INTERNATIONAL JOURNAL OF AUTOMOTIVE AND MECHANICAL ENGINEERING
ISSN: 2229-8649 e-ISSN: 2180-1606
VOLUME 22, ISSUE 3, 2025, 12686 - 12698
DOI: https://doi.org/10.15282/ijame.22.3.2025.12.0969

RESEARCH ARTICLE

Effect of CNG on High-Cycle Fatigue Life of the Piston in a Bi-Fuel Engine Based on Stress Gradient Analysis

H. Ashouri

Department of Mechanical Engineering, Varamin-Pishva Branch, Islamic Azad University, Varamin, Iran

ABSTRACT – The piston is a vital component in the internal combustion engine, subjected to complicated loads. This complex component transfers the pressure resultants from engine combustion into the connecting rod and crankshaft, and due to its exposure to intensive gas temperature and pressure alterations, it is considered a critical component. One purpose of piston design is to ensure durability with a long fatigue life. Thus, simulation and analysis of fatigue cracks are essential. Regarding various bi-fuel vehicles, the effect of compressed natural gas (CNG) on the distribution of piston stress and fatigue should be studied, as well. Therefore, this study evaluates the CNG impact on the high-cycle fatigue (HCF) life of the piston in a bi-fuel engine, with a focus on the stress gradient. For this purpose, the temperature and stress were predicted by finite element analysis (FEA) and ANSYS software. Then, nCode Design Life software and Goodman's criterion were applied to obtain an estimation of HCF life. The piston's mechanical properties were determined using tensile tests at various temperatures. As the FEA showed, the upper areas of the piston pin and ring grooves are critical. According to thermo-mechanical analysis results, the piston in the CNG condition shows a higher tolerance of 30 °C temperature and 7.3 MPa stress than the gasoline. Fatigue analysis results showed that substituting CNG for gasoline reduces piston fatigue life by approximately 1.557×10^9 cycles or 41%. Investigating the HCF safety factor showed no critical area in the piston. No rupture in different parts of the piston was observed through the 800-hour durability test. According to the 800-hour durability test, the piston will not be ruptured in any sections.

ARTICLE HISTORY

Received : 14th Jan. 2025
Revised : 15th July 2025
Accepted : 01st Aug. 2025
Published : 12th Sept. 2025

KEYWORDS

CNG effect
High-cycle fatigue
Piston
Stress gradient

1. INTRODUCTION

A piston in an internal combustion engine is employed to convey the combustion pressure into the connecting rod and crankshaft. Naturally, this fundamental component would be subjected to intensive gas temperature and pressure variations, as well as exposed to high thermal and mechanical loads. Therefore, the piston must be designed appropriately to resist thermo-mechanical stresses induced by the heat and pressure process. It is significant due to the further generation of these stresses superposed on thermal ones by the pressure of the gas, acceleration of the piston, and the side force of the piston skirt [1, 2, 3]. The simultaneous thermal and mechanical loadings cause fatigue that decisively affects the piston's vulnerability. Such an impact can lead to a severe decline in engine efficiency by the slightest damage to the piston, which can increase the emission of pollutant gases. Thus, it would be necessary to investigate the forces and stresses applied to the piston [4]. The previous studies mainly focused on the piston's geometry, strength, materials, and manufacturing technology. However, many pistons are still vulnerable during engine operation. Such vulnerability is mainly rooted in corrosion, temperature gradient, fatigue, and abrasion [1, 5]. Higher octane numbers, lower costs, and reduced pollution have made gaseous fuels the most preferred substitute in bi-fuel vehicles. Currently, CNG gas is the most prominent gaseous fuel around the world. The high global reserves of natural gas have driven the growth of studies on the optimum usage of CNG in internal combustion engines. On the other hand, gas combustion time is higher than gasoline, which increases the thermal durability in the engine combustion chamber [6, 7]. Therefore, the piston tolerates higher temperatures, highlighting the necessity of analysing the piston thermo-mechanically and studying its fatigue in bi-fuel engines. This research explores the gas impact on the piston's HCF life in a bi-fuel engine, considering the stress gradient.

Reviewing the available literature reveals that multiple studies have analyzed the stress and fatigue lifetime of the piston. Ashouri [8] predicted the HCF life of engine pistons. The numerical results showed that the minimum fatigue life can be observed at the piston pin's upper portion. The simulation that Najafi and colleagues [9] performed on the piston's fatigue life in a developed engine demonstrated the piston pin as a critical area. Ashouri [10] studied the evaluation of the oil gallery on HCF life for engine pistons, and its finite element results proved an approximate 33% and 37% higher HCF life for the modified piston at 1000 rpm and 5000 rpm, respectively. In an analysis of the stress and fatigue of engine pistons based on simulation, Chen et al. [10] determined the piston pin and piston bowl as critical areas. In an evaluation of the failure of steel pistons for a diesel engine, Liu et al. [4] suggested the major role of gas pressure in the piston's fatigue life. According to Dagar et al. [2], who used several composites in performing the pistons' thermal stress, the minimum safety factor was achieved in steel alloy. The experimental research by Venkatachalam and Kumarave [5] found

***CORRESPONDING AUTHOR** | **H. Ashouri** | ✉ ha.ashouri@iau.ac.ir

that increasing the silicon percentage enhances the fatigue life of the piston. An integrated simulation methodology, presented by Sahu et al. [11] to predict the piston temperature, confirmed the correspondence of test and simulation. The assessment that Balaji and colleagues [12] conducted on the piston's durability demonstrated that the maximum damage occurs in the middle of the skirt on the thrust side of the piston. The real-time temperature monitoring system presented by Mancaruso et al. [13] verified the fit between the experimental and simulation results of the piston temperature.

Based on the HCF analysis conducted by Gai and Zhao [14] through simulation and experimental assessment of a steel piston's fatigue life, the internal cavity of the cooling oil is the weakest area. Soni et al. [3] studied the thermal analysis of automotive pistons, and their obtained numerical results demonstrated that the area under the piston crown is a critical region. HCF life prediction of a coated piston was done by Tan et al. [15] and validated the potential predictability by the simplified model with less than 15% error. Moreover, Ashouri [16] analyzed a coated piston's HCF life and proved the positive impact of the thermal barrier coating (TBC) on the increment of the HCF life. HCF assessment of a steel piston was done by Liu et al. [17], and it was verified that the cracks observed in the durability test of pistons acknowledged the simulated results of HCF life. Niu et al. [18] predicted the HCF safety factor of pistons by finite element simulation, in which the minimum HCF safety factor was observed at the cooling gallery. Baldissera and Delprete [19] thermo-structurally analyzed a coated piston of a diesel engine and demonstrated the piston pin as a critical area. Wang et al. [20] conducted a simulation of the HCF life of pistons regarding engine knock, stating that the minimum HCF life occurs at the piston pin. Considering the residual stress, Ashouri [21] investigated the thermal barrier coating in fatigue life for an aluminium alloy piston, in which the coated piston showed higher failure cycles (about 12% and 31% at 1000 rpm and 5000 rpm speeds, respectively). The side-thrust load's impact on fatigue analysis was explored by Balaji et al. [22], confirming that the piston's damage is within the acceptable range.

In another study, Moser and colleagues [23] analyzed the opposed piston engines thermo-mechanically. The results showed that a new architecture can prevent losses in heat transfer. According to Wang et al. [24], in the prediction of low-cycle fatigue (LCF) considering the oil gallery structure, the piston crown's throat area is where the fatigue damage commences. Nouby and colleagues [25] assessed the impact of TBC on stress distribution. Notably, this assessment was done for the gasoline engine pistons. The results suggested that augmenting the coating thickness can reduce the stress value. Based on the novel thermal fatigue test method presented by Xiong et al. [26], there was less than a 10% difference between experimental and simulated results. Caldera et al. [27] analyzed a damaged diesel engine piston failure, indicating thermo-mechanical fatigue as the principal cause of failure. Based on the theoretical investigations of Liu et al. [28] on a diesel engine's in-piston thermo-mechanical states, during thermal stress, the highest stress arises at the top of the piston's inner chamber. Yao et al. [29] evaluated enriched high-temperature thermal fatigue properties of the pistons. These pistons were aluminium alloy and were coated using a nano-thermal barrier. The results verified the considerably lower temperature of the substrate of the nano-coated piston. The study of Xuguang et al. [30] on the mechanism of wear resistance of coating the engine piston skirt in cold start conditions showed a reduction of the contact pressure of the coated piston skirt (by about 40%). The investigation that Chen et al. [31] performed to study the oil jet temperature's impact on piston cooling indicated a drop in piston temperature (by about 4 to 6°C) by a reduction of the oil temperature (by 13%). The simulation conducted by Ashouri and Afshari [32] on the oil gallery impact on the thermo-mechanical stresses of the piston presented a 13 MPa reduction in the Mises stress.

In reviewing the literature on piston fatigue and stress analysis, it appears that most studies have focused on gasoline conditions. However, the effect of natural gas on the piston's HCF life in bi-fuel engines remains unexamined. Additionally, it is essential to analyse how such an impact can affect the distribution of piston stress and fatigue across various bi-fuel vehicles. Consequently, this study undertook a thermomechanical and fatigue analysis of the piston under both gasoline and CNG conditions to assess the effects of CNG on the piston's fatigue life. Utilizing natural gas in bi-fuel engines prolongs the combustion period, due to the lower combustion rate of gas than gasoline. As a result, the heat penetrates more deeply into the piston and other components of the combustion chamber [6, 7], showing the high importance of thermo-mechanical and fatigue analysis of combustion chamber components, especially pistons, in bi-fuel engines. Various engine components have geometric discontinuities known as notches that reduce the component's resistance against fatigue. The massive stresses occurring in notches severely reduce the notched components' fatigue life, meaning that the notch's impact must be considered in estimating the piston fatigue life [8, 10]. In this regard, a relative stress gradient can be used as an appropriate method for the effect of the notch on components' fatigue [33, 34]. The objective of this study is to evaluate the gas impact on the distribution of the piston's thermo-mechanical stresses, as well as its HCF life, in a bi-fuel engine, regarding stress gradient. Since using temperature-dependent properties of materials would increase the accuracy of FEA results, this study also included the effect of temperature-dependent properties on pistons.

2. METHODOLOGY

2.1 The Finite Element Model and Behavioral Model

The prediction of the thermal-mechanical fatigue life of the piston is one of the most complicated simulations in the engine. Simulation methods are cost-effective approaches to performing several tests. In this regard, the FEA can estimate the efficacy of various designs and facilitate the procedure [3, 8, 34]. Initially, the piston was modelled by SolidWorks software. Then, the stress distribution and temperature were determined using ANSYS Workbench software. The thermo-mechanical results were used for the ANSYS nCode Design Life software to investigate the HCF life of the piston. The studied piston is shown in Figure 1. Furthermore, the engine characteristics and material properties of the engine pistons made of AlSi alloy can be found in Tables 1 and 2.

(a) (b)

Figure 1. (a) piston used in FEA analysis and (b) the piston model

Table 1. The intended engine characteristics

Parameter	Value
Cylinder diameter (mm)	78.6
Crankshaft radius (mm)	85
Engine volume (cc)	1650
Compression ratio	10.8
Peak torque (N-m)	155@4000rpm
Peak power (kW)	113@6000rpm
No. of valve	16

Table 2. Material properties of the pistons [35]

Parameter	Value
Density (kg/m^3)	2680
Poisson's ratio (-)	0.3
Conductivity@20°C (W/mK)	155
Conductivity @150°C (W/mK)	156
Conductivity @250°C (W/mK)	159
Thermal expansion@20°C(1/°C)	19.6×10^{-6}
Thermal expansion@150°C(1/°C)	20.6×10^{-6}
Thermal expansion@250°C(1/°C)	21.4×10^{-6}

2.2 Thermal Boundary Conditions

The correct boundary conditions lead to more accurate results in limited components analysis. The boundary conditions in piston thermal analysis comprise four areas, including the combustion and the piston ring land, skirt, pin, and underside areas [4, 30]. Among the heat transfer equations, the Woschni equation possesses higher accuracy than other equations available. The conventional heat transfer coefficient inside the cylinder would be obtained through the Woschni equation as follows [6]:

$$h_g=3.26P^{0.8}U^{0.8}b^{-0.2}T_g^{-0.55} \tag{1}$$

In which, h_g, P, T_g, U, are the heat transfer coefficient, the pressure, temperature, and the average piston speed, respectively. Also, the cylinder diameter is shown by b. The thermal boundary condition of the crown, comprising the average of the heat transfer coefficient and the mean cyclic temperature, is expressed using the Woschni equation with 2 and 3 equations [6]:

$$\overline{T_g}=\frac{1}{4\pi\overline{h_g}}\int_0^{4\pi} T_g\, h_g d\theta \tag{2}$$

$$\overline{h_g}=\frac{1}{4\pi}\int_0^{4\pi} h_g\, d\theta \tag{3}$$

The exchange of heat between the crown and gas highlights the necessity of calculating the temperature of the gas. Moreover, the heat transfer coefficient should be calculated for the piston crown, as well. The GT-POWER software was used for the combustion 1-dimensional simulation. The piston crown thermal boundary condition was calculated using GT-POWER software and applied to the piston crown. The differential relation of heat flow (depending on time) is shown in polar coordinates as follows [36]:

$$\frac{1}{r}\frac{\partial}{\partial r}(kr\frac{\partial}{\partial r})+\frac{1}{r}\frac{\partial}{\partial \theta}(kr\frac{\partial}{\partial \theta})+\frac{\partial}{\partial r}(k\frac{\partial}{\partial r})=\frac{1}{\alpha}(\frac{\partial T}{\partial t}) \tag{4}$$

where k, r, and α are the thermal conductivity, the radius distance, and thermal diffusivity, respectively. The principal mechanism for heat transfer in the steady-state thermal analysis of the piston is convection. In this study, the boundary conditions are defined and set as the temperature of the fluid (Tf) and the convection (h), mathematically expressed as follows [36]:

$$-k\frac{\partial T}{\partial n} = h(T-T_f) \tag{5}$$

2.3 The Boundary Conditions of Mechanical Loads

The piston inside the cylinder has a reciprocating motion, and such a motion, due to the engine dynamics, generates the inertia opposite to the piston movement direction, calculated through the equation below, in which F_j is the piston's inertia and m_j represents the mass of components with reciprocating motions [6]:

$$F_j=-m_j a \tag{6}$$

The piston acceleration can be calculated by equation 7, in which R is the crankshaft radius, ω is the engine rotating velocity, λ is the crankshaft radius proportion to the connecting rod length, and α is the crankshaft angle [6]:

$$a=R\omega^2(\cos\alpha+\lambda\cos 2\alpha) \tag{7}$$

The equation below expresses the side force applied due to the connection of the piston and the cylinder wall during engine operation [6]:

$$F_C=(F_{gas}+F_j)\tan\beta \tag{8}$$

In which Fc, Fgas, and β are the piston's side thrust force, the gas pressure, and the angle of the connecting rod and cylinder centerline directions.

2.4 Stress Gradient Consideration using FKM Method

Different parts and components of the engine, including the piston, show geometric discontinuities known as notches. Notches are stress raisers due to increased local stress. Since these notches reduce component resistance against fatigue, their presence requires specific considerations. The massive stresses occurring in notches severely reduce the notched components' fatigue life. The notch is the starting point for fatigue cracks, which typically lead to breaking notched components [33, 34], highlighting the importance of considering notch impact in estimating piston fatigue life [8, 10]. An appropriate method can be using a relative stress gradient [33, 34]. Although various methods are presented to consider the notch impact on the components, the Forschungskuratorium Maschinenbau method (FKM) was used [27,28]. The FKM method was developed in 1994 in Germany and has since continued to be updated. The FKM method was developed for the use of the mechanical engineering community involved in the design of machine components, welded joints and related areas. The authors recommend FKM, among the many methods for the effect of notches on the components [34, 37]. The method was utilized by considering the nCode Design Life's relative stress gradient. The software calculates the notch impact by using the correction factor and considering the relative stress gradient as below [37, 38]:

For $\overline{G}_\sigma \leq 0.1$

$$n_\sigma=1+\overline{G}_\sigma 10^{-(a_G-0.5+\frac{R_m}{b_G})} \tag{9}$$

For $0.1< \overline{G}_\sigma \leq 0.1$

$$n_\sigma=1+\sqrt{\overline{G}_\sigma} 10^{-(a_G-0.5+\frac{R_m}{b_G})} \tag{10}$$

For $1<\overline{G}_\sigma \leq 10$

$$n_\sigma=1+\sqrt[4]{\overline{G}_\sigma} 10^{-(a_G-0.5+\frac{R_m}{b_G})} \tag{11}$$

where n_σ is the correction, $\overline{G}_\sigma$is the relative stress gradient, and R_m is the ultimate tensile strength. Also, a_G and b_G are shown as the constants.

2.5 Model for HCF Life Prediction

The fatigue life is also essential for calculation since it is exposed to repetitive forces and prone to rupture due to fatigue components subjected to thermal and mechanical loads experience low-cycle fatigue when stresses reach the plastic region [1, 4, 12]. However, if stresses remain within the elastic region, the fatigue is classified as high-cycle fatigue [39]. The Goodman criterion was used to estimate the HCF fatigue life [8, 10, 14, 16, 40] since the fatigue analysis results using the mentioned criteria show a good fit with damaged piston samples during in vitro tests [8, 10, 16]. The Goodman criterion is given by equation [39]:

$$\frac{\sigma_a}{\sigma_e}+\frac{\sigma_{mean}}{\sigma_{ult}}=\frac{1}{n} \tag{12}$$

where σ_a is the alternating stress, σ_e is the endurance limit, σ_{mean} is the stress (average), σ_{ult} is the ultimate tensile strength, and *n* is the safety factor. The thermal, mechanical, and HCF life analysis process is given as a flowchart in Figure 2.

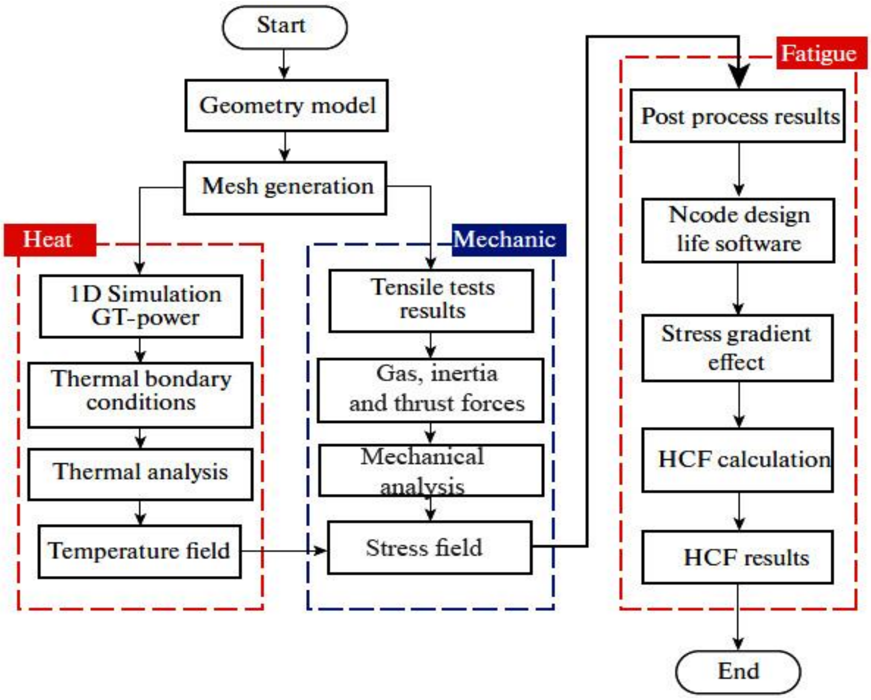
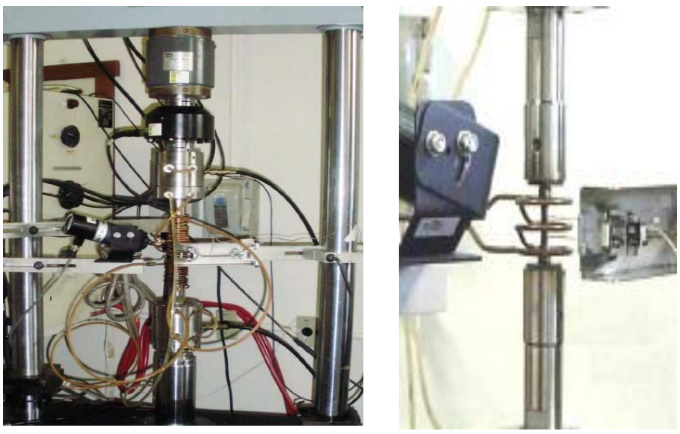


Figure 2. Piston HCF analysis flowchart

2.6 Experimental Tensile Tests

This study investigated the aluminium-silicon cast alloy that can be applied to engine pistons. This material consists of 12.5, 2.4, 2.2, 0.74, 0.41, 0.3, 0.07, 0.02, and 0.01 wt.% of Si, Cu, Ni, Mg, Fe, Mn, Zn, Ti, and Pb, respectively, along with the Al remainder. The tensile tests were performed on the aluminum samples at 25, 150, and 250°C under mechanical strain monitoring based on the E8-E8M standard. The details of tensile test specimens, including the geometry and dimensions, are shown in Figure 3. Furthermore, based on Figure 4, which depicts the tensile test equipment, the sample was heated by the induction system. The mechanical strain and sample temperature were measured using a high-temperature strain gauge and a pyrometer.



(a) (b)

Figure 3. (a) Tensile test equipment and (b) induction coil and extensometer

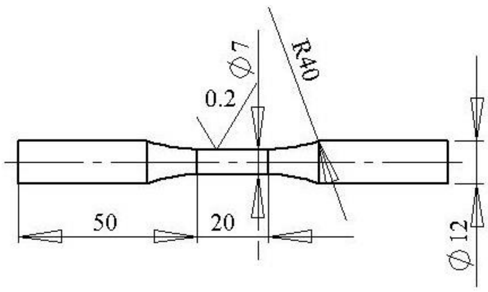


Figure 4. The tensile test specimens (mm-scale)

3. RESULTS AND DISCUSSION

3.1 In-Cylinder Pressure and Temperature

Figure 5 exhibits the in-cylinder temperature and pressure diagrams in gasoline and CNG conditions. The mean gas temperature and heat transfer coefficient were calculated based on equations 2 and 3 using MATLAB software and then applied to the piston crown. The maximum in-cylinder gas pressure in gasoline and CNG conditions is 99.72 and 76.87 bar, respectively, indicating the gas pressure in CNG conditions is approximately 30% higher due to the higher spark advance. The maximum temperatures inside the cylinder in gasoline and CNG conditions are insignificantly different. The in-cylinder gas temperature in CNG conditions, on the other hand, is approximately 20 °C higher.

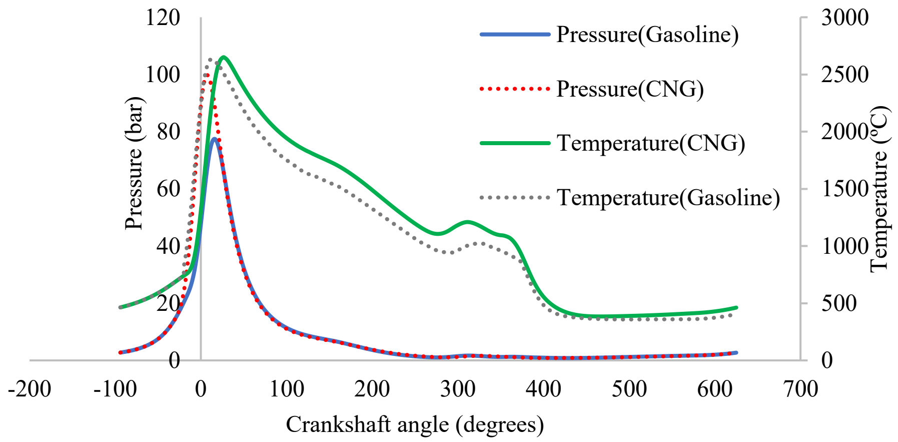


Figure 5. The pressure and temperature inside the cylinder against the crank angle

3.2 Applied mechanical forces

The mechanical forces applied to the piston include gas pressure, inertia force, and side force, as shown in Figure 6 for two conditions: gasoline and CNG, calculated by MATLAB software. The maximum gas force in gasoline and CNG conditions, as given in Figure 6, is 36994.5 and 47440 N, respectively. Such values represent an approximately 28% higher value of the gas force in CNG conditions. However, the inertia force diagrams are similar in gasoline and CNG conditions, where the maximum force is 7263.4 N. The piston's side force is insignificantly different in both conditions.

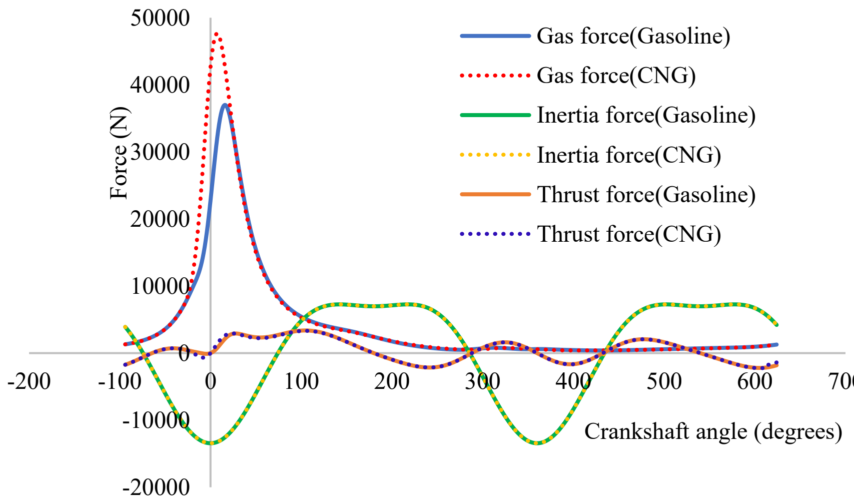


Figure 6. Diagram of piston forces calculated by MATLAB

3.3 Experimental Tensile Test Results

Figure 7 displays the stress-strain curves of the piston at three temperatures (25, 150 and 250°C). Mechanical properties were acquired through the experimental tensile tests and then applied in the ANSYS software.

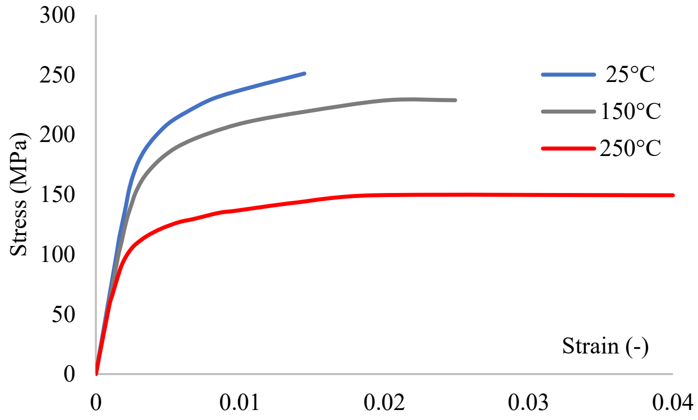
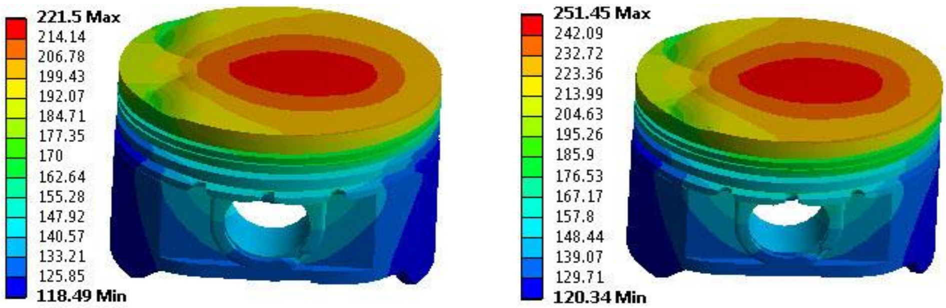


Figure 7. Tensile test results

3.4 Thermal Analysis

Calculating the piston temperature distribution, as a significant issue in engine designing, can provide control of the stress and thermal deformation in the permitted range [8, 10]. The piston temperature distribution enables the piston to be designed optimally (before the prototype stage) with enormous financial savings [11, 13]. The highest temperature of each spot of the piston should be below 66% of the corresponding alloy melting point [41]. Figure 8 demonstrates the piston temperature distribution in gasoline and CNG conditions with maximum piston temperatures of 221.5 and 251.45°C, respectively. According to the figure, a temperature approximately 30 °C higher is imposed on the piston in CNG conditions. By increasing the piston temperature, the applied thermo-mechanical stresses would be increased, which decreases the piston fatigue life. As shown in the figure, the centre of the crown withstands the maximum temperature caused by the exchange of heat of that area with hot combustion gases. By moving from the surface to its skirt, the piston's temperature distribution would be reduced. Thus, the skirt would be subjected to the minimum temperature. The obtained result corresponds with [2, 4, 8, 10, 19, 20] research.



(a) (b)

Figure 8. Temperature field (a) gasoline and (b) CNG

Heightened temperature and temperature gradient at various piston points represent higher thermal stress. Thus, the piston's fatigue life would be reduced due to thermal fatigue. In this regard, the finite element analysis can be used to determine and predict the piston's critical points and fatigue life. In addition to enabling the modification of the critical points, optimization also increases the fatigue life. References [15] and [16] confirmed the reduction of the piston's temperature by coating the crown with a thermal barrier or designing an oil gallery under the piston crown.

3.5 Mechanical Analysis

Based on restricting the piston's pin and utilizing the frictional contact between the pin and piston, the mechanical analysis was executed. Figure 9 illustrates the von Mises stress distribution in pistons in gasoline and CNG conditions. As the figure shows, the upper area of the pin witnessed the highest Mises stress. The results are aligned with [4, 8, 9, 10, 19, 20] studies. The piston's maximum von Mises stresses in the gasoline and CNG conditions, ensuing in the upper area of the pin are 59.485 and 66.803 MPa, respectively. As a result, in CNG conditions, the piston endures a stress of about 7.3 MPa higher. Despite demonstrating the maximum temperature in the thermal analysis, the centre of the piston crown is not a critical point. This part tolerates lower stress than the upper area of the gudgeon pin and piston ring grooves, which are the critical areas in the piston, so it has no considerable importance. The first ring in the ring grooves is mainly responsible for sealing the cylinder and piston, and it is vitally important since it tolerates higher stress due to its higher pressure and temperature. The piston wall is also subjected to high stresses due to its abrasion with the cylinder wall. However, it is not as important as the critical points mentioned above.

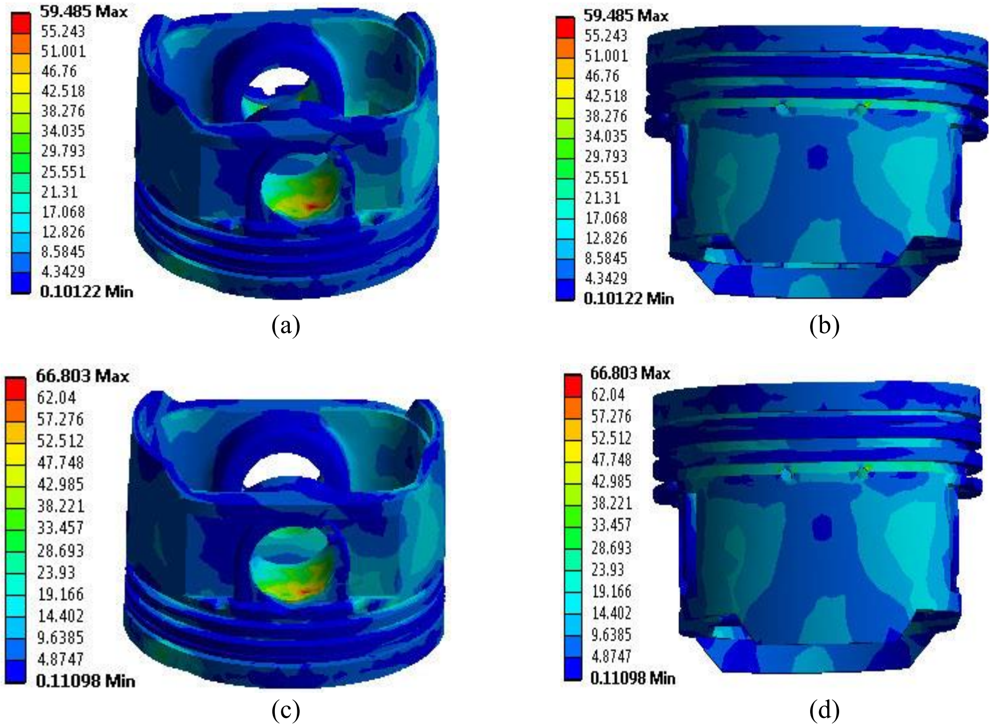


Figure 9. Stress contours in the piston (a, b) gasoline and (c, d) CNG

The stress history of the piston critical node, located on the upper area of the piston pin, is displayed for one engine cycle in Figure 10. As seen, the stress distributions in CNG and gasoline conditions are significantly different in the power stroke, and the piston tolerates about 7.3 MPa higher stress. Such higher stress is caused by the considerable difference in combustion forces shown in Figure 6. The stress distribution in both conditions showed a slight difference in intake, compression, and exhaust strokes.

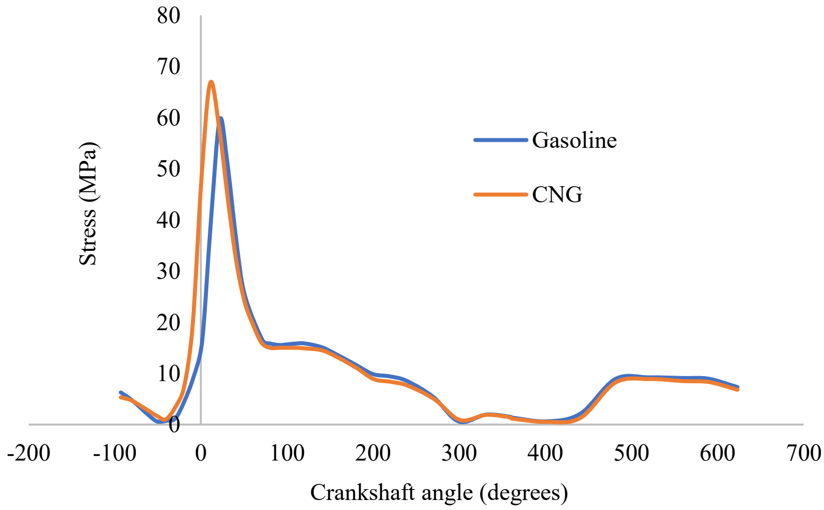


Figure 10. Varied Mises stress distribution of the critical node for the complete combustion cycle

The thermo-mechanical analysis of selecting the number of elements of the piston is provided in Figure 11, in which raising the number of elements causes negligible alterations in the temperature. Moreover, increasing the number of elements to more than 62006 would not considerably change the stress amount, suggesting 62006 as the best number of elements.

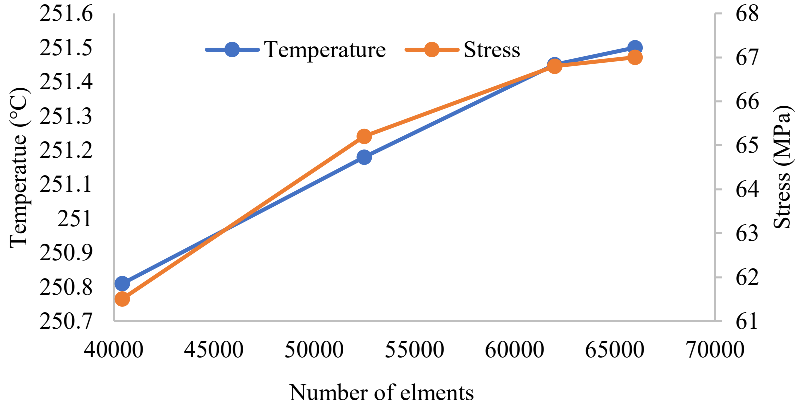


Figure 11. Temperature and stress versus number of elements

3.6 HCF Life Prediction

The fatigue caused by simultaneous thermal and mechanical loads is an essential part of damaging the piston and reducing its life. As seen in the thermo-mechanical analysis, the piston stresses will not surpass the yield limit. Therefore, the piston fatigue is a high-cycle type [9, 16]. Numerous prototypes should be manufactured and tested to achieve a precise design. Nowadays, production should be done faster with lower financial expenditures. In this regard, simulation methods are cost-effective methods that make the performance of multiple tests possible [10, 18]. Predicting the piston's thermal-mechanical fatigue life is one of the most intricate simulation methods of the engine. The piston's HCF analysis is presented in Figure 12. The piston minimum fatigue life in gasoline and CNG conditions occurs at the top of the gudgeon pin by 3.833×10^9 and 2.276×10^9 cycles, respectively. Therefore, substituting CNG for gasoline reduces piston fatigue life by approximately 1.557×10^9 cycles or 41%. As can be seen from the figure, there are more than 10^4 or 10^5 cycles in the critical areas that lead to the failure. Such a thing imposes HCF as the substance for the piston. The fatigue analysis enables the determination of the critical points that show the minimal fatigue life. In this regard, optimization can make the modification of the piston's critical points possible and also increase the fatigue life.

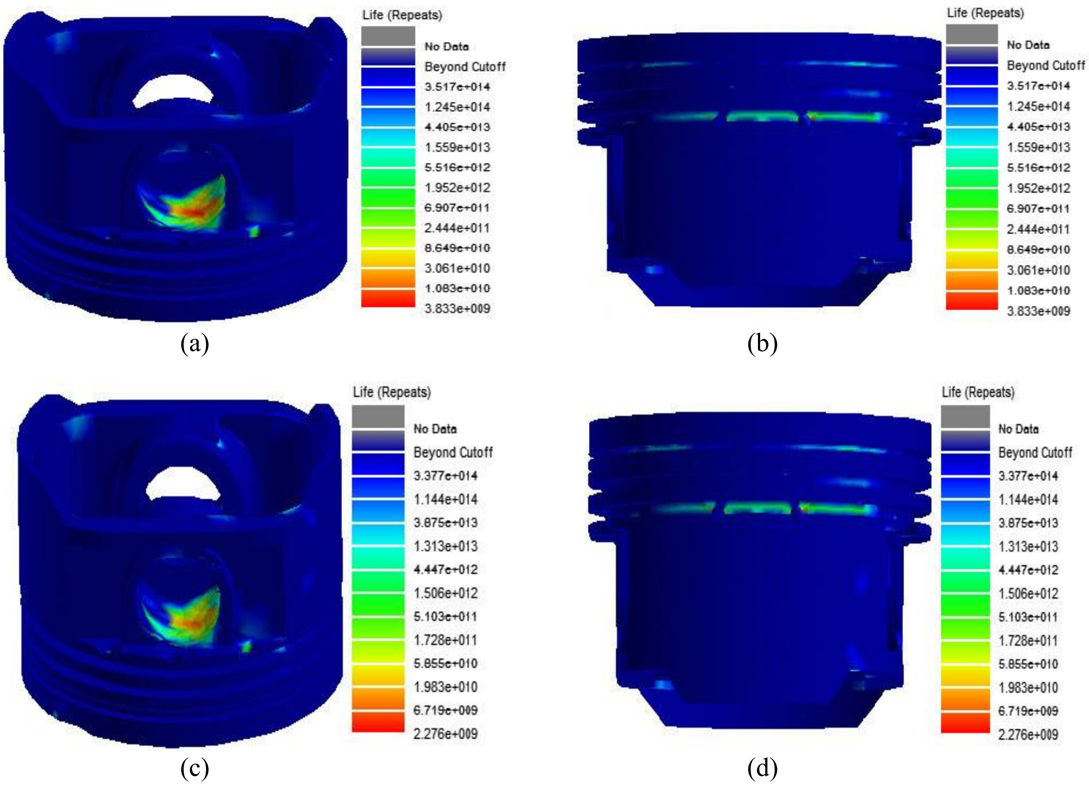


Figure 12. The HCF life prediction in the piston (a, b) gasoline and (c, d) CNG

The safety factor estimation has been performed according to the HCF method using the Goodman equation and considering the stress gradient. According to the safety factor results, as shown in Figure 13, there are no critical areas in terms of safety. The piston minimum safety factor in the gasoline and CNG conditions, in the upper area of the piston pin, are 1.692 and 1.478, respectively.

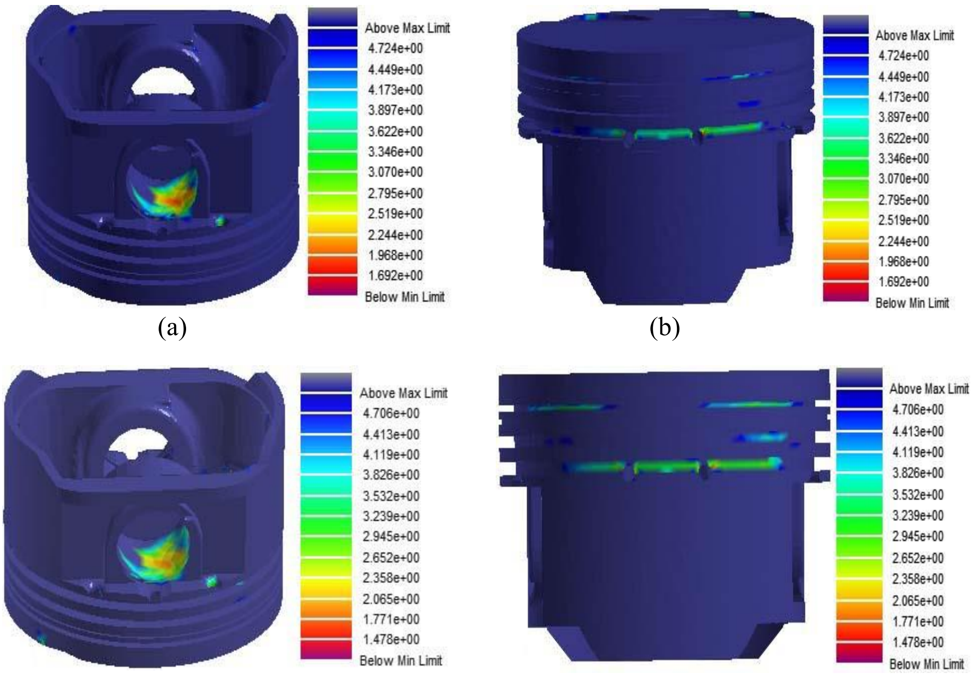


Figure 13. Calculated HCF safety factor in the piston (a, b) gasoline and (c, d) CNG

4. VALIDATION

The maximum piston temperature in the engine trial in gasoline and CNG conditions is 230°C and 255°C, respectively [42], and Figure 8 demonstrates 221.5 and 251.45°C as the maximum piston temperature in gasoline and CNG conditions, respectively. Comparing engine trial results with thermal analysis results shows a perfect fit between them. The most critical values can be observed in the stress and failure cycles at the piston pin's upper part (Figures 9 and 12), corresponding with [16, 20] research. According to the HCF safety factor results, as shown in Figure 13, there are no critical areas in terms of safety factor. Since this research was conducted to investigate the piston's high-cycle fatigue life, the 800-hour endurance test was performed on the engine. The results of the 800-h durability test in gasoline and CNG conditions are given in Figure 14, with no damage in the piston critical areas. Figure 15(a) depicts Silva and colleagues' piston that was ruptured in real engine operation conditions. As seen, the crack sites are observable in the pin's upper zone, correlating with the thermo-mechanical and HCF analysis, in which the maximum mechanical stress and minimum HCF life occurred in the mentioned area. Figure 15(b) presents the critical area of the piston of this research.

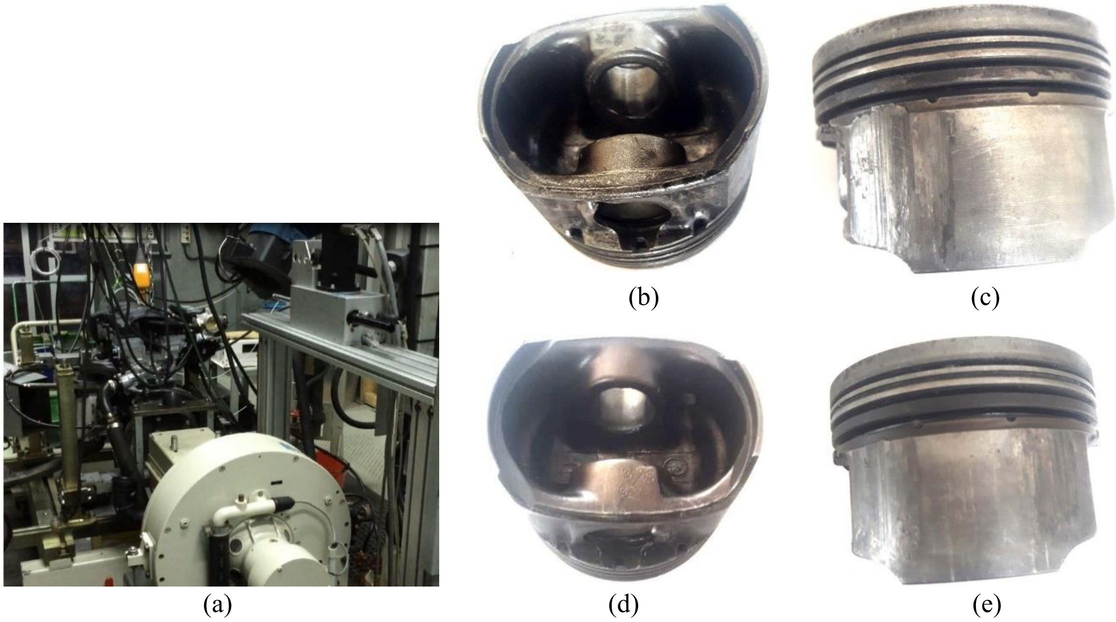
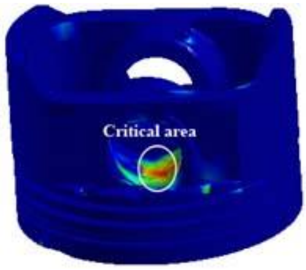


Figure 14. (a) HCF test device and the piston after HCF test (b, c) gasoline and (d, e) CNG

(a)



(b)

Figure 15. (a) The raptured piston in real condition [43] and (b) the critical area of the present study

One of the significant issues in engine design is simulating and calculating the temperature field in components. Calculating the piston temperature distribution is crucial to control its stress and thermal deformation in the allowed range. As the results of various studies have shown, the center of the piston crown is a critical area from the thermal analysis perspective due to the exchange of gases resulting from the combustion, which is also achieved in the present study. The piston in CNG condition tolerates approximately 30°C higher temperature. Raising the piston temperature increases the applied thermo-mechanical stresses, causing a reduction in the piston fatigue life. From the mechanical analysis approach, the most critical area in the piston is the upper area of the piston pin, where the piston tolerates approximately 7.3 MPa higher stress in the CNG condition compared with gasoline. Therefore, using CNG instead of gasoline increases the piston stress by about 12.3%.

Reviewing the literature on fatigue and stress analysis of the piston suggests that the maximum stress and minimum fatigue life are observable in the upper area of the piston pin, consistent with the findings of the present study. The results can be further supported by the cracked piston in the engine experimental test, shown in Figure 15. According to the fatigue analysis results, no critical points were observable in the piston, which is also confirmed by the 800-h durability test results. However, the piston fatigue life would be significantly reduced in the CNG state, compared with the gasoline condition. Notably, such an issue can be addressed effectively through piston optimization.

5. CONCLUSION

This research examined the gas impact on the piston's HCF life in a bi-fuel engine regarding the stress gradient. The piston's mechanical properties were determined using tensile tests at different temperatures. The thermal analysis results proved that the crown center withstands the highest temperature. The thermo-mechanical analysis introduced the upper region of the piston pin and ring grooves as critical regions. According to thermo-mechanical analysis results, the piston in the CNG condition shows a higher tolerance of 30 °C temperature and 7.3 MPa stress than the gasoline. The fatigue analysis can determine the piston's critical points with minimal fatigue life. The HCF results revealed the occurrence of the minimum fatigue life in the pin's upper area, tolerating maximum stress. HCF analysis showed that substituting CNG for gasoline reduces piston fatigue life by approximately 1.557×10^9 cycles or 41%. Investigating the HCF safety factor showed no critical area in the piston. Comparing the FEA with the engine durability test showed the confirmed potency of the piston. The 800-hour test results, as an index of urban driving for the component's high-cycle fatigue assessment, showed no particular damage in the piston. Involving the thermal barrier can reduce the temperature in CNG conditions and boost the fatigue life. Subsequently, the engine performance would be enhanced. Configuring the oil gallery under the piston crown can decrease the piston temperature. Utilizing natural gas instead of gasoline reduces the piston's fatigue life. However, such substitution provides multiple benefits, including lower costs and pollution.

ACKNOWLEDGEMENT

The author would like to thank Islamic Azad University, Iran, for the facility and support.

REFERENCES

[1] Z. Chen, J. Li, J. Liao, and F. Shi, "Stress and fatigue analysis of engine pistons using thermo-mechanical model," *Journal of Mechanical Science and Technology*, vol. 33, no. 93, pp. 4199-4207, 2019.

[2] N. Dagar, R. Sharma, M. L. Rinawa, S. Gupta, V. Chaudhary, and P. Gupta, "Design and analysis of piston using aluminum alloy and composites in Solidworks and Ansys," *Materials Today: Proceedings*, vol. 67, pp. 784–791,2022.

[3] A. K. Soni, S. S. Godara, R. Gade, V. Bernia, R. S. Shekhawat, K. Ksaxena, et al., "Modelling and thermal analysis for automobile piston using ANSYS," *International Journal on Interactive Design and Manufacturing*, vol. 17, pp. 2473–2487, 2023.

[4] Y. Liu, G. Jing, H. Liu, W. Zhang, M. Han, S. Hiao, et al., "Failure analysis and design improvements of steel piston for a high-power marine diesel engine," *Engineering Failure Analysis*, vol. 142, pp. 1-19, 2022

[5] G. Venkatachalam and A. Kumaravel, "Experimental investigations on the failure of diesel engine piston," *Materials Today: Proceedings*, vol. 16, pp. 1196–1203, 2019.

[6] C. R. Ferguson and A. T. Kirkpatrick, *Internal Combustion Engines*, John Wiley & Sons, New York, 2001.

[7] J. Ghorbanian and M. Ahmadi, "Experimental thermal analysis of cylinder block and head of a bi-fuel turbocharged engine," *Meccanica*, vol. 47, pp. 1987–2004, 2012.

[8] H. Ashouri, "Fatigue life assessment for an aluminum alloy piston using stress gradient approach described in the FKM method," *Journal of Solid Mechanics*, vol. 14, no. 1, pp. 57-66, 2022.

[9] M. Najafi, H. Dastani, M. Abedini, and S. Pirani, "Stress analysis and fatigue life assessment of a piston in an upgraded engine," *Journal of Failure Analysis and Prevention*, vol. 19, no. 2, pp. 402-404, 2019.

[10] H. Ashouri, "Improving high cycle fatigue life in a gasoline engine piston using oil gallery with considering stress gradient," *International Journal of Advanced Manufacturing Technology*, vol. 14, no. 4, pp. 73-82, 2021.

[11] A. K. Sahu, S. Chakkamadathil, and D. Das, "Integrated simulation methodology to predict engine head, block, and piston temperatures," *SAE Technical Paper no. 2024-26-0315*, 2024.

[12] Balaji, S. L. Kute, T. Sreenivasulu, and R. Giles, "Piston durability analysis including side-thrust loads," *SAE Technical Paper No. 2019-32-0585*, 2020.

[13] E. Mancaruso, L. Sequino, and B. M. Vaglieco, "Temperature measurements of the piston optical window in a research compression ignition engine to set-up a 1d model of heat transfer in transient conditions," *SAE Technical Paper No. 2019-24-0182*, 2019.

[14] S. Gai and J. Zhao, "Simulation and experimental investigation on fatigue resistance of the forged steel piston in high-duty engine," *Journal of Materials Engineering and Performance*, vol. 32, no. 7, pp. 3202–3214, 2023.

[15] L. G. Tan, G. L. Li, C. Tao, C. Tao, and P. F. Feng, "Study on fatigue life prediction of thermal barrier coatings for high-power engine pistons," *Engineering Failure Analysis*, vol. 138, pp. 1-12, 2022.

[16] H. Ashouri, "Evaluation of thermal barrier coating in fatigue life for an aluminum alloy piston with considering residual stress," *Journal of Solid Mechanics*, vol. 15, no. 3, pp. 343-351, 2023.

[17] Y. Liu, G. Jing, and L. Zhang, "Research on design development and modification of a steel piston in a heavy-duty diesel engine," *SAE Technical Paper No. 2023-01-5023*, 2023.

[18] D. Niu, J. Zhang, P. Xiong, G. Hao, S. Liu, and W. Guo, "High temperature fatigue and oxidation characteristics of forged steel piston materials," *Engineering Failure Analysis*, vol. 97, pp. 220-226, 2019.

[19] P. Baldissera and C. Delprete, "Finite element thermo-structural methodology for investigating diesel engine pistons with thermal barrier coating," *SAE International Journal of Engines*, vol. 12, no. 1, pp. 1-10, 2019.

[20] W. Wang, Y. Lu, Z. Li, and H. Lai, "Simulations of engine knock flow field and wave-induced fatigue of a downsized gasoline engine," *International Journal of Engine Research*, vol. 22, no. 22, pp. 1-15, 2019.

[21] H. Ashouri, "Evaluation of thermal barrier coating in fatigue life for an aluminum alloy piston with considering residual stress," *Journal of Solid Mechanics*, vol. 15, no. 3, pp. 343-351, 2022.

[22] A. Balaji, S. L. Kute, T. Sreenivasulu, and R. Giles, "Piston durability analysis including side-thrust loads," *SAE Technical Paper No. 2019-32-0585*, 2019.

[23] S. Moser, B. Gainey, B. Lawler, and Z. Filipi, "Thermodynamic analysis of novel 4-2 stroke opposed piston engine," *SAE Technical Paper No.2021-24-0096,* 2021.

[24] B. Wang, D. Wang, J. Lei, X. Deng, Y. Liu, H. Yang, "Study on low-cycle fatigue life of diesel engine piston considering oil cooling gallery structure," *Applied Thermal Engineering*, vol. 264, p. 125382, 2025.

[25] N. M. Ghazaly and K. A. Abd El Gwwad, "Evaluation of gasoline engine piston with various coating materials using finite element method," *International Journal of Automotive Engineering*, vol. 9, no. 2, pp. 2942–2948, 2019.

[26] P. Xiong, S. Liu, Z. Li, L. Deng, J. Guo, L. Shi, et al., "A new accelerated thermal fatigue experiment method of pistons and its application," *Engineering Failure Analysis*, vol. 163, p. 108599, 2024.

[27] M. Caldera, J. M. Massone, R. A. Martı´nez, "Failure analysis of a damaged direct injection diesel engine piston," *Journal of Failure Analysis and Prevention*, vol. 17, pp. 979–988, 2017.

[28] X. F. Liu, Y. Wanga, and W. H. Liuc, "Finite element analysis of thermo-mechanical conditions inside the piston of a diesel engine," *Applied Thermal Engineering*, vol. 119, pp. 312-318, 2017.

[29] Z. Yao, K. Hu, and R. Li, "Enhanced high-temperature thermal fatigue property of aluminum alloy piston with Nano PYSZ thermal barrier coatings," *Journal of Alloys and Compounds*, vol. 790, pp. 466-479, 2019.

[30] T. Xuguanga, Z. Jianb, and X. Peiyou, "Wear resistance mechanism of engine piston skirt coating under cold start condition," *Engineering Failure Analysis*, vol. 118, p. 104912, 2020.

[31] Y. C. Simerics, J. Schlautman, and S. D. Simerics, "Experimental and numerical investigation of the multiphase flow and heat transfer in an oil jet cooled engine piston," *SAE Technical Paper No. 2020-01-0165*, 2020.

[32] H. Ashouri and A. Afshari, "Effect of oil gallery on the piston thermo-mechanical stresses," *Journal of New Applied and Computational Findings in Mechanical Systems*, vol. 3, no. 3, pp. 1–12, 2023.

[33] M. Carpenter, P. Jha, and S. Ray, "Fatigue tests of un-notched and notched specimens and life prediction using a variable critical distance method", *SAE Technical Paper No. 2019-01-0801*, 2019.

[34] S. A. McKelvey, Y. L. Lee, and M. E. Barkey, "Stress-based uniaxial fatigue analysis using methods described in FKM-guideline," *Journal of Failure Analysis and Prevention*, vol. 12, pp. 445-484, 2012.

[35] K. Mollenhauer and H. Tschoeke, *Handbook of Diesel Engines,* Springer Heidelberg Dordrecht London New York, 2010.

[36] Z. Yao and Z. Qian, "Thermal analysis of nano ceramic coated piston used in natural gas engine," *Journal of Alloys and Compounds*, vol. 768, pp. 441-450, 2018.

[37] E. Haibach, *FKM-Guideline: Analytical Strength Assessment of Components in Mechanical Engineering*, 5th revised edition: VDMA Verlag, 2003.

[38] Y.L. Lee, M. Barkey, H.T. Kang, *Metal Fatigue Analysis Handbook: Practical Problem-Solving Techniques for Computer-Aided Engineering*, Elsevier, 2012.

[39] R. Stephens, A. Fatemi, H. Fuchs, *Metal fatigue in engineering*, 2nd edition, John Wiley, 2001.

[40] A. Halfpenny, R. Anderson, and X. Lin, "Isothermal and thermo-mechanical fatigue of automotive components," *SAE Technical Paper 2015-01-0548*, 2015.

[41] V. Esfahanian, M. Javaheri and A. Ghaffarpour, "Thermal analysis of an SI engine piston using different combustion boundary condition treatments," *Applied Thermal Engineering*, vol. 26, pp. 277-287, 2006.

[42] M. A. Jamshidi, *The effect of plant and fossil fuels on piston life and reliability*, MSc Thesis, Shahid Beheshti University, Tehran, 2022.

[43] M. Shariyat, J. Fathi Sola, and S.A. Jazayeri, "Experimentally validated combustion and piston fatigue life evaluation procedures for the bi-fuel engines, using an integral-type fatigue criterion," *Latin American Journal of Solids and Struct*ures, vol. 13, pp. 1030-1053, 2016.