagram presented in Fig. 1

```
//
//  Generated by StarUML(tm) C++ Add-In
//
//  @ Project : RollingStockPlant
//  @ File Name : RailTractionVehicle.cpp
//  @ Date : 11-06-16
//  @ Author : Andrzej Erd
/

#include "RailTractionVehicle.h"

void RailTractionVehicle::getRollStockPlant() {
}

void RailTractionVehicle::getVehicSeries() {
}

void RailTractionVehicle::getVehicNo() {
}
```

Fig. 3 Contents of a File „ (RailTracktionVehicle.cpp)"

Depending on the tool applied, the code can be generated in different languages, e.g. C/C++, Java, Visual Basic, Delphi, JScript, VBScript, C#, VB.NET. For the sake of example the StarUML programme was used and the language set for generation was C++. The effect is presented in Figures 2 and 3.

It is noteworthy that all necessary code files (.cpp) as well as header files (.h) were created. The files generated contain all attribute declarations and methods, which can be clearly seen in the presented file "PojazdTrakcyjny.cpp" (RailTrackVehicle.cpp). It is obvious that in the targeted model of the system, the sets of attributes for each class should be more extensive [3]; the same refers to the list of available methods.

It is extremely important for tool programmes to check the correctness of declarations. Correctness is checked in its formal aspect. What is also checked is completeness of all declarations and coherence between individual classes. It is obvious that the method code depends on the programmer's will and must be completed by him. However, support of the tools is also significant

3. Models of system dynamics

The term "system dynamics" denotes system changeability, not only in terms of parameters but also in terms of behavior. What happens to the system is visible outside through the user's interface. The Use Case Diagram (UCD) illustrates users' interaction with the system as well as the system's interactions with environment. In this diagram particular groups of users, the so called "actors" are joined to the rounded rectangles representing activities by means of arrows. The scope and degree of information processing depend on the recipient and the goal. Hence the Use Case Diagram allows us to grasp all forms of collaboration between the environment and the system. In the description of activities which is an indispensable supplement to UCD, all aspects of the user-performed activity should be specified.

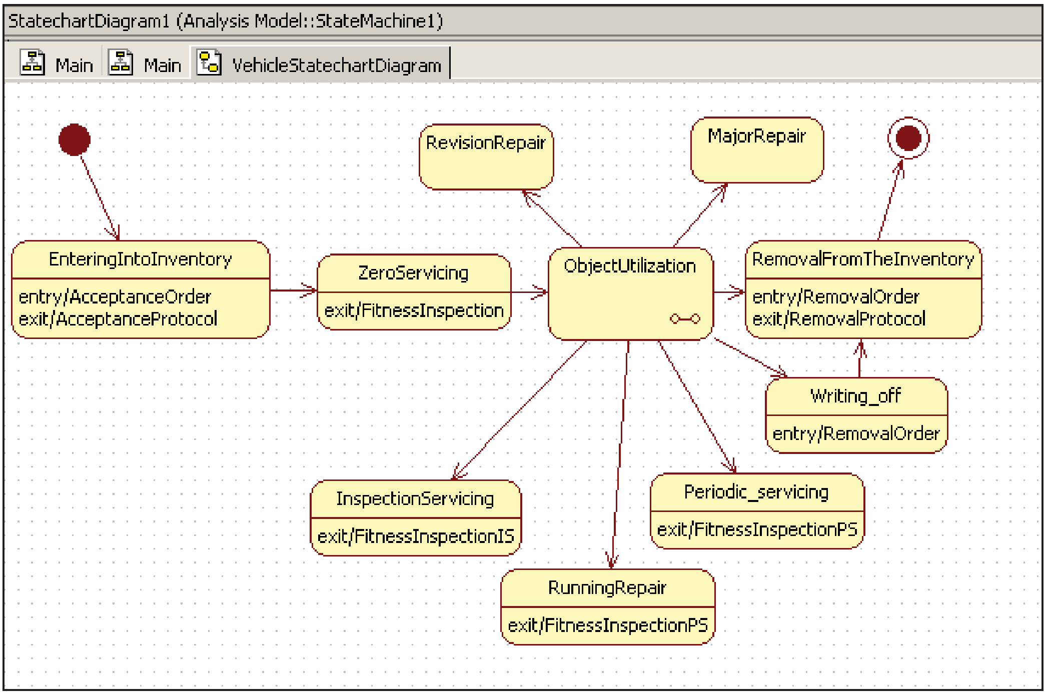

Fig. 4 Statechart Diagram of Vehicle

As a result of entering data by a user (or data obtained independently by a system from other external systems) changes occur in its elements. Generally, individual elements are characterized by certain parameters which allow us to classify them into distinguishable groups. Group affiliation changes depending on activities performed on the system elements. Hence we can say that individual elements are in subsequent states and their changeability (state transition) is illustrated by the State Machine Diagram.

In a rectangle illustrating the state, apart from its name, an activity which is to be performed in an initial state can be added as well as an activity to be performed on exit from a given state. In the case of more extended state diagrams, they can contain sub-states as it is depicted in Figure 5. Basically, the state “Periodic Inspection” takes place when a use-cycle prescribed by regulations for a given series of vehicles is exceeded (this is a condition of entering the state).

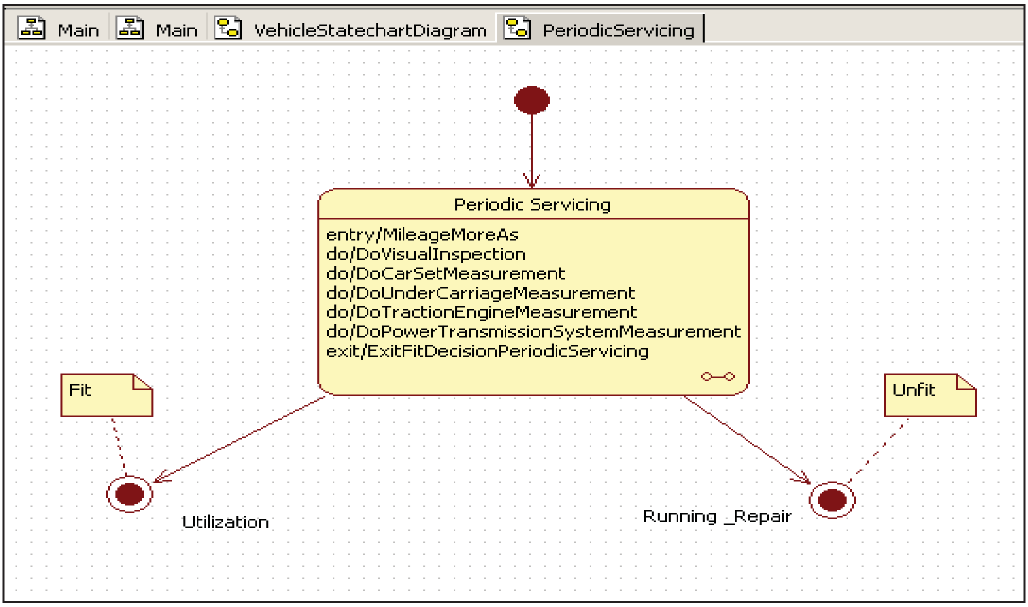

Fig. 5. Diagram of the Periodic_Inspection sub-state

Subsequent methods, such as DoOgledzinyZewn (external inspection), DoPomiaryZK (wheel set measurements), DoPomiarPodwozia (undercarriage measurements), DoPomiarsilnikaTrakc (rail traction engine measurements), DoPomiarUklNapedSS (Diesel engine power transmission system measurement/Diesel engine vehicle) are performed in order to carry out the superior state. In practice they can include:

- records of performing measurement activities on the object,
- methods evaluating the state of particular sub-assemblies on the basis of the system contained data,
- tasks to be performed by external systems, e.g. measurements and inspection.

Depending on whether the method can be implemented immediately or its result will be specified in the future, a decision if a vehicle or object is fit or unfit for further use determines its leaving the final state.

For the sake of example – the method "Measurement of the Diesel Engine Power Transmission System" is a separate task consisting of a number of component procedures covering such measurements as:

- ❖ external characteristic at load state U=f(I) of the main generator ,
- ❖ characteristics at no-load state but at different rotational speeds,
- ❖ dynamic characteristics of the power unit,
- ❖ elementary fuel consumption of a diesel engine at characteristic operation points,
- ❖ fuel consumption per hour at nominal load.

From the above set of tasks which are to be performed one can conclude that obtaining a final result may take even several hours. Consequently, remaining in the Periodic Inspection State can be equally long. Also the Running Repair State may take several days. In such a situation it is important to take into account time interdependencies between events occurring in the system. Sequence diagrams suit this end perfectly well.

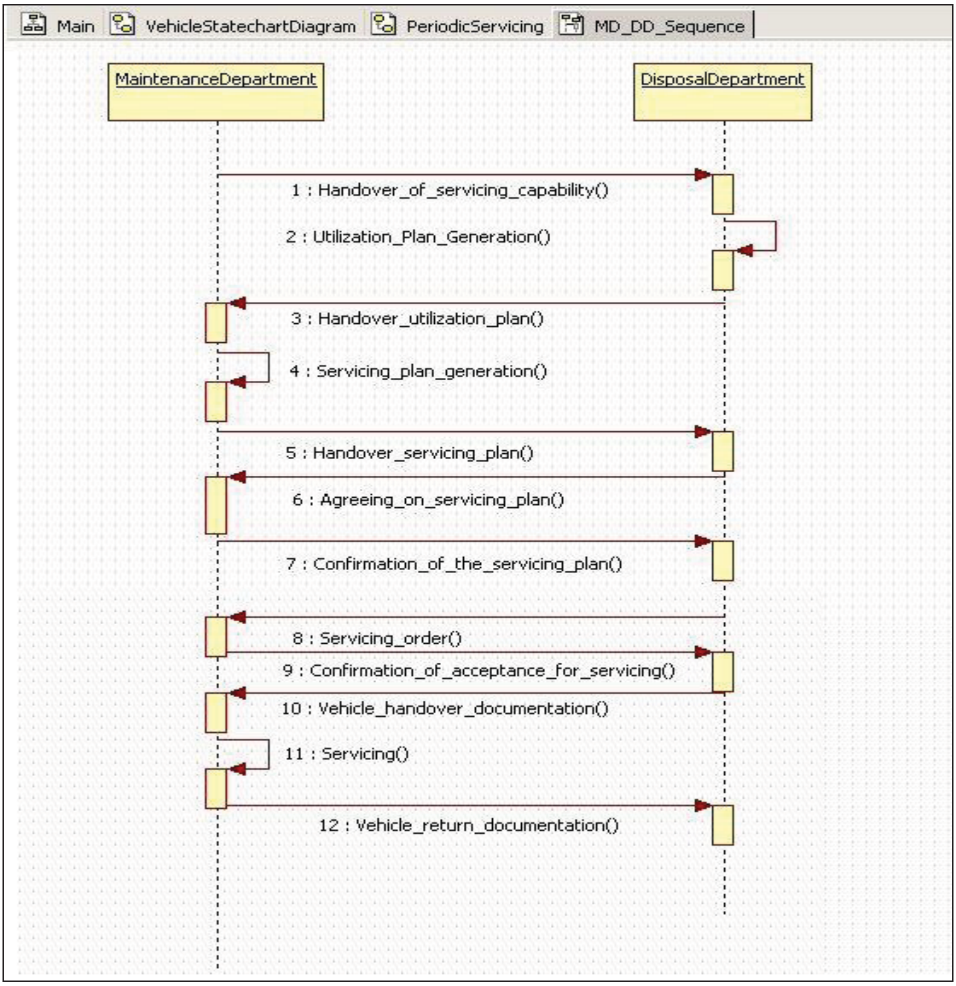

Fig.6 Diagram of communication sequences in the course of a vehicle handover for repair

On the one hand, CAMS should illustrate activities of the Repair Plant and on the other of the Department responsible for the vehicle use. Therefore it seems reasonable to divide the system into two sub-systems: one deals with the Rolling Stock Disposal (DD) and the rolling stock use whereas the other deals with maintenance activities (MD). What combines the two sub-systems are Service Orders on behalf of the Disposal Department and confirmation of service performance and information about the readiness to perform the service on behalf of the Maintenance Department. As services must be performed within a specific time therefore a superior maintenance plan covering time-schedules of the rolling stock use and servicing is necessary. Time interdependencies related to these activities are depicted in Figure 5.

4. Summary

The UML 2.0 specification (ISO/IEC 19501 standard) presented as a model by the Object Management Group, an organization assembling creators of object-oriented methods, includes many other types of diagrams. They allow a model to consider the less common aspects of system operation, e.g. component, collaboration or implementation diagrams. The initial tendency to create separate object-oriented methods depending on application [1] was driven out by the UML language which was originally devised as a tool of modeling and software documentation but it also turned out to be an excellent tool for modeling maintenance and business processes. A further consequence of using the models of the computer-aided maintenance systems seems to be development of design models of such systems which are more universal than the currently existing commercial versions.

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THE PARAMETRIC METHOD OF EVALUATION OF TECHNICAL CONDITION OF THE WORKING TURBOMACHINE BLADE DEPENDING ON THE DISTRIBUTION COURSE REPRESENTING ITS ENVIRONMENT

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Abstract

This article presents tests results of monitoring technical condition of turbomachine blades. The method is based on a diagnostic model $\varphi_{T12,T01}$ that utilizes the difference of phase shift of signals that are the result of blade operation y(t) and a signal x(t) of its environment described with appropriate distribution when the blade moves away from the sensor and when the blade tip approaches the sensor. The adopted diagnostic model indirectly takes into account present blade environment x(t) without the need to measure it [13,15].
The results of diagnosing three blades (worn out to a great extent, considerably worn out and worn out to a little extent) with the use of different distribution courses were presented. Shown results differ to a little extent from each other. However, they unequivocally specify the character of blade wear. Thus, the choice of distribution that represents non-measurable blade environment while determining parameters of a model $\varphi_{T12,T01}$ for evaluation of technical condition of working turbomachine is optional and limited only by observation time.

Keywords: *diagnostics, turbomachine, rotor blades, phase shift, diagnostic model*

1. Introduction

A turbomachine blade is one of basic elements responsible for reliable and safe turbomachine operation. Even slight damage to the blade may lead to damage to a whole machine and in some cases (tearing of a part of blade or the whole blade) even to total destruction of a turbomachine which usually results in tragic catastrophes. Thus, in the process of turbomachine operation a great attention is given to the issues of reliability and diagnostics of turbomachine blades.

Nowadays there are many methods used for diagnosing technical condition of blades during turbomachine operation (the method of eddy currents, the ultrasound method, the radiographical method, the method of colour flaw detection and luminescence flaw detection as well as the vibroacustic method) which work successfully on specific technical objects (SO-3 engines).
Diagnostic inference used so far in the methods of evaluation of blades technical condition is based solely on modification of signals that were measured during diagnostic tests and that are the result of blade operation without sufficient consideration given to signals (of considerable power) of blade variable environment [2÷12, 24, 25, 26].

Measurement of blade environment signals during turbomachine operation is difficult and very often even impossible thus this measurement is not sufficiently considered in blade diagnostics.

Therefore it can be claimed that the methods that have been used so far for evaluation of blades technical condition during turbomachines operation do not entirely fulfill a basic principle of technical diagnostics that demands a test and technical condition analysis of an object in environment (PN-90/N-04002) thus these methods are not accurate nor reliable enough.

Therefore the need arises to develop a new method of diagnostics of blade technical condition during turbomachine operation with taking environment into account, but (if possible) without the need to use the measurement of environment signals that are unavailable and often difficult to measure. This problem can be solved by a blade diagnostics method based on a special diagnostic model that allows to eliminate real existing environment of the blade with the help of special methods.

2. BLADE OPERATION IN NON-MEASURABLE ENVIRONMENT

Fig. 1 shows construction and activity of a blade during its operation in variable environment [18, 21]. The blade consists of two parts: the first one is an operating part, also called a profile part – 1 (a blade) and the second one is a fastening part – 2 (of a lock). The operating part comprises also an edge of attack– 3, a trailing edge – 4, a blade tip – 5, a blade ridge – 6, a trough – 7.

Rotor blades are fastened in a shield with the use of a trapezoidal lock, also called “a dovetail lock”. Blades are covered with an epoxy enamel in order to increase their corrosion resistance.

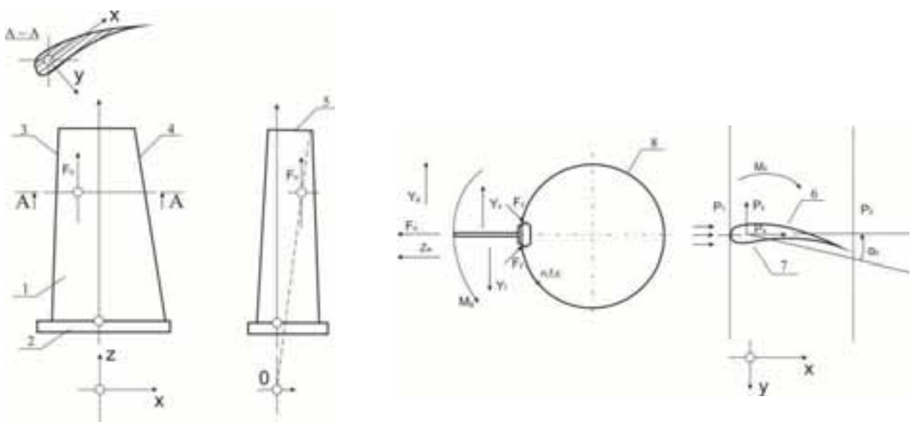

Fig. 1. A turbomachine blade in variable environment.
1 – a blade; 2 – a blade lock; 3 – an edge of attack; 4 – a trailing edge; 5 – a blade tip; 6 – a blade ridge; 7 – a blade trough; 8 – a rotor drum; ; F_0 – centrifugal force; F_z – lock grip force; n - rotational velocity; Y_z- blade aerodynamic lift; P_x - resisting force; M_s – torque moment; M_g - bending moment; P_1 - gas pressure on rotor rim input; P_2 - gas pressure on rotor rim output; Y_g - blade deflection; α_s – a blade torsion angle; Z_w - blade longitudinal displacement; Y_f - a various vibration signal (bending, torsional, longitudinal); Y_c - thermal deformation; f – a vibration signal; c – a temperature distribution signal

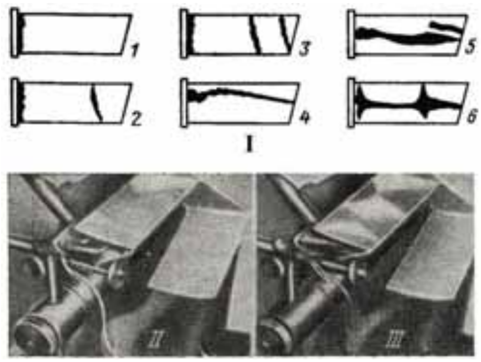

Fig. 2. Forms of vibration and lines indicating traces of vibration nodes.
Chart I: 1,2,3 - first, second and third form of bending vibration; 4 - first kind torsional vibration; 5 - second kind torsional vibration; 6 - combined bending-torsional vibration. Chart II: photograph of nodes traces in second form of bending vibration. Chart III: photograph of nodes traces in third form of bending vibration

During its use, the blade technical condition changes and, with time, various damage appears (such as fractures, deformations, pits, breaks of blade parts) [22].

Fig. 1 and Fig. 2 show that a blade (of a compressor, a turbine) is a technical object with complex principle of operation and that has to be specified with a multidimensional state of blade deformation.

Those deformations originate from environment and are caused by various reasons such as:

- centrifugal force F_0 loads that depend on rotational velocity and cause longitudinal and bending strains (Fig.1) – Z_w, Y_g;
- gas-dynamic Y_z and P_x loads from the stream of air (gas) depending also on flight velocity and altitude (Fig.1) Y_s (if a turbomachine is a compressor or an engine turbine);
- loads Y_g, α_s due to a curvilinear flight path (Fig. 1);
- dynamic loads at mechanical vibration (especially in a resonant range) due to pressure pulsation P_1 and P_2, rotational oscillations etc. (Fig. 2) Y_f;
- blade and casing vibrations f (Fig. 1) and thus Y_g, α_s;
- heat loads c due to uneven temperature distribution (Fig. 2 – complex strain, eg. I – 6) Y_c;

From a synthetic point of view, a state of blade operation in environment can be described by a signal of blade tip displacement y(t) which is a resultant of the following signals: Z_w, Y_g, α_s, Y_f, Y_c, (Fig. 1. and Fig. 2.):

$$y(t) = f\left(Z_w, \alpha_s, Y_g, Y_f, Y_c\right) \tag{1}$$

and an environment signal, x(t) which is a resultant of the following signals: n, F_0, Y_z, P_x, P_1, P_2, f, c (Fig. 1. and Fig. 2.):

$$x(t) = f\left(n, F_o, Y_z, P_x, P_1, P_2, f, c\right) \tag{2}$$

Blade technical condition $S_T(\theta)$ in accordance with diagnostics principles results from relations between the operation signal y(t) and the environmental signal x(t) at the moment of current diagnosing θ_1 and initial diagnosing θ_0.

Therefore the following may be noted:

$$S_T(\theta) = f(y(t)_{\theta_0,}\, x(t)_{\theta_0,}\, y(t)_{\theta_1,}\, x(t)_{\theta_1,}\, \theta, t) \tag{3}$$

Practice proved many times that there are real difficulties in the process of signals measurement, both for y(t), and x(t) (even to a greater extent), and thus also in the evaluation of blade technical condition during machine operation [13, 15, 17, 18].

3. MEASURING POSITION PATTERN

Blade tests were conducted on a turbine engine test bed in the Air Force Institute of Technology (AFIT) in Warsaw. The tested objects were first degree blades of an axial-flow compressor of the SO-3 engine.

In the engine block, a non-contact inductive sensor (or sensor of other type) is mounted for good. (Fig. 3) in order to measure momentary position of compressor blade tips during operation. A signal from the sensor is registered with the use of specialized equipment and saved on a computer. Conducted tests were carried out for minimum rotational speed of 6900 rpm.

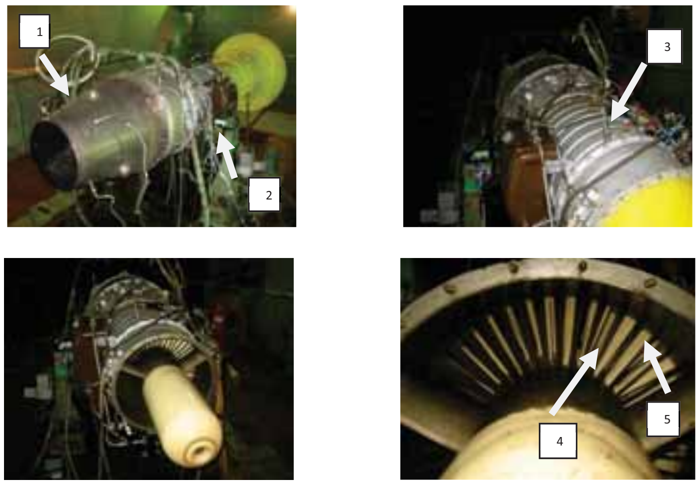

Fig. 3. Measuring position: 1 – SO-3 turbine engine, 2 – measuring device, 3 – non-contact inductive sensor, 4 – compressor blades, 5 – stator blades

4. GROUNDS FOR DETERMINATION OF MODEL $\varphi_{T12,T01}$ PARAMETERS.

A problem of blade diagnostics during turbomachine operation is very complex in its nature as in order to complete a process of blade diagnostics only two signals can be used: the first one is the measurable but interfered signal y(t) and the second one is the environment signal x(t) which is practically immeasurable (except signals n and Δn). It is assumed that environment x(t) is to be represented by distribution $x(t)=\frac{1}{\tau}e^{-\frac{\pi t^2}{\tau^2}}$.

The registered signal of blade tip displacement under the sensor is presented in Fig. 4.

Fixed observation time T_{02} (value T_{02d} or T_{02k}) of blade translocation below the sensor is divided onto two ranges: of blade approaching the sensor T_{01} and receding from it T_{12} (moment T_1 is exactly when the blade tip is below the sensor - Fig. 4.). Adopting long T_{02d} or short T_{02k} time of blade observation is the result of the need to fulfill the conditions of accurate conversion of the signal x(t) into $R_{xx}(\tau)$.

Initially, it is assumed that signals x(t) and y(t) are temporal, stochastic and interfered. In this situation it seems reasonable to switch from space domain "t" of the signals x(t) and y(t) to space domain „τ" of correlation function $R_{xx}(\tau)$ and $R_{xy}(\tau)$ [13, 15, 17, 18, 19, 21].

Afterwards estimates of cross-correlation function R^{*T01}_{xy} and R^{*T12}_{xy} are determined for y(t) translocation in observation periods T_{01} and T_{12} and proper analytic expressions are matched to them [1, 13, 14, 17, 18, 20].

Registered signal courses were multiplied by the Hanning window, then their mutual correlation was calculated. Obtained correlation courses were approximated by fifth degree multinomial with the accuracy of $R^2 > 0{,}997$ described with the coefficient of determination.

$$S_T(\theta) = f(y(t)_{\theta_0}, x(t)_{\theta_0}, y(t)_{\theta_1}, x(t)_{\theta_1}, \theta, t) \tag{4}$$

On the basis of analytic forms of singular correlation functions R^{T01}_{xy} and R^{T12}_{xy}, functions of spectral power density $S^{T01}_{xy}(\omega)$ and $S^{T12}_{xy}(\omega)$ are determined that correspond to them with the use of Fourier transform:

$$F\{R_{xy}\} = \int_{-\infty}^{\infty} R_{xy}(\tau) e^{-j\omega\tau} d\tau \tag{5}$$

$$S_{xy}^{T01}(\omega) = F(R_{xy}^{T01}(\tau)) \tag{6}$$

$$S_{xy}^{T12}(\omega) = F(R_{xy}^{T12}(\tau)) \tag{7}$$

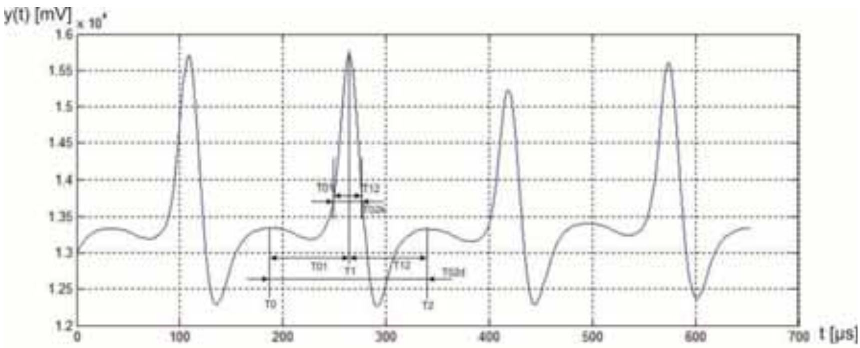

Fig. 4. An inductive sensor signal
T_{02d}, T_{02k} - respectively – a long and short observation period of a blade tip presence in the sensor area, T_0, T_1, T_2 - particular moments of observation of a blade tip under the sensor, T_{01}, T_{12} - observation subperiods of a blade tip for T_{02d} and T_{02k}, respectively, y(t)[mV] – a signal of blade tip displacement, t[μs] - blade displacement time

Expressing functions x(t) and y(t) as $S_{xx}(\omega)$ and $S_{xy}(\omega)$ allows in a very simple manner to take into account relations between diagnostic signals y(t) and environment signals x(t) (Fig. 4).

Thus it may be noted that:

$$\varphi_{T01} = Arg \frac{S_{xy}^{T01}}{S_{xx}^{T01}} \tag{8}$$

$$\varphi_{T12} = Arg \frac{S_{xy}^{T12}}{S_{xx}^{T12}} \tag{9}$$

where:
φ_{T01} – a phase shift of signals x and y while the blade approaches the sensor,
φ_{T12} – a phase shift of signals x and y while the blade moves away from the sensor.

Further it can be assumed that the observation period of T_{12} occurs shortly (ms) after observation time of signals T_{01}.
In this case it may be assumed that:

$$S_{xx}^{T12} = S_{xx}^{T01} \tag{10}$$

Then basing on formulas 8, 9 and 10 and assuming that the environment is eg. noise δ (t, $\hat{t}$) of a great intensity and that it can be correlated with the signal y(t), a new abstract quantity – but the one that can be physically interpreted in the form of phase shifts φ_{T01} and φ_{T12} – can be obtained:

$$\varphi_{T12,T01} = \varphi_{T12} - \varphi_{T01} = Arg\frac{\frac{S_{xy}^{T12}}{S_{xx}^{T12}}}{\frac{S_{xy}^{T01}}{S_{xx}^{T01}}} = Arg\frac{A_{T12}e^{-j\varphi_{T12}}}{A_{T01}e^{-j\varphi_{T01}}} = ArgA_{T12T01}e^{-j(\varphi_{T12}-\varphi_{T01})} \xrightarrow{S_{xx}^{T12}=S_{xx}^{T01}} Arg\frac{S_{xy}^{T12}}{S_{xy}^{T01}} \tag{11}$$

In this manner a new abstract diagnostic model may be determined (of phase shift difference). Its parameters give information about technical condition of the blade being diagnosed [16]:

$$\varphi_{T12,T01} = Arg\frac{S_{xy}^{T12}}{S_{xy}^{T01}} = Arg\frac{B_0 + B_1 s + B_2 s^2 + \ldots + B_5 s^5}{1 + A_1 s + A_2 s^2 + \ldots + A_5 s^5} \tag{12}$$

Difference in the technical condition of the next blades are determined on the basis of relative changes in parameters $M_0 \div M_5$, $L_1 \div L_5$.

$$\Delta\bar{L}_i = \frac{L_{i1} - L_{sr}}{L_{sr}}; \quad i=1,2,3,\ldots n, \tag{13}$$

where:

$L_{śr}$ – average parameter value (reference value, initial)

$$\Delta\bar{M}_i = \frac{M_{i1} - M_{sr}}{M_{sr}}; \quad i=1,2,3,\ldots n, \tag{14}$$

where:

$M_{śr}$ – average parameter value (reference value, initial)

Having calculated relative parameters, μ, σ, 2σ and 3σ are calculated (mean value and standard deviation). Then determined relative value changes into “+” if relative value exceeds σ, „++” if relative value exceeds 2σ, „+++” if relative value exceeds 3σ. In this manner, a blade portrait is obtained which confirms blade aptitude state. If countless “+++” occur, it means that the blade is damaged, "++" means that the blade is considerably worn out, "+" means that the blade is worn out only to a little extent.

Such approach shows clear and unequivocal picture of evaluation of blade damage state.

5. BLADE PORTRAITS FOR DIFFERENT DISTRIBUTION COURSES

Tested objects were first degree blades of the axial-flow compressor of the SO-3 engine. For the purpose of analysis, 3 out of 28 available blades mounted in the rotor drum were taken. The choice was made on the basis of blade wear, i.e. one blade was to be in entirely different technical condition from the remaining ones and thus the most damaged blade was chosen – No. 1, the blade No. 11 that was considerably worn out and the blade No. 3, damaged only to a little extent is.

For the purpose of calculations, distribution $x(t)=\frac{1}{\tau}e^{-\frac{\pi t^2}{\tau^2}}$ was used [23].

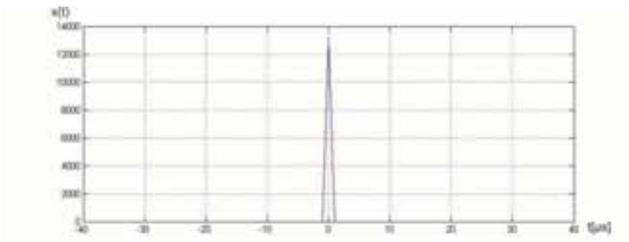

Fig. 5. Graph of distribution for τ =1292

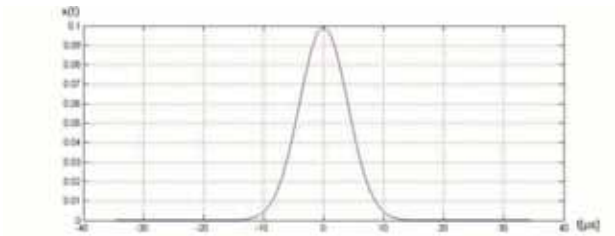

Fig. 6. Graph of distribution for τ =15

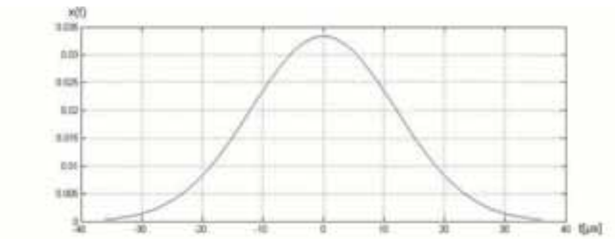

Fig. 7. Graph of distribution for τ =45

Fig. 5 shows distribution diagram for τ =1292 that equals maximum amplitude value of the signal multiplied by the Hanning window, Fig. 6 shows distribution diagram τ =15, Fig. 7 shows distribution diagram τ =45 (τ selected in such a way that the first distribution value was 0 and the number of distribution samples equaled the number of samples of the signal multiplied by the Hanning window). Fig. 8 shows the portrait of the blade No. 1 for distribution course shown in Figs. 5, 6, 7. Fig. 9 shows the portrait of the blade No. 11 for distribution course shown in Figs. 5, 6, 7. Fig. 10 shows the portrait of the blade No. 3 for distribution course shown in Figs. 5, 6, 7. Portraits of blades determined on the basis of formula No.12.

a)

	Portrait of blade No. 1 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1		+	+	+	+							
Cycle 2		++	++	++	++	++	+	+	+	+		+
Cycle 3		+	+	+	++	+			+	+		
Cycle 4		+	+	+	+							
Cycle 5		+	+	++	++							
Cycle 6		+	+	+	+	+						
Cycle 7		++	++	++	++	++	+	+	+	+		++
Cycle 8		++	++	++	++	+	+	+	+	+		+
Cycle 9		++	++	++	++	+	+	+	+	+		+
Cycle 10		++	++	++	++	+	+	+	+	+		+
Cycle 20		++	++	++	++	+	+	+	+	+		+
Cycle 30		+	++	++	++							
Cycle 40		+	+	+	+							
Cycle 50		++	++	++	++	+	+	+	+	+		+
Cycle 60		++	++	++	++	+	+	+	+	+		+
Cycle 70		+	+	+	++							
Cycle 80		++	++	++	++	++	++	++	++	+		++
Cycle 90		+	+	++	++	+		+	+	+		+
Cycle 100		++	++	++	++	+	+	+	+	+		+
Cycle 200		+	+	+	+							
Cycle 300		+	+	+	+						+	
Cycle 400		++	++	++	++	+	+	+	+	+		+
Cycle 500		++	++	++	++	+	+	+	+	+		+
Cycle 600		++	++	++	++	+	+	+	+	+	+	+
Cycle 700		+	+	+	++							
Cycle 800		++	++	++	++	+	+	+	+	+		+
Cycle 900		+	+	+	+					+	+	
Cycle 1000		++	++	++	++	+	+	+	+	+		+
Cycle 2000		+	+	+	+	+	+	+	+	+	+	+

b)

	Portrait of blade No. 1 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1		+	+	+	+							
Cycle 2		++	++	++	++	++	+	+	+	+		++
Cycle 3		+	+	+	+	+						+
Cycle 4		+	+	+	+							
Cycle 5		+	++	++	++	+						+
Cycle 6		+	+	+	+	+						+
Cycle 7		++	++	++	++	++	+	+	+	+		++
Cycle 8		++	++	++	++	+	+	+	+	+		+
Cycle 9		++	++	++	++	+	+	+	+	+		+
Cycle 10		++	++	++	++	+	+	+	+	+		+
Cycle 20		++	++	++	++	+	+	+	+	+		+
Cycle 30		++	++	++	++	+						
Cycle 40		+	+	+	+	+						
Cycle 50		++	++	++	++	+	+	+	+	+		+
Cycle 60		++	++	++	++	+	+	+	+	+		+
Cycle 70		+	+	+	+	+						
Cycle 80		++	++	++	++	++	+	+	+	+		++
Cycle 90		+	+	++	++	+			+	+		+
Cycle 100		++	++	++	++	++	+	+	+	+		+
Cycle 200		+	+	+	+	+						
Cycle 300		+	+	+	+							
Cycle 400		++	++	++	++	++	+	+	+	+		++
Cycle 500		++	++	++	++	+	+	+	+	+		+
Cycle 600		++	++	++	++	+	+	+	+	+		+
Cycle 700		+	+	+	++	+						
Cycle 800		++	++	++	++	+	+	+	+	+		+
Cycle 900		+	+	+	+						+	
Cycle 1000		++	++	++	++	+	+	+	+	+		+
Cycle 2000		+	+	+	+	+	+	+	+	+	+	+

c)

	Portrait of blade No. 1 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1		+	+	+	+	+						+
Cycle 2		++	++	++	++	++	++	++	++	++		++
Cycle 3		+	+	+	+	+	+	+	+	+		+
Cycle 4		+	+	+	+	+						+
Cycle 5		++	++	++	+	++	+	+	+	+		++
Cycle 6		+	+	+	+	++	+	+	+	+		++
Cycle 7		++	++	++	++	++	++	++	++	++		++
Cycle 8		++	++	++	+	++	+	+	+	+		++
Cycle 9		++	++	++	++	++	+	+	+	+		++
Cycle 10		++	++	++	++	++	++	++	++	+		++
Cycle 20		++	++	++	++	++	+	+	+	+		++
Cycle 30		++	++	++	+	++						++
Cycle 40		+	+	+	+	+						+
Cycle 50		++	++	++	++	++	+	+	+	+		++
Cycle 60		++	++	++	+	++	+	+	+	+		++
Cycle 70		+	+	+	+	+						+
Cycle 80		++	++	++	++	++	++	++	++	++		++
Cycle 90		++	++	++	+	++	+	+	+	+		+
Cycle 100		++	++	++	++	++	++	++	++	+		++
Cycle 200		+	+	+	+	+						+
Cycle 300		+	+	+	+	+						+
Cycle 400		++	++	++	++	++	+	+	++	+		++
Cycle 500		++	++	++	+	++	+	+	+	+		++
Cycle 600		++	++	++	+	++	+	+	+	+		++
Cycle 700		+	+	+	+	+						+
Cycle 800		++	++	++	+	++			+			++
Cycle 900		+	+	+	+	+						+
Cycle 1000		++	++	++	++	++			+			++
Cycle 2000		+	+	+	+	+			+			+

Fig. 8. Portrait of blade No. 1 for the model $\varphi_{T12,T01}$
a) with τ =1292, b) with τ =15, c) with τ =45

a)

	Portrait of blade No. 11 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1		+	+	+		+	+	+	+	+		+
Cycle 2		+	+	+	+	+	+	+	+	+		++
Cycle 3		+	+	+		+	+	+	+	+		+
Cycle 4												
Cycle 5						+						+
Cycle 6												
Cycle 7						+						+
Cycle 8						+	+	+	+	+		+
Cycle 9												
Cycle 10		+	+	+		+	+	+	+	+		+
Cycle 20												
Cycle 30						+						+
Cycle 40		+	+	+		+	+	+	+	+		++
Cycle 50						+						+
Cycle 60						+						+
Cycle 70												
Cycle 80						+	+	+	+	+		+
Cycle 90		+	+	+		+	+	+	+	+		+
Cycle 100		+	+	+		+	+	+	+	+		++
Cycle 200												
Cycle 300		+	+	+		+	+	+	+	+		+
Cycle 400		++	++	++		++	++	++	++	++		++
Cycle 500												
Cycle 600		+	+			+	+	+	+	+		+
Cycle 700		+	+	+		+	+	+	+	+		+
Cycle 800		++	++	+	+	++	+	+	+	+		++
Cycle 900												
Cycle 1000		+	+	+		+	+	+	+	+		+
Cycle 2000		+	+	+		++	+	+	+	+		++

b)

	Portrait of blade No. 11 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1		+	+	+		+	+	+	+	+		+
Cycle 2		+	+	+	+	+	+	+	+	+		++
Cycle 3		+	+	+		+	+	+	+	+		+
Cycle 4												
Cycle 5						+						
Cycle 6												
Cycle 7						+	+	+	+			+
Cycle 8						+	+	+	+	+		+
Cycle 9												
Cycle 10		+	+	+		+	+	+	+	+		+
Cycle 20												
Cycle 30						+						+
Cycle 40		+	+	+		+	+	+	+	+		++
Cycle 50						+	+	+	+			+
Cycle 60						+						
Cycle 70												
Cycle 80						+	+	+	+	+		+
Cycle 90		+	+	+		+	+	+	+	+		+
Cycle 100		+	+	+		+	+	+	+	+		++
Cycle 200												
Cycle 300		+	+			+	+	+	+	+		+
Cycle 400		++	++	+	+	++	++	++	++	++		++
Cycle 500												
Cycle 600		+	+			+	+	+	+	+		+
Cycle 700		+	+	+		+	+	+	+	+		+
Cycle 800		++	++	+	+	++	+	+	+	+		++
Cycle 900												
Cycle 1000		+	+			+	+	+	+	+		+
Cycle 2000		+	+	+		++	++	++	+	+		++

c)

	Portrait of blade No. 11 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1		+	+	+		+	+	+	+	+		+
Cycle 2		+	+	+	+	+	++	++	++	+	+	+
Cycle 3		+	+	+		+	+	+	+	+	+	+
Cycle 4												
Cycle 5												
Cycle 6												
Cycle 7							+	+	+	+		
Cycle 8							+	+	+	+		
Cycle 9												
Cycle 10		+	+	+		+	+	+	+	+	+	+
Cycle 20												
Cycle 30							+					
Cycle 40		+	+	+		+	++	++	+	+	+	+
Cycle 50												
Cycle 60												
Cycle 70												
Cycle 80												
Cycle 90		+	+	+		+	++	+	+	+	+	+
Cycle 100		+	+	+		+	++	++	+	+	+	+
Cycle 200												
Cycle 300						+	+	+	+	+		+
Cycle 400		+	+	+	+	++		+	++	++		++
Cycle 500												
Cycle 600						+	+	+	+	+	+	+
Cycle 700		+	+	+		+				+		+
Cycle 800		+	+	+	+	++			+	++		+
Cycle 900												
Cycle 1000						+				+		+
Cycle 2000		+	+	+		+			+	+		+

Fig. 9. Portrait of blade No. 11 for the model $\varphi_{T12,T01}$
a) with τ =1292, b) with τ =15, c) with τ =45

a)

	Portrait of blade No. 3 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1												
Cycle 2												
Cycle 3												
Cycle 4												
Cycle 5		+	+	+								
Cycle 6												
Cycle 7												
Cycle 8												
Cycle 9												
Cycle 10												
Cycle 20												
Cycle 30												
Cycle 40												+
Cycle 50												
Cycle 60												
Cycle 70												
Cycle 80												
Cycle 90												
Cycle 100												+
Cycle 200												
Cycle 300												
Cycle 400											+	+
Cycle 500												
Cycle 600												
Cycle 700												
Cycle 800												
Cycle 900												
Cycle 1000												
Cycle 2000												

b)

	Portrait of blade No. 3 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1												
Cycle 2												
Cycle 3												
Cycle 4												
Cycle 5		+	+	+								
Cycle 6												
Cycle 7												
Cycle 8												
Cycle 9												
Cycle 10												
Cycle 20												
Cycle 30												
Cycle 40												+
Cycle 50												
Cycle 60												
Cycle 70												
Cycle 80												
Cycle 90												
Cycle 100												+
Cycle 200												
Cycle 300												
Cycle 400												
Cycle 500												
Cycle 600												
Cycle 700												
Cycle 800												
Cycle 900												
Cycle 1000												
Cycle 2000												

c)

	Portrait of blade No. 3 for the model $\phi_{T12,T01}$ - minimum velocity											
	L0	L1	L2	L3	L4	L5	M0	M1	M2	M3	M4	M5
Cycle 1												
Cycle 2												
Cycle 3												
Cycle 4												
Cycle 5		+	+	+		+						+
Cycle 6												
Cycle 7												
Cycle 8					+							
Cycle 9												
Cycle 10												
Cycle 20					+							
Cycle 30												
Cycle 40							+	+	+	+		
Cycle 50							+					
Cycle 60												
Cycle 70												
Cycle 80												
Cycle 90												
Cycle 100							+	+				
Cycle 200												
Cycle 300							+	+				
Cycle 400												
Cycle 500												
Cycle 600												
Cycle 700												
Cycle 800												
Cycle 900												
Cycle 1000												
Cycle 2000												

Fig. 10. Portrait of blade No. 3 for the model $\varphi_{T12,T01}$
a) with τ =1292, b) with τ =15, c) with τ =45

On the basis of portrait analysis of blades Nos. 1, 11, 3 shown in Figs. 8, 9, 10, it is found that the choice of the course of a distribution function does not matter to a great extent. However, it is recommended that its course (number of samples) equals the number of signal samples and the initial value is 0.

From portraits it can be also inferred that the blade No. 1 is the most damaged one (has a lot of "++"), considerably worn out is the blade No. 11 (has only few "++") and worn out only to a little extent is the blade No. 3 (almost without "+").

6. CONCLUSION

Method of current assessment of blade technical condition changes basing on diagnostic model $\varphi_{T12,T01}$ is innovative method of blade diagnostics without environment signal measurements.

The equation $\varphi_{T12,T01}$ (11) comprises diagnostic signals y(t) with environment signals x(t) so this is a diagnostic model. Characteristic features of this model are: its determination only on the basis of the measurable signal y(t) in observation periods T_{01} and T_{12} that occur shortly one after another and, what is the most important, taking environment of x(t) into account without the need to measure it as well as sufficient noise suppression of signal y(t) [13, 15, 17, 18].

In order to determine signals $S^{T12}{}_{xy}$, $S^{T01}{}_{xy}$, distribution in the form of the function δ (t, $\hat{t}$) has to be used because it can be easily proven that the quotient of the reciprocal power density function of the signal y and the signal x is insensitive to environment signal x, thus, to a sufficient extent, it eliminates real environment from the model $\varphi_{T12,T01}$ [1,18,23].

Method of blade technical condition monitoring may be based on diagnostic model in form of difference phase shift of output y(t) signal to environment signal x(t) for observation time T_{01} and T_{12}. This method consists in fact that time T_{02} (Fig. 4.) of blade tip movement in sensor area is divided onto two ranges: of blade tip approaching the sensor T_{01} and receding from it T_{12}.

Periods T_{01} and T_{12} of signal y(t) observation are placed so close to each other that the environment x(t) for those periods of signals y(t) observation may be considered identical.

Distinctive feature of model $\varphi_{T12,T01}$ is no necessity of environment signals measurement although these are indirectly taken into account within special research (two observation periods, determination of diagnostic model as a quotient of models binding diagnostic and environment signals to technical condition parameters).

Presented results for the model $\varphi_{T12,T01}$ do not differ when distribution courses are different. Thus, it can be found that there are real opportunities of employing the diagnostic model $\varphi_{T12,T01}$ in diagnosing turbomachine blades during their operation without the need to measure an environment signal.

Blade portraits determined from the model $\varphi_{T12,T01}$ confirm the state of damage presented in the pictures.

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DIAGNOSIS OF GAS TURBINE ENGINES ROTORS SYSTEM IN NONSTATIONARY STATES

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Abstract

Vibration tests of marine gas turbine engines are performed as research of on-line and off-line types. On-line Systems generally monitored one or two vibration symptoms, which asses the limited and/or the critical values of parameters and they, potentially, can warn and/or shutdown engines. Off-line Systems are usually used for vibration analysis during non-steady state of work. The paper presents comparison of different methods of analysis of vibration symptoms measured under run-up and shut-down processes of marine gas turbine engines. Results of tests were recorded on gas turbine engine DR76 type of the COGAG type propulsion system. Main goal of the research was qualified on helpfulness and unambiguous result, from synchronous measurement, order tracking and auto tracking. All vibration symptoms were chosen from the methodology of the diagnosing gas turbine engines operated in the Polish Navy, called Base Diagnosing System. Second purpose of the paper was the estimation of the possibility of usage those analysis methods of gas turbine engines for on-line monitoring systems.

Ke words*: dynamics, gas turbines, rotor vibration, run–up process*

1. Introduction

Operation of marine propulsion systems is a complex issue due to the specific characteristics of the marine environment and the need to maintain a high level of readiness for service and reliability of ships. The use of diagnostic procedures off-line or on-line allows you to use them according to their current condition. This is particularly important in the case of turbine engine, hourly plan and annual plan of technical services is the main usage criteria. This strategy of exploitation makes scheduling maintenance, logistics and security simpler and easier to implement, but also contributes to a significant increase in costs due to the need for replacement of components (often more technically efficient). Furthermore, operating such a exploitation policy makes it impossible for the early detection of other primary causes of faults that occur between appointing terminals.

Diagnostics of gas turbine engines includes a wide range of parameters, controls and maintenance procedures [1]. One of them is the control of unacceptable balance of rotors. Identification of different unbalanced states, determining its value and the accurate placements of corrective masses is commonly known. Such procedures are carried out on Polish ships for over 20 years. Prepared and used test equipment ensures the implementation of diagnostic tests on four types of turbine engines in service. In the case of naval propulsion diagnostic procedures these are limited for several reasons. The most important of these is the need to maintain a constant readiness to start the engine, associated with the tactical requirements. In addition, due to the fact that the engines are foreign construction, there is a lack of information on the structural parameters of the engine, reducing warranty, no spare parts readily available, etc. The use of vibration diagnostics, makes the use of the engine more rational; from a technical point of view, especially towards vitality of service, which in effect will not withdraw, even a technically efficient ship, from service. Measurements and analysis of vibration parameters of marine gas turbine engines can be divided into:

- off-line (measurements performed in free-run mode, periodically);
- on-line (real-time monitoring).

Both methods have their advantages and disadvantages. Off-line Systems are usually offered as a very simple analyzers - data collectors. Measurement path is determined in the collector interface, with preset measuring settings, so that the measurement could be performed by an average technical staff, whose main task is a precise procedure. The analysis of measurement results is carried out of the ship, sending the results to the coast laboratory. Currently, there is not many off-line data collectors, who would engage in that precise diagnostic evaluation. The main advantage of such devices is their price. It should be emphasized that the data collectors are useful mainly to assess the go-state of vibrations of turbine engines.

On-line diagnosis of vibrations provides continuous surveillance of the technical condition of gas turbine engines, including registration, analysis, forecasting and alarming. It allows you to recognize the basic signs of changes in the technical condition with the possibility of analyzing the trend of selected symptoms. On-line vibration systems usually work as part of a complex and symptomatic diagnosis of marine propulsion systems. Proper diagnosis of such structures, for example, turbine engine, depends on various issues, including how the measurement and processing of vibration signals was taken. Important in the further analysis is the fact that internal combustion engines in gas turbine propulsion ships do not run at a constant speed with compressor and turbine rotors.

This is the main reason for synchronizing the processing of selected displacements (of the signals) i.e. the rotational frequency of one or both of the engine rotors [2,3]. This method allows you to identify the most common groups of rotor systems, which allows you to identify their failure. Damages to operating gas turbine engines can be categorized as follows:

- damage or crushing of first-stage compressors' blades or power turbine blades (rare);
- the appearance of unbalance, originating from heating or salinity;
- cracks sealing systems and leakage of lubricating oil to the inside of the drum rotor;
- lack of alignment between the gas-dynamic gas generator and power turbine;
- thermal damage to the combustion chambers – torsion of power turbine rotor;
- damage to the auxiliary engine mechanism.

Some failures can be resolved in the recorded spectra as a change in vibration frequency of rotating engine components, hence the introduction of a synchronous sampling of the transient engine operation, eg in the boot process or in the run.

The occurrence of non-stationary effects, typical for residual unbalance may be due to small, incremental damage whose symptoms may be poorly recognized in the early stages of development. The results of the identification of such phenomena is exemplified in the article comparing the various methods of synchronous signal processing method such as PLD or Order Tracking [7]. The presented method for identification of defects can be introduced into the turbine engine monitoring systems as a tool for early identification of unbalance.

2. The aim and test methods

Monitoring of vibration signals from rotating machinery is a well-known diagnostic procedure, known throughout the world [2,5,7]. Most of rotating machinery and marine gas turbine combustion engines are designed as a supercritical machines, hence, in steady states, are diagnostically limited. Therefore it was decided to analyze the dynamics of rotors of gas turbine engines, using a method of off-line measurements of the unknown states It was expected that the results would yield information on the following areas: unbalance of rotors, lack of concentricity of the rotors, changes in their vibration frequency and changes in the speed of rotor system critical.

Marine gas turbine combustion engines mounted on a DR76 type of propulsion system for ships COGAG class Tarantula Polish Navy were studied using this method. Longitudinal cross section of rotor system is shown in Figure 1.

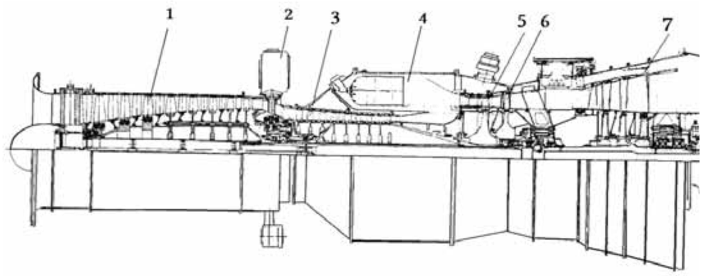

Fig. 1. Longitudinal section of rotor system gas turbine engine DR76 type, where: 1 – low pressure compressor (LPC), 2 – auxiliary drives, 3 – high pressure compressor (HPC), 4 – burning chambers, 5 –high pressure turbine (HPT), 6 – low pressure turbine (LPT), 7 – power turbine (PT)

The study included analysis of the vibration parameters during start-up and run of rotors. Comparison of the results of modeling of dynamic loads using FEM (Final Elements Methods) and measurements of on the real object makes it possible to take correct decisions and give the proper diagnosis

3. Model of the unbalanced rotor

Application of computer simulation to diagnose the condition of turbine engine rotors should be used already during the process of calculation and design, which it is currently implemented. The problem begins when the manufacturer does not provide this kind of know-how in the technical specification for the user. Such a situation arises in the case of exported warships equipped with turbine engines. While placing the engine, rotating parts are assembled with great care. Main objective is to reduce unbalance in rotors. But even the best procedures are not able to prevent factors, such as the inadequacy of heat treatment or the difference of thermal expansion of materials which may cause slight unbalance in rotor, mentioned as residual. Problems in the dynamics of Marine Gas Turbine Engines (MGTE) are associated with the following elements of the engine: rotors, bearings, bearing brackets (bearing struts), engine block, the type of construction, the terms of hydro-meteorological and during sea trials and the aerodynamic parameters inside the engine. Proper and stable work of MGTE engine is mainly connected with these parameters. Loss of energy in rotating machinery is manifested in the form of loss of torque, a decrease in rotor speed, exhaust temperature increase or intensity in vibrations. Vibration energy dissipation is related to: unbalancing of rotors, oversize tolerated shaft misalignment, abrade of blade tips with the inner roller, wear of axis and radial bearings, asymmetry of elasticity and damping asymmetry of the rotor and the gas-dynamic processes anomaly. Emission of vibration yields a lot of information, including the ability to diagnose the technical condition of rotors. Vibration measurement, identification, classification, mathematical analysis, including the use of trend function, give information on the actual technical state and allow the prediction of the wear process in the future.

In the identification an important factor is to compare the results of modeling with the results of the measurements. Each rigid body has six degrees of freedom, whereas the deformable objects have an unlimited number of degrees of freedom. Rotating machinery such as MGTE have a number of degrees of freedom equal to the sum of all degrees of free parts of the engine, minus the number of rigid nodes connecting these elements. Each part of the engine can be described by physical characteristics such as stiffness and damping, obtained from vibration measurements the actual object or model or the modeling of the geometry and properties of materials (the use of rigidly connected structures). The use of a certain type of rigid object model allows the use of the motion ordinary differential equations. Deformable objects require the use of partial differential equations. This second assumption is much more complicated, but can help to achieve to the actual object, especially when it's in a wide range of engine speeds. This was the reason for the choice of the second type of model turbine engine. Diagram of diagnosis using the MGTE model shown in Figure 2.

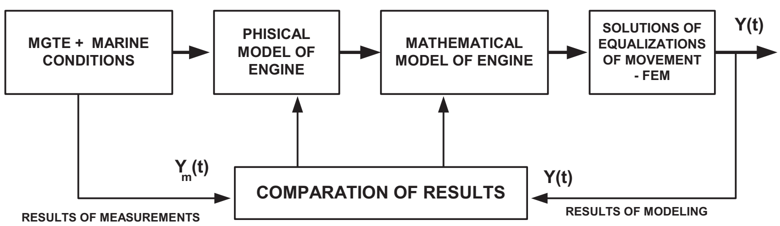

Fig. 2. Scheme of diagnostics model MGTE

Residual unbalance may appear in all sections of the rotor, however, two vectors of unbalance, at both ends of the shaft, may represent the replacement model. These vectors vary in values and phase shifts. Such an FE model allows for dynamic response to unbalance which in effect allows you to compare modeling results with the reports of vibration measurement. The most sensitive point in the unbalance of GT rotor, with respect to vibrations, is the measuring point on the front of the generator exhaust bracket bearing the vertical direction. This is the effect of the minimum thermal expansion of the rigid support used for measurement of radial vibrations at this point. The model is linear so it is clear that response is directly proportional to the value of unbalance The rotor is loaded dynamically and statically from various sources [4].

Identification of the sources and their calculations of the loads were a major problem during the modeling and evaluation of the actual object's vibration. Damage in the objects such as blades, have an impact on changes in the moments of inertia of rotating parts. This results in a shift of the main axis of inertia, which is not parallel to the axis of rotation. It is the main source of unbalance in the form of vibrations of rotor. Implementation of the mathematical model is difficult, mainly due to the problems of determining the stiffness and damping of supports and bearings at different temperatures - Figure 3. Shape of the axis deflection is defined as discrete sets:

- Set of static deflections – u_s;
- Set of dynamic deflections – u_d.

Both sets depend on actual technical state of rotor and geometry, which can change through cracks and wanes of engine parts.

$$\mathbf{u}(\omega t) = \mathbf{u}_s + \mathbf{u}_d(\omega t) \tag{1}$$

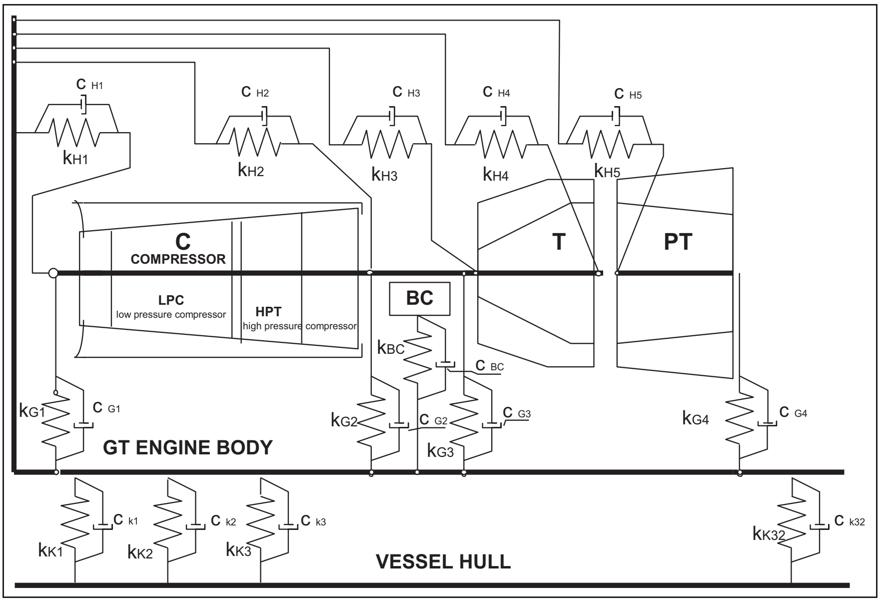

Fig. 3. Axi – symmetric lumped mass inertia model of the MGTE, where: LPC – low pressure compressor, HPT – high pressure compressor, T – turbines (low & high pressure), PT – power turbine, BC – burning chambers, k – stiffness, c – dumping

This equation is a discrete set of points of axis movement of the rotor. Taking into account the damping and stiffness of the support bearings, we can demand that they are functions of temporary positions, namely:

$$k_{ik} = f(u) \qquad c_{ik} = f(u) \tag{2}$$

For the simplification it is assumed that, for a constant speed, these values are constant. Using FEM modeling can provide a three-dimensional discrete model. Rotors MGTE, in the circular symmetry, have been described by one-dimensional, two-beam bar having a symmetrical six degrees of freedom. All parts of the model have geometric and physical properties of the elements. Discrete model of traffic parameters have been obtained by solving the equation:

$$\mathbf{Ku} + \mathbf{C\dot{u}} + \mathbf{M\ddot{u}} = \mathbf{F}(t) \tag{3}$$

where: **K** – matrix of structure's stiffness
C – matrix of structure's damping
M – matrix of structure's inertia
F – vector of forces
$\mathbf{u}, \mathbf{\dot{u}}, \mathbf{\ddot{u}}$ – displacement and their derivatives (velocity and acceleration)

This can be solved as a linear problem, but in MGTE rotor must allow for changes in stiffness and damping, which are functions of motion parameters. In this case equation (3) should be expressed as:

$$\mathbf{K}(\mathbf{u},\dot{\mathbf{u}})\mathbf{u}+\mathbf{C}(\mathbf{u},\dot{\mathbf{u}})\dot{\mathbf{u}}+\mathbf{M}\ddot{\mathbf{u}}=\mathbf{F}(t) \tag{4}$$

Equation (4) indicates that the rotor motion should be described as a nonlinear dynamic problem, and therefore should expect more than one harmonic in both measured and modeled spectrum. [8]

4. Non – steady states vibration signals analysis

To obtain the measurements of the real object Bruel & Kjaer 3560B analyzer was used. Namely, it was used during the collection and processing of measurement data using the PULSE(v.12). Two transducers (accelerometers ICP) have been fitted to the steel girders, situated on the flanges, on the front and on central pillar of the LPC. The fixing cantilevers are characterized by vibration resonance frequency value differing from harmonic frequencies due to rotation speed of the given rotors. Measurements were made perpendicular to the axis of rotation of the rotor. Such a choice was made on the basis of theoretical analysis of unbalance and as a result of analysis of the results of preliminary research on the subject.

Common assessment of the unbalance of rotors was developed through the concept of dimensionless coefficients of diagnosis. Using theoretical analysis of dynamic interactions, as well as using the results of initial diagnostic tests, the following symptoms were selected as the most sensitive to changes in balancing rotors [2]:

- First harmonic of amplitude of the corresponding velocity of the rotor,
- Second harmonic of amplitude of the corresponding velocity of the rotor,
- S 1 - the ratio of the average amplitude of vibration corresponding rotor speed (and harmonic) and the second harmonic of the corresponding rotor,
- S 2 ratio of the average amplitude of vibration corresponding rotor speed (and harmonics) corresponding and the third harmonic of the rotor.

These symptoms can confirm the theoretical assumption of nonlinear rotor dynamics.

5. Vibration analysis of the run-up process

The first test was to analyze the process of starting the engine. The characteristic changes in LPC rotor speed is shown in Figure 4.

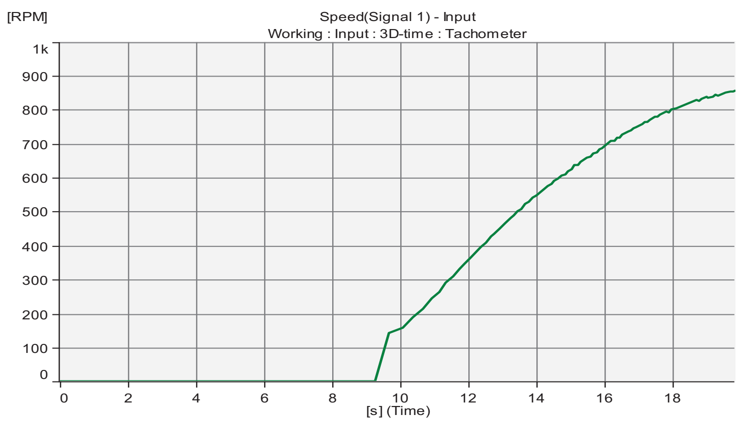

Fig. 4: Rotors LPC rotational speed characteristics during run-up process

Synchronous signal measured by a tachometer connected with the auxiliary drive gear box where the transmission ratio averaged on i=0,125, so the LPC rotor was 8 times greater (in speed) than that shown in Figure 4 The main objective of the analysis of synchronous oscillations in the boot process was to determine the dynamics of the disorder. The impact of "other" signals is shown in Figure 5.

The boot process started at the point t = 7 seconds (see Figure 5), so all recorded vibration signals recorded from the start point contained the signals coming from other sources, i.e. non-rotating motor or frequency of its vibrations or a combination thereof. This allows to identify the main "other" signals, such as: f_1 = 305 Hz, f_2 = 600 Hz, f_3 = 1.6 kHz, and f_4 = 2 kHz. which are associated with sources outside the engine.

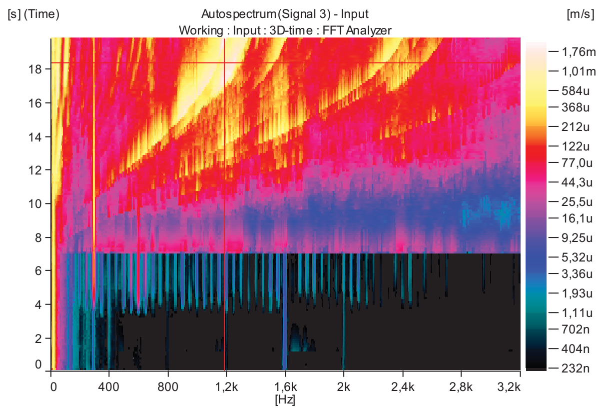

Fig. 5. Synchronous spectra of the velocity of vibration during run-up process with using the band – pass filter of 0,1Hz – 3,2 kHz range

The highest signal during the boot process is the rotor speed and harmonic vibrations, but in Figure 5 it is not clearly visible due to the lack of a synchronous signal tracking.

6. Vibration analysis of the shut-down process

Next test was associated with the analysis of vibration parameters and related to the process runs the motor rotor. Figure 6 shows autospectrum of the velocity measured over the middle LPC bearing using the order tracking procedure. Changes of parameters are presented in the domain of time function, in contrast to the boot process ,where the dominant energy range of vibration signal was 1 / 2 harmonic - seen as a 4th order. The pressure drop of the lubricating oil in the bearing caused an increase in values ranging from displacement and slope between the HPC and LPC rotor (rotating shafts each other, while the shaft rotates within the LPC HPC shaft - see Figure 1) and the typical dominance of the subharmonics .

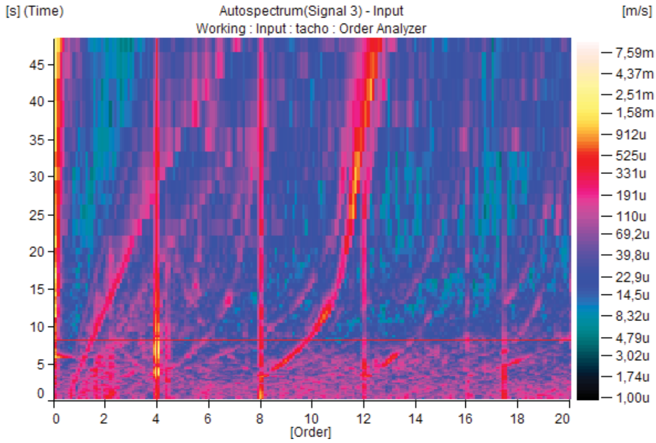

Fig. 6. Autospectrum of velocity of vibration in the shut-down process with the use of order tracking procedure, in the domain of time function

The increase in stiffness of the bearing system confirms the existence of the harmonic "right-hand branches" at the point where t (time) is equal to 4 seconds for the following rows: 4, 8 and 12, which is associated with a pressure drop of lubricating oil in the bearings.

Analysis of the dynamics of the turbine engine rotor in transient states of a system PULSE should be applied in both processes, ie start-up and run. The start-up process helpsto recognize the "other" signals, but the definition of dynamic functions is very difficult due to the significant acceleration of the rotors. Identifying characteristics of rotor system dynamics is much more recognizable in the process runs through the analysis of orders - Figure 7 and 8.

Analysis of the first harmonic (8th order) allows to observe changes in dynamics as trends. Application of the rotational speed function as a field of analysis is the most important factor in the study of the use of the Order Tracking procedure. This allows you to detect changes in the natural frequency, ignoring interference from the signals originating from the thermodynamics processes of turbine engines.

Subharmonics signal analysis is very useful in the diagnosis of rotating machinery. Autospectrum of ½ subharmonic's velocity range (considered in the LPC rotor) indicates the individual characteristics of particular rotors. The nature of changes in order values in the rotor speed can be thought of as an individual fingerprint of each rotor.

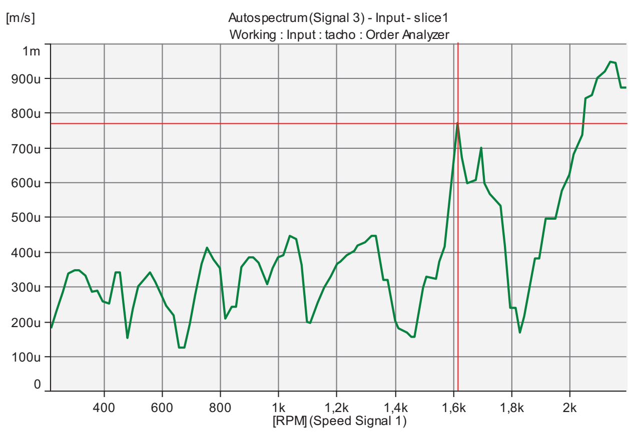

Figure 7. Autospectrum of 8 order (I harmonic) of velocity of vibration in the shut-down process of LPC rotor stoppage

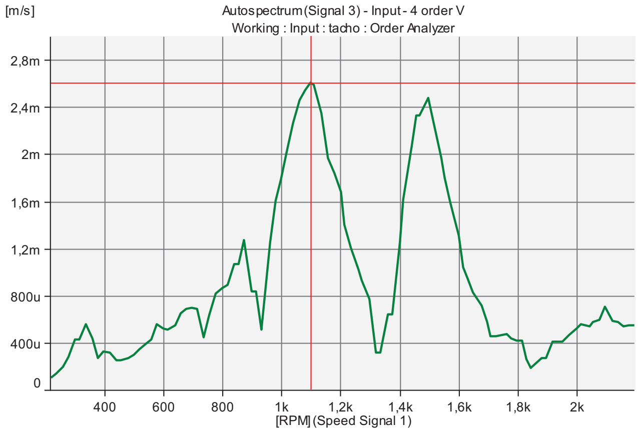

Fig. 8. Autospectrum of 4 order (subharmonic) of velocity of vibration in the shut-down process of LPC rotor stoppage

All changes to the technical condition of rotor system, such as changes in stiffness and damping parameters of alignment, or unbalance result in changes in characteristics of subharmonics - Figures 7 and 8.

Conclusions

All statistical analysis performed on the available population of engines clearly show that the selected parameters analyzed in the non-stationary processes are the basis for predicting changes in the technical condition of rotor system. Implementation of this research turns out to be a credible verification of the technology. Conclusions presented below have been incorporated into operational diagnostics of marine gas turbine engines:

- synchronous measurement of vibration signals during the boot and run processes enables us to recognize symptoms of damage, including the formation of resonance and changes in natural frequencies and unbalanced rotors
- symptoms of S1 and S2 do not have sufficient sensitivity for use in transient states due to the instability of the processes and the need for averaging the results,
- application of auto tracking and monitoring the turbine engine rotor systems can identify a wide range of typical damages, confirmed by the vibro-acoustic diagnostics.

Application of the proposed methods of analysis allows for the rational management of engine life time even in the developed processes of consumption. The analysis of test results obtained gives the following conclusions:

- the approach to the assess the technical condition of gas turbine engines rotor system allows to quickly detect changes in the permitted unbalance and the maintained database enables easier identification of the studied group of engines
- studies on trends in chosen parameters make it possible to reliably detect changes in the value of sensitive operational parameters during the operation of the engine and to evaluate its capabilities.

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COMPUTER-AIDED SYSTEM TO ALLOTMENT OF THE OPERATING TASKS IN THE SHIP ENGINE ROOM

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Abstract
The frequent causes of ships' detentions by port authorities are abnormalities of ship power plant functioning. Each extended ship lay time in port results in a waste of ship operating time thus costs rise to ship owners. This is connected with improper ship power plant management. In order to avoid this, a ship engineer should have at his disposal computer aided system supporting him in the managing of the ship power plant. The prototype of a computer-aided system of operational and maintenance task assignation in ship engine room has been presented in this paper. This system contains three modules that respond of decision-making process for three stages: collecting essential information about operating tasks, selecting of operational and maintenance tasks, generating of the optimal schedule for the set of the determined conditions. For allotment problem was formulated, in the most substantial operating states of a ship like lay time in harbour and sea voyage, the knapsack algorithm' was applied. The mathematical model uses two-criterions optimization for operational and maintenance task assignation in the ship engine room.

Keywords: *Ship engine room, tasks scheduling, AHP method*

1. Introduction

According to many experts to reach correct management of ship power plant involves great difficulties to decision-making persons, i.e. ship chief engineers. This is caused i.e. by:

- increasing number of automated ship systems,
- multiple number of operational processes executed in parallel,
- lack of appropriate information making it possible to quickly master systems and task planning,
- frequent changes of staff members,
- increasing number of requirements for safety of persons, ship and environment.

Moreover changing international maritime law imposes many additional tasks dealing not only with new procedures connected with safety at sea but also with their detail documentation. Such state leads to a situation in which decision-making is more and more difficult and knowledge and experience of ship engineers may appear insufficient. In such conditions making a decision dealing with power plant management may be incorrect or irrational and in consequence causing various losses, e.g. loss of ship service time leading this way to increasing overall cost of ship operation. In order to eliminate such situations ship engineers should have at his disposal a software which could be a „tool" aiding him in organizing ship power plant management process. Such system would collect information concerning realization of all operations in power plant or make use of data bases of already functioning information systems, analyse any limitations associated with their realization and finally advising ship engineer on which tasks and in which sequence they have to be realized.

In ship power plant a team often consisted of several persons performs operations resulting from realization of many tasks of different time horizons, realized in parallel. This requires, from chief engineer, to make rational decisions concerning a.o. determination of a kind, range, sequence and executors of operations. To make such decisions it is necessary to collect and process suitable information. Among other, the following can serve as their sources:

- technical and operational documentation of machines and installations, requirements associated with safety at sea and marine environment protection (conventions, codes, rules of classification societies, rules of maritime administrations, ship owner's regulations etc.),
- data bases of information systems used in ship power plant,
- assessment of technical state of ship power plant machines and systems,
- assessment of state of provisions (fuels, lubricants, spare parts etc.),
- occurrence of a destructive event, e. g. machine failure,
- assessment of feasibility of appropriate actions, e.g. expected time of port staying, deadline of subsequent shipyard's repair etc.,
- assessment of accessibility of an external service in a given shipping region,
- assessment of capability of crew to realize planned operations,
- assessment of crew experience associated with carrying out given kinds of operations [2].

2. Solution of the chief engineer decision-making problem

The main problem to be solved by ship chief engineer within the scope of ship power plant management can be formulated as follows: „*Knowing a set of tasks to be realized as well as taking into account available means (technical, personnel and time resources),operational requirements concerning ship, as well as limitations of different kind, one should make choice of appropriate operations and integrate them into one ordered set of actions*". In other words the thing is that a decision should be taken as to such above mentioned operations whose realization would be most effective from the point of view of ship service.

The decision problem of ship power plant technical management is defined as the following triple:

- the set of decision variables, (i.e. the set of all operations to be executed),
- the set of operators to which appropriate operations should be assigned,
- as well as that of the relations r understood as the relationships between elements of the sets and also containing some features of the elements.

In the process of decision making by ship engineer dealing with assigning the operational tasks to engine room staff the following three main phases should be distinguished:

- collecting and processing all available and necessary data (those earlier mentioned and those presented in [1]),
- selecting the tasks whose realization is constrained by all possible operational limitations as well as ambient conditions in which a given decision is made [3],
- assigning the earlier selected tasks to power plant crew members, in compliance with their competences so as to obtain the best schedule from the operational point of view [2].

3. The data collection module

To generate the appropriate operating tasks schedule it is necessary to use essential information. Such information are collected on a ship in a different form: on the new ships in the computer databases, on the older ships mostly in the paper documentation form.

The data collection module of prototype computer-aided system to support operating tasks scheduling in engine room as an interface is presented on Figure 1. It contains a list of the example-tasks for the selected unit from the engine room systems structure and attributed to each the number of parameters. These parameters are: the area of operation (operating, maintenance, safety, provision), type of the task (planed, emergency, etc.), execution time, frequency of task repetition, the ship's operating stage, the engine room's operating stage in which the task performance can be achieved, the operator executed this task (according to the duties), etc. Data collected by this module are stored in external database created in MS Access. This allows to other systems or programs used the data by SQL. This also permitted to use the databases of other computer systems used in the engine room by the presented system. On the Fig.1 is presented the graphical interface of the computer system consists of two main parts. The first of them, shown on the right-hand side, is characteristic for systems applied in engine room and it demonstrates of power plant structure.

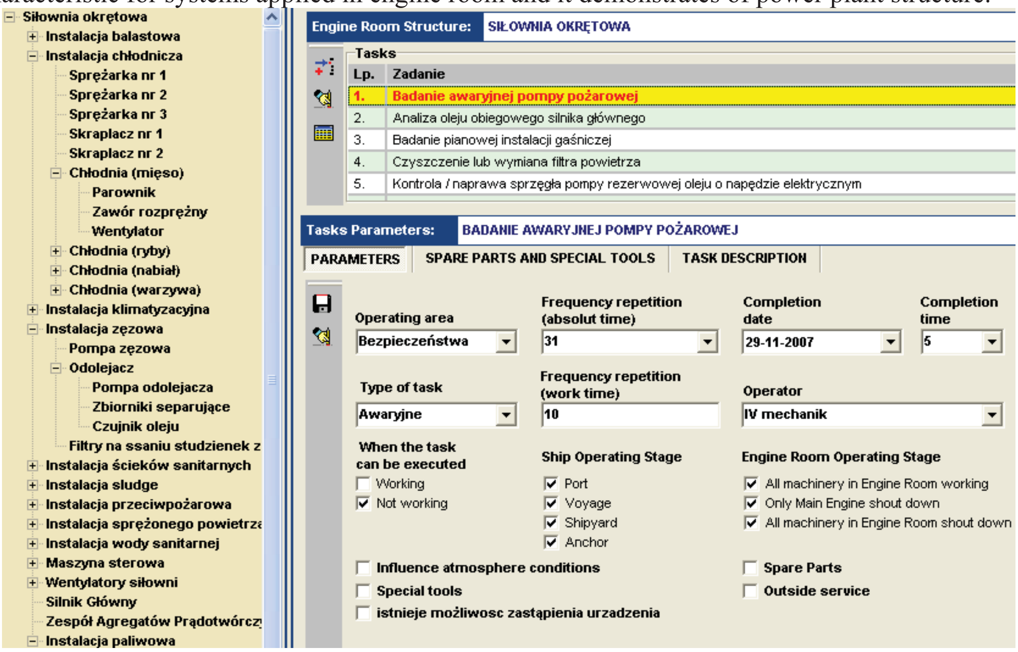

Fig. 1. An example screen of the graphical interface of the computer-aided prototype system for allotment of the operation tasks in ship power plant. Ship power plant structure – in polish (on left hand side), operating task parameters defining (on right hand side)

4. The selection and hierarchy tasks module

Next module of the prototype computer-aided system to support of operating tasks scheduling is designed to stages:

- selection and elimination of the tasks which, cannot be executed in given conditions, due to various circumstances,
- hierarchization of importance the remaining tasks in operational point of view.

In the first stage the chief engineer have to determines the conditions in which the schedule of operating tasks will be done:

- operating stage of ship and operating stage of engine room in which the task performance can be achieved,
- operators, which has at disposal,

- weather conditions,
- time that is available to tasks execution, etc.

After approved these conditions the decision-maker can also limits additionally the tasks because of another reason not involve in computer system i.e. the availability of special tools, etc. Such list of selected operating tasks are automatically give in to hierarchy process because of the importance (coefficient wg_i in equation1) of each tasks in the engine room operation process, which consider for example:

- what kind of task is: planned, forced by breakdown or administration, etc.
- planned time of the maintenance task execution,
- number of the same devices and their technical condition, etc.

So hierarchically list of operating tasks is essential to generate optimal schedule.

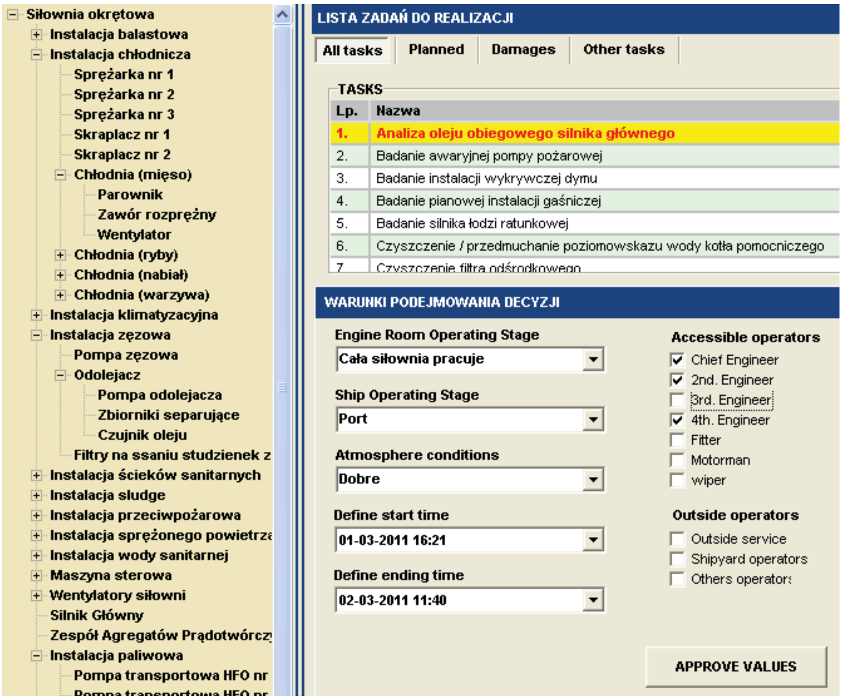

Fig. 2. An example screen of the graphical interface for determine the conditions for which the schedule is generated - selecting and hierarchization module.

5. Module of generating the optimal solution of the schedule

The operating tasks scheduling problem in the engine room is an optimization problem, which like many issues of this type has been implemented as a transportation problem (otherwise also known as: knapsack problem). That is a special case of binary (0-1) issues of combinatorial optimization [5]. In this problem has been defined two criteria for their selection and allocation:

- the most important tasks should be realized, i.e. the schedule should consist of the lists of the tasks assigned to every operator, and having the importance index wg_i of the possible largest value,
- the time of realization of the tasks should be close to the available time for their realization, in other words to obtain the best use of the available time.

In accordance with the way of formulation of optimization function, described in [4], in the presented problem such function can be assumed to be a combination of assessment criteria of scalar form, generally defined as a weighed sum of:

- task importance indices,
- time intervals for their realization.

In the case of the so formulated objective function one has to do with two-criterion optimization. By introducing to it the coefficients ρ1, ρ2 called the criterion weighing factors, a choice on which criterion would be more important, becomes possible. Such choice is made by the decision maker, i.e. chief engineer, depending on needs appearing in a given instant. The coefficients ρ1, ρ2 can take values from the interval <0, 1>, and their sum should be always equal to 1.

Therefore the best schedule, out of all allowable solutions, is that for which the sum of the weighed sums of two presented criteria, for all considered operators, reaches a maximum. The general form of the objective function is as follows:

$$Fj = \max \sum_{j=1}^{o} \left(sk \cdot \rho_1 \cdot \sum_{i=1}^{n} wg_i x_{ij} + \rho_2 \frac{\sum_{i=1}^{n} t_i x_{ij}}{T_S} \right) \tag{1}$$

where:

i = 1,2,3,....,n – number of tasks,

j =1,2,3,....,o - number of operators,

ρ_1, ρ_2 –weighing factor of the criterions,

x_{ij} – factor which determines the assignment of *i-th* task to *j-th* operator,

sk – scale (a coefficient so selected as to obtain balanced values of sum components),

wg_i – task importance index,

T_S – the time available for realization of tasks staying in port, sea voyage time).

In the process of generating the best solutions of the problem takes into account four main constraints:

- total time of the tasks assigned to each of the operators cannot be greater than the available time T_S intended to proceed,
- the task can be assigned only once in the schedule,
- each task is performed by only one of the operators
- way of assigning tasks to individual operators determined,

There are the following assumptions adopted too:

- each operator can perform only one task in a given interval of time,
- each task has a number of attributes stored in a database, or defined in earlier stages (the elimination of impossible tasks to performing in the given conditions, prioritizing tasks) [1], [2],
- allocation of tasks to individual operators to be implemented in accordance with the hierarchy of professional.

The standard knapsack problem consists in filling the “knapsack” of a given limited volume by using elements (blocks) of various dimensions and values in such a way as to fill the knapsack so as to make its value the greatest. In the same way can be formulated the problem faced by ship chief engineer in some specific situations (e.g. short stay in port or short sea voyage), who must assign operational tasks to power plant crew members so as to make the best use of available time and simultaneously to realize the most important tasks out of the set of the tasks whose realization cannot be performed during the available time interval. For solving the problem of optimization of the schedule of operational tasks in ship power plant the last of the presented method, i.e. the method of indirect searching, called also the searching with reversals. The method was selected due to its simplicity, as it contains basic steps of almost all searching methods and simultaneously is one of the quickest among them [4]. In order to check the above presented mathematical model as well as the method of solving of the decision problem usually faced by chief engineers, a prototype computer software for aiding in planning the operational tasks in ship power plant in some definite conditions, was elaborated. In Fig.3 shows the interface of system, which demonstrates solution of the chief engineer scheduling problem for the sample data (sample tasks). The top bar shows two important parameters of the problem: ‘Maximum time’, i.e. the time interval for which the schedule is considered (e.g. port staying time, sea voyage time etc.), ‘Criterion weighing factor’ (the parameter ρ in equation.1) i.e. that determining which choice is of a greater importance: that of the most important tasks or that of the most effective use of the available time. There are a few additional option to choose for the user to presenting the solutions of optimization process. On the remaining parts of window shows the results of the optimal schedule generating process. On the left side, in the area ‘Schedule variants’, is shown the schedules in the text form, on the right graphic form called. Gantt chart, where the height of each row represented each operator skills and the assigned operating tasks are represented by rectangle (different colour and size).

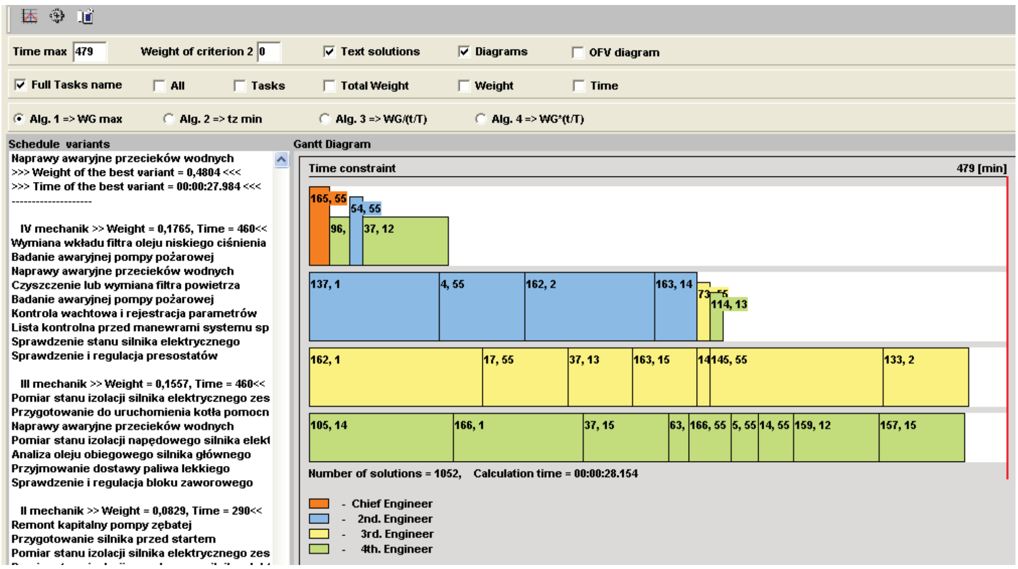

Fig. 3. An example screen of the graphical interface of the operation tasks allotment system Gantt chart – schedule generating module.

6. Conclusion

In this paper has been presented an approach to solving the decision problem associated with scheduling of operational tasks realized by staff in ship power plant, with making into account different conditions. The prototype system consists of three modules corresponding to the stages of decision-making process of operating tasks allotment in engine room and there could be draw the following conclusions:

- the problem formulation as a 'many-knapsack' problem seems to be a natural responding to task allocation process in the ship power plant,
- objective function which takes into account the crucial elements considered by chief engineer in scheduling the operational tasks in ship power plant: i.e. importance of a given task, com-

petences of each of the operators, time available for realization of necessary tasks, has been elaborated.

- advantages of the solution searching method, especially: simplicity, the basic steps of nearly all review methods, convinced to use it for scheduling optimization in the ship engine room.

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MIXTURE OF DISTRIBUTIONS AS A LIFETIME DISTRIBUTION OF A BUS ENGINE

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Abstract

We show that a upside-down bathtub failure rate function can be obtained from a mixture of two increasing failure rate function (IFR) models. Specifically, we study the failure rate of the mixture an exponential distribution, and an IFR distribution with strictly increasing failure rate function. Examples of several other upside-down bathtub shaped failure rate functions are also presented. The method are illustrated by a numerical example of the time between the failures for the bus engines.

Keywords: *Bathtub curve, upside-down bathtub curve, failure rate function, mixture of distributions, reliability function*

1. Introduction

The distributions with non-monotonic failure rate functions are considered frequently in the reliability theory and practice. The distributions with a bathtub shape failure functions (BFR) belong to such a class of distributions. The models with BFR are very useful in the reliability theory and practice. We give the definition of a bathtub shape failure rate function below. It is useful, throughout his paper by increasing or decreasing, understood respectively as non – decreasing or non – increasing.

Definition 1: A lifetime T, with failure rate function r(t) is said to have a bathtub shaped failure rate if there exists t^* such that $0 < t^* < \infty$ and r(t) is decreasing for $0 \leq t \leq t^*$ and r(t) is increasing for $t > t^*$.

A brief discussion and summary for such distributions are given in [1] and [12]. However, there are known many examples of applications of distributions with upside-down bathtub shaped (unimodal) failure rate functions (UBFR). In particular cases, the unimodal failure rate function is used in [9] for data of motor bus failures, in [1] and [4] for optimal burn decisions, and in [5] and [7] for ageing property in reliability.

One of the ways of generating distributions with non-monotone failure rate functions is mixing the standard distributions. It is well known result that the mixture of distributions with a decreasing failure rate functions (DFR) has a decreasing failure rate function (see Prochan [11]). Klutke et al. [8] have been studied the mixture of two Weibull distributions and they suggested that this mixture can be the distribution with unimodal failure rate function. However, in [13], it is stated that the considered mixture failure rate function has a decreasing initial period. The mixture of the two Weibull distributions has also been studied in [14]. For the same values of scale parameter all possible types of shape failure rate function are found. However, for the different scale parameter the numerical computing is performed. Block et al. [3] have been studied the mixture of two distributions with increasing linear failure rate functions.

The paper is organized as follows. In Section 2, the model of the mixture of two distributions is introduced and discussed, while, in Section 3, the particular cases are considered. In last section the numerical examples with technical data are presented.

2. The model of mixture distributions

We consider a mixture of two lifetime T_1, T_2 with densities $f_1(t)$, $f_2(t)$, with corresponding reliability functions $R_1(t)$, $R_2(t)$, failure rate function $r_1(t)$, $r_2(t)$ and weights p and $q = 1 - p$, where $0 < p < 1$. The mixed density is then written as

$$f(t) = p\, f_1(t) + (1 - p)\, f_2(t)$$

and mixed reliability functions is

$$R(t) = p\, R_1(t) + (1 - p)\, R_2(t)$$

The failure rate function of the mixture can be written as the mixture [2]

$$r(t) = \omega(t)\, r_1(t) + [1 - \omega(t)]\, r_2(t)$$

where $\omega(t) = pR_1(t) / R(t)$. Moreover, from [2], we have under some mild conditions, that

$$\lim_{t\to\infty} r(t) = \lim_{t\to\infty} \min\{r_1(t), r_2(t)\}$$

In the following propositions, we give some properties for the mixture failure rate function.

Proposition 1: For the first derivative of $\omega(t)$, we have

$$\omega'(t) = \omega(t)\,(1 - \omega(t))\,(r_2(t) - r_1(t))$$

Proposition 2: For the first derivative of $r(t)$, we obtain

$$r'(t) = (1 - \omega(t))\,((-\omega(t)\,(r_2(t) - r_1(t))^2 + r'_2(t)) + \omega(t)\, r'_1(t)$$

Proposition 3: If $R_1(t) = \exp(-\lambda_1 t)$, then

$$r'(t) = (1 - \omega(t))\,((-\omega(t)(r_2(t) - \lambda_1)^2 + r'_2(t))$$

Proposition 4: If $R_1(t) = \exp(-\lambda_1 t)$, then $r'(0) \geq 0$ if and only if

$$r'_2(0) \geq p\,(r_2(0) - \lambda_1)^2$$

We suppose that $r_2(t) = \gamma t + \alpha t^{\alpha-1} / \beta^{\alpha}$, where $\alpha \geq 1$. The reliability function $R_2(t)$ is a particular case of the reliability function given by Gurwich [6] (see also [10]).Without loss generality, we assume that $\beta = 1$. Hence $r_2(0) = 0$ for $\alpha > 1$.Consequently, the reliability function of corresponding toT_2 is

$$R_2(t) = \exp(-½\,\gamma\, t^2 - t^\alpha) \quad \text{for } t \geq 0.$$

Let $h_1(t) = \omega(t)\,(r_2(t) - \lambda_1)^2$, $h_2(t) = r'_2(t)$. Since $\omega(t) \geq 0$ for $t \geq 0$ and $r_2(t)$ is increasing from 0 to ∞, and $\omega(0) = p$, $\omega(\infty) = 1$, we conclude, that the equation $h_1(t) = 0$ has only one solution t_1. We can also examine the ratio of the function $h_1(t)$ and $h_2(t)$, i.e.

$$\lim_{t\to\infty} \frac{h_1(t)}{h_2(t)} = \infty$$

Since there are t' such that, for all $t > t'$, we have $h_1(t) > h_2(t)$, where as, we have $h_1(t_1) = 0 < h_2(t_1)$. Hence the equation $h(t) = h_1(t) - h_2(t) = 0$ has at least one solution.

3. The particular case of mixture

We shall give the conditions under which the failure rate of the mixture of an exponential distribution and the distribution with failure rate $r_2(t)$ has an UBFR. In this section, we consider three particular cases of a failure rate $r_2(t)$.

Proposition 5: If $2 \leq \alpha \leq 6$ and $p\,\lambda_1^2 \leq \gamma$ then $r(t) \in$ UBFR.

Proof: It is know that the equation $h(t) = 0$ has at least one a solution for $t > t_1$. We consider the ratio

$$u(t) = \frac{(r_2(t) - \lambda_1)^2}{r'_2(t)}$$

It is easy that $u(t_1) = 0$ and $\lim_{t\to\infty} u(t) = \infty$. For the first derivative, we have

$$u'(t) = \frac{r_2(t) - \lambda_1}{[r'_2(t)]^2}\{2[r'_2(t)]^2 - r''_2(t)(r_2(t) - \lambda_1)\}$$

Let $u_1(t) = 2\,[r'_2(t)]^2 - r''_2(t)\,(r_2(t) - \lambda_1)$ and

$$u_1(t) = 2\gamma^2 + \gamma\alpha(\alpha - 1)t^{\alpha-2}(6 - \alpha) + \alpha^3(\alpha - 1)t^{2\alpha - 4} + \lambda_1\alpha(\alpha - 1)\,(\alpha - 2)t^{\alpha-3}$$

If $2 \leq \alpha \leq 6$ then $u_1(t) > 0$ and $u'(t) > 0$ for $t \geq t_1$.

By Proposition 1 $\omega(t)$ is increasing for $t \geq t_1$ and $\omega(t)\,u(t)$ is increasing for $t \geq t_1$. Hence the equation $h(t) = 0$ has only one solution and $r(t) \in$ UBFR.

4. The numerical examples

In this section, we consider four examples to illustrate the theoretical research given in the previous sections.

Example 1: We consider the exponential distribution with failure rate function $r_1(t) = \lambda_1 = 1$ and Gurvich distribution given in the section II with parameters $\beta = 1$, $\gamma = 1$, $\alpha \in \{2.5, 3, 4, 5, 6\}$and mixing proportion $p = 0.8$. Thus r(t) have an upside – down bathtub shaped. Figure 1 shows the five plots of r(t).

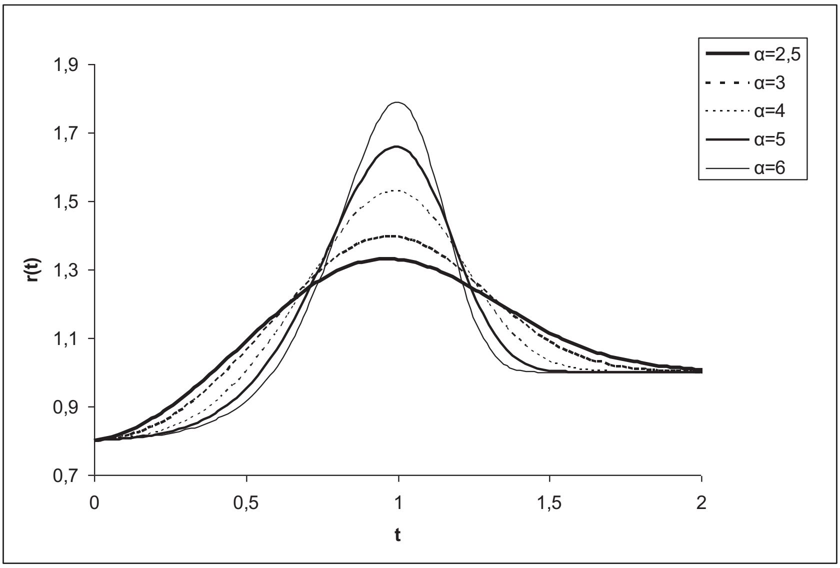

Fig. 1. Mixture failure rate of exponential and Gurwich distribution for $\alpha \in \{2.5, 3, 4, 5, 6\}$ with UBFR shape

Example 2: We consider two failure rate functions namely $r_1(t) = \lambda = 1$ and a failure rate of Gurwich distribution with $\alpha = 2$, $\beta = 1$, $\gamma \in \{2, 3, 4, 5, 6\}$. The mixing proportion $p = 0.8$. Figure 2 shows the plots of $r(t)$ for different values of parameter γ.

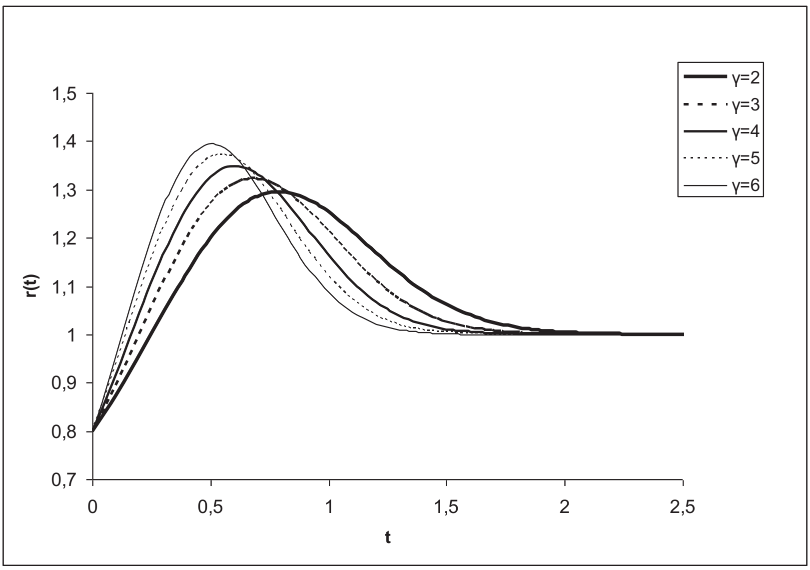

Fig. 2. Mixture failure rate of exponential and IFR distribution for $\gamma \in \{2, 3, 4, 5, 6\}$, with UBFR shape

Example 3: In this example, we consider the mixture of exponential distribution with $\lambda = 2$ and Gurwich distribution with $\alpha = 3$, $\beta = 1$ and $p \in \{0.4, 0.3, 0.2, 0.15, 0.1\}$. Figure 3 contains five plots of failure rate functions for different values of p.

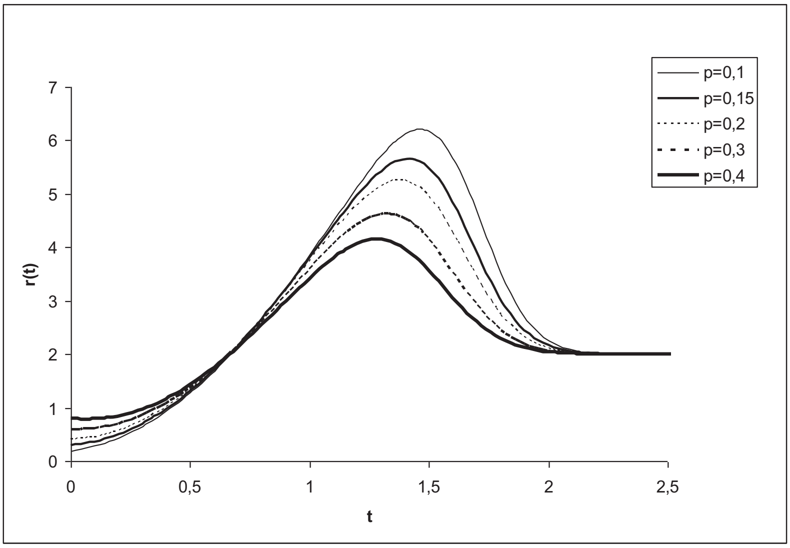

Fig. 3. Mixture of exponential distribution and Gurwich distribution for $p \in \{0.4, 0.3, 0.2, 0.15, 0.1\}$

Example 4: In this example, we consider a real lifetime data. The object of the investigation is a real municipal bus transport system within a large agglomeration. The analyzed system operates and maintains 210 municipal buses of various marks and types. For the investigation purpose, 35 buses of the same make were selected. The data set contains $n = 1081$ times between successive failures of the engine of the bus. We estimate the parameters p, α, γ, β, λ of the model with the reliability function

$$R(t) = p \exp(-\lambda t) + (1 - p) \exp(-0.5\,\gamma t^2 - (t / \beta)^{\alpha})$$

By maximizing the logarithm of likelihood function for grouped data, we calculate $p = 0.76$, $\alpha=2.71$, $\gamma = 15.56$, $\beta = 99.03$, $\lambda=0.082$. For these values of parameters, we prove Pearson's test of fit and compute associated p–value is 0.46. The reliability function R(t) sufficiently precisely describes the empirical reliability function. By Proposition 5, we conclude that the failure rate is UBFR.

5. Conclusions

Sometimes, we have upside-down bathtub estimated failure rates from model which do not have theoretical UBFR. In this paper we ha ve presented flexible and practical model for UBFR. The purpose of this paper is to present a new UBFR as a mixture of two distributions for the first time. The model of UBFR presented in this paper is fully adaptive to the available failure data and this distributions gives reliability engineers and biostatisticians another option for modeling the lifetime. The numerical examples for life time of an engine system of a bus shows that the mixture can be useful to practical applications.

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ENDOSCOPIC IMAGE PROCESSING AND ANALYSIS OF PISTONS' SERVICE FAILURES OF MARINE DIESEL ENGINES

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Abstract

The paper deals with diagnostic issues concerning endoscopic examinations of working spaces within marine diesel engines. In the beginning, endoscopy apparatus being on laboratory equipment of the Department of Ship Power Plants of Gdansk University of Technology has been characterized. The endoscopy considerations have been focused on theoretical bases of a digital image processing and especially - on the "Shadow" measurement method.

Second part of the paper is devoted to operation damages of medium- and high speed engines that have been physically analyzed in many aspects of origin as well as places and character of their occurrence. There have been also considered possibilities of direct and indirect identification of the well known and recognizable operational unserviceable states of the elements of a piston's constructional system by means of endoscopic methods.

The results of the author's own research as well as the accessible results of diagnostic research of the self ignition engines applied in automotive and railway transport have explained and proved the most probable reasons for the piston failures' occurrence.

***Keywords**: technical diagnostics, endoscopic investigation, marine diesel engine, pistons' failures.*

1. INTRODUCTION

A visual examination of surfaces, creating piston engines' working spaces, by means of specialist view-finders called endoscopies represents at present almost the basic diagnostic method for those marine diesel engines which are not equipped with indicator valves in standard [4]. During investigations a surface structure of the constructional material is visible like through magnifying glass, which makes a detection, recognition and also quantitative assessment of the occurring damages and material defects possible. It especially concerns such defects that usually do not generate the observable alterations of diagnostic parameters' values. The endoscopic examination of the engine being stopped (laid off) permits the user to estimate immediately a degree of the constructional elements' waste and fouling. The pistons of cylinder sets represent vulnerable engines elements that, on one hand, are characterized with the significant frequency of damages occurrence and, on the other hand - with high diagnostic susceptibility within the range of the endoscopic method application.

2. ENDOSCOPIC IMAGE PROCESSING

A diagnostic base of the Ship Power Plants' Department of the Gdansk University of Technology is equipped with the EVEREST digital videoendoscope of XLG3 type that has got a special measuring head enabling a qualitative and quantitative identification of the detected surface defects by means of so called the "Shadow" method - Fig. 1.

Fig. 1. Engine's examination by means of EVEREST XLG3 digital measuring videoendoscope [12]

It stands for a milestone of the endoscopy evaluation because eliminates the largest weakness of the traditionally applied optical boroscopes and fiberoscopes, namely, a dependence of the observed object's dimension on a distance between the speculum probe's lens and studied surface. In such a situation there is a lack of appropriate reference patterns that could be used in order to qualify real dimensions of the detected surface defects. In traditional, optical approach to such an issue the comparative method is applied by the calibrated measuring profiles being put on speculum probe's ending [2,4]. However, this is the very inconvenient method that requires a large experience and manual skills of the operator. Moreover, it creates essential threats for the studied engine's reliability, because measuring profiles might be accidentally broken off during manipulating the endoscope optics. Even the smallest part the profile left in an engine's working space excludes its further usage until the left element is removed (but this is not an easy operation and always possible to work out in operation conditions).

The "Shadow" digital image processing method, basing on theory of triangulation[1], enables measuring the seen paintings in such way, to give the quasi three-dimensionality impression, with its depth, the massiveness and the mutual distribution. The ending of videoendoscope's speculum head is equipped with the special optics generating the shadow of a characteristic shape (the most often the shape of a straight line).inside the light stream (like

[1] W. Snellius was the creator of triangulation theory (1615). The measurement method consists in division of the measuring area into adjacent rectangular triangles and marks on the plane the co-ordinates of points by means of application of the trigonometrical functions.

the projector) on surface of the studied element. The projection of the shadow holds at a known angle of the speculum head position, in relation to the observed surface and a known angle of the observation sector. The shadow generated in the vicinity of a detected defect is then located and recorded by a CCD camera which is placed inside an assembly head. The nearer to the observed surface is the speculum head the nearer from the left side of monitor screen is the shadow line. Because, a position of the shadow generating the painting on the matrix of LCD monitor screen is well-known, a magnification of the painting can be simply calculated. Moreover, the linear dimension of distance among individual pixels, and then the real dimensions of detected surface defects can be consequently evaluated.

Advertisement brochures and guides published by the EVEREST Company, as the patent's owner, as well as publications of the present article's Author contain a detailed characterization of the "Shadow" method measurement technology [4,5,12].

Within the "Shadow" method the following measurement options are accessible.

- length,
- multi-segment length, length broken (circuit),
- distance from point to base straight line
- depth (salience),
- area of surface (the area).

In every metrological option the measurement exactness is defined. When the operator possesses high skillfulness it reaches even 95-98% [12,14]. The maximum approach to studied surface of the speculum head represents the most essential factor of a high measurement exactness (the line of shadow moves towards the left side as the speculum head gets closer to the surface) as well as maintaining perpendicular to this surface the position of speculum head (the line of shadow runs perpendicularly to the basis of a monitor screen).

The possibility of taking the immediate decision in case of doubts regarding proper interpretation (unambiguous distinction) of detected surface defects being effective with decrease or accumulation of material is the very essential advantage of the "Shadow" method. Such diagnostic problems step out during the evaluation process of the working spaces' technical shape of combustion engines: piston or turbine.

Often, because of optical and light impressions the usual dirt on internal surfaces of the air and exhaust passages, in figure of mineral settlings or the products of burning the fuel (the carbon deposit), is interpreted as the corrosive or erosive decrements of the constructional material. The depression of surface (its larger distance from the speculum head) is associated with the shadow line's break and shift towards the right side of the screen, and its salience (its larger approaching to speculum head) - the shadow line's refraction and shift towards the left side of the screen.

3. PHYSICS OF THE OPERATION FAILURES

The piston, as a movable bottom of an engine combustion chamber, represents the constructional element that is liable to suffer the largest mechanical and thermal loads. A working medium pressure, achieving 20 MPa and temperature, up to 800 K make a great impact on the piston head's constructional material [10]. It might lead to significant deformations of the piston's shape as well as changes of mechanical properties of its. In the result of their further development serious primary failures of the piston usually occur. They inevitably drive to extensive secondary damages of the piston-crankshaft assembly and the engine as a whole.

In dependence on a place and character of the damages occurrence the piston's constructional form can be divided into elements as follows:

- piston head - cracks, plastic deformations, overheating and overburning,
- annular part - cracks, seizing, ring shelves' breaking off,
- leading part (bearing[2]) - corrosion, cracks, sizing,
- hub of the piston gudgeon pin - hub's cracks and seizing, cracks and chipping of the rings' grooves establishing the gudgeon pin.

However, taking into consideration the possibilities of the failures' direct endoscopic detection they are exclusively limited to the piston's head. Therefore, in order to explain possible reasons of the failures situated in the piston's other elements the piston has to be firstly dismantled from the engine because of the failures being identified (accessible by means of endoscopy) on the cylinder bearing surface (alternatively after disassembling a cylinder block).

Functioning disturbances of the engine's fuel and air fed system represent the most frequent reason of piston damages, particularly during the engine’s start-up process. That kind of disturbances usually leads to the engine’s knocking (detonating the fuel combustion process), which is characterized by the very large speed of flame spreading within the combustion chamber (even a dozen or so times greater than at the normal burning). The intensive growth of pressure and temperature pulsation of the working medium follows. Consequently, the growth of thermal and mechanical loads of the combustion chamber's elements, in peculiarity the piston, occurs [6]. In the result of increasing temperature gradients within the piston's constructional structure its cyclic deformations and growth of thermal tensions appear. They exceed considerably the value limits that correspond to the engine's steady states running (even twice) [3]. Moreover, the piston's internal surface, where the largest material accumulation takes place (a region of the hub of the piston gudgeon pin), represents the particularly vulnerable piston's region, from a thermal tension point of view sight [3,10]. The thermal tension of this piston head's region, in case of high-charged marine diesel engines (a mean effective pressure above 2 MPa), is able to achieve comparable values in relation to internal tensions from gas forces [10].

A considerable, local piston head's overheating, up to exceeding a plasticity border of the constructional material, causes its local upset. If a temperature growth of the piston's head is only temporary there could be also affected a temporary wearing down and even seizing the piston’s crown in the cylinder bearing surface - Fig. 2a [4].

A low-cycle fatigue of the piston's head up to cracks' occurrence, connected with cyclic squeezing and spreading tensions of the constructional material represents a different consequence of this phenomenon - Fig. 2b. They result from cyclic, respectively: the material overheating and widening as well as its cooling down and shrinking, during the engine's work cycle realization.

If a temperature growth of the piston's head is so large that the engine’s cooling system is not able to take over a warmth stream from the combustion chamber's constructional elements a stable growth of external dimensions of the piston's crown occurs. Consequently, lubricating conditions of the friction pair: piston-cylinder are altered. In such a situation, with regard on "favorable” conditions of appearing the second kind adhesive waste, a decay of the surface layer of the piston's constructional material (made of an aluminum alloy) takes place. Such a phenomenon consists in a local joining the frictional surface’s tops of the piston and cylinder liner together as well as a tearing off the aluminum particles and in consequence - their smearing over the cylinder bearing surface - Fig. 2c. Often, a surface of the rolling areas does not encircle the whole circuit of the cylinder liner. It proves, at small rolling thickness, that only a short-lived excess of the flow temperature of the piston's crown happened. The rings, that work properly, take off the piston's material rolled on the cylinder surface after lowering

[2] It transfers a normal force on the cylinder bearing surface.

the piston head's temperature when the lubricating conditions get better. Unfortunately, it usually produces undesirable results - an intensive growth of the aluminum particles quantity in the engine's lube oil system - Fig. 2d.

The introduced course of the failure appearing is the most probable in a situation when the engine runs on idle or on partial loads. In that kind of conditions the engine's fuel fed system works in the especially adverse mode having a direct impact on a course of the combustion process in the engine's cylinders. According to obligatory principles of the marine engines operation exploitation as well as the producers' technological requirements the injection system's regulating parameters i.e.: a fuel dose of fuel and geometrical beginning of the pressing are adjusted to conditions of the nominal load [11].

However, an inequality of the injection pumps' dosage enlarges considerably on partial loads. It means that the fuel dosage delivered to some cylinders is much larger than the required, optimum one foreseen on the assigned, settled engine's load. In case of multi-cylinder engines such a situation even gets worsened because in some cylinders, despite having delivered the fuel, self-ignitions do not come into existence. Hence, a delivering the larger than required fuel dose, either in the result of the dosage inequality or in the result of a fuel gathering during fallen out self-ignitions, leads also to detonating fuel combustion along the above mentioned consequences[3].

Very often, a self-ignition's time decay takes place along with detonating fuel combustion. In case of shifting the fuel burning beyond TDC an intensive temperature's growth of all the combustion chamber elements occurs (particularly the piston). A transfer of the combustion process on the expansion stroke (work), at an incomplete and imperfect character of this process, produces undesirable results with a quantity enlargement of carbon deposits settled on the surfaces that restrict the combustion chamber. A thick layer of the carbon deposit on the piston's head (Fig. 2e), coming into existence particularly intensely during the engine work on partial loads while it is maladjustment, worsens significantly the warmth flow conditions and consequently drives to the piston head's overheating - Fig. 2f.

An insufficient cooling in the vicinity of the first packing ring represents another one reason of the piston's overheating. If the time among starting, coupling and full loading a marine engine is compliant the producer's requirements and the engine cooling system is patent and correctly functioning, the engine should be in a settled thermal state. It means that the cooling system should "keep up" with a warmth stream from the unit: piston-rings-cylinder liner (so called PRC unit).

However, in result of, for example, the excessive deposits of calcium carbonates ($CaCO_3$) and magnesium ($MgCO_3$), creating so called the boiler scale, which is formed in the engine's cooling spaces after a thermal splitting the hydrogen carbonate contained in the water, a total or local patency loss of the cooling system may occur. Additionally, an impact of excessive vibrations transferred on the engine's constructional elements from the vessels properell does not stay without meaning. The vibrations could intensify a process of tearing the boiler scale layers off from internal surfaces of the cooling channels, which choke the cooling medium's flow. An extensive character of piston damages, especially considerable decrements of the piston crown's material (through the high-temperature diffusion) testify about the long-lasting high temperatures' impact. It is also very essential a location of the largest piston damages that, in such cases, occur in direction of the normal force effect i.e. in natural direction of the carrying away heat stream. Extensive damages in this piston's region testify about losses of the possibility of the effective carrying away heat to the cylinder liner through the first sealing ring. In case of temporary engine's torque overload, a different waste image occurs - the

[3] A rings shelves' fracture often stands for an additional consequence of the detonating fuel combustion in engine's cylinders (between the first and the second packing ring) as well as mechanical damages witin the crank arrangement.

piston crown, at the correctly functioning injector, widens uniformly and the waste area on piston's surface is also proportionate on its whole circuit. Moreover, a whole phenomenon of the adhesive waste proceeds at the considerably lower temperature. In such a situation grooves and rollings both on the piston and cylinder bearing surface are clearly visible. However, because of injectors' disfunctioning e.g. in result of losses of the sprayer's tightness or its spraying holes patency, a chronic fuel burning along with piston's overheating is possible. Traces of the fuel after-burning occur on the piston head's surface (fig. 3a), and in the extreme case - even the constructional material's burn out until a total piston melting.

An incomplete fuel burning in engine's cylinders causes the intensity enlargement of the forming carbon deposits which are washed-off from the cylinder bearing surface with lube oil. However, a part of the carbon deposit, gathering in rings' grooves, creates a tight and hard deposable layer in which piston rings are successively bring to a standstill. They lose an ability to adhere springy to the cylinder bearing surface. Consequently, a combustion chamber loses its tightness what follows a blow-by of the hot exhaust into the crankcase along the cylinder bearing surface. Finally, an oil film burns that drives to the total piston rings' immobilization - fig. 3b. In such a situation the extensive seizures within a piston' leading part occur very quickly. There is also another reason of the piston rings' immobilization associated with the metallic particles' penetration into rings' grooves. They represent a product of the adhesive waste of a piston and cylinder bearing surface. Prime reasons of this phenomenon have been precisely described in an initial part of this paragraph.

The often happened damage of the piston's head is the result of direct strikes over its surface the valve heads, injector as well as broken out constructional elements which are placed in a bottom plate of the cylinder head. Usually, such damages do not eliminate the pistons from further operation, but under condition, that their character and dimensions are contained in borders of operation tolerances that are determined by the engine's constructors [11]. A presence of shallow dents on the surface of a piston's crown of which correspond to the valve heads' sizes and shape (fig. 3c), confirms piston strokes over the valves caused by a canceling clearance between the valves being open and piston head, while the piston is in TDC position after the scavenging stroke. Timing angle maladjustment or valve stem's hang-up in the valve guide might represent a prime reason of such strokes. Moreover the strokes often result from the excessive clearances in the assembly of crank-shaft, that occur e.g. in the result of an excessive waste of the piston pin's hub, bearing shells etc. There is also possible a further transfer of impact (shock) loads on other elements of the valve timing assembly: cams, lifter, valve rockers etc. which might also undergo extensive damages, representing a secondary consequence of the piston's strokes.

In selected pistons' solutions the special necks in the shape of valve heads are milled on the pistons' crowns, but they are considerably deeper. The necks aim to eliminate a possibility of mechanical damages of the valve stems (and their secondary consequences in the aspect of extensive cylinder assembly damages) in case of the valve timing maladjustment. However, it makes sense only if a neck's depth is larger than the valve lift.

Piston mechanical deformations represent its relatively often occurred damages. They are usually caused by the valve's part (the exhaust valve generally) e.g. a broken off valve's head, which has fallen into the combustion chamber - fig. 3d. During its operation the cylinder valve is forced to move along its spindle in the guide which undergoes irreversible friction wear. The process goes at high temperature of the spindle whose additional task is to absorb heat from the valve head. As a result an excessive increase of radial clearance is produced between the guide and spindle, which, in all the cases, leads to an undesirable skew of the valve especially, that it is loaded with transverse component from the pressure of the timing lever or cam. Consequently, a loss of cylinder tightness (compression pressure drop), gas eruption, returnable flows, lubricant leakage from the spindle – guide precision pair until an intensive wear of the entire valve unit is reached. The phenomenon may especially intensive develop in the case of supplying the engine with fuel oil of high sulphur content. The cases are known of completely burned-out valves as a subsequent result of extensive wear of the valve guide [8,10,11]. In the extreme case, cracking the valve spindle, its falling down into cylinder space and subsequent failures of the "piston-piston rings-cylinder" system, including piston cracking, can happen. An observable symptom of worn valve guides are smoked valve springs, that indicates a lack of tightness of the combustion chamber.

A presence of the punctual dent on the piston head, that is placed in an injector's axis (fig. 3e,f) testifies about the piston's mechanical strokes over the injector resulting from e.g. the elongated piston stroke (for some high- speed engines, a distance between the piston head and injector's ending at TDC is not more than 3-4 mm [11]). Such a situation might represent a consequence of excessive clearances in an assembly of crank-shaft or intensive wear and tear of bearing pans.

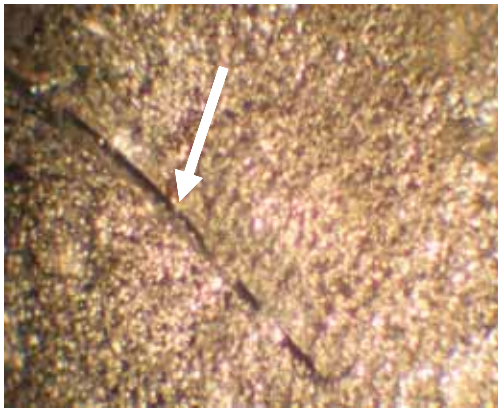
An excessive injector's lowering which is mounted in the head stands for different probable reason of this type dents' formation, caused by the inappropriate-chosen washer under the injector. It in such a situation was one should explain the prime causes of the dent's presence, and independently - the injector has to be definitely exchanged. Further operation in such state usually drives to cracks and even breaking-off the injector's ending. Moreover, an intensive fuel leakage to cylindrical spaces and thinning the lubricative oil with fuel might occur until to development of the extensive secondary damages in cylindrical sets, threatening with the engine's breakdown.

4. FINAL REMARKS AND CONCLUSIONS

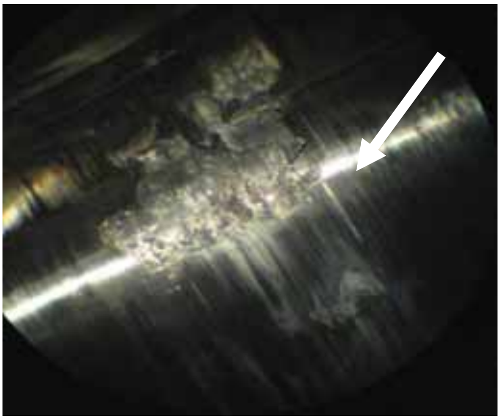
Many material defects detection on the surfaces, which limit working spaces of an IC piston engine, by means of endoscopies represents one of the youngest methods in technical diagnostics. On the basis of a character and dimensions of identified damages it is possible to evaluate a technical state of not only directly accessible constructional elements of the engine's working spaces, but also, in indirect way, it is possible to create an opinion about a technical state of these engine's constructional elements, which are not directly accessible and which co-operate with working spaces [10,11,12]. For example, the diagnosis about technical state of piston's annular or leading part could be formulated in indirect way on the basis of endoscopic investigations of the cylinder bearing surface and a character of detected-on surface defects, even though there is no possibility of carrying out an direct endoscopic evaluation of these piston's regions [4,5,9].

Within a characteristic of endoscope identified pistons' damages of marine diesel engines a lot of attention was devoted to medium- and high-speed engines that characterize a larger intensity of damages' occurrence of this constructional element in relation to slow-speed engines. In order to explain the most probable reasons of a piston damages' formation, the Author's own investigations were used as well as accessible results of diagnostic investigations of the SI engines applied in the road and rail transportation.

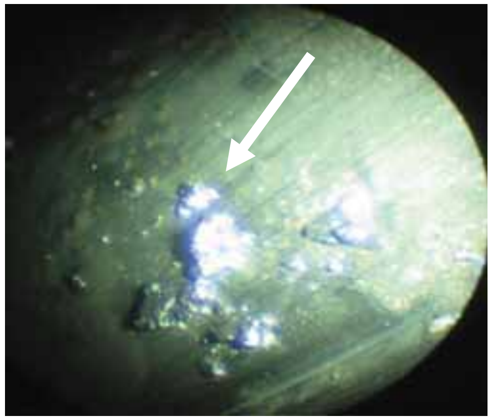
a) cylinder bearing surface - friction traces originated from the piston leading part



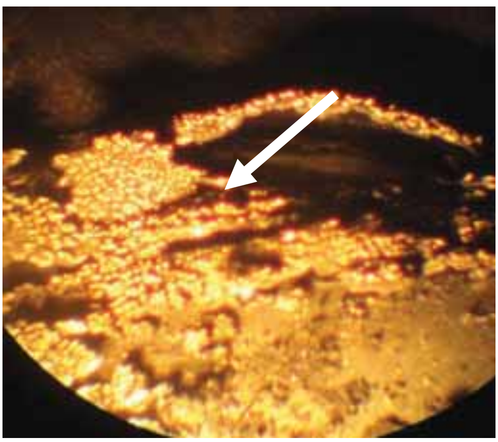
b) piston head – a crack fatigue



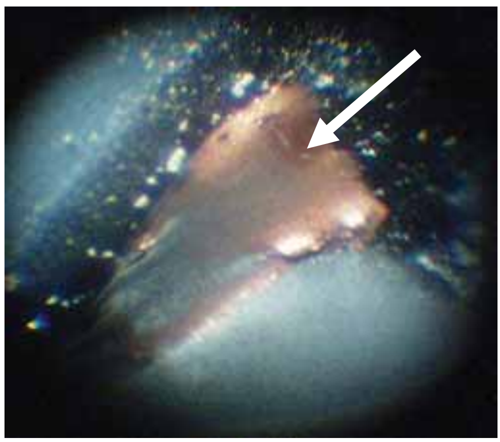
c) cylinder bearing surface – smearing traces of the piston material's layer (aluminum alloy)



d) cylinder bearing surface – metal fillings (aluminum alloy)



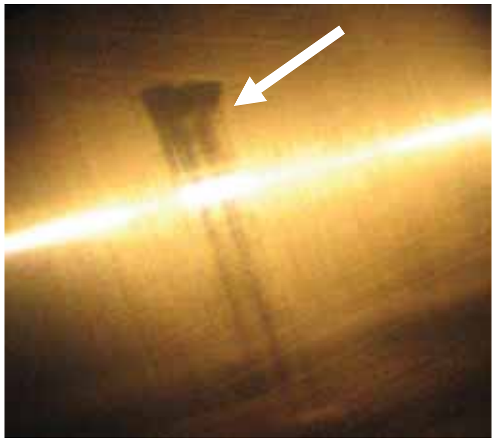
e) piston head – a thick layer of the carbon deposit



f) piston head – a local overheating of the constructional material

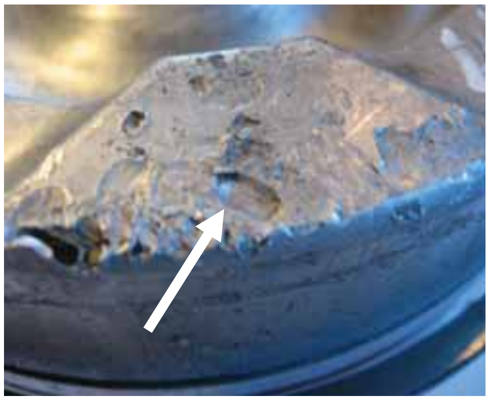
Fig. 2. Endoscopic image of marine engine pistons' damages and their consequences

a) piston head – traces of the fuel after-burning

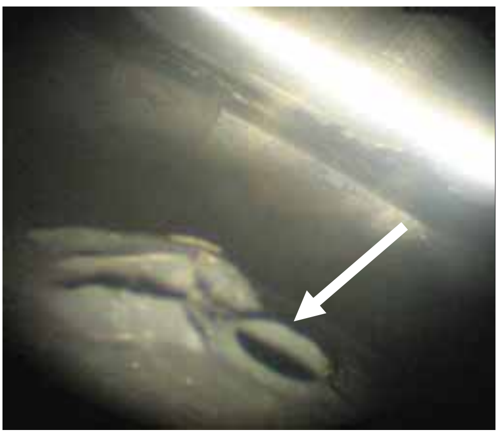


b) cylinder bearing surface – traces of the piston ring's seizing

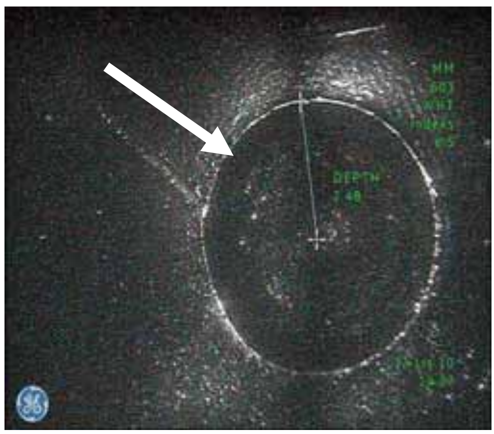
c) piston head – traces of the piston strokes over the valves (a state after the "water hammer")



d) piston head's crown – traces of the strokes made by the broken-off part of the valve head



e) piston head – a dent after the piston stroke over the injection sprayer



f) piston head – a measurement of the dent's depth by means of the „Shadow" method (2,48 mm)

Fig. 3. Endoscopic image of marine engine pistons' damages and their consequences

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FUZZY LOGIC IN THE ASSESSMENT OF HAZARDS TO SHIP POWER PLANT OPERATOR

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Abstract

Complex technical objects such as ship power plants are the source of many hazards to their operators. Identification and elimination of these hazards on the finished objects is a very labor-intensive task and involves significant financial outlays. It would therefore be advisable to carry out these activities much earlier - at the design stage. However, this entails certain difficulties. Depending on the design phase, the designer has a limited amount of information from which the operator's safety can be assessed. Additionally, this information is associated with considerable uncertainty. These difficulties can be overcome by using subjective estimates of persons having practical knowledge in the field of our interest - experts. Such knowledge can be formulated most easily in linguistic categories, i.e. fuzzy logic language.

This paper presents the construction basis for a ship power plant operator risk assessment system at the design stage.

This system was based on a fuzzy inference mechanism. Use of this system will enable the identification of hazards for ship power plant operators and will indicate the necessary corrective actions.

Keywords: *safety, design process, ship power plant, hazard, fuzzy inference*

1. Introduction

Design of complex technical objects is a very complicated task, involving a series of steps. In the early stages of the project a general outline of the object is created Patterns of individual installations are formed. Thus we can say that the resource about the proposed facility is very limited as the project situation is progressively changing. The designer's knowledge expands with the project's progress. During the design process the available information is associated with a high degree of inaccuracy and uncertainty. Therefore, the most accurate analysis of the risks associated with the functioning of this facility should be undertaken at the existing facility during its operation. This approach, however, involves time-consuming and costly modifications in case of any irregularities. The solution may be to analyze operator hazards during the design process of a technical facility. However, as already mentioned, this is subject to certain limitations. By supporting the knowledge of experienced operators of similar technical objects it is possible to analyze operator risk in the early stages of design. This knowledge is not always available in an easy-to-use form. In most cases it is the work experience of experts in the field. Acquiring and recording this knowledge to allow its continued use is not an easy task. In the view of the author, the most convenient solution for collecting a considerable amount of knowledge and its use is to build a computer system. The aim would be to obtain information on the proposed facility during its design and conduct a risk analysis of its operator. A suitable record of the knowledge acquired from experts can be implemented using fuzzy logic. It allows you to reflect the fuzziness of concepts expressed by people in everyday language. Fuzzy logic allows you to perform inference in a manner very similar to human reasoning. Thus, the use of fuzzy logic makes it easier to map the path of experts' inferences.

The specific complex technical object is a ship power plant. It contains a large number of different kinds of machines and devices connected to systems providing propulsion and performing many other important functions. Such an accumulation of potentially hazardous structural units (the term adopted from [6, 7]) in a limited space of engine room poses a real hazard to its operators. Therefore it is advisable to carry out risk analysis within the engine room.

2. A risk analysis during the design process

Design of engine room can be done in the following phases:

- possible conceptual design,
- offer project,
- preliminary design (contract),
- technical-classification design,
- design workshop (working)
- passing documentation (operating).

This paper [6] assumes that the potential impact on the operator's safety during the design process occurs during the initial phase of the project and the technical classification phase. The early stage of the project allows only for a general assessment of hazards. It can be used to pre-determine the hazardous areas [6, 7]. These areas are selected on the basis of operating in a given operating condition of the vessel's structural units in terms of hazards posed to the operator. In areas of potential danger remedies are proposed to lower the level of risk for the operator. At the technical classification design stage, during hazard analysis, the same areas are taken into account again. In this phase, many important details become clearer, such as the construction and placement of structural units. It is therefore necessary to reassess the subsequent identification of hazardous agents. After their assessment, it is possible to identify risks and propose risk mitigation measures.

3. Fuzzy risk assessment

In both stages of the project, the operator's risk assessment is made based on fuzzy logic. Inference in the initial phase of the project is based on information about the types of risks generated by the various structural units. Also factored in is the possibility of operator contact with the manifestation of the hazard. For the purposes of risk assessment, the operator has taken six types of hazards (Fig.1). Each hazard is assigned a corresponding gamut of manifestations.

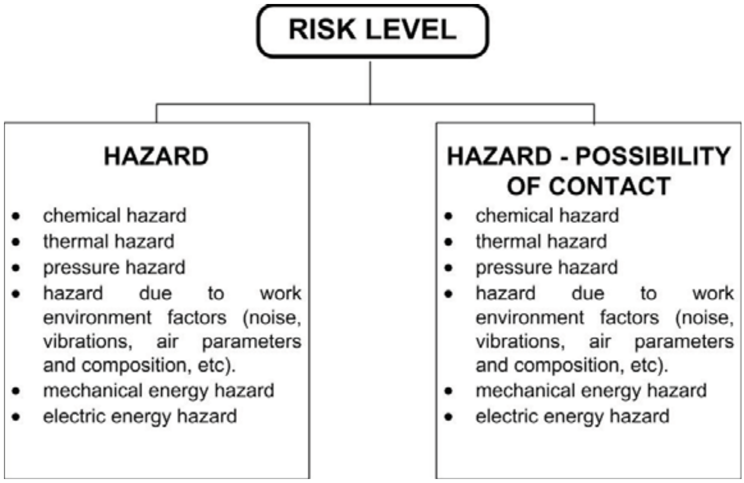

Fig. 1. Risk assessment scheme

In order to determine the level of risk posed by the various manifestations of hazards a survey was built. Based on the results obtained from the responses of experienced ship power operators fuzzy sets have been prepared. Sample collections are shown in Fig.2.

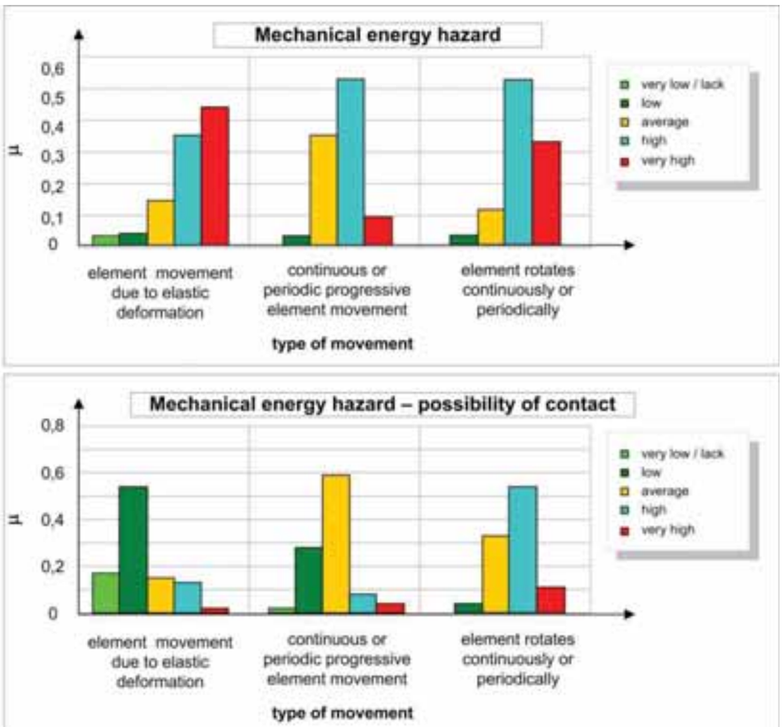

Fig. 2. Fuzzy sets: mechanical energy hazard and mechanical energy hazard – possibility of contact

The resulting fuzzy sets defining the operator's level of risk are reflected using the five linguistic terms:

– very low/lack,

– low,
– average,
– high
– very high.

These sets assume the shape of a triangle (Fig. 3).

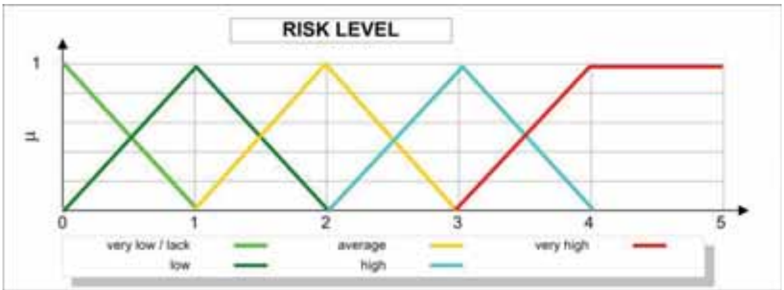

Fig. 3. Fuzzy sets representing level of risk

The result of inference is an acute value between 0-5. Referring this value to the collections from Fig.3, we can assess the level of risk by using linguistic terms. Getting the resulting value of the risk level for the given type of hazard is performed using fuzzy inference mechanism based on the Mamdani model [4, 5, 8].

The overall level of risk resulting from the impact of all types of hazards generated by the considered structural units is obtained after aggregation, using the same model of inference.

Using this method of risk assessment allows the operator to pre-identify areas in which his actions may pose a potential risk to operators. Analysis and risk assessment of operator at the preliminary stage of the project is elaborated in [1].

The next phase of the project includes outlines of individual pipelines installations at a ship power plant, structural solutions for foundations and the mounting of the main machinery and equipment, construction of the shaft together with the necessary strength calculations, vibrations, etc., the plan for dismantling machinery and equipment in the power plant, plans of workshops and storerooms.

At this stage the available knowledge enables a more detailed definition of the risk level of the operator. It can be assumed that it will be a function of factors derived from [8]:

– operation of machinery and equipment,
– accessibility to the site of operational activities,
– position of the operator performing specific operational activity,
– type of operational activities.

Risks associated with the work of machines and equipment are dependent on their function. In this case again the following types of hazards are considered:

– chemical hazard,
– thermal hazard,
– pressure hazard,
– hazard due to work environment factors (noise, vibrations, air parameters and composition, etc),
– mechanical energy hazard,
– electric energy hazard.

We shall take into consideration only those structural units which at the initial stage of the project obtained a significant risk level (**RL**). It is necessary to check whether the remedial measures (**RM**) provided for those units sufficiently reduce the risk to the operator. Depending on the manner in which a unit will be operated, the operator will be exposed to various hazards. The type of activity undertaken by the operator has a significant impact on the type of hazard generated by the structural unit. For each hazard, it is necessary therefore to examine the effects of actions that will be carried out by the operator.

The operator carrying out the operation of the structural unit is mainly exposed to the hazards associated directly with this unit ($\mathbf{H_u}$). However, in view of the specificities of the engine room, in some cases it is necessary to take into account the risks in the surroundings of an operator ($\mathbf{H_s}$). When the immediate environment of the operator contains units that are a potential hazard, their impact should also be taken into consideration.

It is assumed that the accessibility to the place of operational activities in question (**APA)** together with the location of the structural unit at the correct height (**LSU)** significantly affect the course of the work performed by the operator and the position adopted by him when performing maintenance. These values will define the (**ON**) operational nuisance. Forced unusual positions of the operator's body and access restrictions when operating the structural unit increase the possibility of injury to the operator.

With regard to the impact of operations on the operator's risk, one should consider such factors as:

- the degree of activity differentiation (**AD**) - number of different elementary operations performed at the specified structural unit,
- the maximum range of performed movements (**RPM)** – the way activities are performed, for example by hand, arm, with or without the use of tools, etc.,
- the variability of adopted position (**VP**) - information that specifies the dynamics of the movements made by the operator .

The combination of these factors in risk assessment system operator is presented in Fig.4.

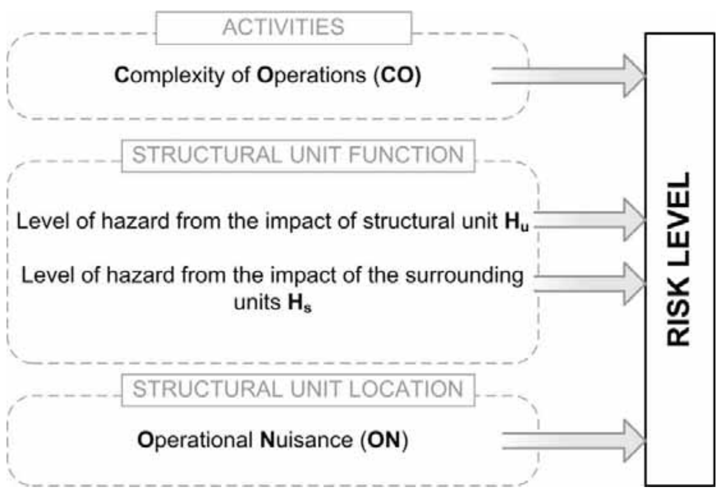

Fig. 4. An overview of the operator's risk assessment system

Building a fuzzy risk assessment system operator requires the creation of fuzzy representation of individual factors and their relationships by the relevant rules. Initially, for purposes of representing individual linguistic variables, fuzzy sets have been adopted in triangle and trapezoid shapes. Examples of fuzzy sets describing the factors that constitute operational nuisance (**ON**) are shown in Figures 5, 6 and 7. Under the numerical values on the abscissa, different scenarios of difficulties in accessing the structural unit are included. For example, the value of 10 was assigned to accessing the unit using only fingers requiring the operator to bend/lean against it at the same time.

Fuzzy rules are created in the form of **IF-THEN** expressions. If there are multiple premises, the **AND** operator is used. Created fuzzy rules have the following form:

IF *the accessibility to the place of operational activities APA is difficult* ***AND*** *the location of a structural unit LSU is below the knee* **THEN** *operational nuisance ON is substantial.*

IF *activities differentiation AD is large* **AND** *the range of performed movements RPM is wide* **AND** *the variability of adopted position VP is high* **THEN** *the complexity of operations CO is high.*

IF *the complexity of operations CO is high* **AND** *the hazard level H_u is high* **AND** *the operational nuisance ON is substantial* ***THEN*** *risk level RL is high.*

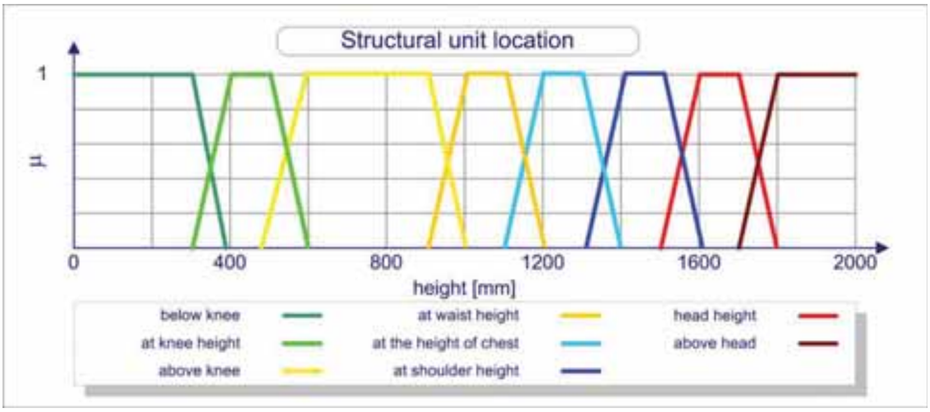

Fig. 5. Fuzzy sets describing the location of the structural unit

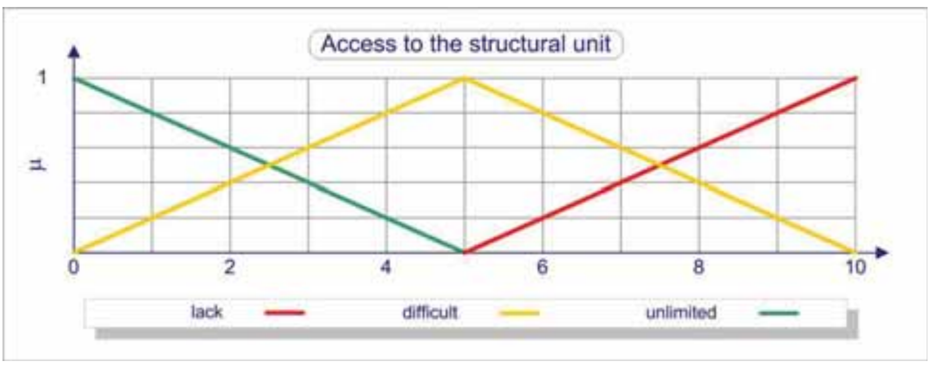

Fig. 6. Fuzzy sets describing the access to the structural unit

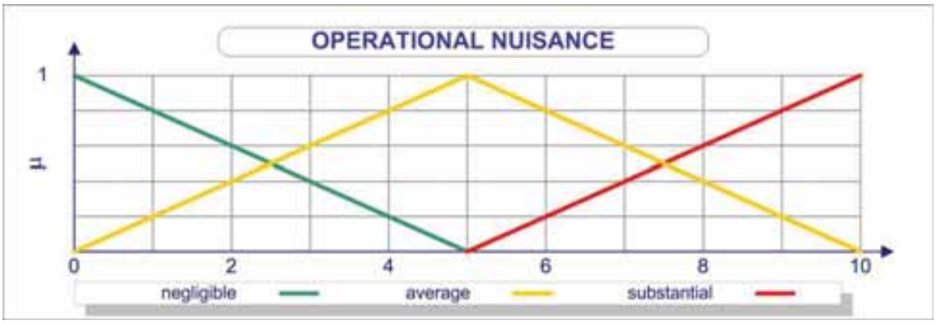

Fig. 7. Fuzzy sets describing the operational nuisance

Depending on the information required, the system will collect information through properly prepared windows that require making a specific choice or, for example, situational diagrams to be complemented by appropriate values (Fig.8).

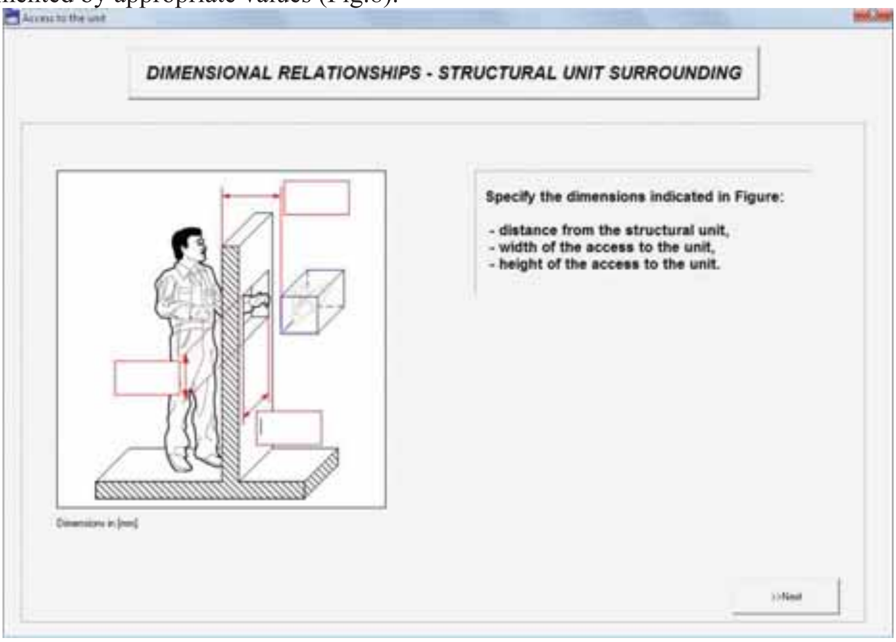

Fig. 8. Sample screen of operator's risk assessment system

The obtained information is converted into corresponding fuzzy sets based on the knowledge accumulated in the base system.

The result of the operator risk assessment process during the implementation of specific operational activities is a value in the interval <0, 10>. This value makes it possible to qualify the level of this risk as low, medium or high. Depending on the result, it will be possible to take appropriate action to improve the safety of any operator. Data collected by the system will enable identification of factors that significantly endanger the operator.

4. Conclusions

The solution based on fuzzy logic offers significant potential in safety modeling in the ship power plant. It is particularly useful in the early stages of the design where information about the safety of the operator is negligible or associated with considerable uncertainty.

The risk assessment carried out at the stage of preliminary design of the engine room, in view of the limited amount of available information, provides an initial opportunity to identify areas of risk. It is very important that specific preventive measures can be provided at this early stage of the project. However, as the project develops, new factors appear that are not included in this assessment and that could significantly threaten the operator. Then it becomes necessary to further evaluate hazardous areas, taking into account precisely these factors. In addition, it is necessary to verify measures already introduced.

Application of fuzzy inference allows interoperability of assessment systems for the various phases of the project. Moreover, inference performed by this method is very similar to human reasoning which makes it easier to process the relevant knowledge into a language understood by the computer.

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THE TAXONOMIC ANALYSIS OF ECOLOGICAL THREATS CAUSED BY A TECHNICAL INFRASTRUCTURE OF MOTOR TRANSPORT BASED ON THE TOTAL QUANTITY OF MATERIAL WASTE

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Abstract

The article presents the matters connected with modelling an ecological strategy using a taxonomic method for a technical infrastructure of motor transport. The ecological problems are exceptionally crucial for this sphere of service activities of transport means. The article shows the essential facilities of a technical infrastructure required for proper service operation with related ecological threats. It also presents the justification for using a taxonomic method to evaluate ecological issues. Based on the total quantity of material waste, the article presents the method, calculation results and their interpretation concerning the choice of an appropriate technology for application.

Keywords: *ecology, motor transport, technical support equipment of transport, taxonomic method*

1. Introduction

Motor transport plays a significant role in functioning of a market, not only a local, regional or domestic one, but it also has a global dimension [9]. Its development is accompanied by specific conditionings and their consequences. The development of a transport infrastructure, increasing number of roads and technical infrastructure facilities in addition to greater and greater motor traffic intensity, cause the necessity to put serious interest in safety of road traffic, reliability of devices and systems as well as protection of natural environment.

The ecological issues are important for the automotive branch, ranging from a vehicle design phase up to a recycling one [3,5]. The dynamic development and the expansion of motorization started to penetrate the environment without limits and without taking negative consequences into account. The direct result is still air, soil, surface and subterranean waters pollution, noise emission, the intensification of the greenhouse effect and the increase in quantity of waste products generated during operation and recycling of motor vehicles. An improperly organized and functioning technical infrastructure may also pose a serious threat to surrounding environment. This article addresses these issues and recommends an appropriate strategy for a logistic system for a technical infrastructure in the aspect of ecological problems.

2. Ecological threats caused by a technical infrastructure of motor transport

A technical infrastructure equipped with installations of adequate quality and special tools is of basic importance for the development of motor transport, because it enables proper operation of vehicles. An unsuitably organized and functioning technical infrastructure causes premature wear or even devastation of vehicles, reduction of transport capabilities, and above all, inevitable losses in individual sectors of a national economy. A properly prepared technical infrastructure keeps motor vehicles in adequate condition, which in turn, enables them to be ready to operate immediately [2,11].

A technical infrastructure involves properly prepared and equipped facilities, which are designated to maintain and supply vehicles, their sale or rent, perform technical services and warranty or periodic inspections, comprehensive or selective diagnostics of functional blocks, perform routine or accident repairs, collect retired vehicles for recycling etc. [4,8].

The most crucial ecological problems concerning a technical infrastructure of transport are [15]:

- air pollution with exhaust fumes in places for servicing or repairing vehicles (operating engines),
- noise emission in aforementioned places (operating engines),
- management and possible recycling of operating fluids (motor oil, gear oils, brake and power steering oils, brake fluid, cooling fluids, AC fluids etc.),
- management of used and replaced units, subsets and elements (steel elements, non-iron metals, polymers, rubber elements etc.) during inspections or repairs,
- protection of environment against harmful influence of disposal sites for used vehicles designated for recycling.

3. The estimation of the total quantity of waste material in a technical infrastructure of motor transport

The flexible and automatic processes that are employed currently in facilities of a technical infrastructure of motor transport are characterized by great variety and innovation. The introduction of new strategies makes it possible to quickly reduce power and material consumption of transport means. A numerical taxonomy method can be used to help achieve optimal efficiency and accuracy [10]. This solution is close to conditions which concern transport means, that is why such an approach differs from other known optimization methods.

The reason for using the taxonomic method to analyse ecological problems occurring in a technical infrastructure was imposed because of parameters that together with descriptions of technological issues, intrinsically have different physical values and different measurement units. The structure of taxonomic models makes it possible to arrange researched technologies in a linear way, so it mirrors their location better within a multidimensional space of parameters. The taxonomic method is employed in natural science to good effect. In technical applications, projects which grasp comprehensively the issues of using those methods appear very rarely. Using this method to develop a logistic system with an ecological aspect in mind, establishes a new quality for technical applications.

In this article, the total quantity of material waste generated during performance of services has been chosen to be analysed with the taxonomic method. 15 companies contributed to the data concerning the quantity of material waste, with the assumption of the quantity of waste in a year, i.e. [kg/year] as a measurement unit. Selected companies belong to different ownership and organizational structures; among them there are economic entities offering wide range of services, as well as technical infrastructures of car showrooms, farm organizations, municipal and intercity transport, transport and spedition companies etc. Some of the analysed companies while giving access to documentation and materials with the range of logistic activities, have not agreed to reveal their names and affiliations; that is why they are named "technologies" in the article.

Since the taxonomic method requires the choice of the most beneficial variant with regard to accepted criteria, three additional parameters (so called exemplary) have been selected, which concern:

- CO_2 emission - WP1,
- overall “quality” of generated waste with regard to toxicity - WP3,
- energy demand with reference to pro-ecological projects - WP4.

The total quantity of material waste generated while providing services remains the fourth exemplary parameter - WP2. In table 1, there are exemplary parameters and their corresponding measurement units, which have been selected for calculations.

Tab. 1. The parameters for pro-ecological assessment of a technical infrastructure of transport selected for analysis

No.	PARAMETER SYMBOL	PARAMETER TYPE	MEASUREMENT UNIT
1.	P1	Emission of carbon dioxide (CO_2) to atmosphere	[kg/year]
2.	P2	Total quantity of material waste	[kg/year]
3.	P3	Overall "quality" of generated waste (toxicity)	[[0-1]*
4.	P4	Energy demand with reference to pro-ecological projects	[kWh/month]

* 0 - lowest, 1 - highest

The acquisition of the information regarding the selected parameters has run into serious difficulties in some companies. The required data has not always been found in available documentation, for instance due to briefness, treatment of waste as a total without selection with regard to a type, the lack of competence of employees responsible for this issue etc. That is why, the information concerning the individual parameters is the result of the additional analyses of statistical data relative to the quantity of performed inspections, repairs, diagnostic assessments and other services within specified time. Accepted data for the selected parameters have been consulted repeatedly with technical supervision of the analysed companies, or even directly with the employees engaged in specific types of work, just to gain the most adequate and reliable results.

3.1. The interpretation of the study results

The description of the methodology for performing study, the basic taxonomic method equations and the algorithm for performing calculations concerning the ecological problems of technical infrastructure of transport have been included in their own elaborations, in treatises among others [12,13,14]. That is why the description of these issues has been omitted. The detailed method for determining the 4 exemplary parameters has been presented in the thesis [13].

The results of the study using the taxonomic method (dendrites) have been verified additionally by the Czekanowski's matrix [6,7]. In the event of the convergence of the results, it can be concluded that the accepted procedure for calculations is correct and it authorizes the unequivocal interpretation during formulation of conclusions.

During the analysis of the results and drawing conclusions, the basic role is played by the sequence of connected points and the values of average differences between these points. Proximity and grouping of the particular technologies which indicate the similarity of the examined parameter enable the choice of the optimal value.

The study results have been presented in a tabular and a graphical way as dendrites and the Czekanowski's matrix, namely:

- tab. 2 - a list of analysed parameters for each surveyed company (with exemplary parameters),
- tab. 3 - values of parameters determined according to the taxonomic method for the lowest quantity of waste in 5 companies among the surveyed,
- tab. 4 - average differences between surveyed technologies (according to table 3),
- tab. 5 - the diagonal matrix of Czekanowski (dendrite verification according to fig. 1),
- tab. 6 - values of parameters determined according to the taxonomic method for the average

quantity of waste in 5 companies among the surveyed,

- tab. 7 - average differences between surveyed technologies (according to table 6),
- tab. 8 - the diagonal matrix of Czekanowski (dendrite verification according to fig. 2),
- tab. 9 - values of parameters determined according to the taxonomic method for the highest quantity of waste in 5 companies among the surveyed,
- tab. 10 - average differences between surveyed technologies (according to table 9),
- tab. 11 - the diagonal matrix of Czekanowski (dendrite verification according to fig. 3),
- tab. 12 - average differences between the analysed technologies for each of the surveyed companies (according to table 2),
- tab. 13 - the diagonal matrix of Czekanowski for each examined technologies (dendrite verification according to fig. 4),
- fig. 1 - a dendrite for a differentiation of a technology according to the sum of waste for the 5 lowest values,
- fig. 2 - a dendrite for a differentiation of a technology according to the sum of waste for the 5 average values,
- fig. 3 - a dendrite for a differentiation of a technology according to the sum of waste for the 5 highest values,
- fig. 4 - a total dendrite for a differentiation of a technology according to the sum of waste for all 15 surveyed companies.

Tab. 2. A list of analysed parameters for each surveyed company (with exemplary parameters)

Parameters / Technology	P1	P2	P3	P4
1	929	58 518	0,35	800
2	1 328	63 372	0,72	900
3	1 679	55 846	0,27	950
4	1 000	31 982	0,26	600
5	2 647	113 516	0,13	975
6	1 221	48 288	0,49	740
7	3 017	106 424	0,31	250
8	1 921	84 961	0,87	1000
9	1 395	62 282	0,59	430
10	896	55 423	0,70	740
11	1 093	63 230	0,18	800
12	4 329	161 326	0,19	940
13	2 247	57 013	0,64	800
14	1 444	65 510	0,25	850
15	2 309	124 893	0,13	900
WP1	672	41 568	1	555
WP2	750	23 987	1	450

WP3	1 441	63 721	1	750
WP4	2 262	79 818	1	187,5

Tab. 3. Values of parameters for the lowest quantity of waste in 5 companies among the surveyed

Technology \ Parameters	P1	P2	P3	P4
4	1 000	31 982	0,26	600
6	1 221	48 288	0,49	740
10	896	55 423	0,70	740
3	1 679	55 846	0,27	950
13	2 247	57 013	0,64	800
WP1	672	41 568	1,00	555
WP2	750	23 987	1,00	450
WP3	1 441	63 721	1,00	750
WP4	2 262	79 818	1,00	188

Tab. 4. Average differences between surveyed technologies (according to tab. 3)

Technology	4	6	10	3	13	WP1	WP2	W3	WP4
4		1572,25	2252,50	2358,06	2502,58	1009,37	783,42	3058,33	4645,56
6	1572,25		1145,92	1195,88	1180,95	970,27	2352,95	1677,59	3221,37
10	2252,50	1145,92		534,46	615,15	1342,40	3019,58	893,54	2626,04
3	2358,06	1195,88	534,46		632,57	1568,22	3105,89	989,99	2590,46
13	2502,58	1180,95	615,15	632,57		1623,59	3238,84	802,90	2539,55
WP1	1009,37	970,27	1342,40	1568,22	1623,59		1689,32	2163,82	3809,71
WP2	783,42	2352,95	3019,58	3105,89	3238,84	1689,32		3837,11	5411,25
WP3	3058,33	1677,59	893,54	989,99	802,90	2163,82	3837,11		2001,91
WP4	4645,56	3221,37	2626,04	2590,46	2539,55	3809,71	5411,25	2001,91	

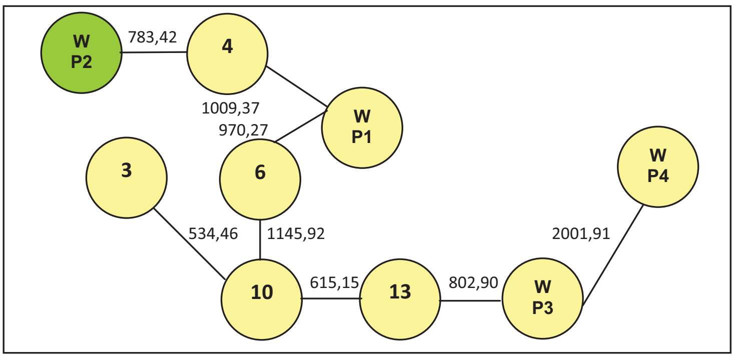
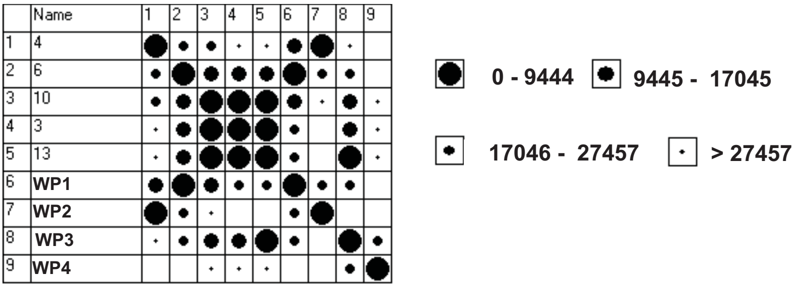


Fig. 1. A dendrite for a differentiation of a technology according to the sum of waste for the 5 lowest values

Tab. 5. The diagonal matrix of Czekanowski (dendrite verification according to fig. 1)



Tab. 6. Values of parameters for the average quantity of waste in 5 companies among the surveyed

Technology \ Parameters	P1	P2	P3	P4
1	929	58 518	0,35	800
9	1 395	62 282	0,59	430
11	1 093	63 230	0,18	800
2	1 328	63 372	0,72	900
14	1 444	65 510	0,25	850
WP1	672	41 568	1,00	555
WP2	750	23 987	1,00	450
WP3	1 441	63 721	1,00	750
WP4	2 262	79 818	1,00	188

Tab. 7. Average differences between surveyed technologies (according to tab. 6)

Technology	1	9	11	2	14	WP1	WP2	WP3	WP4
1		1226,35	1567,27	886,19	966,97	1707,93	3320,58	1080,33	2371,91
9	1226,35		752,88	542,67	951,12	2121,03	3682,79	550,84	1883,30
11	1567,27	752,88		851,67	1264,60	2251,43	3776,22	912,86	1757,25
2	886,19	542,67	851,67		479,62	2127,11	3803,37	357,72	1812,33
14	966,97	951,12	1264,60	479,62		2335,36	4011,45	745,66	1824,96
WP1	1707,93	2121,03	2251,43	2127,11	2335,36		1689,32	2163,82	3809,71
WP2	3320,58	3682,79	3776,22	3803,37	4011,45	1689,32		3837,11	5411,25
WP3	1080,33	550,84	912,86	357,72	745,66	2163,82	3837,11		2001,91
WP4	2371,91	1883,30	1757,25	1812,33	1824,96	3809,71	5411,25	2001,91	

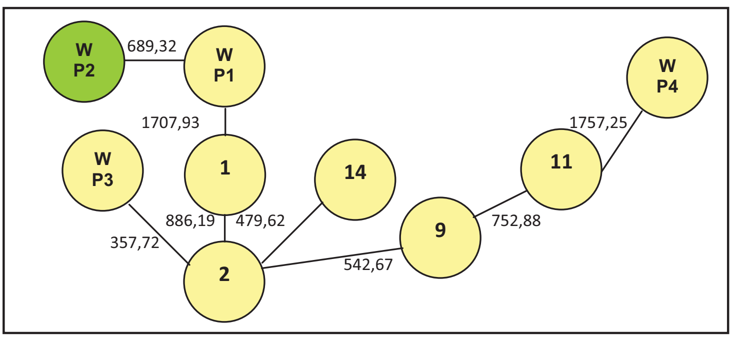
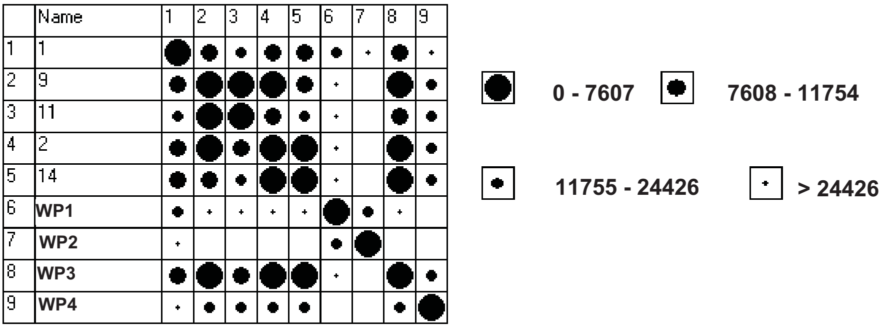


Fig. 2. A dendrite for a differentiation of a technology according to the sum of waste for the 5 average values

Tab. 8. The diagonal matrix of Czekanowski (dendrite verification according to fig. 2)



Tab. 9. Values of parameters for the highest quantity of waste in 5 companies among the surveyed

Parameters / Technology	P1	P2	P3	P4
8	1 921	84 961	0,87	1000
7	3 017	106 424	0,31	250
5	2 647	113 516	0,13	975
15	2 309	124 893	0,13	900
12	4 329	161 326	0,19	940
WP1	672	41 568	1,00	555
WP2	750	23 987	1,00	450
WP3	1 441	63 721	1,00	750
WP4	2 262	79 818	1,00	188

Tab. 10. Average differences between surveyed technologies (according to tab. 9)

Technology	8	7	5	15	12	*WP1*	WP2	WP3	WP4
8		2669,30	3767,05	4225,07	7466,31	4220,54	5896,81	2060,95	1091,67
7	2669,30		2158,48	2009,37	5340,02	6315,27	7981,52	4214,46	2577,86
5	3767,05	2158,48		2029,78	4645,04	7041,35	8650,23	5360,95	3710,97
15	4225,07	2009,37	2029,78		3703,76	8037,54	9714,72	5919,93	4383,47
12	7466,31	5340,02	4645,04	3703,76		11609,48	13285,65	9450,61	7901,15
WP1	4220,54	6315,27	7041,35	8037,54	11609,48		1689,32	2163,82	3809,71
WP2	5896,81	7981,52	8650,23	9714,72	13285,65	1689,32		3837,11	5411,25
WP3	2060,95	4214,46	5360,95	5919,93	9450,61	2163,82	3837,11		2001,91
WP4	1091,67	2577,86	3710,97	4383,47	7901,15	3809,71	5411,25	2001,91	

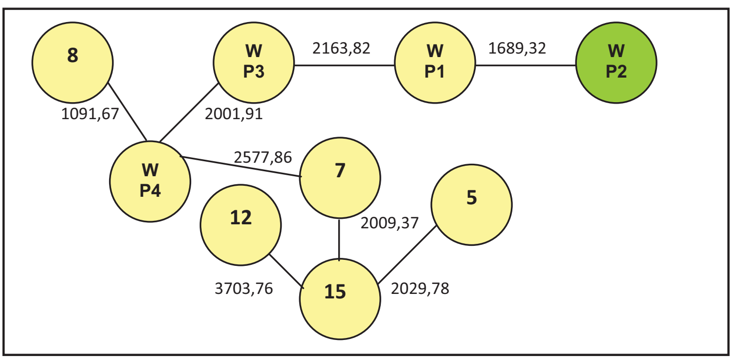
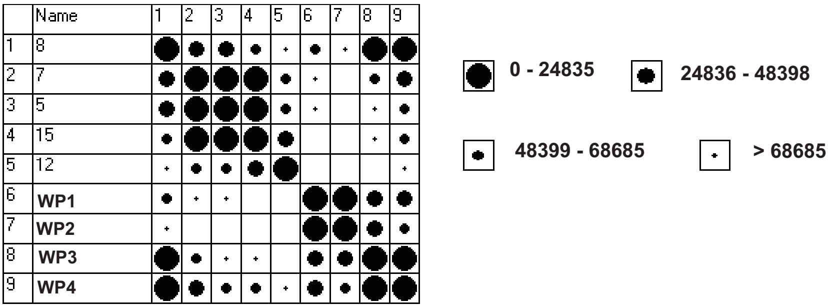


Fig. 3. A dendrite for a differentiation of a technology according to the sum of waste for the 5 highest values

Tab. 11. The diagonal matrix of Czekanowski (dendrite verification according to fig. 3)



Tab. 12. Average differences between the analysed technologies for each of the surveyed companies (according to tab. 2)

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	WP1	WP2	WP3	WP4
1		886,19	819,90	2548,44	5598,28	1319,63	4774,70	2831,19	1226,35	726,35	1567,27	9966,61	839,54	966,97	6397,88	1707,93	3320,58	1080,33	2371,91
2	886,19		911,61	3023,67	5181,97	1554,10	4277,94	2146,98	542,67	836,40	851,67	9483,09	931,81	479,62	5911,77	2127,11	3803,37	357,72	1812,33
3	819,90	911,61		2358,06	5562,80	1195,88	5034,81	2813,09	1101,31	534,46	1315,79	10182,73	632,57	1076,63	6617,01	1568,22	3105,89	989,99	2590,46
4	2548,44	3023,67	2358,06		7866,92	1572,25	7215,07	5116,20	2918,23	2252,50	3085,24	12506,66	2502,58	3232,55	8931,42	1009,37	783,42	3058,33	4645,56
5	5598,28	5181,97	5562,80	7866,92		6405,15	2158,48	3767,05	5332,56	5845,68	4989,51	4645,04	5712,36	5021,38	2029,78	7041,35	8650,23	5360,95	3710,97
6	1319,63	1554,10	1195,88	1572,25	6405,15		5704,15	3544,41	1812,08	1145,92	1993,45	10934,79	1180,95	1804,60	7362,17	970,27	2352,95	1677,59	3221,37
7	4774,70	4277,94	5034,81	7215,07	2158,48	5704,15		2669,30	4301,99	5079,70	4253,62	5340,02	4868,65	4302,35	2009,37	6315,27	7981,52	4214,46	2577,86
8	2831,19	2146,98	2813,09	5116,20	3767,05	3544,41	2669,30		2244,01	2885,16	2120,69	7466,31	2694,37	2204,77	4225,07	4220,54	5896,81	2060,95	1091,67
9	1226,35	542,67	1101,31	2918,23	5332,56	1812,08	4301,99	2244,01		1044,95	752,88	9606,77	900,94	951,12	6037,82	2121,03	3682,79	550,84	1883,30
10	726,35	836,40	534,46	2252,50	5845,68	1145,92	5079,70	2885,16	1044,95		1262,10	10271,86	615,15	1052,61	6695,79	1342,40	3019,58	893,54	2626,04
11	1567,27	851,67	1315,79	3085,24	4989,51	1993,45	4253,62	2120,69	752,88	1262,10		9510,66	1221,75	1264,61	5941,21	2251,43	3776,18	912,82	1757,21
12	9966,61	9483,09	10182,73	12506,66	4645,04	10934,79	5340,02	7466,31	9606,77	10271,86	9510,66		10048,75	9277,03	3703,76	11609,48	13285,65	9450,61	7901,15
13	839,54	931,81	632,57	2502,58	5712,36	1180,95	4868,65	2694,37	900,94	615,15	1221,75	10048,75		1207,25	6476,00	1623,59	3238,84	802,90	2539,55
14	966,97	479,62	1076,63	3232,55	5021,38	1804,60	4302,35	2204,77	951,12	1052,61	1264,61	9277,03	1207,25		5865,67	2335,36	4011,45	745,66	1824,96
15	6397,88	5911,77	6617,01	8931,42	2029,78	7362,17	2009,37	4225,07	6037,82	6695,79	5941,21	3703,76	6476,00	5865,67		8037,54	9714,72	5919,93	4383,47
WP1	1707,93	2127,11	1568,22	1009,37	7041,35	970,27	6315,27	4220,54	2121,03	1342,40	2251,43	11609,48	1623,59	2335,36	8037,54		1689,32	2163,82	3809,71
WP2	3320,58	3803,37	3105,89	783,42	8650,23	2352,95	7981,52	5896,81	3682,79	3019,58	3776,18	13285,65	3238,84	4011,45	9714,72	1689,32		3837,11	5411,25
WP3	1080,33	357,72	989,99	3058,33	5360,95	1677,59	4214,46	2060,95	550,84	893,54	912,82	9450,61	802,90	745,66	5919,93	2163,82	3837,11		2001,91
WP4	2371,91	1812,33	2590,46	4645,56	3710,97	3221,37	2577,86	1091,67	1883,30	2626,04	1757,21	7901,15	2539,55	1824,96	4383,47	3809,71	5411,25	2001,91	

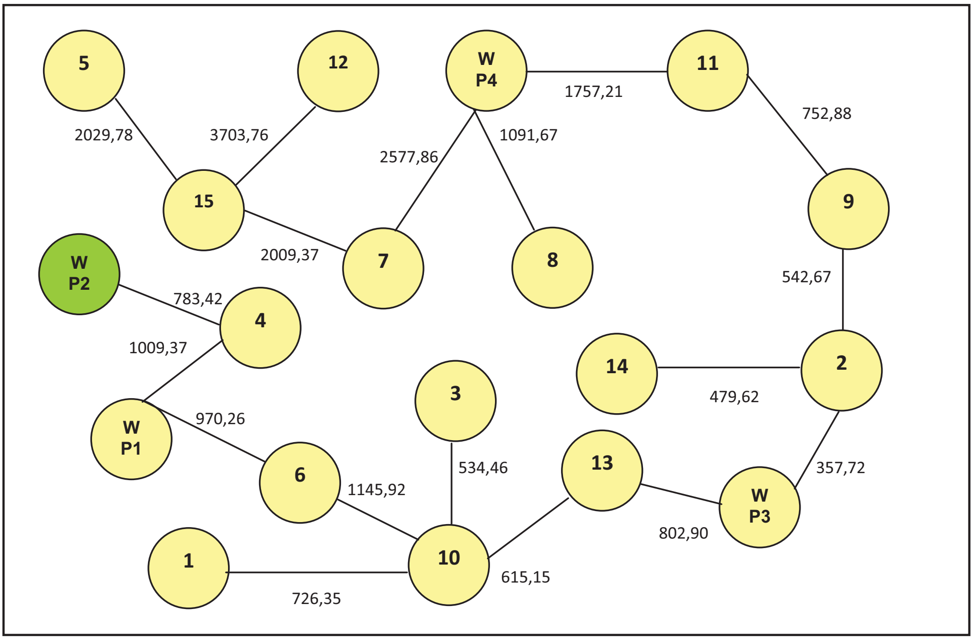
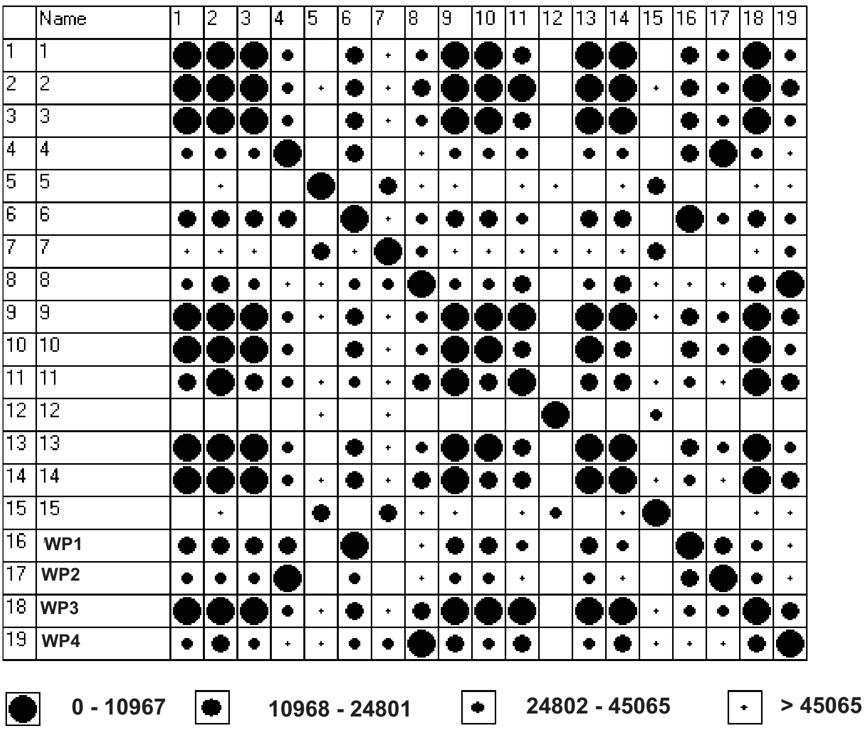


Fig. 4. A total dendrite for a differentiation of a technology according to the sum of waste for all 15 surveyed companies

Tab. 13. The diagonal matrix of Czekanowski for each examined technologies (dendrite verification according to fig. 4)



The analysis of the compiled results enables their interpretation in relation to the usefulness of the examined technologies:

- among the technologies with the lowest quantity of waste, the No. 4 technology is the most beneficial, because it appears a short taxonomic distance away from the WP2 exemplary technology; the No. 4 technology is also preferred due to the proximity to the WP1 exemplary technology and should be recommended for using in other companies,
- the analysis of the technology with the average values of the sum of waste, points at the No. 1 technology as the most beneficial, which places itself relatively close to the WP1 exemplary technology,
- within the group of the technologies with the highest quantity of gathered waste, the No. 8 technology is the most beneficial, showing the lowest values with reference to the WP4 exemplary technology.

Performing the analysis of the diagonal matrix of Czekanowski, in each of the three examined cases regarding the quantity of gathered waste, we confirm the results obtained by the taxonomic method. The diagonal matrix, which also includes all of the examined technologies, verifies the total dendrite positively.

4. Conclusions

The dynamic development of motor transport is connected with the necessity to reduce its negative impact on natural environment [1,15]. The facilities of a technical infrastructure are major participants in degradation of natural environment, due to diagnostic, routine and periodic inspections, repairs and other services concerning vehicles (e.g. car washes, paint shops etc.), which are performed there.

Using the taxonomic method allows for a dendritic arrangement, which mirrors the location of the examined factors within a multidimensional space of parameters, as opposed to any type of optimization method, which only allows for a linear arrangement of the selected problem indicators, occurring in a technical infrastructure of transport. The realisation of the subject matter presented in this article makes it possible to formulate the following conclusions of a general character:

- arranging pro-ecological technologies in a technical infrastructure of motor transport, using the taxonomic method is an effective way to find the point determined by the defined criteria within the space of the selected parameters,
- there is a possibility to verify the results of differentiation of pro-ecological technologies using the method of dendrite arrangement with the help of the diagonal matrix of Czekanowski,
- the dendritic arrangement with the taxonomic method and the matrix one with Czekanowski's method with reference to pro-ecological technologies in a technical infrastructure of motor transport give concurrent results,
- the taxonomic method can be used to choose a logistic system effectively with regard to the ecological problems of a technical infrastructure.

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EFFECT OF ADOPTED RULES OF INFERENCE AND METHODS OF DEFUZZIFICATION ON THE FINAL RESULT OF THE EVALUATION OF RELIABILITY MADE USING THE FUZZY LOGIC METHODS

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Abstract

The object of interest is to solve the problem of risk management of marine systems. But the main trouble is a lack of numerous and sure data on the reliability of the components of such systems. The methods based on the fuzzy logic seem to be helpful here. The goal of the article is to check the effect of using different fuzzy inference rules and methods of defuzzification on the final result of reliability assessment. The three rules of inference are taken into account: the Mamdani rule, the Larsen rule and the Tsukamoto rule. The second problem is the method of defuzzification of the result given in the form of fuzzy number into the real number. The several methods of defuzzification are discussed. The examples of using the above inference rules and defuzzification methods are presented.

Key words: *marine systems, reliability data, fuzzy sets, inference rules, defuzzification methods.*

1. Introduction

The fuzzy seems to be a very good tool when we have in our disposal very imprecise data. In contrast with binary logic, where the truth has only two values (1 – true, 0 – false) fuzzy logic variables of truth may vary from 0 to 1. It gives us possibility to deal with reasoning that is approximate rather than precise. And such situation is very typical in reliability analysis of technical system.

It is worth noting, that such imprecise tool like the fuzzy logic is widely used in control systems that require high precision. The idea arise: if the inference based on fuzzy logic works in control systems, it should also be helpful in drawing conclusions about the reliability of technical systems. Fuzzy inference (in another words fuzzy reasoning or approximate reasoning) uses linguistic rules, which are IF – THAN statements and typical logic AND – OR operators.

For example, the four fuzzy inference rules for car speed control system can take a form: 1st : IF the distance to the car in front is small AND the speed is low THAN maintain the speed level. 2nd: IF the distance to the car in front is small AND the speed is high THAN the speed should be reduced. 3rd: IF the distance to the car in front is large AND the speed is low THAN the speed should be increased. 4th: IF the distance to the car in front is large AND the speed is high THAN maintain the speed level. According to the example above, two fuzzy rules can be written for reliability assessment: 1st: IF the technical element is new THEN its reliability is high. 2nd: IF the technical element is old THEN its reliability is low.

To use such fuzzy rules about reliability it is necessary to build four fuzzy numbers: new technical element, old technical element, high reliability, low reliability. Those fuzzy numbers can be constructed on the basis of available reliability data or based on expert opinions. After that we can use the fuzzy rules of inference like: the Mamdani rule, the Larsen rule and the Tsukamoto rule.

Generally fuzzy inference process consists of five steps: creating a database of rules, writing those rules by using fuzzy numbers (fuzzification), drawing conclusion in the form of fuzzy number, changing the fuzzy number into real number (defuzzification).

2. Fuzzy inference rules for the reliability assessment of marine systems

As it has been already stated above, it is proposed to create two fuzzy inference rules for the reliability assessment. Those rules are:

1st rule: IF the technical element is new THEN its reliability is high.

2nd rule: IF the technical element is old THEN its reliability is low.

To build the rules it is necessary to create four fuzzy numbers: a new element, an old element, high reliability, low reliability.

Of course, we need some reliability data about technical elements being under consideration. Let's take as the example the data given in the fuzzy form about ship pipelines according to [1] that an average lifetime of ship's pipelines is:

- 5 – 7 years for galvanized steel pipelines,
- 5 – 9 years for copper pipelines,
- about 20 years for PVC pipelines,
- more than 20 years for cupronickel pipelines.

Now we focus our attention to galvanized steel pipelines. The task is to create two fuzzy numbers: (t_1) - the new pipeline and (t_2) - the old pipeline. An average lifetime of such pipelines is about six years. In that case the six year period will be essential to differentiate between the new pipelines and the old pipelines. Because we don't know the distribution of time to failure of such pipelines, therefore we have to create the simplest and commonly used fuzzy numbers given in the triangular form, shown in the Fig.1.

In a similar way the next two fuzzy numbers: (R_1) - high reliability and (R_2) - low reliability can be determined. It is obvious that high reliability expressed as a fuzzy number takes the form of "about one". Low reliability takes the form of "about zero". Those fuzzy numbers are presented in the Fig.1. Their membership functions have also typical triangular forms. The rule is that the opposite fuzzy numbers should be partly overlapped.

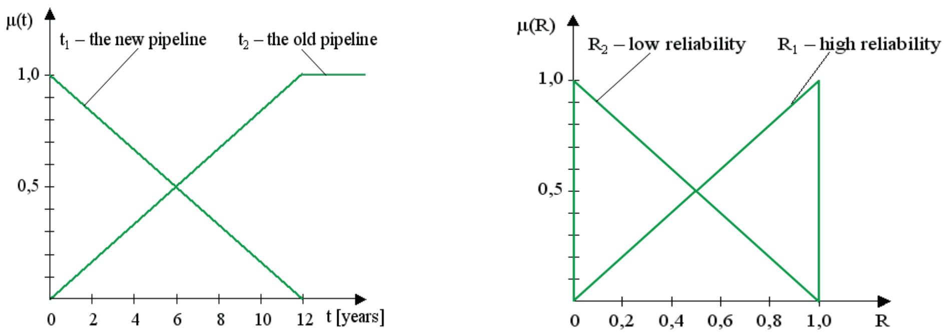


Fig. 1. Fuzzy numbers' membership functions: the new pipeline and the old pipeline

The applicability of fuzzy inference to assess the reliability will be presented with such an example: **to find the reliability of galvanized steel pipeline for the period of time: two years since the moment of installing as a quite new on board**. The two described above rules will be used. 1st rule: IF the pipeline is new THEN its reliability is high. 2nd rule: IF the pipeline is old THEN its reliability is low. And the three methods of inference will be taken into account: the Mamdani method, the Larsen method and the Tsukamoto method [2, 3], used in Fig. 2.

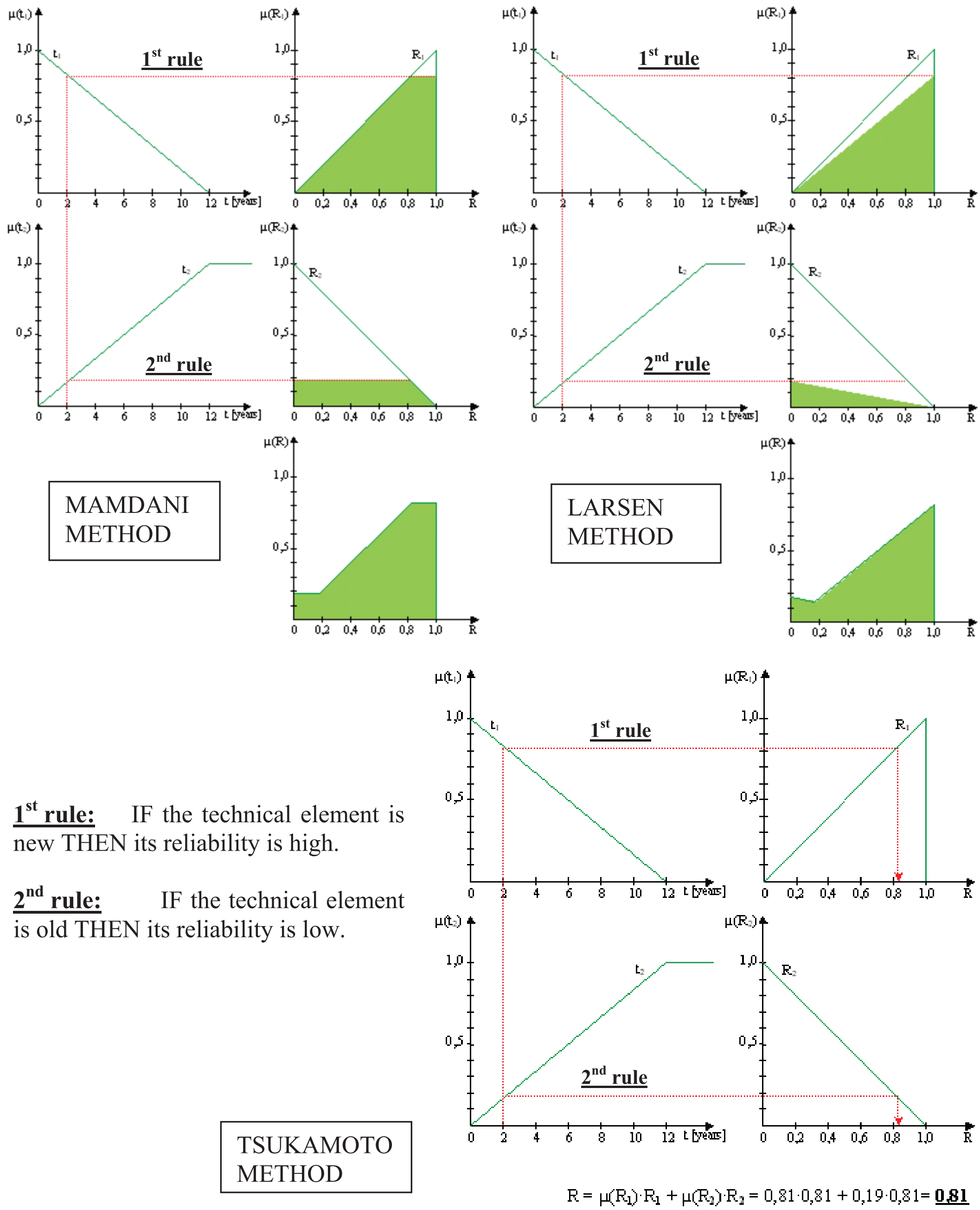
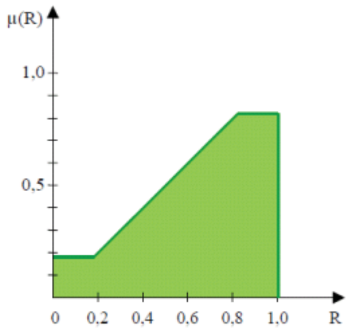


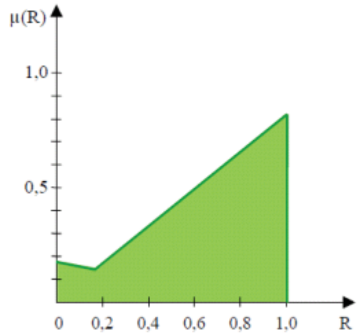
Fig. 2. Reliability of galvanized steel pipeline for two years period of time assessed with fuzzy inference methods

3. Comparison of obtained results

The comparison of obtained results is presented in Fig. 3. We can notice that the results are similar. The Mamdani method and the Larsen method give us results in the form of fuzzy numbers. The Tsukamoto method gives so called crisp value – it means the result is given in the form of real number. The problem is how to understand results, especially those in the form of fuzzy numbers. Looking for the Fig. 3. we can say that the reliability value obtained with the use of the Mamdani method is "about 0.8 – 1"; the reliability value obtained with the use of the Larsen method is "about 1"; the reliability value obtained with the use of the Tsukamoto method is 0.81.



R (2 years) is "**<u>about 0.81 – 1</u>**"



R (2 years) is "**<u>about 1</u>**"

Tsukamoto method

R (2 years) = 0,81

Fig. 3. Comparison of reliability assessment made with the use of Mamdani, Larsen, Tsukamoto inference rules

The results are not contradictory. The question is: which of those methods should be chosen in reliability analysis of marine systems? At first glance, Tsukamoto method seems to be most convenient. It gives a particular result in a form of real number. But we must realize that it is almost impossible the reliability value will be exactly like that. Especially in a situation when we have to our disposal very imprecise data. And such situation is typical in reliability assessment of marine systems.

The results given in the form of fuzzy numbers are imprecise. Thanks to that, they better illustrate the problem we try to solve. Looking at the fuzzy numbers shown above, we can draw conclusions useful in practice. We can conclude: it is most likely that the reliability of the pipeline reach value about 0.81 – 1. But we have to remember that the reliability can also reach the value about 0 – 0.19 what is much less likely. Fuzzy number gives us an idea about the uncertainty of estimation has been made. The consideration can be as follows: the reliability of the pipeline is about 0.81 – 1 with the possibility measure (not probability) 81%.

The Mamdani method is more pessimistic than the Larsen method. Thus applying the principle of worst-case, in the author's opinion the Mamdani method seems to be a good tool to solve a problem of reliability assessment, when available reliability data are very imprecise.

4. Methods of defuzzification

The result obtained in the form of fuzzy number gives the best view of the problem. However, it may be required to provide a result in the form of real number (crisp value). Then some methods of defuzzification can be helpful. Defuzzification is just a transformation from a fuzzy number to a crisp number. There are several methods of defuzzification described in the literature [4, 5, 6]. Some of them will be used to change the reliability calculation result given in the form of fuzzy number into crisp values. The application of those methods for the example being considerate in the article is given in Fig. 4. The results are compared in Tab. 1.

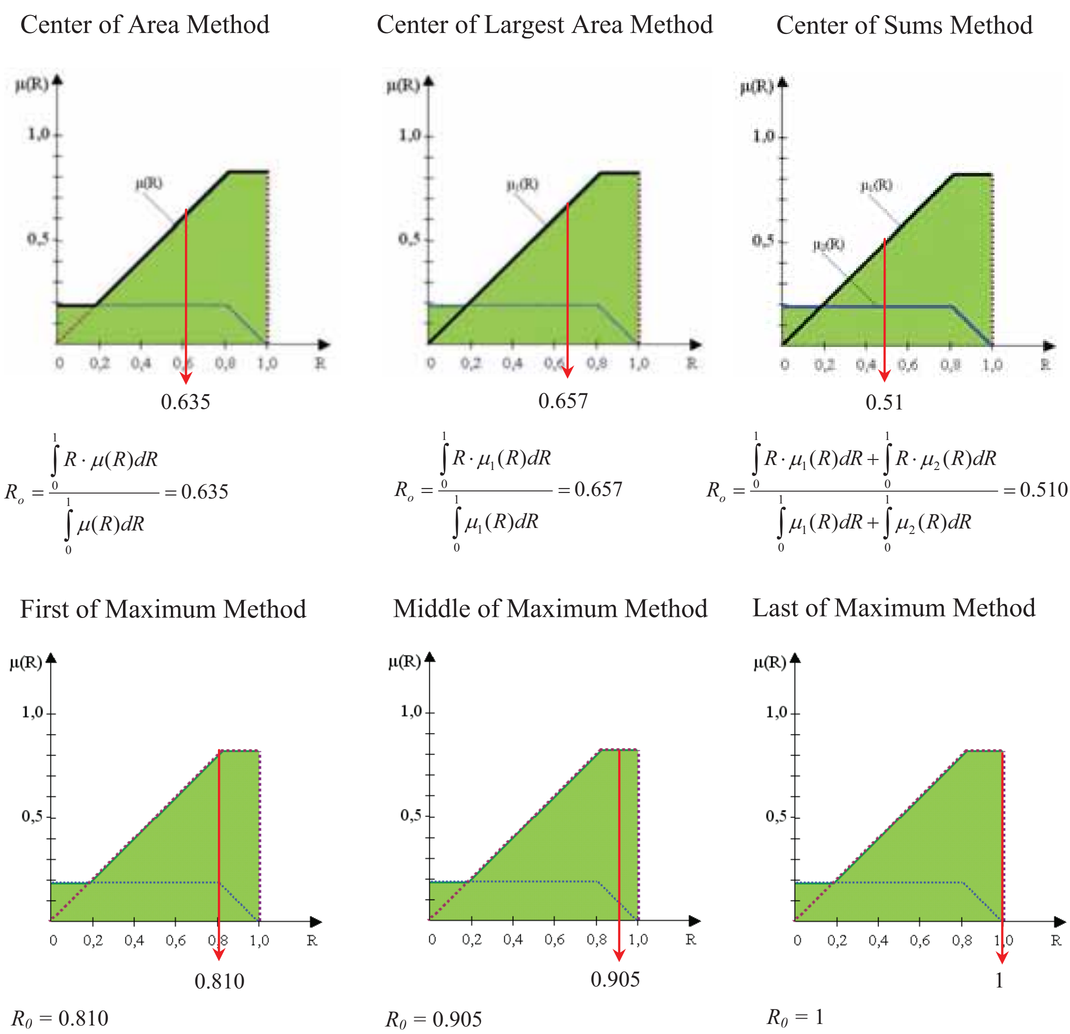


Fig. 4. Methods of defuzzification used to get a crisp value of reliability

Tab. 1. Comparison of reliability assessment results R_0 given in the form of crisp values, with the use of different methods of defuzzification

Defuzzification Method	R_0
Center of Sums	0.510
Center of Area	0.635
Center of Largest Area	0.657
First of Maximum	0.810
Tsukamoto	0.810
Middle of Maximum	0.905
Last of Maximum	1

5. Final remarks

In the author's opinion, it is possible to draw out useful in practice conclusions about reliability of technical items from imprecise data using fuzzy logic inference rules. Such an example has been shown in the article.

For the considered example - the results obtained with the use of the Mamdani method, the Larsen method and the Tsukamoto method are not contradictory. So it is hard to say, which of those methods the best is.

The Mamdani method is a little bit more pessimistic than the Larsen method. Thus applying the principle of worst-case it seems to be an appropriate tool to solve a problem of reliability assessment.

If someone needs the result of calculations in the crisp form, he can use the Tsukamoto method or use the methods of defuzzification of fuzzy number. But the results differ significantly depending on the chosen method, as it has been shown in Tab. 1.

The results given in the form of fuzzy numbers are imprecise, but they much better illustrate the problem of reliability assessment. Fuzzy number gives us an idea about the uncertainty of estimation has been made.

The best way to express the estimated reliability for the given above example is the conclusion: The reliability of the pipeline is about 0.81 – 1 with the possibility measure 81% (against to the conclusion: the reliability of the pipeline is about 0 – 0.19 with the possibility measure 19%).

With the very small data set it is very difficult to talk about probability in the classical sense, even from statistical point of view. We should rather to use fuzzy inference methods, as it has been shown in the article.

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HIGH TEMPERATURE DIAGNOSIS WITH INFRARED CMOS CAMERA

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Abstract

The original setup for high temperature measurements, equipped with a high-class pyrometer and standard CMOS camera was described in this paper. The principles of operation, basic rules and physical fundamentals of temperature measurement were shown and discussed. Chosen research experimental results were presented.

Keywords: *infrared radiation, temperature, CMOS camera, diagnosis*

1. Introduction

Many different kinds of technological processes are accompanied by the occurring of high temperatures. This can take place e.g.: in cutting and grinding - especially - superhard materials, welding processes, mould die techniques. Temperature rating is essential symptom for the proper conduct of the process state. For example; the temperature rise of the cutting edge can be a symptom of tool wear. Actually, there are various measuring techniques, ensuring the assessment process temperature, using both methods (contact or contactless). Particularly important are the non-contact methods of temperature assessment. These methods can be applied to both point measurements and the temperature distribution evaluating. Contactless measurement is based on the infrared radiation analysis, emitted by the surface at a specific temperature. The infrared radiation shall be adopted in the range: 0.76 ÷ 1000 μm, and for visible radiation: in the range 0.4 ÷ 0.76 μm. Any body with a temperature greater than absolute 0K emits electromagnetic radiation, including the infrared one. Spectral distribution of blackbody radiated energy (emitting best) describes the Planck's law [4]:

$$m_{cc,\lambda} = \frac{c_1 \lambda^{-5}}{e^{\frac{c_2}{\lambda T}} - 1}, \tag{1}$$

where:

$m_{cc,\lambda}$ - monochromatic emittance energy of a blackbody for the wavelength λ,

$c_1 = 3.7418 \times 10^{-16}$, W/m^2,

c_2 = 1.43879 x 10^2, mK,
T - temperature, K.

This law can be applied both for sources and receivers of radiation.

With increasing temperature of the radiating body, a maximum radiation intensity moves towards lower wavelengths - the law specifies Viena [4]:

$$\lambda_{\max} T = 2896\ \mu\text{mK}. \tag{2}$$

It refers to black and gray bodies. After integration, Planck's law for all wavelengths can be described by the Stefan-Boltzmann law [4], for total power radiated by a object at a temperature T:

$$W = ST_{cc}^4, \tag{3}$$

where:
S - Stefan-Boltzman constant: 5.67032 * 10^{-8} W/m^2-T,
T_{cc} - the absolute temperature blackbody K.

For real bodies the right takes form:

$$W = ST_{cr}^4. \tag{4}$$

The principle of operation of commonly used devices for contactless surface measurements is based on the use of previously described physical basis. Real bodies' infrared radiation, compared with ideal blackbody radiation is lower. This can be determined by emissivity coefficient:

$$\varepsilon_\lambda = \frac{W_{cc\lambda}}{W_{cr\lambda}}, \tag{5}$$

where:
$W_{cc\lambda}$ - ideal blackbody spectral density power radiation,
$W_{cr\lambda}$ - real-body spectral density power radiation.

The value of emissivity coefficient ε depend on temperature and surface conditions [8], e.g.:
- polished aluminum, at T = 50 ÷ 100 °C, ε = 0.04 to 0.06,
- galvanized sheet metal, at T = 50 °C, ε = 0.2,
- porous red brick, at T = 50 °C, ε = 0.9,
- crystal ice, at T =- 10 °C, ε = 0.98,
- polished copper, at T = 50 ÷ 100 °C, ε = 0.02.

In specially designed, non-contact measuring devices bolometrical infrared radiation detectors [10] are used most often. Determined, using them, temperatures are modified by the emissivity coefficient. Both infrared cameras and pyrometers (for the point measurements) work, basing on this principle [10].

Another technique, based on the CCD cameras use, [1,2,3] with no IR filter is also used. This technique allows the assessment of temperature distribution, basing on the relationship between the energy of infrared object surface radiation at a specific temperature and emissivity. Furthermore, there are dependencies between photometric and electric signal on CCD matrix [7]. Special patterns of black body radiation emission at a specific temperature and emissivity are mostly used for contactless devices for temperature measurements calibration [9]. CMOS camera application for surface temperature distribution test and experimental results are presented in this article.

2. Test stand

The research goal of this study was to evaluate the possibility of using a USB camera with a CMOS sensor, equipped with a filter, for the diagnosis of objects radiation at a visible range in high temperatures. Tests in the first stage consisted object pictures of known ceramic heater temperature. The heater temperature was adjusted by changing the voltage. Tests were performed in the voltage range from 100V to 230V. In addition, the setting– camera's exposure time was changed in range from 20% to 100%. Tests were repeated over the entire voltage of the heater. Original test stand was constructed - Fig. 1.

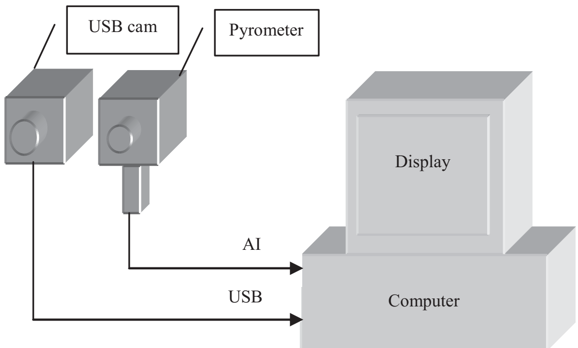

Fig. 1. Test stand diagram

Fig. 2. Minolta/Land Cyclops 152A pyrometer

USB camera was equipped with a visible light permeable filter for infrared radiation of wavelengths above 720 nm. Camera with a filter was directed at a heat source (electric heater). Heater supply voltage was controlled by the autotransformer at U = 100 ÷ 230V.

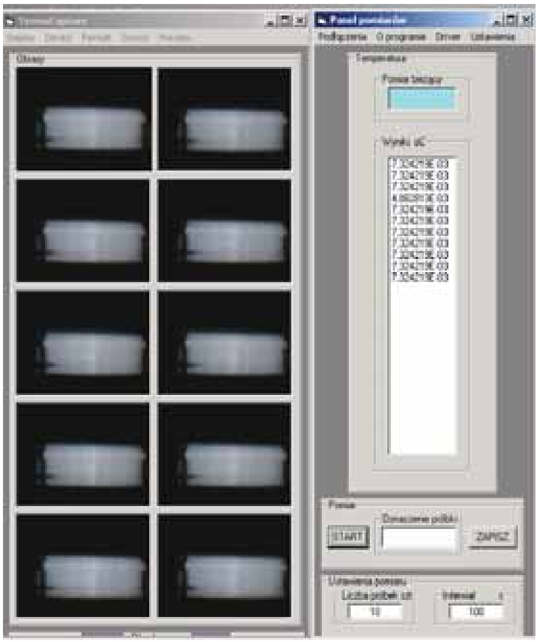

Fig. 3. 'TermoCapture' panel view

Heater temperature at a specific location was measured by Minolta / Land Cyclops 152A pyrometer (range: 550 ÷ 3300°C, spectral range 1 μm, measured area ϕ 5mm for 1m distance, accuracy: 0.25%, measuring modes: instantaneous value, maximum value, minimum value) – Fig. 2. Images from the camera were captured and recorded in grayscale, using original, made in VB6 program 'TermoCapture' – Fig. 3.

Designed program allowed determining of pixels brightness in the selected section of the image. Temperature measurements were conducted in parallel with the radiation sources images recording. Pyrometer, connected through an interface PCLD 8141 with analog input measurement card Advantech PCL-818L was used to record object temperature. VB object, associated with the program 'TermoCapture' was used for temperature measurements. Images were recorded for various camera sensitivity settings (25, 50, 75). Reduced camera resolution; 160 x 120 pixels and 60 s exposure time were used. The program allowed for capture object images capture at specified intervals (60 seconds was used).

3. Results and analysis

Heater temperature measurements results for different transformer settings are presented in the diagram - Fig. 4. Roughly linear relationship between voltage and heater temperature was found. Voltage 110 V was specified as a border value for further tests, due to the pyrometer measure range (550 ÷ 3200 °C).

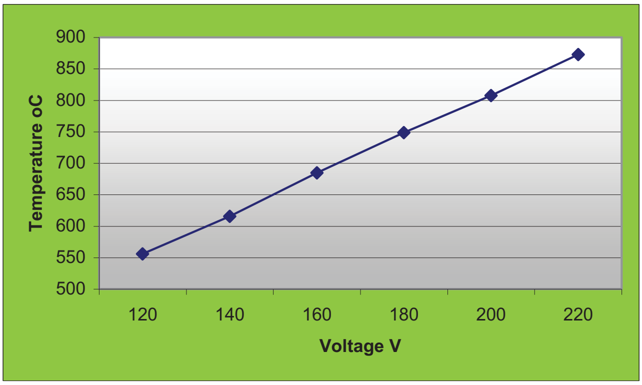

Fig. 4 The surface temperature and the heater voltage dependence graph

Sample, grayscale image of the heater surface obtained by the camera is shown in Fig. 5.

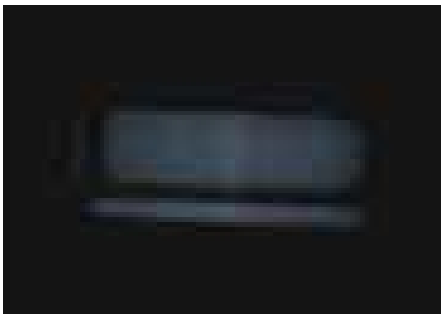

Fig. 5. Heater surface grayscale picture

Image data for measure and visualization of image points the brightness were sent to the graphics program – Fig.6. and Fig. 7. This technique allows for the surface temperatures spatial distribution, registered in the camera, visualization.

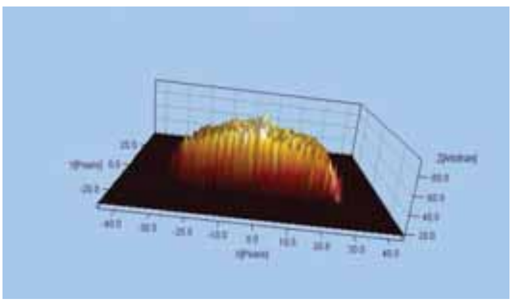

Fig. 6. Picture of the heater surface, processed by the graphic program, taking into account the pixel brightness (U = 180V, sensitivity 25%)

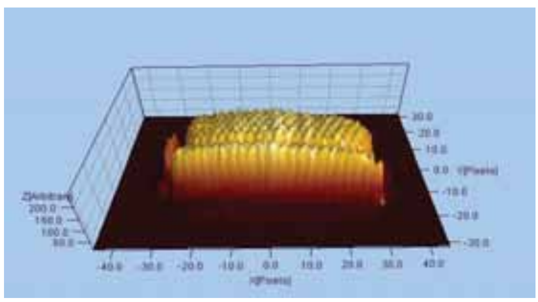

Fig. 7. Picture of the heater surface processed by the graphic program, taking into account the pixel brightness (U = 180V, sensitivity 75%)

Research results indicate a relationship between the object temperature and pixels brightness, recorded using USB camera with a CMOS sensor. At the same time, it was found that the registration process affects the pixels brightness of obtained image. Those results indicate a linear -

under certain test conditions - the relationship between the temperature of the object, and the brightness of pixels, recorded with infrared sensitive USB camera with a CMOS sensor - Fig. 8.

Linear dependence occurs in the range of pixels brightness from 50 to 220. At the same time, it was found that the camera settings (exposure time) affects the image pixels. This indicates the possibility of controlling the camera's sensitivity and allows determination of camera calibration relationships.

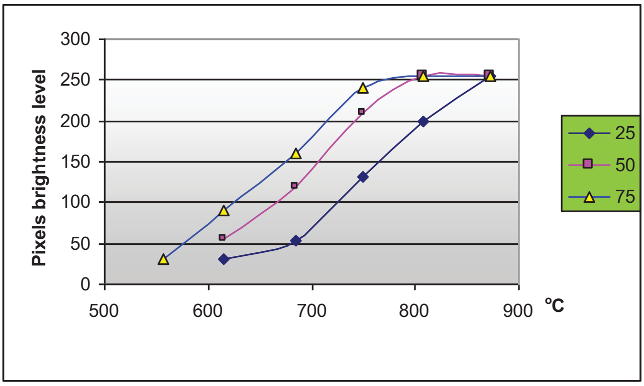

Fig. 8. Surface temperature dependence on image pixels brightness for various sensitivity levels: 25, 50 and 75%

Radiation detection system was saturated at higher temperatures. For this (upper) temperature range smaller exposure set-up time should be used. This indicates the possibility of extending the scope of the camera to determine the temperature of the object by using the maximum exposure time for reduced temperatures ranging from 550°C and the use of reduced exposure time to the possibility of temperature readings using a camera for higher temperatures - Fig. 8. This chart indicates that the effect of exposure time is nonlinear.

The test stand calibration requires further work, but the result of previous work indicates the possibility of using this system for diagnosing the temperature distribution in range above 550°C. It is possible to observe and record objects pictures in visible light by removing filter.

4. Conclusions

The following conclusions were formulated, on the basis of literature and the experimental results:

- ✓ camera with infrared sensitive CMOS sensor, with visible light filter, is useful for visualizing the distribution of surface temperatures in the range above 550°C,
- ✓ range of visible light filter coincides with the pyrometer used for tests,
- ✓ CMOS cameras with visible light filter are suitable for field temperature testing at a specified emissivity,
- ✓ temperatures in the range from 550°C to 880°C were recorded with various camera settings under tests conditions,
- ✓ the camera exposure time can be used to change the camera measuring range, reducing the exposure time allows the evaluation of the surface at higher temperatures,
- ✓ it is advisable to conduct further research on the developed method in the field of different applications, including the possibility of cutting process temperature measurement [5],

- ✓ it is appropriate to develop an image analysis procedure for the direct determination of the analyzed surface temperature,
- ✓ an important advantage of the described diagnostic method for high temperatures is the ability to use one camera to observe the test object in visible light (visible radiation without a filter) and in the infrared range.

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ANALYSIS OF RELATIONS BETWEEN THE COMPRESSION RING CHARACTERISTIC PARAMETERS

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Abstract

A proper design of compression ring secures its correct and long term operation. A good ring contact to cylinder wall along the whole circumference with the required distribution of circumferential pressure at the same time are symptoms of this correctness. The analytical methods and more often numerical ones are applied when designing piston rings. A characteristic parameter most often designated as K, which facilitates the comparison of different ring designs and allows for anticipation of its elastic properties is used at the stage of ring design. The following study presents the most significant mathematical relations between the ring geometry and forces that are acting on ring, and shows that results of force operation could differ relative to the point of their application. Relations between the tangential force and the circumferential one have been established as well. For three compression rings verifying tests consisting in definition of selected parameters using analytical and numerical methods have been carried out. The analysis of attained results and trials on explanation of noticed discrepancies are presented in the study as well.

Keywords*: marine combustion engine, piston ring, oil film, ring wall pressure*

1. Introduction

The compression ring should touch the cylinder wall with all its circumference and move over a layer of lubricating oil in order to serve its purpose, i.e. to keep the combustion chamber tight, to transfer the piston heat and to distribute lubricating oil. According to the hydrodynamic theory of lubrication the formation of oil layer separating working surfaces of ring and cylinder requires a selection of adequate wall pressure as well as micro- and macrogeometry of collaborating surfaces. It is generally considered that thanks to suitable selection of these parameters a period of ring reliable operation could be significantly extended. Other measures helpful in extension of ring life are: application of ring face cover with chromium-ceramic covers or appropriate formation of collaborating surface, e.g. deep honing of cylinder liner or chrome plated ring grooves on piston.

2. Characteristic parameters of compression ring

Characteristic parameters of ring geometry (see Fig. 1) and of ring material (represented by e.g. the modulus of elasticity) are used for definition of ring design. In case of modern compression rings the range of those values is very wide.

For example, the diameters of contemporary engine piston rings could range from dozen millimeters or so to more than meter while the axial width – from a fraction to a few dozens millimeters.

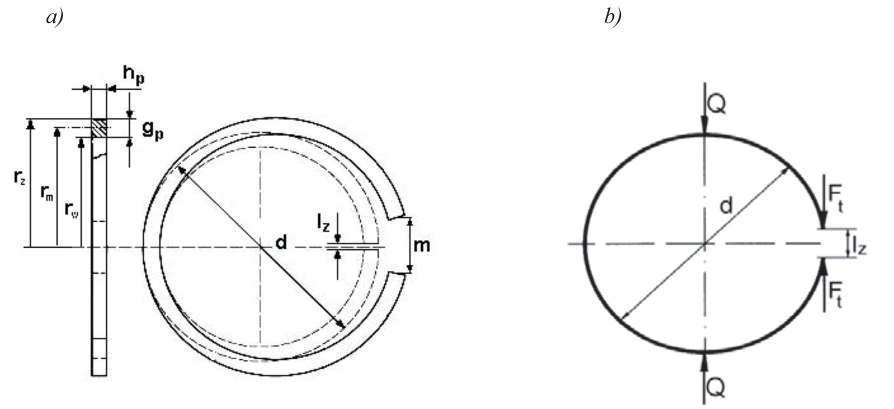


Fig. 1. Sketch of free and clamped compression ring (a) and a ring loaded with tangential F_t and radial Q force (b)

The efforts on definition of an optimum design of compression ring, in particular on relations between ring geometry and its elastic properties have been carried out for years. The most considerable progress was done in the 40-ties of the last century, when so called ring free form was established using analytical formulas. A parameter that could facilitate a comparison of various rings was searched for as well. It was accepted that the quantity further called a ring characteristic parameter and given by the following formula

$$K = 12\frac{p_o}{E}\left(\frac{r_m}{g_p}\right)^3, \tag{1}$$

satisfies these demands. In this formula r_m denotes a radius of neutral layer while p_o is a constant circumferential pressure (resulting from ring installation in liner).

The way the radius of neutral layer r_m was determined needs explanation. If the rectangular ring outer radius was denoted by r_z (see Fig.1a) and the inner radius by r_w then the radius of neutral layer is given by the formula:

$$r_m = \frac{r_z - r_w}{\ln\frac{r_z}{r_w}} = \frac{g_p}{\ln\frac{r_z}{r_w}} = \frac{g_p}{\ln\frac{d}{d-2g_p}}. \tag{2}$$

For majority of piston ring calculations the simplified form of the Eq. (2) reduced to

$$r_m = 0.5\cdot\left(d - g_p\right) \tag{3}$$

is used which means that the neutral layer agrees with the cross-section center of gravity. Eq. (3) could be obtained as a result of the expansion of function „ln (r_z/r_w)" in an exponent series taking into consideration only the first term of this expansion.

Using a well-known relation between the tangential force F_t (this force acts at the ring gap tangentially to the neutral layer) and the circumferential wall pressure p_o ([9])

$$p_o = \frac{F_t}{r_m \cdot h_p} \tag{4}$$

one can obtain other forms of the Eq. (1) (more important ones are presented in Table 1).

Tab. 1. Formulas for calculations of the ring characteristic parameter K

1	2	3	4	5
$K = 12\frac{p_o}{E}\left(\frac{r_m}{g_p}\right)^3$	$K = \frac{p_o \cdot h_p \cdot r_m^3}{E \cdot I}$	$K = \frac{F_t \cdot r_m^2}{E \cdot I}$	$K = \frac{3 \cdot (d - g_p)}{h_p \cdot g_p^3}\frac{F_t}{E}$	$K = \frac{m}{3 \cdot \pi \cdot r_m}$
Terms as in Fig. 1, another ones: I – moment of inertia, given by the formula $I = h_p \cdot g_p^3 / 12$ for a ring of rectangular cross-section				

As verifying calculations show for a compression ring the parameter K takes the value within the range from 0.01 to 0.05, independently on dimensions and material properties.

When trying to define the properties of ring which data are unknown, the formula written in column 5 of the Table 1 linking the K parameter with the clearance of ring gap might be useful (see dimension m in Fig. 1a). To define this one should assume that the ring is a curved rod of satisfactorily big radius r_m relatively to the radial wall thickness g_p. The change in ring gap, i.e. the displacement of ring free ends, could be defined as the derivative of rod potential energy V relative to the force P (according to the Castigliano's theorem):

$$f_y = \frac{\partial V}{\partial P}. \tag{5}$$

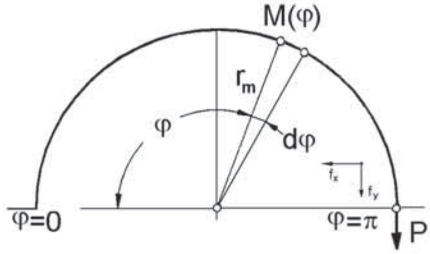


Fig. 2. A schematic draw used for determination of a displacement of ring ends loaded by the force P

An increase in potential energy dV caused by bending moment M(φ) along the increase of angle dφ equals (as very small, other forces and moments acting upon the ring are omitted):

$$dV = \frac{M^2(\varphi) \cdot r_m}{2 \cdot E \cdot I}. \tag{6}$$

A potential energy contained in ring within the section defined by the angle $(0 - \varphi_1)$ equals:

$$V = \frac{P^2 \cdot r_m^2}{2 \cdot E \cdot I} \int_0^{\varphi_1} (1 + \cos\varphi)^2 d\varphi , \tag{7}$$

and the total displacement of a point subjected to the force operation searched for is:

$$f_y = \frac{P \cdot r_m^3}{E \cdot I} \int_0^{\pi} (1 + \cos\varphi)^2 d\varphi . \tag{8}$$

Remembering that the total size of clearance at ring gap is $2 \cdot f_y$, a formula linking the K parameter of ring with the total clearance m has been obtained:

$$K = \frac{m}{3 \cdot \pi \cdot r_m} = \frac{2 \cdot m}{3 \cdot \pi \cdot (d - g_p)} . \tag{9}$$

For a known geometry of ring (and using the formulas presented in Table 1) one can estimate a hypothetic value of ring pressure p_o or tangential force F_t on purely computational way (however the knowledge on properties of ring material given by the Young modulus E is indispensable).

2. Relations linking the tangential and radial force

To determine the ring elastic properties the measuring devices of different construction are being used (e.g. those presented in [8]) which allow to measure a value of tangential F_t or radial Q force. Knowing one of the forces the another one can be determined according to the formula $Q = \kappa \cdot F_t$ (literature gives various values of κ within the range from 2 to 3). In order to assume the factor suitable for certain measurements one should take into consideration the force location and the results of force operation as well. During measurements the force F_t can load the ring evenly (by the clamping band – see Fig. 3a) or on the ring end at the point situated on its neutral layer (Fig. 3b). The value of force should be selected so as to bring about the ring ends as near as gap clearance l_z. On some measuring devices the direction of tangential force is moved from the neutral layer to the ring outer surface (as in Fig. 3a – the force is marked as $F_{t,d}$) which affects the measured value. The following relation takes place between the forces mentioned:

$$F_t = \frac{F_{t,d}}{2} \frac{2d - g_p}{d - g_p} . \tag{10}$$

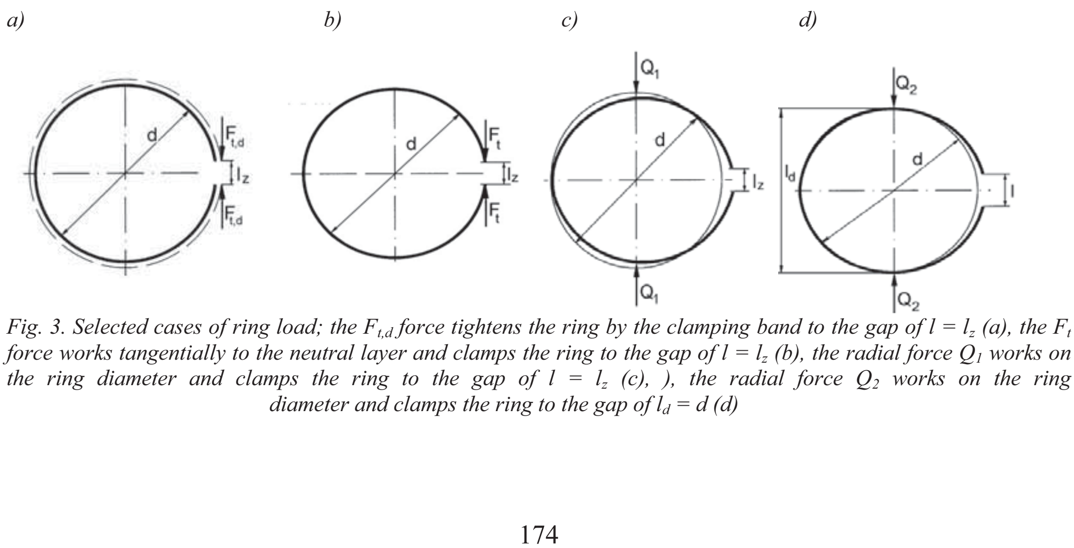


Fig. 3. Selected cases of ring load; the $F_{t,d}$ force tightens the ring by the clamping band to the gap of $l = l_z$ (a), the F_t force works tangentially to the neutral layer and clamps the ring to the gap of $l = l_z$ (b), the radial force Q_1 works on the ring diameter and clamps the ring to the gap of $l = l_z$ (c),), the radial force Q_2 works on the ring diameter and clamps the ring to the gap of $l_d = d$ (d)

On the other hand the radial force should be selected so as to make ring ends come closer to the distance of l_z (the Q_1 force in Fig. 3c) or cause a partial ring closure to the value of $l_d = d$ (the Q_2 force in Fig. 3d). Moreover, it should be mentioned that ring deformation occurs in all cases presented (except the use of clamping band), which additionally affects the variability of the κ factor.

In order to avoid an erroneous selection of the κ parameter the author took a trial to verify its value and variability dependently on analyzed case. A verification of the factor κ_1 value linking the F_t and Q_1 forces (for a case presented in Fig. 3c) was carried out at the beginning.

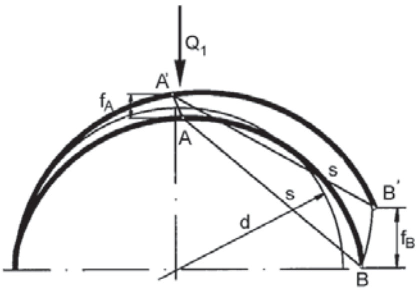


Fig. 4. An auxiliary sketch for determination of the κ_1 parameter

According to the sketch in Fig. 4 the load of force Q_1 brings about a displacement of point A' to A (f_A is the vertical component of this section) and of point B' to B (its vertical component is f_B) at the same time. The length of section s does not change (for simplicity of sketch the l_z slit was not marked because is far shorter than the f_B section) because the right hand side of ring is not subjected to load. Remembering that the same displacement f_B should be brought about by the tangential force F_t acting at the ring gap, a relation linking both forces has been established (relation between the force F_t and displacement f_B was given in [3], for example).

$$\frac{\pi \cdot Q_1 \cdot r_m^3}{4 \cdot E \cdot I} + r_m \left[1 - \cos\left(\frac{Q_1 \cdot r_m^2}{E \cdot I} \right) + \sin\left(\frac{Q_1 \cdot r_m^2}{E \cdot I} \right) \right] = \frac{3 \cdot \pi \cdot F_t \cdot r_m^3}{2 \cdot E \cdot I}. \tag{11}$$

As a solution of (11) the following relation has been obtained:

$$\kappa_1 = \frac{Q_1}{F_t} \approx \frac{6\pi}{4+\pi} = 2.639. \tag{12}$$

Almost identical value, i.e. 2,632 has been given in [4].

The Goetze Company gives other values applied when defining relation between F_t and Q_2 [5]. The value of radial force Q_2 should be big enough to make the ring ends come to the distance of $l_d = d$ (measured on the diameter of ring in cylinder which corresponds to the case presented in Fig. 3d). The relation describing this situation, determined experimentally, has the following form

$$\kappa_2 = \frac{Q_2}{F_t} = 2.2667 \left[1 - 0.33 \cdot K - 7.8 \frac{u}{d} - 24 \left(\frac{g_p}{d} \right)^2 \right], \tag{13}$$

while the value of κ_2 factor should lie within the range from 2.05 to 2.30.

The form of Eq. (13) shows that the initial value of relation Q_2/F_t equal to 2.2667 is consecutively reduced by the individual elements of the formula more the higher is the value of factor K, oval deformation u and radial wall thickness g_p. Test computations carried out for a group of rings proved that the relative decrease in this value could reach 0.2 which explains reasons why the range of variability for this factor was assumed as it was presented earlier.

The author decided to perform the check tests of this factor (using analytical formulas and mathematical model of ring).

Expanding ring causes the displacement of point P (visible at an angle of φ from the start of coordinate system – see Fig. 5) to the point P which corresponds to the Δx and Δy displacements relative to the axes.

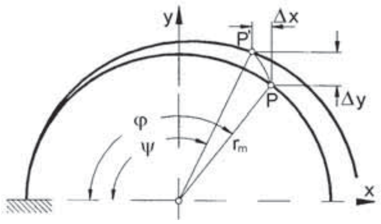


Fig. 5. Supplement sketch for definition of ring free form

Formulas presented in [1] in the form shown below are recognized as one of the best descriptions of the free ring neutral layer, expressing the location of its points relative to axes X and Y:

$$\frac{x}{r_m} = \cos\varphi + K(\varphi\cdot\sin\varphi + \frac{\sin^2\varphi}{2} + \cos\varphi - 1)\,, \tag{14a}$$

$$\frac{y}{r_m} = \sin\varphi + K\cdot(\frac{\varphi}{2} + \sin\varphi - \varphi\cdot\cos\varphi - \frac{\sin\varphi\cdot\cos\varphi}{2})\,, \tag{14b}$$

where K is the characteristic parameter of ring discussed earlier while x/r_m and y/r_m are the relative coordinates of free ring. It results from the Eq. (14) that for the assumptions made initially the form of free ring depends only on the K parameter which means that the rings of the same value of K parameter have the same relative course of the neutral layer. It should be noted here that the set of equations presented below (notation as in Fig. 5) is regarded as the most accurate description of ring free form obtained analytically.

$$\begin{aligned}\frac{dx}{d\varphi} &= \sin(\varphi - K\cdot\varphi - K\cdot\sin\varphi),\\ \frac{dy}{d\varphi} &= \cos(\varphi - K\cdot\varphi - K\cdot\sin\varphi).\end{aligned} \tag{15}$$

Information on the way the Eq. (15) was obtained, methods of its solution and perspectives of implementation when constructing mathematical models of piston ring one may find in [1, 2].

Comparing the ring displacement caused by the force Q_2 with the displacement determined according to the Eq. (14b) one obtains for the angle of $\varphi = \pi/2$

$$r_m \cdot K \cdot \left(1+\frac{\pi}{4}\right)=\frac{\pi \cdot r_m^3 \cdot Q_2}{4 \cdot E \cdot I}, \tag{16}$$

which leads to

$$\kappa_2=\frac{Q_2}{F_t} \approx \frac{\pi+4}{\pi}=2.273 \tag{17}$$

after suitable transformations. It is value close to the one given by the formula (13).

3. Application of numerical method to verification of the κ parameter and description of compression ring shape

A practical implementation of the analytical relations presented in chapter 2 requires a fulfillment of many conditions. For instance, it is being assumed that the ring is installed in an ideally round cylinder and touches to the liner with its entire circumference, and the ring wall pressure is always even. Analytical relations allow to take into consideration changeability of many quantities related to ring geometry and to the material used as well. The numerical methods are free of such limitations. In literature, also in domestic one there are descriptions of numerous methods that facilitate to design the ring of any form and pressure distribution with any accuracy. A method of ring elastic pressure distribution was presented by A. Iskra [1]. The mathematical model developed by the author on the basis of the above method can be found in [6].

A fragment of the model verification process was shown in [9]. This consists in a comparison of computational results accomplished by analytical and numerical methods (implemented to a mathematical model of ring). The comparative analyses embraced among other the magnitude of ring ends displacement resulted from loading forces. The tests concerned rings of three dimensional categories, namely of the automotive engine (the 170A.000 type), of a bulldozer (the DTI-817C type), and the marine one (the L48/60CR type). The ring characteristic parameters were measured by the author or were obtained from related catalogues (see Table 2).

Using a mathematical model of piston ring, the forces extorting the displacement of ring sections (according to the description in Fig. 3) as well as the κ_1 and κ_2 parameters were calculated and the obtained results were compared with the results of analytical calculations (it was assumed that the latter were accurate).

Tab. 2. Technical data of exemplary IC engine compression rings

Quantity		Ring 1 (automotive engine)	Ring 2 (engine of bulldozer)	Ring 3 (marine engine)
cylinder diameter d	[m]	0.08	0.136	0.480
ring neutral radius r_m	[m]	0.0382	0.0655	0.232
axial height h_p	[m]	0.0014	0.003	0.015
radial thickness g_p	[m]	0.0034	0.005	0.016
gap clearance m	[mm]	9.83	14.4	49.0
Young modulus E	[Pa]	$115 \cdot 10^9$	$112 \cdot 10^9$	$105 \cdot 10^9$
mean pressure p_o	[MPa]	0.180	0.095	0.063
tangential force F_t	[N]	9.60	18.6	219
stiffness EI	[Nm^2]	0.527	3.5	537.6
parameter K	[–]	0.0266	0.0229	0.0220

Tab. 3. Summary of calculation results obtained analytically and numerically for selected quantities

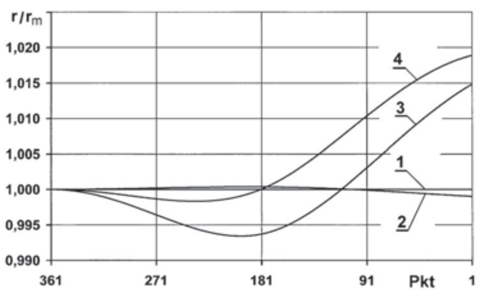
Parameter	Formula	Ring 1			Ring 2			Ring 3		
		Results A	Results N	δ [%]	Results A	Results N	δ [%]	Results A	Results N	δ [%]
Q_1			25.7			49.7			583	
Q_2			22.1			42.5			503	
χ_1		2.639	2.677	1.439	2.639	2.672	1.251	2.639	2.657	0.682
χ_2		2.273	2.302	1.276	2.273	2.284	0.484	2.273	2.296	1.011

As it outcomes from the results summarized in Table 3, there is a high agreement between parameters obtained with analytical (A) and numerical (N) methods, because the relative differences do not exceed 2%. The condition valid for analytical calculations (about keeping the round form by ring) was not fulfilled in numerical calculations which seems to be a probable reason for these differences.

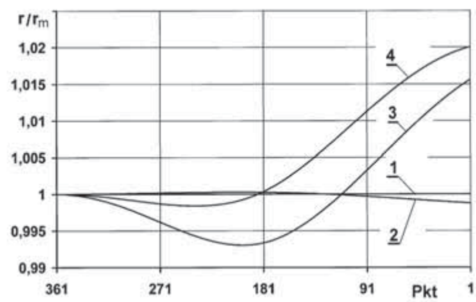
4. Evaluation of ring deformation under the load of external forces

The shape of ring loaded with the forces F_t, and Q_1 and Q_2 differs considerably from the circle. Fig. 6 shows the courses of circumferential relative change in ring radius – calculated analytically – for selected load cases (cases summarized in Fig. 3). The ring deformations compared with the case of even load could reach several percent depending on load case and the highest ones are those for the load of Q_2 force (line 4). It should be emphasized that despite similar shape the courses obtained for various rings differ one from another (Fig. 7). It means that individual characteristic course should be determined for each ring.

a)



b)



c)

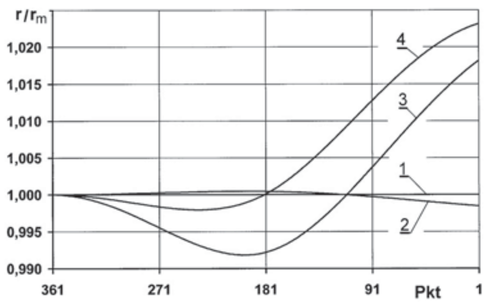


Fig. 6. Courses of the circumferential relative change in ring radius for selected load cases on following engines: a – 170A.000 , b – DTI-817C, c – L48/60CR; 1 – constant load, 2 – loaded with the F_t force, 3 – loaded with the Q_1 force, 4 – loaded with the Q_2 force

a) b)

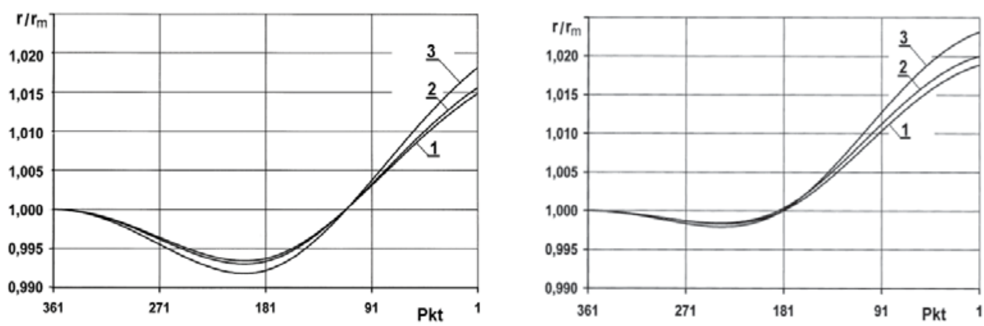


Fig. 7. Courses of the circumferential relative change in ring radius loaded with forces Q1 (a) and Q2 (b) for the rings of following engines: 1 – 170A.000, 2 – DTI-817C, 3 – L48/60CR

Presented analyses and defined relations concern above all the situations relative to the tests of compression rings outside engine but they can be also useful during ring design process and analyses of its behaviour when moving on a running engine.

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LUBRICATION OF ROLLING BEARING PAIRS IN ENVIRONMENTAL ASPECT

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Abstract

Lubrication of bearing pairs is the problem often underestimated. Usually it is discussed in aspect of operational features stability of friction pair. In fact, it fulfils much more functions and it should be considered in wider aspect. In this paper environmental problems of rolling bearings lubrication were discussed.

In analyzed environmental aspects of lubrication, the following problems were discussed: interaction of lubricants (also with additives) and rolling bearing elements, influence of a lubricant on surrounding, acting of lubricating medium on bearing elements, interaction of lubricants' components, ecological and environmental functions of bearing's sealing.

Conducted analysis were illustrated by examples taken from authors' own investigations and from literature.

Keywords: *rolling bearing, lubrication, environment, fatigue life*

1. Introduction

Lubrication of the rolling bearing pairs is the problem often underestimated in the stage of design. During operating it is considered the most often in aspect of stability of usable features of pairs. In substance it fulfils many functions, therefore in both mentioned stages (design and operation) of technical object life one should consider it in wide aspect and take a lot of factors into account.

The main goal of presented in this paper investigations was collect information concerning environmental conditionings of lubricating of bearing pairs - especially rolling ones. In carried out analysis the seals, as elements directly connected with lubrication and environment of work, were also considered. Collected information can be helpful in processes of design and operation of machines with rolling kinematic pairs in their structures.

2. Influence of lubricating medium on rolling bearings elements

Basic tasks of lubricants applied in friction's (both: rolling, as well as sliding) kinematic pairs are:

- decreasing friction coefficient value and thereby – decreasing the wear process intensity,
- heat abstraction generated as result of friction work,
- carrying away waste products from friction zone and eliminating them from circulation in

result of filtration,

- corrosion protection of co-operating elements,
- vibration dumping.

Depending on kind, form and amount of lubricants, degree of realization of above mentioned assignments can be different.

Chemical compounds, with atoms of metals or organic functional groups in structure, perform essential function as additions modifying features of lubricants, among others minimizing results of friction process. In lubricating technique alkilotiophosphate, as well as alkilotiocarbamate of metals, as e.g.: zinc (ZnDTP or ZnDTC), antimony (SbDTP or SbDTC) molybdenum (MoDTP or MoDTC) are commonly used. Their presence as lubricant additions favours formation of antiwear superficial layers. Building of this layers occurs under the influence of different forms of energy, accompanying friction processes [6].

Properties and features of formed layers are functions of mutual relations between metal of foundation and atoms of basic metal of grease or addition [2, 8]. During friction process, as the results of these interactions and activation of additions, antiwear layers are generated in the forms – Fig.1:

- metalic (Me),
- inorganic compounds (MeOx)
- macromolecular compounds [-L-L-]n .

The structure of generated layer has one-, two- or three-phase character.

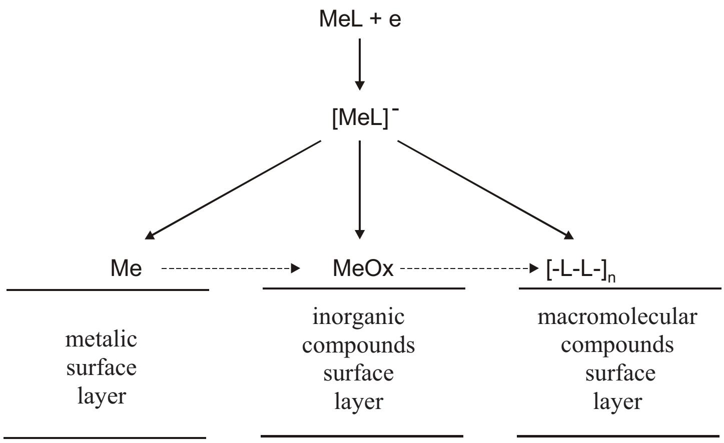

Fig. 1. Generalized mechanisms of changes of tiometalorganic compounds in the process of antiwear surface layers formation [4]

According to different chemical activity of base elements (Me), as well as reactivity of chemical compounds from DTP and DTC groups, antiwear properties of generated layers are diverse. It is affirmed, that they are better for compounds of type MeDTP than for MeDTC. Moreover, all of them in essential degree diminish intensity of wear process - Fig. 2. Conducted experimental investigations were affirmed also, that antiwear layers have reproducible character [4, 5].

Quoted above results of investigations mainly concern conditions of sliding friction. Because slides always accompany to rolling friction, so sliding friction results obtained in presented investigations should indirectly concern phenomena in rolling bearing pairs too. The confirmation of above mentioned assumption may be the fact that in kinematic pairs with non-conformal (concentrated) contact of elements, such asthe rolling bearings, diverse influence of components of lubricants onto material of foundation was also observed. It was affirmed, that lubricating of molybdenic steel by means of grease with sulphuric addition clearly reduced intensity of wear process in comparison to lubricating without such addition [3]. Different action of grease additions was observed in the case of chromium steels, lubricating by grease with chloric additive. In this materials association enlargement of intensity of wear process was affirmed.

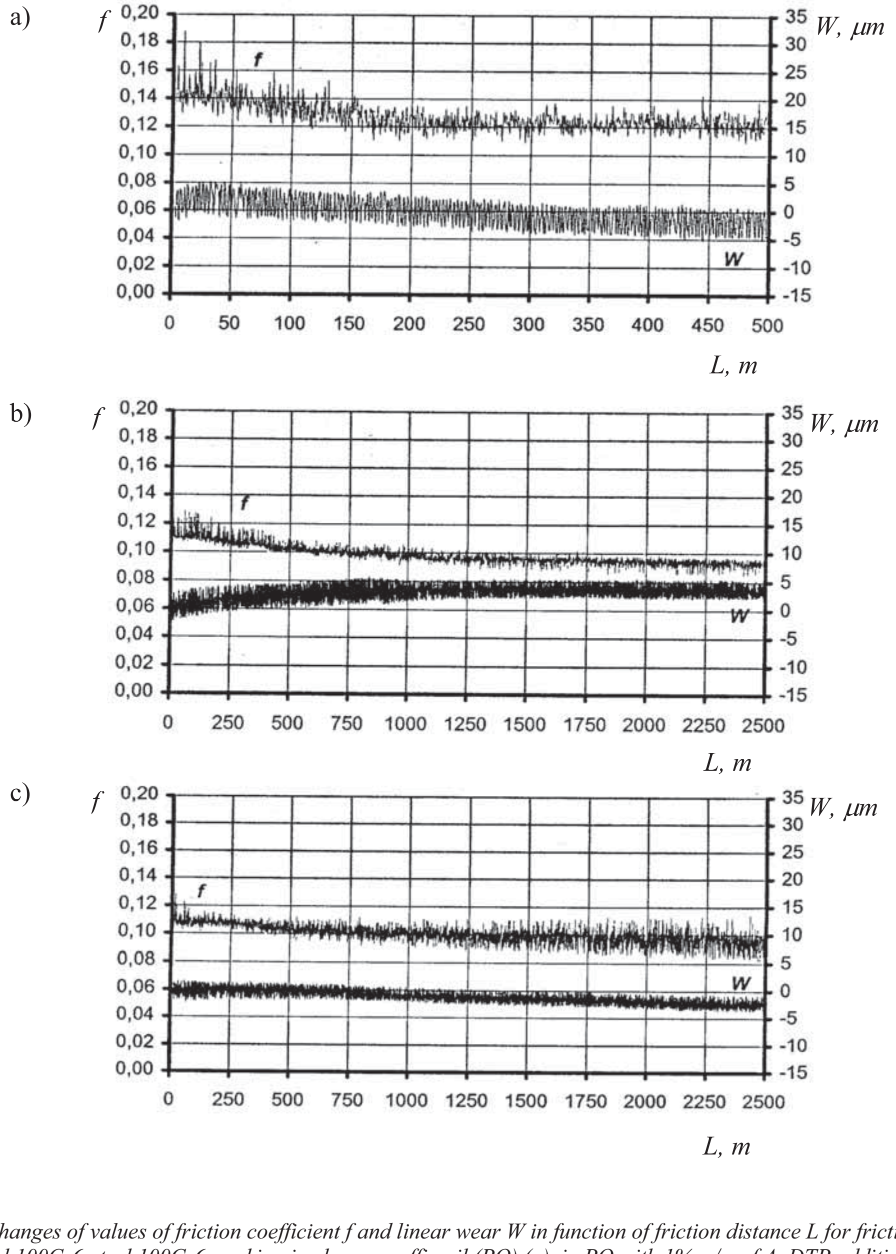

Fig.2. Changes of values of friction coefficient f and linear wear W in function of friction distance L for friction's pair: steel 100Cr6-steel 100Cr6 working in clean paraffin oil (PO) (a), in PO with 1% w/w of AuDTP addition (b) as well as in PO with 1% w/w of ZnDTP addition (c) [4]

The causes of diverse influences of lubricants, as well as their additions onto elements of bearing pairs are different tribochemical reactions between them and elements or phase components of co-operating parts of kinematic pairs. In first above described case, in result of reaction, layer MoS_2 was created, thereby substance which minimizes friction, in second meanwhile – came into being chemical compound $CrCl_3$, accelerating corrosive processes [9].

3. Influence of lubricating medium on surrounding environment

Influence of lubricating medium on environment in which bearing's pair works, the most often has unfavorable character. First of all it results from chemical constitution of lubricants. As the mixtures of hydrocarbons, produced from rock-oil products (mineral greases) or synthetically, they penetrate into environment and cause its pollution and, in further consequence – its degradation. The actions aiming at minimization of this undesirable phenomenon are two-way:

- limitation of quantity of applied lubricant,
- applying of biodegradable lubricants.

In the first case one use special systems of lubricating, e.g. oil fog or so-called minimum lubricating [7]. In both examples amount of lubricants, potential hazard for environment, is smaller than for traditional lubricating, e.g.: immersion system. Moreover, one should to notice, that decrease of amount of oil simultaneously causes smaller loss of power in rolling bearing – Fig. 3.

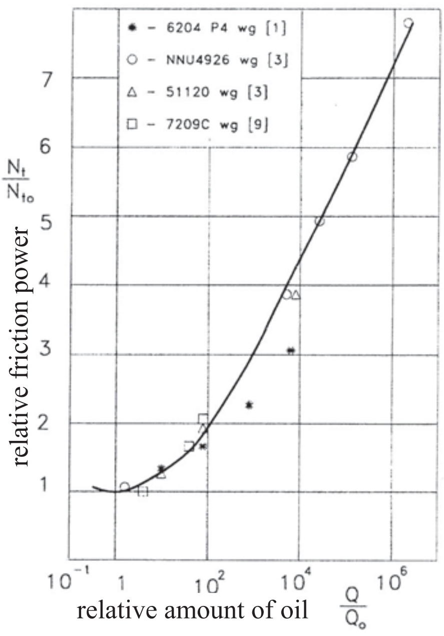

Fig. 3. Changes of friction power N_t in rolling bearings in the function of oil flow Q for different constructional types of rolling bearings (N_{to} – friction power of optimum lubrication, Q_o – optimum amount of oil) [7]

Thus, limiting quantity of lubricants one can observe positive results both, in ecological sphere, as well as – in economic. From investigations, which results are presented on quoted graph appears, that losses of power are diverse and in essential degree they depend on kind of bearing (ball or roller, transverse, angular or axial etc).

Well-known are also the results of investigations showing that resistances of movement in rolling bearings depend on form of lubricant (oil, plastic or solid grease), e.g. [10]. It results directly from different viscosity of lubricants in dependence on their consistence.

Because in practice it is impossible fully eliminate any contact of lubricants with surrounding, therefore biodegradable lubricants are applied [1]. Under influence of natural environmental factors they have ability to biodegradation onto environmentally harmless components. This feature concerns whole lubricant or some of their fractions; concerns lubricant additives too.

Mentioned above activities minimize the negative influence of lubricants on farther surrounding of machines.

4. Influence of environment on elements of bearing

Thin layer of grease, covering elements of rolling bearing, causes situation that environment of work of bearing pairs has not direct contact with co-operating elements. It has very essential importance if environment is chemically aggressive, e.g. sea surroundings, because it minimizes results of chemical corrosion. It limits also destruction of surface induced by electrochemical corrosion.

In the case of chemically neutral environment, but polluted: including dusts, moisture etc. (e.g. in aggregate or other mineral materials mines), presence of lubricants limits also unfavorable influence of farther environment. It is thanks to fact, that dirt or moisture are chemical or physical bonded with oil or grease, and carried away from zone of friction. This way it causes decreasing intensity of wear process, and the same - increasing durability of rolling bearings.

5. Environmental function of seal

The seal of rolling bearing pair is essential element in assurance of its correct functioning. From surrounding reach the pair uncontrolled streams of energy and matter, which as disturbance of controlled parameters of work, generate answer of tribologic system, not always possible to predicting. Therefore, the basic task of bearing pair seal is to minimize surrounding influence on conditions of pair's work. Among environmental function of seal, the essential are following:

- protection against penetration of dirt from environment to lubricant and further – to contact zone of co-operating elements,
- limitation of possibility of lubricant decrement from contact area of rolling elements with raceways,
- making impossible or considerably impeding access of work's environment of bearing to contact area of bearing's elements.

Realization of above mentioned functions makes possible reaching by pair constructively assumed operational parameters, at expected reliability and durability of machine as the whole. Analysis of mentioned tasks of seal shows its importance in environmental aspect. Moreover, it is possible to notice, that above presented functions refer both: closer (zone of contact of elements of bearing) and farther (surrounding of bearing pair) environment.

Very important and desirable feature of seal is its constancy in time. It assures steady conditions of bearing pair work and thus - operating of machine in stabilized conditions, and the same way – machine long life. Registered and described case of thrust (axial) bearing of joint of bus body in serviceability [9] can be representative example of results of lack or insufficient fulfilling by seal mentioned above environmental assignments.

Fractographic analysis of bearing's elements working surfaces showed presence on them numerous decreases - Fig. 4. Initially it was assumed that their nature was corrosive and it resulted from loss of effectiveness of sealing of bearing's pair. In order to verification this assumption necessary investigations and analyses were realized.

The surface of one of the ball after a few hundred hours of bearing operating is shown in Fig. 5. It has very developed sculpture, with number of tops and cavings in its structure.

On the ground of chemical analyses of applied plastic grease, one shows that presence of grease and its reaction with farther environment was a favourable circumstance of destruction of surfaces of bearing's elements. It was affirmed that it was synthetic grease with silicon densifier.

Spectroscopic analysis of lubricant indicates presence of polar groups type: Cl^-, HPO_4^{2-}, $H_2PO_4^{3-}$, SiO_3^{2-} in its constitution.

a) b)

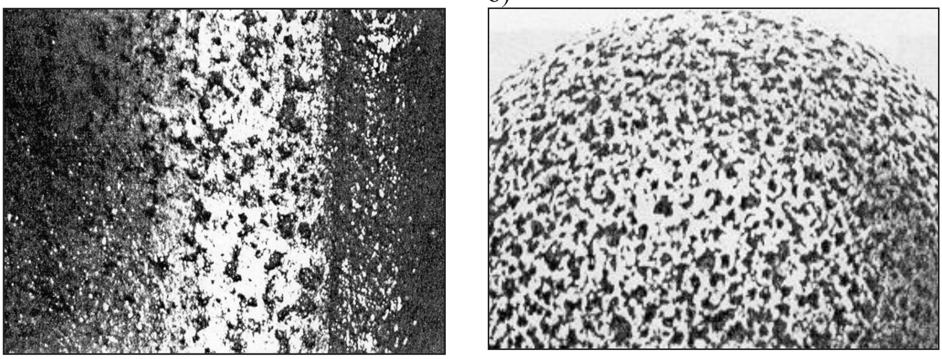

Fig. 4. Macroscopic views of working surfaces of tested bearing with visible characteristic defects on: a) raceway on internal ring, b) ball (mult. 5x)

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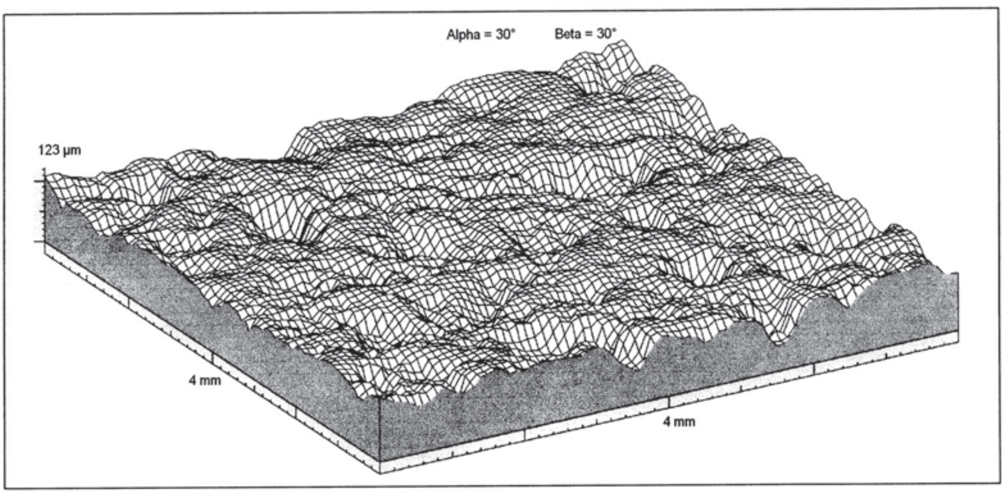

Fig. 5. 3-D view of ball's surface worn out in result of environment acting

Hygroscopic proprieties of lubricant, together with presence of identified polar groups, create conditions to formation of water solutions of electrolytes in whole volumes of grease or zonal, thus they created environment favourable for electrochemical corrosion.

In conditions of polar water environment (dipolar moment of water particle $\mu_{H_2O} = 1{,}84D$) polar groups have tendencies to migration according to locally created electromotive strengths. Presence of this energy, follows from formation of punctual residual currents, generated in consequence of potentials' differences resulting from appearing of oxygenate forms of iron as products of reactions :

$$Fe + nH_2O \rightarrow FeO + 2nH^+ + ne \tag{1}$$

or

$$Fe + mH_2O \rightarrow Fe_2O_3 + 2mH^+ + 3me \tag{2}$$

Then, redox type half-cells arise, e.g. from reactions:

$$Fe \rightarrow Fe^{2+} + 2e \tag{3a}$$

or

$$Fe \rightarrow Fe^{3+} + 3e \tag{3b}$$

in forms $Fe(II) \mid Fe(III) \cdot aq$, or from reaction:

$$Fe(OH)_2 \Leftrightarrow HFeO_2^- + H^+ \tag{4}$$

in forms: $Fe(II) \mid HFeO_2^- \cdot aq$ or $HFeO_2^- \mid Fe(III) \cdot aq$.

Moreover, essential importance have reactions with participation of molecular oxygen, leading to creation of alkaline environment:

$$O_2 + 2H_2O + 4e \Leftrightarrow 4OH^- \tag{5}$$

increasing susceptibility to pinhole corrosion of metal surface.

The components of identified in grease polar groups probably originate from environment of work and they become impurities of grease. Metals such as: lead, copper, zinc, nickel, manganese, are presented in dusts generated during processes of thermal destruction of solid fuels, as well as solid and semi-liquid waste substance. They carry danger of migration of heavy metals to superficial zone of surface layer including water and this way - create conditions to formation of electrolytic cells of metallic type.

In result of chemical analysis it was confirmed that in conditions of ineffective seal and resulting pollution of closer environment of work, as well as its humidity (tested journal bearing of bus bodies joint worked in such conditions), presence of grease inside the bearing created additional, favorable conditions to occure of pinhole corrosion.

Measurements of geometrical structure of balls and bearing raceways surfaces – see Fig. 5, confirmed corrosive genesis of decrements in working surfaces. In the measurements extended TalyMap Expert programme of measuring machine Talyscan 150 was used [9]. The results of this investigations were taken down in Table 1. Together with results of microscopic investigations, as well as chemical analysis, they practically exclude fatigue character of material destruction.

Table 1. The values of some parameters of decrements on surfaces of balls and raceways of rolling thrust bearings

Item	Decrement's parameters	Units	Values of dimensions	
			ball	raceways
1	area	μm^2	446.000	149.000
2	volume	μm^3	12.800.000	3.558.215
3	maximum depth	μm	71,6	59,1
4	average depth	μm	28,7	23,9

Moreover, environmental function of seal consists in limitation above mentioned penetration of pollutions to friction zone. It favors formation of conditions of proper mutual co-operation of elements of rolling pair, because chemical and mechanical cleanness of contact area of co-operating elements is the diminishing factor of intensity of wear processes.

6. Recapitulation

Carried out theoretical analysis, supported by experimental investigations, showed clearly great importance of environmental factors for proper work of rolling bearing pairs. Regarding most of them is indispensable already in stage of design and construction of machine because they usually determine its usable features. The activities in this stage also make independent functioning of machine, its efficiency and reliability from farther environment.

Assurance of possible low noxiousness of machine for environment is essential problem too. In this regard significant function fulfils seal of machine's pairs. Correctly designed and made it protects farther environment against pollution by applied in machine lubricant.

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CONSIDERATION IN DIAGNOSTICS OF THE GREY SYSTEM THEORY

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Abstract

The publication describes Grey System Theory (GST) and takes into account the Grey differential model (GM) and Grey Generating Space (GS). Grey System Theory shows to which extent the vibration signals (deriving from tested objects) influence the evaluation and analysis of condition of the machine. The above-mentioned theory applies under the existence of fix, non-negative and monotonic data correlated with insufficient and uncertain data' sources. In relation to these circumstances, the Forecasting (rolling window) method seems to be appropriate solution, which remains the main subject of this paper. The present research use vibration methods to recognize the technical state of the machines and SVD method (Singular Value Decomposition) as well as GST (Grey System Theory) was used for results validation.

Keywords: *grey systems, vibration diagnostics, forecasting, modeling of condition*

1. INTRODUCTION

The Grey System Theory (GST) was established in 1982 by Chinese scientist known as J.- L. Deng. At the beginning, despite the opportunity for widespread application of GST, the theory did not attract the attention of scientists and researchers base in western countries. In the early nineties of twenty century the circumstances has changed that is why we can observe the wide popularity of the GST which allows easy forecasting the state of the entities, machines respectively. The theory is commonly used across other related areas such as social and natural sciences, demography, hydrology and economy (such as anticipation of the market condition bases on particular data sources). Furthermore the idea of GST offers practical solutions available not only for scientists but also for engineers and entrepreneurs in the way of appropriate decision-making process [1,8,9,10,11].

One of the most adequate tool, which guide the management of algorithm sequence is MATLAB program with an important Toolbox (Statistic) function. The knowledge and the ability of practical usage of MATLAB program is required to understand the algorithm sequence described in following section [1,2,3,4,5].

The algorithm described below is not so complex, that is why the combination of its deep revision and appropriate MATLAB' skills force the correct implementation of the algorithm [6,7,12].

2. THE GREY SYSTEM ALGORITHM TOWARDS CONDITION FORECASTING

Most of the technical algorithms, including Grey System algorithm, can be expressed through the mathematical shape. First of all, it seems necessary to admit that Grey Model (GM) describes the system' behaviors in relation to particular symptom defined as ($x^{(0)}(t)$) in which t means the following obtained symptom, for instance ($t \in$ <1,2,3,…,∞), maybe presented under differential quotation at the bases on k simultaneously forcing e (the same foundation) which can also be shown as GM (k, e). The principle states that k concern the factor which force the changes under differential quotation and e takes the below-presented mathematical forms [2,4,5]:

$$\sum_{i=0}^{e} a_i \frac{d^{n-i}x^{(1)}}{dt^{n-1}} = \sum_{j=1}^{k-1} b_j y_j^{(1)} \tag{1}$$

in which:

$$x^{(1)}(k) = \sum_{i=1}^{t} x^{(0)}(i) \text{ is variable factor of base object} \tag{2}$$

y_j – independent behavior allows correct interpretation of reviewing object
a_i , b_i – polynomials rate estimating form time line $x^{(0)}(t)$, t = 1,2,3…,∞

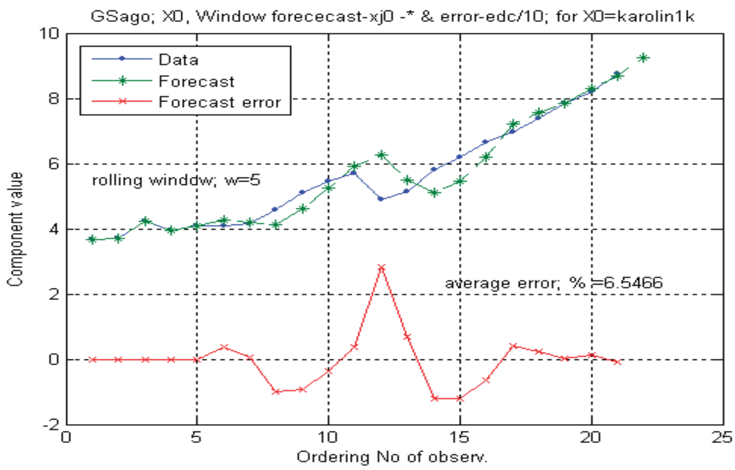

Fig.1. Model GM of monotonic data vibration signal

In the most of circumstances the starting point is obtained as GM (1,1) equation which means the differential quotation with only one forcing factor. The solution of this problem can be guided by following mathematical process:

Stage 1

Determination of observation' vector:

$$x^{(0)} = [x^{(0)}(1), x^{(0)}(2), x^{(0)}(3), \ldots, x^{(0)}(n)], \tag{3}$$

requirements: n ≥ 4,

in which n means number of observations.

Stage 2

Formation of Accumulating Generating Operation (AGO):

$$x^{(1)}(t) = \sum_{i=1}^{t} x^{(0)}(i),\ t = 1,2,3,\ldots n \tag{4}$$

which clearly defines the vector' monotonicity and growth:

$$x^{(1)} = [x^{(1)}(1), x^{(1)}(2), x^{(1)}(3),\ldots,x^{(1)}(n)] \tag{5}$$

requirements: $x^{(1)}(1) = x^{(0)}(1)$.

Stage 3

Basing on above-defined AGO vector, it is appropriate to describe the Grey differential model (GM) in relation to starting position GM (1,1):

$$\frac{dx^{(1)}(t)}{dt} + ax^{(1)}(t) = u \tag{6}$$

in which:

a – growth index,

u – controllable variable factor,

t – uncontrollable factor (e.g. time, asset depreciation).

Stage 4

The solution of the above-presented differential quotation with constant growing variable *t*, as following:

$$\dot{x}^{(1)}(k+1) = [x^{(0)}(1) - u/a] \exp(-ak) + u/a, \tag{7}$$

in which $\dot{x}^{(1)}$ describes the potential forecast of Accumulating Generating Operation

Stage 5

The replacement of the differential complete accretion (Stage 3) relates to *t=1* and composition of the precedent and progressive equations.

- precedent equation: $x^{(1)}(k+1) - x^{(1)}(k) + ax^{(1)}(k) = u$
- progressive equation: $x^{(1)}(k+1) - x^{(1)}(k) + ax^{(1)}(k+1) = u$

The combination of the equations gives us the model of:

$$x^{(1)}(k+1) - x^{(1)}(k) = -a/2[x^{(1)}(k) + x^{(1)}(k+1)] + u \tag{8}$$

k = 1,2,3,…,n.

Stage 6

The conversion of above-mentioned model relates to the following *k* values (basing on previous obtained observation vector - $x^{(0)}$) to estimate the unknown extra differential quotation ratios [*a*, *u*]. These estimation is indicated via selection of 'smaller squares' to final attainment of matrix solution:

$$[a,u]^T = (B^TB)^{-1} B^TY$$

$$\text{in which: } Y = [x^{(0)}(2), x^{(0)}(3), x^{(0)}(4),\ldots, x^{(0)}(n)]^T \tag{9}$$

$$
B = \begin{bmatrix} -[x^{(1)}(1) + x^{(1)}(2)] & 1 \\ -[x^{(1)}(2) + x^{(1)}(3)] & 1 \\ \dots\dots\dots\dots\dots\dots & \dots \\ -[x^{(1)}(n-1) + x^{(1)}(n)] & 1 \end{bmatrix} \qquad (10)
$$

Stage 7

The inverse transformation of AGO presents the potential forecast (basing on AGO' vector) obtaining:

$$\dot{x}^{(0)}(k+1) = \dot{x}^{(1)}(k+1) - \dot{x}^{(1)}(k), \qquad (11)$$

furthermore the conversation with previous-presented progressive and precedent equation (Stage 5) allows to determine the final forecast in relation to starting GM (1,1) model:

$$\dot{x}^{(0)}(k+1) = [x^{(0)}(1) - u/a](e^{-ak} - e^{-a(k-1)}) \qquad (12)$$

$$k = 2,3,4,\dots,n$$

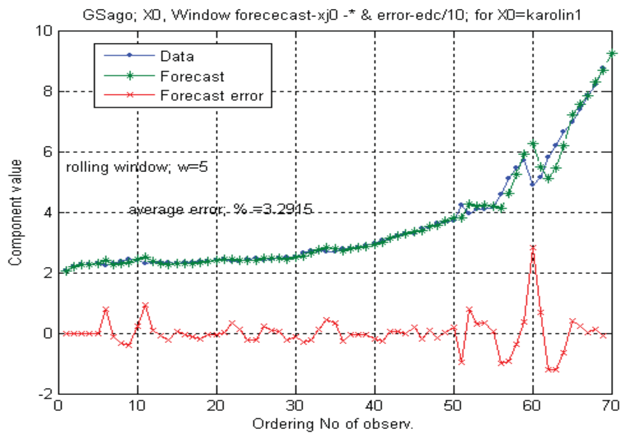

Fig.2. Forecast error in prognosis signal GST

The final stage of these algorithm shows the general principle of the Grey System Theory (GST). With accordance to *Stage 2* it is easily to suggest that the method can fully describe the monotonic and expanding process relates to depreciation of particular asset' or machine' elements. That is why the above-describes method can be used to properly describe and forecast the condition of the machine basing mainly on vibration' symptoms.

What is more important the sufficient forecast can be obtained after few observations, at the same time, the increasing number of observations is positively correlated with growth the possibility of gathering wrong research data – decrease the efficiency of the observation.

The Forecasting (small-size rolling window) method seems to be appropriate solution for short term prediction which simultaneously covers and analyse huge number of data. The determination of appropriate and final forecast (basing on previous-described numerical method) can be possible by cooperation with MATLAB program that is why the adequate researcher skills are highly important factor which definitely determines the forecast' success.

3. ONE – DIMENSIONAL FORECASTING ACCORDING TO GREY SYSTEM METHOD

As it was mentioned before, the already reviewed algorithm can be easily implement across practical basis. This stage of the report describes the influence of the Grey System method on the process of diagnosis and analysis of machines. The excellent example of this phenomenon contains the review of airplane' turbine bearing.

As a results of assets depreciation we can observe the rise of oscillations. The number of observations regarding oscillations' features – velocity of vibration (mm/s) across equal time interval (each 20 hours) – are presented below:

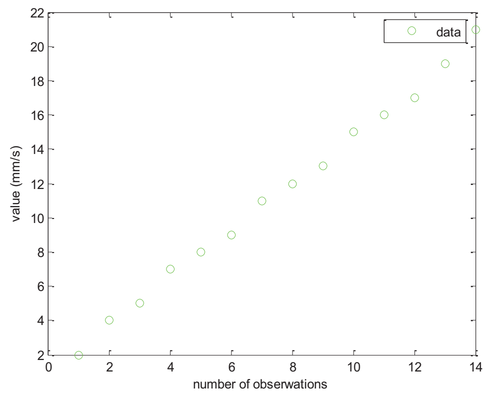

Fig.3. Value of observation signal

The observation' results shows the monotonic and expanding line that is why it seems appropriate to adopts the Grey System Theory (GST). Furthermore, we implement these algorithm at the MATLAB program and upload already gathered data which helps us to obtain the following forecast:

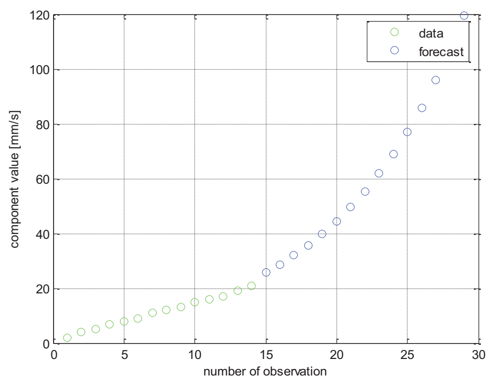

Fig.4. Monotonic line forecasting

Part of the evolution of the airplane' turbine bearing basing on 14 observations and the Forecasting (rolling window) method (15 continual inspections – 300 working hours)

According to above-mentioned graph, the obtained forecast contains several similarities towards quadratic function graph. This phenomenon is fully adequate to practical reality which means that particular unit depreciation is irregular – the higher depreciation ratio drives the velocity of depreciation process. The above-mentioned forecast has visional similarities that allows me to state that the forecast is fully suitable. After measurement of forecast failure' ratio presents the value slightly more than 5%. Basing on this result we can state that forecast fulfill the credibility requirement (the forecast can be accepted with around 5% of failure).

Moreover, it seems also necessary to admit that this method is mainly dedicated for short-term forecast. What is more important, the implementation of measurement should base on continual observations across equal periods of time.

4. MULTIDIMENSIONAL CONDITION' FORECAST

Currently the multidimensional condition forecast is at the stage of early development so that the implementation of related analysis methods is very difficult. Generally, the multidimensional condition forecast bases on the other necessary tools, connected with the artificial intelligence called – neural network, data fusion respectively. Moreover, the Singular Value Decomposition (SVD) is also closely related with the above-mentioned forecast. However, the deep description of that tools and methods are specified in the other reports because of theirs discrepancies with the Grey System algorithm basis.

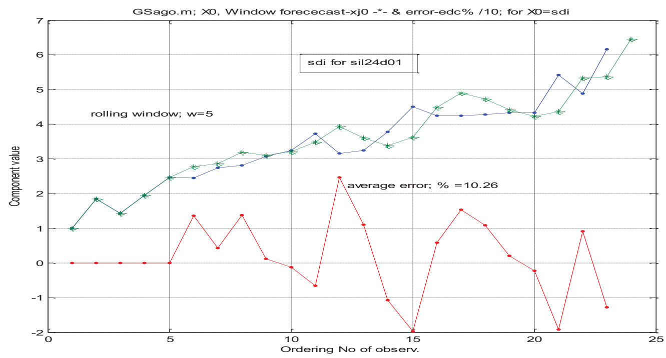

Fig.5. Forecast state of SVD method

The fundamental knowledge of the multidimensional condition forecast is required to become fully capable to interpret processes related with vibration diagnostics' studies.

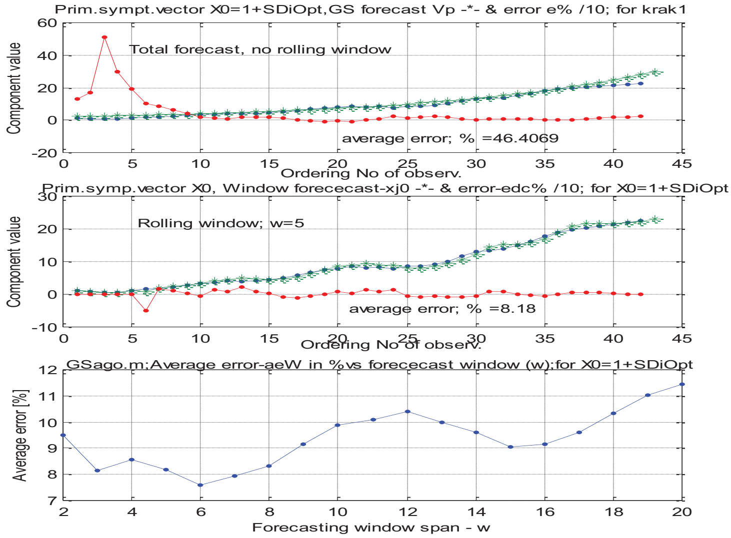

Fig.6. Multidimensional condition forecast GST

5. SUMMARY

This report clearly states the Grey System Theory is especially adequate to forecast one-dimensional unit' condition. The wide opportunity to application of the theory can be utilized across statistical purposes (estimation of revenue or trends) as well as in assets' vibration diagnostics.

The well and suitable performance of Forecasting (rolling window) method allows to decrease the possibility of forecasting failure that is why the appropriate interpretation of Grey System algorithm and increase the knowledge drives the regularity and conformity of the research.

Finally, the report reminds that the proper utilization of above method relates to depreciation of machines' units, with the monotonic and expanding continual values.

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