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Research article

Low cycle fatigue prediction for cylinder head considering notch stress-strain correction proposed by Neuber

Hojjat Ashouri*

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

*ashouri1394@gmail.com

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Abstract

Due to the complex geometry and loading conditions, engines cylinder heads are the most challenging components among all parts internal combustion engines. They must withstand severe cyclic thermo-mechanical loading throughout their lifetime. Low cycle fatigue (LCF) prediction for cylinder head considering notch stress-strain correction proposed by Neuber was investigated. For this purpose, first Solidworks software was used to model the cylinder head. Then Ansys Workbench software was used to determine temperature and stress distribution of the cylinder head. Finally, in order to study the fatigue life based on LCF approach, the results were fed into the nCode Design Life software. The thermo-mechanical analysis showed that the maximum temperature and stress happen in the valves bridge between the two exhaust valves. The results of the FEA correspond with experimental tests performed by researchers, and demonstrated the cylinder heads cracked in this region. The numerical results showed that the area where the maximum temperature and stress is occurred is where the least LCF is predicted.

Keywords: Thermo-mechanical fatigue, Low cycle fatigue, Cylinder head, Neuber method

1- Introduction

One of the main durability of internal combustion engines is cylinder head cracks at valve bridge region. With the rising demand for engine efficiency, emissions standard, increased combustion pressure and elevated temperature make engine cylinder head to be the most critical and complicated part for engine design [1-4]. In engines, cylinder heads are exposed two type of the fatigue loading. The combustion pressure causes mechanical loads, which relates to high cycle fatigue (HCF) behavior of the cylinder head material. The start-stop cycles of engine cause thermal loads, which relate to the low cycle fatigue behavior of the cylinder head material [1, 4,7]. According to these combined thermo-mechanical loadings, the cylinder head material should have some properties such as high resistance to deformation and also high toughness at high temperature to resist fatigue cracking. Thermo-mechanical fatigue (TMF) leading to cracks and possible failure in cylinder head is usually the limiting factor in the conception of new designs [2, 3, 5, 8].

Fatigue fracture of components usually initiates at the stress concentrations, where local stresses and strains are higher than nominal ones and often exceed yielding. Therefore, it is important to accurately estimate the local elastic-plastic stresses and strains at notches. From the linear-elastic finite element analyses and the stress superposition principle, the linear stresses must be corrected using the Neuber equation for the elastic-plastic strain effects [7,9,10].

In the literature, previous studies report several researches related to the stress analysis and fatigue life in the cylinder heads. Ashouri studied fatigue cracks of cylinder heads in diesel engines using the two-layer viscoplasticity model. His simulation showed that the maximum temperature and stress occur in the valve bridge [11]. Evaluation of viscosity effects on thermo-mechanical analysis for cylinder heads was performed by the two-layer viscoplasticity model. The obtained finite element analysis (FEA) results showed that the viscosity strain is more than plastic strain which is not negligible [12]. Chen et al. established the simulation approach for the fatigue life assessment of cylinder heads with integrated exhaust manifolds. Their research proved an acceptable between experimental and simulation results [6]. Thermo-mechanical analysis of a coated cylinder heads via the two-layer viscoplasticity model was done by Ashouri. His study disclosed that thermal barrier coatings decrease distribution of temperature and stress in the cylinder heads [13]. A new fatigue life model for copper aluminum-silicon alloys was presented by Beranger et al. For the critical regions, neglecting ageing effect leads to an acceptable between experimental and simulation results [14]. Satyanarayana et al. used stress analysis to optimization the variable compression ratio of diesel engines cylinder heads. Their simulation showed that the optimum thickness of the part body is 15 mm [15]. Failure analysis of the cast cylinder heads was performed by Jing et al. Their research verified that the failure of the cylinder heads is mainly caused by the thermal fatigue [16]. In another attempt, Fonte et al. analyzed cracked cylinder head studs. Their research revealed that the main reason for crack initiation in studs is high stress concentration at second thread root of the studs [17]. Seifert et al. predicted fatigue life of aluminum cylinder heads considering ageing effects. Their research proved that ageing plays a significant role in the thermo-mechanical fatigue [3]. A complete simulation and analysis process of cylinder heads TMF was carried out by Zeng et al. The TMF analysis proved that the lowest thermo-mechanical fatigue occurs in the intake-exhaust valve bridge [8]. Wang et al. predicted TMF of turbocharged engines cylinder heads. Their simulation showed that the location with low safety factor and TMF is in accord with the cracking location in the experimental tests [5]. Assessing TMF of a aluminum cylinder head was done by Liu et al. Their research proved that the damage due to creep is minimal and can be neglected [2]. Pingale et al. developed a finite element method and HCF to analysis the failure of cylinder heads. Their simulation revealed that the difference between experimental and simulation results is less than 11% [18]. Fatigue life prediction of diesel engines cylinder heads based on thermal fluid solid coupling model was done by Zhang et al. According to the their research the gas pressure and HCF are the dominated factors affecting the fatigue life of the engines cylinder heads [1]. Ashouri performed thermo-mechanical analysis of magnesium alloy diesel engines cylinder heads using a two-layer viscoplasticity model. His study showed that the magnesium cylinder heads tolerate less tensile and compressive cyclic stress compared to the aluminum cylinder heads [19]. Dynamic optimization of load step transient response of a turbocharged spark ignition engine focusing on valves timing was performed by Keshavarz and Keshavarzi. Their study showed that at the end of transient and by closing of engine load to its full load, the valves degrees become constant, approximately [20]. Ren et al. predicted high cycle fatigue failure analysis of cast Al-Si alloy engine cylinder head. The fracture surface proved that several main cracks initiated from large defects near the water jacket surface [21]. Application of scaled specimens in evaluating thermal fatigue performance of cylinder Head was studied by et al. Their simulation showed that thermal fatigue failure occurs on the nose bridge between two exhaust holes and on the nose bridge between the intake and exhaust holes after 80 cycles [22]. Mahajan et al. did optimization of scallop design for cylinder head of a multi-cylinder diesel engine for reduction of combustion deck temperatures and simultaneously enhancing combustion deck fatigue margin. Maximum reduction in water jacket HCF fatigue margin was about 6.5% and combustion deck fatigue margin was enhanced up to 6% [23].

In the literature although lots of paper focused on stress analysis and fatigue life prediction of cylinder heads, there is a lack of science in the field of studying stress analysis and fatigue life prediction in engines cylinder heads considering the notch stress-strain correction. Thermo-mechanical notch stresses in the areas of fatigue cracking exceed the material yield stress and consideration of elastic-plastic material response is crucial for proper prediction of fatigue life. [7, 9, 10]. Notch effect is the main detrimental factor on reducing fatigue life due to the existing of stress concentration near notch roots [7, 9, 10, 20]. Therefore, the aim of this paper is to predict the stress analysis and fatigue life prediction considering notch stress-strain correction expressed by Neuber. For this purpose, Solidworks software was used to model the cylinder heads. Then the thermo-mechanical analysis was performed to get the temperature and stress distribution in ANSYS Workbench software. Finally, the fatigue life prediction that considers stress-strain correction was done.

2- Methodology

2-1 The finite element model and behavioral model

The material used for the engines cylinder heads is aluminum alloy A356.0. This alloy is applied in engines cylinder heads [2, 3, 5, 6]. Finite element analysis provides accurate and reliable prediction of temperature and fatigue life prediction in the engine cylinder heads. FEA allows design engineers to identify engines cylinder heads weakness at the early stage or to find the root cause of cylinder heads failures [1, 8, 11, 12]. The cylinder heads analyzed in this article are shown in Fig. 1. Due to the symmetrical structure of the cylinder head, one of cylinders was meshed. Engines cylinder heads have four valve ports, each with an embedded valve seat; four valve guides; and four bolt holes used to secure the cylinder heads to the engine blocks. The body of the engines cylinder heads are made of cast aluminum alloy

(A356.0), with a Young's modulus of 70GPa, a Poisson's ratio of 0.33, and a coefficient of thermal expansion of 22.6 × 10^{-6} per °C. The four valve seats are made of steel, with a Young's modulus of 200GPa and a Poisson's ratio of 0.3 [11, 12]. Cylinder heads are modeled with three-dimensional continuum elements. The model consists of 35580 elements (Tet10) for improving the accuracy and acceptability of the obtained TMF prediction results.

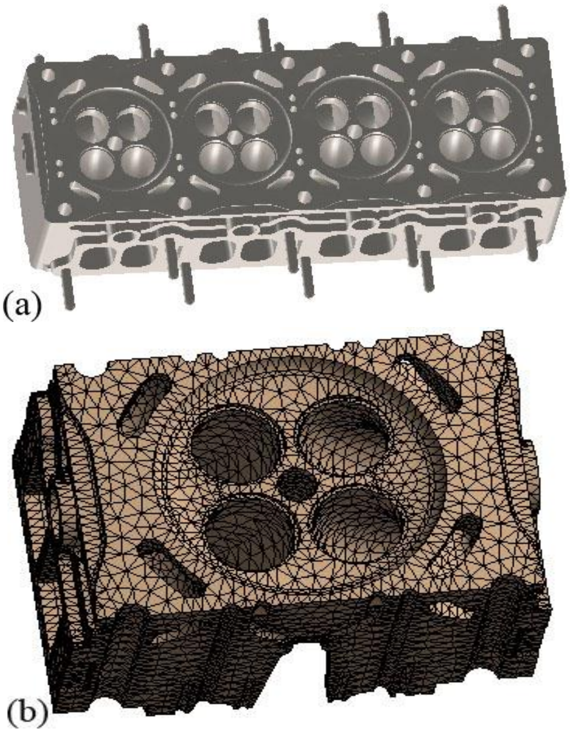

Fig. 1 (a) The cylinder head generated by SolidWorks, (b) Finite element model of the cylinder head

The kinematic hardening parameter (X), describes the translation of the yield surface in the stress space where the isotropic hardening explains the expansion/contraction. The Armstrong–Frederick law has been used to indicate the nonlinear stress–strain equation as below [25]:

$$\dot{X}=\frac{2}{3}C\dot{\varepsilon}_p-\gamma X_{\dot{p}} \quad (1)$$

Where C and γ are material constants. The term $\gamma X_{\dot{p}}$, called the dynamic recovery, causes the nonlinear response of the stress–strain behavior. Integration of Equation (4) with considering the plastic strain, leads to the relationship [25]:

$$X=\upsilon\frac{C}{\gamma}+(X_0-\upsilon\frac{C}{\gamma})+exp(-\upsilon\gamma(\varepsilon_p-\varepsilon_{p0})) \quad (2)$$

Where υ =±1 gives the flow direction. Then, X_0 and ε_{p0} are the values of X and ε_p at the beginning of the saturated cycle [25].

2-2- Notch stress-strain correction

Neuber method assumes that the notch root stress and strain solution near a stress concentration can be expressed in terms of the nominal elastic stress and strain response and the nominal, theoretical stress concentration factor. Upon yielding at the notch tip, the elastic stress concentration factor is approximated as the geometric mean of the stress and strain concentration factors [24]:

$$K_t=(K_\sigma K_\varepsilon)^{0.5} \quad (3)$$

Where K_t is the elastic stress concentration factor, K_σ is the stress concentration factor and K_ε is the strain concentration factor. In equation (3), $K_\sigma = \sigma/S$ and $K_\varepsilon = \varepsilon/e$ where σ and ε are the elastic-plastic stress and strain at the notch root. S and e = S/E are the nominal elastic stress and strain, respectively and E is the modulus of elasticity. By utilizing the relations for K_σ and K_ε, and after rearranging, equation (3) can be written in the following expression [26]:

$$\sigma\varepsilon=\frac{(SK_t)^2}{E} \quad (4)$$

Equation (4) is often referred to as Neuber hyperbola and has two unknowns σ and ε. In order to solve for σ and ε, an additional constitutive equation between stress and strain is needed. Linear elasticity and

Ramberg-Osgood plasticity are used for calculations as follows [26]:

$$\frac{(SK_t)^2}{E}=\sigma(\frac{\sigma}{E}+(\frac{\sigma}{K})^{\frac{1}{n}}) \quad (5)$$

Where K is the cyclic strength coefficient and n is the cyclic strain hardening exponent. To determine the maximum stress and strain for the notched material subjected to cyclic loading, the strain hardening constants must replaced by the appropriate cyclic strain hardening constants [26]:

$$\frac{(S_{max}K_t)^2}{E}=\sigma_{max}\varepsilon_{max} \quad (6)$$

furthermore, to calculate stress and strain range, equation (4) is modified [22]:

$$\Delta\sigma\Delta\varepsilon=\frac{(SK_t)^2}{E} \quad (7)$$

Calculations from equations (6) and (7) constitute a solution set that can be used to directly approximate the mean, valley (or peak) response at the notch root. The graphical solution to Neuber method is the intersection of the Neuber hyperbola and the tensile curve, as shown in Fig. 2.

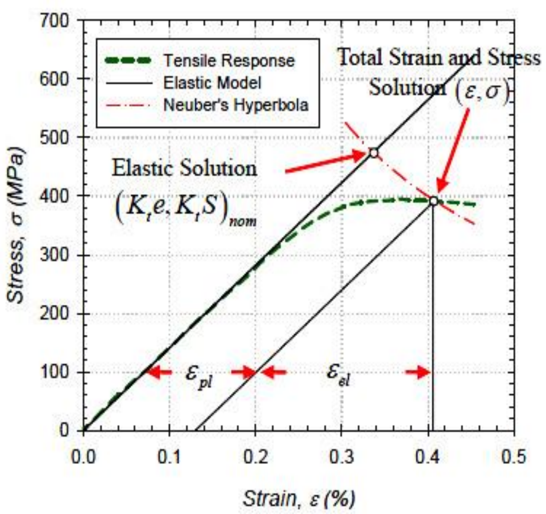

Fig. 2 The Application of Neuber method for determination of notch root stress and strain [9]

2-2- Model for thermo-mechanical life prediction

TMF is the case of fatigue failure due to simultaneous thermal and mechanical loading. The life prediction of TMF loading cases has received considerable attention in recent years mainly in engine parts. The fluctuation of complex thermal and mechanical strains is usually determinant for fatigue life of machine parts. It is well known that the mean stress level has a definite effect on fatigue performance for tensile case being more damaging. Therefore, the results need to be adjusted for mean stress effects [27]. Morrow equation is main method of strain based approach applied widely in engine industry. This method has been used to handle mean stress effects [7]. Fatigue life is estimated with Morrow relationship [27]:

$$\Delta\varepsilon=\frac{\Delta\varepsilon_e}{2}+\frac{\Delta\varepsilon_p}{2}=\frac{\sigma_f'-\sigma_{mean}}{E}(2N_f)^{b}+\varepsilon_f'(2N_f)^{c} \quad (8)$$

where σ_f' is the fatigue strength coefficient, E is the modulus of elasticity, $2N_f$ is the number of reversals to failure, b is the fatigue strength exponent, ε_f' is the fatigue strength coefficient, c is the fatigue ductility exponent, $\Delta\varepsilon$ is the strain amplitude and σ_{mean} is the mean stress [23].

3- Analysis procedure

LCF life prediction for cylinder heads is as follows:

1- Subject the cylinder heads to the transient thermal analysis

3. Return the model to the ambient temperature

4. Gas pressure and temperature distribution data are used to simulate thermo-mechanical stress analysis

5. Prediction of LCF life considering notch stress-strain correction proposed by Neuber

4- Results and Discussion

4-1- Thermal Analysis

Finite element thermal analysis is to predict thermal loading at the engines cylinder heads operation condition. For engines cylinder heads, the main element of the loading is the thermal evolution of the cylinder heads in operating engines conditions [3, 5, 7, 11, 12]. The correct determination of the temperature field is the most critical step for the lifetime prediction of the engines cylinder heads, as the thermo-mechanical loading and thus the damage reacts very sensitive to the temperature level and distribution [1, 3-5, 7].

Cylinder heads thermal boundary conditions consist of the areas of combustion chamber, inlet duct, exhaust duct, areas contacting oil and areas contacting air [16, 27, 28]. In this way, the area of combustion chamber was estimated to be 959°C with a convection coefficient of 1027W/m^{2}K. The inlet duct was defined as 30°C with convection coefficient of 320W/m^{2}K. The exhaust duct was estimated as 650°C with convection coefficient of 640W/m^{2}K. The areas contacting oil was specified as 60°C with a convection coefficient of 150W/m^{2}K. The areas contacting air was defined as 30°C with a convection coefficient of 60W/m^{2}K [16]. The resulting temperature distribution on the cylinder heads is given in Fig. 3. Thermal loading has a considerable effect on the fatigue life prediction and the temperature field identifies critical areas. The highest temperature occurs in the bridge region between the exhaust valves. This corresponds to the results by [1, 3, 5, 6, 8, 16, 27, 28]. The maximum temperature in the valve bridge between the two exhaust valves reaches to 206°C, which is within the limit of acceptance for aluminum material [8].

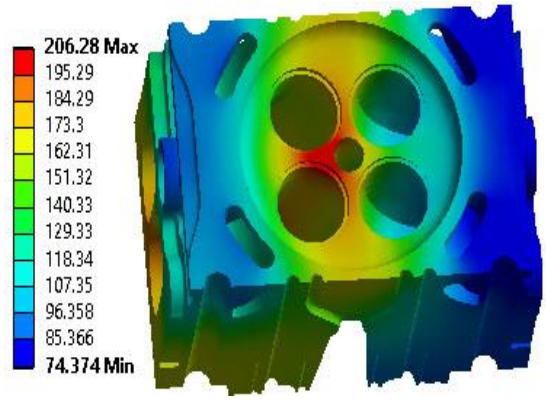

Fig. 3 The temperature distribution in the cylinder head

4-2- Mechanical analysis

The engines cylinder heads bear the mechanical stress and withstands the thermal stress due to the temperature fluctuations. Thus the analysis of thermo-mechanical coupling stress on the cylinder heads is needed [1, 2, 3, 5, 6, 8]. The loads of the thermo-mechanical coupling stress analysis of the engines cylinder heads include gas pressure, bolt preload, thermal load calculated from thermal analysis and loads caused by press-fitting of the seat valves [1, 5, 7, 29].

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 cylinder heads temperature and impose the mechanical stress [1, 2, 3, 5, 6, 8, 11-13]. Then finite element calculation is carried out and the results are studied. It is assumed that the cylinder heads are securely fixed to the engine blocks through the for bolt hoes, so the six degrees of freedom of cylinder heads bolt holes were fully constrained [11-13]. The structural boundary condition are shown in Fig. 4. In addition, Fig. 5 shows the equivalent stress field at end of second step.

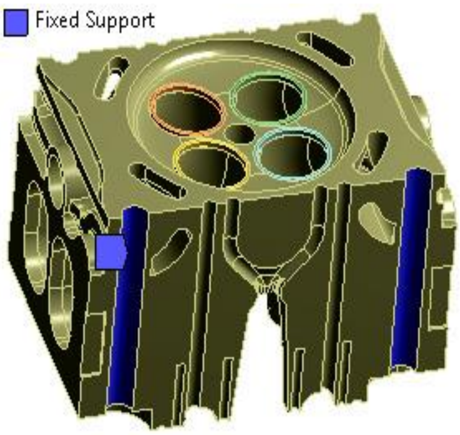

Fig. 4: Structural boundary condition

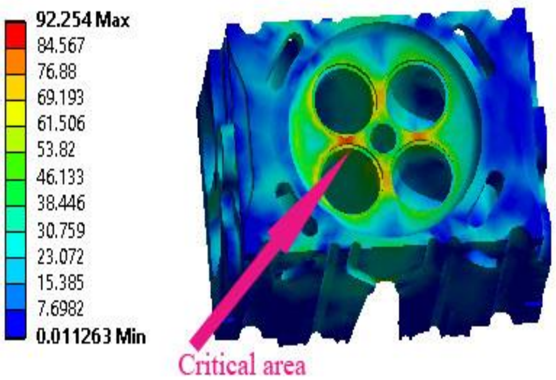

Fig. 5 The Von-Mises stress distribution in the cylinder head

Thermal plastic strain accumulates in the cylinder heads due to cyclic thermal stress, and that eventually adversely affects the lifetime of the cylinder heads as thermal fatigue cracks develop [2, 3, 11, 12]. The equivalent plastic strain distribution in the cylinder heads is shown in Fig. 6.

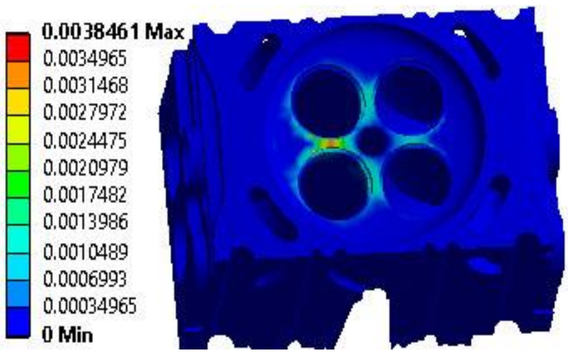

Fig. 6 The equivalent plastic strain distribution

The equivalent plastic strain is greater than zero, indicating that the material is currently yielding. The review of mechanical analysis results, it can be seen that both the stress and plastic strain, which have the dominant effect on the damage evolution are maximum in the valve bridge between the two exhaust valves. This corresponds to the results by [2, 3].

Fig. 7 shows the thermo-mechanical analysis results of selecting the number of elements of the cylinder head. As observed in Fig. 7, as the number of elements increases, the temperature changes is negligible. The stress value does not change significantly by increasing the number of elements more than 35580 elements. Therefore, the best number of elements is 35580.

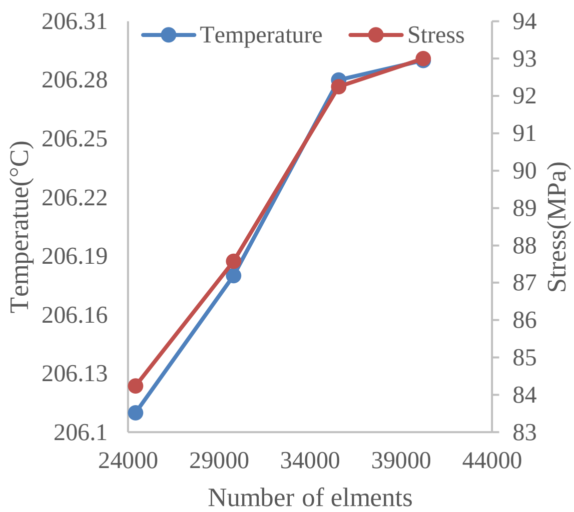

Fig. 7 The Temperature and Von-Mises stress versus number of elements

4-3- Low cycle Life prediction

One of the main durability of internal combustion engines is cylinder head cracks at valve bridge region [1-4, 7, 11, 12, 13]. In engine cylinder heads, HCF is caused by the cyclic firing pressure and LCF is caused by plastic strain induced by thermal cycles during engine start-up and shut-down operations. Due to the constraints imposed on the valves bridges by the surroundings, mechanical strain is introduced and accumulated. Eventually cracks can appear in the valves bridge regions. To avoid these failures, TMF should be assessed to ensure long term cylinder heads durability [1, 4, 5, 6, 7, 28, 29]. The fatigue damage estimation has been performed according to LCF approach, by using the Morrow equation. Fig. 8 represents the number of cycles to failure based on Morrow criterion for cylinder heads.

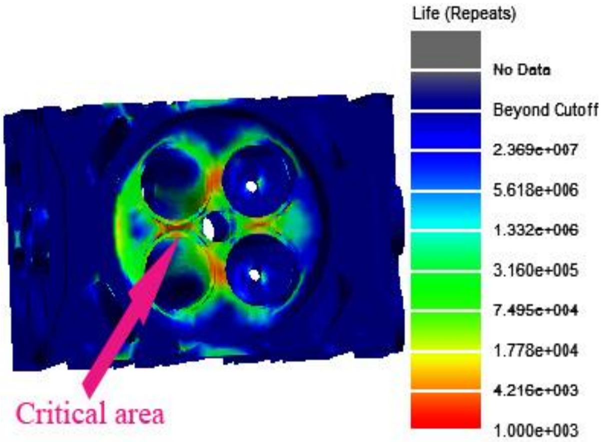

Fig. 8 The number of cycles to failure based on Morrow equation in the cylinder head

Fig. 9 illustrates the fatigue damage for the cylinder head based on Morrow criterion. As it can be seen from this Figure, maximum fatigue damage occurs as 2.372E-4 at the critical zone. 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 [32]. Therefore, cylinder head is under LCF. This corresponds to the number of cycles to failure results.

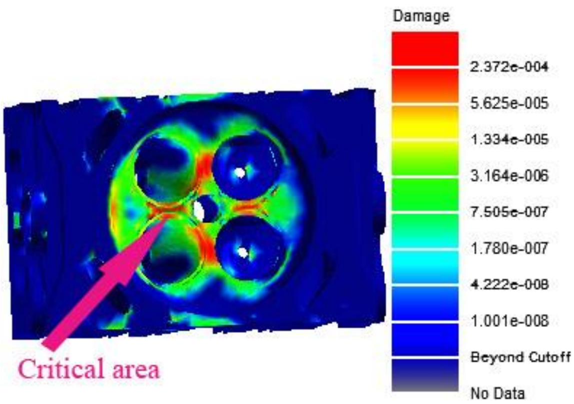

Fig. 9 The fatigue damage based on Morrow equation in the cylinder head

As it can be seen from Fig. 6 the number of cycles to failure in the critical areas is under 10^4 or 10^5 which imposes low cycle fatigue life for the cylinder heads [27]. The area where the maximum temperature and stress is occurred is where the least LCF (valve bridge between the two exhaust valves) is predicted. Fig. 8 gives the crack position of engines cylinder heads in experimental tests. As observed in Fig. 10, engines cylinder heads which has been cracked in the valve bridge between the two exhaust valves. The review of Figs. 4-8 proves that results of FEA and LCF is corresponded with experimental tests carried out by researchers, and illustrate the cylinder heads cracked in this region.

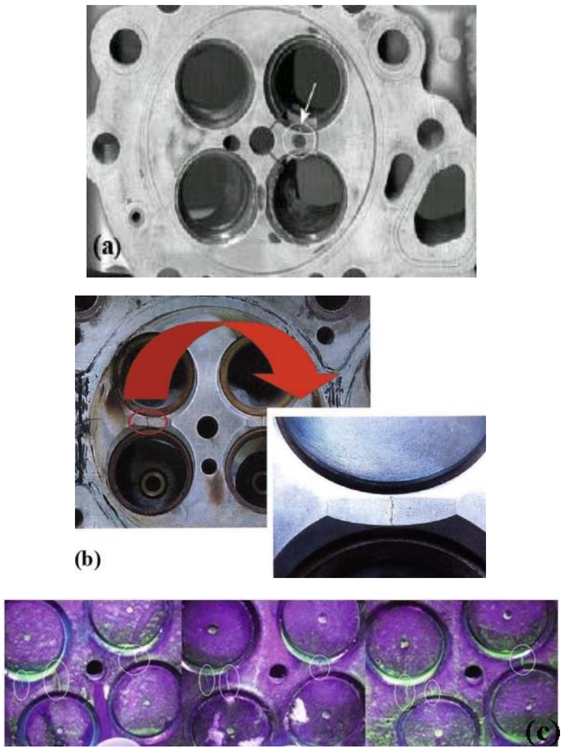

Fig. 10 The cracked cylinder head, a[30], b[31] , c[32]

5- Conclusion

It is shown that cylinder heads are exposed to LCF due to the thermo-mechanical stresses resulted from repeated start-up stop-down cycles of the engines and must be studied via FEA. The purpose of this paper is to predict LCF for cylinder head considering notch stress-strain correction proposed by Neuber. FEA provides accurate and reliable prediction of stress and fatigue life results in the design of engines cylinder heads. The thermo-mechanical analysis showed that the maximum temperature and stress occur in the valves bridge between the two exhaust valves. The numerical results showed that the area where the maximum temperature and stress is occurred is where the least LCF is predicted. To evaluate properly of results, stress analysis and LCF results is compared with real samples of damaged engines cylinder heads and it has been shown that critical identified areas, match well with areas of failure in the real samples. Better understanding of engines cylinder heads fatigue life can improve the development process of a new engine concerning computer aided engineering as well as mechanical testing efficiency.

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