Monitoring of conditions of agricultural machines' parts in operation

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Abstract

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This contribution focuses on problems of monitoring the runout of tractor diesel motor parts used in agricultural operation. Crankshafts from the agricultural tractors of type Zetor 6911 are used as samples for measuring the runout and circularity. The first sample of the crankshaft is loaded in the tractor used in livestock production, and the second one is used in vegetable production. Measured values of the runout and circularity of both samples are evaluated by tables and polar diagrams. Results of the experiment show the amount of runout and the following wear of agricultural machine parts in different operating conditions of agricultural production.

Keywords: crankshaft; deviation, circularity; runout measurement; wear

The importance of usage of agricultural equipment is in elimination of any hard work (Líška 2008; Poničan, Korenko 2008). The wear of several parts and a decrement of machine operability occur during the usage of agricultural equipment in the manufacturing process (Páltik 2007). The lifetime of agricultural machines has to be dealt with when economic effectiveness is the topic. Depreciation charge, return on investment and operation expenditures ratio put into machinery and technico-economic evaluation of used and new machines and technologies along with their appropriate reproduction could not be done without knowing the lifetime. Agricultural machinery lifetime knowledge is essential for manufacturer production amount and selection of machines and spare parts (Kročko et al. 2007). Lifetime optimisation rules are valid generally, in the present economical environment too. The machinery should be replaced after an optimum operation time, in the moment of the lowest unit operation costs (Drlička, Mikuš 2007).

An example can be the crankshaft of the agricultural tractor. This contribution deals with problems of monitoring the runout of tractor diesel motors parts that are used in agricultural operation. Measurements of runout and circularity are accomplished on the crankshafts of the agricultural tractors of type Zetor 6911 (Zetor Brno, Czech Republic). Measured values of runout and circularity are evaluated by tables and polar diagrams in the experiment. The results allow comparing the amount of runout and the following wear of individual machine parts in different agricultural operations.

Turbocharged diesel engine is currently the most preferred for medium-sized drives in drive units (tractors, trucks, marine drives). In addition, continually increasing share in the highly competitive automotive market due to its reliability, combines with very low fuel consumption. During the past decade, mathematical modeling has paved the way for an in-depth study of diesel engines. However, most research has focused on thermodynamics, which has a direct effect on heat release and subsequent performance and pollutant emissions to the environment. On the other hand, questions about engine dynamics, e.g. complex movement of the connecting rod, crank linkage mechanism, crank and its deformation, torsional vibration, etc., are often ignored (Giakoumis at al. 2007).

The optimization of individual operations in the parts manufacturing is a role of engineering technology, which consists in determination of the optimal, the most economic or the most productive conditons for the performance of the defined operation. Economic conditions or conditions of the maximum productivity are studied within optimizing the cutting conditions for machines. The optimal cutting material and tool and the optimal combination of cutting conditions (cutting speed, feed and depth of cut) are defined by the conditions mentioned above (Kotus et al. 2002). An effective balanced solution between external stress and the resistance of parts in the cross-section and surface is certainly a choice of an economical system: basic material – surface layer. Heat treatments, chemical-heat treatments of surfaces and metal coatings are considered as appropriate technologies of creating wear-resistant layers from an economic point of view. These technologies increase the wear resistance of materials significantly but not as much as, for example techniques of diffusional saturation or material surfacing (Kováč et al. 2005).

MATERIAL AND METHODS

Crank mechanism. The crank mechanism is the basic part of the internal combustion piston engine that converts the linear reciprocating movement of the piston into the rotary movement of the crankshaft. The function of the engine is not possible without this mechanism.

Forces that affect the crank mechanism. Gas pressure causes a force impact on surfaces in the combustion chamber. So forces of gas pressure are considered in the calculation of forces in the engine mechanism (force on the piston), of forced oscillation of a valvular train (force on a valvular disk), and of solidity of a cylinder, crankshaft and other parts. The force of gas pressure affecting the piston surface is a very interesting information in terms of forces acting in the engine mechanism:

$$F_{pl} = S \times p_{(\alpha)} \tag{1}$$

where:

F_{pl} – power by gas pressure acting on the piston surface

S – projection of the piston surface area perpendicular to the axis of the piston

$p_{(\alpha)}$ – pressure change in the cylinder depending on the angle of rotation of the crank α

Inertia forces. Inertia forces are created at uneven linear movements, and even or uneven rotative motion (centrifugal forces) in internal combustion engines. All inertia forces have a periodic character in engine operation, which causes engine vibration. This fact urges designers of engines to minimize the inertia force by balancing. Piston engines, in which also linear movements exist besides the rotative motion, are engines that can be balanced only partially. In addition to negative impacts, inertia linear forces have also a positive effect on reducing the load of the crank mechanism because of substraction from the force caused by gas pressure on the piston (especially during the expansion stroke).

Calculation of inertia forces of rotating parts F_{zr} in the crank mechanism:

$$F_{zr} = m_r \times r \times \omega^2 \tag{2}$$

where:

m_r – total weight of rotating parts (kg)

r – radius (mm)

ω – rotation speed of the crankshaft (rotations/min)

Calculation of inertia forces of linear parts F_{zr} in the crank mechanism:

$$F_{zp} = m_p \times a \tag{3}$$

$$m_p = m_{piston} + m_{piston\ pin} + m_{piston\ ring} + m_{op} \tag{4}$$

where:

m_p – total weight of linear sliding parts (piston, piston pin, piston rings and linear sliding part of the piston rod m_{op}; Fig. 1)

a – distance from the axis of the crank pin

Studied object. Crankshafts produce elastic vibrations which stress them to bending, torsion, tension, and sag of the shaft can also occur. Therefore, the function of the crankshaft is changing. Main pins placed into slide bearings are crucial parts with wear and deformation. Therefore, the object of using optical measurements is to measure the runout of crankshafts in the agricultural tractor after its load in operation. The measurement was located in the major places of pins. The measurement of roundness of major pins is related closely.

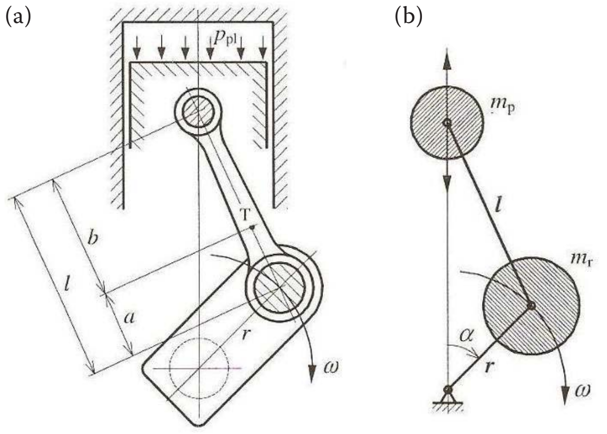


Fig. 1. Crank mechanism (a) and scheme drive crank mechanism (b) (Hlavňa et al. 2000)
p_{pl} – pressure on the piston surface; m_p – torque on tap piston; m_r – torque rod; ω – torque crank; l – length of the rod; α – angle of rotation of the crankshaft; r – space between the connecting rod pivot; α – the axis of rotation of the crankshaft; b – distance from the pivot axis song does gravity rod; a – space between the connecting rod piston pin axis of the connecting rod; T – center rod

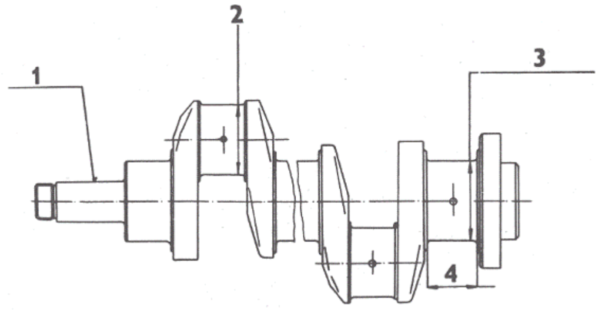


Fig. 2. Crankshaft of the tractor Zetor 6911 (Zetor Brno 1978)
1 – crankshaft; 2 – rod pins; 3 – main pins; 4 – width of the main pin

Methodology. Based on the defined objectives of this work, the basic methodology was chosen for the solution: selection of a specific type of crankshaft, determination of measured parts of the crankshaft, measured quantities depending on the measuring device, and analysis and evaluation of the obtained results.

Characteristic of the monitored crankshaft. Measurements were carried out for two crankshafts (Fig. 2) which were demounted from agricultural machines Zetor 6911 and were in operation for 20,000 Mh. The crankshaft No. 1 is from the tractor in livestock production, and the crankshaft No. 2 is from the tractor in plant production. It is a four-cylinder diesel engine crankshaft with an angular position of bends of the crankshaft (180°). The crankshaft has five main and four rod pins; the main pins are of a larger diameter and width than the rod pins. Each bend of the crankshaft is balanced by a counterweight that balances the uneven run of the crank mechanism and thereby reduces the stress onto the bearings in which the shaft is located. The counterweight is attached to the first, fourth, fifth and the last arm of the crankshaft. The crankshaft is made of grey cast iron by casting; working surfaces of the shaft are finished by grinding, lapping, and superfinishing.

Optical apical device. In order to eliminate the laborious counting, we recommend an adjustment of the zero starting point as follows: to rotate a setting wheel until the bottom window shows 0'0'' and it coincides with the index line; to loose a clamp and a grooved spiral and to tilt a crank to the upper position; to turn a gross adjuster until the top win-

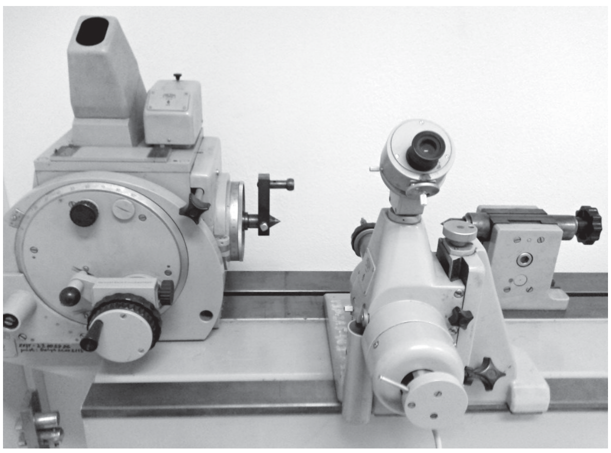


Fig. 3. Optical head with a tailstock and an additive device on a basic board 1,600 mm

Fig. 4. Device for measurements of axial distances

dow shows 0°; to tighten the grooved spiral; to move an adjuster until the measuring spiral is not located between marked double grooves 00; to set an approximate scale to zero that can be turned by a hand and cannot be fasten. After the clamp and the grooved spiral are tightened, we can start to work. The clamps have to be lost for setting the next steps. Each minute a second steps can be done by the setting wheel during the operation. Degrees are set to tens of minutes after work. The approximate scale is used to a gross angular adjustment (Korenko et al. 2010).

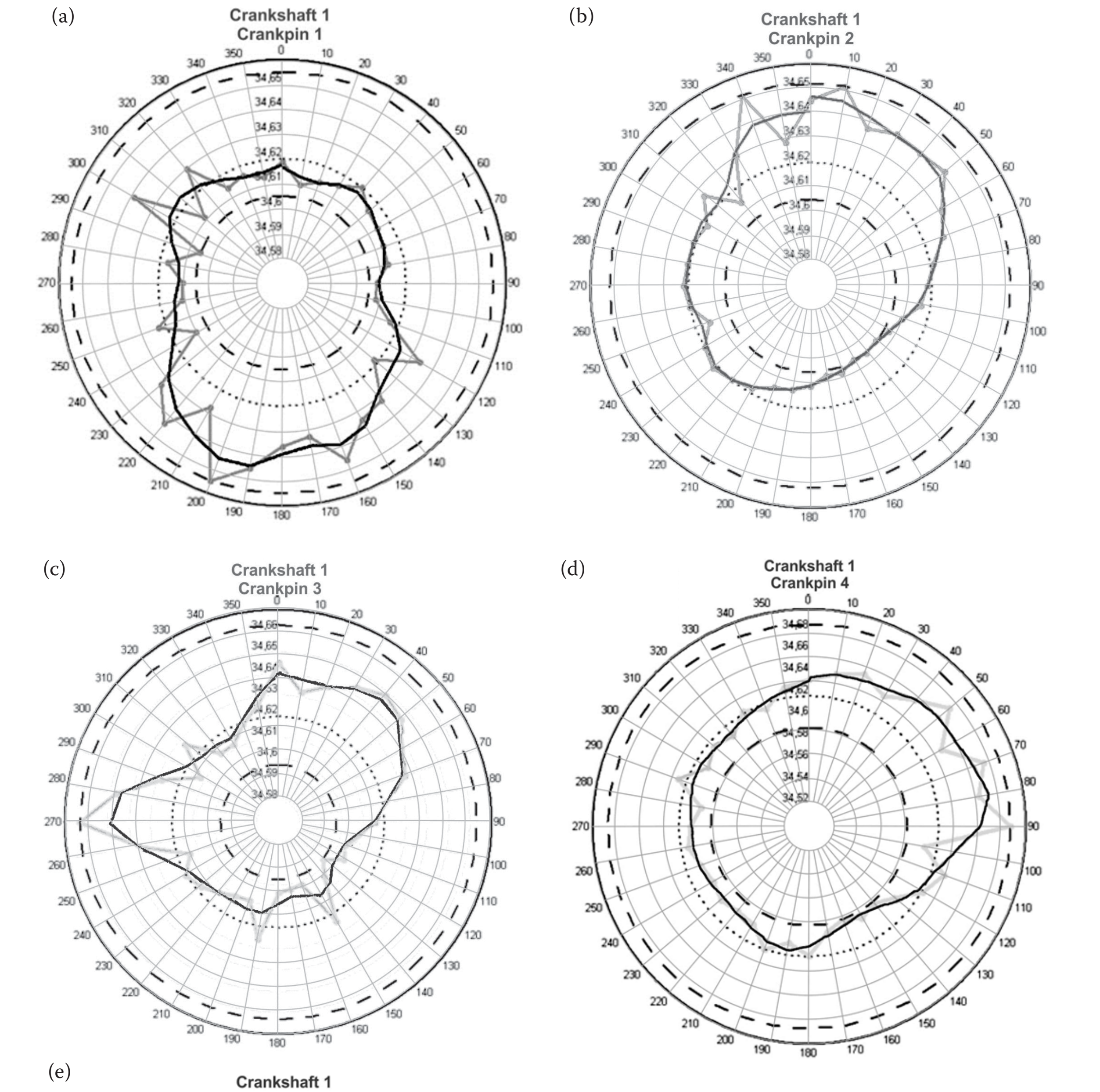
Measurement of circular radial runout (Figs 3 and 4). After reconstitution device to access the procedure for setting up the horse head. Then we can proceed to the actual display of parts, namely the crankshaft between the centers of the optical device and tailstock. Firstly, the crankshaft was thread through the clamping heart and placed on the tip of the optical head. The tip was slided out the tip of the tailstock by turning a screw until it clicks into the centring hole of the crankshaft. The emphasize was put on a force of screw tightening so it could not cause a bend of the shaft – buckling. A dial indicator and a holder provided together with a device marked 24-151-1 were placed on a clean grinded surface of the basic board. The arms of the holder were directed to achieve the horizontal orientation of a sliding measuring rod of the dial indicator. The measurement of runout were performed on points of the main crankshaft's pins. The dial indicator was gently pressed against the shaft so the pointer made one revolution around its axis. Tipping a lever (10) and turning a wheel (9) clockwise showed deflection of the pointer of the dial indicator. The max. value of deflection and the angle of deflection of the shaft were detected within deflection and were subtracted them from the optical head P3. If necessary, a correction and offset the tailstock on a desired value (in accordance with paragraph 3.2.3c (Zetor Brno 1978) was made, and it was necessary to repeat the measurement of eccentricity. The actual measurement proceeded at the exact shoulder of the crankshaft. Tipping the lever and setting the wheel served for setting the display of the optical head P3 to angle 0°. The measurement was be carried out at angular intervals of 60° and repeated up to angle 360°. The dial indicator (marked as 24-151-1) helped to read a measured value at each interval shifted by 60° from the scale. The entire measurement was repeated at the same angular values for each main pin of the other crankshaft. Deviations from circular radial runout were calculated and graphical relationships are demonstrated. The main pins are on the x-axis, and the amount of deviation in millimeters is on the y-axis. The greatest deviation of the crankshaft points could be defined from a curve. Further, the values of the average deviation of runout on both shafts were calculated and compared.

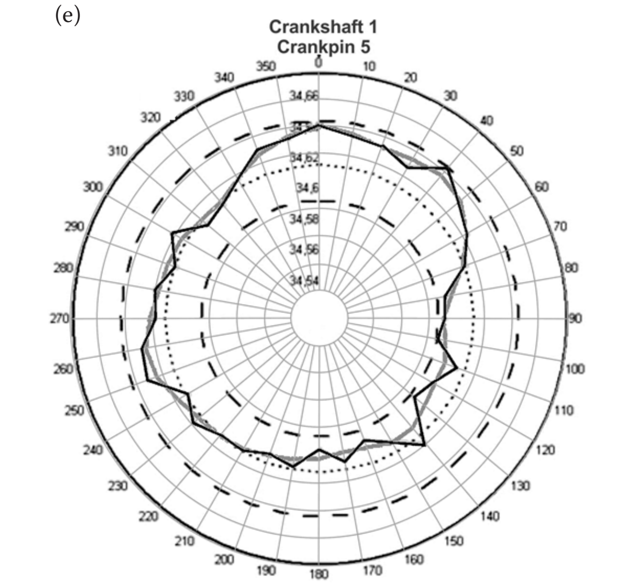
Measurement of roudness. The same installation of the crankshaft as in the case of runout measurement was used. The optical dividing head P3 was used for angular tipping. It was used to set the angle that we will move in the range from 10° to 360°. An additional device for measuring the axial distance – radius was used to measure the radius of the main crankshaft's pins. The device on the basic board in front of the measured pin was moved. The tip for the axial distance of the measuring aditional device was set on the axis of the crankshaft perpendicularly. The cylinder with the tip was pressed against the surface of the pin but not too intensively. Lighting allowed to project the actual value of axial radial distance (r) into ocular from which the values could be read. Graphic relationships in the form of polar diagrams were used for measured values, which reflect the angle of tipping and the radius of pin. Polar diagrams were used for presenting the results of roundness deviations.

Table 1. Circularity deviations (Min; mm) for the main pins of the crankshaft No. 1

r (mm)	Pin 1	Min	Pin 2	Min	Pin 3	Min	Pin 4	Min	Pin 5	Min
Crankshaft No. 1										
r_{max}	34.655	0.05	34.652	0.047	34.663	0.066	34.686	0.095	34.649	0.052
r_{min}	34.605		34.605		34.597		34.591		34.597	
Crankshaft No. 2										
r_{max}	34.767	0.053	34.782	0.086	34.788	0.081	34.781	0.075	34.768	0.053
r_{min}	34.714		34.696		34.707		34.706		34.715	

r – axial distance radius; r_{max} – largest radius measured profile; r_{min} – smallest radius measured profile





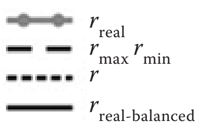


Fig. 5. Polar diagram of the circularity of (a) main pin 1, (b) main pin 2, (c) main pin 3, (d) main pin 4 and (e) main pin 5 of the crankshaft No. 1

RESULTS AND DISCUSSION

Results of circularity measurements and their graphical demonstration

Measurement of circularity of the crankshaft No. 1 (Fig. 5)

The first main pin of the crankshaft No. 1 is characterised by r_{max} = 34.655 at angle 200° and r_{min} = 34.605 at angle 290°. The difference forms the width of the anuloid with values 0.005 mm = 5 mm. Calculations were repeated for all the main pins of both crankshafts.

Table 1 shows r_{max} and r_{min} according to the relation $r_{max} - r_{min} = min$, calculated min. width of anuloids – deviations of circularity min.

Measurement of circularity of the crankshaft No. 2 (Fig. 6)

Table 1 shows r_{max} and r_{min} according to the relation $r_{max} - r_{min} = min$, calculated min. width of anuloids – deviations of circularity *min*.

Results of circular radial runout and its graphical representation (Fig. 6)

The circular radial runout of the circumferential surface's component was determined at the shaft rotation of 360° by the max. value of deviaton v_{max} and minimal value of deviation v_{min}, which were measured by using the dial indicator. The difference between these values was referred to the deviation of circular radial runout ρ_H. It was calculated for the first and the other main pins:

$$\rho_H = v_{max} - v_{min} = 0.307 - 0.3 = 0.007 \text{ mm} \quad (5)$$

where:

v_{max} – max. measured deviation from axial rotation (mm)
v_{min} – min. measured deviation from axial rotation (mm)

Average runout of the main pins of the crankshaft No. 1 (Table 2):

$$\bar{x}_1 = \frac{1}{N}\sum_{i=1}^{N} x_i = (0.007 + 0.012 + 0.012 + 0.016 + 0.013)/5 = 0.0138 \text{ mm}$$

Average runout of the main pins of the crankshaft No. 2 (Table 2):

$$\bar{x}_2 = \frac{1}{N}\sum_{i=1}^{N} x_i = (0.027 + 0.036 + 0.45 + 0.037 + 0.024)/5 = 0.0338 \text{ mm}$$

Difference of runout:

$\bar{x} = \bar{x}_1 - \bar{x}_2 = 0.0338 - 0.0138 = 0.02$ mm

Demands on material attributes grow concurrently with a continuous increase in performance. Development is directed to materials with improved resistance and lifetime, with improved mechanical attributes, less weight and lower cost especially. The crank mechanism also requires its care for a reliable function. The use of appropriate materials and their subsequent chemical-heat

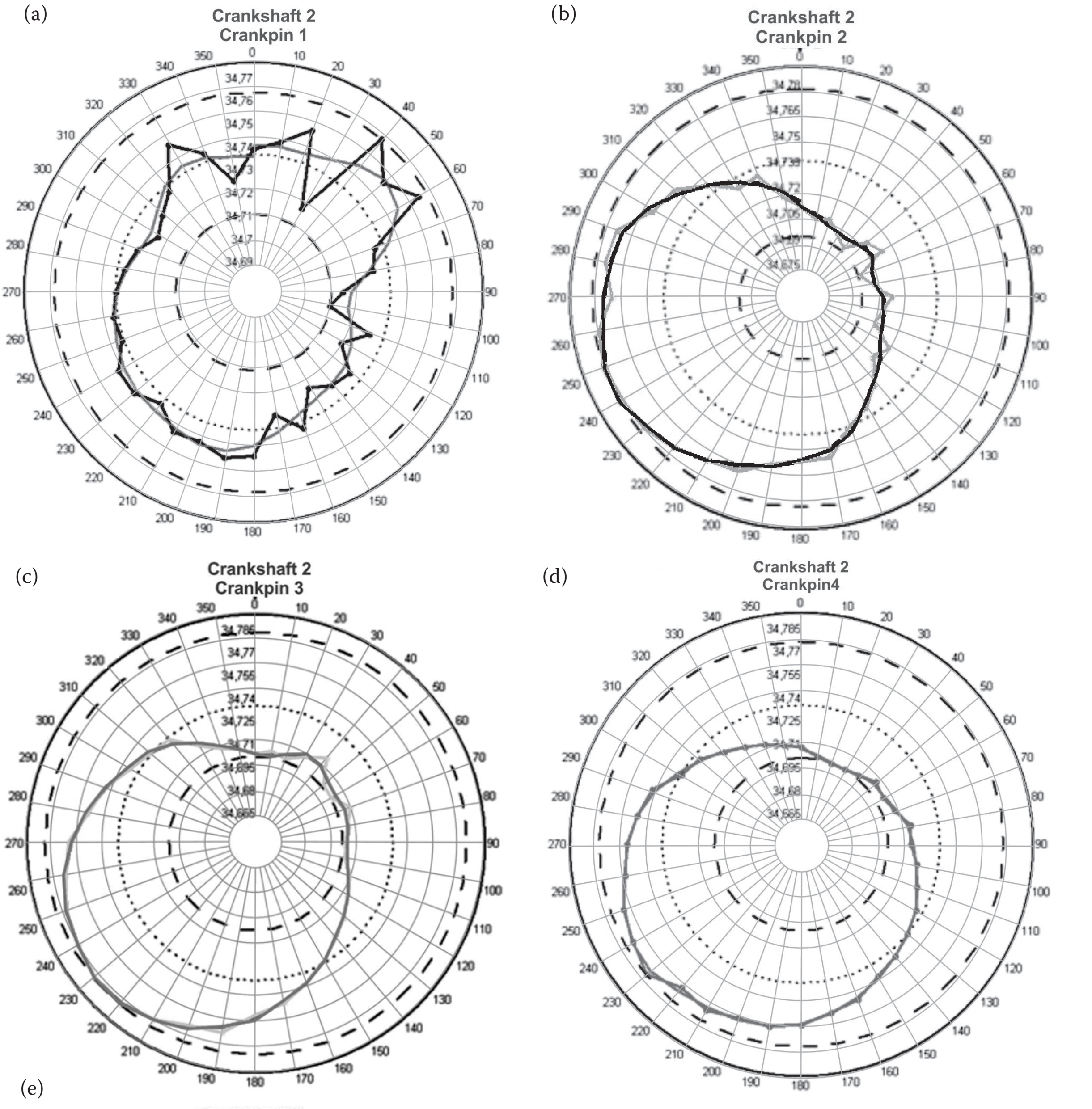
Tab. 2. Deviation v (mm) of runout of the crankshafts No. 1 and No. 2 (Fig. 7)

Angle (°)	Pin 1	Pin 2	Pin 3	Pin 4	Pin 5
Crankshaft No. 1					
60	0.301	0.298	0.303	0.3	0.304
120	0.3	0.302	0.299	0.305	0.299
180	0.302	0.306	0.305	0.308	0.305
240	0.305	0.31	0.314	0.314	0.31
300	0.307	0.308	0.32	0.316	0.312
360	0.306	0.303	0.313	0.308	0.309
ρ_H	0.007	0.012	0.021	0.016	0.013
Crankshaft No. 2					
60	0.528	0.531	0.535	0.520	0.529
120	0.521	0.520	0.519	0.519	0.522
180	0.529	0.527	0.532	0.538	0.530
240	0.540	0.544	0.548	0.554	0.538
300	0.548	0.556	0.564	0.557	0.546
360	0.537	0.545	0.550	0.541	0.534
ρ_H	0.027	0.036	0.045	0.037	0.024

v – measured deviation from the axis of rotation in this section; ρ_H – circular radial run-surface components

(a) (b)

(c) (d)



(e)

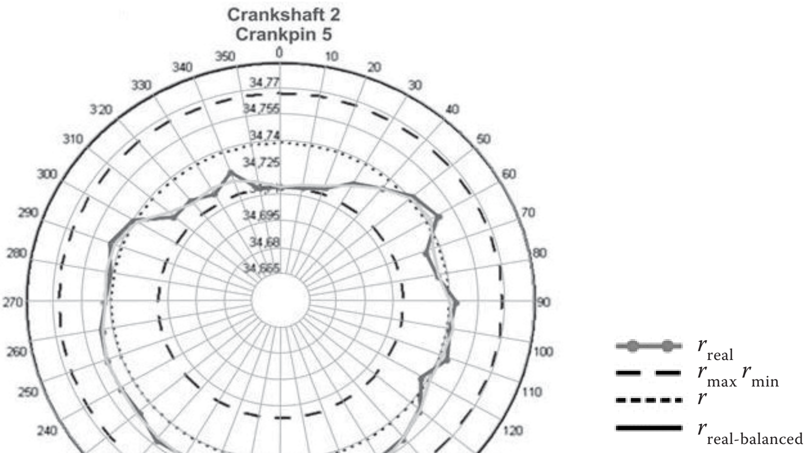


Fig. 6. Polar diagram of the circularity of (a) main pin 1, (b) main pin 2, (c) main pin 3, (d) main pin 4 and (e) main pin 5 of the crankshaft No. 2

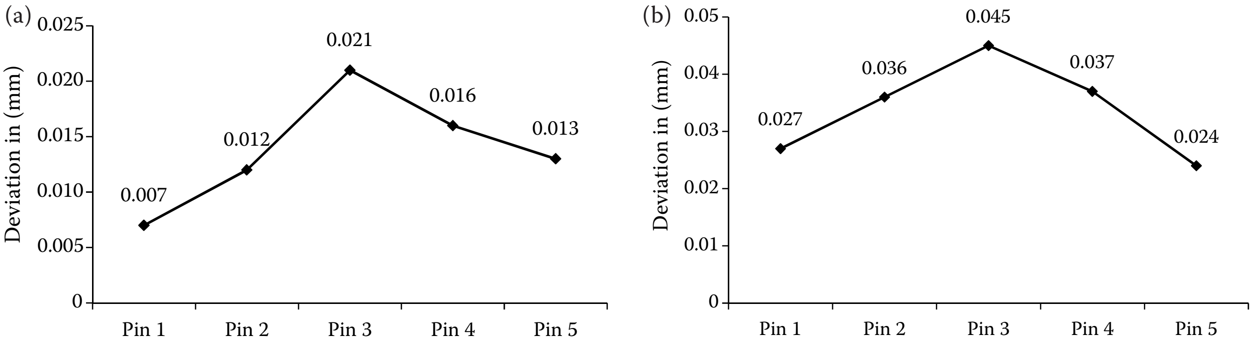


Fig. 7. Relationship between the deviation of runout and the constituent main pin of the (a) crankshaft No. 1 and (b) crankshaft No. 2.

treatment leads to ensuring the lifetime and reliability of the crank mechanism. Important is the quality of lubrication and lubricants. Temperature in the combustion chamber of diesel engines is changing up to 1,000–1,200°C. Pressure can reach values up to 25 MPa in diesel engines. The operating speed of some components must be also taken into account. The piston, valves, crankshaft, etc. are exposed to changes in conditions of nearly 17 times/s at 1,000 rpm. It is up to 50 times/s at 3,000 rpm. Also, a great tribological degradation due to fitting interaction of engine parts must be taken into the account.

According to Giakoumis et al. (2007), in the analysis of the turbodiesel engine under load the following results were obtained:

Moment of inertia affects the crankshaft deformation mainly in surface portion, but it applies only to the total engine with relatively low-speed rotations. It means the max. deformation during one cycle cylinder engine. Significant levels can be expected during transient operation, depending on the intended increase of the load. Instantaneous max. deformation can be up to 50% higher compared with the corresponding mean, as in intermediate cycles. Minor variations in the load, as well as stiffer crankshaft design are key parameters for the reduction of crankshaft torsional deformations.

A significantly degraded factor of crankshafts is also pitting, which usually becomes a deposit of fatigue crack. Therefore, the last two of named steels are usually carburized, nitrided or carbon-nitrided. The crankshaft is a component in the engine that bears the greatest mechanical load. Terms of the crankshaft´s load are static and dynamic. The pressure exerted by an explosion of fuel is transferred through the piston and connecting rods, the inertia force of pistons and connecting rods, vibration, etc. Earlier, nodular cast iron served for crankshafts. Modern engines are equipped with crankshafts of forged steel or cast iron. Micro-alloyed steel Fe-0.5C-0.8Mn-0.25s-0.1V was proved as a suitable alternative and becomes more widespread due to lower cost and excellent mechanical attributes (does not require an additional hardening). Hot forging with cooling in air is sufficient for the creation of carbides and nitrides of vanadium in a structure (Girman 2011).

CONCLUSION

The measurements show that the difference of runout between the shaft No. 1 used within livestock production and the shaft No. 2 used within vegetable production was 0.02 mm. The shafts were in operation for 20,000 Mh. The shaft loaded in vegetable production had the average runout of the main pins of 0.0338 mm while the shaft in livestock production had 0.0138 mm. A greater stress and forces affect the crankshaft from vegetable production in comparison with livestock production.

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