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Research of the efficiency using the model of the spatial hinge in an internal combustion engine

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Abstract. An internal combustion engine is a heat system for many kinds of use in the field of engineering. The difficult economic situation around petroleum products, which is associate with their deficit and modern environmental safety standards require switching to alternative energy sources and looking for ways to modernize internal combustion engines to increase their efficiency. A large number of internal combustion engines belong to the crank group. The analysis of the crank group showed their low technical efficiency. When the piston is in the upper position at the time of ignition of the combustible mixture in the cylinder, the maximum pressure occurs. This leads to maximum driving force with a minimum crank angle. Also increases the friction of the piston on the cylinder wall, the wear of the cylinder wall, additional local heating of the surface of the cylinder and over-load of the crankshaft. Thus, a decrease in the net power resulting from the combustion of fuel in the engine occurs. To increase the efficiency of the crank mechanism, it is propose to apply a new design of the torque transmission mechanism in the engine, which will be built based on a spatial hinge. This paper presents a study of the power and kinematic parameters of the spatial hinge of an internal combustion engine.

Keywords: engine; crankshaft; spatial hinge; polytropic indicator; modeling.

PROBLEM ANALYSIS

The modern development of internal combustion engines occurs in the following areas: increasing fuel efficiency; improving environmental safety; increasing power while reducing their metal consumption, reducing weight and overall dimensions; creation of hybrid propulsion systems of engines; increasing reliability and resource; improving serviceability; the use of alternative fuels and so on [1, 2]. Modern engines are complex technical systems with a large number of units aimed at increasing its efficiency, however, from the technical design of view, they practically do not differ from their classical schemes [3, 4]. Preparation of excellent performance and engine performance is carried out by increasing the efficiency of working cycles, the use of progressive design schemes, use of alternative fuels and modern economical electronic fuel injection systems [5].

The main part of internal combustion engines in construction equipment belongs to the group of crank-connecting rods, the structural schemes of which differ significantly from each other [6] (Figure 1). The disadvantage of this system is that at the time when the piston is in the upper position, then the maximum gas pressure acts on it, which creates the maximum force. This force presses on the crank without doing any work since there is no shoulder of such force at that moment in time. This problem has been solve partially by shifting the ignition process of the mixture and shifting the piston from the vertical [7].

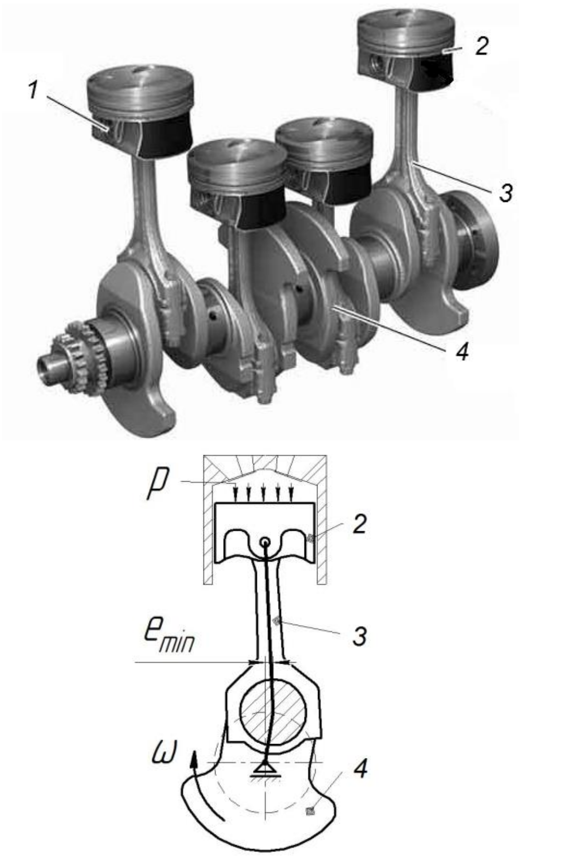

Fig. 1. Crank mechanism: *1* – finger; *2* – piston; *3* – connecting rod; *4* – crankshaft

In this paper, it is propose to consider a constructive modernization of the crank mechanism of the internal combustion engine with a spatial hinge, which allows to significantly shift the angle of rotation of the crankshaft when the piston is in the upper position to obtain the maximum effect from the work of gas expansion [8].

PURPOSE OF THE RESEARCH

Kinematic and power study of the effectiveness of the use of the spatial hinge in the internal combustion engine system.

METHODOLOGY

It is proposing to investigate the spatial hinge, which is capable of transmitting rotation between the shafts, located at an angle relative to each other (Figure 2) [8].

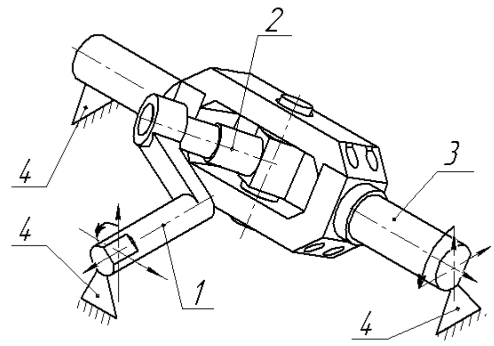

Fig. 2. The spatial joint: *1* – crank; *2* – connecting link; *3* – an oscillatory shaft; *4* – support

The joint consists of a crank *1*, a connecting rod *2* and a vibrating shaft *3*. All the hinge elements are in the housing *4*. The peculiarity of such a mechanism is that the input and output shafts are at an angle of 900 relative to each other. This representation of the spatial hinge is a variation of the Hook hinge [9]. The spatial hinge works as follows: turning the crankshaft 1 leads to the movement of the connecting rod *3*, which has an articulation with a crank and an oscillating shaft (Figure 3). As result of the movement of one end of the connecting rod *2*, the oscillating shaft 3 rotates, and when the crank *1* rotates at 360^0, the oscillating shaft rotates only 90^0. Thus, when the crank rotates, the shaft *3* vibrates and vice versa, when the shaft *3* rotates, the crank *1* rotates.

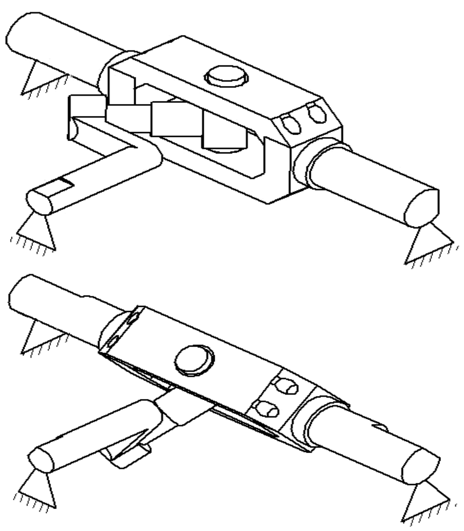

Fig. 3. The principle of operation spatial joint

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The crank radius is indicated by l_1 and the angle between the axis of the connecting rod link and the crank by φ (Figure 4).

If the axis of rotation of the crank intersects the axis of rotation of the connecting rod, the length of the cross-link through which the movement from the crank to the swinging shaft is transmitted is determined by the following relationship $l_2 = \frac{l_1}{\cos\varphi}$.

The work of the spatial hinge from the extreme position of the connecting rod is considered. In this case, the link *2* will be at an angle of 90^0 to the oscillating shaft *3*, which will also occupy its extreme position, will be describe by a maximum deflection angle:

$$\beta_{\max} = \arcsin(\frac{l_1}{l_2}) = \arcsin(\cos\varphi)\,. \qquad (1)$$

The rotation of the oscillating shaft will be determine by the angle of rotation:

$$\beta = \arctan(\frac{y_1}{h}) = \arctan(\frac{l_1\cos\alpha}{l_1\tan\varphi}) = \arctan(\frac{\cos\alpha}{\tan\varphi})\,, \qquad (2)$$

where:

h – the distance will be determine by the design parameters of the mechanism ($h = l_1 tg\varphi$);

$y_1 = l_1\cos\alpha$ – by the projection of the crank on the axis relative to which the displacement from the extreme position is calculated.

The angle of rotation of the oscillating shaft can will be determine by the following formula:

$$\beta = \arccos(\frac{l_1\tan\varphi}{l_2\cos\Theta})\,, \qquad (3)$$

where:

$\Theta = \arcsin(\frac{l_1\sin\alpha}{l_2})$ – the angle of rotation of the connecting rod.

Provide that the torque *3* acts on the oscillating shaft M_3, the magnitude of the torque on the crank *1* is determined:

$$M_1 = \frac{M_3}{l_{pl}} x_1\,, \qquad (4)$$

where:

$l_{pl} = \sqrt{l_1^2\cos^2\alpha + l_1^2 tg^2\varphi}$ – the distance of the action of the force that occurs in the connecting rod due to the transmission of torque from the vibration shaft;

$x_1 = l_1\sin\alpha$ – crank distance.

After the transformations and simplifica-

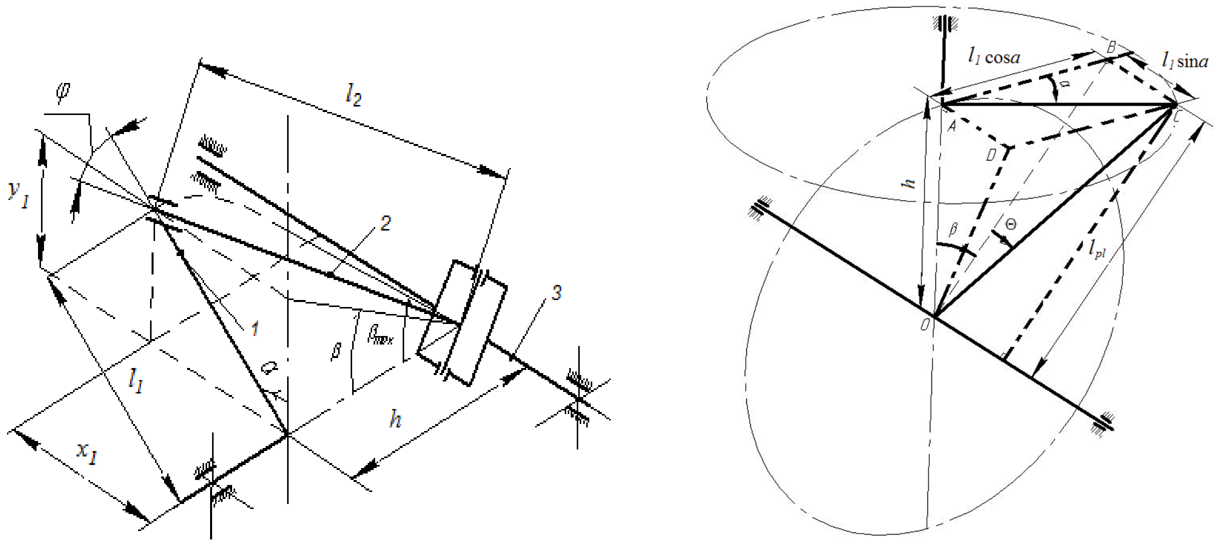


Fig. 4. Schemes of the spatial joint to determine its main characteristics

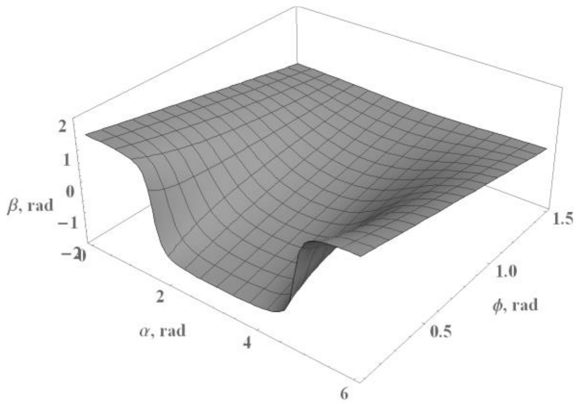
tions of expression (4) it were get

$$M_1 = \frac{M_3 \sin\alpha}{tg\varphi\sqrt{\cos^2\alpha \cdot ctg^2\varphi + 1}}. \qquad (5)$$

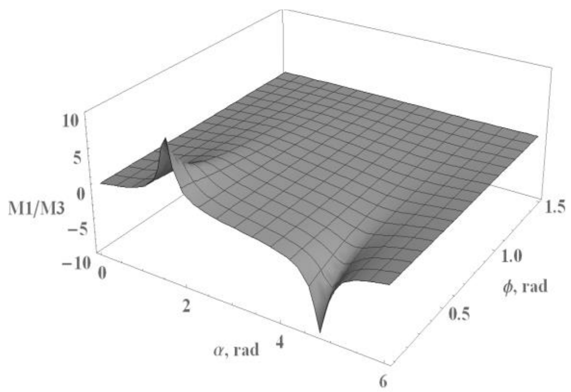
The characteristic of changing the transmission of torque from the drive shaft to the follower:

$$\frac{M_1}{M_3} = \frac{\sin\alpha}{tg\varphi\sqrt{\cos^2\alpha \cdot ctg^2\varphi + 1}} \qquad (6)$$

The graphs of the results of a research the change an angle of rotation β and the nature of the transmission of torque from the oscillating shaft to the crank in the range of a full revolution α of the crank at different angles φ of inclination of the connecting rod to the crank show in Figure 5.



a



b

Fig. 5. Graphs of changes the angle of rotation β (*a*) and transmission of moments M_1/M_3 (*b*)

From the graphs presented, it can be seen that the efficiency of torque transmission in the considered mechanism depends on the angle φ, which is recommended to choose in the range 0,4..0,7 radians. The most efficient transmission of torque from the swinging shaft to the crank occurs at a crank angle of 90^0 regardless of the angle of the connecting rod (Figure 6).

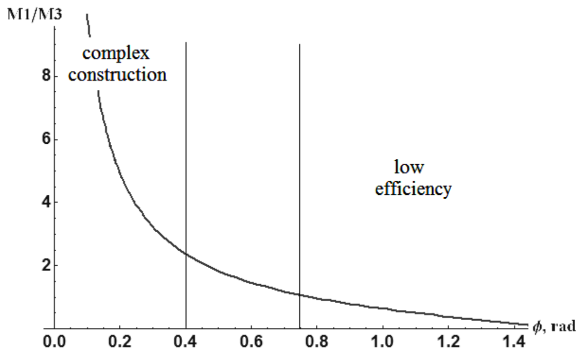


Fig. 6. The variation of gear ratio on the moment M_1/M_3 for $\alpha = 90^0$

When an angle φ between the connecting rod and the crank is less than 40^0, a sharp change in the angle of rotation of the oscillating shaft near the position of the crank 90^0 occurs.

Taking the time derivative of expression (2), we will be have [10]:

$$\frac{d\beta}{dt} = -\frac{\sin\alpha \cdot ctg\varphi}{1 + \cos^2\alpha \cdot ctg^2\varphi} \cdot \frac{d\alpha}{dt}. \qquad (7)$$

It is known that $\frac{d\beta}{dt} = \omega_\beta$ and $\frac{d\alpha}{dt} = \omega_\alpha$ then we will have kinematic gear ratio:

$$\frac{\omega_\beta}{\omega_\alpha} = -\frac{\sin\alpha \cdot ctg\varphi}{1 + \cos^2\alpha \cdot ctg^2\varphi}. \qquad (8)$$

The operation of the crank mechanism of an internal combustion engine with a central (Figure 7, *a*) and with a displaced piston placement (Figure 7, *b*) is investigated.

Piston stroke with central placement:

$$S_1 = l_3 + l_4 - l_3\cos\beta_{11} - l_4\cos\varphi_{11}. \qquad (9)$$

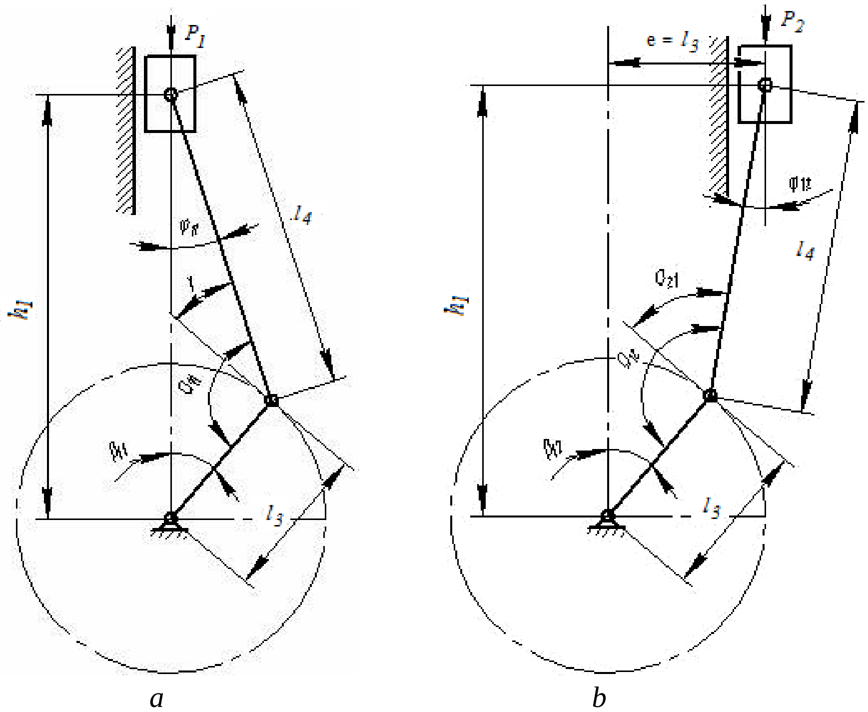


Fig. 7. The crank mechanism with the central (*a*) and with a displace location of the piston (*b*)

The deflection angle φ_{11} of the connecting rod is express by the angle β_{11} of rotation of the crank:

$$\varphi_{11} = \arcsin\left(\frac{l_3}{l_4}\sin\beta_{11}\right), \qquad (10)$$

where:

$$\cos\varphi_{11} = \sqrt{1-\sin^2\varphi_{11}}. \qquad (11)$$

Piston stroke with displacement with displacement:

$$S_2 = \sqrt{l_4^2 + 2l_3 l_4} - l_3\cos\beta_{12} - l_4\cos\varphi_{12}. \, (13)$$

The deviation angle φ_{12} of the connecting rod is express by the angle β_{12} of rotation of the crank:

$$\varphi_{12} = \arcsin\left(\frac{l_3}{l_4}(1-\sin\beta_{12})\right). \qquad (14)$$

Neglecting the weight of the piston, connecting rod and crank, the torque moment on the output shaft of the crank mechanism is determined.

For the central location of the piston:

$$M_{kp.c.p.n.} = P_1 \cdot \frac{\cos\gamma}{\cos\varphi_{11}} l_3, \qquad (15)$$

where:

P_1 – the force create by the pressure of gases as they expand;

γ – the projection angle of the connecting rod to normal to the crank ($\gamma = \frac{\pi}{2} - \beta_{11} - \varphi_{11}$).

For displacement of the piston with displacement:

$$M_{kp.e.p.n.} = P_2 \cdot \frac{\cos\Omega_{21}}{\cos\varphi_{12}} l_3, \qquad (16)$$

where:

P_2 – the force create by the pressure of gases as they expand;

Ω_{21} – an angle

($\Omega_{21} = \arccos\left(\frac{l_3}{l_4}(\sin^2\beta_{12} - \sin\beta_{12})\right) - \frac{\pi}{2}$).

The forces P_1 and P_2 are create by the pressure of the gases as they expand in the combustion chamber of the engine. The value and the change in gas pressure in the combustion chamber defined by the equation polytropic expansion [11, 12]:

$$p_z V_z^k = p_b V_b^k, \tag{17}$$

where:
p_z is the gas pressure in the combustion chamber at the beginning of the expansion process (for diesel engines $p_z = 5\ldots\ 12$MPa);
p_b is gas pressure at the end of the expansion process; V_z and V_b are respectively the initial (combustion chamber volume) and the final (expansion chamber volume) of the cylinder volume; k is polytropic indicator (for diesel engines 1,18…1,28).

It is known that the volume of the cylinder depends on its cross-sectional area and height (displacement of the piston):

$$p_b = p_z \frac{V_z^k}{V_b^k} = p_z \frac{S_0^k \left(\frac{\pi D^2}{4}\right)^k}{(S_0+S)^k \left(\frac{\pi D^2}{4}\right)^k} = p_z \frac{S_0^k}{(S_0+S)^k}, \tag{18}$$

where:
S – the stroke of the piston;
S_0 – the size of the combustion chamber;
D – piston diameter.

Accordingly, the forces acting on the piston from the gas pressure will be determined by the following dependencies:

$$P_1 = p_z \frac{S_0^k}{(S_0+S_1)^k} \frac{\pi D^2}{4}, \tag{19}$$

$$P_2 = p_z \frac{S_0^k}{(S_0+S_2)^k} \frac{\pi D^2}{4}. \tag{20}$$

For engine with dimensions $D = 130$mm, $l_3 = 70$mm, $l_4 = 175$mm, $S_0 = 5$mm, $k = 1,28$, $p_z = 6$MPa built dependence stroke crank mechanism according to the rotation of the crank shaft (Figure 8) and indicator diagram of the enlargement process of working gas in the engine cylinders (Figure 9).

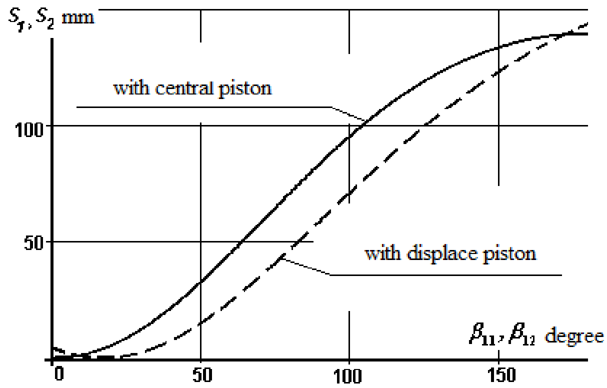


Fig. 8. Graphs changes of stroke

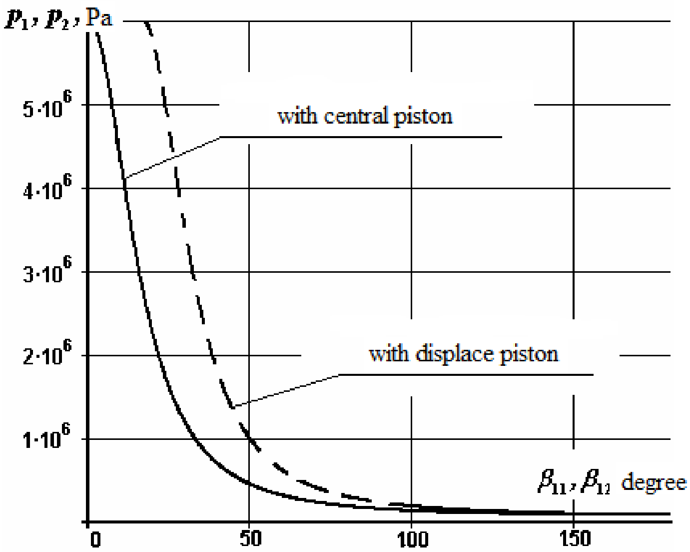


Fig. 9. Graphs change in gas pressure in the cavity of the cylinder

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Between the angle of rotation of the crank of the crank mechanism and the swinging shaft of the spatial joint there is the following relationship:

$$\beta_{11}(\beta_{12}) = \beta + \xi . \qquad (21)$$

where:
ξ - the angle of deviation of the crank from the angle of rotation of the swinging shaft of the spatial hinge.

The properties of motion transfer from the crank mechanism through the spatial hinge to the consumer for angle $\varphi = 45^0$ are investigate.

Figure 3 shows the graphs of the moment indicator on the crank of the spatial hinge (Figure 10).

From the graph (Figure 10) it can be seen that the maximum indicator torque in the engine with the traditional crank mechanism (curve 3) is approximately 1200 Nm, while in the engine using the spatial hinge the indicator torque can be increased to 5500 Nm (curve 2).

An engine with a displaced piston and a space hinge has been developed to transmit torque to the consumer (Figure 11).

CONCLUSION

To improve the efficiency of internal com-

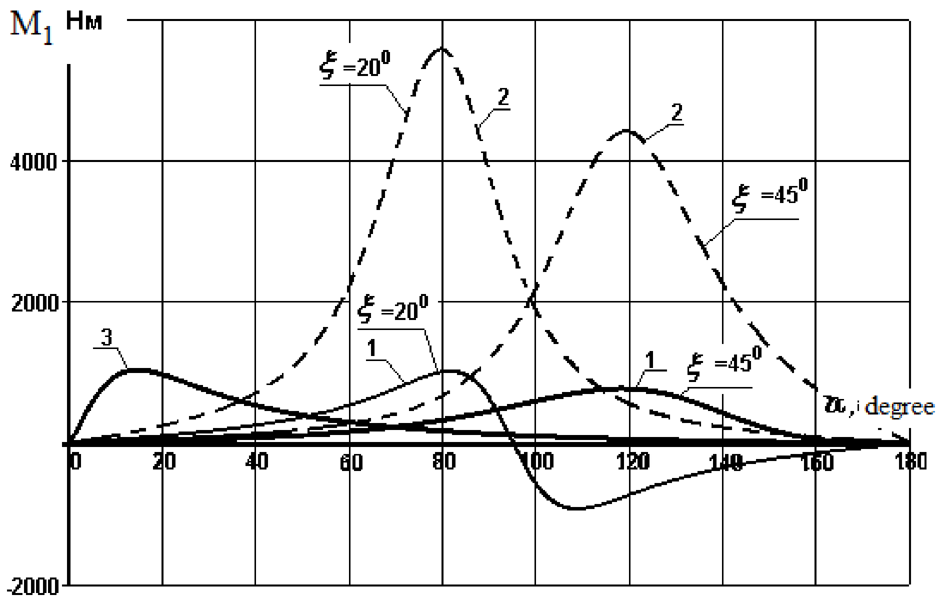

Fig. 10. Graphs change the moment indicator on the crank hinge spatial

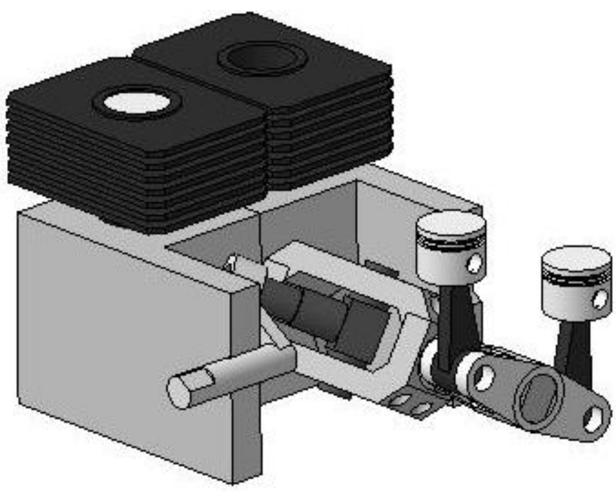

Fig. 11. The design of the engine with a spatial hinge and tilting shaft

bustion engines is propose to develop its structure shifted to accommodate the piston from a vertical crank length on the value and use advanced spatial joint for transmitting torque to the consumer. In the future, it is necessary to study in more detail the processes in the operation of such systems for a clearer understanding of their work.

REFERENCES

1. **Oglieve, Callum & Mohammadpour, Mahdi & Rahnejat, Homer**. (2017). Optimisation of the vehicle transmission and the gear-shifting strategy for the minimum fuel consumption and the minimum nitrogen oxide emissions. Proceedings of the Institution of Mechanical Engineers, Part D. Journal of Automobile Engineering. 231. 095440701770298. https://10.1177/0954407017702985.
2. **Mariani A., Foucher F.** (2014). Radio frequency spark plug: An ignition system for modern internal combustion engines. Applied Energy, Vol.122, 151-161. https://doi:10.1016/j.apenergy.2014.02.009.
3. **Mossa A. A., Hairuddin M., Nuraini A. A., Zulkiple A. A., Tobib J., & M., H.** (2019). Effects of Hot Exhaust Gas Recirculation (EGR) on the Emission and Performance of a Single-Cylinder Diesel Engine. International Journal of Automotive and Mechanical Engineering, Vol. 16(2), 6660-6674. https://doi.org/10.15282/ijame.16.2.2019.14.0501.
4. **Knoll G., Schönen R., Wilhelm K.** (1997). Full dynamic analysis of crankshaft and engine block with special respect to elasto-hydrodynamic bearing coupling. ASME ICE, Nr.28(3), 1-8.
5. **Abramchuk F. I., Gutarevich Ju. F.** (2006). Automobile engines. Kyiv, 476.
6. **Wahab A. M. A., Yusof Z., Rasid, Z. A., Abu A., & Rudin N. F. M. N.** (2018). Dynamic Instability of High-Speed Rotating Shaft with Torsional Effect. International Journal of Automotive and Mechanical Engineering, Vol. 15(4), 6034-6051. https://doi.org/10.15282/ijame.15.4.2018.23.0460.
7. **De la Cruz M, Theodossiades S and Rahnejat H.** (2010). Aninvestigation of manual transmission drive rattle. Proc IMechEPart K: J. Multi-body Dynamics, Vol.224(2), 167–181.
8. **Vlasov A. N.** (1995). The internal combustion engine. Patent No. 2046197.
9. **Heywood J. B.** (1988). Internal combustion engine fundamentals.New York, McGraw-Hill, 198.
10. **Ovchinnikov P. P. and other.** (2002). Higher Mathematics: Part 1: Linear and Vector Algebra: Analytic geometry. Introduction to mathematical analysis: Differential and integral calculus. Kyiv, Technika Publ., 592.
11. **Lenz, J. R.** (2002) Polytropic Exponents for common refrigerants. International Compressor Engineering Conference. Paper, 1528. https://docs.lib.purdue.edu/icec/1528.
12. **Rozhdestvensky Y, Khozeniuk N, Surovtsev S.** (2018). Dynamics and Lubrication Problem Analysis of Main Bearings for IC Engines Based on Coupling between a Crankshaft and a Flexible Whole Engine Block. Tribology in Industry, Vol.40(1), 139-148.

Дослідження ефективності застосування моделі просторового шарніра в конструкції двигуна внутрішнього згоряння

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Анотація. Двигун внутрішнього згоряння – це силова система, що застосовується в багатьох видах машин в галузі будівництва. Складна економічна ситуація навколо нафтопродуктів, пов’язана з їх дефіцитом, і сучасні стандарти екологічної безпеки вимагають переходу на альтернативні джерела енергії та пошуку шляхів модернізації двигунів внутрішнього згоряння для підвищення їх ефективності. Велика кількість двигунів внутрішнього згоряння відноситься до кривошипно-шатунної групи, проведений аналіз якої показав їх низьку технічну ефективність, що пояснюється їх конструктивною недосконалістю. Коли поршень знаходиться у верхньому положенні в момент займання горючої суміші в циліндрі, виникає максимальний тиск. Це забезпечує максимальну рушійну силу з мінімальним кутом кривошипа. Також збільшується тертя поршня об стінку циліндра, знос стінки циліндра, додатковий місцевий нагрів поверхні циліндра і перевантаження колінчастого вала. Таким чином, відбувається зменшення корисної потужності в результаті згоряння палива в двигуні.

Для підвищення ефективності кривошипно-шатунного механізму, пропонується застосувати нову конструкцію механізму передачі крутного моменту в двигуні, який буде побудований на основі просторового шарніра. У роботі представлено дослідження силових та кінематичних параметрів просторового шарніра двигуна внутрішнього згоряння.

Ключові слова: двигун, колінчастий вал, просторовий шарнір, політропний показник, моделювання.