CONDITION IDENTIFICATION IN SERVICE AND MAINTENANCE OF MARINE ENGINE

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Abstract

Marine machine sets with combustion engine have to fulfill requirements of safety, quality and efficiency. It is reflected in all elements of maintenance: conservation, inspections and repairs. Main points of interests are repair technologies and all types of condition identification: estimation of free element quality, elements before and after repair, estimation of assembling quality and diagnosis. In case of marine engines does not exist any offer of system of identification, which fulfill requirements, especially safety requirements.

There exist necessary to analyze, development and systemize methods of identification and generate a system of identification for machine set with machines with piston – crank mechanism. In the papers will be presented uniform classification of deviations include deviations of free element, deviations of elements built in set/machine set and diagnostic symptoms.

There will be presented, specific for ship machine sets with engine with crank-piston mechanism, symptoms and deviations, and relations between them:

- *it will be presented model based on measured spectrum of axial vibrations and measured spectrum of loads on engine cylinders. This model was generated to condition monitoring of crank shaft position;*
- *it will be shown deviations and symptoms of tribological junction condition and diagnostic models based on defined and measurable deviations and vibration symptoms. Generated model will allow to monitoring the condition of connecting-rod head bearing, crosshead bearing, connecting-rod big end bearing and main bearings.*

Keywords: *crank-piston mechanism, deviations, axial vibration, condition monitoring*

1. Introduction

Production systems are designed to satisfy requirements concerning quality, efficiency and safety. These requirements have been formalized to a large extent. It particularly concerns the requirements of safety which, among others, are regulated by international conventions. Non-fulfillment of those requirements, mainly formal, puts a system into inability condition.

The machine set is the smallest production system. Machine elements (components) are subject to wear. An element whose wear reached the limit value, is considered as damaged, and the whole machine set as a system is in inability condition.

A damaged element may be replaced by a spare element i.e.:

– a newly-produced element made as a spare element;
– an element after repair or reconditioning offered as a spare element;
– or it may be repaired, that requires it’s dismounting and mounting.

The choice of a spare part is determined by: time of access to part, price of part, quality of part (including reliability).

The decision whether to repair or recondition a particular element has to be justified in economic and technical terms. The costs of repair or reconditioning have to be lower than the cost of a new element, and the quality of the repaired or reconditioned element has to be comparable with the quality of a new element.

The instant of time of the element exchange should slightly precede the instant of time of the element failure.

2. Identification of elements

Advisable operations on machine set elements may be reduced to the stabilization and prolongation of life time and to due exchange of damaged elements. The necessary requirement for these operations is the assessment of element quality – identification of element condition in all the phases of its life. The identification can refer to:

- free element condition: new, damaged, repaired;
- mounted element condition (assembled in a machine);
- condition of an element in operation.

Identification requires measurable magnitudes, which describe the condition of an element. Also, there should be a correlation between the magnitudes used for the description of the condition in different phases of element life. This correlation is indispensable for effective maintenance of machine sets.

3. Magnitudes for describing condition of element

Magnitudes for describing the condition of a free element may be called deviations. These may be the effects of such processes as the solidification of alloy, plastic forming, machining, putting a surface layer etc. Various groups of deviation can be pointed out. Those referred to as elementary ones are as follows:

- deviations from nominal chemical structure,
- deviations from homogeneity of structure (macrostructure),
- deviations from nominal internal stresses,
- deviations from nominal linear and angular dimensions,
- deviation from nominal shape and roughness.

Elementary deviations are identified with methods derived from material sciences, nondestructive testing and metrology.

When one or more elementary deviations occur, they may cause appear or rising of undesired characteristic/property of an element. Magnitudes describing such characteristic, may be called complex deviations. These deviations may include:

- run-out of shafts and impeller discs,
- unbalance of shafts and impeller discs,
- leak of tightness,
- mechanical characteristics, e.g. transfer function.

During the assembly of machines, elements are collected into sets; stationary and movable connections are made. The essence of stationary connections, (excluding shape connection) is that there are generated in them states of stresses with designed values and in described directions. These stresses should ensure recommended stiffness and leak tightness of connections.

When a connection is improperly made, the result is, that the connected elements do not have mutual positions as designed, the elements are deformed or undesired internal stresses occur. There occur deviations of run-out and unbalance of impellers, deviations of leak tightness of sets, deviations of mechanical properties.

In movable connections the most important is to get a designed clearance and mutual position of one movable element relative to the other or to the machine casing. The quality of assembling and effects of assembling may be assessed by making measurements of appropriate elementary or complex deviations. It is advisable to use indirect deviations: clearances (negative allowances) and deviations of alignments of movable elements. Indirect methods are often methods specific for each type of mechanism.

In running machine set machines improper clearances, improper alignments of impeller shafts and unbalance of impellers cause the increase of bearing load, additional reactions in a bearing, additional bending moments which affect impellers shafts. These forces and moments add to operational loads and generate additional vibrations of the machine set. Deviations of leak tightness and deviations of mechanical properties may result in a failure to reach satisfactory characteristics, especially in terms of safety and quality. During machine operation, the unavoidable wear brings about an element failure and the machine failure. The intensity of wear process depends on:

– value and character of a course of operational mechanical and thermal machine loads affecting machine elements;
– stresses, forces and moments which are effects of production and assembling deviations of elements;
– condition of lubrication and cooling fluids and working media;
– quality/loadability of aggregate elements.

Operating machine sets generate signals, which contain information about technical condition of the aggregate and its elements. Oriented for specified failure, signals descriptors are named diagnostic symptoms. Diagnostic symptoms are defined for a specified type of machine set. Diagnostic inference requires the knowledge of relations between symptoms and deviations of production and assembling of elements.

4. Deviations and symptoms of the position of marine machine set shafts

Supported in bearings, machine set impeller shafts are connected through various types of couplings, from rigid couplings to flexible ones.

It is necessary that:

– axes of bearings that support both shafts be aligned,
– production and assembly deviations of couplings should not be source of additional forces or moments and should not load shafts connected by them.

For non-crankshafts and rigid flange couplings, the above requirements, which partly refer to couplings, may be limited to the requirements: frontal plane of flange should be perpendicular to shaft axis, coupling flange and shaft axis should overlap, holes for connection bolts in both coupling flanges should be made with the same pitch.

The misalignment of two shaft axes consists of two particular deviations: deviation of parallel shift of axes and deviation of axis bend. Both deviations may occur separately or together (twisted axes). The measurement of shift and bend of two shaft axes is performed with pitch-surface generators (shaft fit method).

In case of shafts connected by rigid flange couplings measurements are generally based on pitch-surface generators and frontal planes of coupling flanges (coupling flange fit method). The measurement of misalignment may be done with mechanical accessories, using holders, bars and dial gauges or by optical methods, e.g. by laser systems. A laser system may be used, for example, for the measurement of misalignment between a crankshaft and the shaft of reduction gear. Both shafts were connected by a flexible coupling. Elements of the laser system were mounted on one side of the flying wheel, on the other side on the coupling flange, with special magnetic holders.

During the operation of machines such phenomena as the misalignment of bearings and shafts, additional reaction forces in bearings and additional shaft bending moments may take place. These are caused by unequal wear of support bearings and/or movements of shaft main bearing supports.

In case when one of the machines has a crankshaft, then bearing wear, machine body deformation or misalignment of aggregate shafts will result in crankshaft deflection. The crank deflection Δa is defined as the difference between the value of a distance for two positions of crank in one plane: vertical $\Delta a = a_{TDC} - a_{BDC}$ and in the horizontal plane $a_{SB} - a_{PS}$.

The crankshaft deflection is measured by dial gauges. The measurement of crankshaft deflection requires that the machine be out of operation and the crankshaft must be rotated slowly.

An assumption may be made that the courses of a value between crank arms, depending on the shaft angle position $a=f(\alpha)$ are sine curves.

Distance changes between crank arms mean that one of the main bearing journals or both are moving in the axial direction. If we imagine that the crank is fixed on one side so that one of the main bearing journals may rotate only and the axial movement is blocked, then an assumption can be made: displacement Δl of the free main bearing journal of the crank fixed on one side, as a function of crankshaft angle position is a sine curve, Figure 1.

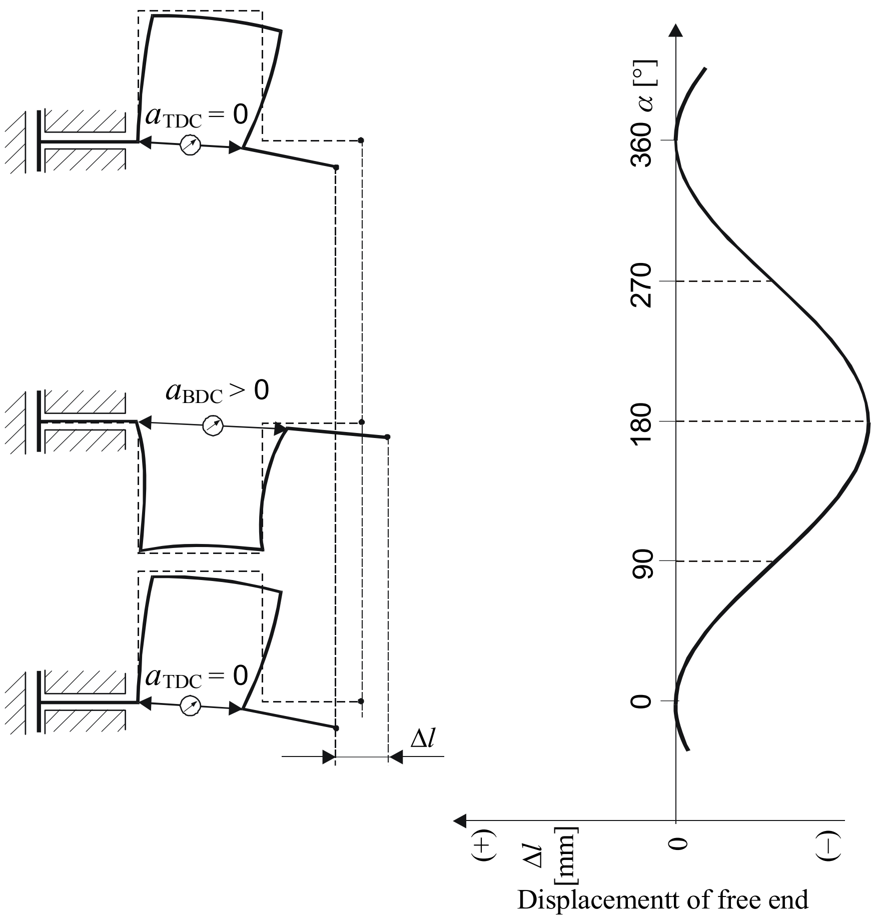

Fig. 1. Negative deflection (–Δa) and course of displacement Δl of crank free end: if $a_{BDC} = 0$, then $(–)\Delta a = a_{BDC}$ and $\Delta l = (–)a_{BDC}$

To see the influence of summed up crank deflections on the total displacement of the free end, let us make a theoretical decomposition of the crankshaft to separate one-side fixed cranks and then to sum up geometrically the sine curves of displacements.

The imagined fixing of the first main bearing journal is justified in reality because:

– first main bearing journal is rigidly connected with a heavy flying wheel and often with a shaft of another machine incapable of generating axial vibrations,
– in direct vicinity of the first journal the thrust bearing or at least fix bearing is located. Both bearings have the property of damping the vibrations and blocking bigger axial displacement of the shaft.

Figure 2 shows the decomposition of shaft with three cranks positioned in one plane with the deflection in the vertical plane (Fig. 2a) and the course of displacements of free ends and their sum (Fig. 2b). The sum of sine curve displacements of each crank is sine curve displacement of the crankshaft free end.

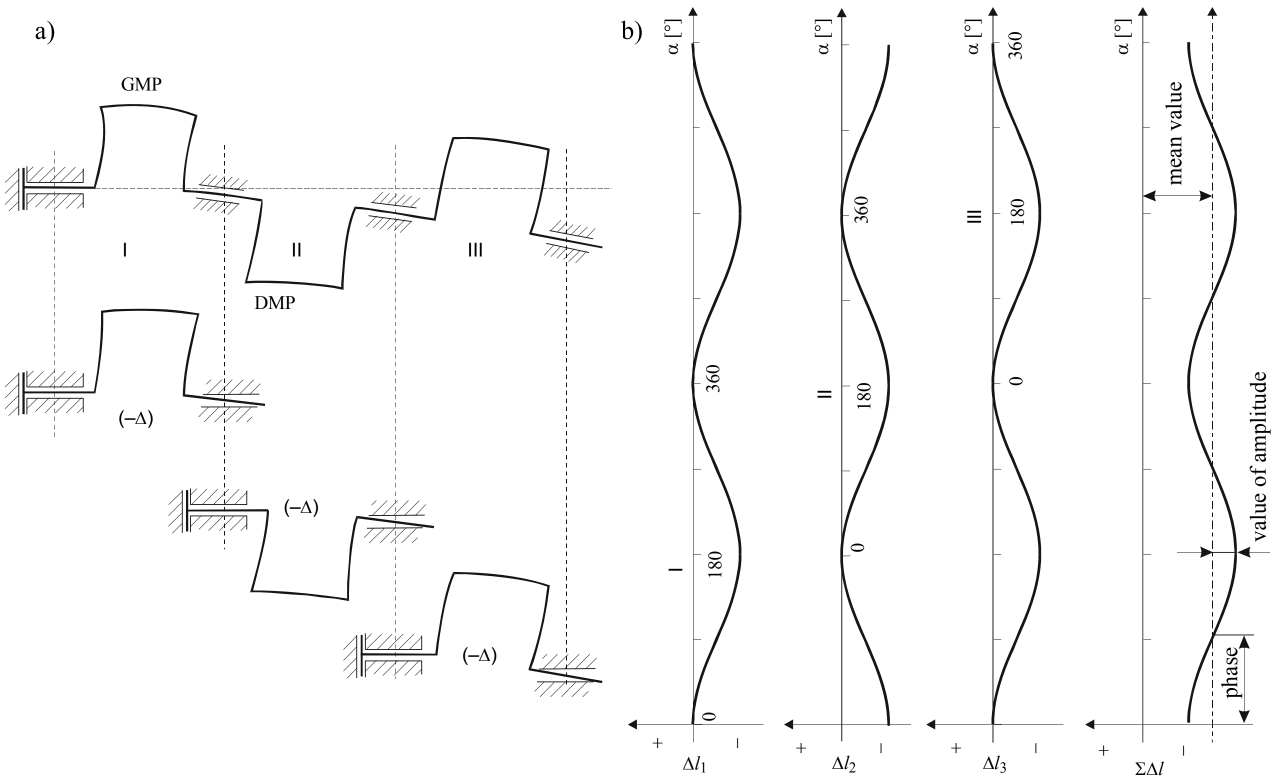

Fig. 2. a) Decomposition of crankshaft deflection of three-cylinder engine shaft (real axis of shaft arc tangential to the desired axis on the first bearing), b) the sum of axial displacements(γ = 180°)

The model thus made gives the sum of sine curves of deflection in vertical and horizontal planes for each crank. Moreover, the geometrical sum of resultant sine curves for all cranks of the shaft gives in effect a sine curve as well, which represents the displacement of free end of the crankshaft.

Important values of this displacement are:

- value of amplitude,
- phase shift relative to TDC of first crank,
- mean value.

During the operation of a machine with the piston-crank mechanism, whose shaft has some deflection, axial displacement of free end caused by that deflection is part of axial vibrations of the shaft. The period of vibration caused by deflection is equal to one revolution, and in an FFT spectrum of vibrations the first harmonic will be dominant (also higher harmonics are possible).

Axial vibrations of piston-crank machines are generated mainly by radial components of mass and pressure forces acting on crankpins of the crankshaft. The spectrum and value of those vibrations depend not only on the load of the whole machine, but also on the distribution of the load of each cylinder. With high inequality of load distribution in the spectrum of axial vibrations generated by pressure forces, there will occur significant values of harmonics with orders lower than the number of machine cylinders; the first harmonics may reach a significant value.

It seems possible to build a reverse model and to draw conclusions about the position of crankshaft, based on the measured spectrum of axial vibrations and the measured spectrum of loads on engine cylinders.

5. Deviations and symptoms of the tribological junction condition in a machine with the piston-crank mechanism

One of the problems typical of large machines, particularly four stroke engines, is seizure of tribological junction in the piston-crank mechanism. This problem may be solved with the use of vibration symptoms.

The effect of wear in tribological pairs is that the clearance increases. In case of a sleeve bearing, the change of force direction acts on the pin, causing the detachment of the pin from one wall of the bearing shell and its movement, within the range of clearance, to the other wall of the bearing shell.

The pin collides with the other wall of the bearing shell.

The force of collision depends on the pin velocity, on the bearing shell stiffness and stiffness of oil film. The stiffness of oil film decreases with increasing of bearing clearance.

In machines with the piston-crank mechanism the forces and moments transferred in tribological junctions have a periodical course; and it is possible for the sign of force or moment to change.

An example of a course of the resultant force *F* acting on the engine piston is shown in Figure 3a.

The force *F* may be decomposed as shown in Figure 3b.

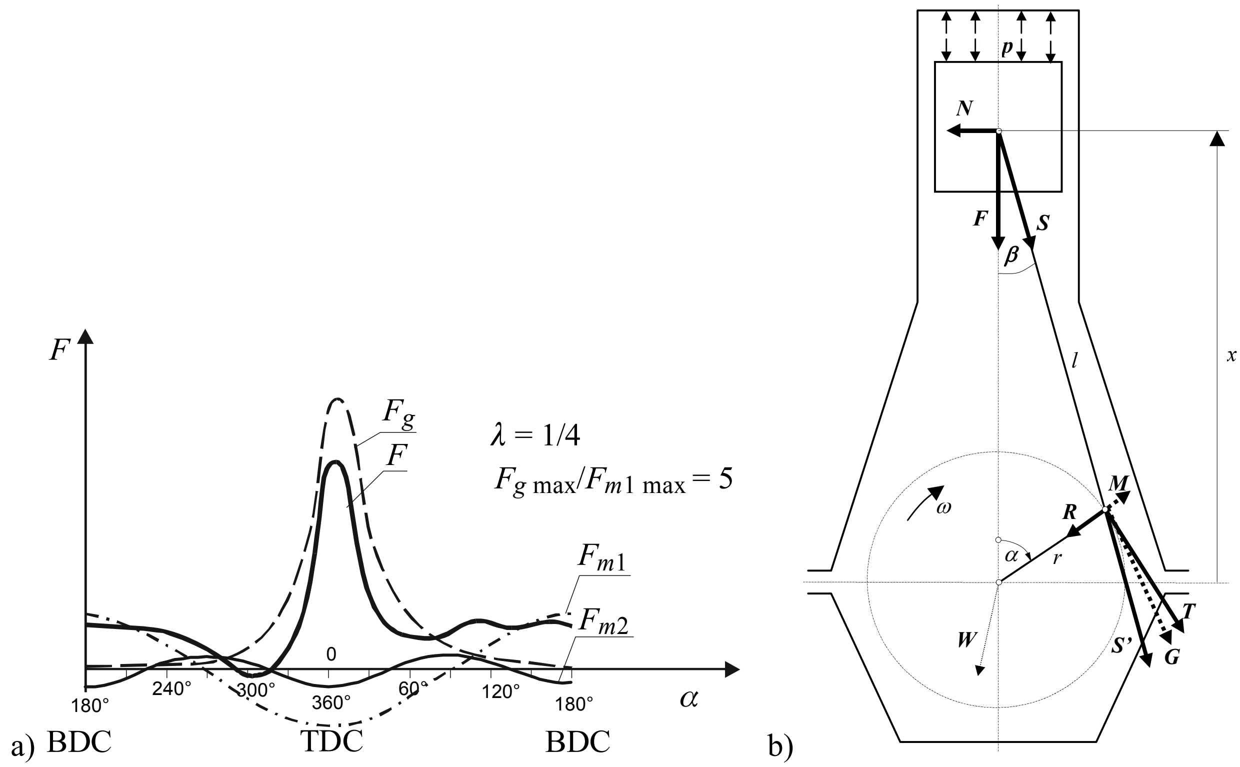

Fig. 3. a) A course of piston force F in a two stroke engine as a function of crankshaft angle position (CAP), b) decomposition of forces affecting the piston-crank mechanism [1]

The analysis of forces acting on the elements of piston-crank mechanism, presented in works [1, 2], shows that for a specific angle position or positions of crankshaft the signs of forces *F, N, S* change (passing through the zero value).

For a mechanism without crossheads, the force *F′* acting through a pin on the bearing shell of the connecting-rod head, is the force *F* modified by bearing clearance, Fig. 4a. The force *F′* generates vibrations of the connecting-rod. After properly decreasing of clearance in the bearing of the connecting-rod head, the resultant force S' acting on the connecting-rod big end is composed of gas forces, mass (inertia) forces of piston, piston pin and part of connecting-rod performing oscillating movement. The clearance in the bearing of connecting-rod big end makes the force S'' *act* on the crank pin. The force S'', modified by clearance, generates axial vibrations related, among others, with crankshaft shape (to put it more precisely, axial vibrations are caused by the radial component R of the mentioned force, Figure 3b). It is possible to measure these vibrations.

a)

b)

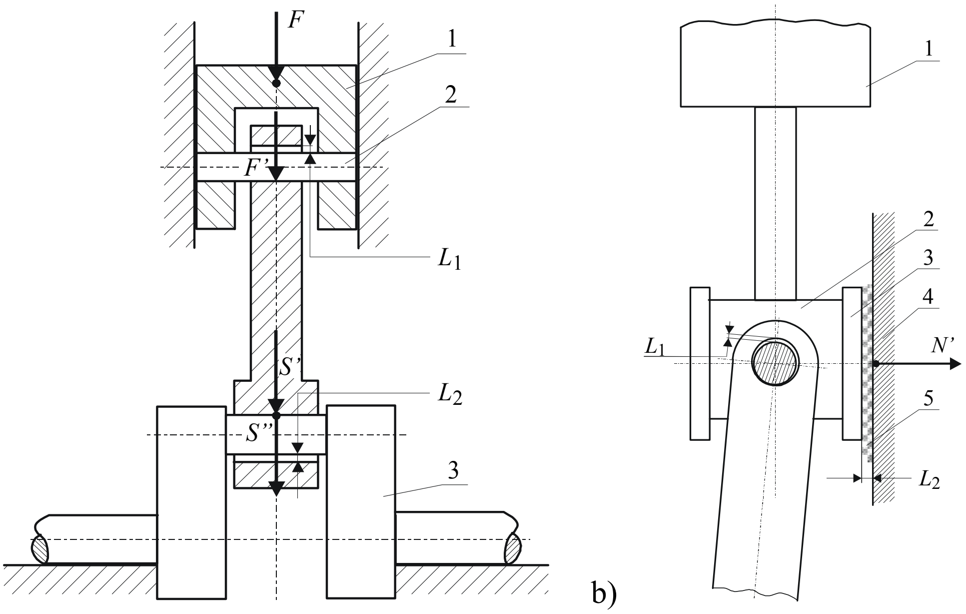

Fig. 4. a) Forces generating vibrations of connecting-rod and crank: 1 – piston, 2 – piston pin, 3 – crank; L_1 – clearance in connecting-rod head bearing, L_2 – clearance in connecting-rod big end bearing, b) forces generating vibrations of slideways: 1 – piston, 2 – crosshead, 3 – slide, 4 – slideway, 5 – separation liquid; L_1 – clearance in connecting-rod head bearing, L_2 – clearance between slides and slideway [1]

The influence of clearances is similar in a crosshead mechanism, Fig. 4a. The forces modified by clearance the in connecting-rod head bearing are gas forces and inertia forces of the piston, piston rod and crosshead.

In a machine without the crosshead, the force N is taken over by the cylinder liner through oil film, mainly between the guiding part of the piston and the cylinder liner. Because the value of force N goes through zero value, the clearance between the piston and the cylinder liner modifies the course of force N, which, modified by clearance, generates vibrations of the cylinder liner.

The application of this effect for diagnostic purposes is problematic because of the following reasons:

- as a rule, the cylinder liner is cooled by liquid on the outer side and it is not directly accessible;
- force F is partially transferred to the cylinder liner through piston rings. There is gas between the piston and piston rings (compressed or exhaust gasses) which, being under very high pressure, has some elasticity and damping properties;
- positioning of the piston pin center (point through which the force N is transferred) in the center of piston gravity does not guarantee equal distribution of thrust in the guide part of the piston.

The positioning of the piston pin axis in the axis of piston guide part symmetrically below the piston center of gravity causes impacts of piston edges on the cylinder liner when the force N changes its sign, (see [3]).

In mechanisms with crossheads the force N acts along in the center line of the crosshead. The force N' acting on crosshead slides is the force N modified by clearance between slides and slideways, Fig. 4b. The dynamic component of force N' generates slideways vibrations. These vibrations can be measured. Because the force N is a component of force F', the course of force N' also depends on the clearance in the connecting-rod head bearing.

The influence of clearance on vibrations was confirmed in practice. The previous research, among others, was focused on the influence of clearance in crank bearings on axial vibrations of crankshaft [1]. It was found that there is an optimal value of clearance for which axial vibrations of the crankshaft are the lowest. When clearance is larger or smaller than the optimal one, axial vibrations increase, and noticeably, for clearances lower than optimal the increase is much higher.

The increase in axial vibrations may be explained as follows:

- optimal clearance is contained in the range of nominal clearance (between minimum and maximum values);
- below nominal clearance, the quantity of lubricating oil flowing through the bearing is insufficient to ensure "steady thermodynamical conditions"; the intensity of oil flow is too small to carry off all friction heat without the rising of oil temperature. Higher oil temperature causes a considerable decrease of oil viscosity and increase (modification) of the force taken over through the crank (rise of axial vibrations). Generally, it should be expected that during seizing the clearance will be rising and lubrication will be insufficient (or lacking) which will bring about a rise of impulse values of forces and a corresponding rise of vibration measures.

Conclusion

1. Deviations and symptoms may be classified in terms of deviation images and in terms of consequences arising for an element or set of elements and the machine. Elementary and complex deviations can be distinguished.
2. It is also possible to classify deviations in terms of the element or aggregate life. It is purposeful to assume a uniform classification of deviations including deviations of free elements, deviations of elements built in set/machine/machine set and diagnostic symptoms. It will facilitate the assessment of element quality during inspection and verification. Besides, it will contribute to better estimation of loadability and safety coefficients.
3. In case of machine sets including an engine with the piston-crank mechanism it is necessary and possible to generate diagnostic models based on defined and measurable deviations and symptoms. The generated model will allow to:
   - make diagnose a connecting-rod head bearing, crosshead bearing, connecting-rod big end bearing and main bearings;
   - make diagnose crankshaft position;
   - assign real forces and moments acting on elements and to forecast fatigue failures of elements [4].

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