Improving High Cycle Fatigue Life in A Gasoline Engine Piston using Oil Gallery with Considering Stress Gradient

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Abstract: Fatigue due to thermo-mechanical stresses plays an effective role in causing damage and reducing piston fatigue life. The effect of oil gallery on the thermal stress and High Cycle Fatigue (HCF) life in a gasoline engine piston using oil gallery with considering stress gradient was investigated. For this purpose, coupled thermo-mechanical analysis of a gasoline engine piston was carried out. Then HCF life of the component was predicted using a standard stress-life analysis and results were compared to the original piston. 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. The obtained thermo-mechanical analysis results proved that the oil gallery reduces the stress distribution in the piston about 7MPa and 12MPa at engine speed 1000rpm and 5000rpm, respectively. The results of high cycle fatigue life showed that the number of cycles of failure for modified piston is approximately 33% and 37% higher than original piston at 1000rpm and 5000rpm, respectively. To evaluate properly of results, stress analysis and high cycle fatigue results is compared with real sample of damaged piston and it has been shown that critical identified areas, match well with areas of failure in the real sample.

Keywords: Gasoline Engine Piston, High Cycle Fatigue Life, Oil Gallery, Stress Gradient

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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 [1-3]. 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 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 [1], [5]. 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], [4-5]. Different cooling methods result in different distributions of heat flux in pistons. Traditional spray cooling and free jet cooling cannot fully meet the new requirement of highly intensified piston cooling. The most effective way is the employment of cooling gallery inside the piston head, which is able to significantly reduce the heat loading of pistons [3], [6]. When the engine is running, the cooling oil is injected from oil jet nozzle into the gallery through the inlet hole, flows around in the circumferential direction and exits the gallery through the outlet hole back into the crankcase, as illustrated in “Fig.1(d)”. The heat flux removed by the cooling gallery can occupy as much as 60%~70% of the total thermal energy passed to the piston from the combustion gases, which avoids the possibility of piston instability being induced by large thermal stresses during engine operation [7-8].

In recent years, considerable attention has been focused on piston gallery design all over the world in order to meet the requirements of lower emissions and higher power density of internal combustion engines [3], [9-10]. Various experimental and numerical studies on heat transfer performance of cooling galleries have been conducted. Zhu et al. performed a simulation and optimization on oscillating cooling characteristics in high-enhanced piston oil Cooling gallery. The calculation showed that the error between optimal results and calculating result is less than 1% [6]. Effect of cooling gallery on the piston temperature in a gasoline direction injection engine was studied by Han et al. Their analysis confirmed that the cooling gallery played an important role in determining the piston temperature [11]. Wang et al. conducted a simulation on flow and heat transfer characteristics of engine oil inside the piston cooling gallery. Their simulation proved that the optimal jet velocity of cooling oil should be slightly higher than the maximum speed of engine piston during the reciprocating motion [12].

Luff et al. studied the effect of piston cooling jets on diesel engine piston temperatures, emissions and fuel consumption. The use of piston cooling jets reduced piston temperature by up to 80ºC. This cooling had little effect on HC but produced a small reduction in engine-out emissions of NOx at the expense of an increase in CO [9]. Numerical investigation on the oscillating flow and uneven heat transfer processes of the cooling oil inside a piston gallery was investigated by Deng et al. The results revealed that the instantaneous oil charge ratio decreases and the area-weighted heat transfer coefficient increases as the engine speed increases [13]. 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 oil filling ratio match [14]. Optimization of the location of the oil cooling gallery in the diesel engine piston was investigated by Deng et al. The research showed that the best position of oil cooling gallery is where the distance of the oil cooling gallery and the piston top surface is 12.5mm [4]. Peng et al. performed a numerical study on the flow and heat transfer process nano-fluids inside a piston cooling gallery. The numerical simulation results in general corresponded well with the empirical formula [7].

Multi-objective optimization of cooling galleries inside pistons of a diesel engine was carried out by Deng et al. Their research uncovered the fact that the gallery cross section should stay far away from the grooves and close to the bottom of combustion chamber along with the top region of inner chamber [3]. In another attempt, heat transfer coefficient in piston cooling galleries was examined by Binder et al. A good correlation between experimental and simulated results was shown [10]. Wang et al. investigated a numerical study on cooling process of conventional engine oil and nano-oil inside the piston gallery. The results demonstrated that the nano-oil is able to improve the heat transfer capacity by a large margin [8].

Based on the reviewed literature, the effect of oil gallery on the fatigue life of the piston has not been studied yet. The oil gallery declines the temperature distribution in the piston. The lower temperature of the piston, the less thermo-mechanical stress. Thus, fatigue life of the piston will increase. Therefore, it is necessary to study the effect of oil gallery on the high cycle fatigue life of the piston. Therefore, the aim of this study was to evaluate the effect of oil gallery on the high cycle fatigue life of XU7JP/L3 engine piston. For this purpose, first Solidworks software was used to model the XU7JP/L3 engine piston. An oil gallery was also created on the piston head. Then ANSYS software was used to determine temperature and thermo-mechanical stress distribution of the piston. Finally, in order to study the high cycle fatigue life of the piston, the results were fed into the ANSYS nCode Design Life software. 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 [15-16]. In this study, the notch effect is considered based on the stress gradient approach described in the FKM method. The effect of engine speed on thermo-mechanical stress and fatigue life of the piston is also investigated in this work.

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 are impossible. Many researchers have used finite element method to obtain stress-strain distribution in geometrically complex components [1], [4-5].

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 [17]. The characteristics of the engine under study are summarized in “Table 1”.

Table 1 Specification of the engine under study [18]

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

Due to the symmetrical structure of the piston, a 1/4 3D solid model was created [1], [3], [19]. The piston analyzed in this article is shown in “Fig. 1”. Piston is made of AlSi alloy with a thermal conductivity 155W/mm°C, Young's modulus of 90GPa, a Poisson's ratio of 0.3, and a coefficient of thermal expansion of 21×10^{-6} per °C [17]. 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.

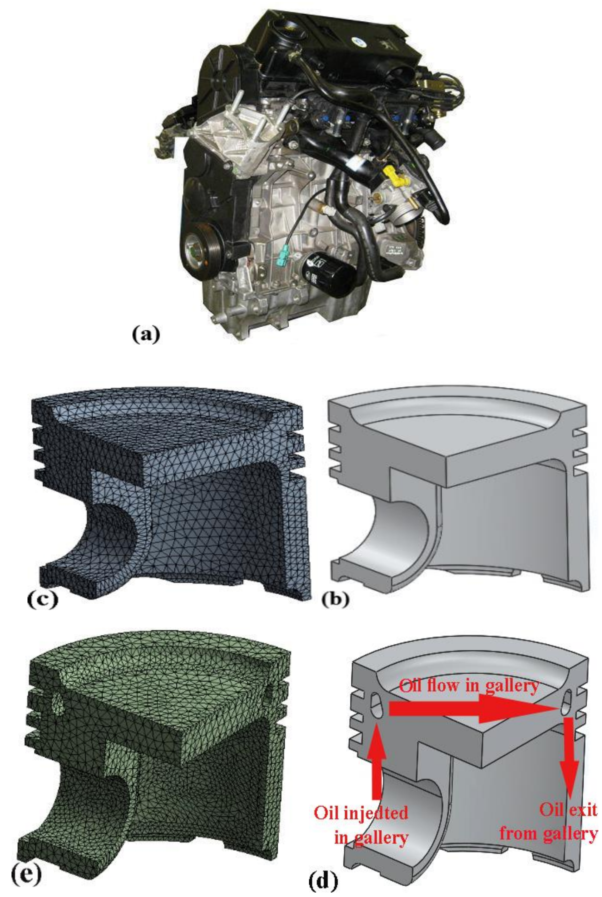

Fig. 1 (a): XU7JP/L3 gasoline engine, (b): The piston generated by SolidWorks, (c): Finite element model of the piston, (d): Modified piston with oil gallery, and (e): Finite element model of the modified piston.

The differential equation of time dependent heat flow is given in polar coordinate by [17], [19]:

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

For the steady state thermal analysis of the piston, the major mechanism for heat transfer is convection. In this paper, the fluid temperature T_f and the convection coefficient h are considered as the boundary conditions of the problem. These conditions can be mathematically expressed as follows [17], [19]:

$$-k\frac{\partial T}{\partial n} = h(T-T_f) \quad (2)$$

Since 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 [1], [20]:

$$a=-r\omega^2(\cos\alpha+\lambda\cos2\alpha) \quad (3)$$

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 [20]:

$$F_c=(F_{gas}-F_j)\tan\beta \quad (4)$$

The prediction of the fatigue life of components by calculation is a common step within the design process of machines [2], [5]. 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 [15-16], [21]. It is widely recognized that the stress gradient is of paramount importance for assessing fatigue strength in notched parts. 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 [22-23].

In ncode Design Life software, an alternative approach has been implemented based on the stress gradient approach described in the FKM [24]. 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 $\overline{G}_\sigma \leq 0.1$

$$n_\sigma=1+\overline{G}_\sigma 10^{-(a_G-0.5+\frac{R_m}{b_G})} \quad (5)$$

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

$$n_\sigma=1+\sqrt{\overline{G}_\sigma} 10^{-(a_G-0.5+\frac{R_m}{b_G})} \quad (6)$$

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

$$n_\sigma=1+\sqrt[4]{\overline{G}_\sigma} 10^{-(a_G-0.5+\frac{R_m}{b_G})} \quad (7)$$

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 [20], [25].

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 equation to account for mean stress effects. Goodman criterion is used to evaluate the fatigue life of aluminum alloy piston [4], [26-27]. The Goodman equation is given by Equation [21]:

$$\frac{\sigma_a}{S_e}+\frac{\sigma_m}{S_u}=1 \quad (8)$$

3 RESULTS AND DISCUSSION

3.1. 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 [3], [25]. 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. In this way, the inner temperature was estimated to be 650°C with a convection coefficient of 800 W/m^{2}K. The upper ring land temperature of the piston was specified as 300°C with a convection coefficient of 230 W/m^{2}K. The lower ring land temperature of the piston is defined as 110°C with a convection coefficient of 200 W/m^{2}K. The piston skirt, piston inside surface and piston pin temperatures are defined as 85°C with convection coefficient of 60 W/m^{2}K [17], [19], [28].

The resulting temperature distributions on the original piston are given in "Figs. 2 and 3". The piston temperature distribution as in "Figs. 2 and 3" changes between 121.34°C to 61.256°C and 190.44°C to 112.23°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 [29-30].

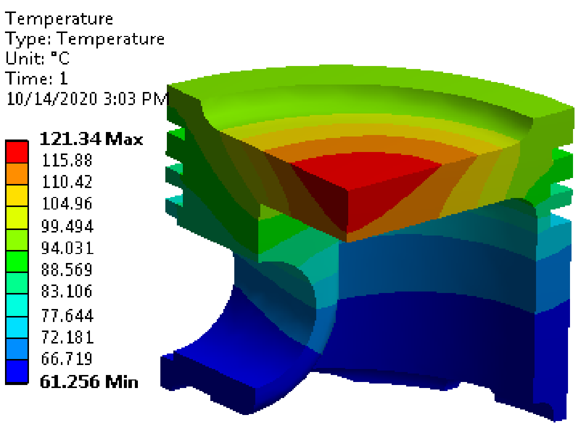

Fig. 2 The temperature distribution in the original piston at engine speed of 1000rpm.

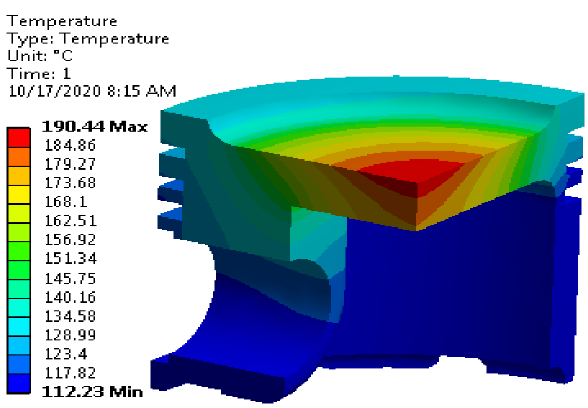

Fig. 3 The temperature distribution in the original piston at engine speed of 5000rpm.

The resulting temperature distributions on the modified piston are shown in "Figs. 4 and 5" As seen in "Figs. 4 and 5", the oil gallery reduces the temperature distribution in the piston about 8°C and 11°C at engine speed of 1000rpm and 5000rpm, respectively. It is clear that the oil gallery causes the piston temperature to decrease as the engine speed increases. This is corresponding to the result by [12].

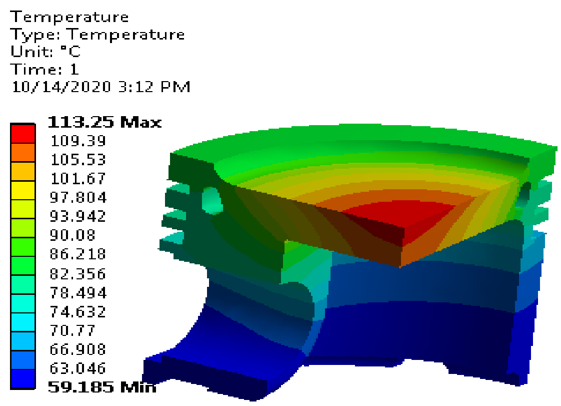

Fig. 4 The temperature distribution in the modified piston at engine speed of 1000rpm.

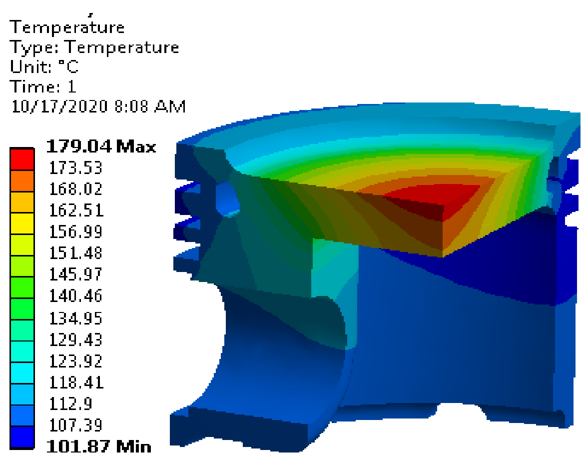

Fig. 5 The temperature distribution in the modified piston at engine speed of 5000rpm.

3.2. 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], [5], [20]. Among them, thermal load was the temperature field, which had been analyzed previously. The support reaction on the inner surface was instead of the displacement constraint [5], [20], [26]. 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 [20]. 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 [17], [20] [,26]. Figure 6 shows the structural boundary conditions applied to finite element model of piston for structural analysis. Lateral planes are fixed in their normal directions [26], [29].

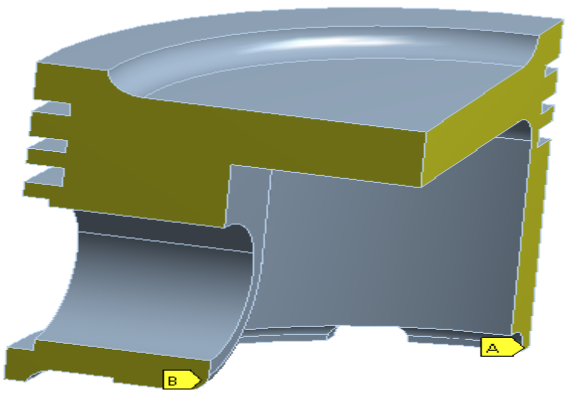

Fig. 6 Structural boundary conditions.

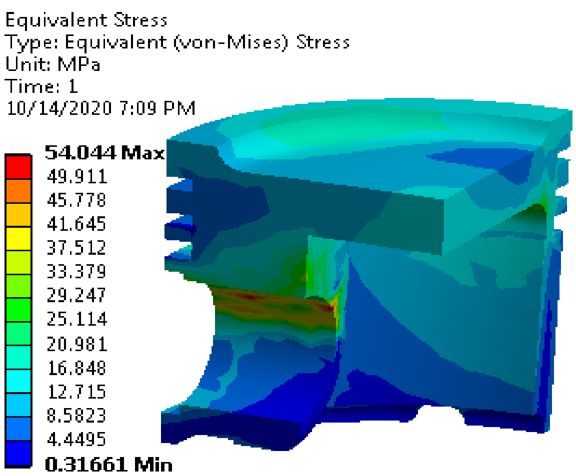

Fig. 7 The Von-Mises stress distribution in the original piston at engine speed of 1000rpm.

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. The calculated results of the piston temperature are imported and the mechanical stress is imposed. Then finite element calculation is carried out and the results are studied. The Von-Mises stress distributions in the original piston are exhibited in “Figs. 7 and 8”. Stress contour results for the modified piston are presented in “Figs. 9 and 10”. Although the maximum temperature occurs in the piston crown center, this area in not critical.

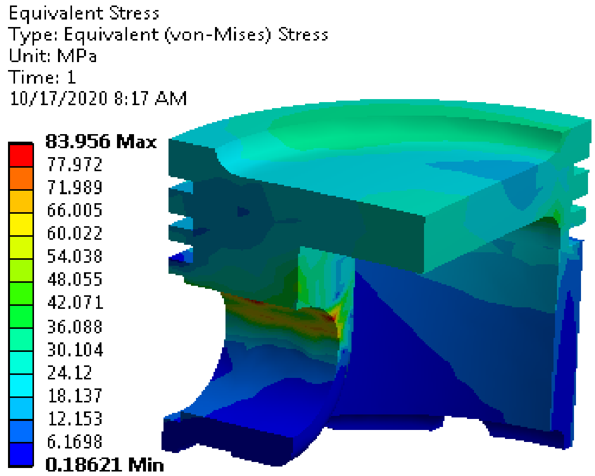

Fig. 8 The Von-Mises stress distribution in the original piston at engine speed of 5000rpm.

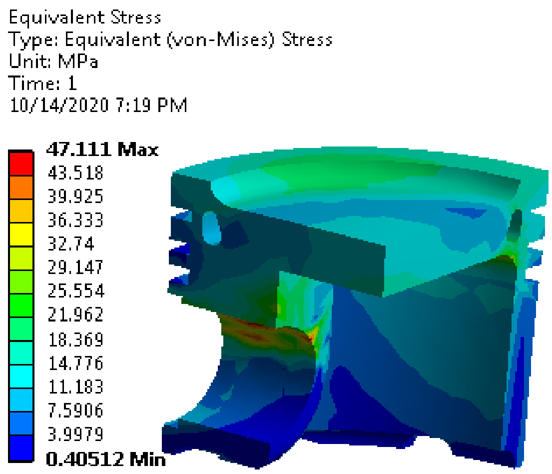

Fig. 9 The Von-Mises stress distribution in the modified piston at engine speed of 1000rpm.

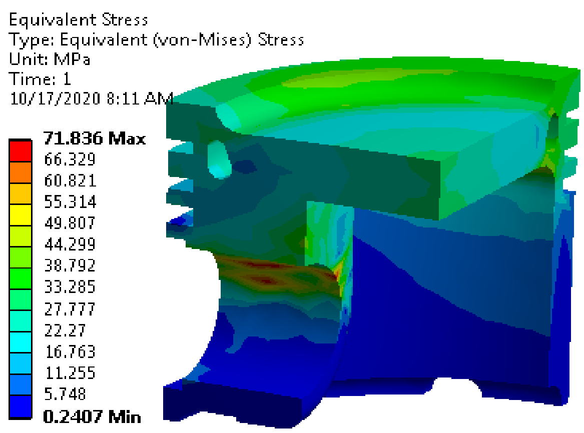

Fig. 10 The Von-Mises stress distribution in the modified piston at engine speed of 5000rpm.

As shown in stress contours, maximum stress occurs at the upper portion of piston pin. This corresponds to the results by [17], [20], [31]. As seen in “Figs. 9 and 10”, the oil gallery reduces the stress distribution in the piston about 7MPa and 12MPa at engine speed of 1000rpm and 5000rpm, respectively.

3.3. Verification of FEA Results

As shown in "Fig. 7", under the condition of thermo-mechanical coupling, the maximum stress value in the piston is 54.044Mpa 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 50Mpa. Comparing this result, proves a good agreement between thermo-mechanical analysis and simulated results carried out by Golbakhshi et al. The numerical simulation results for the original piston are compared on "Table 2" with results found on previous work from the literature survey [32]. As it observed from "Table 2", there is a good compromise between FEA results and data observed on the literature survey [32].

Table 1 Comparison of the FEA results and Source [32]

Original piston	Numerical simulation	Source[32]
Maximum temperature at engine speed of 1000 rpm	121.34°C	124°C
Maximum temperature at engine speed of 5000 rpm	190.44°C	184.2°C
Maximum stress at engine speed of 5000 rpm	55.044MPa	55.9MPa
Maximum stress at engine speed of 5000 rpm	83.959MPa	78.6MPa

3.4. HCF Life Prediction Using Stress Gradient Approach Described in the FKM Method

From the above thermo-mechanical coupling analyzes, it can be drawn that the maximum stress concentration occurs at the upper portion of piston pin. Although it does not 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], [5]. For this purpose, fatigue tests are usually performed on the specific fatigue machine, but they are complex, high cost and time-consuming [1], [33]. In this paper, the HCF prediction, based on the Goodman equation, 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. Figures 11 and 12 represents the number of cycles to failure in the original piston. In "Figs. 13 and 14", the number of cycles to failure in the modified piston is shown. As it can be seen from "Figs. 11 to 14", the number of cycles to failure in the critical areas is above 10^4 or 10^5 which imposes HCF for the piston material [5], [23]. As shown in failure contours, minimum HCF life occurs at the upper portion of piston pin. This corresponds to the results by [5], [26]. The oil gallery increases the fatigue life of the piston about 33% and 37% at 1000rpm and 5000rpm, respectively.

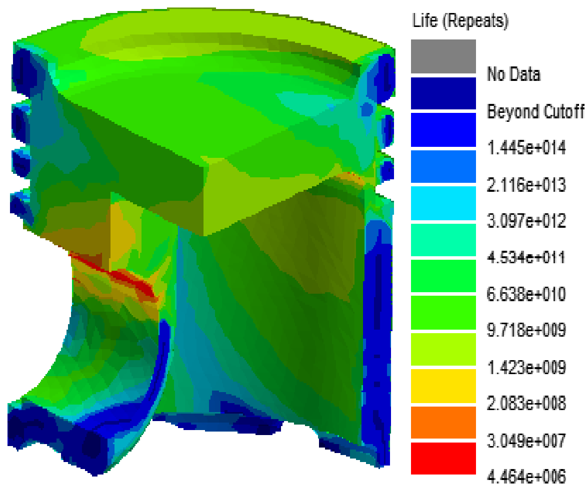

Fig. 11 The number of cycles to failure in the original piston at engine speed of 1000rpm.

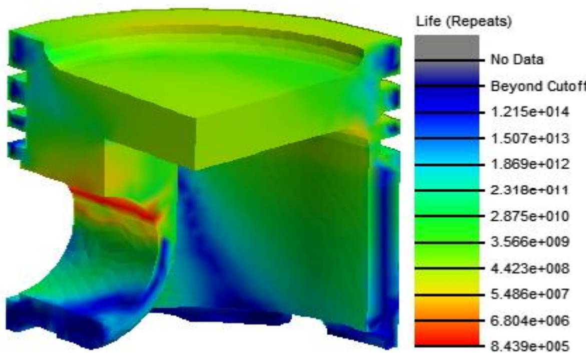

Fig. 12 The number of cycles to failure in the original piston at engine speed of 5000rpm.

As observed in "Figs. 7 to 14", 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. Figure 15 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. Figure 16 shows one damaged Piston in this area. Comparison of these Figures with "Figs. 7 to 14" concludes that the HCF results have a good agreement with the real samples.

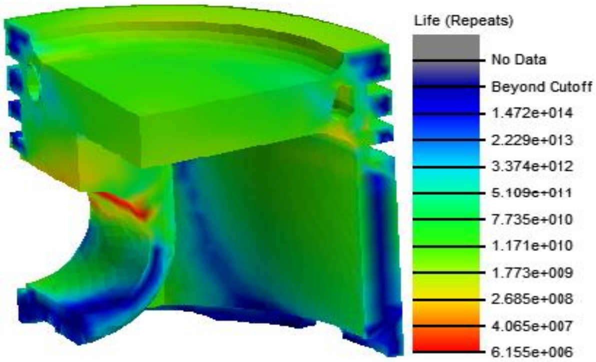

Fig. 13 The number of cycles to failure in the modified piston at engine speed of 1000rpm.

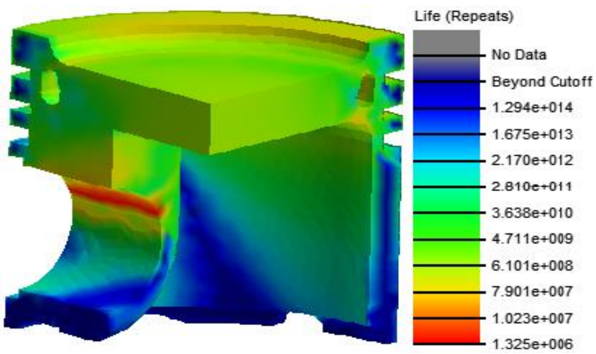

Fig. 14 The number of cycles to failure in the modified piston at engine speed of 5000rpm.

Fig. 15 A cracked engine piston in upper portion of piston pin [34].

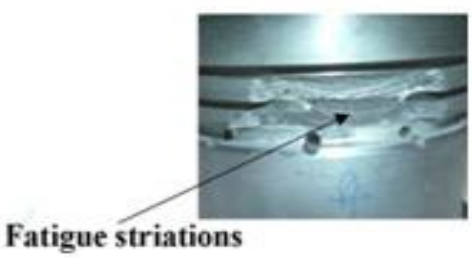

Fig. 16 A cracked engine piston in upper portion of piston pin [34].

4 CONCLUSION

The aim of this study is to investigate the effect of oil gallery on the thermal stress and HCF life in a gasoline engine piston with considering stress gradient. The numerical results showed that the temperature maximum occurred at the piston crown center. The results of FEA demonstrated that the temperature distribution in the modified piston dwindles about 8°C and 11°C at engine speed of 1000rpm and 5000rpm, respectively. Therefore, the piston endures less temperature and fatigue life will increase.

The thermo-mechanical analysis proved that Von-Mises stress decreases about 7MPa and 12MPa at engine speed of 1000rpm and 5000rpm, respectively, which can lead to higher fatigue lifetime. 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.

The results of high cycle fatigue life showed that the number of cycles of failure for modified piston is approximately 33% and 37% higher than original piston at 1000rpm and 5000rpm, respectively. To evaluate the results properly, 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. Computer aided engineering plays an important role to find the weakness of a piston layout at the early stage of the engine development.

5 NOMENCLATURE

k	Thermal conductivity
α	Thermal diffusivity
r	Radius distance
T	Temperature
n	Exterior normal vector
a	Piston acceleration
ω	Rotating speed of engine
λ	Ratio of crank radius to the length of connecting
F_c	Side thrust force of the piston
F_{gas}	Gas pressure of the piston crown
Fj	Inertia force
β	Angular displacement of connecting rod
n_σ	Correction factor
$\overline{G}_\sigma$	Relative stress gradient
a_G	Material constant
b_G	Material constant
R_m	Ultimate tensile stress

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