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KINEMATIC OF MARINE PISTON-CRANKSHAFT SYSTEM

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Abstract

Two-stroke, slow speed main engines are often installed on merchant ships, because of its very high efficiency. That kind of engine has an output of about 5500 kW per cylinder. The mass of piston-crankshaft system reaches over a dozen tons. That reciprocating masses are source of high level of dynamic inertia forces (mass forces). Those forces have big influence on engines working parameters and characteristics. One of them is instability of crankshaft rotational speed, which leads to dangerous torsional vibrations of propulsion system. Some inconsistency can be observed during analysis of piston-crankshaft system kinematic. In the theoretical engine books, the piston speed and acceleration has only two harmonic components, the inertia forces are depended on engine rotation speed and they doubled rotation speed. However, empirical formulas, published by engines producers, give us at least five harmonic components of mass forces. The author tries to find out the theoretical reason of existing (measured) higher harmonic orders of engines inertia forces. It is a first step for developing monitoring system of propulsion system's torsional vibrations coupled with axial vibration, dynamic shaft line alignment and crankshaft springing. In the paper two analytical methods of piston displacement, speed and acceleration are presented. Well-known (from literature) equations are compared with more-detailed analytical procedures. The analysis was performed for one of the biggest marine MAN B&W engine, type 7K98MC. A discussion about the analysis results was included in the final part of the paper.

Keywords: *marine propulsion system, piston-crankshaft kinematic, slow speed engines, inertia forces, propulsion system vibrations*

1. Introduction

Nowadays, mostly two-stroke, slow speed main engines have been installed on merchant ships [6]. The engines are connected to a directly driven propeller by a relatively short shaft line. These engines are powerful and heavy. The cylinder diameter reaches a diameter value equal to 960-980 mm. That kind of engine has an output of about 5500 kW per cylinder. The crankshaft pin is greater than 1000 mm. An example of FEM (Finite Element Method) model of crankshaft of MAN B&W engine, type: 7K98MC (seven cylinders, bore diameter – 980 mm) is presented in Fig. 1. The mass of piston with piston rod is approximately equal to 6 ton; the mass of connecting rod (all set) is approximately equal to 8 ton. Therefore, reciprocating motion of piston-crankshaft system generates high level of dynamic inertia forces [1]. Those forces have big influence on crankshaft springing [11] and torsional vibration of power transmission system [12]. Torsional vibrations of the marine power transmission system are usually most dangerous for the shaft line and the crankshaft [1]. What is more, some authors [3] try to use crankshaft's rotational speed as a diagnostic data for damage monitoring, for instance, of elements of exhaust system. This is a reason, why detailed, mathematical description of piston motion is important.

Some inconsistency can be observed during analysis of piston-crankshaft system motion. In the theoretical engine books [13-17], the piston speed and acceleration has only two harmonic components. It is mean that inertia forces are depended on engine rotation speed, and doubled rotation speed. However, empirical formulas, published by engines producers, give us five harmonic components [1, 2]. The author tries to find out the theoretical reason of existing higher harmonic orders of engines inertia forces. It is a first step for developing monitoring system [4, 5, 10] of propulsion system's torsional vibrations coupled with axial vibration [7], dynamic shaft line alignment [8, 9] and crankshaft springing [11].

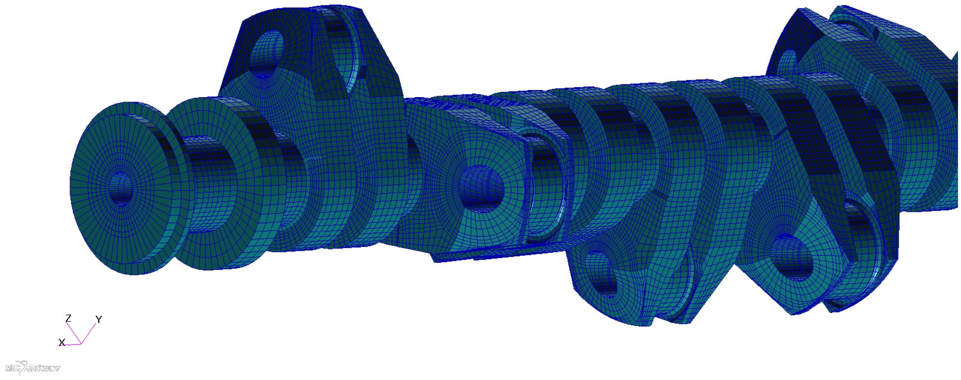

Fig. 1. Detailed 3-D FEM model of a crankshaft of 7K98MC main engine

2. Basic assumptions

Changing in oscillating movement of piston on the rotatable of crankshaft is main task of piston-crankshaft system. Piston movement is caused by fuel combustion (gas forces), but during analyses of propulsion system dynamic characteristics, also inertia forces (mass forces) should be taken into account. Both these forces are periodical but not harmonics. After simple analysis (e.g. FFT – Fast Fourier Transformation) we can decompose dynamic forces functions on several harmonic components. The components are depended on engine rotational speed multiplied by natural numbers (there are half-components for four-stroke engines). Usually, 24 harmonic components of gas forces are determined for two-stroke marine engines and 5 harmonic components for mass forces [1].

Piston is connected with crankshaft by piston rod (only for slow-speed marine engines), slider, connecting rod and crank pin. Kinematic scheme of piston-crankshaft system is presented in Fig. 2. Piston rod has no effect on the system kinematic. Therefore, the x value determining the location of piston is associated with slider. The starting point of the axis is located in the piston top dead centre.

Constant rotational speed of the crankshaft ($d\alpha/dt = \omega$ = const.) is the main assumption, in the paper. The system is moving without any frictions and without any excitations, only by inertia. Determination of the function of piston location, speed and acceleration is the target of this paper. Crankshaft location angle (α) is independent variable of the function. Knowledge of the acceleration function of piston allows us to determine the function of system inertia forces. And then, the natural (without excitations) instability of rotational speed of engine can be determined.

The piston is moved between top and bottom dead centre. The stroke ($S=2\times R$) is the distance of the piston moving. Very important coefficient describes dependence between crankshaft radius, length of connecting rod and it is defined as:

$$\lambda = R/L, \tag{1}$$

where:

λ – piston-rod coefficient,
R – crankshaft radius,
L – length of connecting rod.

λ value of the analysed engine (7K98MC) is equal to 0.413. Slow speed marine engines have greater λ coefficients in comparison to other engines (usually λ = 0.2-0.35).

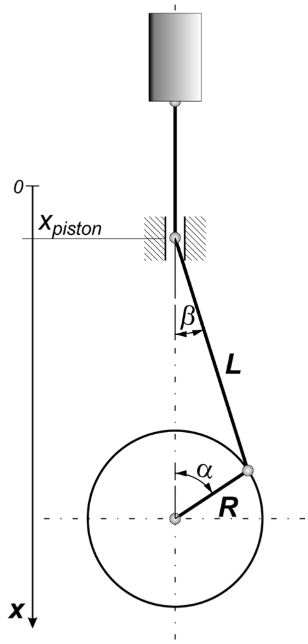

Fig. 2. Scheme of piston-crankshaft system movement

3. Piston kinematic

The location of the piston can be described (see Fig. 2) by the following equation:

$$x = R(1-\cos\alpha) + L(1-\cos\beta)\,. \tag{2}$$

Piston moving parameters should be described as a function of crankshaft rotational speed. The speed is depended directly on crankshaft piston angle (α). If the crankshaft speed is constant, the dependence between crankshaft speed and location is obvious. Dependence between crankshaft position angle (α) and connecting position angle (β) is also necessary, and can be described (with using equation 1) as:

$$\cos\beta = \sqrt{1-\lambda^2\sin^2\alpha}\,. \tag{3}$$

In most theoretical engines books [13, 14] the right side of the equation 3 is simplified to the following form:

$$\sqrt{1-\lambda^2\sin^2\alpha} \cong 1-\frac{1}{2}\lambda^2\sin^2\alpha\,. \tag{4}$$

Therefore, two different equations can be derivate on the base of equations 1-4. First one (well known) is approximation road of piston (equation 5), and second one – exact equation 6. Equations 5 and 6 were used to solving motion of piston of engine type 7K98MC.

$$x_{app} = R\left[1-\cos(\omega t)+\frac{1}{2}\lambda\sin^2(\omega t)\right], \tag{5}$$

$$x_{ex} = L\left\{1+\lambda\left[1-\cos(\omega t)\right]-\sqrt{1-\lambda^2\sin^2(\omega t)}\right\}, \tag{6}$$

where:
x_{app} – approximated piston displacement,
x_{ex} – exact piston displacement.

The piston road was compared with clear sinusoid trajectory which was named “x”. The piston displacement as a function of crankshaft rotation angle was presented in Fig. 3.

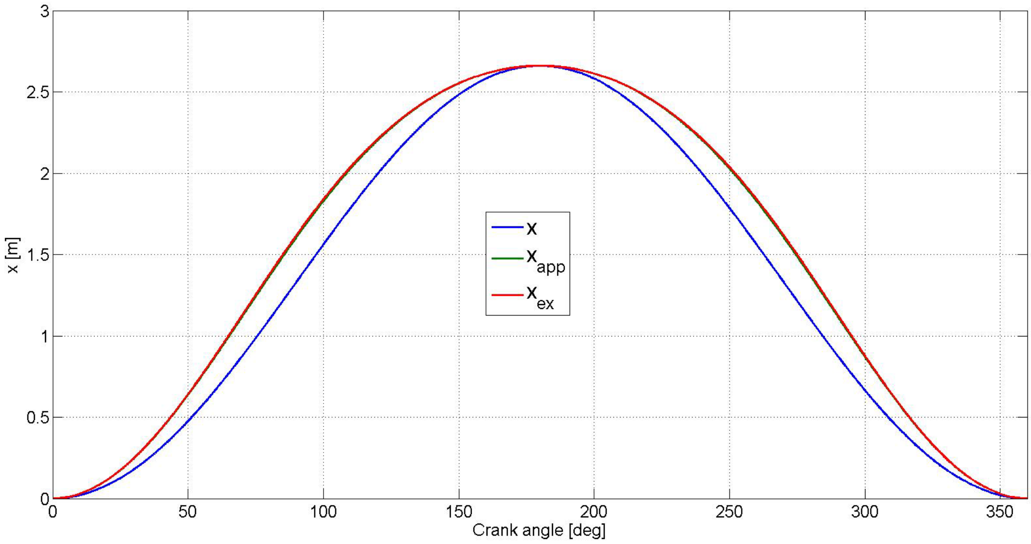

Fig. 3. Piston displacement as a function of crankshaft rotational angle

Approximated and exact piston displacements are nearly overlapped – the differences reaching values of the half of one percent. The values of relative errors were shown in Fig 4.

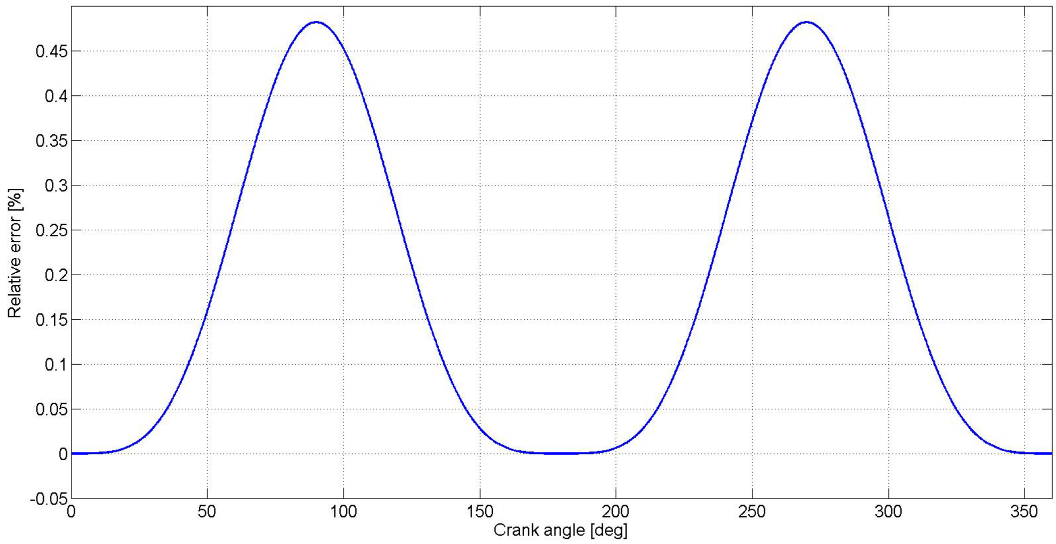

Fig. 4. Relative errors of piston displacement determination

Commonly used approximation (equation 4) is acceptable from quasi-static analysis of engine working parameters. In the next step, piston speed and acceleration should be determined, and they have the form as:

$$v_{app} = \frac{\mathrm{d}x}{\mathrm{d}t} = R\,\omega\left[\sin(\omega t) + \frac{1}{2}\lambda\sin(2\omega t)\right], \tag{7}$$

$$v_{ex} = R\,\omega\left\{\sin(\omega t) + \frac{1}{2}\lambda\sin(2\omega t)\cdot[1-\lambda^2\sin^2(\omega t)]^{-1/2}\right\}, \tag{8}$$

$$a_{app} = \frac{\mathrm{d}v}{\mathrm{d}t} = R\,\omega^2[\cos(\omega t) + \lambda\cos(2\omega t)], \tag{9}$$

$$a_{ex} = R\,\omega^2\left\{\cos(\omega t) + \lambda\cos(2\omega t)\cdot[1-\lambda^2\sin^2(\omega t)]^{-1/2} + \frac{1}{4}\lambda^3\sin^2(2\omega t)\cdot[1-\lambda^2\sin^2(\omega t)]^{-3/2}\right\}, \tag{10}$$

where:
v_{app} – approximated piston speed,
v_{ex} – exact piston speed,
a_{app} – approximated piston acceleration,
a_{ex} – exact piston acceleration,

The piston speed (approximated and exact) was also compared with clear sinusoid trajectory, which was named “v”. The piston speed as a function of crankshaft rotation angle was presented in Fig. 5. The errors values between approximated and exact piston speed is a little bit greater (up to 1.2%) but it is still not very high.

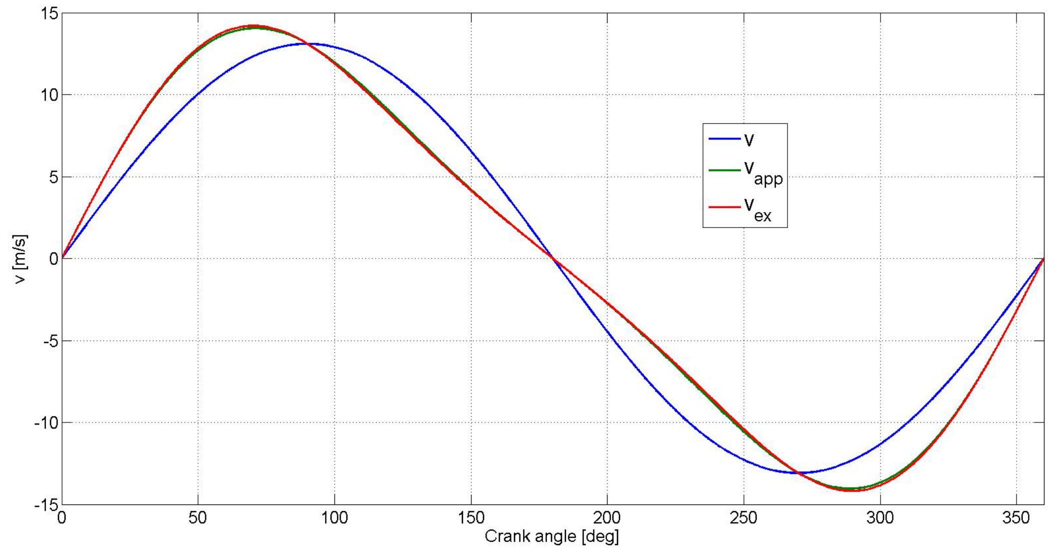

Fig. 5. Piston speed as a function of crankshaft rotational angle

Similar (like displacement and speed) comparison was performed for piston acceleration. Clear harmonic trajectory was marked by “a”. The piston accelerations as a function of crankshaft rotation angle were presented in Fig. 6. Noticeable differences between acceleration determined on the base of exact and approximated equations can be observed. The relative error reaches 3%. It is mean that acceleration is determined with absolute error, which exceeds 5 m/s^2, for the analysed slow-speed engine. The values of absolute errors were shown in Fig 7.

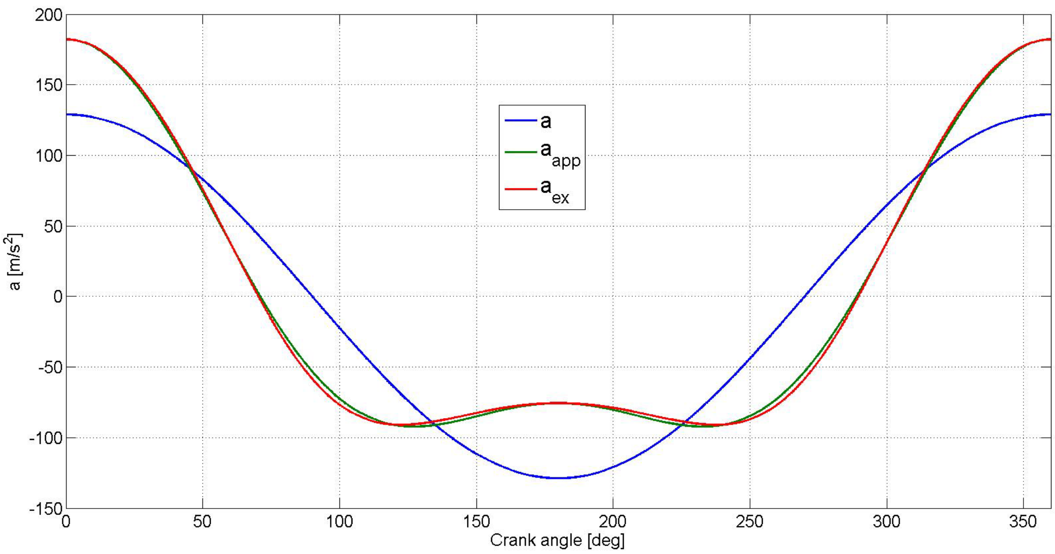

Fig. 6. Piston acceleration as a function of crankshaft rotational angle

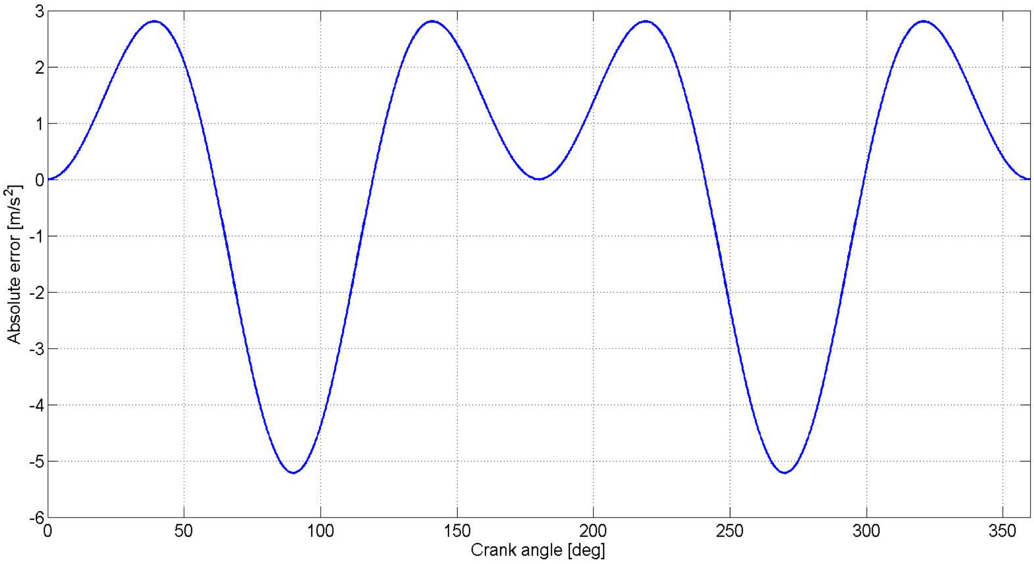

Fig. 7. Absolute errors of piston acceleration determination

The values of piston acceleration are very high (up to 200 m/s^2). Those accelerations, together with high piston-crankshaft system masses give us very high level of dynamic mass forces. During vibrations analyses of propulsion system (and also ship hull and deckhouse), not only excitations amplitudes are important. Forces frequencies are even more important due to resonance vibration threat. Especially, during torsional vibration analysis of power transmission system, frequencies of gas and mass forces are essentials. Frequencies characteristics (FFT) of piston accelerations, for both assumptions, were presented in Fig. 8. Logarithmic scale of vertical axis was used to highlighting higher harmonic components.

First two harmonic components are similar for both analysis methods. However, exact equation gives us additionally 4th, 6th and even 8th harmonic components. This is consistent with engines producers data based on experience. Piston kinematic analysis cannot be based on approximated equations, if vibrations analyses of propulsion system are our target.

4. Conclusions

In the theoretical engine books based on simplified equation of the piston movement (kinematic), its speed and acceleration, have only two harmonic components. Inertia forces are depended on engine rotation speed and they doubled rotation speed. Analysis based on exact equations (No. 6, 8 and 10) show that movement disorders are more complicated. At least five harmonic orders should be taken into account. This is in line with empirical formulas published by engines producers. Inertia force error of the analysed engine, coming from one cylinder, is greater than 50 kN, if our analysis is based on simplified equations. Forces values and frequencies are very important during propulsion system vibrations analyses.

Presented work is a first step for developing monitoring system of marine propulsion system's torsional vibrations coupled with axial vibration, dynamic shaft line alignment and crankshaft springing. In the next step, the mathematical model of the piston-crankshaft system will be improved. Real rotational speed of the crankshaft is not constant, and mass moment of inertia of the system is also variable. Specialized software will be developed for solving those problems.

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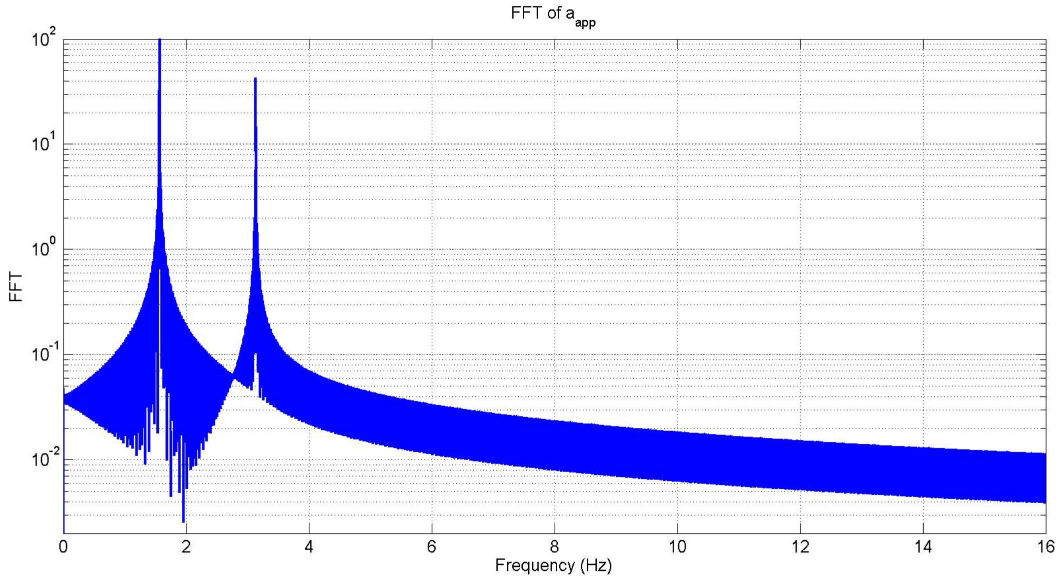

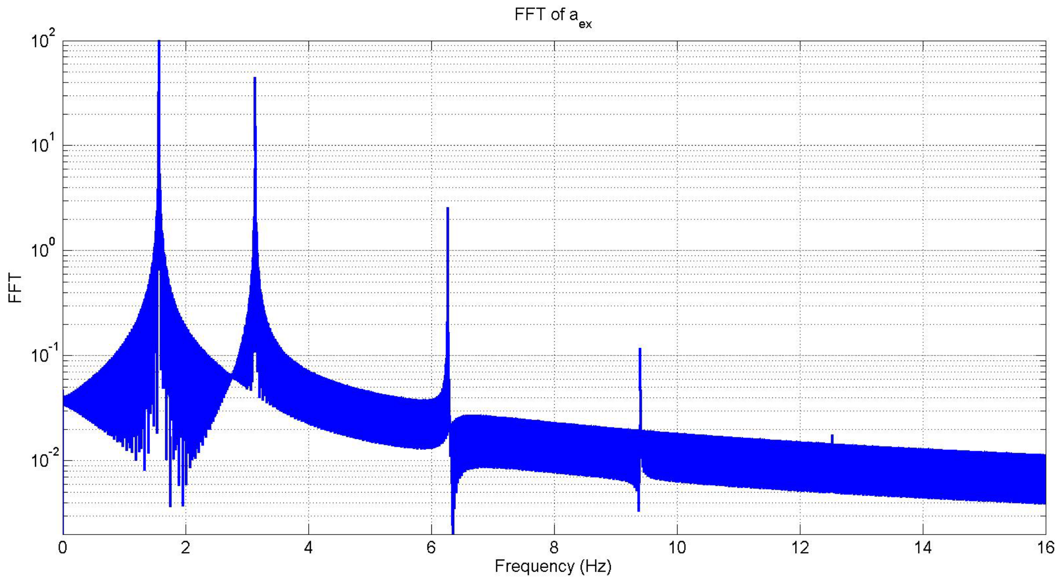

Fig. 8. Frequencies characteristics of piston accelerations determined on the base of approximated and exact equations

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