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Estimation and Assurance of Machine Component Design Lifetime

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Abstract

The authors offer a kinetic model of machine components degradation failure development process by the material strength energy criterion at the stage of research and development. On the basis of this model, they developed an algorithm and a program, which will make it possible to estimate the design service life of the load element in various conditions of external loading. Theoretical results of the research work were verified and their practical application under production conditions was described.

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Keywords: Failure pattern; Machine components; Energy strength criterion; Durability; Design service life; Lifetime; Verification.

1. Introduction

The main task of modern machine-building is to develop competitive machines and to provide their high quality mainly due to improvement of their reliability level, which is calculated at the stage of research and development. One of the main preconditions for high reliability of machines in the process of their development is the reasonable choice of the machine structure and its main components. At one of the stages of research and development, which is referred to as component development, the principle of double designed protection is implemented as well as such criteria as minimization, limitedness, design modularity, and protection diagnostics [1]. Among other things, this principle includes verification check-out (checking calculation) of reliability of individual elements and the machine as a whole. The designed level of reliability of the developed machines is determined on the basis of reliability and durability prediction for the most critical components of their structure by the material strength criterion [1, 2]. For this purpose, beside deterministic model, probability failure models [1-8] are used, which are obtained on the basis

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of statistical processing of numerous experimental data. These models make it possible for a mechanical design engineer to estimate the changes in the probability of failure-free operation of various elements at possible fluctuations of external loads and properties of materials on the basis of past operating experience of similar products or sample tests.

Analysis of domestic and world experience shows that to describe the processes of failure development in machine components under various conditions of external loading, the following stress models are used:

- operating loading H - material resistibility R [1];
- stress s - strength S [2, 3];
- load S - resistibility R [4, 5];
- operating stress σ - breaking stress σ_{np} [6, 7];
- estimated stress σ_a - breaking stress σ_∂ [8] etc.

The main reliability index determined by such models is the probability of failure-free operation of a component, the magnitude of which (in terms of [7]) is determined by the condition $P = P(\sigma < \sigma_{np}) = P(\sigma_{np} - \sigma > 0)$, where σ and σ_{np} are stochastic variables distributed by some law. This condition of the component non-failure determines the probability that maximum normal, tangential or equivalent operating stress σ will not reach the corresponding limiting value σ_{np} of ultimate resistance, yield stress, fatigue endurance limit, creep rupture limit, etc. This reliability index of the components to be developed is introduced instead of the somewhat arbitrary deterministic safety factors (safety margins) $n = \sigma_{np} / \sigma$, as even when the safety factor n is the same, the value of P can vary widely [2].

As the probability of failure-free operation depends only on the kind of distribution of operating σ and breaking stresses σ_{np}, a great number of experimental investigations were devoted to the laws of distribution of damageability parameters σ and damage criteria σ_{np}, viewed as stochastic variables. In particular, analysis of investigations carried out by scientists from American Society for Testing Materials [2] shows that ultimate resistance, yield stress, fatigue limit, etc. can be distributed according to different laws: normal law, logarithmic-normal law, Weibull law, gamma law, and others [9-11]. However, it is impossible to offer the same generalization for specific distributions of operating stresses [2].

Mechanical (static) approach is used to determine operating stresses in the process of estimation of design component reliability by long-term static strength and fatigue strength criteria. This approach is based on continuum mechanics described by the objective laws of elasticity theory and plasticity theory on the basis of the limiting state hypothesis [12].

That is why static models do not take into account the current time of the damageability process development or structural changes of real defective nonhomogeneous engineering materials, especially in the deformed surface layers during contact loading of components. Limiting theories are unable to explain the causes of material damage when the loads are less than the critical values (when $\sigma < \sigma_{np}$). The values of σ and σ_{np} are not properties of materials, they characterize the process of component loading and can vary widely depending on the experiment conditions, they have a very wide scattering (more than double and even triple scattering) [12, 13]. These models are unable to predict the expected service life of load elements and in most cases they have very low practical value because they reflect the results of past events, when the machine has already outlived its useful life. That is why, in the opinion of many specialists these models are not reliable, especially for newly developed components, and do not meet the test of experiment without selecting corresponding empirically determined coefficients [13 - 15].

Further investigations of solid body behavior under load resulted in a new thermofluctuation approach, which describes failure as a kinetic dynamic process of gradual damaging of materials and defect accumulation in their structure [12, 13, 16-18].

One of the most experimentally justified and best developed complex theories offered within the frame of the ergodynamic concept of deformable solid bodies is the thermodynamic failure theory of V.V. Fedorov [13, 18]. It consists of the structure-energy analysis of the kinetic process of damaging and material damage. The theory is based on general laws of thermodynamic for irreversible processes, on molecular-kinetic theory of Y.I. Frenkel and thermofluctuational strength theory of S.N. Zhurkov and it makes use of fundamental Arrhenius equation as well as dislocation theory in their mutual dialectical connection on the basis of the basic law of nature - energy conservation law [19].

However, practical application of contemporary energy theories in the process of development of failure patterns for machine components at the design stage requires either a prior model or full-scale tests of engineering samples or special experimental investigations, in order to obtain the physical quantities making up the models. The problem of purely analytical description of load elements degradation and forecasting of their failure-free operation and durability has not been solved yet.

An effort to develop such a model on the basis of the simplified kinetic equation of component damaging of V.V. Fedorov [13] without taking into account strengthening of the deformed material was made in the research work [20]. However, satisfactory results were obtained only for a narrow range of operating stresses in static load strength conditions.

On the basis of the kinetic concept development the authors offer a variant of the universal analytical model of machine component failure by the energy criterion of material strength for various conditions of their external load.

2. Kinetic model of forecast failure of machine components

Unlike the empirical relationships described above, the offered kinetic model is based on the energy condition of the element operational integrity of the following form: «damageability parameter u_t – failure criterion u_*». Where [13, 18]:

$u_t = u_0 + \Delta u_t = u_0 + \Delta u_{et} + \Delta u_{Tt}$ is the current magnitude of internal energy density of the local, most loaded material regions where the operating stresses σ reach their maximum values; u_0 is internal energy density u_t of component material in its initial state (when $t = t_0$); Δu_t, Δu_{et} and Δu_{Tt} is the gain in energy density u_t, its stored (potential) u_{et} and kinetic (heat) u_{Tt} component for the time period t; $u_* = \Delta H_{TB}$ is critical energy intensity of the component material under these loading conditions, its value correlates with the value of the enthalpy of melting of the material in solid state ΔH_{TB}.

In this case, the condition of probability estimation of failure-free operation of a component at any instant t of its future operation and failure condition take the following respective forms:

$$P(t) = P(u_t < \Delta H_{TB}) \text{ and } u_t = \Delta H_{TB} . \quad (1)$$

Consequently, the expected gamma-percentile life t_γ and the operation life $t_{сл\gamma}$ for the predetermined acceptable value $[P(t)] = \gamma$ and the corresponding fractile Z_γ, are determined by solution of equations:

$$u_* - (\overline{u}_0 + \overline{\dot{u}}_t \cdot t_\gamma) = Z_\gamma \cdot \sqrt{\sigma^2_{u\,0} + \sigma^2_{\dot{u}} \cdot t^2_\gamma} \; ; \; t_{сл\gamma} = (1 + \Pi) \cdot t_\gamma , \quad (2)$$

where ($\overline{u}_0$, $\sigma^2_{u\,0}$) and ($\overline{\dot{u}}_t$, $\sigma^2_{\dot{u}}$) are the mean value and the random variable dispersion of u_0 and $\dot{u}_t$ respectively; Π is the share of scheduled downtime of the machine. Here $\dot{u}_t$ is the density accumulation rate of the material internal energy.

Physical stochastic model (1) - (2) in its general form describes the process of gradual structural damage of materials of loaded component at the time of t, their failure at the instant $t = t_\gamma$ (or $t = t_{сл\gamma}$) and reflects the kinetics of changes of their energetic state.

In order to obtain the analytical dependence (2) for estimation of the design service life t_γ (or operation life $t_{сл\gamma}$) of machine components under various conditions of external loading and to simplify mathematical manipulation, the following assumptions were made:

- the following mean variables will be used $\overline{u}_{e0} = u_{e0}$, $\overline{u}_t = u_t$, $\overline{\dot{u}}_t = \dot{u}_t$, $\overline{t} = t_*$ and $\overline{t}_{сл} = t_{сл*}$, setting the acceptable value of the probability of failure-free operation, which is equal to $[P(t)] = \gamma = 0{,}5$, when $Z_\gamma = 0$, $\sigma^2_{u\,0} = \sigma^2_{\dot u} = 0$;

- assume that the conditions of loading of the component under study are steady-state, when the maximum stress and temperature of the component can be considered constant within the definite operating period ($\sigma = const$ $T = const$). In this case, taking into account the relationships above, the failure condition (1) of the element under study will take the following form [20]:

$$\Delta u_{et} = \Delta u_{e*}, \tag{3}$$

where Δu_{e*} is the critical increment of hidden energy density of structural defects of component material:

$$\Delta u_{e*} = \Delta H_{TB} - u_{e0} - \Delta u_{Tt}; \tag{4}$$

u_{e0} is the initial value of hidden energy density of defects u_{et} when $t = t_0$:

$$u_{e0} = \left((0{,}067 \cdot HV)^{1,2} \cdot k_\sigma\right)^2 / (6 \cdot G); \tag{5}$$

k_σ is the complex structural parameter:

$$k_\sigma = \left(6{,}47 \cdot 10^{-6} \cdot HV + 0{,}12 \cdot 10^{-2}\right)^{-1}; \tag{6}$$

G and HV are the modulus of elasticity in shear of the material and the mean value of Vickers hardness;

$\Delta u_T = \rho(T) \cdot c(T) \cdot T$ is the constant incremental size of thermal variable of internal energy of the element material at operating temperature $T = const$;

$\rho(T)$ and $c(T)$ are density and heat capacity of the material.

Equations (2) for estimation of the average operating life and service life taking into account (3) - (6) will take the following form:

$$t_* = (u_{e*} - u_{e0}) / \dot u_e \,;\; t_{сл*} = (1 + П) \cdot t_*, \tag{7}$$

where $\dot u_e$ is the mean accumulation rate of potential energy of structural defects of the material in the process of component operation.

To estimate the rate of element degradation $\dot u_e$, the authors made use of the kinetic equation of solid body damaging in quasi-steady-state approximation derived in [13] :

$$\dot u_e = A_1 \cdot sh[a \cdot (\alpha \cdot \sigma_i^2 / \nu_0 - u_e^{cp})], \tag{8}$$

where A_1 , a , α and ν_0 are parameters calculated below; σ_i is the deviatoric part of the stress tensor:

$$\sigma_i = M_R \cdot \sigma; \tag{9}$$

M_R is the coefficient of impact equivalence of static and cyclic stresses. For static stresses $M_R = 1$. At cyclic loads [18]:

$$M_R^2 = \sigma_T \cdot (65 + 0{,}46 \cdot HV) / \sigma_R^2 \tag{10}$$

σ_T , σ_R are the yield stress and endurance limit stress of the material; $u_e^{cp} = (u_{e*} + u_{e0})/2$ is the mean value of hidden energy accumulated for all the period of deformation up to the moment of failure.

$u_{e*} = \Delta H_{TB} - \Delta u_T$ is the critical value of hidden energy density of defects u_{et} at the instant of failure t_*.

However, because of mathematical difficulties, the authors simplify the equation (8) changing the hyperbolic sine by an argument expression for conditions of multi-cycle component loading and by an exponential function for low-cycle component loading. Besides, the equation (8) loses its meaning for the range of stresses σ_i , where the argument of the hyperbolic sine is negative, which contradicts to practical operation and to the basic principles of kinetic concept [12].

The authors believe that, in order to adjust the differences and to take into account resistance u_e^{cp} of the strengthening material structure in all the range of values of operating stresses according to Le Chatelier principle, the equation (8) can be modified in the following way.

If in equation (8) $u_e^{cp} = 0$, then we will obtain an expression for estimation of the rate $\dot{u}'_e$ of energy density accumulation of structural defects without taking account its strengthening [20]. Then at the failure instant t_* the energy density of the defects will achieve the value of $u'_{e*} = \dot{u}'_e \cdot t_*$, which will exceed the true critical value of density u_{e*} by $u_e^{cp} = (u_{e*} + u_{e0})/2$, that is, it will be equal to:

$$u'_{e*} = u_{e*} + (u_{e*} + u_{e0})/2. \qquad (11)$$

It follows from this, that the coefficient of rate reduction $\dot{u}'_e$ to the value of $\dot{u}_e$, taking into account the resistance of material structure can be determined by the following relationship:

$$K_C = \frac{\dot{u}_e}{\dot{u}'_e} = \frac{u_{e*} - u_{e0}}{u'_{e*} - u_{e0}} = \frac{2 \cdot (u_{e*} - u_{e0})}{3 \cdot u_{e*} - u_{e0}}. \qquad (12)$$

In this case, the specified equation (8) can take the following form:

$$\dot{u}_e = K_C \cdot \dot{u}'_e = K_C \cdot A_1 \cdot sh\,[a \cdot \alpha \cdot \sigma_i^2 / v_0]. \qquad (13)$$

Parameters comprising equation (12) are determined according to the recommendations [13, 18].

Parameter A_1 characterizing damageability rate as a result of the impact of the ball portion of stress tensor σ_0 :

$$A_1 = \frac{2 \cdot k \cdot T_f \cdot U(\sigma_0, T_f)}{h} \exp\left[-\frac{U(\sigma_0, T_f) \cdot V_{am}}{k \cdot T_f} \right]; \qquad (14)$$

$\sigma_0 = M_R \cdot \sigma / 3$ is the ball portion of stress tensor; k is Boltzmann constant; h is Planck constant; V_{am} is the volume of one atom of material; T_f is the mean self-heating temperature of component material. In case of static loading it is equal to the initial temperature $T_f = T_0$;

In case of cyclic loading:

$$T_f = T_0 + \frac{T_* - T_0}{44 \cdot v_0}; \qquad (15)$$

$T_* = T_0 \cdot \sigma_{np}^2 / (\sigma_{np}^2 - \sigma_i^2)$ is the heating-up temperature of local, highly loaded volumes of material; $v_0 = T_0 / 870$ nonuniformity coefficient of internal energy density in terms of volume of component material in its initial state;

$U(\sigma_0, T_f)$ is the activation energy of the material damage process for the predetermined load conditions:

$$U(\sigma_0, T_f) = U(p_{T0}) - \Delta U_T - \Delta U_{\sigma 0}; \tag{16}$$

$U(p_{T0}) = -2{,}05 \cdot 10^{-8} T_0^3 + 5{,}055 \cdot 10^{-5} T_0^2 + 0{,}0103 T_0 + 10{,}43$ is the initial activation energy of the damage process taking into account thermal pressure at $T = T_0$ and $\sigma = 0$;

$\Delta U_T = 3 \cdot \alpha_0 \cdot K(T_f) \cdot T_* / 2$ is the change in the activation energy at the temperature T_*;

α_0 is the linear expansion coefficient;

$K(T_f) = E(T_f) / (3 \cdot (1 - 2 \cdot \mu(T_f)))$ is the modulus of volume elasticity of the material at the temperature T_f;

$E(T_f) = E \cdot e^{-0{,}0007 \cdot T_f}$; $\mu(T_f) = \mu \cdot e^{-0{,}0005 \cdot T_f}$ are the modulus of linear elasticity and Poisson ratio of the material;

$\Delta U_{\sigma 0} = \beta \cdot \sigma_0^2$ is the change in the activation energy of the ball portion of stress tensor;

$\beta = \varphi_\sigma^2 / (2 \cdot K(T_f))$ is the coefficient of strain energy due to change of volume;

$\varphi_\sigma = k_\sigma \cdot v_0^{0,5}$ is the coefficient of overstress on interatomic bonds.

Coefficients determining the distortion strain energy a and α in the equation (12):

$a = v_0 \cdot V_{am} / (2 \cdot k \cdot T_*)$; $\alpha = \varphi_\sigma^2 / (6 \cdot G(T_f))$;

$G(T_f) = E(T_f) / (2 \cdot (1 + \mu(T_f)))$ is the modulus of transverse elasticity of the material at the temperature T_f;

The set of equations (7) - (15) represents a failure pattern of loaded machine components in predetermined stationary operating conditions. The model was used to develop an algorithm and an application "Operating life DM", which make it possible to estimate their design service life in static load conditions, multiple-cycle fatigue and low-cycle fatigue, contact loading, bearing failure, creeping, creep-rupture strength and others. The comparative analysis of the design service life of various kinds of the design/structure of components under study allows design engineers to provide the service life specified in the specification for such components.

The application interface is shown in Fig. 1, where beside loading diagram, kind of cycle, material of the component and its properties, one can see the calculation of its mean operating life $t_{cp} = t_*$ in graph form – in the form of a nomographic chart. Its magnitude is shown on the axis of ordinates indicated by arrows, by the values of maximum normal σ, equivalent $\sigma_{экв}$ or shearing τ (increased by 1.75 times) stress and initial temperature T_0.

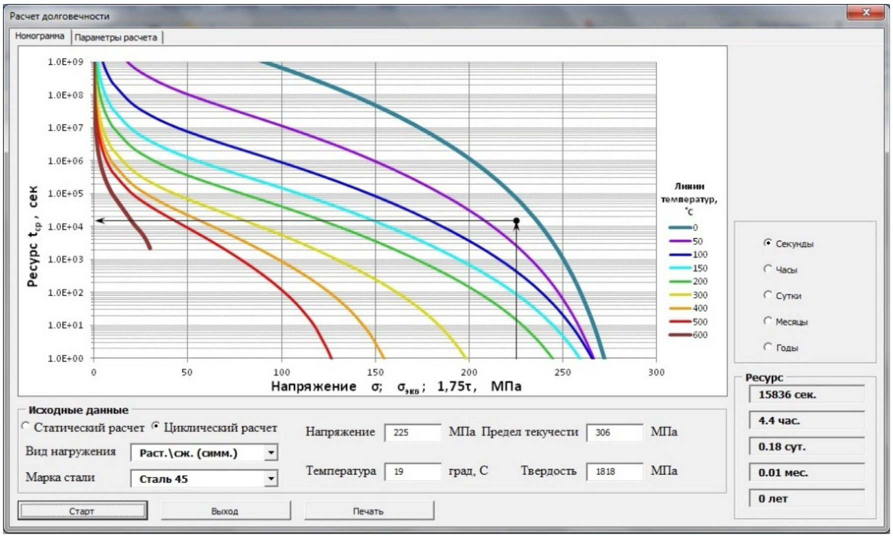

Fig. 1. Application interface "Operating life DM"

3. Verification of theoretical results

To estimate the reliability of the offered methods, the research group compared the design t_*^p and the experimental $\bar{t}_*^э$ values of the sample service life (Table).

Table Comparison of experimental and design data

№	Cycle amplitude σ , MPa	Design service life t_*^p , h	Experimental service life $\bar{t}_*^э$, h	Computational/Theoretical error, δ , %
1	225	4,40	5,69	22,7
2	231	2,68	2,96	9,19
3	239	1,61	1,96	17,7
4	248	1,16	1,31	11,6
5	253	0,53	0,62	14,5

Empirical results were obtained when the research group studied the process of damaging and fatigue failure of samples made of annealed steel 45(o) in the symmetrical compression – expansion cycle under various loading conditions described in [13]. The design values were calculated using equation (7) taking into account (8) - (15). The result of the design service life estimation of the sample №1 was given in Fig. 1 in the form of a nomograph. The service life of the sample equal to 4.4 hours was determined for the specified kind of load, component material and its properties for the values of the operating stress $\sigma = 225 MPa$ and the initial test temperature $T_0 = 19°C$ in the direction of arrows in the nomograph field.

The error range of the operating life calculation for the samples under study is up to 25%, which proves the validity of the theoretical work. The suggested method of service life forecasting makes it possible to carry out comparative analysis of service life of various design variants of the highly loaded components at the stage of component design and can provide the necessary level of their reliability and durability specified in the technical assignment. It allows the design engineer to avoid extra model and field tests of samples and to cut material costs, financial expenses and time necessary for development.

4. Practical application of the analytical method

The method of analytical forecasting and providing of the service life of machine components was used to develop the project justification of retrofitting the rotary actuator group of one of the clinker kilns of Magnitogorsk cement plant. The retrofitting was necessary due to simultaneous failure of №5 kiln actuator anchor screws subjected to elongation after four years of failure-free operation. Analytical calculations carried out in accordance with the offered method showed that:

- maximum operating tensile stress in the screw shanks was $\sigma = 104 MPa$, the safety margin was $n = 2{,}1$ for mean values of operating temperatures $T_0 = (25...28)°C$;
- the design period (calculated by equations 7) until failure of the actuator tension bolts for the specified loading conditions was: service life - $t_*^p \approx (3{,}19...4{,}05)$ years , operating life for $\Pi \approx 0{,}13$ - $t_{сл*}^p \approx (3{,}6...4{,}6)$ years, because at the end of this period, energy density of defects in the material of the damaged components reached the critical value by condition (3).

As a result of comparative analysis of loading of all actuators of the enterprise, it was proved that №5 kiln actuator operated under the most severe conditions because in the process of the kiln installation the actuator had to be transferred to the opposite side of the kiln (compared with the design) due to the lack of space. To eliminate the overload, it was offered to change the direction of the kiln rotation to the opposite. Thus the maximum operating stress decreased from 104 MPa to 15 MPa, while the design service life increased to $t_*^p > 10^2$ years.

The actuator mechanism has been in operation for almost three years since its retrofitting was completed.

5. Conclusions

1. The research group carried out analysis of well-known failure patterns of machine components and came to the conclusion, that their application at the design stage requires extra field research and increases the cost of development.

2. The authors developed a kinetic model of failure development process for loaded elements as well as an algorithm and an application for analytical estimation of their design service life and operating life for different conditions of external loading without carrying out experiments.

3. They proved the reliability of the theoretical method, which makes it possible to provide the level of durability specified in the technical assignment for the most critical machine components by the energy criterion of material strength at the design stage.

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