ISSN 1648-4142 TRANSPORT
www.transport.vtu.lt

TRANSPORT – 2005, Vol XX, No 6, 225–231

THE RESEARCH INTO THE DISAXIAL CAM MECHANISM FOR DIESEL FUEL – INJECTION PUMP

Evgeny Tausenev[1], Andrew Svistula[2]

Altai State Technical University, Lenin St., 46, 656038 Barnaul, Russia,
Tel/fax: (+3852368475); e-mail: [1]tayceheb_e_m@mail.ru, [2]sae59@mail.ru

Received 10 April 2005; accepted 30 September 2005

Abstract. A cam mechanism is named disaxial due to the offset of a plunger axis concerning the axis of a camshaft. The named offset allows to reduce the maximal angle of pressure. There is decrease of contact pressure in a pair of a cam and a roller and the pressure upon the side face of a guide sleeve. Theoretical researches into the influence of the offset and other elements of a cam mechanism on the specified pressure and kinematics of a plunger are resulted. Examples of practical realization of disaxial designs are shown. The results of comparative tests of fuel-injection pumps with disaxial and without disaxial are shown.

Keywords: diesel engine, fuel-injection pump, a disaxial cam mechanism, angle of pressure, contact pressure in a pair a cam – a roller, pressure upon the side face of a guide sleeve.

1. Introduction

In the perfection of fuel systems of diesel engines there is a tendency to increase injection pressure as it is one of the effective and rare factors providing inconsistent requirements for a diesel engine of ecological and economic parameters. And in this direction systems of direct action and accumulative are improved [1].

When the injection pressure increases, there is growth of loadings with other things being equal that results in the reduction of serviceability of details of a cam mechanism.

In this paper a greater attention will be given to fuel-injection pumps (further – pump) with spool-type steering of delivered quantity of fuel, with a tangential cam profile and a guide sleeve with a roller.

After small phase of operation tracks of destruction of loading faces of a cam and a roller have appeared in the experimental pump. On the side surface of a guide sleeve, in the zone of the most loaded and thin walls, which was formed by a flute under a roller, crazes were formed. The maximal pressure of fuel after the holder delivery-valve was equal to 686 bars.

In this connection development engineers put the problem not only of the selection of means to increase the injection pressure, but also to count the efforts in the specified joints and to develop provisions of their lowering, and without the increase of pump sizes.

It is possible to define the loading of a pair a cam – a roller evaluated by the value of the maximum contact pressure σ_{max}, and the side face of a guide sleeve – by pressure q_{max}. If the parameters are above the allowance, there are crazes, shelling-out, burrs, that result in losses of the pump functional properties.

It is known that the offset of the axis of the plunger concerning the axis of a power shaft allows to lower the tension in a plunger drive mechanism.

For example, the pump (Fig 1) of system Common Rail of L'Orange GmbH corporation of a series 4 000 for diesel engines of MTU corporation has 8 sections and everyone has the offset of axes of plungers concerning the axis of the eccentric shaft which permits to lower the plunger warping moment.

The disaxial mechanism (Fig 2) is the unused redundancy for lowering contact tensions on a surface of a cam in traditional pumps. A contact tension decrease is 12,5 % for pump model UTNI (Fig 3). There is optimum value ε=14,5 mm where R_0=16 mm, r_0=10 mm, h=9 mm. For traditional pumps $\gamma_{max} < 27 \div 40°$.

The degree k_ε of a contact unload is 1,1÷1,5. Reduction of γ and k_ε under offset is evaluated:

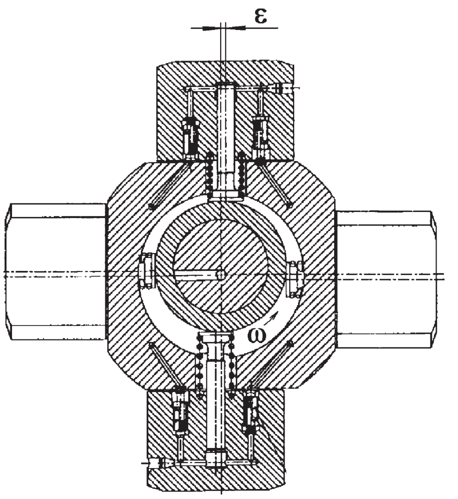

Fig 1. The Fuel-injection pump of system Common Rail l'Orange GmbH corporations of a series 4 000: ε – offset, ω – direction of rotation of the eccentric shaft

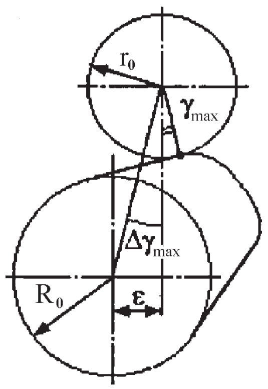

Fig 2. The scheme of a disaxial cam mechanism

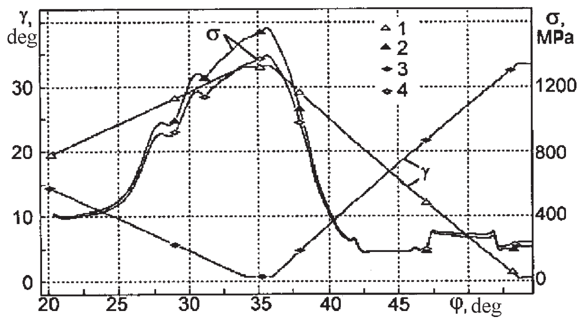

Fig 3. The angle of pressure (1, 3) and the contact tension (2, 4) in a fuel-injection pump of diesel engine D245.12C (n = 1 200 min^{-1}, Q = 79 mg/stroke): 1, 2 – axial; 3, 4 – disaxial, ε = 14,5 mm

$$\Delta\gamma_{max} = \left| arctg\left\{ \varepsilon \Big/ \left[h + \sqrt{(\rho + r)^2 - \varepsilon^2} \right] \right\} \right|;$$

$$k_\varepsilon = \left(cos\,\gamma_{\sigma\ max} \right)^{-0,5}, \text{ [1]}.$$

Thus, the offset diminishes the maximum angle γ_{max} of tension which reduces a contact effort.

2. Theoretical research

Value γ_{max} is an important parameter of a cam mechanism. It determines efficiency and intensity, and as corollary, reliability and longevity of operation of the mechanism with other things being equal. Limitation of γ_{max} diminishes abrasion and wear, prevents jamming a guide sleeve and diminishes the power which is expended for its driving.

It is possible to draw the diagram of efficiency of the mechanism $\eta = f(\gamma_{max})$ which is introduced in Fig 4, when the accounts of the cam mechanisms with different γ_{max} at the same tension and friction coefficient will be carried out. Losses are diminished by overcoming abrasion up to some limit, and then it is augmented and the mechanism can be jammed.

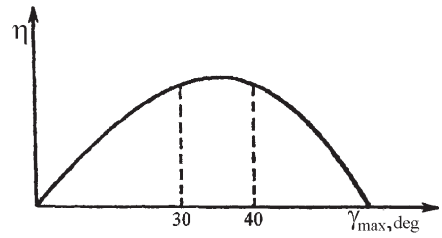

Fig 4. Diagram $\eta = f(\gamma_{max})$

It is proved that the maximum value of η for slow-speed and high-speed cam mechanisms with a roller guide sleeve corresponds to γ_{max} = 30÷40 °, η_{max} = 0,68÷0,73.

It is necessary to mark that different characteristics of a guide sleeve (plunger) driving almost do not influence the optimum limits of γ_{max} [2].

Function $\eta = f(\gamma_{max})$ has the maximum. There are values of γ_{max}, which are corresponding to it, but it is necessary to reduce γ_{max} as much as possible to decrease the level tension of details of the mechanism. According to operational experience the limits of an angle γ_{max} for cam mechanisms with a bodily moved roller guide sleeve and an external cam profile are limited by 30÷37 °.

The angle γ_{max} is a geometrical parameter, it depends on the constructive sizes of the mechanism. It signifies that it is possible to provide acceptable values of γ_{max} by these sizes.

Let's consider a pair of a cam – a roller.

According to Hertz theory the calculated contact compression tension σ_{max} in a pair of a cam – a roller can be defined by the formula:

$$\sigma_{max} = 0{,}418\sqrt{\frac{E \cdot P_{t\,max}}{b_o \cdot cos\gamma} \cdot \left(\frac{1}{r_o} \pm \frac{1}{R^*} \right)}, \text{ MPa}, \quad (1)$$

where $P_{t\ max}$ – the maximal total force which is operating at plunger along axis, N; E – reduced coefficient of elasticity, $E = 2E_1E_2/(E_1+E_2)$, MPa; E_1 and E_2 – coefficients of elasticity of materials of a cam and a roller accordingly; b_o – length of a contact line between a roller and a cam, mm; γ – the tension angle which is corresponding to $P_{t\,max}$, deg; r_o – radius of a roller, mm; R^* – radius of curvature of a cam at the point of contact with a roller, mm; in the formula the sign "plus" is accepted at the calculation of a cam with the external profile, and the sign "minus" – with internal.

Robert Bosch GmbH corporation allows σ = 1 500 MPa, Friedmann – Maier corporation – 1 600 MPa. According to the data of CNITA (St. Petersburg) the reliable operation of tractor diesel engine pumps during 10 000 hours is achieved with $\sigma = 1\,400 \div 1\,500$ MPa.

The time in use of the pump before outage depends on industrial, constructive and operation factors. It can exceed the period of operation before the occurrence of visible with the naked eye destructions of the surface 1,5–4 times. The level supposed of σ should be specified for each construction at holding bench trials in scale of real-time because reliable methods of accelerated tests are not created yet.

Theory and practice have proved that corresponding $P_{t\,max}$ time coincides with the beginning of fuel shutdown.

When the beginning of fuel shutdown coincides with the extremity of a tangential section of a cam, it will be $\gamma=\gamma_{max}$ in the formula (1), because this position corresponds to the maximum value of γ.

It follows from the formula (1), that the reduction of σ_{max} can be achieved increasing the radius of curvature of a convex section of a cam profile, the radius and breadth of a roller, decreasing normal force $P_{n\,max} = P_{t\,max}/\cos\gamma$ (Fig 5).

The active stroke of a plunger should come to an end up to an exit of a roller on a small radius arc. So the lower value of σ_{max} is ensured. Thus it is possible to reduce the radius of a round, to increase an active stroke and velocity of a plunger.

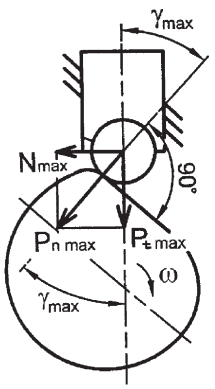

Fig 5. The Scheme of the operation of powers

The increase of breadth and radius of a roller augments the diameter of a guide sleeve and sizes of the pump, what is intolerable in an observed event.

Let's consider power $P_{n\,max} = P_{t\,max}/\cos\gamma$. Power $P_{n\,max}$ is augmented at the growth of injection pressure and (or) angle γ. It is necessary to diminish γ for the decrease of σ_{max}. In this case it is an unique way.

Let's consider a guide sleeve. It transmits an axial thrust to a plunger and accepts side power from a cam. Roller guide sleeve has the greatest extending in view of an optimum combination of speed, reliability and compactness. Supposed pressure q on the side surface of a guide sleeve is 10÷18,5 MPa. This value should be improved for each construction.

The tense level of the side surface of a guide sleeve may be estimated under the maximum pressure at the lower edge of the bearing surface:

$$q_{max} = \frac{N_{max}}{d \cdot l} + \frac{6N_{max}}{d \cdot l^2} \cdot \left(\frac{l}{2} \pm a\right), \text{ MPa}, \qquad (2)$$

where N_{max} – the greatest normal power to a side surface, N; d – diameter of a guide sleeve, mm; l – bearing length of a guide sleeve, mm; a – distance from the center of a roller up to the lower edge of the bearing side surface of a guide sleeve, mm; there will be a sign "minus" before a, if the center of a roller is displaced up from the lower edge of a bearing side surface of a guide sleeve and a sign "plus" in the contrary case (Fig 6) [3].

It is possible to reduce q_{1max} only having lowered $N_{max} = P_{t\,max}\,\mathrm{tg}\gamma$, and N_{max} – by drop γ under the condition of invariance of pump sizes.

The decrease of a pressure angle can be realized changing the geometrical sizes of a cam, a roller, an offset. Therefore it is necessary to investigate the agency of these sizes of the mechanism to γ_{max} and kinematics parameters of a plunger. For this purpose the mathematical model of a plunger kinematics of a disaxial cam mechanism with a tangential cam profile was built. The computer program was created. The program allowed to investigate the mechanism theoretically. The most important results are the following:

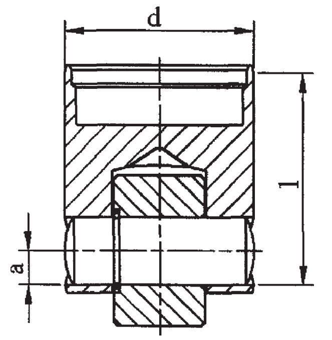

Fig 6. The basic sizes of a guide sleeve

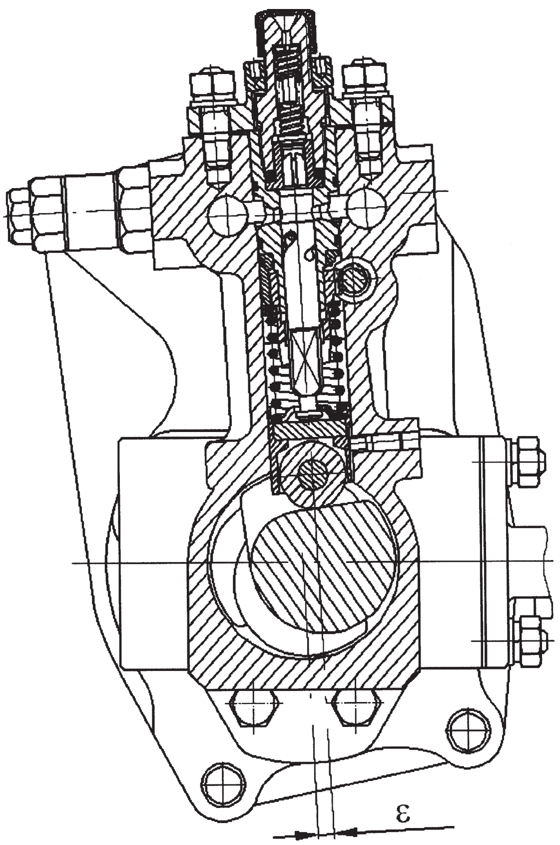

Fig 7. The disaxial fuel-injection pump

sizes of offset, a cam and a roller have the influence not only on γ_{max}, but also on v_{max} – the maximum velocity of a plunger. The offset is the most effective way to decrease a pressure angle. Offset reduces v_{max} less than other parameters. It is known that the velocity of a plunger influences on injection pressure. It is detected that the selection of sizes allows to decrease γ_{max} and to keep v_{max} on a former level. Offset diminishes the acceleration of a plunger, it decreases inertial power and allows to reduce locking tense of a plunger return spring.

Mathematical modeling research has shown that it is possible to lower γ_{max} from 39,4 ° up to 33,9 ° without reducing v_{max}; σ_{max} has decreased 4 %, q_{max} – 18 %, acceleration of a recoil – in 17,5 % (Fig 7–9) as contrasted to an axial cam mechanism of the pump [4, 5]. Calculated pressure of fuel above a plunger remained constant. It is visible that the offset depresses q_{max} the most considerably.

3. Experimental research

The disaxial pump is shown in Fig 7. The cam has no sizes providing results figured in Fig 8–10 by the technological reason. Nevertheless, comparative trials of pumps No 1, No 2 and No 3 are lead on the diesel engine to obtain intermediate results. Design features of pumps are shown in Table 1.

Table 1

Design features	Pump No 1	Pump No 2	Pump No 3
Diameter of a plunger d (mm)	10		
Stroke of a plunger h (mm)	12		
Units spacing (mm)	27	32	40
Type of the body of a fuel-injection pump	compact		with a manhole
Diameter of the delivery-valve (mm)	7	6	6
Preliminary stroke of a plunger hpr (mm)	5	5,4	5
Cam profile	tangential		
Radius of pitch circle of a cam (mm)	18	15,5	16
Radius of a roller (mm)	10	11	10
Radius of a rounding (mm)	5	4	3
Size offset (mm)	4	0	0
Type of a governor	Mechanical variable		
Sizes (mm): length width height	 303 181 265,5	 381 173 255	 511 193 286
Weight (kg)	14,80	15,5	18,472

3.1. Purpose: functional trials of the pump No 1, comparison of injection parameters for different pumps with sizes d×h = 10×12 mm, the estimation of pumps loading.

3.2. Technique: load regulation characteristics were registered at n=2 000, 1 850, 1 500, 1 200 min $^{-1}$, the optimum profitability angle θ of injection advance with several alternatives of the complete equipment of the diesel engine was defined (see Table 2), parameters of injection were measured. It is θ=28° for the pump No 1 with a preliminary stroke of a plunger 5 mm; smoke K_x of burnt gases is defined according to State Standard 17.2.2.02- 98 "Norms and methods of definition of a smoke of burnt gases of diesel engines of tractors and self-moving agricultural machines".

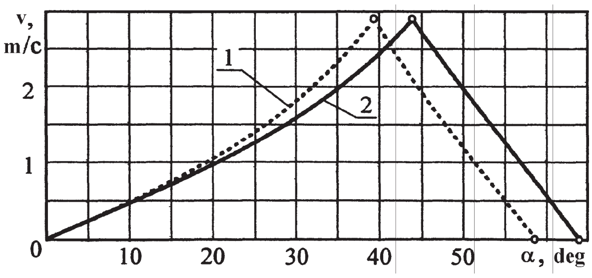

Fig 8. Velocity of plungers for regimes of a diesel engine at n = 2 000 min^{-1}:1 – axial, 2 – disaxial

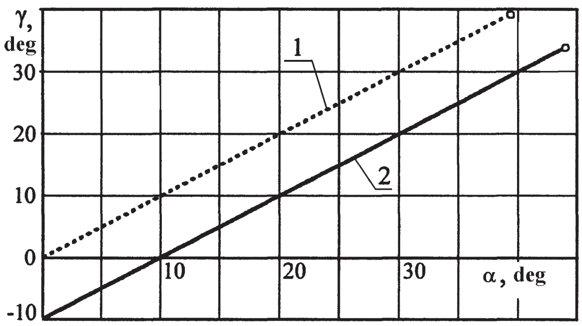

Fig 9. Angles of pressure:1 – axial, 2 – disaxial

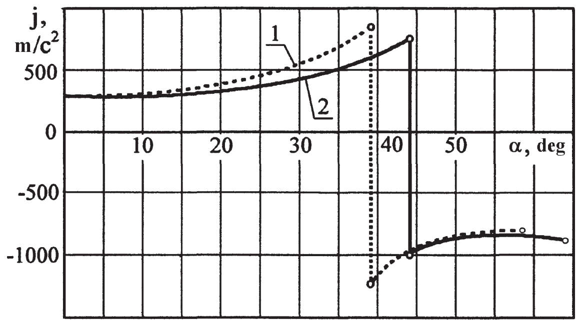

Fig 10. Acceleration of plungers for regimes of a diesel engine at n = 2 000 min^{-1}:1 – axial, 2 – disaxial

Table 2

Preliminary stroke of a plunger h_{pr} (mm)	Pump No 1				Pump No 2	Pump No 3
	4,2	5	5,4	6	5,4	5,0
Parameters of the injector: d_i = 6 mm, h_i = 0,25 mm, μf = 0,3…0,33 mm², P_{s0} = 2 450 $^{+78,4}$ N (250 $^{+8}$ kg)						

3.3. Results

3.3.1. Influence of a preliminary stroke of a plunger of the pump No 1

Change of h_{pr} influences on fuel pressure $p_{f\,max}$ after the holder delivery-valve as follows:

- for regime M_{max}: h_{pr} = 4,2 mm, $p_{f\,max}$ = 463 bar; h_{pr} = 5 mm, $p_{f\,max}$ = 454 bar; h_{pr} = 5,4 mm, $p_{f\,max}$ = 475 bar; h_{pr} = 6 mm, $p_{f\,max}$ = 466 bar (see Fig 11);
- for regime $N_{e\,nom}$: h_{pr} = 4,2 mm, $p_{f\,max}$ = 515 bar; h_{pr} = 5 mm, $p_{f\,max}$ = 527 bar; h_{pr} = 5,4 mm, $p_{f\,max}$ = 543 bar; h_{pr} = 6 mm, $p_{f\,max}$ = 549 bar (see Fig 12).

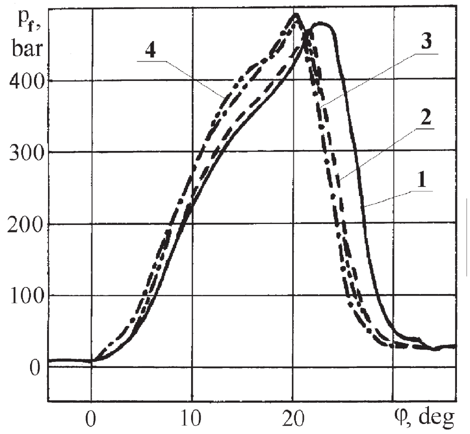

Fig 11. Fuel pressure p_f in the pump No1 for M_{max} regime of the diesel engine (n = 1 500 min^{-1}, N_e = 125 kW):1 – h_{pr} = 4,2 mm, 2 – h_{pr} = 5 mm, 3 – h_{pr} = 5,4 mm, 4 – h_{pr} = 6 mm

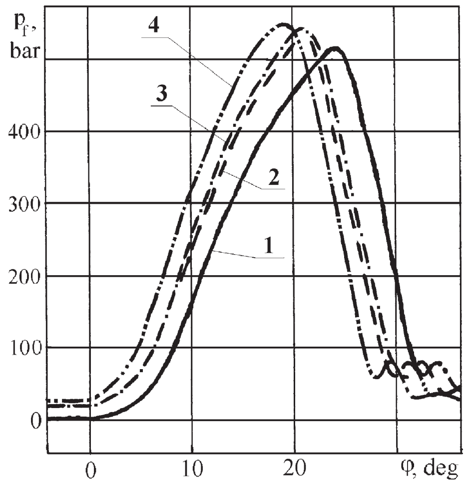

Fig 12. Fuel pressure p_f in the pump No1 for regime N_e = 132,4 kW of the diesel engine (n = 2 000 min^{-1}):1 – h_{pr} = 4,2 mm, 2 – h_{pr} = 5 mm, 3 – h_{pr} = 5,4 mm, 4 – h_{pr} = 6 mm

The increase of h_{pr} reduces specific effective consumption g_e of fuel 0,8–2,0 g/(kW·h) at all regimes of load regulation characteristics. The best results of fuel efficiency are obtained for the pump No 1:

- when setting h_{pr} = 6 mm: g_e = 219…223 g/(kW·h) for regimes n = 2 000 min^{-1} and N_e = 125…132,4 kW; g_e = 216,2…219 g/(kW·h) for regimes n = 1 850 min^{-1} and N_e = 117,6…132,4 kW;
- when setting h_{pr} = 5,4 mm: $g_{e\ min}$ = 208,1 g/(kW·h), n = 1 500 min^{-1} and N_e = 125 kW (regime M_{max}).

3.3.2. Comparison of fuel-injection pumps by economic parameters of a diesel engine

Optimization of angles θ at velocity regime n = 2 000 min^{-1}:

the pump No 1 (h_{pr}=5 mm) – θ=28 °; the pump No 2 (h_{pr}=5,4 mm) – θ=26 °; the pump No 3 (h_{pr}=5 mm) – θ=28 °.

At nominal regime (n = 2 000 min^{-1}, N_e = 132,4 kW) specific consumption is g_e = 223 g/(kW·h) for the pump No 1, it is more 1,0 g/(kW·h) than for the pump No 2 and it is less 1,1 g/(kW·h) than for the pump No 3.

At regime n = 1 850 min^{-1}, N_e = 180 kW consumption is g_e = 216,9 g/(kW·h) for the pump No 1, it is more 0,5 g/(kW·h) than for the pump No 2 and it is less 1,5 g/(kW·h) than for the pump No 3.

At regime n = 1 500 min^{-1}, N_e = 117,6 kW (μ= 17 %) consumption is g_e = 211,3 g/(kW·h) for the pump No 1, it is more 1,0 g/(kW·h) than for the pump No 2 and it is less 1,5 g/(kW·h) than for the pump No 3.

At regime n = 1 200 min^{-1}, N_e = 95,6 kW consumption is g_e = 221,7 g/(kW·h) for the pump No 1, it is more 3,5 g/(kW·h) and 5,8 g/(kW·h) than for the pump No 2 and No 3 accordingly.

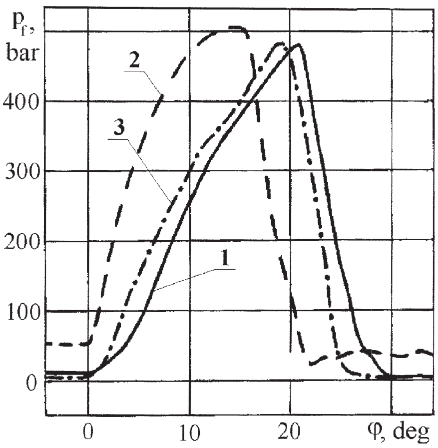

Fig 13. Fuel pressure p_f at M_{max} regime of a diesel engine (n = 1 500 min^{-1}, N_e = 125 kW): 1 – pump No 1, 2 – pump No 2, 3 – pump No 3

At identical regimes of load regulation characteristics (n = 2 000, 1 850, 1 500 min^{-1}) the pressure $p_{f\ max}$ differs insignificantly for the pump No 1, No 3 and it is less 60–90 bar for the pump No 2 in comparison (see Fig 13, 14).

There are $p_{f\ max}$ = 527 bar for the pump No 1; $p_{f\ max}$ = 600 bar for the pump No 2; $p_{f\ max}$ = 533 bar for the pump No 3 at a nominal regime; the duration φ_i of fuel splash is 22,5; 22,5; 21,5 ° accordingly.

The parameters of a diesel engine with the pump No 1 differ insignificantly from the parameters with the pump No 2 at resetting h_{pr} = 5,4 mm, thus $p_{f\ max}$ is lower 70 bar for the pump No 1; parameters of a diesel engine do not differ for pumps No 1 and No 2 at h_{pr} = 6 mm.

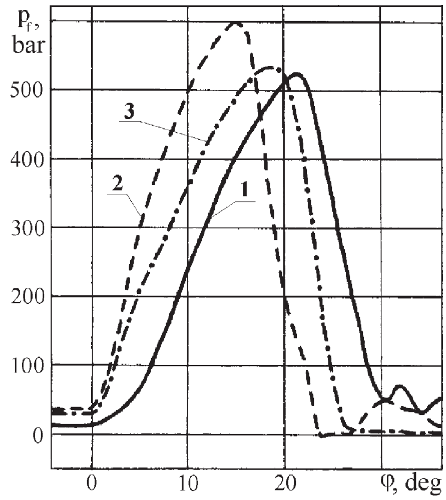

Fig 14. Fuel pressure p_f of a diesel engine N_e = 132,4 kW, n = 2 000 min^{-1}:1 – pump No 1, 2 – pump No 2, 3 – pump No 3

Table 3

n (min^{-1})	N_e (kW)	G_{air} – conditional expenditure (dm^3/c)	K_x standard (%)	K_x real (%)
2 154	2,0	133,3	43	5
2 100	75,2	130,2	43,3	6
2 000	133,6	124	44	8
1 900	134,9	117,8	44,8	7
1 800	137	111,6	45,6	6
1 700	138,2	105,4	46,6	8
1 600	134,6	99,2	47,5	10
1 500	128	93	48,7	11
1 400	118,7	86,8	50	13
1 300	106,6	80,6	51	9
1 200	94,3	74,4	52,4	6

It is necessary to note stable, steady work of a diesel engine with the pump No 1 at all velocity regimes. Smoke K_x of burnt gases of a diesel engine with the pump No 1 corresponds to the requirements of State Standard 17.2202 98 (see Table 3). The graph of plunger velocities, angles of pressure and plunger accelerations at n = 2 000 min $^{-1}$ are submitted in Fig 15–17.

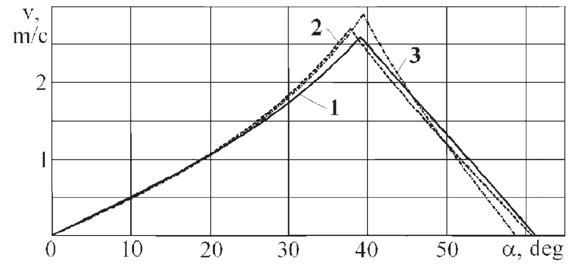

Fig 15. Velocities of a plunger n = 2 000 min^{-1}:1 – pump No 1, 2 – pump No 2, 3 – pump No 3

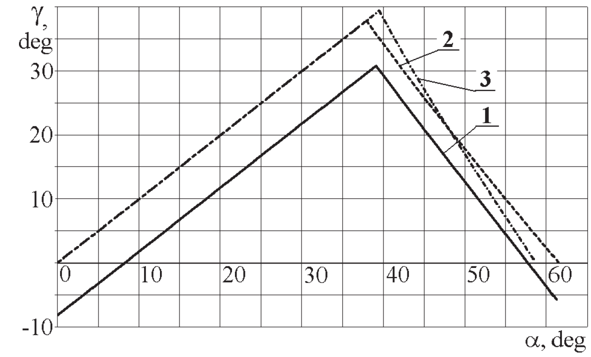

Fig 16. Angles of pressure: 1 – pump No 1, 2 – pump No 2, 3 – pump No 3

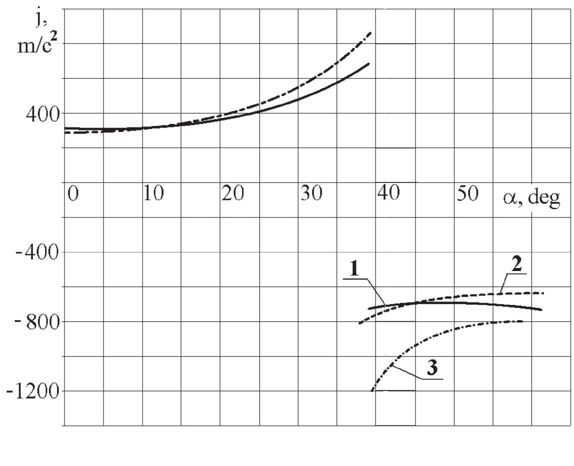

Fig 17. Plunger acceleration n = 2 000 min^{-1}:1 – pump No 1, 2 – pump No 2, 3 – pump No 3

4. Conclusion

Identical parameters of a diesel engine with the pump No 1 (with offset) and the pump No 2 (without offset) are provided. The pump No 2 creates greater fuel pressure. The lower fuel pressure p_f is connected with the lower plunger velocities what is realized in the design of the pump No 1 by the technological reasons of camshaft manufacturing.

The analysis of the test results and recommendations for the design of the disaxial fuel-injection pump are given. It is possible to correct fuel pump design. But the opportunity of reducing σ_{max} and q_{max} in a cam mechanism without the deterioration of diesel engine parameters is now confirmed.

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