Journal of Solid Mechanics Vol. 17, No. 2 (2025) pp. 197-210
DOI: 10.60664/jsm.2025.1125964

Research Paper

Thermo-Mechanical Stress Analyses of An Aluminum Alloy Piston Using Thermal Resistance Circuit Model

H. Ashouri *

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

Received 6 July 2024; Received in revised form 23 February 2025; Accepted 27 February 2025

ABSTRACT

Piston has to withstands thermo-mechanical cyclic stresses in a wide range of engine operating conditions. The thermo-mechanical stresses distribution enables us to optimize the piston design at lower cost Thermo-mechanical stress analyses of an engine piston used in a gasoline engine was studied. The boundary conditions of thermal loads of piston obtained by thermal resistance circuit model and GT-POWER and MATLAB software products. The thermal analysis results showed that the piston crown center withstands the maximum temperature. The results of finite element analysis (FEA) indicated that the stress has the most critical values at the upper portion of piston pin and piston compression grooves. The numerical results showed that the stress maximum occurred at the upper region of piston pin. The results of the Von-Mises and Tresca criterions proved that the upper region of piston pin and ring grooves are critical areas. The distribution of the safety factor demonstrated no critical point in the piston, and the minimum safety factor of 1.1061 occurred in the upper area of piston pin. The results of the FEA are match with experimental damaged piston in these regions. To evaluate properly of results, stress analysis results is compared with real samples of damaged piston and it has been shown that critical identified areas, match well with areas of failure in the real samples.

Keywords: Finite element analysis; Piston; Thermal circuit resistance model; Thermo-mechanical stress analysis.

1 INTRODUCTION

THE piston in the engine industry is the most important component subjected to high thermal and mechanical loads. In addition, gas pressure, piston acceleration, and piston skirt side fore can develop thermo-mechanical stresses superposed on thermal stresses. Therefore, the piston must be designed to withstand thermo-mechanical

*Corresponding author. Tel.: +98 09127235677.
E-mail address: ashouri1394@gmail.com (H. Ashouri)

stresses induced by the heat and pressure process [1,2]. The thermo-mechanical stresses and fatigue distribution enable us to optimize the piston design at lower cost before the first prototype is constructed [1,3,4].

Multiple investigations studied the simulation of thermo-mechanical stresses and fatigue on the piston. Liu et al., investigated Failure analysis and design improvements of steel piston for a high-power marine diesel engine. Their research results showed that the excessive stress amplitude caused by the alternating gas pressure is the root cause of steel piston failure [1]. Stress and fatigue analysis of engine pistons using a thermo-mechanical model was carried out by Chen et al. The numerical results showed that the piston bowel is a critical region [2]. Ashouri performed a fatigue life assessment for an aluminum alloy piston. The numerical results verified that the piston crown center is a critical area [3]. Ashouri studied the effect of oil gallery on high cycle fatigue life in an engine piston. The results of the thermo-mechanical analysis suggested that the oil gallery reduces the stress distribution [4]. The wear resistance mechanism of engine piston skirt coating under cold start conditions was conducted by Xuguang et al. Through the dynamic simulation of the piston, the contact pressure of the piston skirt coated is reduced by about 40% [5]. Jiana et al. investigated the wear behavior of graphite coating on the aluminum piston skirt of the engine. The results revealed that the deposition of graphite coating plays a principal role in decreasing friction and resisting the scuffing of the cylinder bore [6]. Yao et al. evaluated enhanced high-temperature thermal fatigue properties of aluminum alloy pistons with Nano thermal barrier coatings. Their study proved that the substrate temperature of the Nano-coated piston is considerably lower than that of the uncoated piston [7]. A fatigue life study of an engine piston was performed by Najafi et al. Piston pine and ring grooves are critical regions [8]. Liu et al. used FEA to evaluate the thermo-mechanical stresses of a diesel engine piston. The results confirmed that the piston pine region is a critical area [9]. Chen et al. did a numerical simulation on heavy-duty engine piston cooling gallery oil filling ratio. According to their study, the experimental and simulated results of the oil-filling ratio match [10]. Ashouri evaluated the thermal barrier coating in fatigue life for an aluminum alloy piston regarding residual stress. The simulation indicated that the TBC system reduces the stress distribution in the piston by about 2.4 MPa and 8.5 MPa at engine speeds of 1000 rpm and 5000 rpm, respectively [11]. The effect of the oil gallery on the piston thermo-mechanical stresses was simulated by Ashouri and Afahari. Their simulation showed that the Von-Mises stress in the modified piston is reduced by about 13 Mpa [12]. Liu et al. studied the failure analysis of steel pistons for a high-power diesel engine. The cause of the failure of the piston was the alternating gas pressure [1]. Dagar et al. used several composites to evaluate the thermo-mechanical stresses of a gasoline engine piston. Al-SiC had the least deformation among all the materials [13]. Thermodynamic analysis of opposed piston engines was investigated by Moser et al. It was found that the novel architecture can reduce heat transfer losses by up to 5.2% [14]. Balaji et al. investigated the effect of side-thrust loads on fatigue analysis in an engine piston. Their study confirmed that the damage in the piston is within the acceptable limit [15]. Baldissera and Delprete performed thermo-mechanical analysis to investigate the thermo-mechanical stresses of diesel engine-coated pistons. The thermal analysis pointed out a decrease of temperature up to 40°C in the upper part of the coated piston [16]. An experimental and simulation study on aluminum alloy pistons based on thermal barrier coating was conducted by Liu et al. The results indicated that the maximum temperature of the coated piston is 12.2% lower than that of the aluminum alloy piston under the rated power [17]. The thermo-mechanical behavior of an aluminum-silicon alloy piston was investigated by Khan et al. The maximum stress is observed at the upper part of the pinhole of the piston [18]. Tan Studied fatigue life prediction of thermal barrier coatings for engine pistons. Experimental results suggested that the fatigue life prediction model can predict the life of the TBCs with the error being less than 15% [19].

In the literature, there are a lot of numerical and experimental evaluations on thermo-mechanical stress analysis and fatigue in the piston. In the current study, thermo-mechanical stress analysis of the engine piston XU7JP/L3 is studied using the thermal circuit resistance model. The thermal resistance circuit model is an appropriate method to simulate the thermal systems' static and dynamic behavior. While this model has been used widely in various thermal systems, this pattern has been recently prevalent in internal combustion engines. The thermal resistance circuit model is an accurate and simple method to evaluate the temperature in different piston sections [1,20]. The boundary conditions of piston thermal loads were obtained using the thermal circuit resistance model and GT-POWER and MATLAB software products.For this purpose, SolidWorks software was initially used to model the piston. Then, ANSYS Workbench software was used to determine the temperature and stress distribution of the piston. It should be noted that using temperature-dependent properties of materials would increase the accuracy of FEA results. Therefore, the effect of temperature-dependent properties for pistons is also considered in this work.

2 METHODOLOGY

2.1 The finite element model and material properties

FEA makes accurate and reliable assessment of thermo-mechanical stresses and fatigue life results in the engines parts. Finite element analysis allows engineers to find piston weakness at the primary step or to detect the root reason of piston failures [1,9,20]. The piston analyzed in this article is shown in Figure 1. The model consists of 39989 elements (Tet10) for improving the accuracy and acceptability of the obtained results The characteristics of the engine under study are listed in Table 1. Material properties of the engine pistons made of AlSi alloy are listed in Table 2.

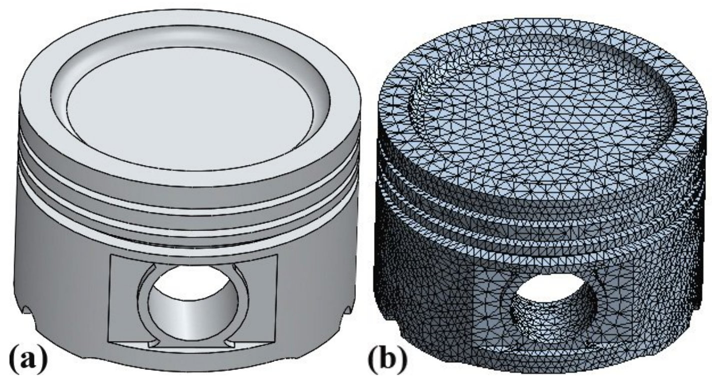

Fig. 1
(a) piston used in FEA analysis and (b) finite element model of piston.

Table 1
Performance parameters of the engine [24]

Parameter	Value
Cylinder diameter (mm)	83
Crankshaft radius (mm)	40.7
Engine volume (cc)	1761
Compression ratio	9.3
Peak power (kW)	70.8@6000rpm
Peak torque (N-m)	153.4@2500rpm
No. of valve	8

Table 2
Material properties of the engine piston [25]

Parameter	Value
Young's modulus@20°C (GPa)	
Young's modulus@150°C (GPa	76
Young's modulus@250°C (GPa)	72
Poisson's ratio (-)	0.3
Density (kg/m^3)	4680
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 *The boundary conditions using thermal resistance circuit model*

The thermal resistance concept has extended applications in heat transferring. The similarities between thermal and electrical resistances facilitate and shorten many heat transfer problems [1,20]. The correct thermal boundary condition can guarantee accurate FEA results. Piston thermal boundary conditions are as follows [3,4,9,23]:
1- Combustion side condition
2- The ring land and skirt condition
3-Underside condition
4-Piston pin condition

In this study, the following equation proposed by Woschni is used to predict instantaneous heat transfer coefficient as following equation [24]:

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

Where h_g is the heat transfer coefficient, P and T_g are the cylinder pressure and temperature respectively, U the mean piston speed and b is the bore.

The mean cylinder gas temperature is computed using a cycle simulation to predict instantaneous gas temperature and then integrated according to [24]:

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

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

The piston crown thermal boundary condition was calculated in GT- POWER software and applied to the piston crown. The thermal resistance circuit method is applied to model the heat transfer in the piston ring land and skirt [1, 20]. The resistances are calculated as follows according to Figure 2.

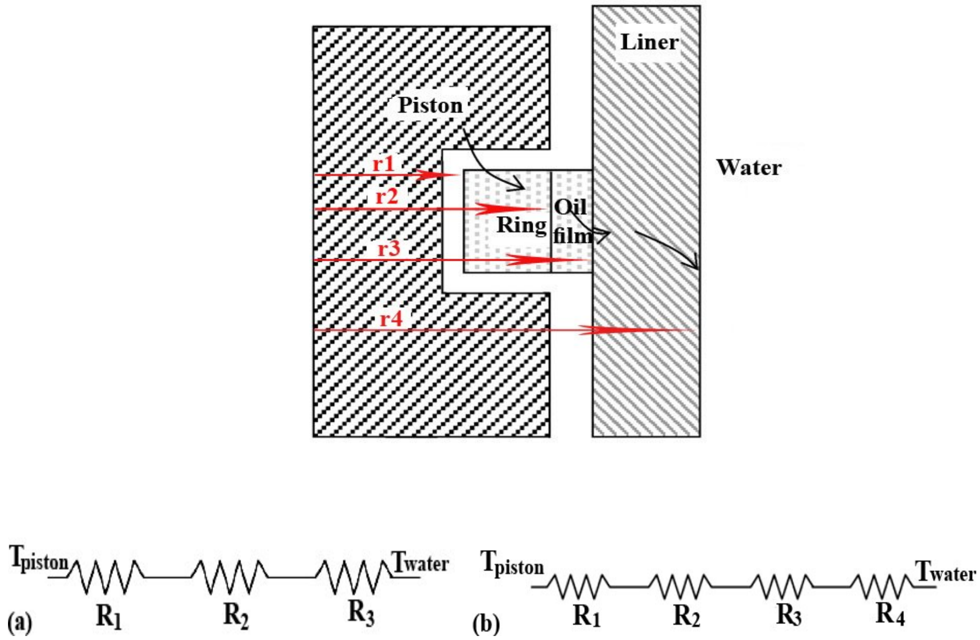

Fig. 2
Thermal resistance circuit model for heat transfer from the (a) skirt and (b) rings.

The resistances are [20]:

$$R_1=\frac{\ln(\frac{r_2}{r_1})}{2\pi H_1 k_{ring}} \quad \text{ring resistance} \tag{4}$$

$$R_2=\frac{\ln(\frac{r_3}{r_2})}{2\pi H_2 k_{oil}} \quad \text{oil film resistance} \tag{5}$$

$$R_3=\frac{\ln(\frac{r_4}{r_3})}{2\pi H_3 k_{block}} \quad \text{block resistance} \tag{6}$$

$$R_4=\frac{1}{h_{water} A_s} \quad \text{water-jacket resistance} \tag{7}$$

Where r_1, r_2, r_3 and r_4 are inner radius of ring, outer radius of ring, bore radius and inner radius of water-jacket, respectively. H_1, H_2 and H_3 are the widths of the heat transfer paths, respectively and As is the effective area in contact with the coolant. The heat transfer coefficient in the first ring is obtained by calculating each of the mentioned resistances using the equation below [20]:

$$h_{eff}=\frac{1}{R_{total} A_{eff}} \tag{8}$$

Where h_{eff} effective convective heat transfer coefficient on the piston, A_{eff} is the piston surface in contact with the ring and R_{total} is the sum of the thermal resistance values. Notably, the other rings are modeled by repeating the same procedure. The underside areas of pistons are cooled through oil jet. This area can be divided into two sections. The heat transfer coefficients of the top and bottom of this section are obtained using equations 9 and 10, respectively [25]:

$$h_{uc1}=900\left(\frac{N}{4600}\right)^{0.35} \tag{9}$$

$$h_{uc2}=240\left(\frac{N}{4600}\right)^{0.35} \tag{10}$$

Where h_{uc1} is the crown underside convective heat transfer coefficient, h_{uc2} is the skirt underside convective heat transfer coefficient and N is the engine speed. The piston has no contact with the cylinder in the area between the piston crown and the cylinder. Heat is transferred by the gas inside the cylinder, which is of conduction type, expressed as follows [20]:

$$k\frac{T_{piston-}T_{wall}}{\delta}=h(T_{piston-}T_{wall}) \tag{11}$$

Where δ is the crevice clearance, h is the convective heat transfer coefficient, k is the conduction heat transfer coefficient and T_{wall} is the wall temperature.

2.3 The boundary conditions of mechanical loads

The piston does the reciprocating motion in the engine which can produce the inertia force. This force is given by equation [26]:

$$F_j=-m_jR\omega^2(1+\lambda\cos2\alpha) \tag{12}$$

The acceleration is given as [26]:

$$a=R\omega^2(\cos\alpha+\lambda\cos2\alpha) \tag{13}$$

Where R represents the radius of crank, ω is the rotating speed of engine, and λ is the ratio of crank radius to the length of connecting rod. Since the piston skirt is in contact with the cylinder wall, it withstands a side thrust force [26]:

$$F_C=(F_{gas}+F_j)\tan\beta \tag{14}$$

Where F_c is the side thrust force, F_{gas} is the gas pressure and β is the angular displacement of connecting rod.

2.4 Analysis procedure

Thermo-mechanical analysis for piston is as follows:

1- Piston modeling using SolidWorks software
2- Determining the boundary conditions in thermal analysis using the thermal resistance circuit model
3- Determining the boundary conditions in mechanical analysis
4- Thermal analysis of the piston using ANSYS software
5- Mechanical analysis of the piston using ANSYS software

3 RESULTS AND DISCUSSION

Due to the heat exchange between the piston and gas, heat transfer coefficients for the piston top surface and gas temperature must be defined. For this purpose, GT-POWER software is applied to establish a one-dimensional simulation model to simulate the engine combustion process. Figures 3 and 4 demonstrate the cylinder's internal gas temperature and internal gas pressure versus crank angle, respectively. Then the instant heat transfer coefficient h_g and instant temperature T_g of gas in one working cycle are obtained as shown in Figure 5. As may be noted from Figures 3 and 4, the maximum temperature and pressure are exerted slightly after the combustion initiation. According to these Figures, the curves slopes drops immediately after spark. The mechanical forces on the piston consist of gas pressure, inertial force, and side thrust force. Figure 6 shows that all forces were applied to piston.

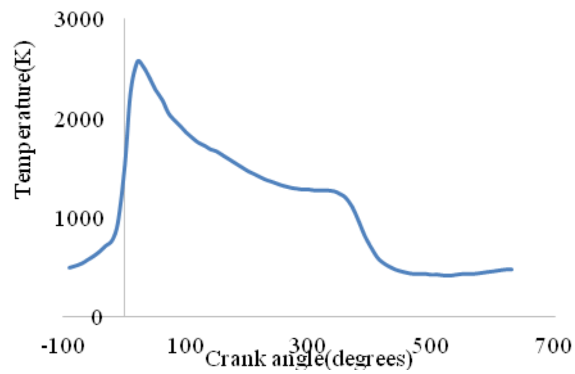

Fig. 3
In-cylinder temperature versus crank angle calculated by GT-POWER.

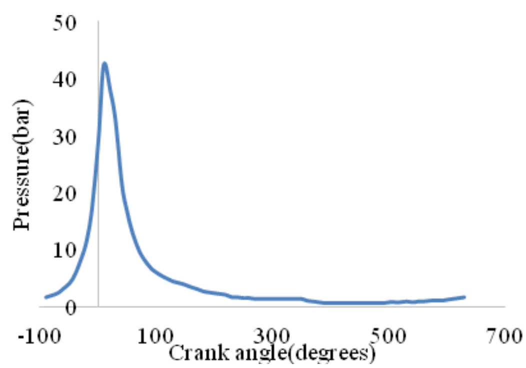

Fig. 4
In-cylinder pressure versus crank angle calculated by GT-POWER.

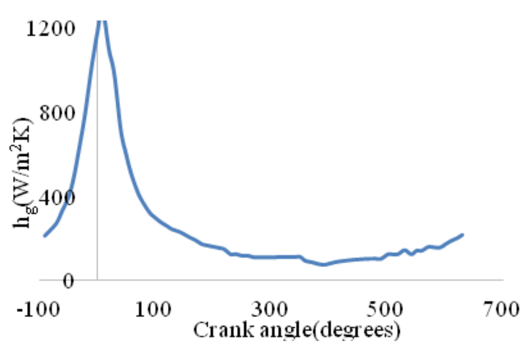

Fig. 5
Piston crown convection heat transfer coefficient versus crank angle calculated by MATLAB.

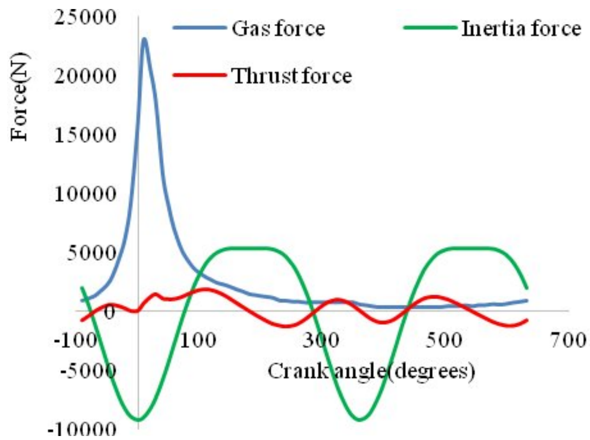

Fig. 6
Diagram of piston forces calculated by MATLAB.

3.1 Thermal analysis

Thermal stresses in the pistons are the dominant stresses, leading to fatigue in the pistons. As a result, thermal loading is the most significant one in the thermo-mechanical analysis of pistons. It is also necessary to evaluate the piston temperature field to detect the thermal stresses and fatigue life within the permitted limit. Thus, the first step of thermo-mechanical analysis is the thermal analysis that evaluates the piston temperature field [1,3,4,14]. The resulting temperature distribution on the piston is given in Figure. 7. As exhibited in Figure 7, the maximum temperature occurs in the center of the piston crown due to the heat exchange of that area with hot combustion gases. The temperature distribution reduces from the piston surface to its skirt. The skirt, therefore, tolerates the minimum temperature. The obtained result corresponds with [3,4,18] research.

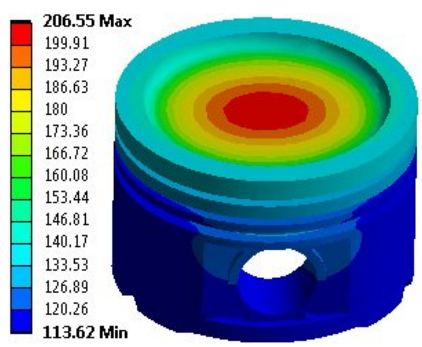

Fig. 7
The temperature distribution in the piston.

3.2 Mechanical analysis

Thermo-mechanical stresses are applied to the piston due to engine temperature fluctuations. Thus, detailed analysis of thermo-mechanical stresses is essential. The mechanical forces on the piston consist of gas pressure, inertial force, and side thrust force [1,2,8,9]. Import the simulated results of the thermal analysis and impose the mechanical loads. Then FEA calculation is performed and the results are investigated. The mechanical analysis is performed in three sections of thermal, mechanical, and thermal-mechanical stresses investigation, and the results are presented in Figures 8, 9, and 10.

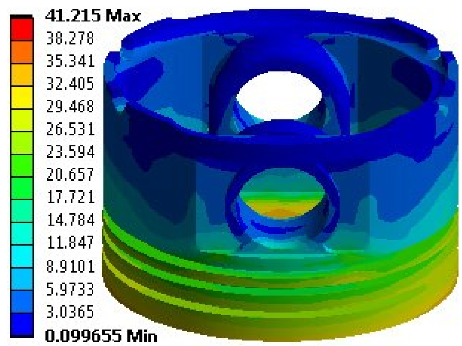

Fig. 8
The Von-Mises stress distribution in the piston under thermal loads.

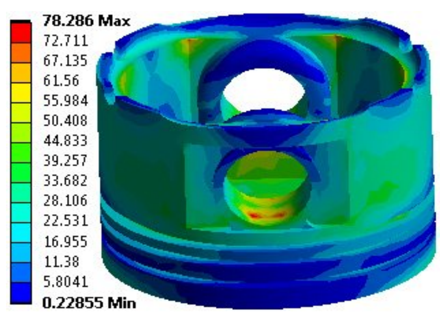

Fig. 9
The Von-Mises stress distribution in the piston under mechanical loads.

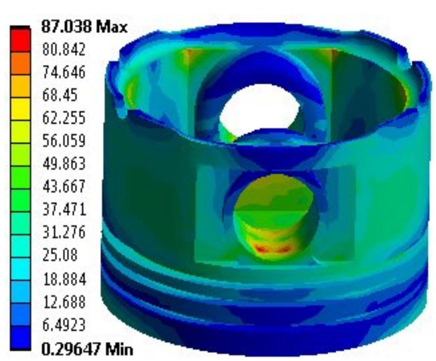

Fig. 10
The Von-Mises stress distribution in the piston under thermal and mechanical loads.

The thermal stress results show that the piston ring groove and the regions below the piston crown are critical areas. The oil ring groove tolerates higher thermal stresses than the compression ring groove. The thermal stress is insignificant in the piston pin area. The upper regions of the piston pin and piston ring groove tolerate high mechanical stress, and the maximum mechanical stress occurs in the upper area of the piston pin. The lower area of the piston crown, which tolerates high thermal stress, tolerates insignificant mechanical stress. The stress distribution of the combined thermal and mechanical stresses is similar to mechanical stress distribution. Therefore, the prominent stress applied to the piston is rooted in the mechanical stresses. This result corresponds with the study conducted by Liu et al. Thus, the upper area of the piston pin is the most critical area of the piston in which the maximum stress occurs. At the combustion moment, the upper areas of the piston pin and piston are in touch with each other. Due to the considerable force of combustion than other forces, it can be expected that the highest stress occurs in the upper area of the piston pin. The maximum thermal, mechanical, and thermomechanical stresses are 41.215, 78.286, and 87.038 MPa, respectively.

Despite demonstrating the maximum temperature in the thermal analysis, the center 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's 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. According to the Tresca criterion, also known as the maximum shear stress criterion, failure occurs when the maximum shear stress exceeds half of the ultimate failure limit of the material [18]. The Tresca stress distribution in the piston is shown in Figure 11. This Figure indicates that the maximum stress value is 50.029Mpa and the position is at the upper region of piston pin. So, this region is a critical area. On the other hand, the results of the stress distribution in the Tresca criterion confirm the results simulated by the Von-Mises criterion.

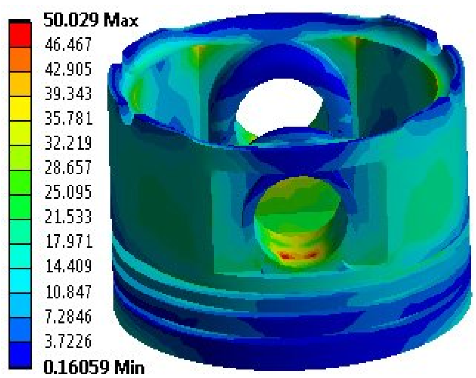

Fig. 11
The Tresca stress distribution in the piston.

Figure 12 shows the normal stress in the center of the piston crown. The produced stress is compressive due to combustion pressure applied on the piston crown. The maximum compressive stress that the piston crown tolerates occurs at the combustion moment. The Von-Mises stress for the critical node located in the upper area of the piston pin is shown in Figure 13. As can be seen, the highest stress is related to the combustion moment.

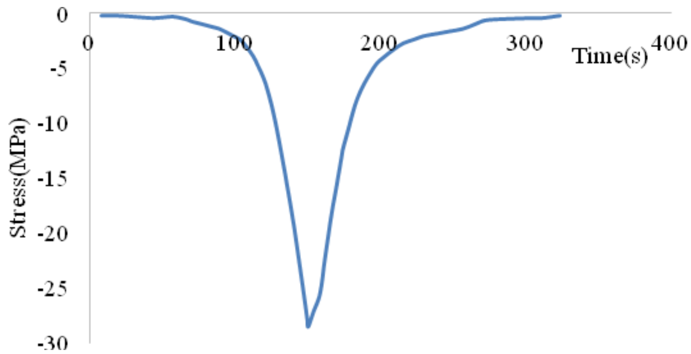

Fig. 12
The normal stress in the center of the piston crown.

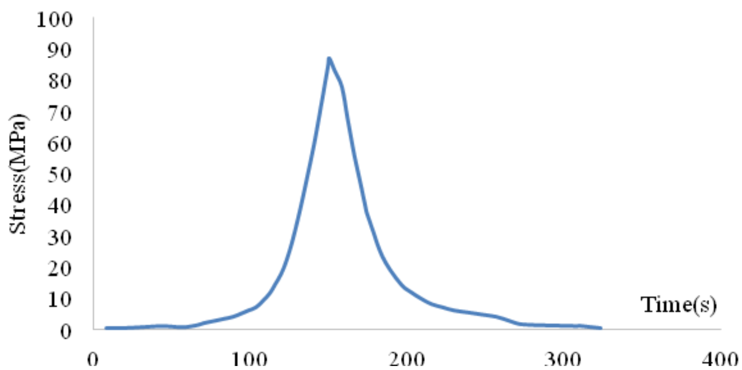

Fig. 13
The Von-Mises stress in the critical node of the piston.

As shown in Figure 14, the maximum thermo-mechanical deformation is 0.12311 mm, which occurs perpendicular to the center line of the piston pinhole at the lower end of the piston skirt. This result corresponds with the study conducted by Liu et al. It can be explained by the easy deformation of the end of the piston skirt in this area. However, this deformation would not be significant along the piston pin axis due to the mechanical boundary conditions applied to the piston pin.

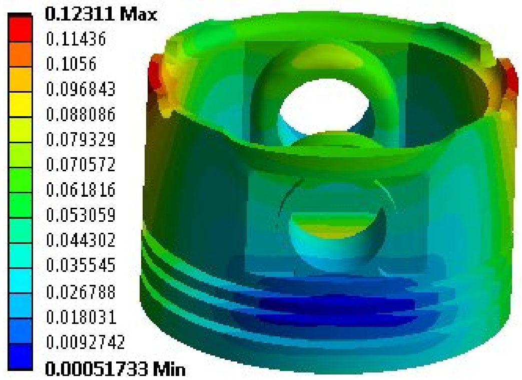

Fig. 14
The thermo-mechanical deformation in the piston.

The safety factor distribution is shown in Figure 15. It can be seen that the minimum safety factor is 1.1061. According to the thermodynamic analysis results, the upper area of the piston pin tolerates the highest stress and is a critical area. Therefore, as expected, the lowest safety factor occurred in this area. Considering that the safety factor is greater than one, there is no possibility of failure of the piston.

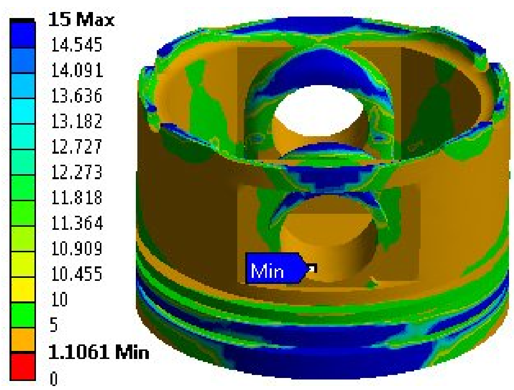

Fig. 15
The safety factor distribution in the piston.

Figure 16 displays the thermo-mechanical analysis results of the piston based on the element number. Increasing the element number would not significantly change the temperature. Since the increment of the element number to 39989 does not significantly alter the stress value, it can be regarded as the best element number.

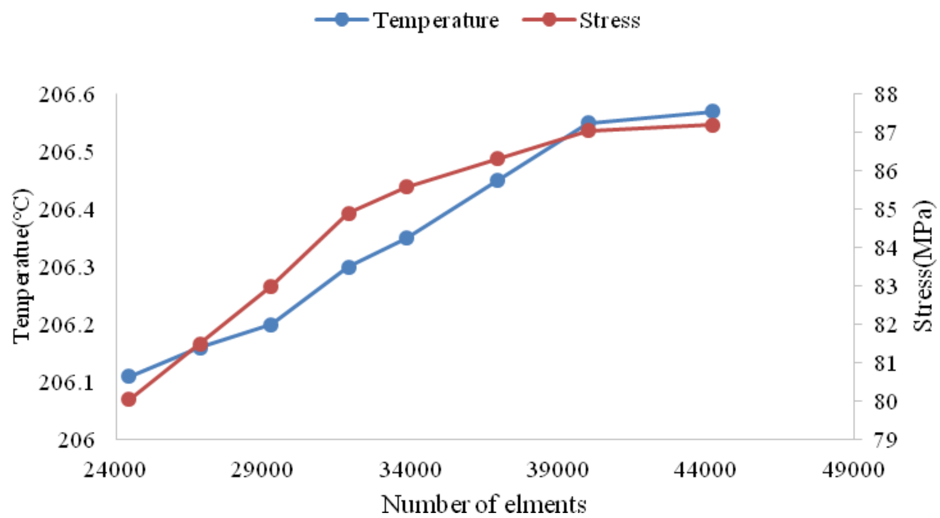

Fig. 16
The Temperature and Von-Mises stress versus number of elements**.**

3.3 Verification of FEA results

As observed in Figures 8-11, two areas of the piston are serious: upper region of piston pin and compression grooves. Different engine pistons where cracks initiated in those areas will be presented subsequently. Thermo-mechanical stress analyses on the piston proved the same serious areas. Two damaged pistons in the upper region of piston pin and compression grooves is shown in Figure 17. Comparison of damaged pistons with Figures 8-11 proves that the thermo-mechanical stress analysis results have a very good agreement with the real damaged pistons.

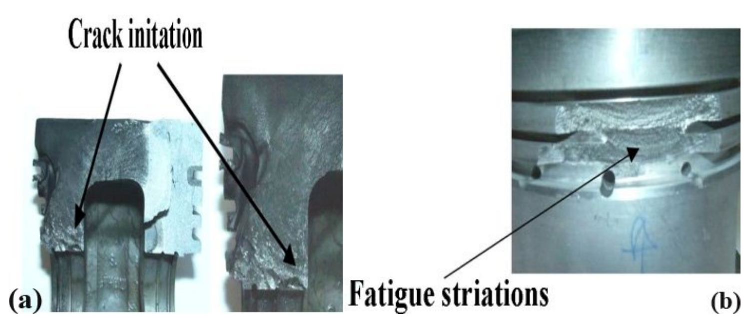

Fig. 17
A damaged engine piston (a) in the upper portion of the piston pin and (b) on piston compression grooves [27].

4 CONCLUSION

The study aimed to investigate the thermo-mechanical stresses using a thermal resistance circuit model in a gasoline engine piston. The thermal resistance circuit model is an accurate and simple method to evaluate the temperature in different piston sections [1,20]. The thermal analysis results proved that the piston crown center withstands the maximum temperature. The Obtained FEA results showed that the stress has the most critical values at the upper portion of the piston pin and piston compression grooves. The results of the Von-Mises and Tresca criteria showed that the upper region of the piston pin and ring grooves are critical regions. Accordingly, the minimum safety factor was 1.1016, which occurs in the upper area of the piston pin. The results of the stress analysis were compared with the real damaged piston samples and indicated that the identified critical areas matched well with areas of failure in the real samples.

REFERENCES

[1] Liu Y., Jing G., Liu H., Zhang W., Han M., Xiao S., Zhang Z., 2022, Failure analysis and design improvements of steel piston for a high-power marine diesel engine, Engineering Failure Analysis, https://doi.org/10.1016/j.engfailanal.2022.106825.

[2] Chen Z., Li J., Liao J., Shi F., 2019, Stress and fatigue analysis of engine pistons using thermo-mechanical model, Journal of Mechanical Science and Technology 33(9):4199-4207.

[3] Ashouri, H., 2022, Fatigue life assessment for an aluminum alloy piston using stress gradient approach described in the FKM method, Journal of Solid Mechanics 14(1): 57-66.

[4] Ashouri, H., 2021, Improving High Cycle Fatigue Life in A Gasoline Engine Piston using Oil Gallery with Considering Stress Gradient, International Journal of Advanced Design and Manufacturing Technology 14(4): 73-82

[5] Xuguanga T., Jianb Z., Peiyou X., 2020, Wear resistance mechanism of engine piston skirt coating under cold start condition, Engineering Failure Analysis, https://doi.org/10.1016/j.engfailanal.2020.104912.

[6] Jiana Z., Zhong-yub P., Shi-yingc L., Sheng-weid S., Li-juna D., 2019, Investigation of wear behavior of graphite coating on aluminum piston skirt of automobile engine, Engineering Failure Analysis 97: 408–415.

[7] Yao Z., Hu K., Li R., 2019, Enhanced high-temperature thermal fatigue property of aluminum alloy piston with Nano PYSZ thermal barrier coatings, Journal of Alloys and Compounds, https://doi.org/10.1016/j.jallcom.2019.03.177.

[8] Najafi M., Dastani H., Abedini M., Pirani S., 2019, Stress analysis and fatigue life assessment of a piston in an upgraded engine, Journal of Failure Analysis and Prevention, https://doi.org/10.1007/s11668-019-00583-4.

[9] Liu XF., Wanga Y., Liuc W H., 2017, Finite element analysis of thermo-mechanical conditions inside the piston of a diesel engine. Journal of Applied Thermal Engineering, doi: http://dx.doi.org/10.1016/j.applthermaleng.2017.03.063.

[10] Chen Y., Moghe S., 2018, Heavy Duty Engine Piston Cooling Gallery Oil Filling Ratio Measurement and Comparison of Results with Simulation. ASME Internal Combustion Engine Division Fall Technical Conference, USA, November 4-7.

[11] Ashouri H., 2023, Evaluation of thermal barrier coating in fatigue life for an aluminum alloy piston with considering residual stress, Journal of Solid Mechanics, 15(3): 341-351.

[12] Ashouri H., Afshari A., 2023, Effect of oil gallery on the piston thermo-mechanical stresses, Journal of New Applied and Computational Findings in Mechanical Systems, 3(3): 1-12.

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

[14] Moser S., Gainey B., Lawler B., Filipi Z., 2021, Thermodynamic Analysis of Novel 4-2 Stroke Opposed Piston Engine, SAE Technical Paper No.2021-24-0096.

[15] Balaji A., Kute S.L., Sreenivasulu Giles, T.R., 2019, Piston Durability Analysis including Side-Thrust Loads, SAE Technical Paper No.2019-32-0585.

[16] Baldissera P., Delprete C., 2019, Finite Element Thermo-Structural Methodology for Investigating Diesel Engine Pistons with Thermal Barrier Coating, SAE International Journal Engines, 12(1): 1-12.

[17] Yang L., Jilin L., Xiaoqiang N., Xiwen D., Jun W., Zhigao W., 2022, Experimental and simulation study on aluminium alloy piston based on thermal barrier coating, Scientific Reports, 12: 1-13.

[18] Khan S.N., Usman A., Mourad A.H., Park C.W., Liwicki M., Almqvist A., 2022, Numerical Investigation of Thermo-Mechanical Behaviour of an Aluminum-Silicon Alloy Piston in an IC Engine, 13th International Conference on Mechanical and Aerospace Engineering, 210-216.

[19] Tan L.G, Li G.L., Tao C., Feng P.F., 2022, Study on fatigue life prediction of thermal barrier coatings for high-power engine pistons, Engineering Failure Analysis, https://doi.org/10.1016/j.engfailanal.2022.1063.

[20] Esfahanian V., Javaheri M. Ghaffarpour, A., 2006, Thermal analysis of an SI engine piston using different combustion boundary condition treatments, Journal of Applied. Thermal Engineering, 26: 277-287.

[21] Mirmohammadi A., Kalhor A., 2017, Studying turbocharging effects on engine performance and emissions by various compression ratios, American Journal of Energy and Power Engineering 4(6): 84-88.

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

[23] Ashouri H., 2019, Evaluation of thermal barrier coating in low cycle fatigue for exhaust manifold, Journal of Simulation & Analysis of Novel Technologies in Mechanical Engineering, 12(4): 41-51.

[24] Ferguson C.R, Kirkpatrick A.T., 2001, Inernal Combustion Engines, John Wiley & Sons, NewYork.

[25] Liu Y., Reitz R.D., 1997, Multidimensional modeling of combustion chamber surface temperatures, SAE paper No. 971539.

[26] Heywood J.B., 1998, Internal combustion engine fundamentals, McGraw-Hill press.

[27] Silva, F.S., 2006, Fatigue on engine pistons – A compendium of case studies, Journal of Engineering Failure Analysis, 13: 480-492.