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Investigation of the piston vibration modes of high-performance 2.2 L gasoline engine under different skirt lengths

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Abstract This paper aims to perform modal and harmonic response analyses to show how the piston skirt length reacts. The studied aluminium piston was generated in CATIA CAD Software and consequentially this was simulated in ANSYS software using modal and harmonic response tools. The piston finite element model was built to predict the basic modal parameters such as: natural frequencies, vibration modes and deformations. Different grid sensitivity tests have been done to improve the accuracy of the piston model. The piston with larger skirt has shown 77% higher vibration deformations than piston with smaller skirt. The proposed methodology can be easily used by a design engineer to perform dynamic behaviour studies of moving components and assemblies in Internal Combustion engines and not only.

Keywords: piston skirt, CAD, FEM, modal analysis, natural frequencies, deformations, vibration modes.

1. Introduction

The combination of electrical and Internal Combustion (IC) engine known as hybridization has become the new standard on automotive engines. However, understanding the benefits of hybridization is still insufficient due to complex interaction of mechanical and electric propulsion systems [1, 2]. This requires the improvement of engine components, which will play a key role in the Noise, Vibration and Harshness (NVH) behavior. Thus, this has led the engine designers to new smaller turbo engines, where the thermal and mechanical stresses have great influence on the engine components, such as pistons, piston rings and cylinder bores [3]. An understanding of piston dynamics is a fundamental part of the development process to improve the generated friction, noise and vibration. Therefore, it is necessary to create advanced simulation models for the optimization of the piston design.

In the topic of piston-bore tribology, the current literature has enough results including piston friction, thermo-elastic distortion and elastohydrodynamic lubrication. In 1983, Furuhama et al. [4] presented some interesting experimental results regarding the piston friction comparing a small gasoline engine and a diesel engine. Improving the floating liner method, they found that the engine size has a great impact on piston friction. To understand the piston tribological performance more emphasis should be given in fired conditions. Mufti et al. [5] studied the piston assembly friction using the indicated mean effective pressure method. Their measurements were validated by a simple and a more realistic approach giving remarkable results. Further works were also developed in order to examine complex physical phenomena into piston-bore contact. Li et al. [6], Mazouzi et al. [7] and Littlefair et al. [8] solved mixed-elastohydrodynamic conditions considering piston skirt structure and piston's dynamic motion. At the same time, with the increasing in-cylinder pressures through downsizing, the contribution of piston slap is also studied numerically and experimentally. Geng et al. [9] compared numerical predictions of the complex piston dynamics with experimental measurements in a diesel engine. In particular, they measured piston slap using three accelerometers along the cylinder showing good agreement with numerical predictions. Teraguchi et al. [10] described the effect of the oil supply on piston slap using a test engine. They tested some oil quantities under idle engine conditions, and they found that for oil supply of 6 ml/min, there is a reduction of 50% to cylinder vibration. More recently, Dolatabadi et al. [11], Zavos et al. [12], and Korkos et al. [13] captured experimental data regarding generated noise and vibrations in two motorcycle engines. They proposed that the operating frequencies of piston slap can be varied between 450 Hz and 3500 Hz using wavelet analysis. With extremely thin lubricant film at the dead centers, it was found that piston slap accounts for a large proportion of the noise in a full engine cycle. Thus, the contribution of the piston crown and the piston skirt geometry are critical [14, 15].

Lightweight pistons with smaller piston skirt area are frequently used in high-performance engines. The modal density of this piston is high and, therefore, it is necessary to investigate the modal parameters during the design. When these vibration modes are equal to the natural frequency of the piston, resonance can occur. As a results, this can lead to structural deformation or damage and, then to abnormal noise or vibration. Simulation models and experimental studies try to predict the continuous dynamic characteristics of the piston such as resonance, fatigue, and other harmful factors of forced vibration. Limited modal and harmonic analyses can be found in the literature including the effect of the piston skirt structure. Kuppuraj et al. [16] provided some numerical results regarding the piston modal behavior using two different materials. They showed that the piston with SiC reinforced ZrB2 composite material has more stiffness than aluminum piston. In the same field, Zheng et al. [17] used a real piston geometry for modal analysis. It was found that the natural frequency of the piston is in the range of 6,000–10,000 Hz, and the piston skirt deformation is higher as the piston is twisted in y-axis. A different approach was also considered by Zheng et al. [18], who employed a harmonic analysis to obtain the vibration modes in a titanium alloy piston taking into account the maximum combustion load. It was clearly observed that the skirt has large deformation, and its design has a dominant role in engine vibration.

Based on the literature survey, it is evident that existing research primarily focuses on investigating tribological parameters such as friction and film thickness, as well as fault diagnosis of pistons. However, limited attention has been given to the effect of skirt geometry on dynamic behavior during the design process. This paper aims to address this gap by developing a design procedure for modal and harmonic piston analysis. The study considers the impact of piston structure using input data from a high-performance 2.2L gasoline engine. Key aspects covered in this paper include: (i) detailed design of piston geometry, (ii) assessment of skirt length influence on vibration modes through harmonic analysis, and (iii) comparison of skirt deformations across different lengths. The methodology offers the advantage of integrating a CAD model with Finite Element analysis, providing designers with a comprehensive tool for in-depth dynamic analyses of various mechanisms or components in Internal Combustion (IC) engines.

2. Methodology

2.1. Modal and harmonic response theory

To demonstrate the modal parameters in the piston structure, a finite element piston model was developed in ANSYS Workbench for this study. Modal analysis can represent the natural frequencies and mode shapes; therefore, the structural deformations can be obtained. Additionally, to avoid resonant vibrations, the prediction of the modal parameters is significant for the design process. The vibration equation of the piston structure is:

$$(\mathbf{K} - \omega_i^2 \mathbf{M})\boldsymbol{\varphi}_i = 0, \tag{1}$$

where $\mathbf{K}$ is the stiffness matrix, $\mathbf{M}$ is the mass matrix, ω_i is the natural frequency and $\boldsymbol{\varphi}_i$ is ith vibration mode. Regarding the steady-state response of the piston structure under the dynamic load, a harmonic analysis has been done. The motion of the structure under simple harmonic load is given as:

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{f}\sin(\theta t), \tag{2}$$

where $\mathbf{x}$ is the displacement response, $\mathbf{f}$ is the amplitude vector of the sinusoidal load and θ is the excitation frequency. Considering the natural frequencies from the modal analysis and the dynamic load of the combustion pressure, the frequency response curve was obtained using the method of the mode superposition. This can predict the frequency response curve of the piston structure and find how the skirt profile can react due to the external excitation.

2.2. Engine data and piston design

The high-performance engine presented here is a 2.2 L four-stroke gasoline engine. Every component of the engine was designed and introduced to CATIA V5R21 environment. Figure 1 shows the full design of this engine, and Table 1 lists the basic specifications. Basic design drawings for the studied piston are illustrated in Figure 2. The reference piston has flat crown shape, and its material is aluminum alloy. All dimensions of the piston are included in Table 2.

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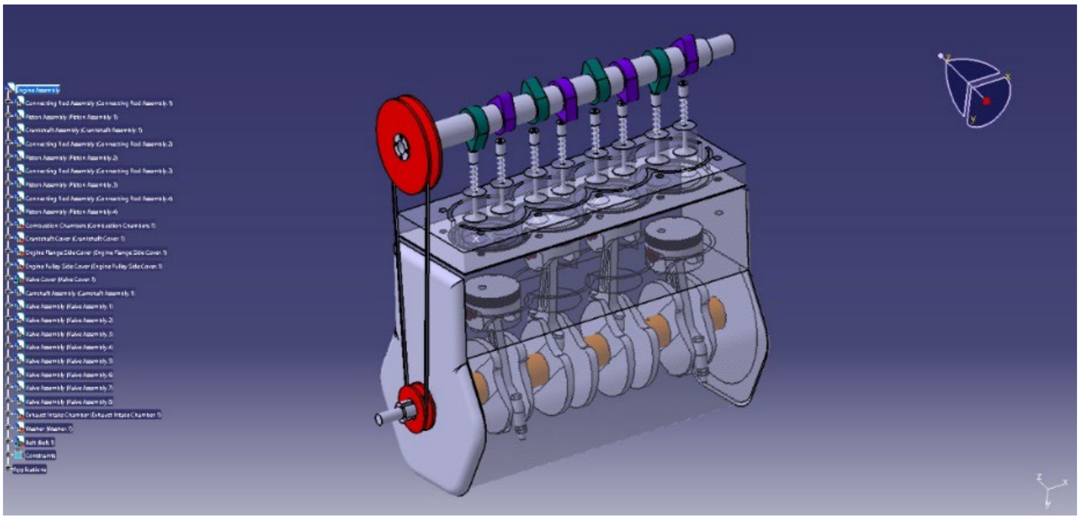

Figure 1. Engine assembly in CATIA CAD Software.

Table 1. Engine specifications.

Parameter	Value	Unit
Volume	2.2	L
Compression ratio	9:1	–
Bore × Stroke	87 × 84	mm × mm
Rod length	149.65	mm
Idle speed	2000	rpm
Max HP	237 at 7800 rpm	HP

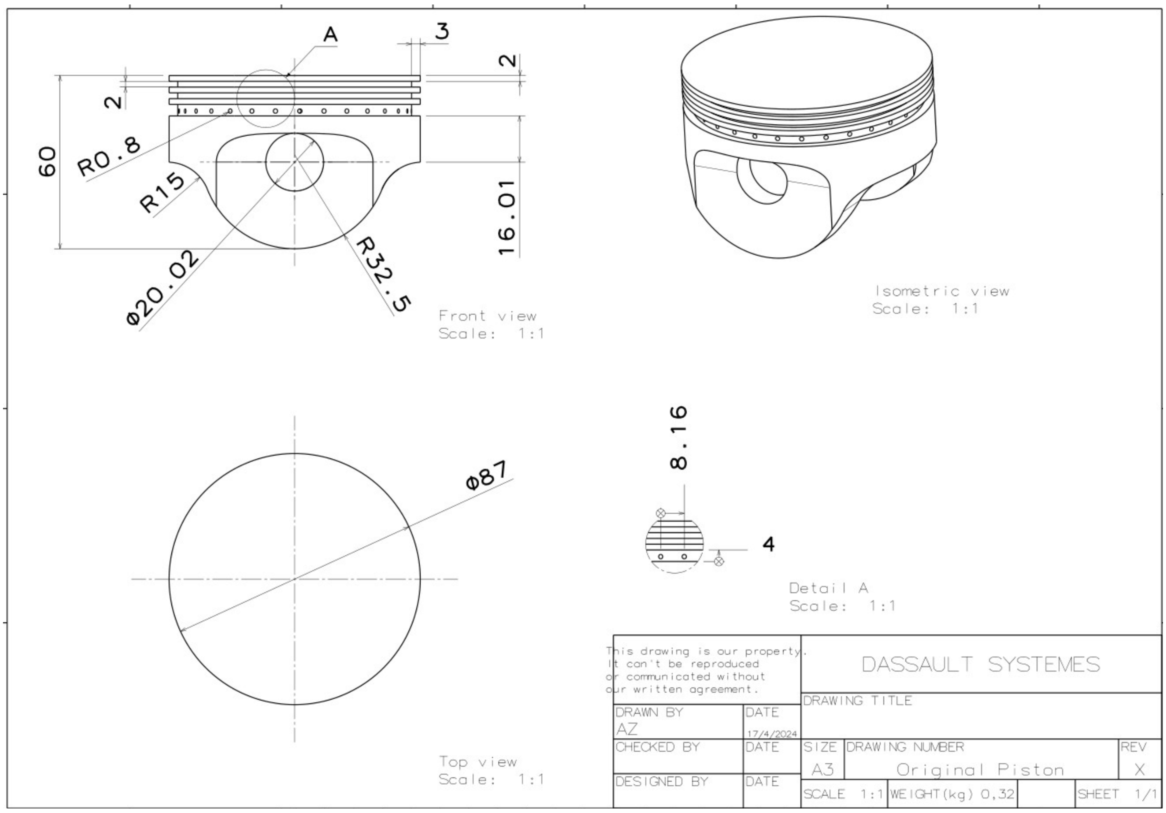

Figure 2. Design drawings of the original piston.

Table 2. Original piston dimensions and material data.

Parameter	Value	Unit
Elastic modulus	70	GPa
Piston density	2770	kg/m^3
Piston diameter	87	mm
Piston length	60	mm
Piston crown height	2	mm
Pin diameter	20.02	mm
Skirt contact length	16.01	mm

Nowadays, there are many techniques for improving structural integrity, noise and vibration on piston structure. Two main design factors are piston crown and skirt (see Figure 3). Initially, the piston crown has a dominant role on generated thermo-mechanical stresses owing to the combustion pressure and the heat transfer. Additionally, the skirt profile has a critical factor owing to the surface area of the piston that comes into contact with the cylinder wall, where the operational piston stability can be changed rapidly leading to abnormal noise and vibration. Thus, the main goal is to have just enough skirt-to-cylinder contact to stabilize the piston, which reduces surface area and friction. For this reason, the effect of the skirt profile on modal parameters and generated deformations through harmonic analysis was examined and discussed in this study. The influence of the piston crown geometry will be investigated in a future work.

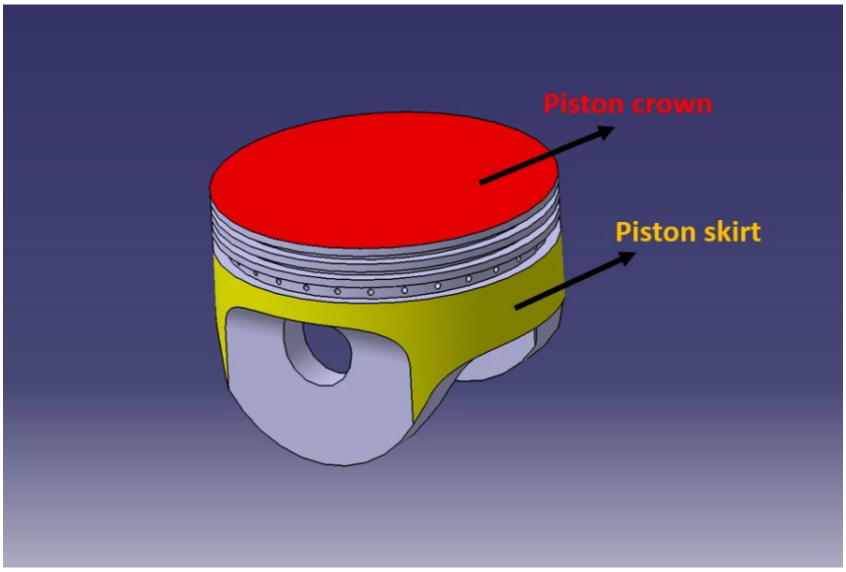

Figure 3. Piston critical design factors.

2.3. Finite element model of the piston

Figure 4 shows the finite element piston model in ANSYS Workbench. The piston model was meshed using second-order tetrahedral elements SOLID187. SOLID187 is a 3D 10-node high-order tetrahedral element that offers improved accuracy and convergence compared to lower-order tetrahedral elements. Regarding the input boundary conditions in the modal analysis, there is a fixed support on the inner surface of the piston pin hole. On the other hand, for the harmonic response analysis, the specified tool (ANSYS Workbench) was also used, where the maximum in-cylinder pressure of 12.2 MPa was applied to the top of the piston crown at an engine speed of 3,500 rpm, as per manufacturer data.

Through grid sensitivity tests, the appropriate element mesh size was selected in order to improve the simulation accuracy. Figure 5 illustrates the mesh sensitivity tests conducted for key performance parameters, including natural frequencies. Following convergence tests, an element size of 1.5 mm was selected for the original piston, deemed suitable for the intended calculations.

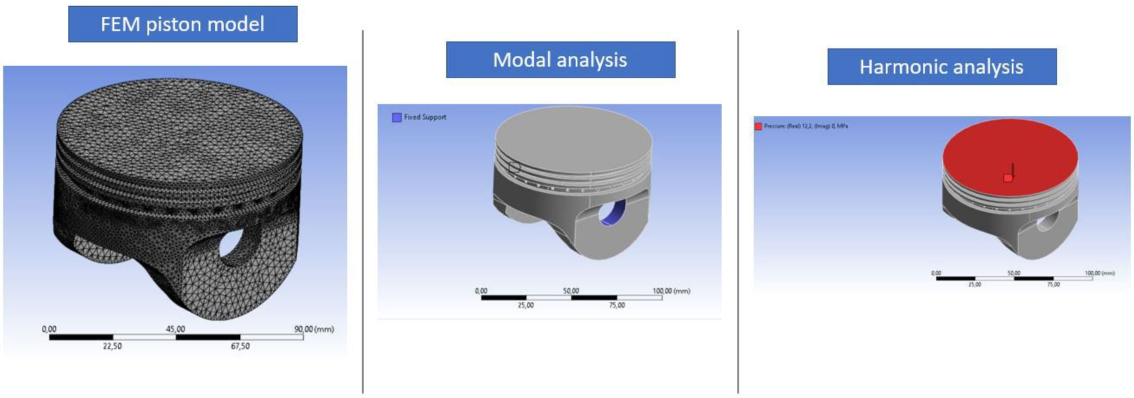

Figure 4. FEM piston model: modal and harmonic analysis boundary conditions.

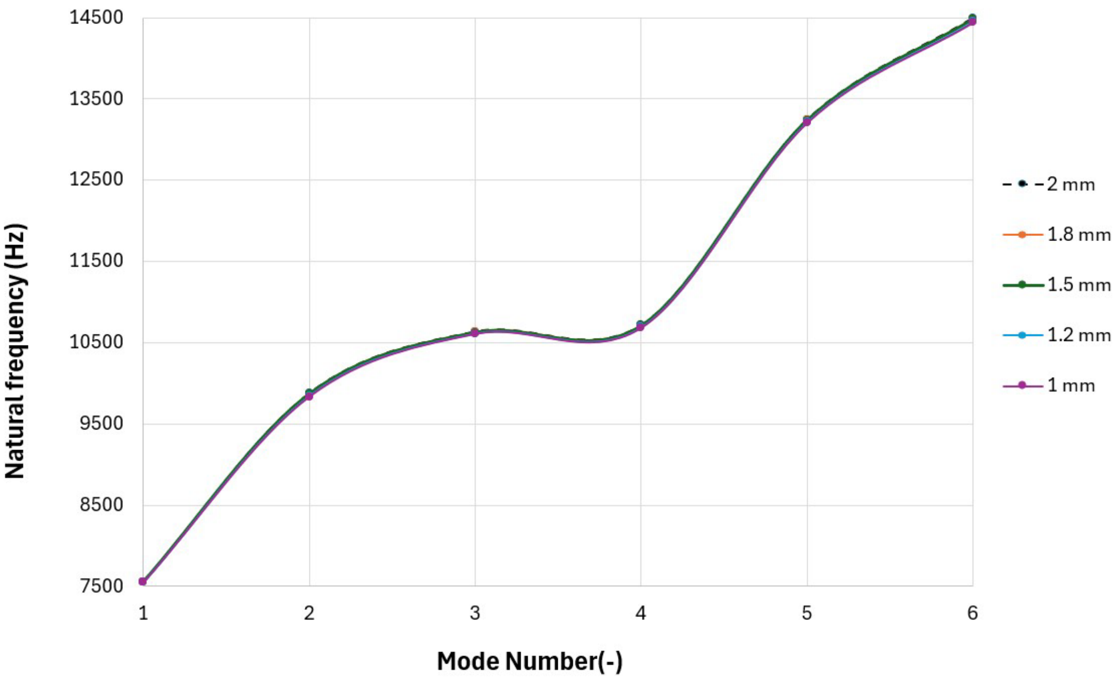

Figure 5. Mesh sensitivity tests for original piston (Type A).

3. Results and discussion

The numerical results reported here correspond to the original piston of a high-performance 2.2 L gasoline engine. The dynamic characteristics of the piston structure were obtained through modal and harmonic studies. To determine the steady-state response of the piston under the dynamic combustion load, the harmonic response analysis was carried out at medium driving speed of 3,500 rpm applying the maximum pressure load of 12.2 MPa. This engine speed was selected according to the world-wide harmonized Light vehicles Test Cycle (WLTC) [19]. To investigate the piston skirt geometry, two different pistons were designed, and the corresponding piston skirt deformations were compared related to the original case.

3.1. Natural frequencies and vibration modes

Figure 6 shows the natural frequencies of the original piston (Type A). It is obvious that the natural frequencies are high in the range of 7,000–15,000 Hz. The first natural frequency is 7,555.2 Hz with the maximum deformation at the top land of the piston crown, as well as the other natural frequencies showed larger deformations in the piston skirt. Therefore, this reflects the critical effect of the piston skirt design in vibration deformations and noise generation.

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To correlate the piston skirt design and vibration modes, harmonic analysis was used in ANSYS Workbench tool. Using the method of mode superposition, the variation of natural frequencies from modal analysis and the corresponding combustion load were applied in the set-up tool. In the original piston (Type A), the following conditions were used: (i) the frequency interval was set to 7,000–15,000 Hz, (ii) a combustion load of 12.2 MPa was used and (iii) the solution interval was set to 80 [18].

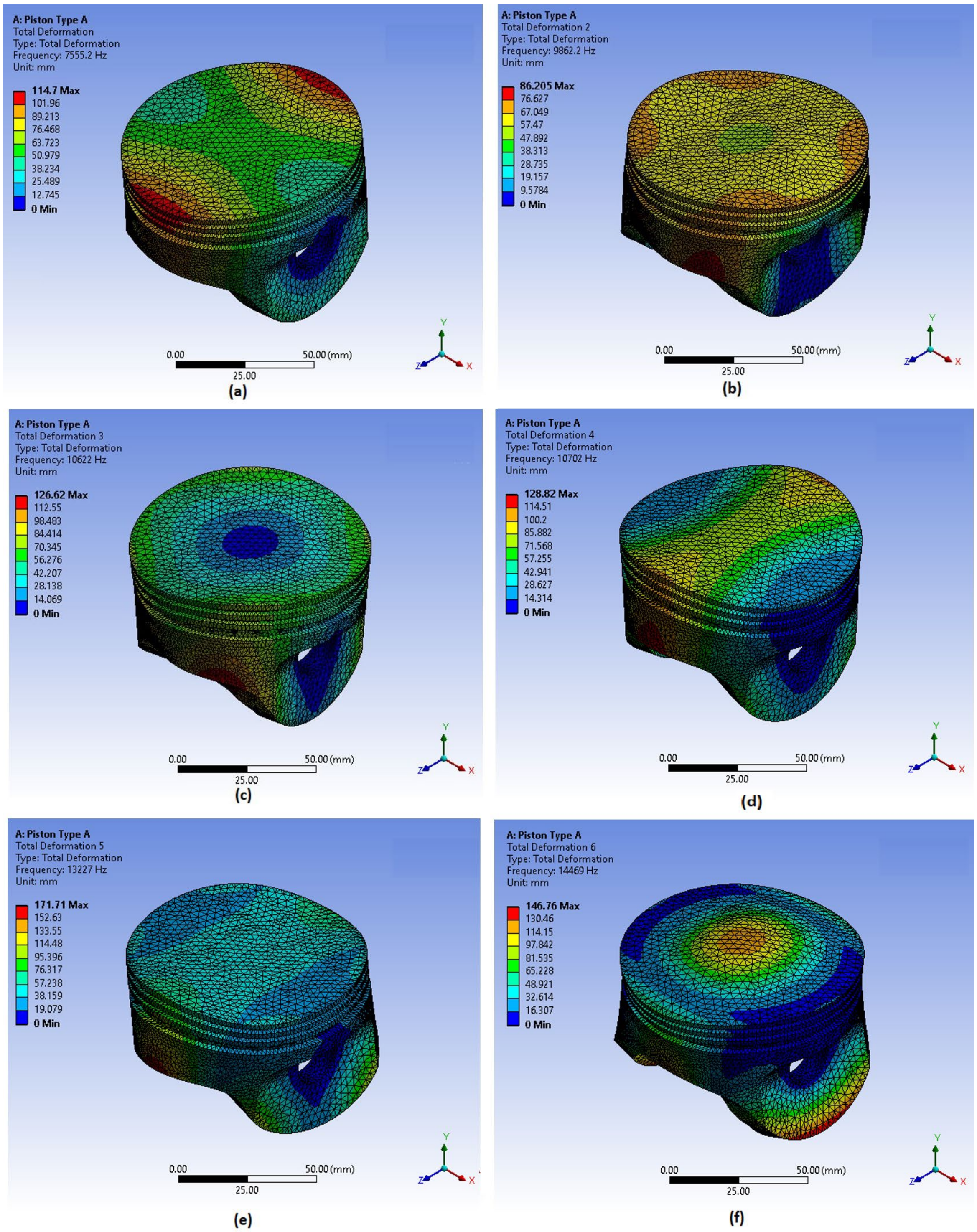

Figure 6. Natural frequencies of the original piston model (Type A).

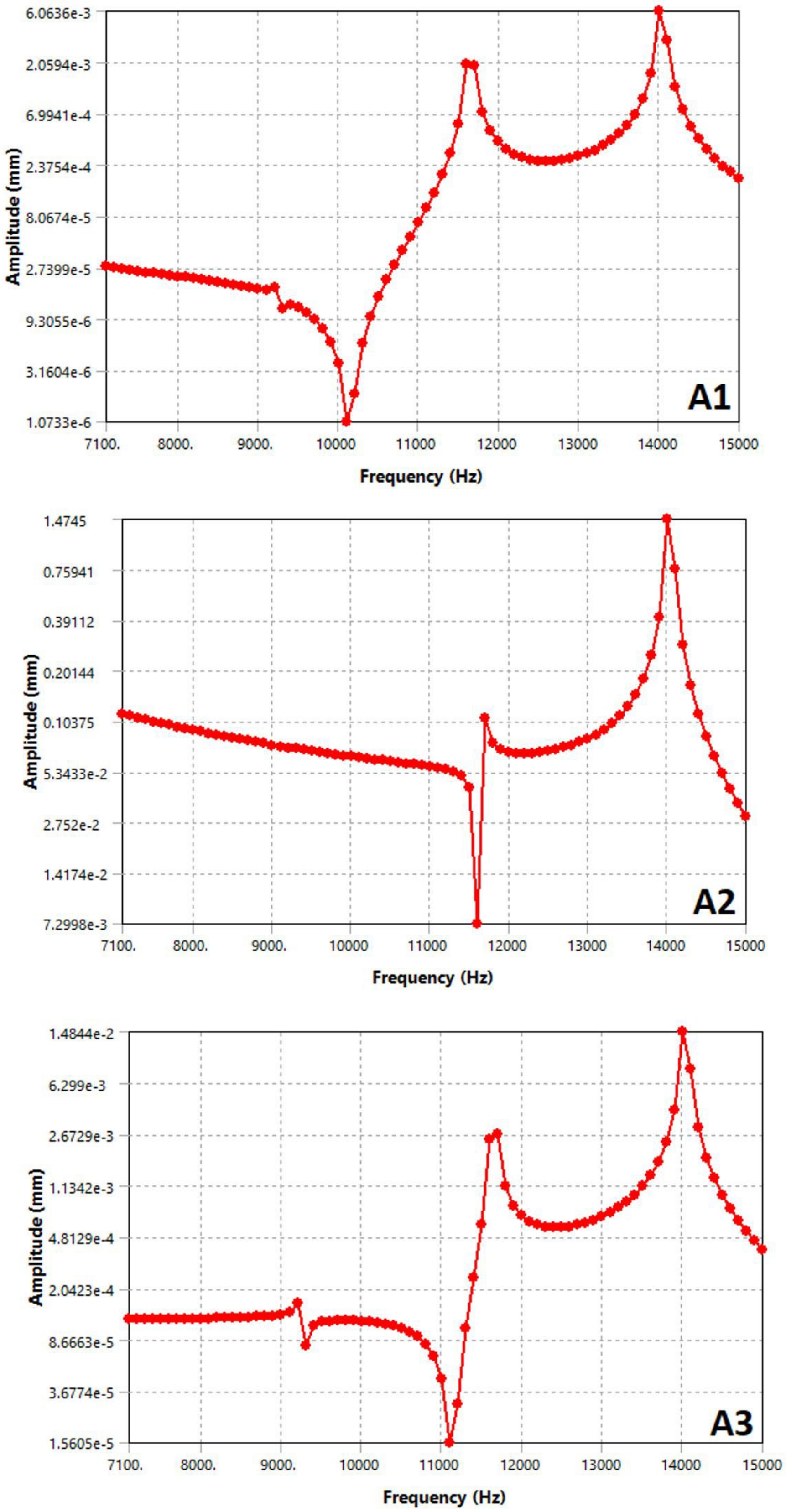

Figure 7. Deformation frequency response curves of the original piston skirt (Type A) in *x*-axis (A1), *y*-axis (A2) and *z*-axis (A3).

Figure 7 shows the curves of the piston skirt deformation frequency of the original piston skirt (Type A). As observed, the skirt deformation amplitude reaches the maximum at 14,000 Hz, where resonance can occur. In more details, the maximum vibration deformations are: 0.006 mm in *x*-axis (Figure 7A1), 1.47 mm in *y*-axis (Figure 7A2) and 0.014 mm in *z*-axis (Figure 7A3). The results showed several small fluctuations with an upward behaviour and, after the maximum value there is a fast reduction. The same trend is seen in the extracted results from a diesel engine [18]. Again, higher displacement values were observed at the piston skirt when the skirt twisted in *y*-axis with the value of 8.77 mm near to 12,100 Hz. The reason for this is that the simulations were taken for a high-performance diesel engine with a combustion pressure of 22 MPa. As a result, to improve the stability of the piston, the piston skirt should avoid these maximum frequencies and these deformations for both cases.

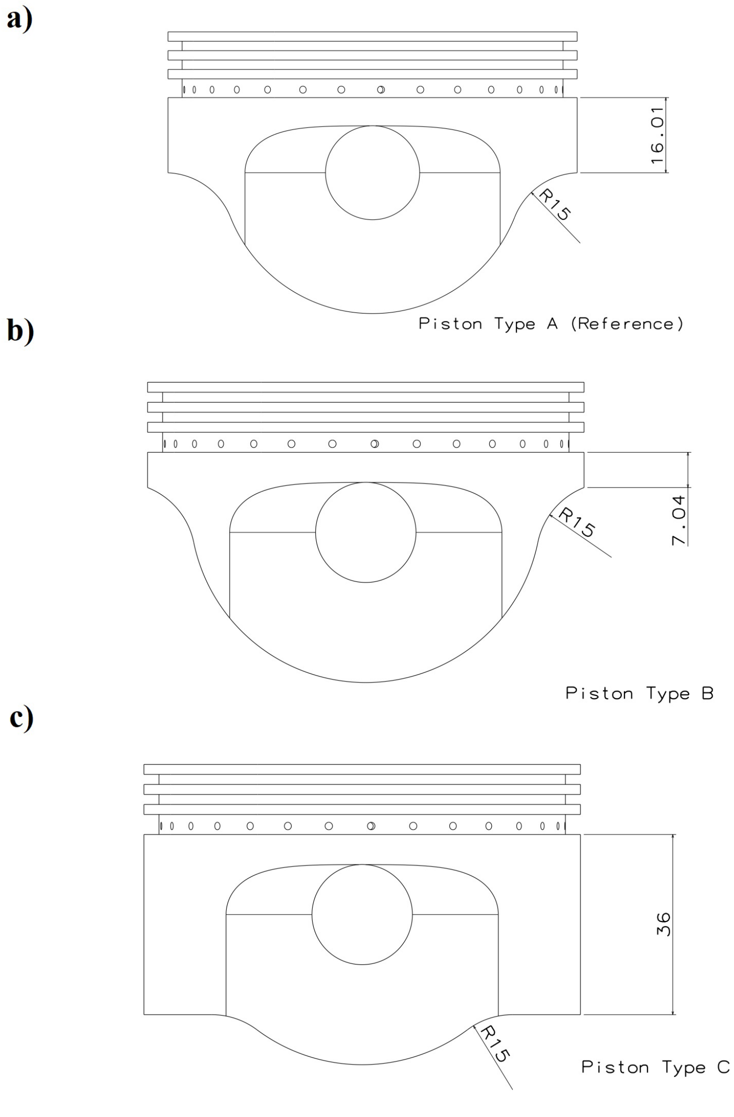

Figure 8. Design of the piston skirt: a) Type A (original), b) Type B, c) Type C. The main dimensions are in mm.

3.2. Effect of piston skirt structure

To study the effect of the piston skirt, it is highly required to correlate skirt geometry with vibration deformations. In this analysis, there are two geometries considered for piston skirt. Figure 8 presents the designs of two different skirt structures. The piston skirt type B has length of 7.04 mm (smaller length from the original piston) and the skirt type C has length of 36 mm (larger length from the original piston). Through suitable grid convergence tests, the natural frequencies of the piston skirt B and C are also presented in Figure 9. In each case, the tetrahedral type was selected. Regarding the modal analysis, the frequency range has shown different values for each examined piston skirt. It is obvious that the piston skirt type C has lower natural frequencies due to larger mass (0.38 g) than other two types of pistons (0.32–0.33 g). Using the appropriate frequency range, the curves of the displacement frequency response are also determined through harmonic analysis.

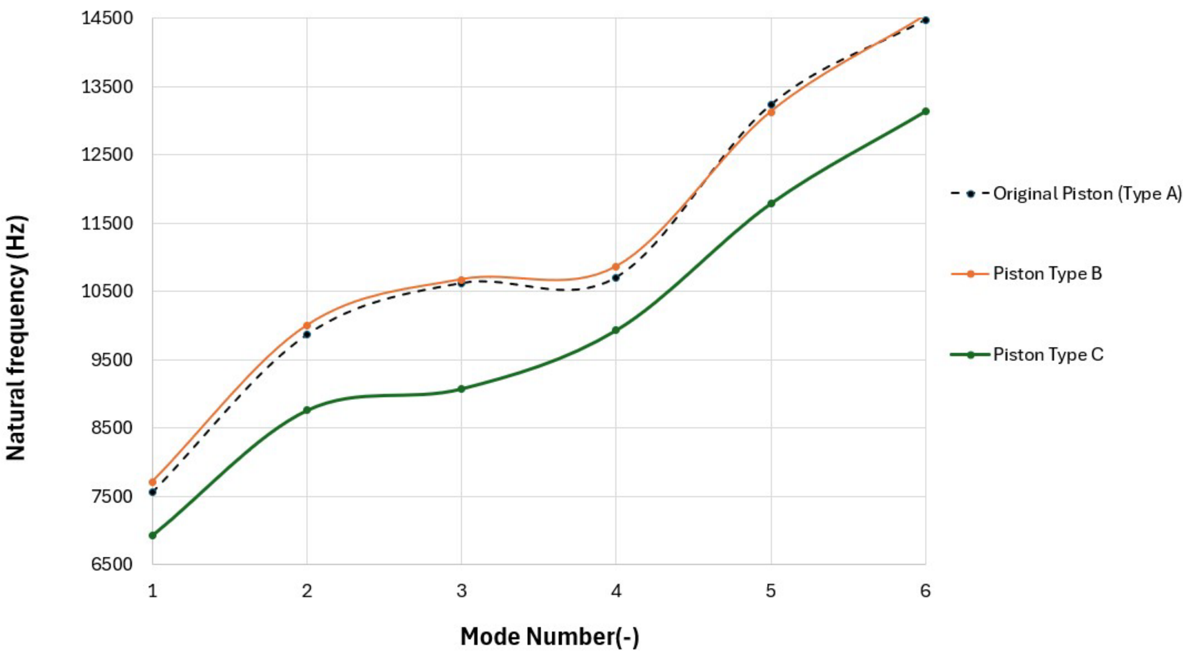

Figure 9. Comparison of natural frequencies for the piston Types A, B, and C.

Figure 10 shows the deformation frequency response curves for all piston cases. The original piston and piston type B have shown same fluctuations in *x, y* and *z*-axis respectively. The piston type B with smaller skirt reaches the maximum deformation at 14,500 Hz with stronger fluctuations. In particular, the maximum vibration deformations are: 0.0065 mm in *x*-axis (Figure 10B1), 3.83 mm in *y*-axis (Figure 10B2) and 0.036 mm in *z*-axis (Figure 10B3). This means that the piston type B has lower stability with higher displacements than original skirt. On the other, the piston type C with higher length provided lower fluctuations but higher vibration deformations in *x, y* and *z*-axis accordingly. The maximum deformations occur at 13,200 Hz with values of 0.015 mm in *x*-axis (Figure 10C1), 6.35 mm in *y*-axis (Figure 10C2) and 0.056 mm in *z*-axis (Figure 10C3). This result means that the piston skirt type C has more stability but larger vibration deformations than other two types of pistons due to small fluctuations in the frequency range. In the case of the current analysis, the contribution of the skirt length to vibration deformations has mainly increased to 77%, which would result in increased noise. This is a crucial finding in the current investigation. Therefore, the multi-objective optimization of the skirt design needs to be introduced in order to improve piston performance without increasing the mass.

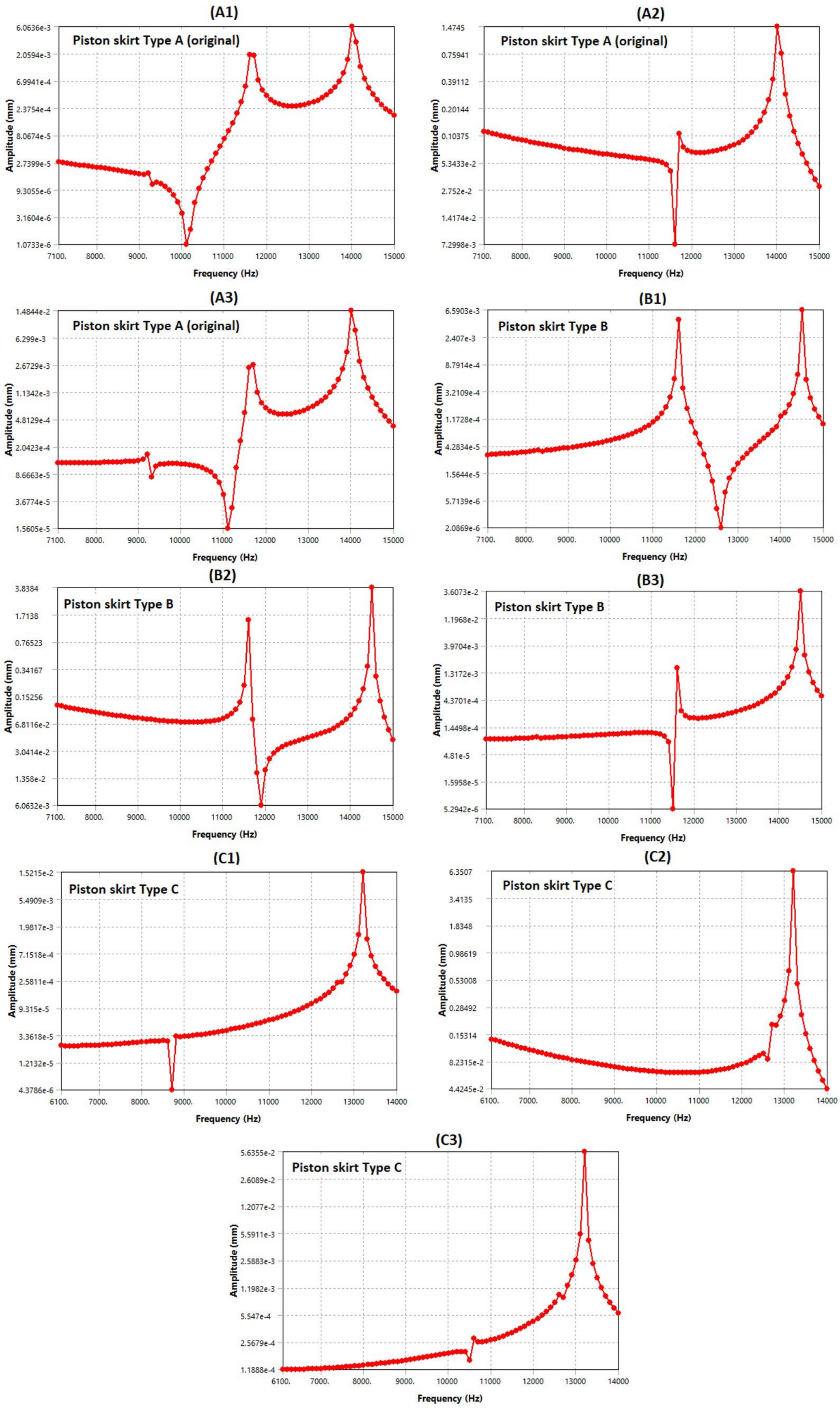

Figure 10. Deformation frequency response curves of (i) the original piston skirt (Type A) in *x*-axis (A1), *y*-axis (A2) and *z*-axis (A3), (ii) the piston skirt (Type B) in *x*-axis (B1), *y*-axis (B2) and *z*-axis (B3) and (iii) the piston skirt (Type C) in *x*-axis (C1), *y*-axis (C2) and *z*-axis (C3).

4. Conclusions

In this work, a piston finite element model of a high-performance 2.2 L engine has been developed and analysed using modal and harmonic response tools. The paper investigates the effect of the skirt length using three different pistons. The pistons were created in CATIA CAD Software, and the simulations were made in ANSYS Workbench. The natural frequencies, the vibration modes and the skirt deformations are determined and compared.

The numerical results showed that the piston has high range of natural frequencies. These values can be varied between 6,000–15,000 Hz. Regarding the skirt length, the displacement frequency curves have shown different behaviour through harmonic response analysis. The piston with a larger skirt offers improved stability, albeit with higher vibration deformations, amounting to a 77% increase compared to pistons with smaller skirts. This implies that the generated noise may amplify under working conditions. Additionally, the piston with a smaller skirt exhibited greater fluctuations and lesser deformations compared to the piston twisted along the y-axis.

The present study shows that more work is needed to optimize the piston skirt design. These observations should be compared using noise and vibration measurements. The presented methodology does not include comparable experimental results; hence, the current work needs to proceed further. The measurement of the radiated noise and vibration would give an advanced tool for the operating piston conditions. This is highly important to improve future high-performance engines in terms of stringent requirements on durability and acoustics. Suggestions for future work include the development of a 3D full simulation framework considering tribodynamic and thermo-mechanical models to support a more detailed piston design process.

Additional information

The author declares: no competing financial interests and that all material taken from other sources (including their own published works) is clearly cited and that appropriate permits are obtained.

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