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MATERIAL CONSUMPTION CRITERIA OF PISTON – CON – ROD ELEMENTS SYSTEM OF COMBUSTION ENGINE

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Abstract

The following article includes the analysis of mechanical and thermal load criteria of combustion engine. Those criteria have been divided into tribological, technical in detail and technical in general. The above criteria have been related to piston-con-rod system, in which, two tribological centres were specified. One centre included, piston-piston rings and cylinder, in which, piston rings serve as labyrinth seal and the cylinder constitutes piston guides. Crank bearings and main bearings of engine crankshaft constitute the second centre. From the point of tribological view, piston-piston rings and cylinder constitute slide and thrust bearing lubricated hydrodynamically, while crankshaft bearings a-c-t as slide and journal bearings also lubricated hydrodynamically. In the description of thermal load of the above-mentioned tribological centres, special attention was paid to criteria that ensure their motion reliability and durability of engine operation. Whereas in the description of mechanical load of elements constituting investigated centres, some criteria were determined resulting from their functionality and fatigue strength of the material they were made of. Criteria of thermal loads were based on mean density flux of heat conducted through the walls of engine elements, paying attention to their low precision of evaluation. It has been emphasized that synthetic criteria indicators are the most precise. They are determined on the basis of engine load measurements during its operation. Those criteria allow us to describe the operating process of con-rod-piston diesel engines and at the same time it is possible to expose their technical condition which is of essential practical meaning. The wear process of tribological centres that takes place during operating in piston-con-rod system, affects the course of engine working process and its techno-economic indicators.

Keywords: *criteria of mechanical and thermal load of the engine, tribological criteria of piston-crank combustion engine system*

1. Introduction

Elements of piston-crank systems are subjected to mechanical and thermal load during engine operation. It causes their wear and tear.

Acknowledgment criteria of elements consumption in piston–crank system, accepted as permissible, result from the following requirements:

– tribological – preserving the conditions of normal process of consumption in friction couples and their motion reliability,
– technical in detail – correct operation of an engine,
– technical in general – the ability of function realization, resulting from engine operation [2, 3, 7].

Piston – piston rings and cylinder of combustion engines constitute, on the one hand, labyrinth seal of the cylinder working area, and on the other hand, longitudinal slide bearing, lubricated hydrodynamically. The above centre, as a labyrinth seal must ensure the tightness of working area of cylinder and thus prevent scavenge of exhaust gases under the piston. Whereas, as a slide bearing, the above centre, should be characterized by minor friction values, taking part in fluid friction [2, 5, 6, 8].

Main bearings and the crankshaft ones are slide journal bearings. Their friction force depends on physic-chemical qualities of oil, relative speed of friction surface, and the shape of oil slit [7, 8].

2. Tribological criteria

The elements of piston – crank system of an engine operate in the conditions of fluid friction. A change of these conditions is connected with the change of increase of the pressure on the friction surface which is equal to speed increase of temperature. It means that speed and the character of temperature changes of friction surface, constitute the reliability criteria of engine motion. The above criteria, according to Aue [1] is written down in the following way:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = K \cdot \frac{\mathrm{d}T}{\mathrm{d}t}, \tag{1}$$

where:
$\mathrm{d}P/\mathrm{d}t$ – speed increase of pressure on friction surface,
$\mathrm{d}T/\mathrm{d}t$ – speed of temperature increase of the friction surface,
K – coefficient of proportionality.

The criteria (1) shows that hydrodynamic pressure determining load capacity of oil film in engine cylinder depends only on relative speed of cooperating surfaces of the ring and the cylinder, and does not depend on piston clearance in the cylinder and oil viscosity.

It means that with excessive clearance values, in spite of an increase of engine rotational speed, the oil wedge may not come into being. In this case fluid friction transforms into a limiting one. It causes an increase of intensity of cooperating surfaces wear and tear.

In conditions of lubrication destabilization, there can follow a seizure between cooperating surfaces. The seizure, as a rule, is the result of scarce lubrication or can be caused by exceeding of permissible load.

The surface of friction undergoes seizing when its temperature reaches melting temperature of the material it was made of [4, 7, 8]. Therefore the criteria of seizing danger has been determined by the melting temperature of the friction surface and expressed by the following formula:

$$T_{top} = \frac{p \cdot c_{av} \cdot \mu}{k} + T_0, \tag{2}$$

where:
p – unitary pressure in [Pa],
c_{av} – medium speed of sliding in [m/s],
μ – coefficient of friction in [–],
k – heat transfer coefficient in [W/(m^2·K)],
T_0 – absolute temperature of one of cooperating surfaces in [K].

The dependence (2) shows that seizing danger rises with an increase of unitary pressure p also material association, characterized by a friction coefficient μ.

Fluid friction and mechanical load caused by surface pressure and its dynamics of change, influence considerably the ability of load transfer by main bearings in defined conditions of operation.

Fluid friction is conditioned by minimal thickness of the oil film in the bearing and greater than critical value of Hersey parameter of the formula:

$$\lambda = \frac{\eta \cdot n}{p_{med}}, \tag{3}$$

where:
η – dynamic viscosity coefficient of lubricating oil in [N·s/m^2],
n – rotational speed of engine crankshaft in [rps],
p_{med} – medium pressure in bearing determining the state of thermal balance [Pa].

Pressure values on the bearing surface should be lower than permissible value. They constitute the criteria of mechanical load of the bearing expressed by:

$$p_{med} \le p_{permis}, \tag{4}$$

where p_{med} and p_{permis} is, respectively, medium and permissible pressure on cooperating surfaces of the bearing in [Pa].

The criteria expressed by the formula (4) are connected with the bearing stiffness and the material it was made of [7].

Operating reliability of the crankshaft bearings depends also on thermal load, which causes temperature increase in the zone of friction and temperature gradient on the surface and also on the thickness of the bearing walls. Values of temperature and its gradient should be lower than permissible and they constitute criteria of the bearing thermal load.

Radius speeds of crankshaft pins whose values should be lower than permissible in the bearing, because of cavitation and fatigue damages, constitute criteria of cavitation resistance and bearing fatigue, which can be determined from the inequality:

$$\frac{\mathrm{d}h}{\mathrm{d}t} \le |a|, \tag{5}$$

where:
$\mathrm{d}h / \mathrm{d}t$ – radius speed of the pin in [m/s],
$-a$ – negative values indicate bearing resistance to cavitation in [m/s],
$+a$ – positive values indicate bearing resistance to fatigue in [m/s].

The quantity a appearing in the formula (5) depends on the pressure value in the oil film and with an increase of this quantity, maximal pressure of bearing lubricating oil, increases too [4].

The above criteria belongs to exploitation criteria determining durability of the bearing, conditioned by intensity of wear and tear processes of bearing shell and pin.

3. Technical criteria in detail

Engine elements during operation undergo natural wear, which after achieving permissible values, disturb its correct functioning.

Tightness criteria of engine combustion chamber is determined by the temperature of lakes isolation from cylinder oil. This temperature is called a criterion of lakes creation and expressed by the following dependence:

$$K_L = \frac{T_{lak}}{T_p} > 1, \tag{6}$$

where:
T_{lak} – temperature of lakes isolation from cylinder oil in [K],
T_p – surface temperature of cylinder liner over the first sealing ring in [K].
The criteria of lakes creation determines conditions of oil lubrication [7].

Pressure criteria of combustion chamber tightness is determined by a dependence:

$$p_{comp} \le p_{st\,comp}, \tag{7}$$

where:
p_{comp} – air compression pressure in engine cylinder with piston position in TDC in [Pa],
$p_{st\,comp}$ – standard value of air compression pressure in an engine in [Pa].

Pressure differences of the air compressed in an engine cylinder expressed by formula (7), indicate the stage of tightness loss by piston rings.

Part of exhaust gases gets into under piston space and causes an increase of pressure. Therefore, the amount and overpressure of exhaust gases penetrating the space under the piston is known as the criteria of combustion chamber tightness [8], expressed by the formulas:

$$\Delta p_{exhgas} > \Delta p_{stexhgas}, \tag{8}$$

where:

Δp_{exhgas}, $\Delta p_{stexhgas}$ is overpressure of exhaust gases in sub-piston area and its standard quantity in the conditions of engine operation, when the standard value was measured in [Pa].

Attention should be paid to the fact that the highest influence of gases blown into the sub-piston space, except immobilizing of piston rings, has the clearance value of the first ring joint. This clearance depends on the wear stage of the cylinder liner [3].

4. General technical criteria of an engine

General technical criteria define combustion engine operation in a limiting state. Among them there are thermal and mechanical criteria. Thermal and mechanical criterion of the engine load is defined by the K_e quantity, supported on a medium thermal density flux on cooled surfaces of cylinder liner, cylinder head and the piston, during nominal load of the engines of the same type, expressed by the formula [3, 5]:

$$K_e = \frac{p_{eav} \cdot c_{av}}{z}, \tag{9}$$

where:

c_{av} – medium speed of an engine in [m/s], $c_{av} = (S \cdot n)/60$, where S is the piston stroke in [m] and n is the engine speed in [rpm],

p_{eav} – medium effective pressure in [Pa],

z – number of ignitions in cylinder per one rotation of the shaft: $z = 1$ for two-stroke engines, and $z = 1/2$ for four-stroke engines.

Quantity K_e characterizes mechanical load, because with an increase of effective pressure, maximal combustion pressure rises too. In addition, with an increase of piston medium speed, inertia forces of movable parts of the engine increase too [5].

Quantity of criteria K_e is proportional to piston power indicator, expressed by the following dependence:

$$K_{pist} = \frac{N_{ecyl}}{F_{pist}} = \frac{z}{2} \cdot c_{av} \cdot p_{eav}\,, \tag{10}$$

where:

F_{pist} – area of piston crown in [m^2],

N_{ecyl} – effective power of one piston-crank assembly in [W],

remaining designations as in formula (9).

Piston power indicator (10) defines heat flux affecting the unit of piston area in a time unit. By way of statistic investigation, one can fix, so called, corrected piston power indicator of the following form:

$$K_{spist} = \frac{N_{ecyl}}{F_{pist} \cdot \sqrt{\frac{S}{D}}}, \tag{11}$$

where:

S – piston stroke in [m],

D – piston diameter in [m],

remaining designation as in formula (10).

Formula (11) enables us to compare maximal thermal load in engines of different forms of piston displacement, characterized by the proportion of displacement to piston diameter.

Simple and popular criterion of piston thermal load is, so called, a criterion of Gincburg [2, 3] of the following forms:

$$K_D = \frac{N_e}{D}, \tag{12}$$

where:
N_e – effective power of one piston-crank assembly in [W],
remaining designations as in formula (10).

Formula (12) is achieved on the basis of the hydro dynamical similarity theory, assuming that the quantity which determines piston thermal load, is nothing else, but temperature differences in its optional points. In formula (12) the quantity of effective pressure is characterized by the power given to the outside.

To evaluate vulnerability to piston heads cracking in similar engines, one can use a simplified criterion of thermal damages, introduced by Morland in the form of the following parameter:

$$K_p = p_{eav} \cdot S \cdot n \cdot \sigma , \tag{13}$$

where:
σ – thickness of the wall, through which thermal flux is passing in [m],
n – rotational speed of the engine in [rps],
remaining designations as in formulas (10) and (11).

Thermal load criterion, basing on, interdependence between heat-flux density, passing through the walls of the engine, and some parameters of combustion process, was expressed by Kostin [3, 4, 8] in non-dimensional form:

$$K_k = B \cdot c_{av}^{0.5} \cdot \left(p_{eav} \cdot q_e \cdot \frac{T_d}{T_0} \right)^{0.88} \cdot \left(\frac{D}{p_d} \right)^{0.38} , \tag{14}$$

where:
B – constant value: 5.73 for four-stroke engines, and 10.2 for two-stroke engines,
c_{av} – medium rotational speed of an engine [m/s],
D – piston diameter [m],
p_d – pressure of supercharging air in [MPa],
T_d – absolute temperature of supercharging air in [K],
T_0 – temperature of neighbourhood, $T_0 = 293$ K,
q_e – fuel consumption unit [kg/kWh],
p_{eav} – medium effective pressure [MPa].

Examination results of many engines are confirmed by linear dependence of temperature, in different points of piston crown, on non- dimensional parameter of thermal load. Dependences (14) are used by Norwegian Marine Institute, as a criterion of thermal load NSFI of the form [8]:

$$T_i = A + B \cdot \frac{T_d^{1.25} \cdot p_{iav}^{1.58} \cdot n^{0.35}}{p_d^{1.2}} , \tag{15}$$

where:
T_i – temperature of selected point of combustion chamber in[K],
n – rotational speed of an engine in [rps],
A, B – constant coefficients determined experimentally, for a given point of combustion chamber,
remaining designation as in formula (14).
Formula (15) allows us to determine temperature in an optional point of combustion chamber.

With a defined intensity of cooling, medium temperature of cylinder liner is proportional to a medium density of heat flux, conducted by, the above mentioned, cylinder liner. This fact permits to determine the thermal load of engine cylinder liner [3], by means of, the following coefficient:

$$K_c = \frac{p_{iav} \cdot n \cdot \sigma \cdot T_d}{\eta_v \cdot p_d} , \tag{16}$$

where:
σ – thickness of cylinder liner wall in [m],

η_v – coefficient of filling the engine cylinder with air in [(m·K)/s],
remaining designation as in formula (15).

Formula (16) is often used as a synthetic indicator for a quick evaluation of cylinder liner consumption.

Some producers, like for example, Sulzer, recommend for their products to use comparative criteria to evaluate thermal load of the engine [4, 5] of the following form:

$$K = \frac{p_{eav}}{p_d + p_b}, \tag{17}$$

where:
p_b – barometric pressure of the neighborhood in [Pa],
remaining designation as in formula (14).

Calculating the K-quantity from the formula (17), standard data, and comparing it with a calculated one, during measurement of the engine in operation, with a similar load, one can evaluate a general stage of thermal load.

Mechanical criteria include stress criterion, which is determined by the stress ratio permissible for the material in engine operation temperature, to stresses in its walls, at a given load [3]. This criterion is expressed by formulas:

$$K_{mat} = \frac{\delta_{T_d}}{\delta_T} = \frac{2\delta_{Td} \cdot \lambda_s (1-\mu)}{\beta \cdot E \cdot \sigma \cdot q} \geq 1.1, \tag{17}$$

where:
β – coefficient of linear expansion of engine parts material in [1/K],
E – module of transverse elasticity of material in [Pa],
μ – Poisson coefficient in [–],
σ – thickness of engine parts walls conducting the heat flux in [m],
λ_s – coefficient of heat conduction through engine parts material in [W/mK],
q – density of heat flux in the wall of the engine part in [W/m^2],
δ_{Td} – permissible stress of a given material of the engine part in a defined temperature in [Pa].

The greatest gas force occurs in piston position about TDC before the stroke of work, and constitutes one of the quantities that defines mechanical load of the engine [5]. For comparative aims as a criterion indicator of mechanical load by gas force, the following product is used:

$$K_g = p_{eav} \cdot D, \tag{18}$$

where designations as in formula (14).

Besides, mechanical load of piston –crank system depends on dynamics of combustion pressure increase, from the moment of ignition to obtainment of maximal value [3]. For this reason, criterion of mechanical overloading in piston – crank engine , is expressed by the formula:

$$K_m = \frac{\Delta p_s}{\Delta \alpha} > K_{mperm}, \tag{19}$$

where:
$\Delta p_s / \Delta \alpha$ – speed of pressure increase in engine combustion chamber in [Pa/HVAC],
K_{mperm} – permissible speed of combustion pressure increase which does not cause mechanical overload,
α – angle of rotation of the engine crankshaft in [HVAC].

Torsional vibrations caused by the moment, constitute additional source of torsional stress of crankshaft dynamic load, affecting parts of engine piston-crank system, which can exceed permissible value [4]. It permits to formulate a criterion of torsional vibrations:

$$\frac{16 \cdot M_{\max}}{\pi \cdot d^3} < \tau_{perm}, \tag{20}$$

where:
d – diameter of crankshaft pin in [m],
$M_{\max}$ – the highest amplitude of rotational moment change in [Nm],
τ_{perm} – permissible contact stress in engine crankshaft in [Pa].

5. Summary

General technical criteria should be approached critically, because on their basis, one can sometimes draw misleading conclusions, concerning engine load. Engine loads are defined by values describing, at the same time, its technical conditions and operating as well, which is of essential practical meaning.

In high-pressure engines, operating wear of piston rings and cylinder cause worsening of tightness in combustion chamber space.

Thermal loads criteria are based on dependences between density of heat flux passing through the walls of engine parts and parameters of its work.

A criterion defined by formulas (12) can be used as a thermal load parameter, only on condition, when differences of mechanical efficiency in comparable engines will be of not higher difference, than 5%.

The quantity of non-dimensional thermal load parameter (14) permits us to define changes of piston temperature during operation.

An increase of calculated comparative quantity K from the formula is the evidence of engine thermal load rise, and its worse technical condition due to wear of its parts.

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