Journal of Polish CIMAC
Gdansk University of Technology
The Faculty of Ocean Engineering and Ship Technology

ANALYSIS OF PISTON-CRANK SYSTEM BALANCING IN V-VR ENGINES

Sławomir Wierzbicki

University of Warmia and Mazury in Olsztyn
ul. Słoneczna 46A, 10-710 Olsztyn, Poland
tel. +48 89 5245222, fax. +48 89 5245150
e-mail: slawekw@uwm.edu.pl

Abstract

The advances made in materials science and modern material production and processing technologies support the design and serial production of structural solutions which could not have been applied in the past on account of their complexity or high cost.

The piston-crank system of a combustion engine remained practically unchanged over decades, and the only structural modifications involved the use of new materials for improved fatigue strength or attempts to limit the mass of system components to minimize inertial forces. Since the invention of the piston engine, in-line engines have been the predominant type of engine configuration on the market. Larger engines are built sporadically, and most of them have a "V" arrangement of cylinders and pistons for easier mounting. Selected vehicles, in particular sports cars, are equipped with boxer-type piston-crank systems. Recent years have witnessed the advance of new engine configurations with VR or V-VR piston-crank systems which are often referred to as W engines due to a unique arrangement of the connecting rod. The V-VR crankshaft-piston configuration supports the design of engines characterized by high displacement and reduced size for easier assembly in the vehicle's engine compartment. Such solutions are often deployed in upper class cars.

There is a general scarcity of information about V-VR piston-crank systems in literature, therefore, this study presents an overview of the above configuration and analyzes inertial forces impacting the discussed system.

Keywords*: piston-crank system, V-VR (W) engine, inertial force, engine balancing*

1. Introduction

The piston-crank system is one of the main structural units of a combustion engine. Crankshaft-piston configurations have to meet numerous requirements, the most important being:

– smooth engine operation,
– small size and weight relative to engine power,
– ease of assembly in a vehicle,
– low level of vibrations transmitted to the power transmission system and engine block,
– dynamic engine operation within a wide range of speeds and loads [1, 4, 5].

Cylinder arrangement is one of the characteristic features of a piston-crank system. Since the invention of the piston engine, in-line engines have been the predominant type of engine configuration characterized by simple structure of the crankshaft, the engine block and the cylinder head. Owing to the simplicity of the applied solutions, in-line engines are relatively inexpensive to build. However, in in-line engines with a higher number of cylinders, the piston-crank system and the entire engine are characterized by relatively low rigidity. The above poses assembly problems, in particular in selected types of passenger cars.

A different configuration of the piston-crank system is found in a V engine. The cylinders are aligned in two separate banks that are positioned towards one another at a certain angle. This configuration produces engines with reduced length and high rigidity. The discussed solution is ideal for vehicles with high performance requirements and a small engine compartment. A V engine incorporates two separate cylinder heads, an expanded engine block and cooling system, therefore, its main disadvantage are high production costs.

Boxer-type crankshaft-piston configurations are often encountered in sports cars. They are characterized by small overall height, which makes them the ideal solution for vehicles with a small engine compartment. Similarly to V engines, boxer configurations are quite expensive, and they are relatively rarely used.

2. Kinematic analysis of a piston-crank system

A piston-crank system in a combustion engine may rotate around its axis (simple setup) (Fig. 1) or off the central axis (eccentric setup) with a relatively small axial offset. According to the dependences described in literature [1, 4, 5], piston movement as a function of crankshaft rotation in a simple system can be expressed as:

$$x = r\left(1 - \cos\alpha + \frac{1}{2}\cdot\lambda\cdot\sin^2\alpha\right), \tag{1}$$

where:

α – crank angle,

$\lambda = r/l$,

r – crank radius,

l – length of connecting rod.

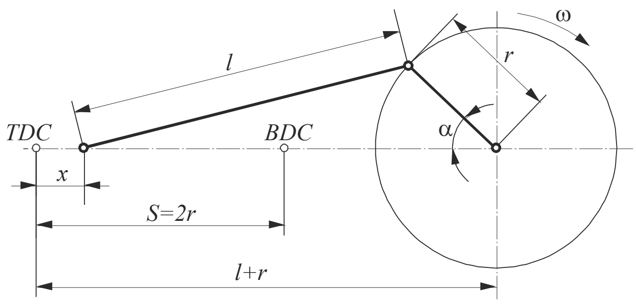

Fig. 1. Diagram of a simple piston-crank system

Instantaneous piston velocity is:

$$v = r\cdot\omega\cdot\left(\sin\alpha + \frac{\lambda}{2}\cdot\sin 2\alpha\right). \tag{2}$$

Instantaneous piston acceleration can be expressed as:

$$a = r\cdot\omega^2\cdot(\cos\alpha + \lambda\cdot\cos 2\alpha). \tag{3}$$

In kinematic analyses of eccentric piston-crank systems, the above equations take on a more complex form, but since the values of eccentricity which are practically applied in engines have an insignificant effect on piston location, velocity or acceleration, the formulas for simple crankshaft-piston arrangements can be used instead [1, 4].

Eccentric crankshaft-piston setups are less popular, and they are used mostly in light engines to minimize lateral pressure on cylinder bearing surface and facilitate piston movement past a dead center position. In eccentric arrangements, maximum acceleration does not take place at extreme positions of the piston. Piston accelerations for a simple system and an eccentric setup with eccentricity of $e = 0.3 \cdot r$ are presented in Fig. 2. The diagram clearly indicates that eccentricity has a minor influence on acceleration. In practice, even smaller eccentricity values are applied below $e = 0.2 \cdot r$ [1, 5].

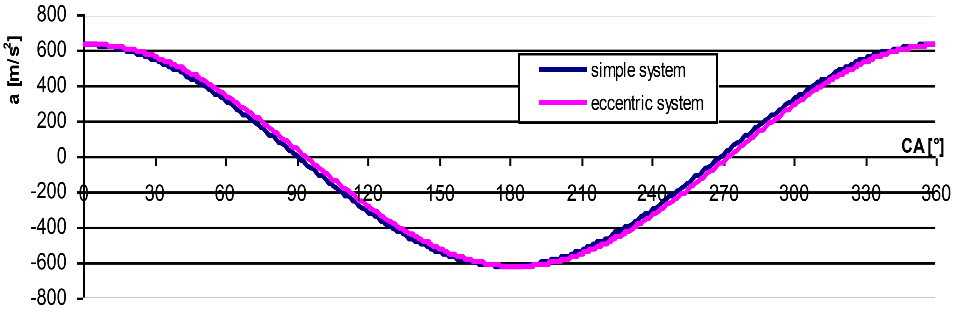

Fig. 2. Effect of crankshaft eccentricity on piston acceleration at e=0.3r

3. Piston-crank system in a V-VR engine

Recent years have witnessed the advance of new engine configurations with a V-VR piston-crank system which are often referred to as W engines due to a unique arrangement of the connecting rod. The architecture of a V-VR engine differs from that of a W engine which is equipped with three cylinder banks [1, 5]. A V-VR engine is characterized by a unique arrangement of the piston-crank system where cylinders are located in four banks. The discussed configuration is largely based on the original VR setup which has the features of an in-line engine and a V engine and combines the advantages of both solutions [6]. The main advantage of in-line engines is the relative structural simplicity of the engine block, timing gear and cooling system as well as the option of using only one cylinder head, which significantly lowers production costs. A V engine is characterized by reduced length and greater rigidity. A VR configuration is basically a V engine with a relatively small offset angle between cylinder banks, which reaches 10.5° to 15° in most applications. This solution produces a highly compact and rigid structure, a much shorter engine with only one cylinder head. A V engine with 6 cylinders is significantly shorter and less prone to vibrations than an in-line engine with 6 cylinders [6].

A VR engine evolved into a V-VR engine which combines two VR engines in a V-type arrangement. The piston-crank system of a V-VR engine is presented in Fig. 3, and the kinematic diagram of the discussed cylinder setup is shown in Fig. 4. The contemporary V-VR engines have 8, 12 or 16 cylinders. This solution produces a short, compact and rigid engine with low weight and relatively high capacity.

The engine block (Fig. 5) and the cylinders are manufactured of aluminum alloys, and this structure ensures high engine rigidity. The crankshaft is mounted to the engine block with a single compact cover which significantly increases engine rigidity.

In the analyzed arrangement, the crankshaft has non-standard architecture (Fig. 6a). The cylinders are arranged in four asymmetric banks for steady operation, the crankshaft has a spatial structure and the shared crankpins are mutually offset (Fig. 6b). Their offset is determined by the number of cylinders in the engine, and it is necessary to ensure uniform distribution of ignition force. A common crankpin is used for connecting rods of cylinders separated by an angle of 72°. In the discussed configuration, the ignition distance is 90°, and the crankpin serving the right-hand cylinder is offset by an angle of 18° in a direction opposite to crankshaft rotation. The above solution guarantees uniform ignition distances. In 12 cylinder engines, crankpins are separated by an angle of 12° but in an opposite direction to that noted in 8 cylinder engines, therefore in cylinders offset by 72°, the ignition distance is 60°. To guarantee adequate compression and shared use of the cylinder head in adjacent cylinder banks, pistons have beveled ends (Fig. 6c) – a single flat head can be used when one piston rotates relative to the other. The firing order in an 8 cylinder engine is 1-5-2-6-4-8-3-7.

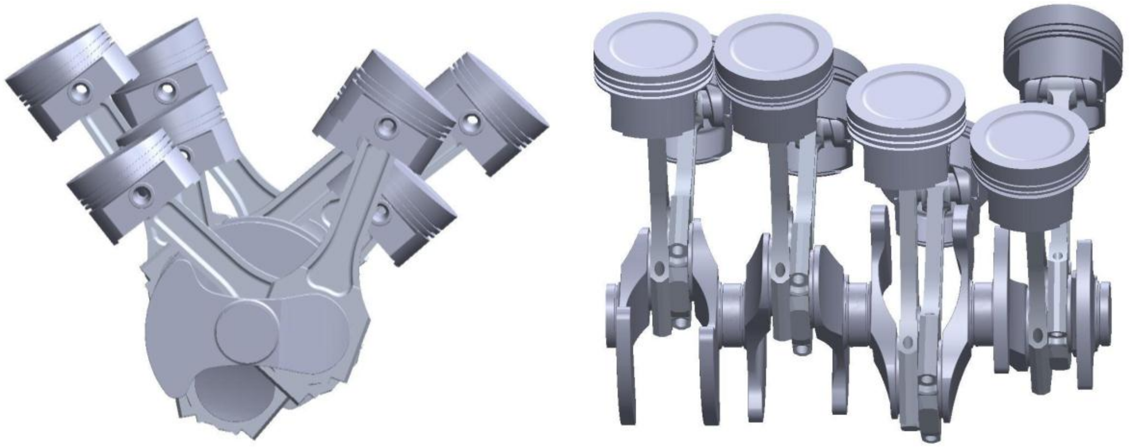

Fig. 3. 3D model of a piston-crank system in a V-VR8 engine

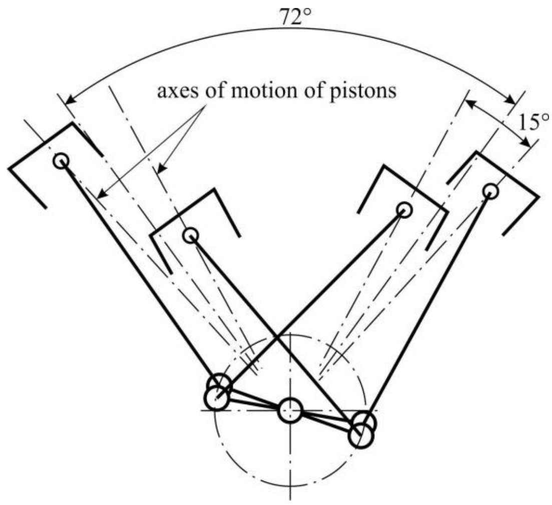

Fig. 4. Kinematic diagram of a piston-crank system in a V-VR engine

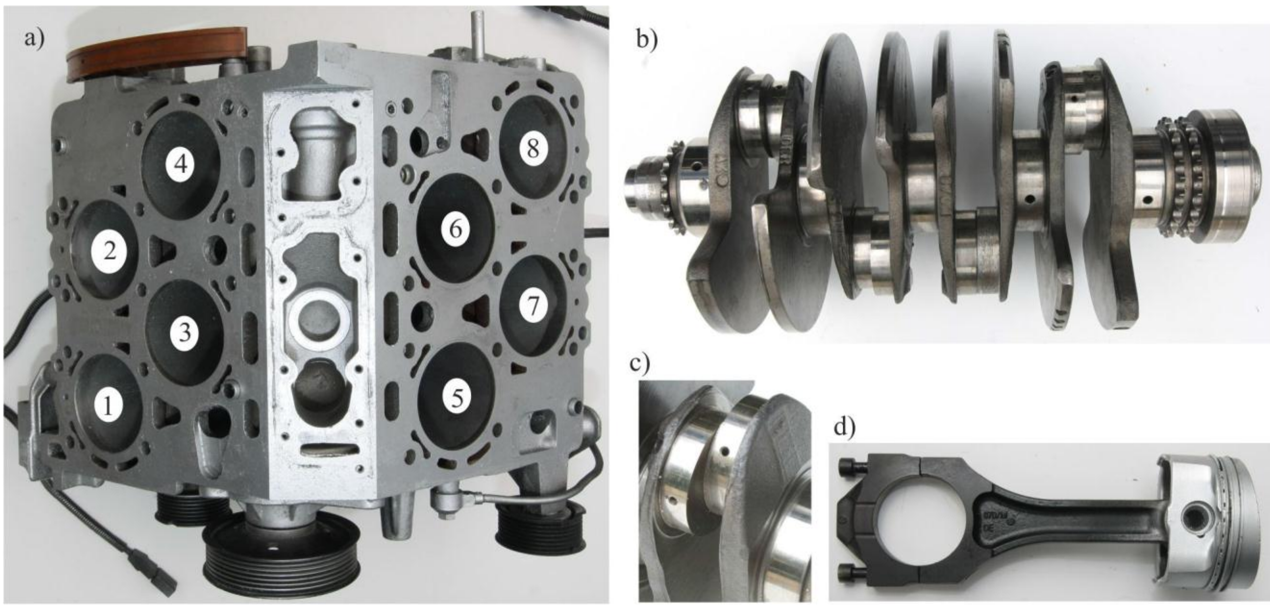

Fig. 5. Components of a piston-crank system in a V-VR engine: a) block with cylinder order, b) crankshaft of a W8 engine, c) crankpin, d) piston with connecting rod

4. Analysis of inertial forces in a piston-crank system of a V-VR engine

In a combustion engine, pistons move with variable acceleration, as per formula (3). For this reason, in multiple-cylinder engines, the inertial forces of reciprocating masses create a spatial force system and moments of inertia. Subject to the number of cylinders and their position, those forces can be mutually reduced or they can create an unbalanced system where a significant load is imposed on the piston-crank system and transferred to the vehicle frame.

In line with the principles of mechanics, the inertial forces of reciprocating masses in an engine are determined by the mass of those components and their instantaneous acceleration. Inertial forces can be minimized by reducing the mass of each component and distributing cylinders in a way that supports the self-balancing of inertial forces in the engine.

Most analyses of inertial forces in piston-crank systems in combustion engines focus on first-order and second-order inertial forces and their moments. Higher-order forces are omitted due to their insignificant value [1, 4, 5].

In a V-VR engine, cylinders are positioned in four banks, and in order to determine inertial forces in the analyzed crankshaft-piston arrangement, the formula for piston acceleration (3) has to be generalized by accounting for the displacement between the cylinder and the main axis γ and the displacement between individual cranks and the first cylinder crank ρ (Fig. 7). In this case, the acceleration of a single piston in its axis of motion is calculated using the following formula:

$$a = r \cdot \omega^2 \cdot [\cos(\alpha + \rho - \gamma) + \lambda \cdot \cos 2(\alpha + \rho - \gamma)], \tag{4}$$

where:

- γ – displacement between the cylinder and the main axis.,
- ρ – angular displacement between a crank relative to the assumed position of the first cylinder crank.

The piston acceleration formula can be written as the sum of first-order *a'* and second-order *a"* accelerations which equal:

$$a' = r \cdot \omega^2 \cdot (\cos(\alpha + \rho - \gamma)), \tag{5}$$

$$a'' = r \cdot \omega^2 \cdot \lambda \cdot \cos(2(\alpha + \rho - \gamma)), \tag{6}$$

Piston acceleration and reciprocating mass *m* can be then used to determine the inertial forces acting on the axis of one of the cylinders:

$$F' = m \cdot r \cdot \omega^2 \cdot (\cos(\alpha + \rho - \gamma)), \tag{7}$$

$$F'' = m \cdot r \cdot \omega^2 \cdot \lambda \cdot \cos(2(\alpha + \rho - \gamma)). \tag{8}$$

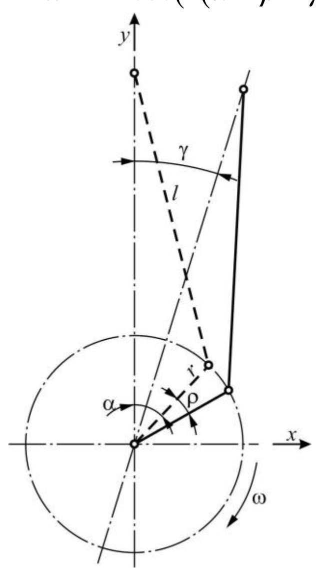

Fig. 6. Diagram of the analyzed piston-crank system

In the analyzed engine, vectors of inertial forces of reciprocating masses are not found in a single plane, therefore, the projections of individual forces onto two perpendicular axes should be analyzed for greater convenience. The inertia components of every piston $F[i]$ will have the following projection on the coordinate axes (Fig. 6):

$$F'_x[i] = \sin\gamma[i]\cdot m\cdot\left(r\cdot\omega^2\cdot\cos(\alpha+\rho[i]-\gamma[i])\right), \tag{9}$$

$$F'_y[i] = \cos\gamma[i]\cdot m\cdot\left(r\cdot\omega^2\cdot\sin(\alpha+\rho[i]-\gamma[i])\right), \tag{10}$$

$$F''_x[i] = \sin\gamma[i]\cdot m\cdot\left(r\cdot\omega^2\cdot\lambda\cdot\cos(2(\alpha+\rho[i]-\gamma[i]))\right), \tag{11}$$

$$F''_y[i] = \cos\gamma[i]\cdot m\cdot\left(r\cdot\omega^2\cdot\lambda\cdot\sin(2(\alpha+\rho[i]-\gamma[i]))\right), \tag{12}$$

where:

$a[i]$, $\gamma[i]$, $\rho[i]$ – acceleration, cylinder displacement and angular displacement of the crank of the i-th cylinder relative to the crank of the first cylinder, respectively.

A procedure was developed in the Matlab application for determining the moment of inertia in any crankshaft-piston configuration. The following parameters of the piston-crank system were used:

r = 45 mm,
l = 168.5 mm.

The vector of angular displacement between individual cranks and the first cylinder crank in an 8 cylinder engine was:

$\rho[i]$ = [0, 180, 195, 15, 342, 162, 177, 357].

The vector of displacement between cylinders and the y-axis was:

$\gamma[i]$ = [316.5, 316.5, 331.5, 331.5, 28.5, 28.5, 43.5, 43.5].

The projections of first-order inertial forces on two perpendicular axes for each cylinder in an 8 cylinder engine are presented in Fig. 7. This diagram clearly indicates that the forces of inertia are mutually balanced. In the analyzed engine, the ignition distance is 90°CA, and similarly to a V engine where the banks form a 90° angle, those forces are mutually balanced [1, 5]. The projections of second-order inertial forces on each axis are shown in Fig. 8. Second-order forces of inertia are mutually balanced only in one plane, whereas in the second plane, they are summed, causing the system to be out of balance. In practice, inertial forces are balanced with the use of additional balance shafts which rotate at twice the speed of the crankshaft [1, 5, 6]. The forces of inertia, presented in Figs. 8 and 9, are expressed in relative units, and the actual values of those forces are proportional to the square of the instantaneous velocity of the crankshaft.

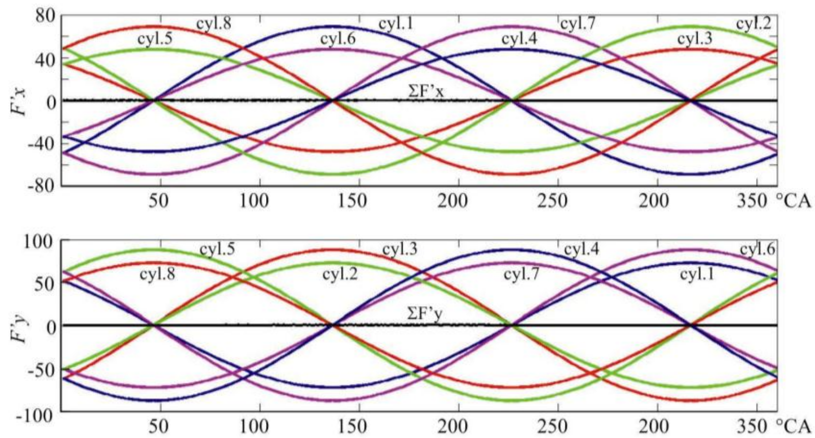

Fig. 7. First-order inertial forces of cylinders in a V-VR8 engine

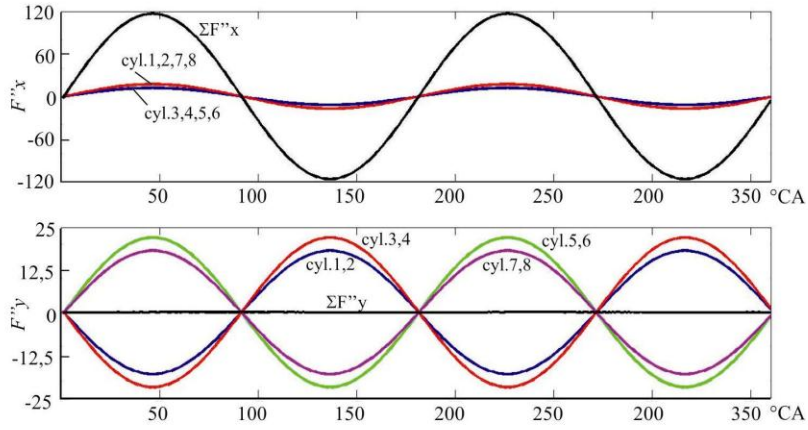

Fig. 8. Second-order inertial forces of cylinders in a V-VR8 engine

The moments of inertia relative to the longitudinal center axis of the crankshaft are summed to determine the resultant moment of inertia in the analyzed engine. The resultant moments of inertia relative to the x-axis and the y-axis are:

$$M_x = \sum_{i=1}^{8} \left(F_y[i] \cdot k[i] \right), \tag{13}$$

$$M_y = \sum_{i=1}^{8} \left(F_x[i] \cdot k[i] \right). \tag{14}$$

First-order and second-order moments of inertia in each bank of an 8 cylinder engine are shown in Fig. 9 and Fig. 10 (similarly to the forces of inertia, also the values of the moments of inertia are given in relative units). They were calculated using the following vector of distance between the cylinders and the longitudinal center axis of the crankshaft $k[i]$:

$k[i]$ = [-0.104, 0.026, -0.039, 0.091, -0.091, 0.039, -0.026, 0.104] [m].

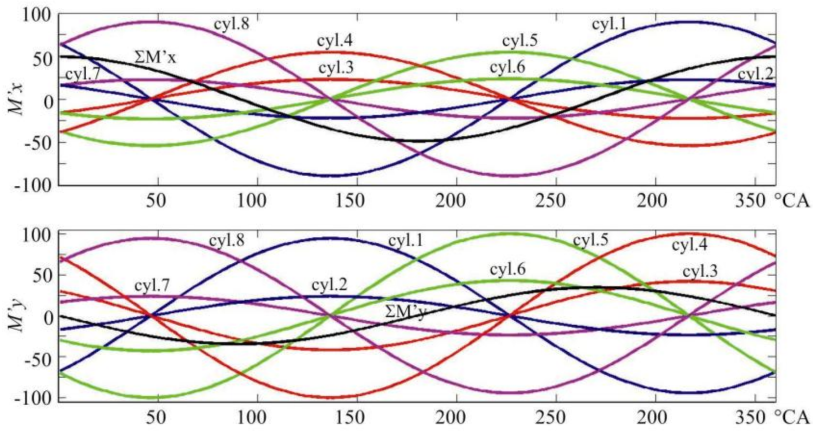

Fig. 9. First-order moments of inertia in a V-VR8 engine

An analysis of moments of inertia in the discussed engine indicates that the first-order moment is relatively small and it changes with the crankshaft rotation angle, therefore, it can be reduced by applying additional counterweights to the crankshaft [1, 2, 3, 5]. Second-order moments of inertia are even smaller, and they do not exert significant loads on the system.

In a 12 cylinder engine with a V-VR piston-crank system, first-order and second-order inertial forces are balanced out. Each of the four banks contains three cylinders whose crankpins are separated by an angle of 120°, and similarly to a 3-cylinder in-line engine, the resulting inertial forces are balanced [1, 5]. The moments of inertia remain unbalanced, but their sum is relatively small and it does not exert a significant load on the engine.

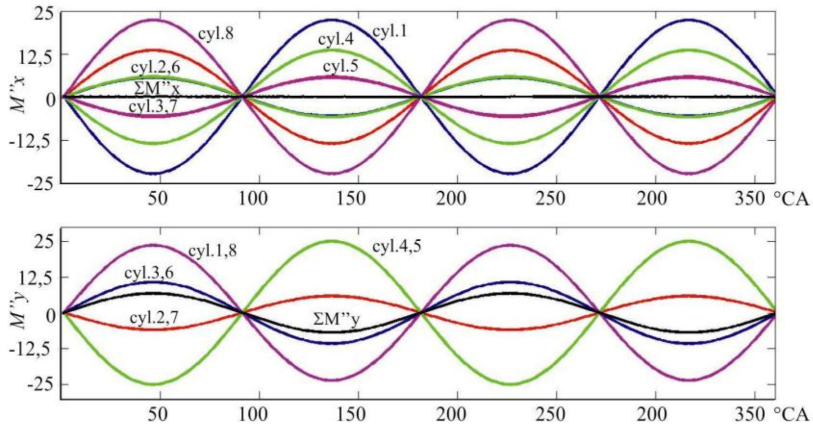

Fig. 10. Second-order moments of inertia in a V-VR8 engine

5. Conclusions

In an 8 cylinder engine with a V-VR piston-crank system, first-order inertial forces are balanced, and additional balance shafts are applied to equalize second-order forces. Balance shafts are increasingly often used in 4 cylinder engines to ensure smooth engine operation. First-order and second-order moments of inertia are not completely balanced in a V-VR8 engine where cylinders are not symmetrically distributed relative to the longitudinal center axis of the crankshaft. In part, the moments of inertia are mutually balanced, the resulting moments are very small, and they do not exert a significant load on the system.

Owing to its specific architecture, a 12 cylinder engine is characterized by mutually balanced first-order and second-order inertial forces, and similarly to an 8 cylinder engine, its moments of inertia remain unbalanced.

The crankshaft-piston configuration of a V-VR engine contributes to smooth engine operation without exerting inertial load on the engine or the engine suspension system. The discussed cylinder arrangement produces compact engines with reduced length and high capacity which can be easily assembled in vehicles. The compact architecture of the crankshaft guarantees low inertia of the entire system, and it contributes to high engine performance.

References

[1] Jędrzejowski, J.: *Mechanika układów korbowych silników samochodowych*. WKiŁ, Warszawa 1986.

[2] Mosakowski, R.: *Analiza wyrównoważenia silników rzędowych dwusuwowych czterocylindrowych*. Archiwum Motoryzacji, nr. 4/2006, pp. 343-358, 2006.

[3] Mosakowski, R.: *Analysis of balancing of six-cylinder in-line two-stroke internal combustion engines*. Combustion Engines – Silniki Spalinowe, nr 4/2009 (139), pp. 22-33, PNTSS, 2009.

[4] Niewiarowski, K.: *Tłokowe silniki spalinowe*. WKiŁ, Warszawa, 1983.

[5] Wajand, J. A., Wajand, J. T.: *Tłokowe silniki spalinowe średnio- i szybkoobrotowe*. WNT Warszawa, 1993.

[6] VW, Audi – materiały reklamowe i prasowe.