Multibody analysis of the opposed-piston aircraft engine vibrations

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Abstract. One of the characteristic features of piston engines are vibrations caused by the pistons moving in the cylinders. During the engine design process, it is necessary to determine the level of vibration that can occur in the engine. This is especially important for aircraft engines. Due to the minimization of the weight of the aircraft, it is necessary to limit the factors that may cause damage to the structure. One of these factors is engine vibration, which can cause resonance and, consequently, a dangerous stress concentration. Long-term action of variable loads may also lead to the formation of fatigue cracks. The article presents the results of a multibody analysis of an opposed-piston diesel engine. It is a two-stroke three-cylinder aircraft engine. The engine has two crankshafts and six pistons that run opposite each other, but the rotation of the shafts is shifted in phase 14°. Engine vibration will also be caused by crankshafts which, to reduce weight, are not equipped with counterweights. The calculation results are presented in the form of time courses of forces and displacements on the engine supports and FFT analysis of the vibration velocity. The results show that the maximum vibration velocity is 7 mm/s and occurs at a frequency of 140 Hz, which corresponds to twice the rotational speed of the crankshafts. The results obtained from the tests allow for the selection of the flexible elements used in the real prototype engine supports.

1. Introduction

Opposed piston two-stroke engines have been around for over 100 years. From 1890, they were first produced in Germany, and then in the USA, Great Britain and France. They are used to propel civilian and military vehicles, ships and even aircraft [1]. However, since the beginning of the 1970s, as a result of stricter exhaust emission standards, especially for piston engines, manufacturers have gradually abandoned their production [2]. This situation was reversed after Achates Powers presented a three-cylinder engine with a capacity of 2.7 dm³, dedicated to delivery vans. This engine was presented for the first time at the Detroit show in 2015. Its mass production is to start in 2024 [3].

The engine with opposite pistons has many advantages compared to classic engines. The most important are: the combustion chamber is limited by two heads of movable pistons (reciprocating), which does not require the use of a head, often characterized with a complicated structure. These engines do not require an advanced valve mechanism, which translates into no loss of power necessary for its drive. The positioning of the pistons, connecting rods and crankshafts (a crank-piston mechanism) opposite each other, helps to balance the engine. On the other hand, significant disadvantages include
the need to mechanically connect two crankshafts, no space for any placement of fuel injectors inside the combustion chamber (the need for tangential mounting of the injector to the cylinder sleeve) [4, 5]. The downsizing of internal combustion engines has developed rapidly in recent years. This concept mainly refers to increasing the power generated by the drive unit for its total mass minimized to a minimum [6, 7]. For this purpose, many scientific institutions, design offices and enterprises have started developing innovative control, power, cooling systems or flue gas treatment plants. Apart from system modifications, new materials for engine parts are implemented, their geometry and mechanical, thermal and fatigue strength are improved [8, 9, 10]. However, to accelerate the optimization process and reduce the number of experiments or prototypes, the numerical modelling technique is commonly used [11]. This technique uses computer units with dedicated software to integrate the design and prototyping phase with manufacturing. They allow for solving the engine problems related to thermodynamics, fluid mechanics and multi-object dynamics. The authors used such tools to design an opposed pistons engine. The parameters and characteristics of the vibration signal generated by the internal combustion engine are increasingly used to assess the technical condition and operating conditions (functional and operational parameters) of the engine [12, 13, 14]. Difficulties are encountered in identifying and inferring the technical or operating condition of an engine part or component based on commonly used parameters and characteristics of the vibroacoustic signal. The result, among others, is from the complex character of the vibrations generated by the engine, processes occurring inside it, the coexistence of many vibration sources and complex kinematics [15, 16]. This paper presents the concept of diagnosing the designed internal combustion engine with the use of frequency analysis of free vibrations.

The CATIA v5 software was used in the first stage of designing an opposed-piston engine. In this software, solid models of individual parts of the assembly of the crank-piston system were developed and they were fully assembled (taking individual degrees of freedom from selected parts and assigning mass). Then, using the one-dimensional AVL Boost software, the working processes in the engine were simulated. A similar process of engine simulation tests is described in [17] and [18]. Then, based on the data obtained in the AVL Boost program, the values of the forces acting on the piston crowns, which resulted from the engine operation process, were entered into the MSC Adams software. Thus, it was possible to determine the course of dynamic forces resulting from the operation of the engine based on the model developed at MSC Adams. This allowed for determining the vibration level of the analysed engine.

The engine vibration level was analysed for a reduced model in MSC Adams. In this model, other components such as the mass of the block, the mass of units mounted on the engine, screws, gears were replaced by concentrated mass (Figure 3). The value of the concentrated mass and the position of the centre of gravity were determined in the CATIA software based on the model shown in Figure 1. The main aim of the research was the preliminary selection of the susceptible elements of the engine supports.

2. Computational model

The model of the PZL-100 opposed-piston engine was developed with the use of the Catia V5 software in cooperation with WSK "PZL-KALISZ" S.A. aircraft engine manufacturer. It is a 100 kW diesel engine intended for the propulsion of light aircraft (Figure 1). The model of the crank-piston system of the engine is presented in Figure 2. The model consists of the following components:

- two crankshafts,
- six connecting rods with feet and connecting rod bolts,
- six pistons with piston pins,
- three cylinders.
Figure 1. PZL-100 counter-rotating engine model in the CATIA v5 software.

Figure 2. Model of the engine crank-piston system in the CATIA v5 software.

The developed geometry for the assembly of the crank-piston system was exported from the CATIA environment in the STEP format. The models were then converted to the parasolid format using the conversion software. The intermediate process was performed due to the lack of a uniform file format between the CATIA and MS Adams software. The prepared geometry was imported to the MSC Adams environment using the Adams View module (Figure 3). This module enables simulation tests of both static and dynamic systems. It allows solving the problems of kinematics, statics and dynamics of mechanisms.
As a result of the simulation tests carried out with the use of AVL Boost software, the course of changes in the value of the working pressure for cylinders was generated (Figure 4). The engine operating conditions for the working pressure constituted the full engine load for the rotational speed of the crankshaft 4,000 rpm. The waveforms were imported into the Adams View software in numerical form.

The scope of work also included the extension of the piston force course to more than 120 cycles, which makes it possible to carry out simulation tests up to a time of values not exceeding 1.5 s. The simulation tests of engine vibrations were carried out to prepare spectral characteristics in the frequency domain for different values of the crankshaft lead angle and different stiffness coefficients of the flexible elements of the supports.

In addition, the model was equipped with weightless shafts located in the front of the engine, i.e. along the axis of the crankshafts (Figure 5). Between each of the additional shafts located in a short distance from the front of the crankshaft, there is a “torsion spring” connection. It is a connection simulating a flat spring, which enables the registration of the torque waveform between two elements in rotation. For flat springs, the stiffness coefficient was assumed at the level of $k = 7 \times 10^7$ Nmm/deg to obtain no deformability effect, while the damping was omitted. The plane spring was used only to measure the value and course of the torque generated on individual drive shafts (the MSC ADAMS software allows measurement of torque only between kinematic ties, while the lack of damping and high stiffness does not affect the behaviour of the model).

In order to obtain the correct registration of the torque curve, the additional shafts rotated at a constant rotational speed of $n = 4,000$ rpm, and the crankshafts moved at the speed resulting from the applied piston forces. Due to software limitations, the real speed of the crankshafts was measured on a kinematic constraint (flat spring) between the crankshaft and the auxiliary shaft.
Models of engine supports (Figure 5) were also used for the simulation tests. These models are made of metal-rubber sleeves, inclined towards the centre of gravity of the engine (the axes of symmetry of all four vibro-isolators run through the centre of gravity of the engine).

The model of the crank-piston system with the cylinders, the concentrated mass (the mass located in the centre of gravity with the sum of the masses of the block and aggregates installed on the engine), suspension, were simulated for the crankshaft lead angle of 14 and 20° using the MSC ADAMS software (Figure 6). The performed numerical analysis was aimed at determining the influence of engine operation on the number of vibrations transferred to the aircraft structure in the places where the engine is supported.

Parameters for the supports were assumed for the tests:
- axial stiffness coefficient $k = 890 \text{ N/mm}$,
- axial damping coefficient $c = 100 \text{ Ns/mm}$,
• angular stiffness coefficient $k_\alpha = 97 \text{ Nmm/rev}$
• angular damping coefficient $c_\alpha = 100 \text{ Nmm s/rev}$.

The values of the characteristic parameters were selected based on the catalog data of the J-7401-27 engine cushions [19]. In turn, for the second series of tests, the parameters of the J-3049-67 vibroisolators were assumed, characterized by greater (subsequent model from the catalog [20]) stiffness:
• $k = 1930 \text{ Ns/mm}$,
• $k_\alpha = 240 \text{ Nmm/rev}$,
• $c = 100 \text{ Ns/mm}$,
• $c_\alpha = 100 \text{ Nmm s/rev}$.

The tests were carried out in two stages: for the lead angle of the crankshafts 14 and 20 cad left in relation to the right.

The flattest characteristic of the function was chosen as the criterion of the spectral analysis, following the general principle of frequency function analysis (Figure 7). If the measurement and analysis of the vibrations of the object concern some unknown frequency range and if they are not imposed in advance, e.g. by measuring a specific parameter such as displacement, velocity or acceleration, the general rule is to measure the quantity that has the flattest characteristic as a function of frequency. This allows the measurements to cover the largest dynamic range of the tested system. However, if the character is unknown, the vibration velocity parameter should be used [21]. Therefore, the results of the simulation analysis are presented as a function of the frequency characteristics for the speed in the engine supports, i.e. in the place where mechanical vibrations are transmitted to the aircraft structure.

![Figure 7](image)

**Figure 7.** Selection of the parameter of the measured vibrations due to the course of the spectral characteristics [21]:
- a) displacement, b) velocity, c) acceleration.

### 3. Results

Figure 8 shows the course of changes in the torque generated in the engine crankshafts. Figure 9 through Figure 14 show the course of changes in load, speed and acceleration in the supports of the designed engine, for the lead angle of the shafts of 14 cad and flexible elements with lower stiffness. Figures 9 to 14 show the results of the frequency spectrum analysis for the speed of one of the engine supports in the variants for two lead angles of the crankshafts and two materials of the flexible elements of the engine supports.

The torque generated during engine operation for the left and right shafts (Figure 8) adds up to an average value of 325 Nm. This summation is made possible by the kinetic coupling of the crankshafts. On the other hand, the linear velocity in the kinematic links of the individual supports varies cyclically. This velocity in the "X" axis (Figure 9) varies from -48 to +95 mm/s, obtaining the average values for left supports of 27 mm/s, and right supports of 24 mm/s. In the "Y" axis (Figure 10) the average velocity
for all supports does not exceed the value of 0.1 mm/s. In the "Z" axis (Figure 11), the average velocity value for all supports does not exceed the value of 0.3 mm/s.

Variable velocity in the kinematic nodes of the engine supports also translates into a change in acceleration. In the 'X' axis, the value of the acceleration of the supports varies from -25 to +25 m/s² (Figure 12), where the average values oscillate at the level of 12-14 m/s². In the "Y" axis, the average value for all supports is 2 m/s² (Figure 13), while in the "Z" axis, the value does not exceed 1.5 m/s² (Figure 14).

![Figure 8](image8.png)

**Figure 8.** The course of the torque value for the left and right shafts and total torque for both crankshafts, for lead angle 14 cad.

![Figure 9](image9.png)

**Figure 9.** The course of speed changes in the motor supports in the vertical axis "X", for the coefficients k = 890 N/mm and c = 50 Ns/mm.

![Figure 10](image10.png)

**Figure 10.** The course of speed changes in the motor supports in the vertical axis "Y", for the coefficients k = 890 N/mm and c = 50 Ns/mm.
Figure 11. The course of speed changes in the motor supports in the vertical axis "Z", for the coefficients $k = 890$ N/mm and $c = 50$ Ns/mm.

Figure 12. The course of acceleration in the motor supports in the vertical axis "X", for the coefficients $k = 890$ N/mm and $c = 50$ Ns/mm.

Figure 13. The course of acceleration in the motor supports in the vertical axis "Y", for the coefficients $k = 890$ N/mm and $c = 50$ Ns/mm.

Figure 14. The course of acceleration in the motor supports in the vertical axis "Z", for the coefficients $k = 890$ N/mm and $c = 50$ Ns/mm.
4. Conclusion
During the simulation tests of vibrations of the opposed-piston engine mounted in oblique supports, spectral characteristics were generated in the frequency domain. The model of a crank-piston system with cylinders and a concentrated mass imitating the mass of the engine block and other aggregates was made. The engine was mounted in oblique supports in such a way that the symmetry axes of the metal-rubber sleeves coincide in the centre of gravity of the drive unit. The model was loaded with piston forces corresponding to the maximum power, the thrust from the propeller, and the force of gravity. The characteristic coefficients of the flexible elements of the engine supports were selected based on the available data from the catalogues. The simulation tests were carried out until 123 engine operating cycles were achieved, which gives a simulation time of approx. 1.5 s. The comparison of the vibration spectrum of the engine with counter-rotating pistons depending on the stiffness coefficient of the supports for the crankshaft leads angle 14 cad is shown in Figure 15.

Figure 15. Comparison of the vibration spectrum depending on the stiffness coefficient of the supports for the 14 cad crankshaft lead angle.

On the basis of the analysis of the results from simulation tests using the MSC ADAMS software, the following conclusions can be drawn:

- engine vibrations occur for times the frequency of approx. 70 Hz, which corresponds to the rotational speed of the crankshaft 4,200 and its times, i.e. 8,400, 12,600, 16,800, 21,000 rpm,
- the highest level of vibrations occurs at the rotational speed of 8,400 rpm, i.e. at the frequency equal to 140 Hz,
- increasing the lead angle of the crankshaft from 14 to 20 cad increases the vibration level by an average of 230%,
- increasing the support stiffness coefficient (selecting "harder" engine cushions) increases the level (regardless of the crankshaft lead angle) of engine vibrations [22],
- the highest spectral speed of the engine supports does not exceed the value of 10 mm/s, the vibration level is in the typical range for an internal combustion engine, i.e. 5-13 mm/s,
- two-piece mounts J-7401-27 were correctly selected for configuration with the opposed piston design engine.

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