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EXPERIMENTAL AND THEORETICAL RESEARCH CONCERNING THE INFLUENCE OF CYLINDERS WORKING PROCESSES ON NEIGHBOR CYLINDERS MAIN BEARING

Ioan Száva[1)], Ioan Curtu[2)], Vasile Ciofoaia[3)], Marius Botiş, Florin Dogaru, Dan Druţă, Laurenţiu Vasu, Petre Hlipcă, Dana Luca-Motoc[*]

Abstract

The entire engines durability generally depends on that of the crankshaft one. Consequently, the necessity of a precise evaluation of its stress-strain state represents an important problem both for designers and for engine manufacturers as well.

In this paper the authors bring in (hold) their contribution to demonstrate that, for a preliminary calculus, the crankshaft can be considered as a single zone, corresponding to a single cylinder, namely a beam (simply supported frame) with two supports only, because the influence of taking into account a great number of subassemblies corresponding for many cylinders has a reduced relevance. The experimental research has been carried out by authors for an 6 in-line tractor engine. That was way an original hydraulic stand has been designed and manufactured. The small displacements of main bearing house were recorded by displacement transducers and the fixing screws force values evaluation have been recorded using electrical strain gauges.

The experimental data were compared with the those obtained within a numerical simulations with FEM. There resulted a negligible influence of the neighbor cylinders on the main bearings solicitations. Due to the fact such an influence is small and can be neglected, one can take into account only one subassembly, corresponding to a single cylinder.

Evidently, for dynamic behavior analysis one must take into account the overall system, as a single part.

Keywords : Crankshaft, Durability, Stress-strain state, Electrical strain gage, Displacement transducer

1. Introductive aspects

Starting of the increased importance of the crankshaft on durability and competitivity of the whole motor ensemble, the authors tried to give their contribution at elaboration of one simplified methodology for crankshaft calculus from statically point of view.

In a usual abordation of this problem, the crankshaft is considered as a 3D devious beam with a number of nedetermination greater than 1, and solving of such a problem, even in hypothesis of perfectly stiffened supports, requires software and computers with enhanced performances or even graphical stations. If however some preliminary calculations are done on computers with low features, then are needed some important simplifications of the initial structure. Such a simplification, often used in papers is to consider only a bend (a zone afferent to one cylinder) as a simply supported beam. It is obvious that such a simplification, even using the hypothesis of short/ narrow bearing can be applied only with a preliminary experimental comparation. It can be mentioned also the fact that every structure so considered has stiffness very different from case to case and from this reason it must exist as much as

[*] "Transilvania" University of Brasov, B-dul Eroilor nr.29, 2200 Brasov, Romania : Tel:00 40 68 412921/171; Fax: 00 40 68 410525 1)eet@unitbv.ro; 2) curtui@unitbv.ro; 3) ciofoaiav@unitbv.ro.

possible the results of the experimental measurements afferent of some constructive solutions alike, from stiffness of the bend point of view as much as from breadth of cover bearing point of view. (divided on cover diameter).

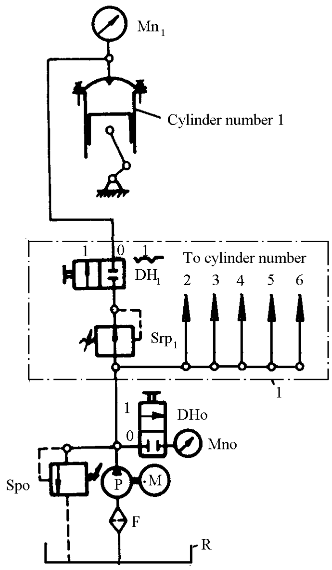


Fig.1

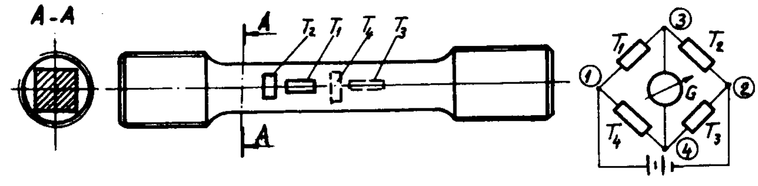


Fig.2

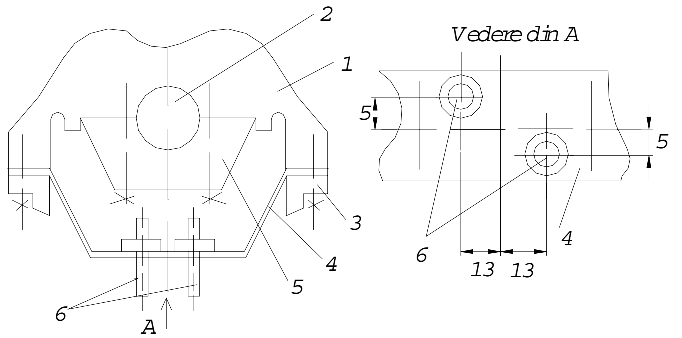


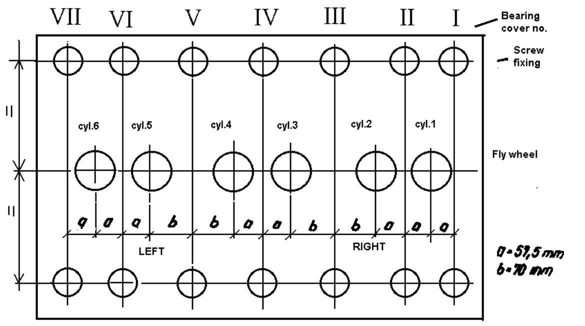
Fig.3

For a better acceptation of the simplified hypothesis above mentioned, with reference on further development and modernization of the crankshaft used in tractor engine made in Romania, model 2601-050, the authors used an own methodology. For this scope it was made an universal hydraulic stand for cold testing of the engines (from one cylinder engine to 12 cylinders engines) [1] see fig.1**.** The hydraulic oil is absorbed into the tank **R**, through the filter **F**, by pump **P**, which is driven by electrical engine **M**. In the present hydraulic scheme there are 6 stations tied with 6 cylinder. Every of them is functioning to 6 important pressures for an engine. It can be noticed that usually there aren't more than 3 meaningful pressures to a engine with 6 line-cylinder. As a result, the achievement of the 6 distinct pressures (with individual adjustment) is enough to ensure the using of this stand for all usually automotive engines. It can be adjusted, afferent to a given position for the crankshaft, the meaningful individual pressures from the cylinders, watching the manometers $\mathbf{M_{n,j}}$, with $j = 1,\ldots 6$. The crankshaft is fasten/blocked in this position till the measurement cycle is done and the engine's suspension is identically with the real one.

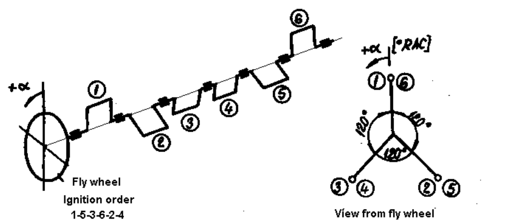
The measurements for the forces from the fastening screws for main bearing cover was achieved with equivalent bolts from stiffness point of view, on which was applied 4 electrical strain gauges, mounted in full Wheatstone bridge (Fig.2). The displacements measurement for main bearing cover was done with some displacements inductive transducers type **TI-2-BB**, ***Microlimit***, settled like in fig. 3**.** Here, ***1*** represents motor block, ***2*** is the crankshaft, ***3***- tank of oil , ***4*** is the support for the strain gauges, put between the tank of oil and motor block, ***5*** is main bearing cover, and ***6*** represents displacements inductive transducers, mounted in a diagonally scheme for highlighting the rotations of the main bearing cover in the two perpendicularly planes.

On this constructive solution for the engine it could be distinguished three type of zone for the cylinders:

I – cylinders having both own bearings or the neighbor bearings with symmetrically positions;
II – cylinders having an asymmetrically position both for own bearings level or for the neighbor level;
III – cylinders having asymmetrically position for own bearings and symmetrically position for neighbor bearings level (Fig.4).

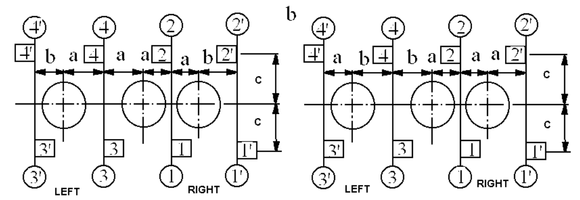


Sight of up engine block-2601-050



Cylinder type I

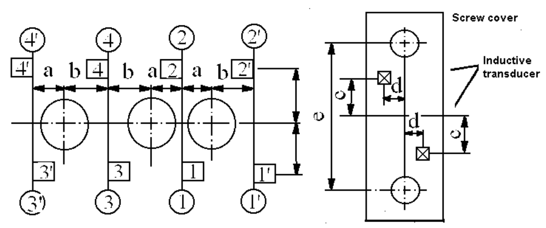
Cylinder type II



Cylinder nr.	T_1	T_2	T_3	T_4	T'_1	T'_2	T'_3	T'_4
1	1	2	3	4	–	–	5	6
6	11	12	13	14	9	10	–	–

Cylinder nr.	T_1	T_2	T_3	T_4	T'_1	T'_2	T'_3	T'_4
2	3	4	5	6	1	2	7	8
5	11	12	9	10	13	14	7	8

Cylinder type III



Cylinder nr.	T_1	T_2	T_3	T_4	T'_1	T'_2	T'_3	T'_4
4	7	8	9	10	5	6	11	12
3	7	8	5	6	9	10	3	4

c = 13 mm
d = 5 mm
e = 104 mm

Fig.4

It has been used the notation $\mathbf{T_j}$ for electrical strain gauges bolts, and with $\mathbf{M_j}$ the displacements inductive transducers. In the same picture is presented: the crank star (their spatial displacement), respective the upper view for the motor block. In fig. 5, 6, and 7 are presented, connected with type **I, II,** and **III** cylinder**,** the changing of the forces from own electrical strain gauges bolts and of the main bearing cover displacements. It was used the following notations: $\mathbf{T_j}$ – the forces from own electrical strain gauges bolts; $\mathbf{M_j}$ – the indications of the inductive transducers at the level of the overs **c** for analyzed cylinder, and through $\mathbf{T'_j}$ şi $\mathbf{M'_j}$ – are replaced the values afferent neighbor bearings. Through statistical analysis of these values it can be obtained also *the probable intervals of their sizes*, that is in authors opinion an usual element for designing purposes. Based on simplified scheme from fig. 8 it can be used very well the indications of the displacements inductive transducers which are average liniar displacement of main bearing cover Δ_m, as well as his rotation in his own plane φ_1, respectively in the transversal one φ_2. Another set of measurements watched the changing in reaction values at a constant loading of the piston with a force $F = 50000\ N$

simultaneuosly with the rotation of the crankshaft. In fig. 9 are given the results of the measurements afferent cylinders of type **I**.

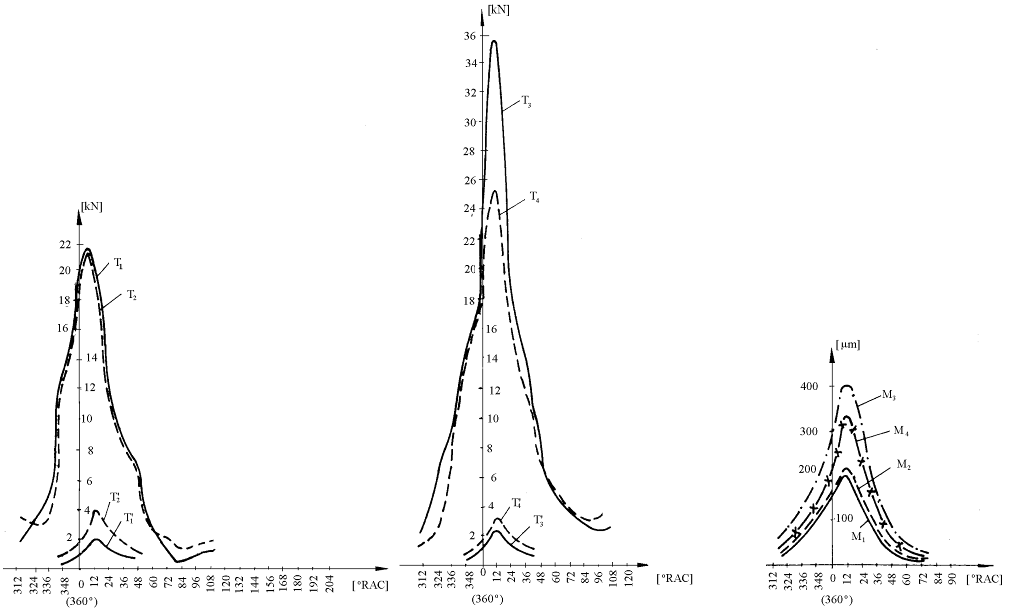


Fig.5

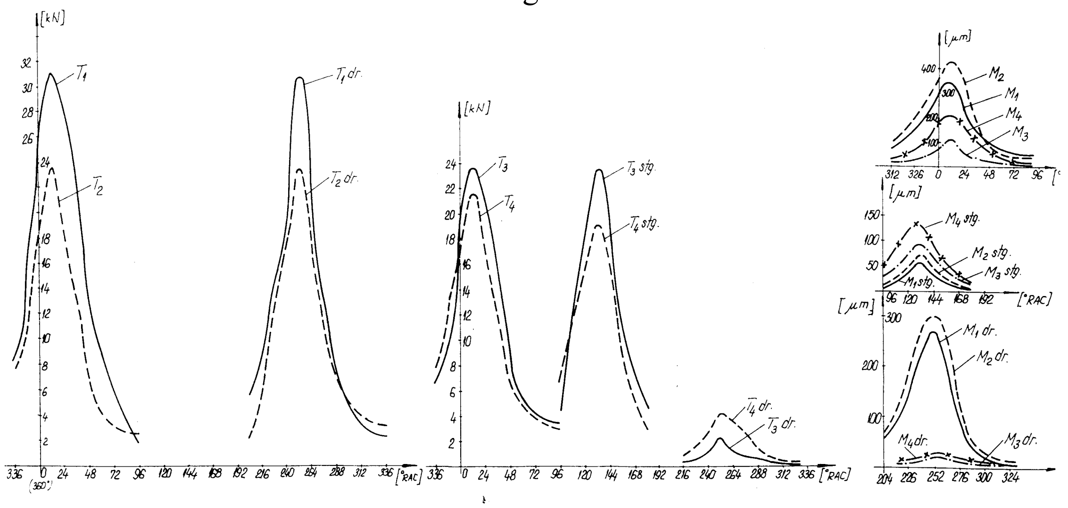


Fig.6

2. Numerical modeling using MEF method

It were used tetrahedral finite elements, with 10 nodes, having 3 DOF (degrees of freedom) each per node. The total number of finite elements was 5126. The geometrical modeling was done using AutoCaD software and numerical analysis was done using DesignStar. For a closer reality evaluation, in the zone of the main bearing cover the contact between the crankshaft and the bearing was represented using some jacks.

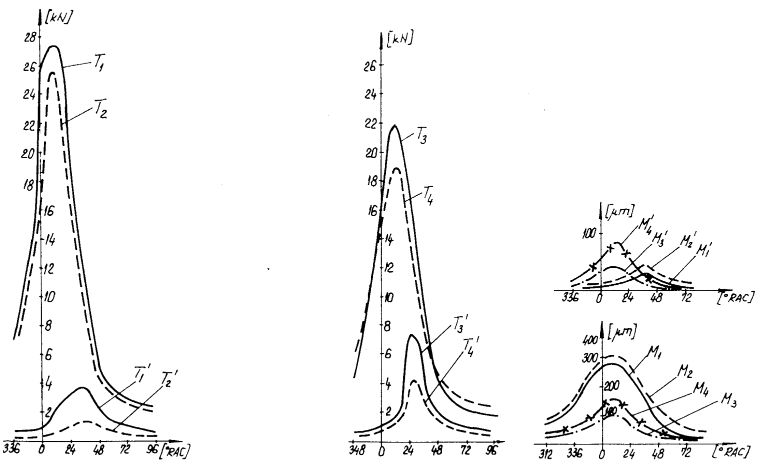
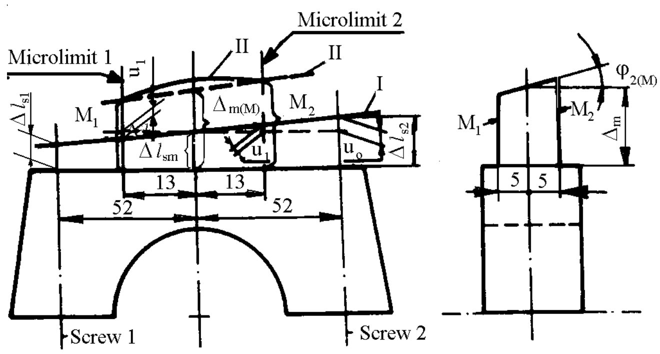


Fig.7



$$\Delta l_{si} = \frac{N_i\, l_i}{E\, A_i}$$

$$\Delta l_{sm} = \frac{N_{med}\, l}{E\, A}$$

$$u_o = \Delta l_{32} - \Delta l_{sm}$$

$$u_1 = u_o/4$$

$$\Delta_{m(M)} = (M_1 + M_2)/2$$

$$\varphi_{1(s)} = \mathrm{arctg}\,\frac{u_1}{13} \quad \text{[rad]}$$

$$\varphi_{1(M)} = \mathrm{arctg}\,\frac{M_2 - M_1}{26000} \quad \text{[rad]}$$

$$\varphi_{2(M)} = \mathrm{arctg}\,\frac{M_2 - M_1}{10000} \quad \text{[rad]}$$

$$\varphi_{1\,cover} = \varphi_{1(M)} - \varphi_{1(s)}\ ;\ \varphi_{2\,cover} = \varphi_{2(M)}\ ;\ \Delta_{m\,cover} = \Delta_{m(M)} - \Delta l_{sm}$$

Fig.8

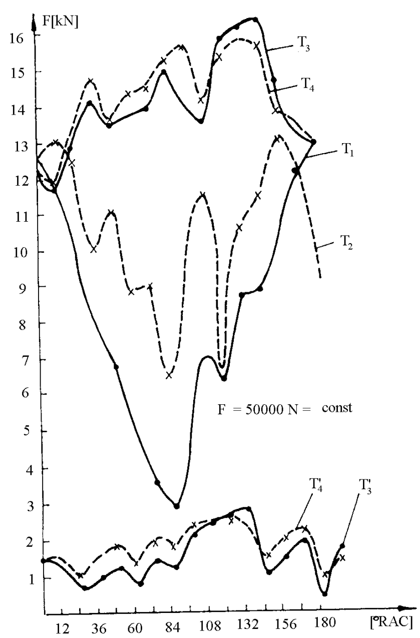


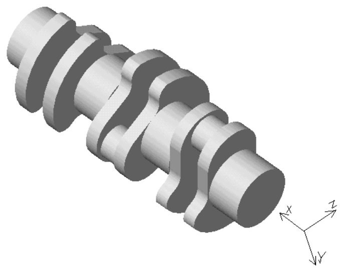
Fig.9

In fig. 10 is given the equivalent field of von Mises stress in crankshaft and ractions force. In the Fig. 11 is represented the total displacements field and the . Fig. 12 represents the effect of the constant loading force of 50 000 *N* in vertical plane in 2 position rotated with 90 ^{0} each.

3. Conclusions

The results of the experimentally researches as well as in numerically way are coming to justify the validity of the preliminary calculus scheme made in hypothesis with one bend. This scheme can be used with good results only at preliminary calculus from statically point of view, respective with errors of 5...10% at complete statically analysis. For a preliminary dynamic calculus it must take into consideration whole crankshaft, placed only on stiffened supports.

Model of cranckshaft



Field of stresses Von Mises

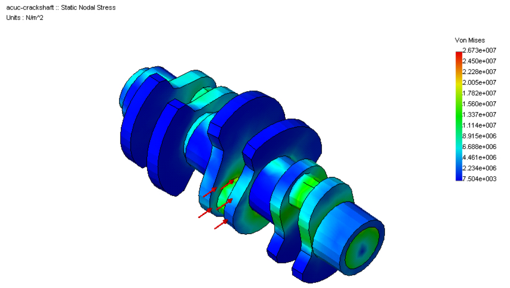


Fig.10

Total displacements

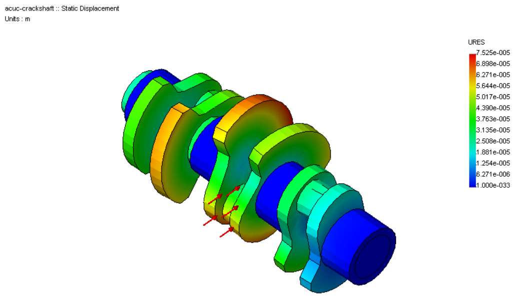
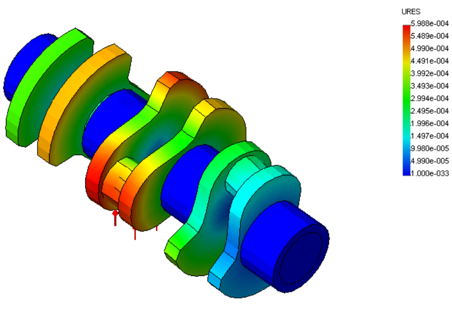


Fig.11

Field of displacement Ures at 0^0



Field of dispacement Ures at 90^0

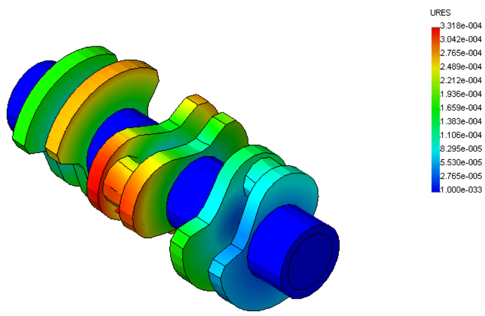


Fig.12

A complete dynamic calculus must take into consideration also the unstiffened support, but not only as a uniform stiffness and also with a variable stiffness with the center angle. This last aspect, especially with the evaluation of the meaningful size of rotation angles φ_2 of the main bearing cover, will lead at the apparition of some parametric vibrations Hill-Mathieu, with hard to estimate effects on the stressed-deformed state of the crankshafts [2].

4. References

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