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THEORETICAL RESEARCH RESPONSE TIME OF THE MECHANISM FOR COMPRESSION RATIO CHANGING OF THE CONROD-FREE ENGINE

Summary. The mathematical model for calculating the response time of the compression ratio of the mechanism for compression ratio changing (MCRC) is presented in this article. This revealed the influence of various engine factors with

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the connecting rod and crank mechanism (CRCM) on the operation of the MCRC (for example, the rate of the compression ratio change). The results of the study of the operation of the MCRC indicate a strong influence of the values δ (relative area of flow passage of channels) and p (pressure) on the response time τ in the field of their small values. This indicates that with insignificant changes of the area of flow passage of channel of the hydraulic lock and the minute oil pressure in the cavities, a significant response rate of the MCRC is ensured. The results demonstrate the possibility of speedy compression ratio change in the engine with the MCRC. Calculation studies showed that the mechanism full operation occurs quickly (0.02 s per unit ε_x), which indicates the expediency of using such a high-pressure pump in a four-stroke gasoline engine with CRCM. The mechanism movable body complete movement ($S = 4$ mm) at the oil temperature in the hydraulic system of 45°C and pressure on the body of $p = 60$ bar are stated to occur for 0.2 s.

Keywords: mechanism for compression ratio changing, connecting rod and crank mechanism, conrod-free engine, response time, pressure, oil temperature.

1. INTRODUCTION

Substantial progress in fuel efficiency improvement and exhaust gas toxic components reduction in automobile engines have become topical issues in recent times [1-3]. Increased state control over environmental cleanliness and the economic use of nonrenewable natural resources favoured it [4-6]. Key automotive companies participate in the state standards development and compete for consumers; they have to improve their engines and use up-to-date design and technology solutions [7-11].

Innovations are implemented in up-to-date engines [12,13]. They focus improvements in the engine systems and mechanisms and transmission units. Fuel equipment elements that improve operating procedures play a special role in the engine design perfection. In this case, the piston motion law is push-type and depends only on the connecting rod and crank mechanism (CRCM) constant parameters [14-16]. This circumstance does not allow the use of compression ratio regulation being a powerful reserve to optimise engine operating parameters in its running regimes range [17-19].

The compression ratio is believed to be the internal combustion engine (ICE) constant design parameter as, for example, the cylinder diameter is. The compression ratio quantity is definitely determined by the CRCM dimensions and the cylinder head location relative to the crankshaft axis in the conventional ICE [20,21].

The ICE power and fuel efficiency are known to rise with the compression ratio growth due to the indicated efficiency increase. The ICE performance improvement breaks off when the compression ratio reaches 13...14 due to increase in the mechanical losses. Therefore, the compression ratio values are optimal.

The compression ratio value is built into the engine design and differs from the optimum value. The compression ratio in gasoline engines is limited by detonation. It is less than optimal and, as a rule, does not exceed 10.

Numerous calculations and experimental studies show that the compression ratio regulation can ensure 20% of fuel economy improvement.

The boost pressure in the gasoline engine can be increased without detonation when the compression ratio is reduced, and the power-to-volume ratio is increased with a positive effect

along with it. The fuel economy improved under light-load conditions with the compression ratio increase [22,23].

The compression ratio maximum value does not exceed $\varepsilon = 14$ in most ICE designs with variable compression ratio. The frictional losses increase with the further increase of the engine compression ratio. The indicator efficiency, in this case, does not compensate these losses, so effective efficiency decreases as a result.

2. THE CONSTRUCTION OF THE CONROD-FREE ICE

The conrod-free ICE (CFICE) is given realisation of a change compression ratio more than $\varepsilon = 14$. This is one of the ICE possible designs. The crank and rocker arm mechanism (CRAM) is used in the engine instead of the traditional CRCM (Figure 1).

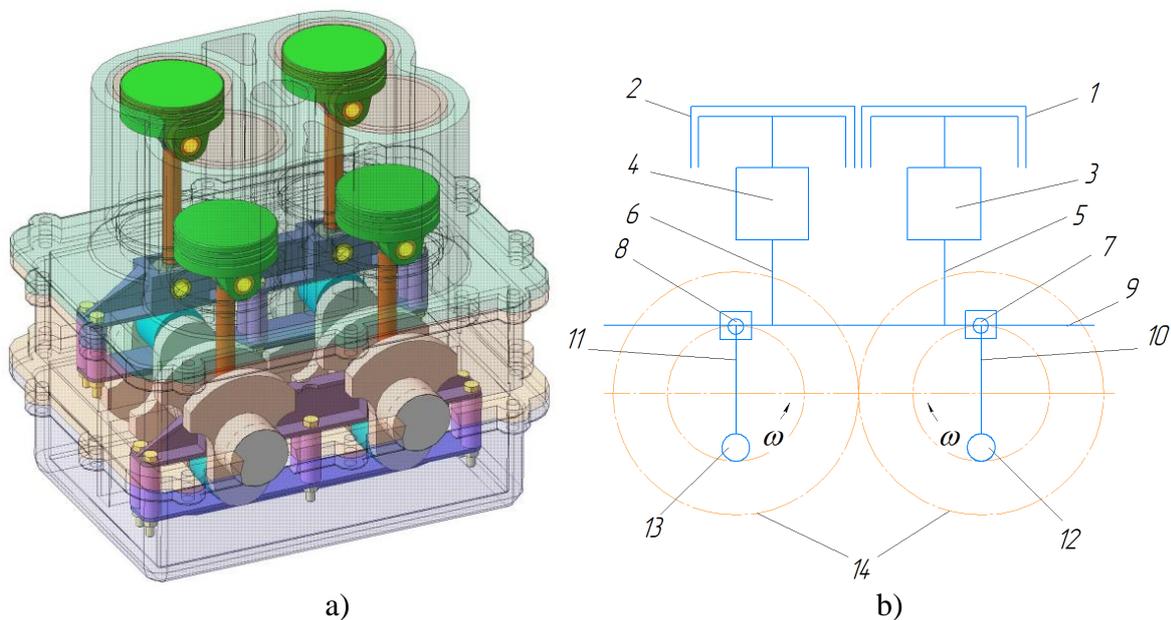a)   b)

Fig. 1. The graphic presentation of the conrod-free engine: (a) general view; (b) diagram of the conrod-free engine with the MCRC; 1, 2 – piston; 3, 4 – MCRC; 5, 6 – piston rod; 7, 8 – crosshead; 9 – rocker arm; 10, 11 – crankshaft; 12, 13 – crankcase counterweight; 14 – gear

Experimental design efforts to create conrod-free engines are carried out in Germany, the USA [24,25], Ukraine [26] and other countries [9,18,27]. The engine distinctive features are as follows:

- low friction losses in the entire load range and shaft speed
- full dynamic balance and stroke uniformity
- compactness and low specific gravity

The piston strictly rectilinear motion ensuring absence of lateral loads is the advantage of the CFICE.

The analysis of the CFICE with a CRAM shows:

- increase effective efficiency at $\varepsilon > 14$ small mechanical losses
- constructively is implemented variable compression

The mechanism for compression ratio changing (MCRC) was designed in the CFICE. The mechanism has to be studied theoretically to identify the engine rational parameters.

2.1. The construction of the mechanism for compression ratio changing

The MCRC is located in the engine piston.
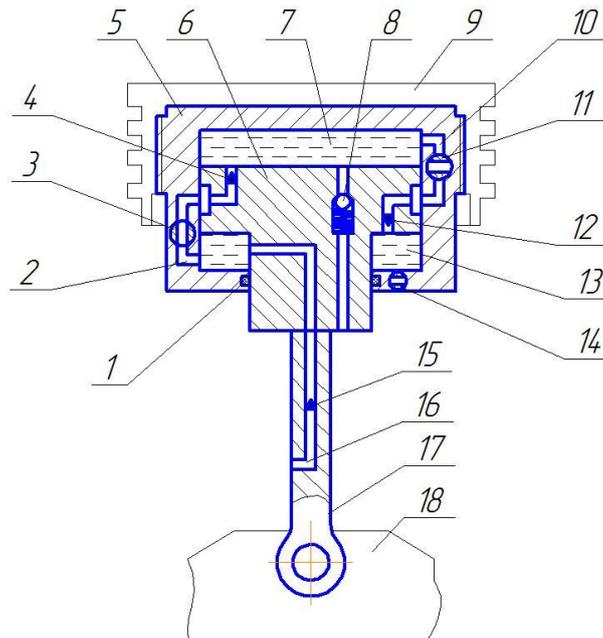The MCRC construction diagram is shown in Figure 2.

Fig. 2. The construction diagram of the MCRC: 1 – sealing; 2, 10, 16 – oil channel; 3, 11, 14 – hydraulic lock; 4, 12, 15 – check valve; 5 – moving body; 6 – piston in moving body; 7 – top oil cavity; 13 – bottom oil cavity; 8 – emergency valve; 9 – engine piston; 17 – piston rod; 18 – rocker arm

Heater was installed on the experimental model of the engine, which allowed investigating the operation of the mechanism at different oil temperatures (the oil temperature changed from 20 to 70 degree Celsius). This allowed us to provide realistic conditions for the MCRC operation.

A valve was installed in the oil supply line to the mechanism that performed the function of a check valve. The check valve prevents the creation of loads on the oil pump shaped like overpressure when the mechanism responds. Experiments were conducted on the mechanism with an open and closed crane. Oil was fed into the mechanism from the oil pump through a connecting pipe. The oil pressure in the bottom oil cavity was controlled by a manometer and maintained at 4.0 atm. In an automobile engine, an oil lubrication system pump creates such pressure. The turn of the moving body of the mechanism to activate the hydraulic lock was carried out using an electromagnetic relay. Due to this, the MCRC response time throughout the experiment remained unchanged.

Displacement of the moving body (Figure 2) occurs under the influence of applied gas forces ΔP_g and inertial forces $P_{j\Sigma}$, and frictional forces P_{fr} between the piston rings and the cylinder walls and the oil pressure P_{oil}.

The signal from the detonation sensor reaches the MCRC control mechanism when detonation combustion in the engine cylinder appears. This causes the turn of the hydraulic locks 3 and 11. The engine piston together with the mechanism moving body moves down under the force ΔP_g. The compression ratio decreases until the hydraulic locks return to their original position (position in Figure 3) and the cycle is repeated.
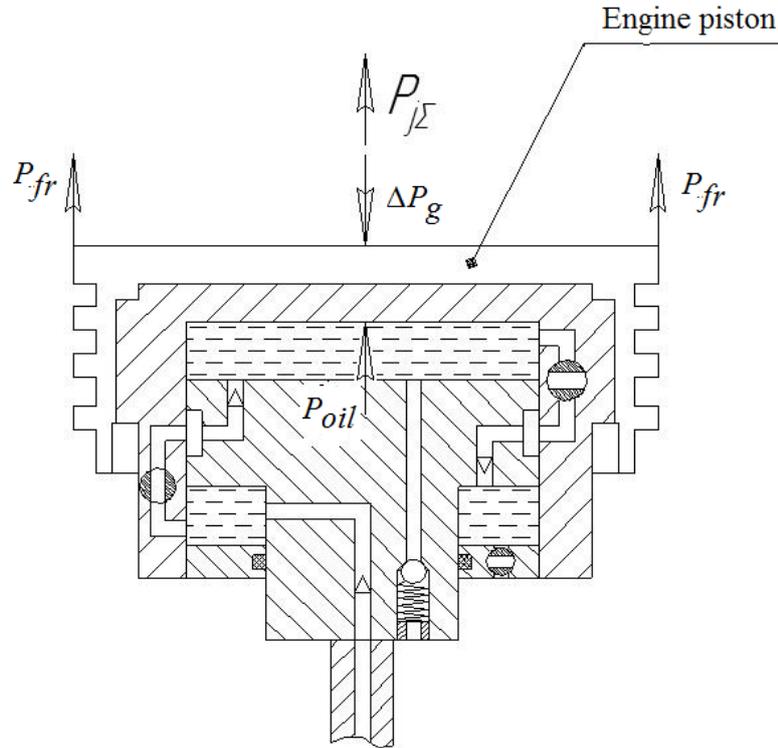

Fig. 3. The scheme of forces operating in the MCRC

The compression ratio change occurs in the engine successive cycles until force closure between the body and the movable thimble occurs with the help of the hydraulic lock upon the signal from the detonation sensor.

3. CALCULATION OF THE MCRC RESPONSE TIME

The MCRC response time t is determined according to the dependence:

$$t = \frac{\Delta S_c}{v_{mov}} \tag{1}$$

where ΔS_c is MCRC moving body relative movement or, as a consequence, the engine combustion chamber height change [m] and v_{mov} is MCRC moving body average speed [m/s].

The compression ratio current value is determined by the formula:

$$\varepsilon_x = \frac{V_h + V_c}{V_h} \tag{2}$$

where V_h is cylinder volume [m³] and V_c is combustion volume [m³].

The compression ratio rate change per second is determined with regard to the Equation (2):

$$v_{\alpha} = \frac{\dfrac{V_h + V_{c2}}{V_h} - \dfrac{V_h + V_{c1}}{V_h}}{t} = \frac{V_{c2} - V_{c1}}{t \cdot V_h} \qquad (3)$$

where V_{c1} is a combustion volume 1 [m^3] and V_{c2} is combustion volume 2 [m^3].

Taking into account the fact that MCRC moving body relative movement (Figure 4) is determined by:

$$\Delta S_c = \frac{V_{c2} - V_{c1}}{F_p} \qquad (4)$$
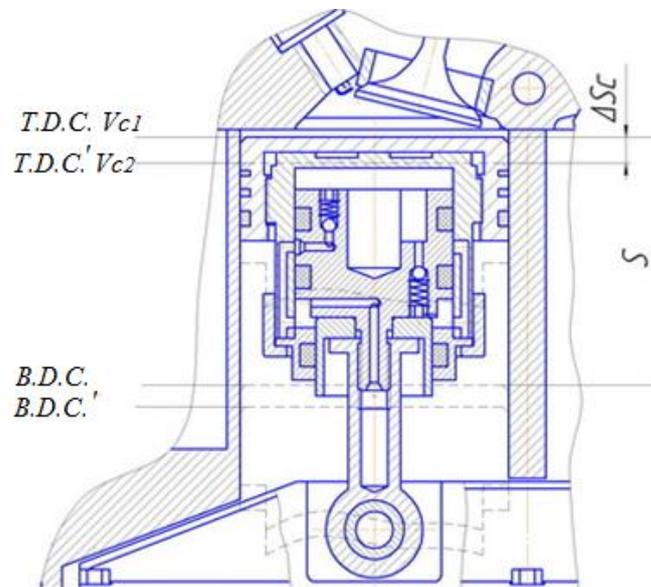
where F_p is an engine piston area [m^2].

Fig. 4. Engine diagram to determine the combustion chamber height change:
S is a piston stroke; ΔS_c is MCRC moving body relative movement; *TDP* is top dead point; *BDC* is bottom dead centre; *TDP′* is piston position at the top dead point under the compression ratio change; *BDC′* is piston position at the bottom dead centre under the compression ratio change

The compression ratio rate change is determined with regard to the Equation (1):

$$v_{\alpha} = \frac{v_{mov}}{S} \qquad (5)$$

where S is piston stroke [m].

The compression ratio rate change per cycle $v_{\alpha c}$ is a very important factor for MCRC. One cycle in a four-stroke engine is known to occur in crankshaft two revolutions. Therefore, compression ratio rate change is determined:

$$v_{\alpha c} = \frac{720 \cdot v_{\alpha}}{n} \qquad (6)$$

where *n* is Engine crankshaft speed, min^{-1}

Thus, to obtain the MCRC response time, it is necessary to determine the moving body movement speed v_{mov} by the amount ΔS_c. The following MCRC motion equations (Figure 5) have been worked out to solve this problem:

$$m_M \frac{dv_{mov}}{dt} = \Delta P_G + P_J + P_{fr.r} - P_{hydr} - P_{fr.s} \tag{7}$$

where m_M is the mechanism moving body mass, kg; x is the mechanism moving body movement, m; ΔP_G is force of gas pressure in the engine cylinder, H; P_J is total inertia force of the mechanism, H; P_{gr} is gravity force of the mechanism, H; $P_{fr.r}$ is friction force of piston rings, H; P_{hydr} is force, which describes hydraulic losses in the mechanism, H; $P_{fr.s}$ is frictional force in the seals of the mechanism, H.
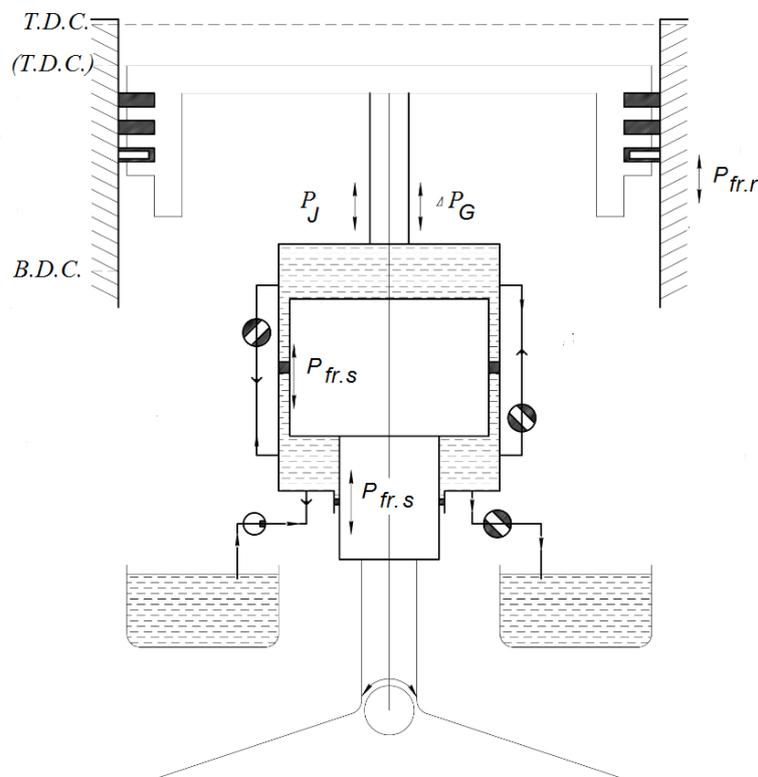

Fig. 5. Scheme for the MCRC calculation: P_J is total inertia force;
ΔP_G is force of gas pressure; $P_{fr.r}$ is friction force of piston rings;
$P_{fr.s}$ is frictional force in the seals of the mechanism

The mechanism moving body mass is determined:

$$m_M = m_p + m_{p.r} + m_{mov} + m_c + m_n + m_{t.b} \tag{8}$$

where m_p is the engine piston mass, kg; $m_{p.r}$ is piston rings mass, kg; m_{mov} is MCRC moving body mass, kg; m_c is cover mass, kg; m_n is nut mass, kg; $m_{t.b}$ is turning bushing mass, kg.

Gas pressure force is determined:

$$\Delta P_G = (p_G - p_0) \cdot F_p \tag{9}$$

where p_G is gas pressure in the engine cylinder at an arbitrary time interval, MPa; p_0 is atmospheric pressure, MPa.

Total inertial force $P_{J\Sigma}$ is determined:

$$P_{J\Sigma} = P_{J.mov} + P_{J.oil} \tag{10}$$

where $P_{J.mov}$ is inertial force from the MCRC moving masses, H; $P_{J.oil}$ is inertial force from oil in the MCRC, H.

The inertial force from the MCRC moving masses is determined:

$$P_{J.mov} = -m_M \cdot j = -m_M \cdot R \cdot \omega^2 \cdot (\cos\varphi + \lambda \cdot \cos(2 \cdot \varphi)) \tag{11}$$

The conrod-free engine design characteristics (the connecting rod coefficient λ, the crank radius R and the connecting rod length L) are the following:

$$\lambda = \frac{R}{L} = \infty \tag{12}$$

The inertial force from moving masses of the MCRC is determined:

$$P_{J.mov} = -m_M \cdot R \cdot \omega^2 \cdot \cos\varphi \tag{13}$$

where R is crank radius, m; ω is the crankshaft angular velocity, s^{-1}

The inertial force from oil in the MCRC is determined:

$$P_{J.oil} = -(m_{oil.b} + m_{oil.t}) \cdot R \cdot \omega^2 \cdot \cos\varphi \tag{14}$$

where $m_{oil.b}$ is oil mass in the mechanism bottom cavity, kg; $m_{oil.t}$ is oil mass in the mechanism top cavity, kg.

The piston ring set frictional force is determined:

$$P_{fr.r} = \left[-sign(v) \cdot c_1 \cdot \sqrt{|v|} \right] \cdot \left[1 - c_2 \cdot \frac{t_w - t_r}{t_r} \right] \cdot \left[1 + c_3 \cdot \frac{p - p_0 + \sigma - \sigma_r}{p_0} \right] \cdot \left[\frac{d}{d_r} \right]^2 \tag{15}$$

where p_a is pressure at the end of inlet, MPa; c is the constant coefficient; v is piston speed, m/s; t_r is reduced temperature, °C ($t_r = 40$°C [28]); t_w is cylinder wall temperature, °C; σ is the ring radial stress, MPa; σ_r is the ring reduced radial stress, MPa ($\sigma_r = 0.1$MPa [29]); d_r is reduced cylinder diameter, mm ($d_r = 165$mm [29]).

The hydraulic losses force P_{hydr} takes into account the loss of oil flow through the channels P_{ch}, back-pressure valves P_{b-p} and hydraulic lock P_{h-l} that occur when the oil moves in the mechanism body. The force is formed as:

$$P_{hydr} = \rho \cdot g \cdot a_i \cdot Q_i^2 \cdot F_{p.mov} \tag{16}$$

where ρ is oil density, kg/m^3; g is free fall acceleration, m/s^2; a_i is the direct or backward channel hydraulic resistance, s^2/m^5; Q_i is oil consumption under the direct or backward mechanism body motion, m^3/s; $F_{p.mov}$ is active area of piston in moving body, m^2

The rubber seals frictional force under the mechanism moving body motion is determined by the Equation (9):

$$P_{fr.s} = \mu \cdot sign(v_{mov}) \cdot (|p_1 - p_2| \cdot F_{r1} + p_2 \cdot F_{r2}) \tag{17}$$

where μ is friction coefficient of the pair of steel-rubber in oil, $\mu = 0.4$ [30]; p_1 is pressure in the MCRC top oil cavity, MPa; p_2 is pressure in the MCRC bottom oil cavity, MPa; F_{r1} & F_{r2} is gasket ring area, m^2

Total calculated force is formed as:

$$P_\Sigma = \Delta P_G + P_J + P_t - P_{fr.r} - P_{hydr} - P_{fr.s}$$ (18)

4. THE DESIGN-THEORETICAL RESEARCH RESULTS

The MCRC operation speed was determined to be affected by the following parameters:
- pressure on the mechanism on the side of the springing attachment
- oil temperature
- channels flow passage area δ
- the MCRC moving body rotation angle when the hydraulic lock is turned on
- back pressure in the mechanism bottom oil cavity

The dependence of the MCRC response time on the relative area of flow passage of channels δ of the hydraulic lock, pressure p and oil temperature t were obtained experimentally (Figure 6).

The empirical dependence of the response time of the MCRC is determined:

$$\tau(\delta, p, t) = \tau_A(p, t) \cdot \left[\frac{\delta_A}{\delta} \right]^{m(\delta, p, t)}$$ (19)
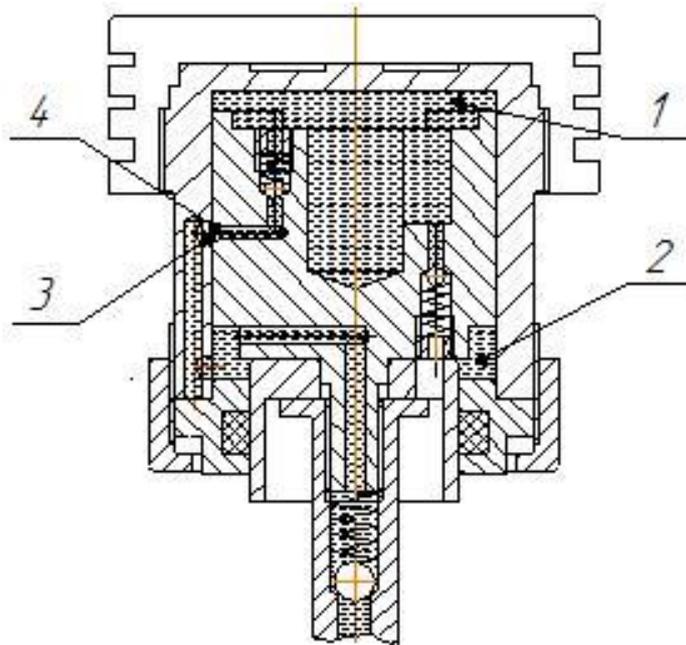

Fig. 6. The MCRC channels scheme for increasing the compression ratio:
1 – top cavity; 2 – bottom cavity; 3 and 4 – channels

The MCRC response time τ calculated dependencies on values (δ, p, t) were built in Figures 7 and 8.
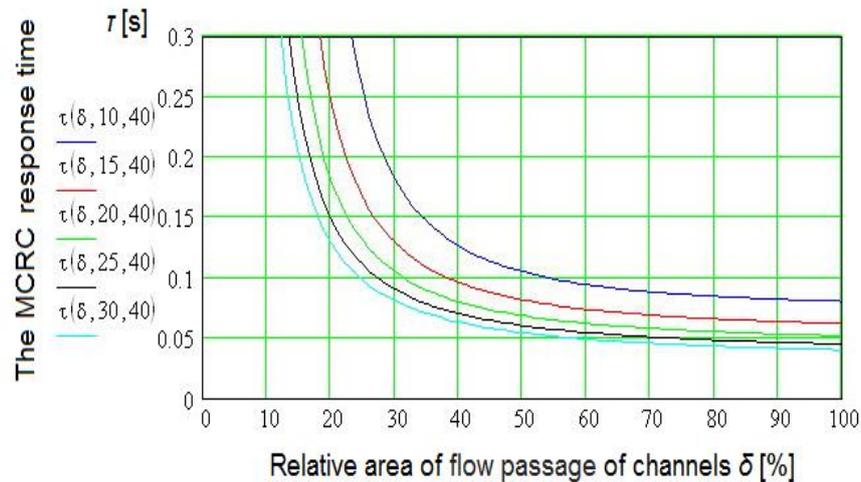
Fig. 7. Dependence of the response time τ [s] of MCRC on the relative area of flow passage of channels δ [%] of the hydraulic lock at different pressures p [MPa]: $t = 40°$C

The mechanism operating conditions investigation shows that the MCRC response time τ decreases under the increasing values (δ, p, t). The response time τ decreases sharply over the range from 0 to 40% (value δ) and pressure p over the range 0 to 30 MPa (Figures 7 and 8). Further increase of values δ and p has little effect on the response time τ. Therefore, when the MCRC is running in the engine, it is not necessary to turn on the hydraulic lock on 100% when the compression ratio is reduced. This enabled reduction of the MCRC response time τ and the compression ratio. The hydraulic locks are to be 100% turned on with the compression ratio increase in order to increase the compression ratio rapidly. Consequently, the engine operated less under nonoptimal conditions. The compression ratio increase is the result of the inertia forces action from the body and oil pressure in the mechanism cavities and the compression ratio change. Therefore, the mechanism response time τ for the compression ratio rise will be less than that for the value ε decrease.
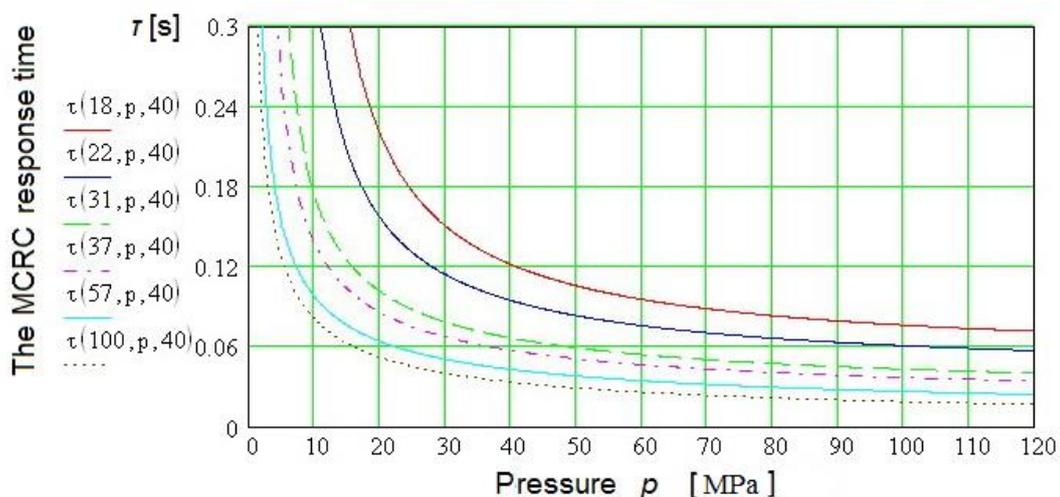

Fig. 8. The MCRC response time τ [s] dependence on pressures p [MPa] in the oil cavity at different relative area of flow passage of channels δ [%] of the hydraulic lock: $t = 40°$C

Depending on oil temperature t (Figures 9 and 10) has a different character. The curves slope indicates significant effect of the temperature t in the whole range of its values. The relation between the value τ and value t varies greatly depending on the MCRC operating conditions (values δ and p change). The response time τ increases under low temperatures (0...40°C) and especially at the hydraulic lock small flow passage. Nevertheless, this occurs during engine warm-up.
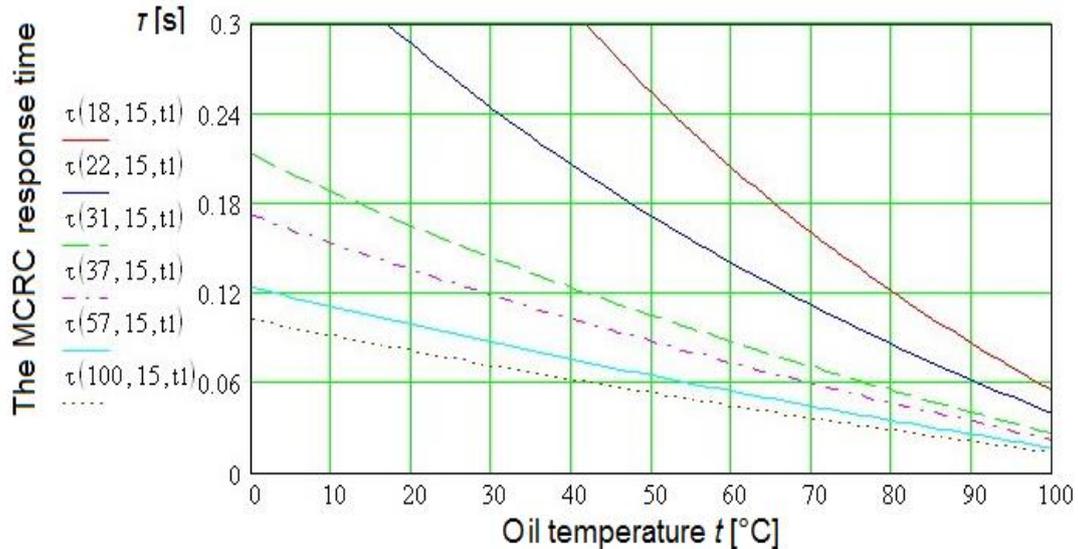

Fig. 9. Dependence of the response time τ [s] on oil temperature t [°C] in the oil cavity 1 (Figure 8) at different relative area of flow passage of channels δ [%] of the hydraulic lock: p = 15 MPa
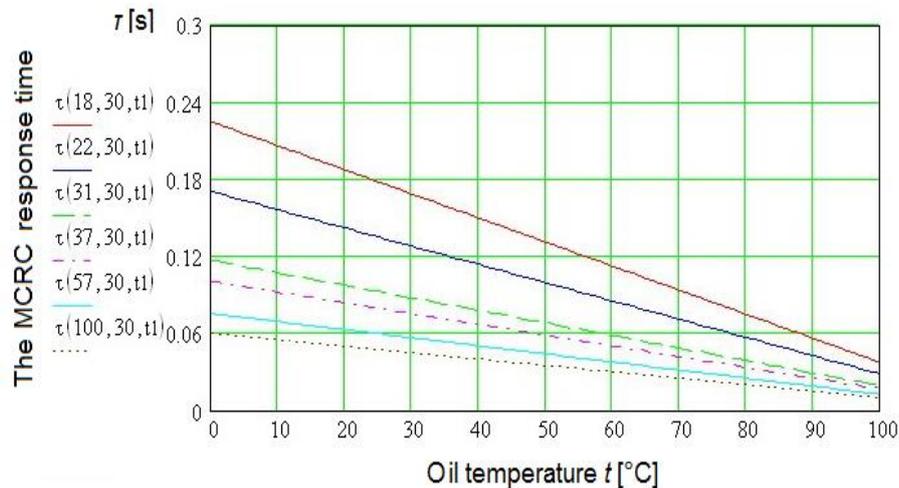

Fig. 10. Dependence of the response time τ [s] on oil temperature t [°C] in the oil cavity 1 (Figure 8) at different relative area of flow passage of channels δ [%] of the hydraulic lock: p = 30 MPa

The MCRC operation study results indicate strong influence of the values δ and p on the response time τ in the field of their small values. This indicates that the MCRC significant response rate is ensured under insignificant changes in hydraulic lock flow passage area and little oil pressure in cavities.

Empirical dependence of the oil temperature t (see Figures 9 and 10) on the response time τ is determined:

$$\tau = f(\delta, p, t) \tag{20}$$

A graphical method was used to determine the Equation (20). The empirical dependence of the response time of the MCRC is determined by the Equation (19).

The oil pressure in the bottom oil cavity was controlled by a pressure gauge and maintained at 4 atm. In an ICE, the pressure can be generated by an oil pump of a lubrication system. The oil temperature was controlled by an electrical thermometer.

In the experiment, the MCRC response time was tested at different section of canals and at different viscosity engine oil. Full operation time is influenced by section of canals, channel form and viscosity oil. The larger-section channels and the lower the viscosity oil the 0.2 s of full operation time is enough.

The results demonstrate the possibility of compression ratio change in the engine with the MCRC.

5. CONCLUSIONS

Theoretical studies were conducted. These studies were aimed at improving the MCRC for a gasoline four-stroke engine.

The developed mathematical model of the MCRC calculation revealed the influence of various factors of the engine with the CRCM on the MCRC operation (for example, the compression ratio rate change). The experimental data corroborates with the compiled mathematical model operation.

Calculation studies show that the mechanism full operation takes place quickly (0.02 s per unit ε_x), which indicates the expediency of using such a high-pressure pump in a four-stroke gasoline engine with CRCM.

The mechanism movable body complete movement ($S = 4$ mm) at the oil temperature in the hydraulic system of 45°C and pressure on the body of $p = 60$ bar are stated to takes place 0.2 s. The MCRC response time is average time 0.23 s. An important parameter for the MCRC response time is the response time until the appearance of detonation combustion with increasing motor load.

Heat losses in the CFICE are less than in a classical engine since the piston of the CFICE is shorter than in a classical engine, it is cut to piston rings.

Currently, the patent application of the MCRC is being considered in the patent office for the grant of a patent for the invention, hence, the description of the mechanism is not in full detail.

The mathematical model of the MCRC and the design-theoretical research results can be used to calculate engines of the same type in order to obtain optimal parameters for further design or improvement of engines of similar structures.

CFICE can be used as an internal combustion engine for motor vehicles and other mobile and fixed users as well as a pump or compressor for creating the working fluid and transport excess pressure (pumping) of gases, liquids, suspensions, mixtures, suspensions and other agents in various industries and farms.

Acknowledgements

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