ISSN 1052-6188, Journal of Machinery Manufacture and Reliability, 2023, Vol. 52, No. 3, pp. 271–280. © Allerton Press, Inc., 2023.
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Fundamentals and Some Results of Numerical Modeling the Cylinder-Piston Group Jet-Oil Supply in a High-speed Four-Stroke Internal Combustion Engine

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Received November 9, 2022; revised February 8, 2023; accepted February 20, 2023

Abstract—Scientific substantiation of the design and simulation techniques for one of the understudied aspects of oil supply, namely, jet supply of engine oil from the gaps of the rotating journal bearing to the "cylinder–piston" pairing of a high-speed four-stroke universal diesel engine, is given. The simulation resulted in visualization of the trajectory and zones of oil ingress on the friction surfaces of parts of the cylinder–piston group presented in this article, as well as a quantitative assessment of the oil volume in these zones and in the crankcase. The simulation technique and the results of its application can be of both scientific and practical interest for specialists in the field of lubrication design and reliability of piston machines.

Keywords: friction, lubrication, oil supply, journal bearing, diesel engine, numerical simulation

DOI: 10.3103/S1052618823030159

Uninterrupted and rationally organized oil supply, considered as a set of methods and technical solutions that ensure the supply of lubricant to friction and wear zones, is an essential condition for energy-efficient and reliable operation of machines, the principle of which is not based on friction. The information mentioned above is true in large measure for internal combustion engine (ICE), the most energy-intensive technical unit of which is the cylinder–piston group (CPG) in general and the cylinder–piston mating, in particular. It is known that friction losses in the CPG make the greatest contribution to the total mechanical losses of high-speed piston engines [1, 2]. The reason for the quantitative predominance of mechanical losses in this group of parts is the imperfection of the oil supply of the latter, which is carried out to a lesser extent due to oil mist and to a greater extent by oil ejection from the gaps of the rotating journal bearing [3, 4]. The imperfection of oil supply by means of splashing is associated with the specifics of the design and kinematics of the traditional crank mechanism. Even the visually predictable flight path of a single oil jet under the action of oil pressure in the connecting rod bearing gap and the centrifugal inertial forces during crank rotation shows that oil jets should mainly fall on one of the two sides of the inner surface of the cylinder opposite in the swing plane of the connecting rod (the so-called unloaded side) of the cylinder, leaving the other (loaded) side in a state of lubrication deficiency. (In the text and figures below, the loaded and unloaded sides of the cylinder wall are designated as L and UL, respectively.) This oil supply anomaly was detected by direct measurements of the amount of oil removed by the piston rings from the cylinder walls in the engine cranking mode from an external source, i.e., under conditions of equal load on the left and right sides of the cylinder wall in the swing plane of the connecting rod [5]. Indirect evidence of such an imperfection in the oil supply of CPG parts, which manifests itself in asymmetry, or, more precisely, in the discrepancy between the amount of oil supplied to the cylinder and the nature of loading of the cylinder sides by the lateral force of the piston, is often observed in the operation of piston engines in the form of a difference in the intensity of wear of the cylinder's opposite sides and the piston skirt in the swing plane of the connecting rod [6–8].

Cases of damage from wear and scuffing of the piston skirts, the dry friction surface of the cylinders, the appearance of discoloration on the piston pin, found after disassembly and inspection of failed CPG parts [9], indicate an unresolved problem of oil starvation and the need for closer study of the oil supply process. After revealing the limitations of experimental methods for visualizing the oil supply to CPG parts under conditions of high rotation speed [5], the choice was made in favor of mathematical modeling of the process under consideration.

Of all sources (or methods) of supplying engine oil to the CPG friction surfaces in high-speed four-stroke piston engines [2], this article discusses only the last of the six sources of oil supply to the CPG:

(A) condensation of oil droplets from oil mist;

(B) oil spraying with a bucket located on the connecting rod bearing cap;

(C) inertial oil injection from the inner surface of the piston bottom, when the latter stops at the bottom dead center (BDC);

(D) oil jet supply from a special channel in the connecting rod body;

(E) oil splashing from the gaps of the fixed main bearing by the rotating surfaces of the crank checks and counterweights;

(F) oil ejection from the gaps of a rotating connecting rod bearing. This choice is due, firstly, to the guaranteed availability of this source of oil supply in any design of a piston engine with a traditional crank mechanism, which cannot be said, for example, about sources 2–4. Secondly, this choice is justified by the largest (especially in comparison with the first source) contribution of the ejection of oil jets from the gap of the rotating connecting rod bearing to the oil supply to the main rubbing CPG surfaces, which directly follows from a simple analysis of the geometric relationships in the crank mechanism, indicating the following conditions:

(A) The maximum proximity of the upper semicircle of the orbit of the central axis of the crank neck to the friction surfaces of the cylinder and piston.

(B) The intersection of the trajectories of oil jets in the swing plane of the connecting rod on the inner surface of the cylinder, which perceives the lateral piston force, which is not obvious in the case of source 5.

As a result of experiments on a breadboard setup, created on the basis of a small-sized high-speed diesel engine with an optically transparent cylinder, which makes it possible to observe visually the formation of a jet in the gaps and special oil channels of the lubrication system, it was found that a single oil jet is ejected from the point of the maximum gap that occurs as a result of displacement of the shaft neck under the action of an applied external load.

The problem of calculating the coordinate of this point is posed and solved in this article in the first approximation based on application of the provisions of solid mechanics and, in the second approximation, by refining the obtained solution using the provisions of the hydrodynamic theory of lubrication. The array of coordinates of the ejection point of the oil jet found in this way for the working cycle of a four-stroke piston engine made it possible, based on the use of the developed mathematical models, to calculate not only the trajectory, but also the amount of oil entering the characteristic CPG friction zones with the jet, including zones that urgently need lubrication to minimize friction and wear and to reduce the risk of scuffing. The results of numerical simulation of the oil jet supply obtained for the input data of a single-cylinder high-speed four-stroke diesel engine confirmed the experimentally observed anomaly of oil supply to the cylinder, which consists in the fact that significantly less oil enters the loaded inner side (half-surface) of the cylinder than the opposite unloaded side. This study made it possible for the first time to evaluate by means of calculation the quantitative ratios of the oil volumes entering the main CPG friction and wear zones. The use of the developed computer program for simulating the process of oil jet supply to CPG parts makes it possible to obtain information on the influence of a large number of factors on this process, including oil viscosity and temperature, speed and load modes of the engine, and the design of its CPG and crank mechanism, which opens up additional opportunities for improving fuel efficiency and engine reliability.

The purpose of this study was to determine the amount (volume) of oil that got into the main CPG friction and wear zones during the working cycle due to the dominant source of oil supply for this group of parts, that is, the oil ejection from the gaps of the rotating connecting rod bearing.

There are the following tasks to be solved to achieve this goal:

(A) Formation and solution of equations of solid mechanics to determine the angular coordinates of the ejection point of the oil jet from the gap of the rotating connecting rod bearing;

(B) Development of a mathematical model describing the trajectory of the oil jet from the ejection point to the lubrication point of the friction surface;

(C) Creation of a computer program implementing the mathematical model according to section 2 and, with its help, simulation of the supply of the oil jet to the CPG of a high-speed internal combustion engine.

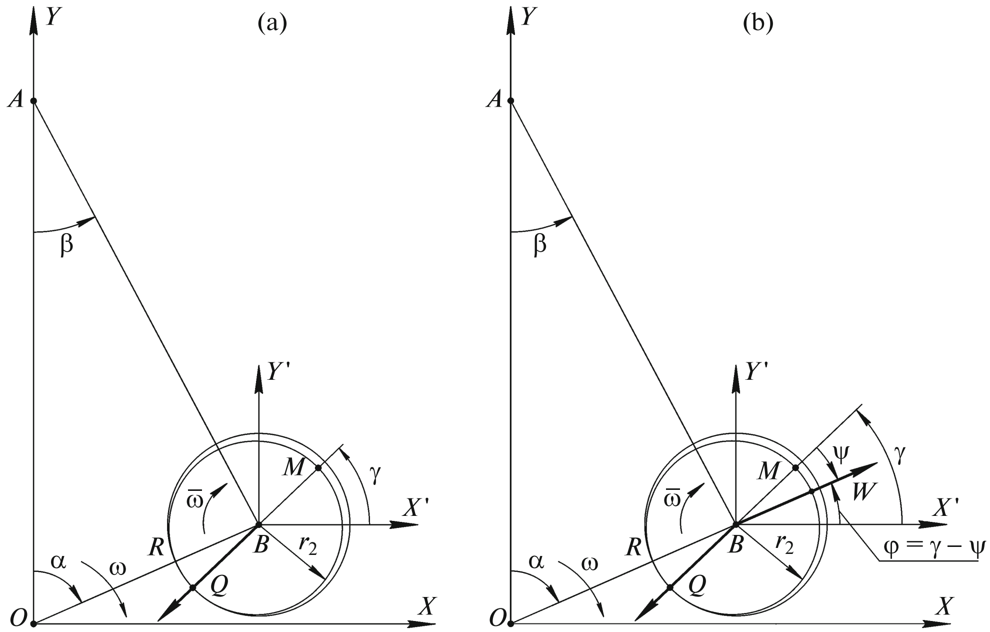

Fig. 1. Angular coordinate of the ejection point of the oil jet from the gap of the rotating connecting rod bearing: (a) without regard for the hydrodynamics (angle γ); (b) with regard for the hydrodynamics (angle φ).

PROBLEM SOLVING METHODS

The solution of problem 1 was divided into two stages (or two approximations):

Stage 1. Obtaining a mathematical expression for the angular coordinate of the ejection point of the oil jet from the gap of the rotating connecting rod bearing without regard to the influence of hydrodynamic forces on the position of the shaft relative to the support in the connecting rod bearing.

Stage 2. Correction of the solution obtained in Stage 1, based on taking into account the influence of the hydrodynamic reaction of the oil layer in the gap on the position of the shaft in the connecting rod bearing.

Stage 1. The solution was obtained as a result of applying the provisions of the mechanics of a rigid body to the design scheme of the traditional crank mechanism (Fig. 1)

$$\gamma = \arctan\frac{Q^y}{Q^x}, \tag{1}$$

where γ is the angular coordinate of the ejection point of the oil jet from the gap of the rotating connecting rod bearing; Q^x and Q^y are the projections of the force Q acting on the connecting rod neck onto the corresponding axes of the XOY coordinate system.

Stage 2. At this stage, the values of the angular coordinate (1) were refined by taking into account the influence of the hydrodynamic reaction W applied to the crank pin from the side of the oil layer in the gap of the connecting rod bearing and balancing the external load Q (Fig. 1). The effect of the hydrodynamic reaction W is that it causes an additional angular displacement of the point of the maximum gap in the direction of shaft rotation by angle ψ, which is called the load angle (Fig. 1). As a result of this angular displacement, the exit point of the oil jet passes from point M to point N, which is characterized by the angular coordinate $\varphi = \gamma - \psi$.

The estimation of the load angle ψ was obtained based on the use of ready-made expressions for the tangent of this angle, which was presented in the chapter "Journal bearings" of work [10] separately for the Reynolds solution (infinitely long bearing model) and the Michell solution (infinitely short bearing model) in expressions (2) and (3), respectively,

$$\tan\psi = \frac{2\left(1-\varepsilon^2\right)^{\frac{1}{2}}\left[\sin\xi - (\pi+\xi)\cos\xi\right]}{\varepsilon\left(1+\cos\xi\right)^2}; \tag{2}$$

$$\tan\psi = \frac{\pi}{4}\frac{\left(1-\varepsilon^2\right)^{\frac{1}{2}}}{\varepsilon}, \tag{3}$$

where ε is the relative eccentricity of the journal bearing and ξ is the dimensionless parameter, which takes the values from 0.887 to 1.352, according to [10].

Numerical expressions for the values ε and ξ were found from the tabular arrays of these parameters presented in [10] based on the known values of the external load Q determined in the course of dynamic calculation, as well as solving the equations of hydrodynamic equilibrium of the crank pin on the oil layer in the gap of the connecting rod bearing according to the first and second models, in expressions (4) and (5), respectively,

$$Q = \frac{6\mu v r^2 l}{c^2}W^*; \tag{4}$$

$$Q = \frac{\mu v l^3}{c^2}W^*, \tag{5}$$

where μ is the dynamic viscosity of the engine oil; v is the linear speed of rotation of the crank pin; l and r are the axial length and radius of the crank pin, respectively; c is the diametrical clearance of the "crank pin—inserts" mating; and W^* is the relative (dimensionless) function of the hydrodynamic reaction of the oil layer in the gap of the connecting rod bearing, determined according to [10] for infinitely long and narrow bearing models by dependencies (6) and (7), respectively,

$$W^* = \frac{\varepsilon(1+\cos\xi)^2}{2\left(1-\varepsilon^2\right)(1+\varepsilon\cos\xi)}\sqrt{1+\tan^2\psi}; \tag{6}$$

$$W^* = \frac{\pi}{4}\frac{\varepsilon}{\left(1-\varepsilon^2\right)^2}\left[\left(\frac{16}{\pi}-1\right)\varepsilon^2+1\right]^{\frac{1}{2}}. \tag{7}$$

The linear speed of rotation of the crank pin relative to the connecting rod big end v, which is included in the numerator of the right parts of equilibrium equations (4) and (5), differs from that for the crank and depends on the angular speed of rotation of the engine crankshaft ω, the radius of the crank pin r, the crankshaft rotation angle (CRA) α and crank mechanism constant λ, equal to the ratio of the radius of the crank R to the length of the connecting rod L (Fig. 1), determined from the formula

$$v = \omega\left(1+\frac{\lambda\cos\alpha}{\sqrt{1-\lambda^2\sin^2\alpha}}\right)r.$$

The solution to problem 2 was based on the calculation scheme (Fig. 2), which makes it possible to determine the velocity vector v_o of the oil jet ejected from the gap of the rotating connecting rod bearing, using the expressions for the components (vectors) of the velocity v_p and v_c, determined, respectively, by the oil pressure and the rotation of the crank pin

$$\vec{v}_o = \vec{v}_p + \vec{v}_c; \quad \vec{v}_p = \sqrt{\frac{2p}{\rho}}; \quad \vec{v}_c = \frac{\overline{M_{i-1} - M_i}}{\Delta T},$$

where p is the oil pressure generated by the oil pump; ρ is the oil density; $M_{i-1} - M_1$ are the shortest distance between the previous and current position of the point M at the ith step of the calculation by the CRA α; ΔT is the time of moving point M from position M_{i-1} to position M_i.

To solve problem 3, the engine oil jet ejected from the point of the maximum gap of the rotating connecting rod bearing M was represented by a set of mathematical points that make the complex movement in a plane tangent to the end surface of the connecting rod big end and parallel to its swing plane (Fig. 3).

Within the framework of this model, the jet was divided into conditional portions in accordance with the time step of calculation (CRA angle α). Each mathematical point in the set was an elementary part of

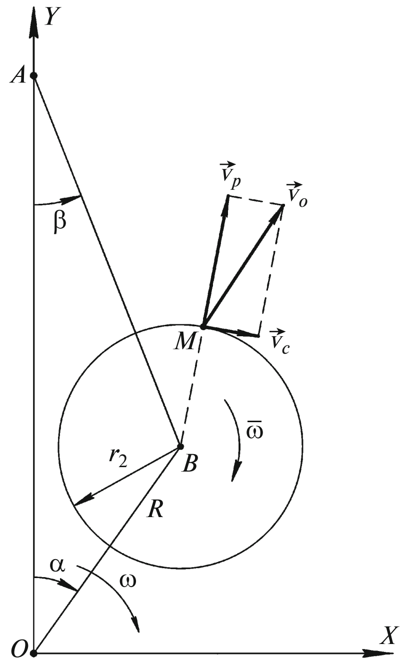

Fig. 2. Calculation scheme for determining the oil jet velocity.

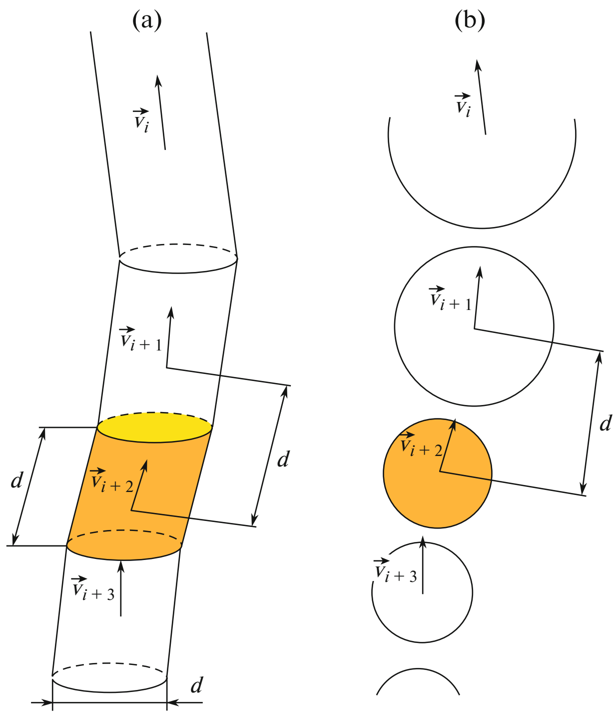

Fig. 3. Mathematical points set model for evaluating the (a) quantitative and (b) visual characteristics of the oil jet.

the jet, commensurate in size with the diameter of the hole d, inscribed in the maximum gap of the connecting rod bearing (Fig. 3).

The volume of an elementary portion of the oil formed over the period of time ΔT was determined as

$$V_e = Sv_o\Delta T,$$

where S is the cross-sectional area through which the oil jet flows.

Table 1. Specifications (input data) for the calculation of the supply of the oil jet to the CPG of the diesel engine 1Ch 8.5/8.0

Name	Value
Crankshaft speed, min^{-1}	3000
Oil pressure, bar	4
Oil temperature, °C	80
Oil density at 20°C, $\mathrm{kg/m}^3$	890
Oil kinematic viscosity at 100°C, cSt	10
Weight of piston assembly, kg	0.813
Weight of connecting rod assembly, kg	0.987
Crank radius, mm	40
Connecting rod length, mm	136
Piston diameter, mm	85
Piston stroke, mm	80
Cylinder length, mm	153
Crank pin diameter, mm	50
Crank pin length, mm	21
Diameter gap in connecting rod bearing, mm	0.025

The total volume of oil in the jet V was determined by multiplying the volume of an elementary portion of oil by the number of portions in the jet. Since the volume of an elementary portion of oil could be different at each calculation step, for clarity and additional information, it was decided to visualize the oil jet in the form of circles of different diameters, the size of which is proportional to the volume of the portion of oil (Fig. 3).

The friction surfaces of the CPG and crank mechanism, which could potentially be affected by an oil jet, were divided into eight characteristic zones: 1, the loaded side of the cylinder (L); 2, the unloaded side of the cylinder (UL); 3, the inner surface of the piston skirt on the loaded side of the cylinder; 4, the inner surface of the piston skirt on the unloaded side of the cylinder; 5, the inner surface of the piston head; 6, the piston pin; 7, crankcase; 8, internal space of the crankcase. The volume of oil that got into each of these zones was determined separately for each stroke and in general for the full working cycle of a four-stroke internal combustion engine.

The described positions and calculation models formed the mathematical basis of the computer program CSJet (Crank Shaft Jet) developed and debugged in the course of this study, which provides a numerical simulation of the oil jet supply from the gaps of the rotating connecting rod bearing to the CPG parts.

SIMULATION RESULTS: DISCUSSION

The simulation task was to determine the amount (volume) of engine oil that fell into eight characteristic friction, wear, and cooling zones of the CPG due to the ejection of oil jets from the gaps of the rotating connecting rod bearing. The object of the study was a high-speed four-stroke universal diesel engine 1Ch 8.5/8.0, the main technical characteristics and indicators of which are given in Table 1.

As a design mode, the operation of a diesel engine at constant power and speed (7 kW and 3000 min^{-1}, respectively) was chosen when the latter was used as a power unit of the AC generator.

The simulation results showed that the vast majority (up to 86%) of the total volume of oil ejected during the working cycle from the gaps of the rotating connecting rod bearing (Fig. 4) falls on the inner surface of the engine crankcase walls and, thus, does not participate directly in the lubricating parts.

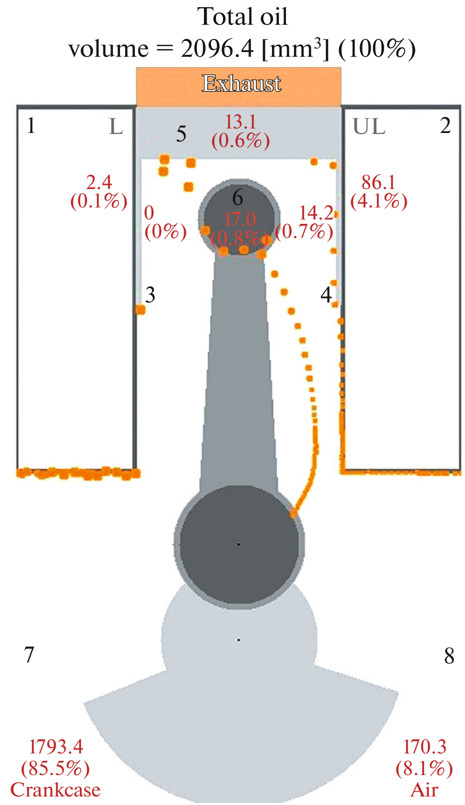

Fig. 4. Distribution of the oil volume ejected from the gaps of the rotating connecting rod bearing during the working cycle of the 1Ch 8.5/8.0 diesel engine over characteristic zones 1–8 of the CPG and crank mechanism.

The indirect participation of this oil volume, as well as the oil volume that did not have time to reach the CPG and crank mechanism surfaces (rounded 8%), can manifest itself in the lubrication of moving mates only in the formation of oil mist after the fact of oil jets hitting the fixed surface of a part. A very small proportion (about 5%) of the total volume of oil jets emitted from the rotating connecting rod bearing enters directly into the zones of the rubbing GPG mates "piston skirt–cylinder" and "piston pin–connecting rod bush." A little more than 1% of the oil reaches the inner surfaces of the walls of the piston head and skirt, participating in its cooling.

The results of the calculation in the "stroke by stroke" mode showed that the oil ejected from the gaps of the rotating connecting rod bearing enters the zones of the CPG parts only during two strokes of the working cycle, namely, on the exhaust stroke and the intake stroke, when the angular sector of the crank pin circle, covering the points of oil jet ejection from the connecting rod bearing, is directed to the unloaded side (UN) of the cylinder wall in the direction of crank rotation (Fig. 5, Table 2).

Comparison of the amount of lubricant that during the working cycle got on the loaded (L) and unloaded (UL) sides of the cylinder wall (Fig. 4) with the results of the estimates of the minimum engine oil volume necessary to prevent oil starvation in this lubrication zone for this diesel engine showed that the volume of lubricant on the loaded side of the cylinder wall (2.4 mm^3) is clearly not enough to provide a favorable hydrodynamic lubrication regime in the "cylinder–piston skirt" mating.

Taking into account the relevance of the trend associated with the transition to the use of low-viscosity motor oils in modern piston and hybrid engines [11], it was of particular interest to obtain information on the effect of the high-temperature (at 100°C) kinematic viscosity of oil ν_{100} on the supply of the oil jet of the CPG parts. The influence of this viscosity on the parameters of supply of the oil jet of the diesel engine, the object of this study, was evaluated in a wide range of its value variation of 2–14 cSt. The control output values were taken as follows:

(A) The absolute (in mm^3) total amount of oil ejected with the jet during the operating cycle is q_0.

(B) The amount of oil that got into the CPG friction zones during this time (i.e., into zones 1–6) is q_{1-6}.

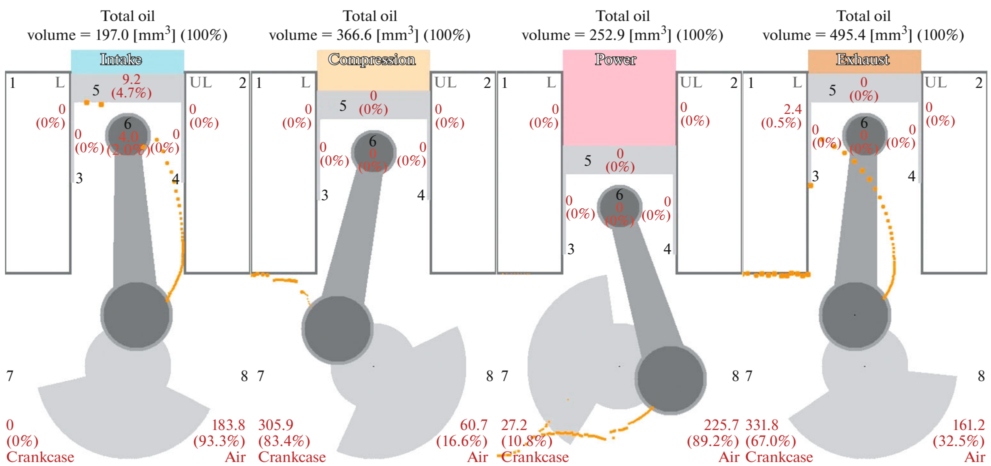

Fig. 5. Visualization of the oil-jet supply process from the gaps of the connecting rod bearing to the parts of the CPG during (by strokes) the working cycle of the diesel 1H 8.5/8.0.

As the engine oil viscosity decreases linearly, the total amount of oil ejected with the jet and the amount of oil supplied by the jet to the CRG parts increase nonlinearly (Fig. 6). The analysis showed that such a change in q_0 and q_{1-6} is explained by the influence of the oil viscosity, as a measure of the internal friction of the fluid layers, on the jet velocity component v_c due to rotation of the crank pin: with a decrease in friction inside the oil, the average value of the oil jet velocity component increases.

CONCLUSIONS

The numerical simulation of oil supply of CPG parts, carried out by ejection of an oil jet from the gaps of the rotating connecting rod bearing of the high-speed four-stroke diesel engine, showed the following during its operating cycle:

(A) CPG parts receive only about 15% of the total volume of oil supplied with the jet (100%) on the friction surface.

Table 2. Result of calculation of oil volumes in zones 1–8 of the CPG and crank mechanism at the end of each of the four-stroke of the working cycle of the diesel engine 1Ch 8.5/8.0

Stroke number and name	Total per stroke	Absolute mm^3 and relative (%) amount of oil that got into the CPG zone of and crank mechanism with an oil jet							
		zone							
		1	2	3	4	5	6	7	8
(1) Intake	668.6 (100)	0.0 (0.0)	86.1 (12.9)	0.0 (0.0)	14.2 (2.1)	13.1 (2.0)	4.0 (0.6)	463.0 (69.3)	88.1 (13.2)
(2) Compression	483.6 (100)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	385.0 (79.6)	98.6 (20.4)
(3) Power	426.8 (100)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	335.3 (78.6)	91.4 (21.4)
(4) Exhaust	517.4 (100)	2.4 (0.5)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	0.0 (0.0)	13.0 (2.5)	331.8 (64.1)	170.3 (32.9)

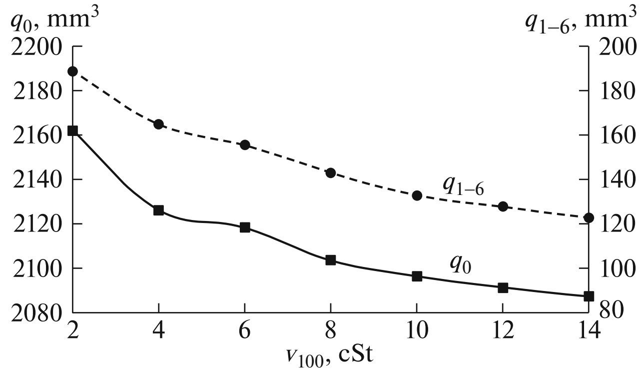

Fig. 6. The effect of changes of high-temperature viscosity variation of the engine oil ν_{100} on the total amount of oil q_0 ejected by a jet during the diesel operating cycle, and the amount of oil q_{1-6} supplied by a jet at the CPG friction surface during the same time.

(B) There is an anomaly in the oil supply of CPG parts, in which significantly less lubricant is supplied to the loaded side of the cylinder wall than to the unloaded one: in particular, 0.1% versus 4.1% of the total amount of oil ejected by the jet, respectively.

(C) Oil is supplied to the friction and cooling zones of the CPG parts only during two strokes of the working cycle, when the jet ejection point is oriented during rotation of the crankshaft to the unloaded side of the cylinder wall.

This study showed that a decrease in the viscosity of engine oil leads to an increase in the amount of lubricant supplied to the friction zones of CPG parts, which can be considered as one of the aspects of justifying the feasibility of switching from medium-viscosity to low- and ultra-low viscosity engine oils for modern high-speed piston and hybrid engines.

The logical continuation of research in this area may be the development and verification of the effectiveness of technical solutions aimed at eliminating anomalies in the oil supply of CPG parts. In addition, it is of practical interest to develop a method for quantitative determination of the extremely low viscosity of engine oil, which is able to reduce friction losses while maintaining the reliability of the piston engine.

ACKNOWLEDGMENTS

The authors are grateful to A.F. Biktashev, Master of Engineering and Technology, who made the invaluable contribution to the development of the algorithm and the interface of the CSJet program.

CONFLICT OF INTEREST

The authors declare that they have no conflicts of interest.

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Translated by A. Kolemesin