Journal of Solid Mechanics Vol. 14, No. 1 (2022) pp. 57-66
DOI: 10.22034/jsm.2021.1898031.1582

Fatigue Life Assessment for an Aluminum Alloy Piston Using Stress Gradient Approach Described in the FKM Method

H. Ashouri *

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

Received 6 September 2021; accepted 8 December 2021

ABSTRACT

Engine piston is one of the most complex components among all automotive. The engine can be called the heart of a car and the piston may be considered the most important part of an engine. In fact, piston has to endure thermo-mechanical cyclic loadings in a wide range of operating conditions. This paper presents high cycle fatigue (HCF) life prediction for an aluminum alloy piston using stress gradient approach described in the Forschungs Kuratorium Maschinenbau (FKM) method. For this purpose, first Solid works software was used to model the piston. Then Ansys Workbench software was used to determine temperature and stress distribution of the piston. Finally, in order to study the fatigue life of the piston based on HCF approach, the results were fed into the nCode Design Life software. The numerical results showed that the temperature maximum occurred at the piston crown center. The results of finite element analysis (FEA) indicated that the stress and number of cycles to failure have the most critical values at the upper portion of piston pin and piston compression grooves. To evaluate properly of results, stress analysis and HCF 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. The lifetime of this part can be determined through FEA instead of experimental tests.

Keywords: Finite element analysis; Piston; FKM method and high cycle fatigue.

1 INTRODUCTION

THE increasing needs for higher power density, low emission and low fuel consumption impose many restrictions on the design process of engine components. Therefore, the design and analysis methods of engines have become substantially more complicated [1, 2, 3]. Piston is one of the most challenging components in engine which is subjected to high thermal and mechanical loads. Large temperature difference between piston crown and cooling galleries induces significant thermal load in piston. Besides, the firing pressure, piston acceleration and

*Corresponding author. Tel.: +98 912 7235677.
E-mail address: ashouri1394@gmail.com (H. Ashouri)

piston skirt side force can develop cyclic mechanical stresses which are superimposed on former thermal stresses. So, the piston should be well designed to withstand the thermal and mechanical stress resulted from extreme heat and pressure of the combustion chamber [3,4]. Thermal loads due to the high temperature gradient in the piston and the mechanical loads as a result of high gas pressure and piston acceleration make this component vulnerable to failure. This damage is created both by thermal and mechanical stresses [5,6]. To improve the life and reliability of pistons, a large number of fatigue tests are carried out when designing new pistons. However, to reduce the cost and time involved in fatigue testing, many engine manufactures use FEA to predict the distribution of temperature and stress on engine components. Therefore, this simulation and analysis of fatigue cracks in the design of piston is of paramount importance [1,4,6]. Numerous papers have been presented on analysis of stress and fatigue in piston. Stress analysis and fatigue life assessment of a piston in an upgraded engine was carried out by Najafi et al. The results showed that increasing the rotational speed and gas pressure inside the cylinder by 30% will decrease the life of the piston by 4000 times [6]. Mancaruso et al. collected temperature measurements at different engine speeds and loads to set-up a theoretical correlation and 1d model of Heat Transfer in Transient Conditions. For motoring conditions, a theoretical correlation to approximate the normalized curves of temperature transient was found [7]. Evaluation of thermal barrier coating (TBC) on stress and deformation distribution in gasoline engine piston was studied by Nouby et al. Their simulation proved that the values of the maximum stresses are reduced by increasing the coating thickness [8]. Yao and Qian investigated thermal analysis of nano ceramic coated piston used in natural gas engine. Results showed that the temperature at the top surface of coated piston is significantly higher than that of the uncoated piston. The lower metallic substrate temperature provides better thermal fatigue protection for the piston [9]. Liu et al. used finite element analysis to investigate the thermo-mechanical conditions inside the piston of a diesel engine. The results showed that in different thermal, mechanical, and thermal-mechanical coupling conditions, their relationships obeyed Arrhenius model, Inverse power law model, and Generalized Eyring model, respectively [4]. Failure analysis of a damaged direct injection diesel engine piston was studied by Caldera et al. It was concluded that thermo-mechanical fatigue along with some degree of degradation suffered by the material is responsible of the reported failure [5]. Dudareva et al. examined thermal protection of internal combustion engines pistons. Simulation and motor tests have showed micro arc oxidation coatings are efficient for thermal protection of pistons from fusions [10]. Lu et al. analyzed thermal temperature fields and thermal stress under steady temperature field of diesel engine piston. The steady-state temperature field of finite element simulation proved a good agreement with the experimental results [3]. Multi-body dynamic simulation and fatigue analysis of the unique crank-train for a creative two-stoke opposed piston diesel engine was studied by Changming and Sichuan. Calculated results showed that the minimum safety factor on crank journal fillet can meet relevant evaluation criterion [11]. Golbakhshi et al. evaluated the coupled thermo-mechanical stresses for an aluminum alloy piston used in a gasoline engine XU7. According to the obtained results, the temperature of lubricating oil has not a considerable effect on thermal stress gradient of the piston. The maximum normal stress takes place at the middle surface of the piston crown [12]. Shariyat et la. established an integral-type fatigue criterion for life evaluation of the bi-fuel engines piston. Results of the TMF fatigue analyses proved that the piston life decreases considerably when natural gas is used instead of gasoline [13]. Kakaee et al. investigated thermo-mechanical analysis of an SI engine piston using different boundary condition treatments. The results of their study disclosed that resistor−capacitor model with a smaller number of equations and consequently less solution time, is an appropriate method for solving problems of engine piston heat transfer [14]. Improving heat transfer and reducing mass in a gasoline piston using additive manufacturing was studied by Reyes Belmonte et al. Their analysis showed that the weakest piston region is the pin bore [2]. Influence of different temperature distributions on the fatigue life of a motorcycle piston was studied by Giacopini et al. They come to the conclusion that both the location of the minimum Dang Van safety factors and the minimum Brown–Miller numbers of cycles to failure correspond to effective crack initiation points [15]. Sharma et al. performed Thermal and mechanical failure analysis of a two-stroke motocross engine piston. Their analysis showed that fatigue due to cold start can be minimized if the engine is allowed to warm at idle speeds [16].

In the current study, the fatigue life of the piston of gasoline engine XU7JP/L3 is investigated. For this purpose, first Solid works software was used to model the piston. Then Ansys Workbench software was used to determine temperature and stress distribution of the piston. Finally, in order to study the fatigue life of the piston based on high cycle fatigue approaches, the results were fed into the nCode Design Life software. It should be noted that using time-dependent of material properties would increase the accuracy of FEA results [13]. Therefore, the effect of time-dependent properties for piston, is considered in this work. The effect of inertia and side thrust forces on stress and fatigue fields of the engine piston is also investigated in this article. Most engine components have complicated geometries and contain different kind of notches. In fatigue life estimation of these components, the effects of notch-like features must be taken into account [17,18]. In this study, the notch effect is considered based on the stress gradient approach described in the FKM method.

2 THE FINITE ELEMENT MODEL AND MATERIAL PROPERTIES

Fatigue life prediction of each component needs the cyclic stress-strain distribution. Hot components of engines had complex geometry and loading, and the applying analytical methods for the detection of stress-strain distribution in them is impossible. Many researchers have used finite element method to obtain stress-strain distribution in of geometrically complex components [1,4,6].

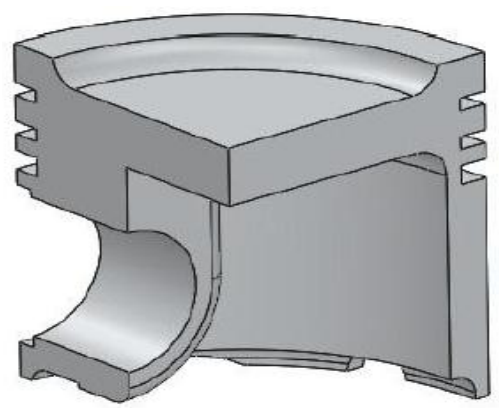
The object of the study is taken from the piston of gasoline engine XU7JP/L3. This engine is assembled in cars Samand, Pegout 405 and Pars models as a widespread car in Iran transport section [12]. The characteristics of the engine under study are summarized in Table 1.

Table 1
Specification of the engine under study [19].

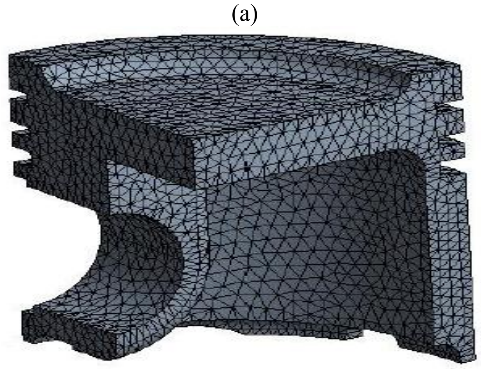
Parameter	Value
Bore (*mm*)	83
Stroke (*mm*)	81.4
Connecting rod (*mm*)	150.5
Engine volume (*cc*)	1761
Compression ratio	9.3
Max power (*kW*)	70.8@6000 *rpm*
Max torque (*N-m*)	153.4@2500 *rpm*
No. of valve	8

Due to the symmetrical structure of the piston, a 1/4 3D solid model was created [1,20]. The piston analyzed in this article is shown in Fig. 1. Piston is made of AlSi alloy with a Young's modulus of 90 *GPa*, a Poisson's ratio of 0.3, and a coefficient of thermal expansion of 21×10^{-6} *per* °*C* [12]. Piston is modeled with three-dimensional continuum elements. The model consists of 43571 elements (Tet10) for improving the accuracy and acceptability of the obtained results.

(a)



(b)



(c)

Fig.1
(a) XU7JP/L3 gasoline engine, (b) The piston generated by SolidWorks and (c) Finite element model of the piston.

3 MODELS FOR FATIGUE LIFE PREDICTION

In the real engineering world, engine components mostly operate under complex thermal-mechanical loading conditions where temperature and mechanical loads change simultaneously with time such as during engine start up and shutdown. The existence of thermal gradients due to uneven heat transfer by material and structural design may cause complex loading situations between thermal expansion and mechanical constraints and loads [4,15].

Fatigue cracks are caused by the repeated application of loads which individually would be too small to cause failure. Each of the cyclic fatigue failures, low or high cycle fatigue, occurs by evidently different stress-strain situation. The high cycle fatigue happens when the stress or strain cycles are largely limited to the elastic range. This domain is linked with low loads and long lives and is commonly referred to as high cycle fatigue. The other type of cyclic loading, the low cycle fatigue, happens when significant plastic strain occurs during at least some of the loading cycles. This fatigue involves some lower number of cycles, relatively short lives, so it is usually referred to as low cycle fatigue. The stress-based approach to fatigue is typically used for life prediction of components subject to high cycle fatigue, where stresses are mainly elastic. This approach emphasizes nominal stresses rather than local stresses. It uses the material stress-life curve and employs fatigue notch factors to account for stress concentrations, empirical modification factors for surface finish effects, and analytical equations such as Goodman and Gerber equation to account for mean stress effects. The Goodman and Gerber equations are given by Eq. (1) and (2) respectively:

$$\frac{\sigma_a}{S_e}+\frac{\sigma_m}{S_u}=1 \tag{1}$$

$$\frac{\sigma_a}{S_e}+\left(\frac{\sigma_m}{S_u}\right)^2=1 \tag{2}$$

where σ_a is alternating stress in the presence of mean stress, S_e is alternating stress for equivalent completely reversed loading, σ_m is mean stress and S_u is ultimate tensile strength [21,22].

4 NOTCH EFFCT COSIDERATION USING STRESS GRADIENT DESCRIBED IN THE FKM METHOD

The prediction of the fatigue life of components by calculation is a common step within the design process of machines [5,6,13]. Notch effect is the main detrimental factor on reducing fatigue life due to the existing of stress concentration near notch roots. Machine components usually contain stress raisers that are known as a notch. Due to high stress gradients around the notch root, there are more difficulties to solve the fatigue problem of such components compared to smooth specimens [17,18,21,22].

It is widely recognized that the stress gradient is of paramount importance for assessing fatigue strength in notched parts. In fact, knowledge of the stress gradient allows us to determine in a complete way the stress field at a notch root: knowledge of the stress concentration factor is not enough by itself to evaluate fatigue strength of notched components. Stress gradients have been shown to be quite important also for evaluating fatigue strength in the presence of a multiaxial stress field as has been shown by Munday and Mitchell. There are different theories which concern on fatigue failure of the notched specimens [21,22]. 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 [23,24,25].
In ncode Design Life software, an alternative approach has been implemented based on the stress gradient approach described in the FKM [26]. The FKM method describes a method in which the fatigue strength of a material is increased by a factor based on the surface normal stress gradient and the strength and type of material. There are several approaches to estimate the fatigue notch factor, among which FKM is recommended by the authors. According to FKM method, the correction factor can be calculated in dependence of relative stress gradient as follows:

For $\bar{G}_\sigma \le 0.1$

$$n_\sigma = 1 + \bar{G}_\sigma 10^{-(a_G - 0.5 + \frac{R_m}{b_G})} \tag{3}$$

For $0.1 \prec \bar{G}_\sigma \le 0.1$

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

For $1 \prec \bar{G}_\sigma \le 10$

$$n_\sigma = 1 + \sqrt[4]{\bar{G}_\sigma 10^{-(a_G - 0.5 + \frac{R_m}{b_G})}} \tag{5}$$

where n_σ is the correction factor, $\bar{G}_\sigma$ is the relative stress gradient and a_G and b_G are constants [24,25].

5 RESULT AND DISCUSSION

5.1 Thermal analysis

It is important to calculate the piston temperature distribution in order to control the thermal stresses and deformations within acceptable levels. The temperature distribution enables us to optimize the thermal aspects of the piston design at lower cost, before the first prototype is constructed [7,15]. The temperature $T\,(x, y, z, t)$ as a function of coordinate system parameters and time satisfies a parabolic differential equation, so called heat equation:

$$k_x \frac{\partial^2 T}{\partial x^2} + k_y \frac{\partial^2 T}{\partial y^2} + k_z \frac{\partial^2 T}{\partial z^2} + Q = \rho c_p \frac{\partial T}{\partial t} \tag{6}$$

where k is the thermal conductivity, $Q(x, y, z, t)$ is the source or sink rate of heat in a domain, ρ is density and c_p is the volumetric specific heat [9,20]. On the boundary the essential boundary condition on the boundary and the natural boundary condition can be defined, respectively, as:

$$T(x,y,z,t)=T_1(x,y,z,t) \tag{7}$$

$$k_n \frac{\partial T}{\partial t} + q_p + h(T - T_\infty) + \sigma\varepsilon(T^4 - T_\infty^4) = 0 \tag{8}$$

where k_n is the thermal conductivity normal to the surface, $q_p(x, y, z, t)$ is a prescribed flux, h is the heat transfer coefficient for convection, σ is Stefan–Boltzmann constant, ε is the emissivity and T_∞ is the ambient temperature for convection and/or radiation [20].

Piston thermal boundary conditions consist of the combustion side thermal boundary condition, upper ring land, lower ring land and skirt thermal boundary condition, underside thermal boundary condition, inside piston surface and piston pin thermal boundary condition [9,12,20]. In this way, the inner temperature was estimated to be 650°*C* with a convection coefficient of 800 W/m^2K. The upper ring land temperature of the piston was specified as 300°*C* with a convection coefficient of 230 W/m^2K. The lower ring land temperature of the piston is defined as 110°*C* with a convection coefficient of 200 W/m^2K. The piston skirt, piston inside surface and piston pin temperatures are defined as 85°*C* with convection coefficient of 60 W/m^2K [20].

The piston temperature distribution as in Fig. 2 changes between 237°*C* to 327.54°*C* with the maximum temperature at the piston crown center and minimum temperature at the lower part of the piston skirt. This figure indicates that the temperature distribution of the piston surface tends to decrease from the center to the edge of the piston. This situation is valid for the spark ignition engines at real working condition. This result is similar to that obtained on a similar type of piston in an earlier study [10].

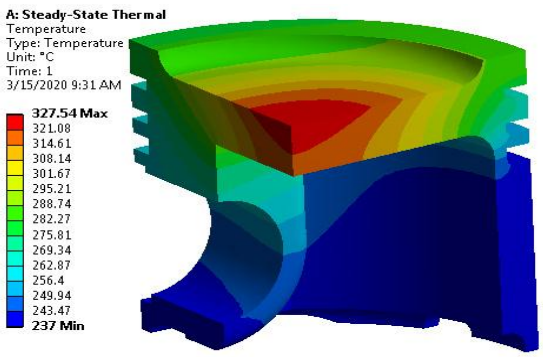


Fig.2
The temperature distribution in the piston.

5.2 Mechanical analysis

The piston bears the mechanical stress and withstands the thermal stress because of the change of temperature. Therefore, the analysis of thermo-mechanical coupling stress on the piston is needed. The loads on the piston include gas pressure, reciprocating inertial force, side pressure, thermal load and support reaction on the inner surface of the pin hole [1,4,6,14]. Among them, thermal load was the temperature field, which had been analyzed previously. The support reaction on the inner surface was instead of by the displacement constraint [3,4,6]. The gas pressure was loaded on the piston top, combustion chamber surface, field of fire and ring grooves. Since the gas pressure will reduce gradually after the piston rings, the explosion pressure imposed on the first ring groove is 75% of the total pressure, and 25% of the pressure is imposed between the first ring bank and the second ring groove. Gas pressure under the second ring groove is negligible [4]. Because the piston does the reciprocating motion in the cylinder, according to the dynamics of engine, this process could produce the reciprocating inertial force. Its value is proportional to the acceleration of the piston, but the orientation is opposite to the acceleration. The acceleration of reciprocating movement is:

$$a = r\omega^2\left(\cos\alpha + \lambda\cos 2\alpha\right) \tag{9}$$

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 [1,4]. If the engine is working, the piston does linear reciprocating movement along the cylinder. Since the piston skirt is in contact with the cylinder, it is subjected to a side thrust force due to the force of connecting rod. The solving formula of side thrust force is as follows:

$$F_C = \left(F_{gas} - F_i\right)\tan\beta \tag{10}$$

where F_c is the side thrust force of the piston, F_{gas} is the gas explosion pressure of the piston crown, F_j is the inertia force and β is the angular displacement of connecting rod [4]. The selection of the displacement boundary condition is very important to the finite element analysis. If the selection is not correct, it will affect the calculation precision [3,4,12]. On the moment of the maximum gas pressure, the pin contacted mainly to the upper surface of the pin hole, but the support reaction of the pin hole didn't act on the whole upper surface. In the cylindrical coordinate system, the displacements of *X, Y* directions of all nodes in the contact region were restricted. Moreover, by including the of cylinder model in the analysis, the displacements of *Z* direction on lateral side of piston pin boss were also restricted [12].

The analysis of the thermo-mechanical coupling stress is based on the results of the analysis of mechanical stress. The temperature distribution and the mechanical loads are taken into consideration at the same time. Import the calculated results of the piston temperature and impose the mechanical stress. Then finite element calculation is carried out and the results are studied. As shown in Fig. 3, under the condition of thermo-mechanical coupling, the maximum stress value in the piston is 57.456 *Mpa* and the position is at the upper portion of piston pin. Based on the work by Golbakhshi et al., the maximum value of the stress in the XU7JP/L3 engine piston is 50 *Mpa*. Comparing these results, proves a good agreement between thermo-mechanical analysis and simulated results carried out by Golbakhshi et al.

As it's shown in Fig. 3, the maximum stress is 57.456 *MPa*, which does not exceed the material yield strength. The second ring groove has a high stress, and the stress value is 37.57*Mpa*. The maximum stress on the third piston ring groove reaches 31.98*Mpa*, and the piston skirt have a small stress.

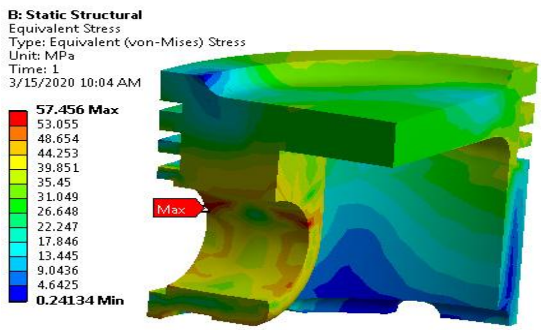


Fig.3
The Von-Mises stress distribution in the piston.

The contour of the safety factor (FS) in the piston is shown in Fig. 4. The safety factor in the piston is more than 1.1378. This indicates that the design of piston can meet the life limitation. As it can be seen from Figs. 3 and 4, the area where the minimum FS is occurred is where the maximum stress is predicted. This corresponds to the results by He et al. [1].

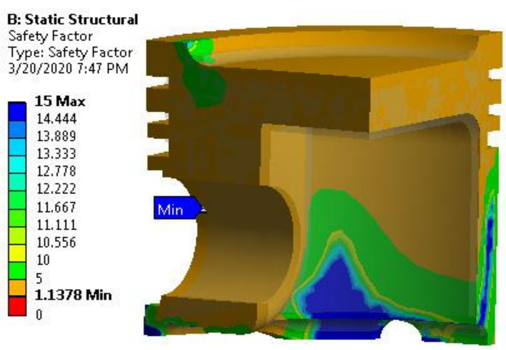


Fig.4
The safety factor distribution in the piston.

5.3 HCF life prediction using stress gradient approach described in the FKM method

From above the thermo-mechanical coupling analyzes, it can be drawn that the maximum stress concentration occurs at the upper portion of piston pin. Although it doesn't exceed the yield strength of the material, the thermo-mechanical fatigue breakdown would most likely occur at the place. Therefore, it is necessary to carry out the thermo-mechanical fatigue checkout for this piston [1,6,13]. For this purpose, fatigue tests are usually performed on the specific fatigue machine. But they are complex, high cost and time-consuming [1,18]. In this paper, the HCF prediction, based on the Goodman and Gerber equations, is conducted to calculate the fatigue life instead the experimental fatigue tests.

In order to study the fatigue life of the piston based on HCF approach, the stresses histories were fed into the nCode Design Life software. The dominant fatigue mode was found to be HCF in this article as the number of cycles is relatively high and the maximum stress obtained did not exceed the yield stress of the material. Therefore, HCF method was used for evaluation fatigue life in the current work.

Fig. 5 represents the number of cycles to failure based on Goodman criterion. As it can be seen from this Figure, minimum life (4.752E8 cycles) has been determined at the critical zone of the piston. In Fig. 6, the number of cycles to failure based on Gerber equation is shown. Minimum life (4.808E8 cycles) has been predicted at the critical area of the piston. As it can be seen from Figs. 5 and 6, the number of cycles to failure in the critical areas is above 10^4 or 10^5 which imposes HCF for the piston material [21,22].

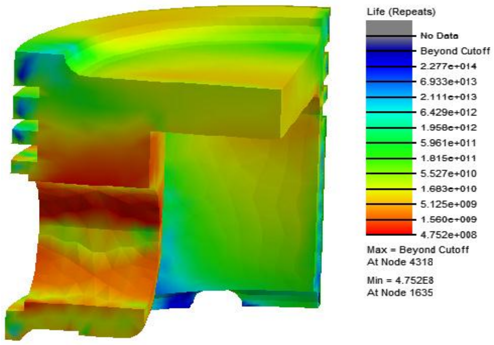


Fig.5
The number of cycles to failure based on Goodman criterion.

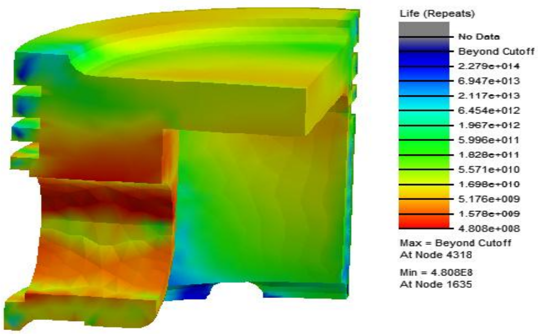


Fig.6
The number of cycles to failure based on Gerber equation.

Figs. 7 and 8 illustrates the fatigue damage for the piston based on Goodman and Gerber criterions, respectively. As it can be seen from these Figures, maximum fatigue damage occurs as 6.408E-10 and 6.339E-10 at the critical zones. If the value of the fatigue damage is greater than 1, it indicates that the part will fail from fatigue before the design life is reached [26]. Therefore, piston is under HCF. This corresponds to the number of cycles to failure results.

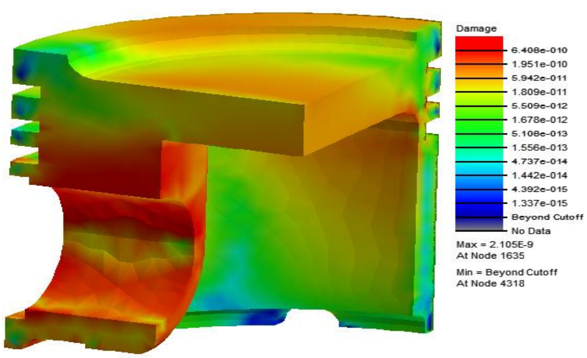


Fig.7
The fatigue damage based on Goodman criterion.

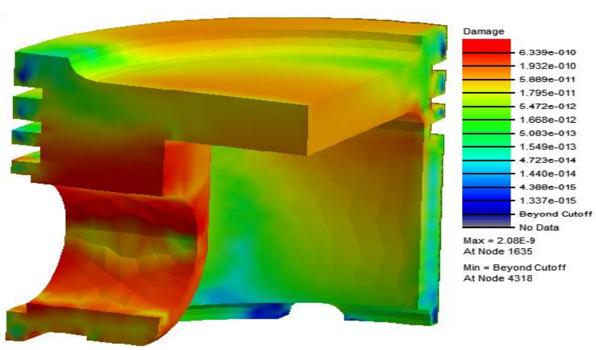


Fig.8
The fatigue damage based on Gerber equation.

As observed in Figs. 5 to 8, due to the thermo-mechanical stresses, there are mainly two critical areas: upper portion of piston pin and piston compression grooves. Subsequently will be presented different engine pistons where the cracks initiated on those areas. Stress analyses on the piston showed the same critical areas. Fig. 9 depicts a piston which has been cracked in the region of upper portion of piston pin. Another typical fatigue damage occurs on piston compression grooves. Fig. 10 shows one damaged Piston in this area. Comparison of these Figures with Figs. 5 to 8 concludes that the HCF results have a good agreement with the real samples.

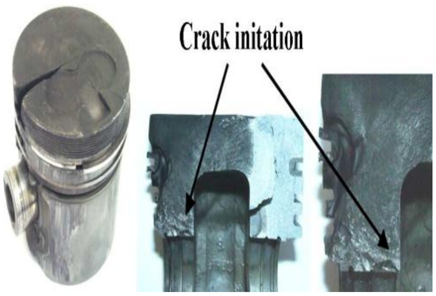


Fig.9
A cracked engine piston in upper portion of piston pin [27].

Fig.10
A damaged engine piston in area of rings groove [27].

6 CONCLUSION

In this study HCF life prediction for an aluminum alloy piston using stress gradient approach described in the FKM method is investigated. Finite element analysis provides accurate and reliable prediction of temperature and fatigue life results in the piston. The numerical results showed that the temperature maximum occurred at the piston crown center. The Obtained FEA results showed that the stress and number of cycles to failure have the most critical values at the upper portion of piston pin and piston compression grooves. To evaluate properly of results, stress analysis and HCF 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. The lifetime of this part can be determined through finite element analysis instead of experimental tests. Computer aided engineering plays an important role to find the weakness of a piston layout at the early stage of the engine development. In order to prevent piston cracking it is recommended to modify geometry of material in crucial parts. TBC might also be used in the regions which not only reduce temperature, but also increase the fatigue life of piston. Since they reduce thermal stress, fatigue life of the piston grows.

REFERENCES

[1] He T., Lu X., Zou D., Guo Y, Li W., Huang M., 2014, Thermo-mechanical fatigue life prediction for a marine diesel engine piston considering ring dynamics, *Journal of Advances in Mechanical Engineering* **6**: 429637.

[2] Reyes Belmonte M.A., Copeland C.D., Hopkins G., Schmieder A., Bredda S., Akehurst S., 2015, Improving heat rtransfer and reducing mass in a gasoline piston using additive manufacturing, *SAE Technical Paper* **2015**(01): 0505.

[3] Lu Y., Zhang X., Xiang P., Dong D., 2016, Analysis of thermal temperature fields and thermal stress under steady temperature field of diesel engine piston, *Journal of Applied Thermal Engineering* **113**: 796-812.

[4] Liu X.F., 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* **119**: 312-318.

[5] Caldera M., Massone J.M., Martı´nez R.A., 2017, Failure analysis of a damaged direct injection diesel engine piston, *Journal of Failure Analysis and Prevention* **17**: 979-988.
[6] 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* **19**: 402-411.
[7] Mancaruso E., Sequino L., Vaglieco B.M., 2019, 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* **2019**(24): 0182.
[8] Nouby M., Ghazaly K., Abd El-Gwwad A., 2019, Evaluation of gasoline engine piston with various coating materials using finite element method, *International Journal of Automotive Engineering* **9**(2): 2942-2948.
[9] Yao Z., Qian Z., 2018, Thermal analysis of nano ceramic coated piston used in natural gas engine, *Journal of Alloys and Compounds* **768**: 441-450.
[10] Dudarevaa N.Y, Enikeeva R.D., Ivanov V.Y., 2017, Thermal protection of internal combustion engines pistons, *Journal of Procedia Engineering* **206**: 1382-1387.
[11] Changming H., Sichuan X., 2016, Multi-Body dynamic simulation and fatigue analysis of the unique crank-train for a creative two-stroke opposed piston diesel engine, *SAE Technical Paper* **2016**(01): 2332.
[12] Golbakhshi H., Namjoo M., Dowlati M., Khoshnam F., 2016, Evaluating the coupled thermo-mechanical stresses for an aluminum alloy piston used in a gasoline engine XU7, *The Journal of Engine Research* **42**: 33-41.
[13] Shariyat M., Fathi Sola J., Jazayeri S.A., 2016, Experimentally validated combustion and piston fatigue life evaluation procedures for the bi-fuel engines, using an integral-type fatigue, *Latin American Journal of Solids and Structures* **13**: 1030-1053.
[14] Kakaee A.H., Gharloghi J., Foroughifar A., KhanlariA., 2015, Thermo-mechanical analysis of an SI engine piston using different boundary condition treatments, *Journal of Centeral South University* **22**: 3817-3829.
[15] Giacopini M., Sissa S., Rosi R., Fantoni S., 2015, Influence of different temperature distributions on the fatigue life of a motorcycle piston, *Journal of Automobile Engineering* **229**(9): 1276-1288.
[16] Sharma D., Abdulaziz A., Mohammad A., Marwan A.A, Siniawski M.T., 2014, Thermal and mechanical failure analysis of a two-stroke motocross engine piston, *Journal of Failure Analysis and Prevention* **14**: 21-26.
[17] Ayatollahi M.R, Mohammadi F., Chamani H., 2013, Notch effects on high cycle fatigue life of engine components, *7th International Conference on Internal Combustion Engines*, Iran.
[18] Carpenter M., Jha P., Ray S., Nienhuis M.D., 2019, Fatigue tests of un-notched and notched specimens and life prediction using a variable critical distance method, *SAE Technical Paper* **2019**(01): 0801.
[19] 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.
[20] Durat M., Kapsiz M., Nart E., Ficici F., Parlak A., 2012, The effects of coating materials in spark ignition engine design, *Journal of Materials and Design* **36**: 540-545.
[21] Stephens R., Fatemi A., Fuchs H., 2001, *Metal Fatigue in Engineering*, John Wiley.
[22] Lee Y.L, Pan J., Hathaway R.B., Barkey M.E., 2005, *Fatigue Testing and Analysis: Theory and Practice*, Elsevier Butterworth-Heinemann.
[23] McKelvey S.A, Lee Y.L, Barkey M.E., 2012, Stress-based uniaxial fatigue analysis using methods described in FKM-guideline, *Journal of Failure Analysis and Prevention* **12**: 445-484.
[24] Lee Y.L, Barkey M., Kang H.T., 2012, *Metal Fatigue Analysis Handbook: Practical Problem-Solving Techniques for Computer-Aided Engineering*, Elsevier.
[25] Haibach E., 2003, *FKM-Guideline: Analytical Strength Assessment of Components in Mechanical Engineering*, VDMA Verlag.
[26] Ansys (V.18), 2018, User᾽s Manual.
[27] Silva, F.S., 2006, Fatigue on engine pistons – a compendium of case studies, *Journal of Engineering Failure Analysis*, **13**: 480-492.