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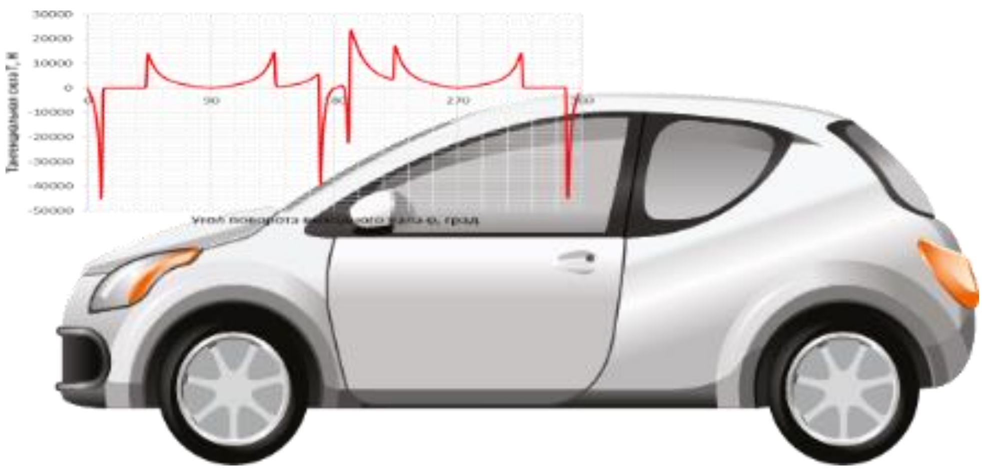
Calculation of a petrol engine for a car

with a power of 71 kW (97 hp),

equipped with a new power take-off device



1 Background of the calculation

A preliminary comparative evaluation of an engine equipped with a patented new type power take-off device can be made on the basis of calculating the operating cycle parameters of a piston engine and determining the kinematic and dynamic parameters of a power take-off device.

2 Initial data for design

2.1 Power and rotational speed of the crankshaft

When calculating the rated power of the engine N_e is usually set. The choice or setting of the rated power is determined primarily by the purpose of the engine (for cars, trucks or tractors), its type (gasoline, diesel), operating conditions.

The nominal rotational speed of the crankshaft is taken to be reduced by 2 times compared with the rotational speed of the engine being compared for a similar purpose

$n_н$ = 3000 minute $^{-1}$.

2.2. Number and arrangement of cylinders

The choice of the number of cylinders and their location depend on the power, diagnostic and design factors. The most common four-cylinder and six-cylinder tractor diesel engines. Select in accordance with the characteristics of the kinematics of the PTO number of cylinders i = 4.

The location of the cylinders in accordance with the structural scheme.

2.3 Cylinder dimensions and piston speed

The dimensions of the cylinder (diameter and stroke) are the main design parameters of the engine. The diameter D (mm) of various engines varies widely: from 60 to 150 mm.

The piston stroke is usually characterized by the S/D ratio, which is directly related to the piston speed. Depending on the S/D value, there are distinguished short-stroke (S/D <1) and long-stroke (S/D> 1) engines. In short-stroke engines, the engine height and weight are lower, the indicator efficiency and filling ratio are higher, the piston speed is slower and the engine component is more durable.

Considering the above, we assume a preliminary cylinder diameter D = 60 mm and S = 56 mm, i.e. the engine is short-stroke

The piston speed $V_{п.ср}$ is a criterion for engine speed.

$$V_{п\,ср} = \frac{4 \cdot S \cdot n_н}{60} = \frac{4 \cdot 56 \cdot 10^{-3} \cdot 3000}{60} = 11{,}2 \text{ m/s}$$

Since the obtained value $V_{п.ср}$ = 6,5 m / s, the engine is high-speed.

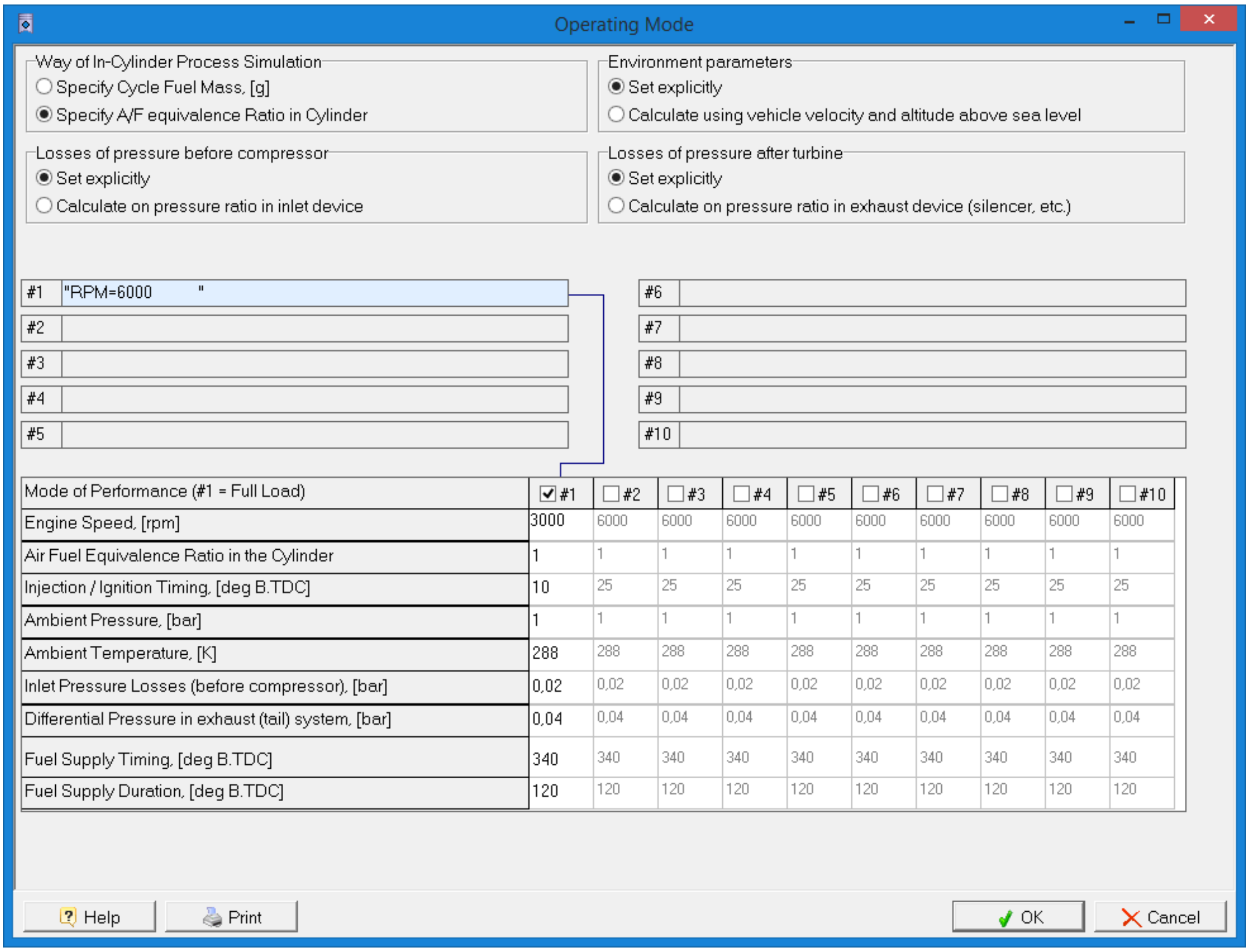
2.4 Compression ratio

The compression ratio is one of the most important characteristics of the engine. Its choice depends, first of all, on the method of fuel mixing. In addition, the degree of compression is chosen taking into account the presence or absence of pressurization, engine speed, cooling system and other factors.

Accepted for the calculated engine compression ratio ε = 11,5 as for engines of a similar purpose, which allows the use of similar fuel.

3 Calculation of engine workflow

The calculation was performed using the program Diesel-RK. The initial data for the calculation of the workflow are presented in Fig. 1.


Operating Mode

Way of In-Cylinder Process Simulation
- ○ Specify Cycle Fuel Mass, [g]
- ◉ Specify A/F equivalence Ratio in Cylinder

Environment parameters
- ◉ Set explicitly
- ○ Calculate using vehicle velocity and altitude above sea level

Losses of pressure before compressor
- ◉ Set explicitly
- ○ Calculate on pressure ratio in inlet device

Losses of pressure after turbine
- ◉ Set explicitly
- ○ Calculate on pressure ratio in exhaust device (silencer, etc.)

#1 "RPM=6000 "
#2
#3
#4
#5
#6
#7
#8
#9
#10

Mode of Performance (#1 = Full Load)	☑ #1	☐ #2	☐ #3	☐ #4	☐ #5	☐ #6	☐ #7	☐ #8	☐ #9	☐ #10
Engine Speed, [rpm]	3000	6000	6000	6000	6000	6000	6000	6000	6000	6000
Air Fuel Equivalence Ratio in the Cylinder	1	1	1	1	1	1	1	1	1	1
Injection / Ignition Timing, [deg B.TDC]	10	25	25	25	25	25	25	25	25	25
Ambient Pressure, [bar]	1	1	1	1	1	1	1	1	1	1
Ambient Temperature, [K]	288	288	288	288	288	288	288	288	288	288
Inlet Pressure Losses (before compressor), [bar]	0,02	0,02	0,02	0,02	0,02	0,02	0,02	0,02	0,02	0,02
Differential Pressure in exhaust (tail) system, [bar]	0,04	0,04	0,04	0,04	0,04	0,04	0,04	0,04	0,04	0,04
Fuel Supply Timing, [deg B.TDC]	340	340	340	340	340	340	340	340	340	340
Fuel Supply Duration, [deg B.TDC]	120	120	120	120	120	120	120	120	120	120

Help Print OK Cancel

Figure 1 - Window for setting the parameters of the engine operating mode

The results of the calculation of the workflow

```
2019-05-02 15-47-42 "Yaris_1"
 Mode: #1 :: "RPM=3000            ";
 Title: "A/F eq. defines m_f"
 www.diesel-rk.bmstu.ru
  Fuel:      Petrol regular

   ----------------- PARAMETERS OF EFFICIENCY AND POWER ----------------
   3000.0     - RPM      - Engine Speed, rev/min
   33.615     - P_eng    - Piston Engine Power, kW
   10.647     - BMEP     - Brake Mean Effective Pressure, bar
  0.01176     - m_f      - Mass of Fuel Supplied per cycle, g
  0.18114     - SFC      - Specific Fuel Consumption, kg/kWh
  0.17940     - SFC_ISO  - Specific Fuel Consumption in ISO, kg/kWh
  0.32578     - Eta_f    - Efficiency of piston engine
   12.755     - IMEP     - Indicated Mean Effective Pressure, bar
  0.39029     - Eta_i    - Indicated Efficiency
   11.200     - Sp       - Mean Piston Speed, m/s
   1.7302     - FMEP     - Friction Mean Effective Pressure, bar (Intern.Exp)
  0.86021     - Eta_m    - Mechanical Efficiency of Piston Engine

   --------------------- ENVIRONMENTAL PARAMETERS ----------------------
   1.0000     - po_amb   - Total Ambient Pressure, bar
   288.00     - To_amb   - Total Ambient Temperature, K
   1.0000     - p_Te     - Exhaust Back Pressure, bar (after turbine)
  0.98000     - po_afltr - Total Pressure after Induction Air Filter, bar

   ------------------ TURBOCHARGING AND GAS EXCHANGE -------------------
  0.98000     - p_C      - Pressure before Inlet Manifold, bar
   288.00     - T_C      - Temperature before Inlet Manifold, K
  0.03528     - m_air    - Total Mass Airflow (+EGR) of Piston Engine, kg/s
   0.0000     - Eta_TC   - Turbocharger Efficiency
   1.0459     - po_T     - Average Total Turbine Inlet Pressure, bar
   908.10     - To_T     - Average Total Turbine Inlet Temperature, K
  0.03737     - m_gas    - Mass Exhaust Gasflow of Pison Engine, kg/s
  0.99999     - A/F_eq.t - Total Air Fuel Equivalence Ratio (Lambda)
   1.0000     - F/A_eq.t - Total Fuel Air Equivalence Ratio
 -0.37799     - PMEP     - Pumping Mean Effective Pressure, bar
  0.96608     - Eta_v    - Volumetric Efficiency
  0.94676     - Eta_vo   - Volumetric Efficiency defined by Ambient
Parameters
  0.04592     - x_r      - Residual Gas Mass Fraction
   1.0002     - Phi      - Coeff. of Scavenging (Delivery Ratio / Eta_v)
   1.1533     - BF_int   - Burnt Gas Fraction Backflowed into the Intake, %
  0.69309     - %Blow-by - % of Blow-by through piston rings

   -------------------------- INTAKE SYSTEM --------------------------
  0.95986     - p_int    - Average Intake Manifold Pressure, bar
   299.72     - T_int    - Average Intake Manifold Temperature, K
   27.887     - v_int    - Average Gas Velocity in intake manifold, m/s
   349.72     - Tw_int   - Average Intake Manifold Wall Temperature, K
   102.82     - hc_int   - Heat Transfer Coeff. in Intake Manifold, W/(m2*K)
   290.87     - hc_int.p - Heat Transfer Coeff. in Intake Port, W/(m2*K)
   140.20     - v_int.p  - Max Velocity in a Middle Section of Int. Port, m/s
   3.1813     - A_v.thrt - Total Effective Valve Port Throat Area, cm2
Valve Dim. Estim.: Num=1 Dv= 26.4 Dt= 24.8 Ds= 6.2 Lv=  8.6 Lv_max=  6.6 mm

   ------------------------- EXHAUST SYSTEM --------------------------
   1.0398     - p_exh    - Average Exhaust Manifold Gas Pressure, bar
   906.86     - T_exh    - Average Exhaust Manifold Gas Temperature, K
   59.552     - v_exh    - Average Gas Velocity in exhaust manifold, m/s
   19.139     - Sh       - Strouhal number: Sh=a*Tau/L (has to be: Sh > 8)
   824.50     - Tw_exh   - Average Exhaust Manifold Wall Temperature, K
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   237.40     - hc_exh   - Heat Transfer Coeff. in Exhaust Manifold, W/(m2*K)
   1080.8     - hc_exh.p - Heat Transfer Coeff. in Exhaust Port, W/(m2*K)
   353.91     - v_exh.p  - Max Velocity in a Middle Section of Exh. Port, m/s
   2.7000     - A_v.thrt - Total Effective Valve Port Throat Area, cm2
Valve Dim. Estim.: Num=1 Dv= 24.0 Dt= 22.4 Ds= 5.7 Lv=  5.7 Lv_max=  6.0 mm

   ---------------------------- COMBUSTION -----------------------------
   1.0000     - A/F_eq   - Air Fiel Equival. Ratio (Lambda) in the Cylinder
   1.0000     - F/A_eq   - Fuel Air Equivalence Ratio in the Cylinder
   64.706     - p_max    - Maximum Cylinder Pressure, bar
   2537.8     - T_max    - Maximum Cylinder Temperature, K
   18.000     - CA_p.max - Angle of Max. Cylinder Pressure, deg. A.TDC
   26.000     - CA_t.max - Angle of Max. Cylinder Temperature, deg. A.TDC
   1.6336     - dp/dTheta- Max. Rate of Pressure Rise, bar/deg.
   4.8709     - Ring_Intn- Ringing / Knock Intensity, MW/m2
   1853.3     - F_max    - Max. Gas Force acting on the piston, kg
   10.000     - SOI      - Start Of Injection or Ignition Timing, deg. B.TDC
  0.30792     - Phi_ign  - Ignition Delay Period, deg.
   9.6921     - SOC      - Start of Combustion, deg. B.TDC
   44.000     - Phi_z    - Combustion duration, deg.
   2.3425     - m_w      - Wiebe's Factor in the Cylinder
   72.390     - ON       - Minimum Octane Number of fuel (knock limit)

   ------------------------ ECOLOGICAL PARAMETERS ----------------------
   3328.7     - NOx.w,ppm- Fraction of wet NOx in exh. gas, ppm
   14.695     - NO       - Specif. NOx emiss. reduc. to NO, g/kWh (Zeldovich)
   0.0000     - SO2      - Specific SO2 emission, g/kWh

   ------------------------- CYLINDER PARAMETERS -----------------------
   1.5899     - p_ivc    - Pressure at IVC, bar
   385.57     - T_ivc    - Temperature at IVC, K
   28.055     - p_tdc    - Compression Pressure (at TDC), bar
   775.17     - T_tdc    - Compression Temperature (at TDC), K
   6.7620     - p_evo    - Pressure at EVO, bar
   1555.4     - T_evo    - Temperaure at EVO, K

   ------------------ HEAT EXCHANGE IN THE CYLINDER --------------------
   1258.2     - T_eq     - Average Equivalent Temperature of Cycle, K
   738.90     - hc_c     - Aver. Factor of Heat Transfer in Cyl., W/m2/K
   468.81     - Tw_pist  - Average Piston Crown Temperature, K
   413.00     - Tw_liner - Average Cylinder Liner Temperature, K
   434.08     - Tw_head  - Average Head Wall Temperature, K
   412.22     - Tw_cool  - Average Temperature of Cooled Surface
                 head      of Cylinder Head, K
   386.65     - Tboil    - Boiling Temp. in Liquid Cooling System, K
   12235.     - hc_cool  - Average Factor of Heat Transfer, W/(m2*K)
                           from head cooled surface to coolant
   1721.6     - q_head   - Heat Flow in a Cylinder Head, J/s
   1649.0     - q_pist   - Heat Flow in a Piston Crown, J/s
   1671.4     - q_liner  - Heat Flow in a Cylinder Liner, J/s

   --------------- MAIN ENGINE CONSTRUCTION PARAMETERS -----------------
   11.500     - CR       - Compression Ratio
   71.000     - EVO      - Exhaust Valve Opening, deg. before BDC
   16.000     - EVC      - Exhaust Valve Closing, deg. after  DC
   16.000     - IVO      - Intake Valve Opening, deg. before DC
   70.000     - IVC      - Intake Valve Closing, deg. after BDC
____________________
Versions:     Kernel 04.01.13;  RK-model Not used;  NOx-model 22.02.13
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4 Kinematic and dynamic calculations

Diagrams of forces acting in the new power take-off mechanism are presented in Fig. 1.

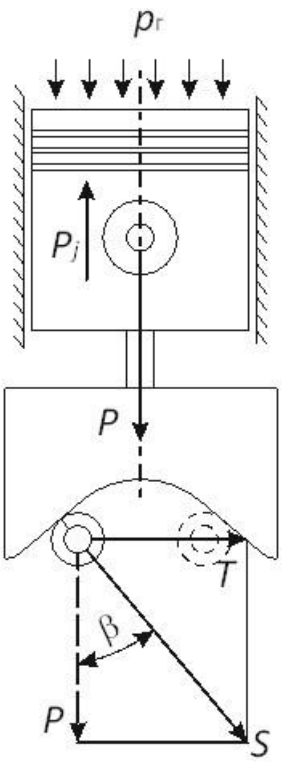

Рисунок 1

Let us describe the scheme mechanism with a power take-off device (PTO): the gas pressure force $P_г$ is transferred from the piston to the guide, which interacts with its profile surface with a roller. In turn, the roller is located at the end of the lever. As a result of this interaction, the roller, rolling along the guide, turns the lever in a plane perpendicular to the plane of the drawing, doing useful work.

The main driving force is the pressure force of gases $P_г$:

$$P_г = p_г \cdot F_п ,$$

where $p_г$ – cylinder gas pressure; $F_п$ – piston bottom area.

From the assumption of the identity of the working processes of engines with the mechanisms under study, we first assume that the forces $P_г$ in the mechanisms under study are equal.

When transferring the pressure force of gases, one should take into account the force of inertia of reciprocating moving masses P_j, arising as a result of a change in the velocity and direction of movement of these masses. This power is defined as:

$$P_j = -m_{вп} \cdot j_п ,$$

Where $m_{вп}$ – mass of reciprocating moving parts; $j_п$ – acceleration of parts (piston).

The resulting force is transformed in the mechanism:

$$P = P_r + P_j.$$

Graphs of the resulting forces are presented in Fig. 2 and 3.

4.1 Kinematic analysis of the new power take-off

The kinematic parameters for the new power take-off mechanism will be determined by constructing the profile of the sweep of the guide shown in Fig. 2.

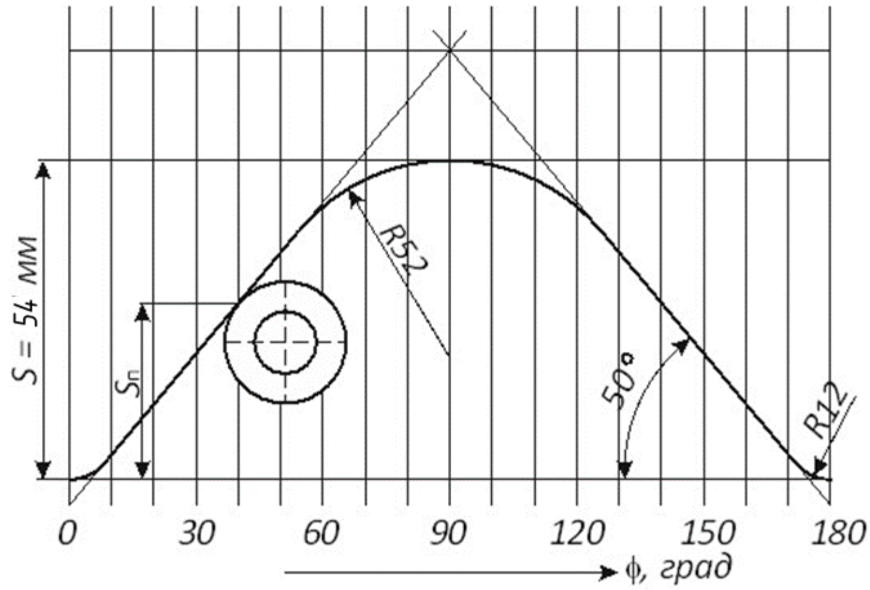

Figure 2 - Scanning of the profile of the guide mechanism with PTO.

The movement of the piston $S_п^{УОМ}$ is determined by measuring the coordinates at appropriate angles ϕ of rotation of the roller relative to the guide.

Speed $v_п^{УОМ}$ and acceleration $j_п^{УОМ}$ determined by numerical differentiation of the table function $S_п^{УОМ} = f(t)$:

$$v_п^{УОМ} = \frac{dS_п^{УОМ}}{dt}; \qquad j_п^{УОМ} = \frac{dv_п^{УОМ}}{dt},$$

where t – time determined by the angle of rotation of the roller ($t = \frac{180 \cdot \varphi}{\pi \cdot \omega}$).

The calculation results are presented in Fig. 3. A feature of the kinematics of the engine with the new mechanism is that the full working cycle (4 piston strokes) takes place in one revolution of the lever with the roller, i.e. at 360 °, while in the crank end of the engine the working cycle takes place in two revolutions of the crankshaft, i.e. over 720 °.

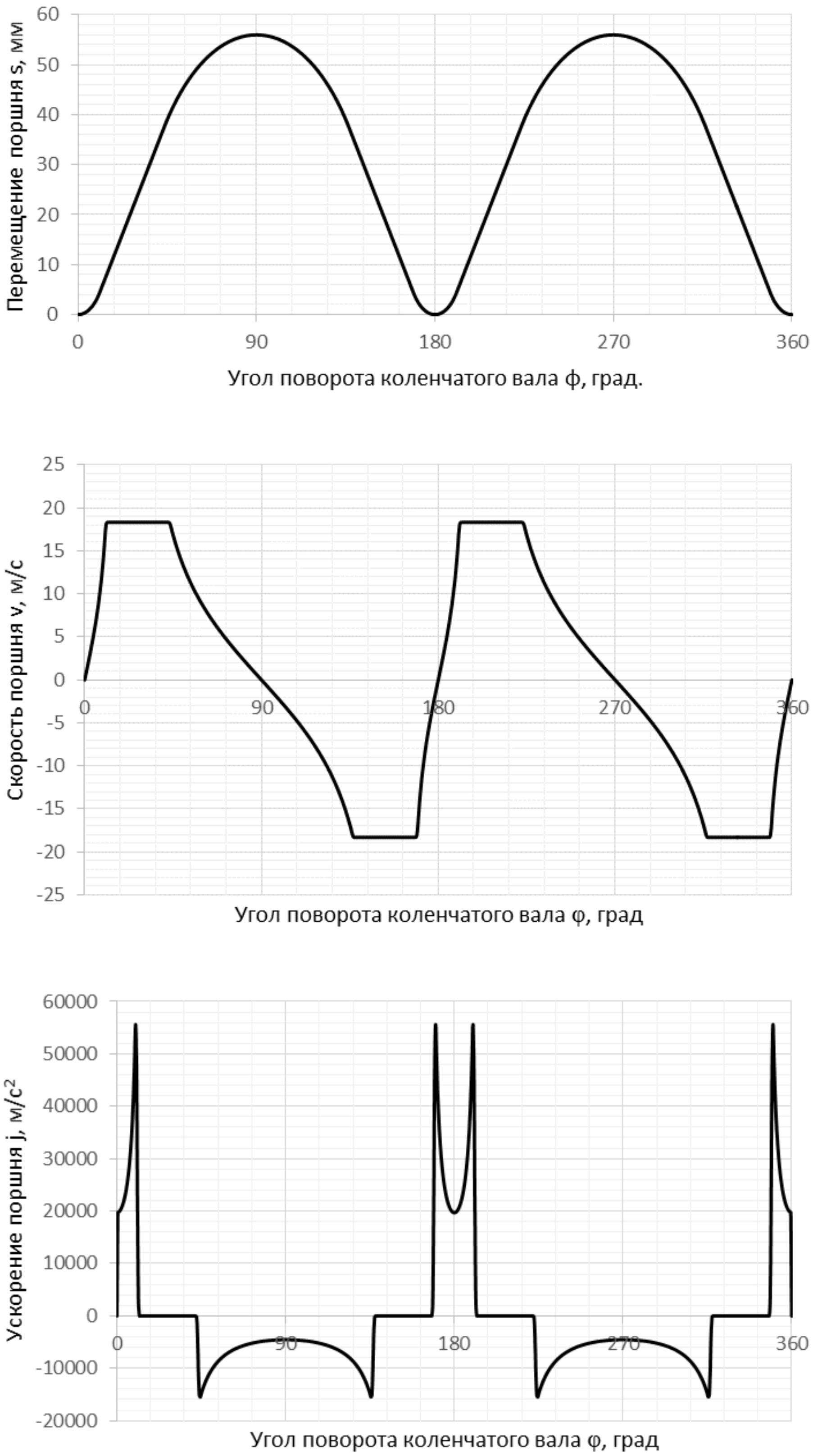

Figure 3 - Kinematic parameters of the engine with PTO

4.2 Dynamic analysis of mechanisms

Torque on the output shaft of the engine is created by the tangential force T (Fig. 1). Analytical dependence of tangential force:

$$T = P \cdot \mathrm{tg}\beta .$$

It should be noted that the K_r inertia force of the rotating CM masses acting in the plane of action of the force T (Fig. 1) is absent in an engine with an PTO, since acts in a plane perpendicular to the plane of figure 1, b and does not affect the tangential force T.

The average value of the total tangential force of a four-cylinder engine is determined by summing the tangential forces of individual cylinders in accordance with the order of operation of the cylinders..

$$T_{\Sigma} = \sum_{1}^{i=4} T_i .$$

In this case, the phase shift angle for an engine with a new power take-off mechanism — 90 ° rotation of the output shaft.

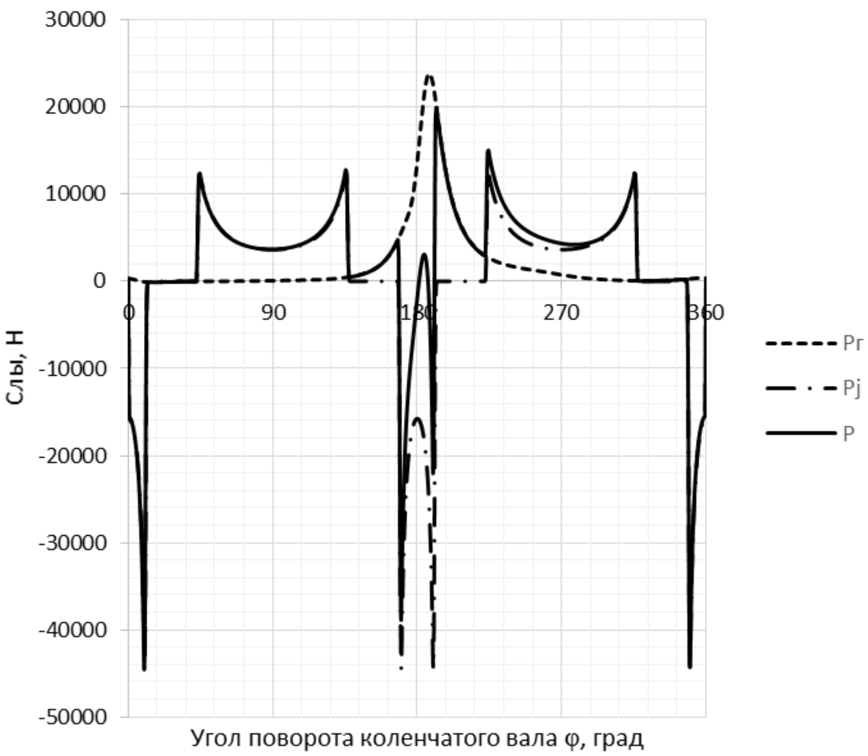

Figure 4 - Graphs of the resulting forces.

Figure 5 - Graphs of tangential forces

5 Power indicators

Indicator torque was defined as

$$M_i = T_\Sigma \cdot r_к ,$$

Effective power:

$$N_e = \frac{2 \cdot \pi}{60} \cdot M_i \cdot n_н \cdot \eta_м \cdot 10^{-3} ,$$

where $\eta_м$ – mechanical efficiency of the engine..

Comparative results of calculations for the 1NR-FE engines and the engine with a new PTO are presented in the table.

Name	Values for:	
	engine with CM (1NR-FE)	engine with PTO
Bore *D*, mm	110	60
Piston stroke *S*, mm	125	56
Number of cylinders *i*	4	
Litrage, l	1,329	0,633
Output shaft speed nн, rpm	6000	3000
Maximum workflow pressure p_z, MPa	7,0	6,7
Maximum workflow temperature T_z, K	2690	2538
Mean effective pressure p_e MPa	1,11	1,06
Indicator efficiency of the workflow η_i	0,395	0,390
Mechanical engine efficiency $\eta_м$	0,86	0,70
Average total tangential force T_Σ, N	3206	6633
Crank radius $r_к$, mm	40,25	48,91
Indicator torque M_i, N·m	129,1	324,4
Effective power N_e, kW	70,0	97,0
Hourly fuel consumption $G_т$, kg / h	18,9	13,6