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Issue of changes in technical states of a Diesel engiene as the result of wear of its tribological systems

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Abstract

The paper shows a necessity to consider the processes of load on and wear of tribological systems in Diesel engines (and therefore the changes in their states) in real conditions as random processes. This means that mathematical models for these processes can be random functions with an argument which is time (t) and with values which are random variables such as loads and wear of particular tribological systems. The load properties for the tribological systems have been presented as the main causes of their failures. It has been signaled that a stochastic dependence should be expected between mechanical load and thermal load, whose the intensity (force) can be defined by Czuprow's convergence coefficient. A proposal has been submitted to accept that the tribological system load in Diesel engines is a stationary process with asymptotically independent increments. The reasons for this proposal have been presented in the form of relevant hypotheses and an exemplary graph of changes in pressure and temperature in an engine cylinder under operation. Also it has been signaled that the models of wear processes in tribological systems may be stochastic processes with special properties called semi-Markov processes. The wear properties for these systems have been explained in the form of relevant hypotheses.

Introduction

The rational operation of marine Diesel engines involves in particular a proper control over changes in their technical and energy states. The changes in these states depend mainly on load on their basic tribological systems, such as pistons-piston rings-cylinder liners and main and crank bearings. This implies a duty to supervise realizations of the load processes and the associated wear processes in the systems and to control these processes. The studies show that realizations of load and wear processes for these systems (thus also the changes in their technical and energy states) in real conditions must be considered as random processes [1, 2, 3, 4, 5, 6, 7, 8, 9]. Thus, it becomes important to determine the properties of the processes to enable development of their relevant mathematical models. Defining the properties of the load processes and the resulting wear processes for these systems is possible in case of formulation of proper hypotheses that would explain the facts collected during examination of the processes. The hypotheses formulated herein show that mathematical models of the processes should be presented as random functions with an argument which is time (t) and with values which are random variables such as loads and wears of particular tribological systems. The studies show that they may be stochastic processes with special properties called semi-Markov processes [5, 6, 7, 8, 10, 11, 12, 13, 14, 15]. This enables application of the theory of controlled (decision) semi-Markov processes to control the load processes and the associated wear processes in tribological systems [11, 16]. Thus, the considerations herein have been focused on the concerns of tribological systems wear, but with regard to the properties of loads on the systems.

Properties of loads on tribological systems in engines as a major cause of their failures

Both, mechanical (Q_M) and thermal (Q_C) loads on the mechanisms of tribological systems in main marine engines need to be classified as the major causes of wear of these systems, as well linear (in surface) as in volume. For this reason, the effect of load resulting in occurring damages in this type of engine systems should be analyzed carefully in particular. This follows from the fact that the loads change randomly, sometimes within wide limits [1, 2, 4, 8, 10, 12, 14, 15, 17].

As a result, different values of indices (parameters) of Diesel engine operation can be registered, which are random events [12, 14, 15, 18, 19].

Thus, during operation, the load $Q(t)$ on engine tribological systems should be considered as a stochastic process, paying attention that it comprises a mechanical component $Q_M(t)$ and thermal $Q_C(t)$ component, which can be expressed in the form of the following relation:

$$Q(t) = f[Q_M(t), Q_C(t)] \tag{1}$$

where: Q – engine load; Q_M –engine mechanical load; Q_C –engine thermal load; t – engine operating time.

Thus, in empirical studies of the load, two stochastic processes $\{Q_M(t): t \geq 0\}$ and $\{Q_C(t): t \geq 0\}$ can be considered as components of the process $\{Q(t): t \geq 0\}$ [3, 5, 6, 11].

In case of Q_M and Q_C loads it can be written down that they are characterized by the operating indices (parameters) and other physical quantities at any time t as follows:

$$Q_M : [p_{\max}, p_z, p_e, c_{śr}, \varphi, \varphi_p, n, P_g, P_b, \ldots] \tag{2}$$

$$Q_C : [\dot{q}, \nabla T, \rho, p_e, c_{śr}, T_{\max}, T_z, T_{sw}, \dot{Q}, \ldots] \tag{3}$$

where: $p_{\max}$ – maximum combustion pressure; p_z –combustion pressure; p_e – mean useful pressure ($p_e = \eta_m p_i$; η_m – mechanical efficiency; p_i – mean indicated pressure); $c_{śr}$ – average piston speed; φ – rate of (izochoric) pressure increment; φ_p – instantaneous rate of pressure increase; n – crankshaft rotational speed (in engine); P_g – force from gas pressure; P_b – force of inertia; $\dot{q}$ – thermal flux (thermal energy) density; ∇T – temperature gradient; ρ – initial (isobaric) expansion rate; $T_{\max}$ – maximum combustion temperature; T_z – combustion temperature; T_{sw} – exhaust gas temperature; $\dot{Q}$ – thermal flux; T_{ol} – oil temperature; T_w – cooling water temperature.

Previous research shows that some quantities, e.g. p_e, $c_{śr}$ [2, 12, 18, 19, 20] characterize both, mechanical and thermal loads. Therefore, it is obvious that dependencies exist between mechanical load and thermal load. Due to the fact that they are random processes, the conclusion is that a stochastic dependence must be expected between them. Therefore, in order to explain this dependence the following hypothesis $\boldsymbol{H_1}$ can be formulated: "**a stochastic dependence exists between mechanical load $\boldsymbol{Q_M(t)}$ and thermal load $\boldsymbol{Q_M(t)}$, as the defined variants of mechanical load $\boldsymbol{Q_M}$ are accompanied by different variants of thermal load $\boldsymbol{Q_C}$.**" Hence, the conclusion that the dependence between the processes of $Q_M(t)$ and $Q_C(t)$ loads cannot be described by applying a usual method of algebraic equations [21]. The dependences existing between the loads on the system are affected by a large number of factors, including these that cannot be measured [8, 12, 13, 18, 19]. Thus, the degree to which the load Q_M is connected with the load Q_C may be very different. It follows from that there is a need to take into account the intensity (strength) of the stochastic relation between Q_C and Q_M.

The intensity (strength) of the stochastic relation between $Q_M(t)$ and $Q_C(t)$ can be determined during empirical studies, from the formula [21]

$$T_{MC}^2 = T_{CM}^2 = \frac{\chi^2}{N\sqrt{(k-1)(l-1)}} \tag{4}$$

where: k – number of variants of the variable Q_M; l – number of variants of the variable Q_C; N – boundary number of variants of the variable Q_M or Q_C; χ^2 – value calculated from the chi-square formula; $T_{(\cdot)}{}^2$ – Czuprow's convergence coefficient.

It can be shown [21] that T_{MC} takes values from the interval [0, 1]. This ratio is equal to zero ($T_{MC} = 0$), where there is no relation between the values of the process (Q_M and Q_C), while the ratio equal to one ($T_{MC} = 1$) proves the existence of a functional dependence.

It results from previous studies [3, 6, 9, 17, 21, 22] that the following hypothesis $\boldsymbol{H_2}$ can be formulated: **"load is a process with asymptotically independent increments, as with increasing the time gap between time intervals in which the load is examined (measurements of load are performed), its values become less and less dependent on each other".**

The studies also show that there is no monotonicity in changes of engine load in a longer period of operation, which allows to formulate a hypothesis $\boldsymbol{H_3}$ stating that: **"load in a longer period of operation (work) is a stationary process, as at such time there is no monotonicity in changes of engine load"** [21].

Additionally, the studies of Diesel engines provide that when the time interval (Θ) between measurements of values of these loads increases, the correlation between the loads decreases. Thus, the values of load measured at time intervals considerably distant from each other can be regarded as independent. This property is called asymptotic independence of the load value. It occurs when the gap Θ increases. This independence can be stated by analyzing the values of engine loads (Fig. 1) in time intervals considerably distant from each other (by the gap Θ), e.g. in the time interval (t_i, t_{i+1}) corresponding to the interval (α_i, α_{i+1}) and in the time interval (t_k t_{k+1}) corresponding to the interval (α_k, α_{k+1}). The dependence between these loads will be small and the less the greater the time interval (gap) Θ is. An example of such dependence is illustrated in figure 1.

The presented view on the engine load properties may lead to new opportunities for obtaining a probabilistic description of wear with regard to its dependence on load. The studies show that a model describing the changes (over time) in value of wear at a given wear rate of tribological systems in Diesel engines can be a model known as the Lorenz curve (Fig. 2) [2, 4, 8, 10, 20, 23].

The development of wear is, however, a random process which is irreversible and with values dependent on engine load [2, 12, 15, 18, 19].

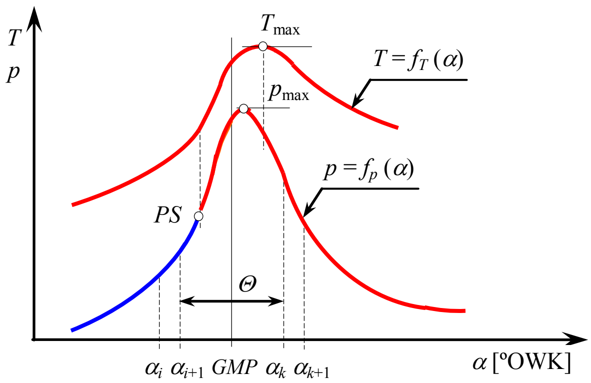

Fig. 1. Graph of changes in pressure and temperature in a Diesel engine cylinder: p – pressure, T – temperature, α – crank angle, p_{max} – maximum combustion pressure, T_{max} – maximum temperature; PS – combustion start in a cylinder; Θ – time interval of engine operation.

Thus, the hypothesis ($\boldsymbol{H_4}$) can be accepted as follows: ***when Diesel engine is operated rationally, wear (Z) of each tribological system and its load (Q) are random variables closely related to each other because it can be stated that between them there is such a linear stochastic dependence that the correlation coefficient $r_{qz} = 1$.*** This means that in rational operation, with increasing load the wear increases, and inversely – less load causes less wear of tribological systems in the same time interval of engine operation.

Different (random) loads cause irregular course of wear of particular tribological systems, and at the same a random range of wear rate of Diesel engine systems. For this reason, it is necessary to identify the properties of the processes of wear for engine tribological systems.

Properties of the process of wear of tribological systems in engines

The process of wear of any tribological system is affected by: initial system quality and a random range of wear rate, resulting from variable loads and lubricating conditions.

The wear process in tribological systems for any Diesel engine can be presented in a form of the Lorenz curve, which is of course one of many models that can be used for studies of changes in states of these systems at a defined time [8, 14, 23, 24]. All of these models reflect a linear development of wear in a time interval in which the wear is normal (stable) wear. In other time intervals wear may have a different course, e.g. logarithmic, exponential, etc. This means that when recognizing the process of wear of a given tribological system the following can be considered: 1) state z_0, in which break-in proceeds, 2) state z_1, in which normal stable wear proceeds and 3) state z_3, which causes accelerated (damage, catastrophic) wear and hence it is undesirable.

Thus, an assumption can be made as follows: *a process of changes in states of tribological systems $z_i \in Z^*$ ($i = 0, 1, 3$) is a stochastic process with discrete realizations.*

The process of wear in all these states depends on many factors, but mainly on load on particularly tribological systems. Therefore, taking into account the described properties of load, a hypothesis ($\boldsymbol{H_5}$) with the following wording can be regarded as acceptable: ***increments of wear of tribological systems (in their particular states) in Diesel engines are the increments of small dependence between each other and this is the smaller the larger the time interval (gap) (Θ)is between the increments, since such a course of autocorrelation function can be observed for each tribological system, that with increasing the gap Θ the function decreases rapidly at first and then oscillates around zero with a relatively small and getting smaller – with increasing Θ – amplitude.*** Thus, this hypothesis is a consequence of the previously accepted hypothesis $\boldsymbol{H_4}$ with a linear dependence of load and wear of tribological systems in Diesel engines. This follows from that the hypothesis with asymptotic independence of increments of wear of tribological systems (not only in Diesel engines) may be accepted if the reasons for its rightness follow from the analysis of the properties of loads and the associated other phenomena occurring in such systems [4, 8, 19, 23]. It is important to bear in mind that the quantitative analysis of the wear process in the mentioned systems cannot provide evidences that the investigated process is the process with asymptotically independent increments. Such an analysis can only show that there is no reason to reject a hypothesis about asymptotic independence of increments of wear of the studied tribological systems.

The processes of wear occurring in particular Diesel engines' tribological systems of the same type are the processes with intertwining realizations [4, 23]. Wear increments in subsequent moments of each engine operating time are dependent on each other. However, when distinguishing two time intervals of engine operation (t_j, t_{j+1}) and (t_k, t_{k+1}) between which the gap Θ is large enough, the dependence of wear increments Δz_j and Δz_k is negligible and decreases with the increase of the period [4, 8, 19, 20, 23]. Thus, the process of engine tribological systems wear can be considered as a process with asymptotically independent increments. This is important because in the theory of processes with asymptotically independent increments it has been proved that the variance of such processes increases linearly over time [4, 9].

Basing on results of one analysis of a wear process it is unfortunately impossible to determine whether the obtained course of the process is affected mainly by the initial quality of tribological systems or by accidental changes in their loads, which cause (as already said), an accidental range in the systems wear rates. In order to determine this it is necessary to perform a statistical analysis of several realizations of wear process for the systems of the same type [4, 21].

Analyzing a wear process for any tribological system in accordance with the Lorenz curve (Fig. 2), some boundary values can be distinguished, i.e. such values of a given system wear, which if exceeded, cause a quality change in the properties of the system. The values which are certainly of practical usefulness, include:

– initial (preliminary) value of wear (z_p), which occurs after the time interval $(0, t_p]$, called a break-in period (D), so after duration of the state z_0 of a tribological system, that is the state which ensures its proper break-in;
– permissible value of wear (z_d), which occurs after the time interval $(t_p, t_d]$, so after duration of the state z_1 of a tribological system, that is the state ensuring regular (quasi-stable, normal) wear;
– non-permissible (destructive, extreme) value of wear (z_n), which occurs after the time interval $(t_d, t_n]$, so after duration of the state z_2 of the tribological system, that is the state causing accelerated (emergency, catastrophic) wear such as seizing up of the system, or local or extensive welding, melting, etc, of the surfaces of its components, when the initial stadium of destruction in the system rubbing surfaces does not occur yet;
– catastrophic value of wear (z_k), which occurs after the time interval (t_n, t_k), so after duration of the state z_3 of the tribological system, when a catastrophic (definitive) damage occurs, making further system operation impossible.

An interpretation of the boundary values is shown in figure 2.

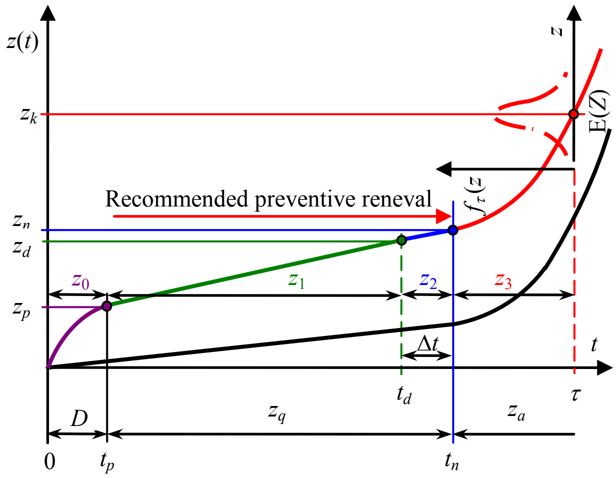

Fig. 2. Exemplary curves of typical wear development for sliding tribological systems: z_d – permissible value of wear, z_n – non-permissible (destructive, extreme) value of wear, z_k – catastrophic value of wear; D – break-in period, z_q – quasi-statistic (normal, regular) period of wear, z_a – period of catastrophic (accelerated, destructive) wear, z_0 – state of system's partial ability that results from break-in, z_1 – state of system's full ability, z_2 – state of system's partial ability, z_3 – state of system's inability, $f_T(z)$ – density function of system wear, $E(Z)$ – expected value of wear, Z – wear as a random variable, Δt – time interval in which preventive maintenance should be started, t_p – time of reaching the value z_p, t_d – time of reaching the value z_d, t_n – time of reaching the value z_n

For practical reasons, a tribological system should be regarded as damaged, when the value of its wear reaches a permissible value. This follows from the fact that an average wear rate for the system, in case of existing state z_3, increases with the time of its operation due to that the wear process in the system finding just in this state is additionally influenced by processes accelerating the system wear.

On the assumption that the wear rate for tribological systems is sufficient to be described with the equation [4, 23]

$$v(t)=u(t)\rho(t) \tag{5}$$

the system wear rate is a function consisting of a determined function $u(t)$ and a stationary random process $\rho(t)$. With so interpreted wear rate for the system, its wear can be expressed with the following formula [4, 23]:

$$Z(t)=\int^{t} v(\tau)\mathrm{d}\tau \tag{6}$$

Due to [4] $\mathrm{E}\{v(t)\} = u(t)$, the expected value of wear in the time interval $[0, t]$ can be defined as follows:

$$\mathrm{E}\{Z(t)\}=\int_0^t \mathrm{E}\{v(\tau)\}\mathrm{d}\tau=\int_0^t u(\tau)\mathrm{d}\tau \tag{7}$$

In operating practice, wear of tribological systems may proceed with different rates [4, 8, 12, 19, 23]. Sometimes, it can be accepted that the wear rate $u(\tau) = a\tau + b$ for $0 < \tau \le t$. Then, the equation (7) takes the following form:

$$\mathrm{E}\{Z(t)\}=\int_0^t (a\tau+b)\mathrm{d}\tau=\frac{1}{2}at^2+bt \tag{8}$$

The variance is then defined with the formula [4]:

$$\mathrm{D}^2\{Z(t)\}=\mathrm{D}^2(a)t^2+\mathrm{D}^2(b) \tag{9}$$

The equation (9) shows that if realization of the wear process depends only on the initial quality of a tribological system, then the variance $\mathrm{D}^2\{Z(t)\}$ increases proportionally to the second power of time t (i.e. t^2).

On the assumption that realization of the wear process in tribological systems is linear (as for the period of normal wear), their expected value of wear at any time t can be described with the equation (8) where the quantities a and b are independent random variables. The variance of such a process of wear, in case it can be accepted that this is a process with asymptotically independent increments, can be defined as follows:

$$\mathrm{D}^2\{Z(t)\}=\mathrm{D}^2(a)t^2+\mathrm{D}^2(b)=At+B \tag{10}$$

where: A, B – constant coefficients.

It results from the equation (10) that the variance of a wear process with asymptotically independent increments and with uniform initial quality of tribological systems, grows linearly depending on time, so it is a linear function of time.

The assumption of realization linearity of a wear process for tribological systems of Diesel engines does not infringe the general principle of the statistical analysis. This follows from that in case of a variable rate of wear of these systems, it is possible (by changing the operating time scale for the given system) to transit to the process with a constant wear rate. Therefore, a similar analysis can be performed for a wear process during the periods of break-in and catastrophic wear of each sliding tribological system.

The presented properties of the load and wear processes for tribological systems in Diesel engines and the existing capabilities of measuring the loads and wear of these systems [4, 16, 17, 19, 20, 23, 24, 25, 26], empower to formulate the conclusion that a model of the wear process for tribological systems in Diesel engines can be a semi-Markov process with a corresponding function matrix and a known initial distribution, whose values are the distinguished earlier states $z_i \in Z^*(i = \overline{1,3})$.

Final considerations and conclusions

The presented model of the process of changes in the states $z_i \in Z^*(i = 1, 2, 3)$ of tribological systems inside Diesel engines is a process with a finite set of states and continuous time, and considerations relate mainly to such mechanisms and systems like main bearings and crank bearings, and piston – piston rings – cylinder liner.

It has been shown herein that the wear process and the associated wear of tribological systems depend significantly on their load and that the load on each Diesel engine, thus also its tribological systems, examined at any time of its operation (work) can be recognized as a multidimensional random variable. The loads analyzed in subsequent moments of the operating time of such engines, can be considered as realizations of a load process. Therefore, the load process for each engine should be investigated by assuming that it is a multidimensional stochastic process.

The proposed hypotheses explain why it can be accepted that the load process for any tribological system can be regarded as a stochastic and stationary process with asymptotically independent increments, and – that there is a stochastic relation between its mechanical and thermal loads, whose the intensity (force) can be determined during research by using the Czuprow's test of convergence.

It has also been shown that the process of wear of engine tribological systems can be regarded as a process with asymptotically independent increments. This is important in operating practice, because it has been proved in the theory of the processes with asymptotically independent increments that the variance of such processes increases linearly with time. This is of significant importance because the variance includes information about the process of wear.

The attention has also been paid that the model of the process of wear of tribological systems in Diesel engines can be a semi-Markov process. This has a significant importance for practice because a big advantage from application of semi-Markov processes (like in case of using Markov processes) is that there are available professional computer tools which enable solving different systems of equations of states for this type of models of real processes. As a result of the above, the probabilistic characteristics for tribological systems can be easily determined.

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