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

Evgeny Tausenev¹, Andrew Svistula²

Altai State Technical University, Lenin St., 46, 656038 Barnaul, Russia,
Tel/fax: (+3852368475); e-mail: ¹tayceheb_e_m@mail.ru, ²sae59@mail.ru
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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 σ₂₃max and the side face of a guide sleeve – by pressure q₂₃max. 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 Rₚ = 16 mm, rₚ = 10 mm, h = 9 mm. For traditional pumps γ₂₃max < 27°+40°.

The degree kₚ of a contact unload is 1.1=1.5. Reduction of γ and kₚ under offset is evaluated:
2. Theoretical research

Value $\gamma_{\text{max}}$ 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 $\gamma_{\text{max}}$ 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 $\eta=f(\gamma_{\text{max}})$ which is introduced in Fig 4, when the accounts of the cam mechanisms with different $\gamma_{\text{max}}$ 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.

![Diagram](image)

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

It is proved that the maximum value of $\eta$ for slow-speed and high-speed cam mechanisms with a roller guide sleeve corresponds to $\gamma_{\text{max}} = 30 \pm 40^\circ$, $\eta_{\text{max}} = 0,68 \pm 0,73$.

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

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

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

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

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

$$\sigma_{\text{max}} = 0.418 \sqrt{\frac{E \cdot P_{t_{\text{max}}}}{b_{c} \cdot \cos \gamma} \left(\frac{1}{r_{o}} \pm \frac{1}{R^*}\right)}, \text{MPa},$$

where $P_{t_{\text{max}}}$ – 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; $b_s$ – length of a contact line between a roller and a cam, mm; $\gamma$ – 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 $\sigma = 1\ 500$ 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 $\sigma = 1\ 400+1\ 500$ 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 $\sigma$ 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 $\gamma$.

It follows from the formula (1), that the reduction of $\sigma_{\max}$ 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 $\sigma_{\max}$ is ensured. Thus it is possible to reduce the radius of a round, to increase an active stroke and velocity of a plunger.

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 $\gamma$. It is necessary to diminish $\gamma$ for the decrease of $\sigma_{\max}$. 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:

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

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 $\sigma$, 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 $q_{\max}$, only having lowered $N_{\max}$, 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 $\gamma_{\max}$ 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 5. The Scheme of the operation of powers

Fig 6. The basic sizes of a guide sleeve
sizes of offset, a cam and a roller have the influence not only on $\gamma_{\text{max}}$, but also on $v_{\text{max}}$ – the maximum velocity of a plunger. The offset is the most effective way to decrease a pressure angle. Offset reduces $v_{\text{max}}$ 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 $\gamma_{\text{max}}$ and to keep $v_{\text{max}}$ 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 $\gamma_{\text{max}}$ from 39,4 ° up to 33,9 ° without reducing $v_{\text{max}}$; $\sigma_{\text{max}}$ has decreased 4 %, $q_{\text{max}}$ – 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 $q_{\text{max}}$ 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.

![Fig 7. The disaxial fuel-injection pump](image)

<table>
<thead>
<tr>
<th>Table 1</th>
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<tbody>
<tr>
<td>Design features</td>
</tr>
<tr>
<td>Diameter of a plunger d (mm)</td>
</tr>
<tr>
<td>Stroke of a plunger h (mm)</td>
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<tr>
<td>Units spacing (mm)</td>
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<tr>
<td>Type of the body of a fuel-injection pump</td>
</tr>
<tr>
<td>Diameter of the delivery-valve (mm)</td>
</tr>
<tr>
<td>Preliminary stroke of a plunger hpr (mm)</td>
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<tr>
<td>Cam profile</td>
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<tr>
<td>Radius of pitch circle of a cam (mm)</td>
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<tr>
<td>Radius of a roller (mm)</td>
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<tr>
<td>Radius of a rounding (mm)</td>
</tr>
<tr>
<td>Size offset (mm)</td>
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<tr>
<td>Type of a governor</td>
</tr>
<tr>
<td>Sizes (mm):</td>
</tr>
<tr>
<td>length</td>
</tr>
<tr>
<td>width</td>
</tr>
<tr>
<td>height</td>
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<tr>
<td>Weight (kg)</td>
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</table>
3.1. Purpose: functional trials of the pump No 1, comparison of injection parameters for different pumps with sizes $d \times h = 10 \times 12$ mm, the estimation of pumps loading.

3.2. Technique: load regulation characteristics were registered at $n = 2000, 1850, 1500, 1200$ min$^{-1}$, the optimum profitability angle $\theta$ 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 $\theta = 28^\circ$ for the pump No 1 with a preliminary stroke of a plunger 5 mm; smoke $K_s$ 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 $n = 2000$ min$^{-1}$: 1 – axial, 2 – disaxial](image1)

![Fig 9. Angles of pressure: 1 – axial, 2 – disaxial](image2)

![Fig 10. Acceleration of plungers for regimes of a diesel engine at $n = 2000$ min$^{-1}$: 1 – axial, 2 – disaxial](image3)

![Fig 11. Fuel pressure $p_f$ in the pump No 1 for $M_{\text{max}}$ regime of the diesel engine ($n = 1500$ min$^{-1}$, $N_2 = 125$ kW): 1 – $h_{pr} = 4.2$ mm, 2 – $h_{pr} = 5$ mm, 3 – $h_{pr} = 5.4$ mm, 4 – $h_{pr} = 6$ mm](image4)

![Fig 12. Fuel pressure $p_f$ in the pump No 1 for regime $N_2 = 132.4$ kW of the diesel engine ($n = 2000$ min$^{-1}$): 1 – $h_{pr} = 4.2$ mm, 2 – $h_{pr} = 5$ mm, 3 – $h_{pr} = 5.4$ mm, 4 – $h_{pr} = 6$ mm](image5)

Table 2

<table>
<thead>
<tr>
<th>Preliminary stroke of a plunger $h_{pr}$ (mm)</th>
<th>Pump No 1</th>
<th>Pump No 2</th>
<th>Pump No 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.2</td>
<td>5</td>
<td>5.4</td>
<td>6</td>
</tr>
<tr>
<td>5.4</td>
<td>5</td>
<td>5.0</td>
<td></td>
</tr>
</tbody>
</table>

Parameters of the injector: $d_i = 6$ mm, $h_i = 0.25$ mm, $\mu = 0.3...0.33$ mm$^2$, $P_{\text{inj}} = 2450 \pm 70.4$ N (250...18 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 $p_{f\text{,max}}$ after the holder delivery-valve as follows:

- for regime $M_{\text{max}}$: $h_{pr} = 4.2$ mm, $p_{f\text{,max}} = 463$ bar; $h_{pr} = 5$ mm, $p_{f\text{,max}} = 454$ bar; $h_{pr} = 5.4$ mm, $p_{f\text{,max}} = 475$ bar; $h_{pr} = 6$ mm, $p_{f\text{,max}} = 466$ bar (see Fig 11);
- for regime $N_{2\text{,nom}}$: $h_{pr} = 4.2$ mm, $p_{f\text{,max}} = 515$ bar; $h_{pr} = 5$ mm, $p_{f\text{,max}} = 527$ bar; $h_{pr} = 5.4$ mm, $p_{f\text{,max}} = 543$ bar; $h_{pr} = 6$ mm, $p_{f\text{,max}} = 549$ bar (see Fig 12).
The increase of $h_{pr}$ reduces specific effective consumption $g_c$ 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: $g_c = 219.0–223$ g/(kW·h) for regimes $n = 2,000$ min$^{-1}$ and $N_e = 125.0–132.4$ kW; $g_c = 216.2–219$ 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: $g_{c\text{ min}} = 208.1$ g/(kW·h), $n = 1,500$ min$^{-1}$ and $N_e = 125$ kW (regime $M_{\text{max}}$).

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

Optimization of angles $\theta$ at velocity regime $n = 2,000$ min$^{-1}$:

- the pump No 1 ($h_{pr} = 5$ mm) $- \theta = 28^\circ$; the pump No 2 ($h_{pr} = 5.4$ mm) $- \theta = 26^\circ$; the pump No 3 ($h_{pr} = 5$ mm) $- \theta = 28^\circ$.

At nominal regime ($n = 2,000$ min$^{-1}$, $N_e = 132.4$ kW) specific consumption is $g_c = 223$ 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 $g_c = 216.9$ 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 ($\mu = 17\%$) consumption is $g_c = 211.3$ 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 $g_c = 221.7$ 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.

At identical regimes of load regulation characteristics ($n = 2,000$, $1,850$, $1,500$ min$^{-1}$) the pressure $P_{l_{\text{max}}}$ 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 $P_{l_{\text{max}}} = 527$ bar for the pump No 1; $P_{l_{\text{max}}} = 600$ bar for the pump No 2; $P_{l_{\text{max}}} = 533$ bar for the pump No 3 at a nominal regime; the duration $\phi$ of fuel splash is $22.5^\circ; 22.5^\circ; 21.5^\circ$ 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 $P_{l_{\text{max}}}$ 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 13. Fuel pressure $P_l$ at $M_{\text{max}}$ 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](image)

![Fig 14. Fuel pressure $P_l$ 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](image)

Table 3

<table>
<thead>
<tr>
<th>$n$ (min$^{-1}$)</th>
<th>$N_e$ (kW)</th>
<th>$G_{\text{pr}}$ – conditional expenditure (dm$^3$/c)</th>
<th>$K_s$ standard (%)</th>
<th>$K_r$ real (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2,154</td>
<td>2.0</td>
<td>133.3</td>
<td>43</td>
<td>5</td>
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<tr>
<td>2,100</td>
<td>75.2</td>
<td>130.2</td>
<td>43.3</td>
<td>6</td>
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<tr>
<td>2,000</td>
<td>133.6</td>
<td>124</td>
<td>44</td>
<td>8</td>
</tr>
<tr>
<td>1,900</td>
<td>134.9</td>
<td>117.8</td>
<td>44.8</td>
<td>7</td>
</tr>
<tr>
<td>1,800</td>
<td>137</td>
<td>111.6</td>
<td>45.6</td>
<td>6</td>
</tr>
<tr>
<td>1,700</td>
<td>138.2</td>
<td>105.4</td>
<td>46.6</td>
<td>8</td>
</tr>
<tr>
<td>1,600</td>
<td>134.6</td>
<td>99.2</td>
<td>47.5</td>
<td>10</td>
</tr>
<tr>
<td>1,500</td>
<td>128</td>
<td>93</td>
<td>48.7</td>
<td>11</td>
</tr>
<tr>
<td>1,400</td>
<td>118.7</td>
<td>86.8</td>
<td>50</td>
<td>13</td>
</tr>
<tr>
<td>1,300</td>
<td>106.6</td>
<td>80.6</td>
<td>51</td>
<td>9</td>
</tr>
<tr>
<td>1,200</td>
<td>94.3</td>
<td>74.4</td>
<td>52.4</td>
<td>6</td>
</tr>
</tbody>
</table>
It is necessary to note stable, steady work of a diesel engine with the pump No 1 at all velocity regimes. Smoke $K_v$ 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 $n = 2000 \text{ min}^{-1}$ are submitted in Fig 15–17.

![Fig 15. Velocities of a plunger $n = 2000 \text{ min}^{-1}$: 1 – pump No 1, 2 – pump No 2, 3 – pump No 3](image)

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 $q_{\text{max}}$ and $q_{\text{max}}$ in a cam mechanism without the deterioration of diesel engine parameters is now confirmed.

References


