



Thermo-mechanical analysis of magnesium alloy diesel engines cylinder heads using a two-layer viscoplasticity model

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ABSTRACT

Loading conditions and complex geometry have led the cylinder heads to become the most challenging parts of diesel engines. The aim of this study is to compare the distribution of temperature and stress in the aluminum and magnesium cylinder heads under thermo-mechanical loads. The three-dimensional model of the cylinder heads was simulated in Abaqus software and a two-layer viscoplasticity model was utilized to investigate the elastic, plastic and viscous behavior of the cylinder heads. The temperature and stress results of magnesium alloy were compared to aluminum alloy results. The results of finite element analysis (FEA) showed that surface temperature of the magnesium cylinder heads is about 23°C lower than the aluminum cylinder heads. As a result, the fatigue lifetime of the magnesium cylinder heads can be improved in comparison to the aluminum cylinder heads. The thermo-mechanical analysis showed that the magnesium cylinder heads tolerate less tensile and compressive cyclic stress compared to the aluminum cylinder heads. The stress reduction value in the magnesium cylinder heads was about 10 MPa which can lead to higher fatigue lifetimes in comparison to the aluminum cylinder heads.

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1. Introduction

Cylinder heads are the important parts of the internal combustion engines which are under thermo-mechanical stresses for the sake of their working environment [1]. Therefore, selection of materials is of paramount importance since they must have sufficient mechanical strength at high temperatures to be able to withstand cyclic stresses caused by heat and pressure [2].

High output capacity, low fuel consumption, low emission and reducing the cost of maintenance are among the restrictions making the design of cylinder heads a complicated task [3]. Thus, detailed analysis and design are essential. Escalation in environmental concerns and fuel costs underlines the need for research on more efficient engines with less energy dissipation and emission [4]. One way to decrease the fuel costs is to reduce the weight of vehicles. Hence, lighter alloys must be used in pursuit of this goal [5]. Recently, the use of aluminum alloys has increased for economic reasons and for improvement of engine power by weight reduction. Aluminum-Silicon is a casting alloy which has extensive use in the automotive industry, especially in cylinder heads of diesel engines. These materials have been replaced by a variety of cast iron which were previously used in the manufacture of cylinder heads. Thermal deformation is the greatest challenge faced by the aluminum cylinder heads [6].

Magnesium is lighter than aluminum and has superior casting properties that make it attractive and has almost the same strength, creep and fatigue properties. Nowadays, magnesium alloy is developed for power-train applications such as cylinders and cylinder heads. The apparent advantage of using magnesium alloy components is reduction in the engine weight up to 40 percent and a potential increase in the efficiency, power density, fuel consumption and economy improvements due to its lower density in comparison with cast irons and aluminum alloys [7,8]. The yield strength, ultimate strength, fatigue strength and hardness of magnesium alloy is less than aluminum alloy, however, its mechanical properties are in the required range to be used in the cylinder heads [9].

Cylinder heads are exposed to thermal and mechanical loads. The temperature difference, which is the result of turning the engine on and off, begets thermo-mechanical fatigue (TMF) loads on the cylinder heads [10] and consequently reduces their lifetime, especially in thinner regions [11]. The crucial regions include the valves bridge and areas near spark plugs and injectors [2]. Cylinder heads endure out-of-phase TMF. Namely, the maximum stress occurs at the minimum temperature and the minimum stress occurs at the maximum temperature. When the engine shuts off and the temperature is low, the tensile stresses arising from assembly loads will be applied to cylinder heads. As the engine starts and temperature increases the compressive stresses produced by thermal loading (σ_{th}) and combustion pressure (σ_p) will be applied to them [3]. This type of loading is displayed in Figure 1. As the figure reveals the changes in stress caused by thermal load are very high. The fluctuating stresses come out of the engine which is being heated and cooled [12,13].

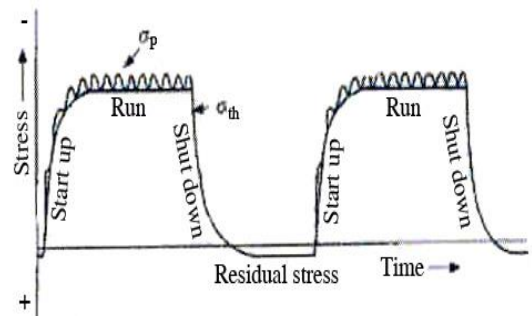


Figure 1. The cyclic loading of cylinder head [13].

Plastic deformation is observed in structures such as cylinder heads which bear high temperature fluctuations and assembly loads. Classical models are used to obtain steady response of these structures. This approach is very expensive. Because many loading cycles are required to obtain a steady response. Cyclic analysis is used in order to avoid the cost of transient analysis [5]. Numerous papers have been presented on analysis of stress and fatigue in cylinder heads. TMF of cylinder heads was studied by Thomas et al. using the energy model and elasto-viscoplastic law. Their research proved a good agreement between experimental and simulated results of the fatigue life of the

cylinder heads and the location of crack initiation [14].

Thermo-mechanical analysis of cylinder heads and cylinders of AFV diesel engines was conducted by Venkateswaran et al. Their research demonstrates that the cylinder heads and engine blocks can tolerate more stress caused by pressure and thermal loads increase, and the next generation of engines do not need further alteration [15].

Zieher et al. simulated the complete process of lifetime. They used energy model to predict the fatigue life of cast iron cylinder heads. Their research shows the simulated results of the number of cycles of crack initiation and the location of crack initiation are in accord with experimental results. The minimum lifetime was observed in the valves bridge [16]. The analysis of high/low cycle fatigue of cylinder heads was performed by Ghasemi using the thermo-mechanical analysis results. His study verified that the cracks observed in the experimental test of low-cycle of cylinder heads acknowledged the simulated results of low-cycle fatigue. The simulated results of low-cycle fatigue of cylinder heads after modification of cooling systems indicate that high levels of damage parameters are do not observed [17].

Shojaeefard et al. experimentally measured the stress in cylinder heads and compared it with simulated results. Their research confirmed the concordance between the experimental and simulated results at low temperature. The simulated stress at temperatures exceeding 200°C was estimated to be greater than the experimental stress due to inelastic material deformation [18]. Prediction of the fatigue life of cylinder heads of two-stroke linear engines was done by Rahman et al. using finite element analysis (FEA) and stress-life approach. Their research refuted the possibility of failure in all spots. Compressive mean stress increases the fatigue life and tensile mean stress lessens the fatigue life [19].

Gomez and Pischinger investigated the sophisticated interaction effects of thermal and mechanical loads, geometry of cylinder heads and TMF behavior of cylinder heads material. They optimized the valves bridge based on the ratio of mechanical to thermal strain. Their

research indicates that the vertical temperature gradients are mainly determined by the thickness of the valves bridge which plays a role in distribution of temperature. Geometric dimensions of the valves bridge and thermal conductivity were the most outstanding parameters in the thermo-mechanical analysis of cylinder heads [2].

Mirslim et al. calculated low cycle fatigue life by finite element simulation of cylinder heads based on various criteria of the strain state. Their experiments show that by cutting the valves bridge, one can increase the fatigue life of cylinder heads [12]. Trampert et al. studied the effects of thermo-mechanical loads on cylinder heads. Their research indicated concordance between experimental and simulated results of the fatigue life of cylinder heads. Crucial locations in the analysis of fatigue were the same locations of crack initiation in the experimental conditions. There was conformity between the number of cycles of calculated failure and the experimental results of macroscopic observation of cracks [20].

Zahedi and Azadi compared the stress and low-cycle fatigue life of aluminum and magnesium (AZ61) cylinder heads of diesel engines. Their research showed that the strain in magnesium cylinder heads was greater in comparison with the aluminum ones, while the magnesium cylinder heads had less stress. The fatigue life of the both cylinder heads was almost identical [5]. Azadi et al. analyzed cracked cylinder heads of gasoline engines. Examining materials and doing finite element analysis of cracked cylinder heads stress, they determined the cause of cracks and provided some solutions. Their research revealed that the main reason for cracks initiation in cylinder heads was high stress and plastic strain caused by assembly loads of cylinder heads bolts [4].

TMF analysis of gray cast iron cylinder heads were conducted by Lee et al. An acceptable agreement between experimental and simulated results of TMF life was proved. Improving and optimizing the structure of cylinder heads doubled their fatigue life [3]. Xuyang et al. predicted TMF life of diesel engines cylinder heads. Their research revealed that the

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discrepancy between experimental and simulated results was 3%. The energy criterion accurately predicted fatigue life in the valves bridge compared with thermal shock tests [21].

Metzger et al. predicted the lifetime of cast iron cylinder heads under thermo-mechanical loads and high-cycle fatigue. According to their study the experimental and simulated results of temperature matched. The mechanical analysis correctly anticipated the position and direction of cracks in the valves bridge. Comparing with experimental results, the anticipation of fatigue life was rather conservative [1].

Thermo-mechanical analysis of aluminum cylinder heads using a two-layer viscoplasticity model were carried out by Ashouri. His study showed that maximum temperature and stress occurs in the valves bridge [22]. Aluminum and magnesium cylinder heads must be adequately robust to tolerate gas pressure, assembly loads and high temperature resulting from ignition to avoid cracking the valves bridge [9,23]. Thermo-mechanical loading cylinder heads can only be controlled through modern cooling systems or protective coatings such as thermal barrier coating (TBC) that reduces heat stress and thereby reduces the temperature gradient [24].

According to the introduction, due to the lack of information on the behavior of hardening, softening and viscosity of materials, the analysis of cylinder heads is mostly based on simple models of material behavior like elastic-plastic, and the effects of viscosity and creep of cylinder heads are less taken into consideration. magnesium alloy has creep behavior at about 150°C and viscosity should also be considered [9,25]. The main objective of this study is to compare the distribution of temperature and stress in the aluminum and magnesium cylinder heads under thermo-mechanical loads. In addition, few studies have been conducted on the distribution of temperature and stress of magnesium cylinder heads. In some analyses, it is assumed that temperature changes have no effect on the stress-strain curves and thermo-mechanical analysis of cylinder heads is non-coupled. Since changes in temperature influence the stress-strain curves, the thermo-mechanical analysis of cylinder heads in this study is coupled.

2. The material and its behavioral model

In this study the cast alloy of magnesium has been used to simulate the thermo-mechanical behavior. The alloy is known as AZ91 which is applied in diesel engines cylinder heads. The chemical composition of the AZ91 is 9.00% Al, 1.05% Zn, 0.06% Mn, 0.04% Si, 0.68% RE and remainder Mg [26]. The two-layer viscoplasticity model divides the elastic and viscosity effects into two elastic-viscous and elastic-plastic networks. As displayed in Figure 2, this model is presented by Kichenin [27]. This model makes the cyclic stress-strain behavior of the material predictable with reasonable accuracy [28].

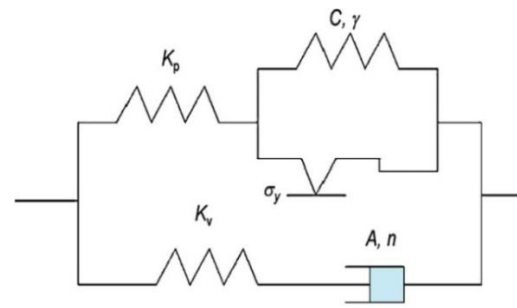


Figure. 2 The two-layer viscoplasticity model [28].

This model consists of a network of elastic-plastic parallel to a network of elastic-viscous. Plastic deformation and creep can be seen in structures such as cylinder heads of engines which are under assembly loads and temperature fluctuations. The two-layer viscoplasticity model is the best to examine the response of materials such as aluminum cylinder heads which have remarkable dependent behavior on temperature and plastic at high temperatures [1,10,16].

In the plastic network nonlinear kinematic/isotropic hardening model is applied which predicts the behaviors such as hardening, softening, creep and mean stress relaxation and it is a suitable model for the plastic behavior of materials [10, 28].

Kinematic hardening has both linear and nonlinear isotropic/kinematic model. The first model can be used with Mises or Hill yield surface while the second one can only be used with the Mises yield surface and it is the most accurate and comprehensive model to examine some issues with cyclic loading including cylinder heads of engines. The kinematic

hardening model assumes that the yield surface, proportional to the value of α , moves as back stress in yield zone but it does not deform [29]. Abaqus software uses Ziegler linear model [29] to simulate this model as following equation shows:

$$\dot{\alpha} = C \frac{1}{\sigma_0} (\sigma_{ij} - \alpha_{ij}) \dot{\varepsilon}^{PL} + \frac{1}{C} \dot{C} \alpha_{ij} \quad (1)$$

Where C is kinematic hardening modulus, \dot{C} is the exchange rate of C in temperature and $\dot{\varepsilon}^{PL}$ is the rate of equivalent plastic strain. In this model σ^0 (the size of the yield surface) remains constant. In other words, σ^0 is always equal to σ_0 (that is yield stress in zero plastic strain) remain constant. Nonlinear isotropic/kinematic hardening model includes motion of yield surface proportional to the value of α in stress zone and also changes in the size of yield surface is proportional to the plastic strain [29]. This model has been extracted from Chaboche experience [30, 31]. In order to introduce this model a nonlinear term is added to equation (1) to indicate the size of yield surface [29].

The Abaqus software uses nonlinear isotropic/kinematic hardening model as following equation shows:

$$\dot{\alpha} = C \frac{1}{\sigma_0} (\sigma_{ij} - \alpha_{ij}) \dot{\varepsilon}^{PL} - \gamma_{ij} \dot{\varepsilon}^{PL} + \frac{1}{C} \dot{C} \alpha_{ij} \quad (2)$$

Where C and γ are material constants. In order to introduce this model in Abaqus software the isotropic and the kinematics parts are required to be defined separately [10]. In order to define the isotropic part, the equation (3) is used in which b and Q_∞ are material constants [28]:

$$\sigma^0 = \sigma_0 + Q_\infty (1 - \exp(-b \dot{\varepsilon}^{PL})) \quad (3)$$

The overall back stress is computed from the relation (4) [29]:

$$\alpha = \sum_{K=1}^N \alpha_K \quad (4)$$

In equation (4) if we consider N equal to 3, the hardening variable is divided into three parts which increases the accuracy of the model [10].

Norton-Hoff law is used viscous network in order to consider the effect of strain rate, the equation of which is the following [32]:

$$\dot{\varepsilon}_V = A(\sigma_V)^n \quad (5)$$

Where the $\dot{\varepsilon}_V$ is viscous strain rate, A and n are material constants and σ_V is the viscous stress.

According to equation (6) the rate of the elastic modules in the two viscous and plastic networks is expressed by f . Where k_v and k_p are elastic modules in the elastic-viscous and elastic-plastic networks respectively [28].

$$f = \frac{k_v}{k_v + k_p} \quad (6)$$

3. The finite element model and material properties

Traditionally, optimization of engine components such as cylinder heads was based on building a series of physical prototypes, and performing a series of different experiments and tests. Unfortunately, this method is time consuming and building a prototype in the early stages of the design is arduous. Many samples must be constructed and tested in order to achieve the precise design. This process is costly. These problems have been resolved using finite element analysis to evaluate the effectiveness of various designs. This technique is accepted for the design and development of geometrically complex components such as cylinder heads in a shorter period and with the least cost. Cylinder heads are complex and challenging components of engines, for which the finite element analysis plays a critical role in optimization [18]. TMF analysis of each component needs the cyclic stress-strain distribution. Diesel engines hot components hold 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 geometrically complex components [33]. Nowadays, simulation techniques are substitute to validation tests so as to decrease the cost and time of production [20]. Cylinder heads examined in this study are shown in Figure 3.

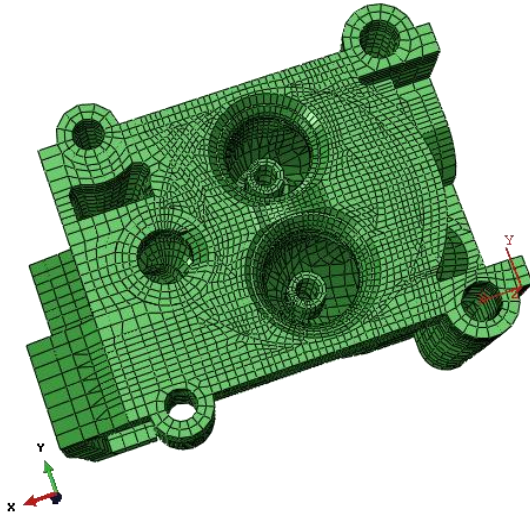


Figure.3 The meshed cylinder head [34].

The Cylinder heads have three valve ports, each with an embedded valve seat, two valve guides, and four bolt holes used to secure the cylinder heads to the engine blocks. The Cylinder heads are made of aluminum alloy (AZ91). The two valve guides are made of steel, with a Young's modulus of 106 GPa and a Poisson's ratio of 0.35. The valve guides fit tightly into two of the cylinder heads and their behavior is presumed elastic. The three valve seats are made of steel, with a Young's modulus of 200 GPa and a Poisson's ratio of 0.3. The valve seats are press-fit into the cylinder head valve ports. This is accomplished by defining radial constraint equations [34].

All of the structural components (the cylinder head, the valve guides, and the valve seats) are modeled with three-dimensional continuum elements. The model consists of 19394 first-order brick elements (C3D8) and 1334 first-order prism elements (C3D6). The model consists of 65580 nodes and approximately 80000 degrees of freedom. The loading of the Cylinder heads is applied in two phases involving thermal analysis and mechanical analysis. The values of f , n , A and Q_{∞} are extracted from the experimental results of AZ91 by [26,35] and they are entered into the Abaqus software.

There are several methods to insert the values of C and γ into Abaqus software. One of them is entering yield stress at plastic strain using the tensile test results [34]. The yield stress at plastic

strain are extracted from tensile test results and entered into the Abaqus software.

4. Results and Discussion

4.1 Thermal Analysis

Thermal stresses in the cylinder heads are the dominant stresses, leading to low cycle fatigue. Low cycle fatigue of cylinder heads is caused by repeated start-up and shut-down cycle of the engine [17]. The main part of cylinder heads stresses is the result of the thermal loading and the rest is caused by the combustion pressure and mechanical constraints (Figure 1) [18]. Therefore, thermal loading is the most important loading in the thermo-mechanical analysis of cylinder heads. Knowing the precise distribution of temperature in the cylinder heads increases the accuracy of thermal analysis [12]. Accurate prediction of the temperature of the engine is very crucial and increases the precision of the FEA results [17]. As the accuracy of thermal analysis increases, the accuracy of mechanical analysis and fatigue life estimation rises [14]. The combustion pressure causes high cycle fatigue in the cylinder heads [1]. Many researchers believe that the combustion pressure has a secondary effect on the TMF of cylinder heads [14]. In finite element simulation the valves bridge, where the greatest thermal concentration exists, is subjected to thermal loading ranging from a minimum of 35°C to a maximum of 300°C [5].

*CFLUX and *FILM options are used for the boundary conditions in the thermal analysis [22]. *CFLUX option is used to apply a heat flux to any node of the model in fully coupled thermal-stress analysis. *FILM option is used to provide film coefficients and sink temperatures for fully coupled thermal-stress analysis [34]. The cyclic thermal loads are obtained by performing an independent thermal analysis. In this analysis three thermal cycles are applied to obtain a steady-state thermal cycle. Each thermal cycle involves two steps: heating the cylinder heads to the maximum operating temperature and cooling it down to the minimum operating temperature using the *CFLUX and *FILM options. The nodal temperatures for the last two steps (one thermal cycle) are assumed to be a steady-state

solution and results are stored for use in the subsequent thermal-mechanical analysis [5]. The temperature distribution when the cylinder heads are heated to its peak value is shown in Figure 4(b). Contour results of the temperature distribution in the magnesium cylinder heads are shown in Figure 4(a).

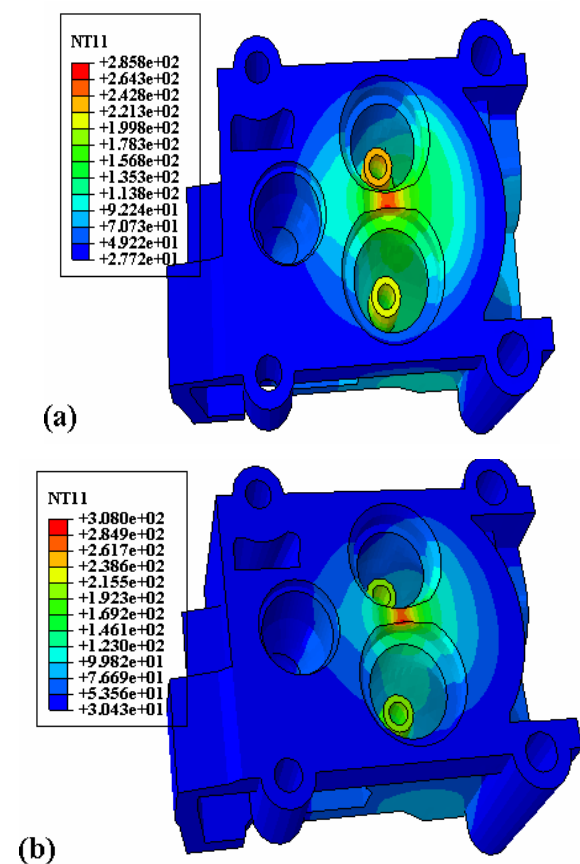


Figure. 4 The temperature distribution in the: (a) magnesium cylinder head and (b) aluminum cylinder head(A356.0) [22].

Thermal loading has a considerable effect on the fatigue life and the temperature field identifies critical regions. Crack initiation is due to the changes in the temperature field [20]. The lower temperature of the flame and the gradient temperature of the parts of cylinder heads, the less thermal stress. Thus, low cycle fatigue life of the cylinder heads which is mainly affected by thermal fatigue will increase [36]. Figure 4 shows surface temperature of the magnesium cylinder heads is about 23°C (from 308°C to 285°C) lower than the aluminum cylinder heads. This can lead to lower thermal stress values in the magnesium cylinder heads. Thus, the fatigue lifetime of the

magnesium cylinder heads can be improved in comparison to the aluminum cylinder heads.

The maximum temperature in the aluminum and magnesium cylinder heads occurred in the valves bridge. The temperature in this region (node 50420) for aluminum and magnesium cylinder heads is shown in Figure 5 as a function of time for a steady-state cycle, representing a cycle of turning the engine on and off. This Figure shows that the valves bridge of the magnesium cylinder head tolerates less temperature compared to aluminum cylinder head.

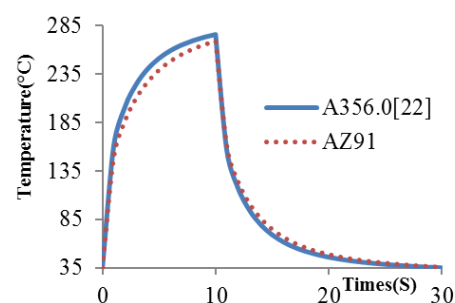


Figure. 5 The temperature at node 50147 versus time.

The lower temperature of the flame and the gradient temperature of the parts of cylinder heads, the less thermal stress. Thus, low-cycle fatigue life of the cylinder heads which is mainly affected by thermal fatigue will increase [36]. Plastic deformation and creep are observed under such conditions. The two-layer viscoplasticity model is ideally suited to examining the response of materials in these conditions [1,10].

4.2 Mechanical analysis

Mechanical analysis has been carried out in two stages. In the first stage the three valve seats are press-fit into the corresponding cylinder heads valve ports. This is accomplished by defining radial constraint equations of the form $u^r = u^p - u^s$ between the nodes on the valve seat surface and the nodes on the valve port surface, where u^p is the radial displacement on the valve port, u^s is the radial displacement on the valve seat, and u^r is a reference node. A static analysis procedure is used for this purpose [5,34]. Figure 6 demonstrates vectors of the maximum principal stress at this stage in the valves bridge. As the figure represents the maximum principal stress in

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the valves bridge is tensile. This corresponds to the results by [3,22].

The cyclic thermal loads are applied in the second step of the analysis. It is assumed that the cylinder heads are securely fixed to the engine blocks through the four bolt holes, so the nodes along the base of the four bolt holes are secured in all directions during the entire simulation [5]. Von-Mises stress distribution at the end of the second stage is shown in Figure 7(b). Contour results of the stress distribution in the magnesium cylinder heads are shown in Figure 7(a). The maximum stress, the same as maximum temperature, occurred in the valves bridge.

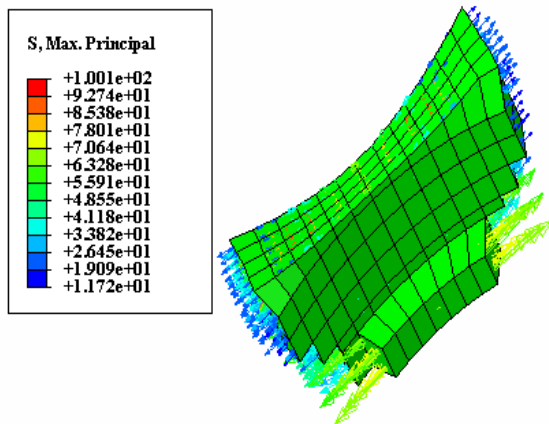


Figure. 6 The maximum principal stress vectors in the valve bridges due to the assembly loads.

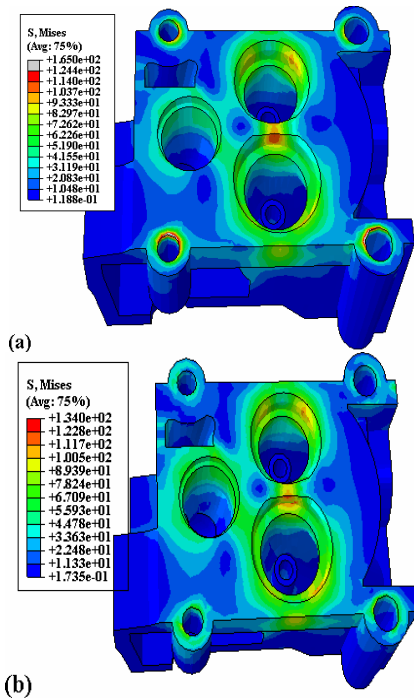
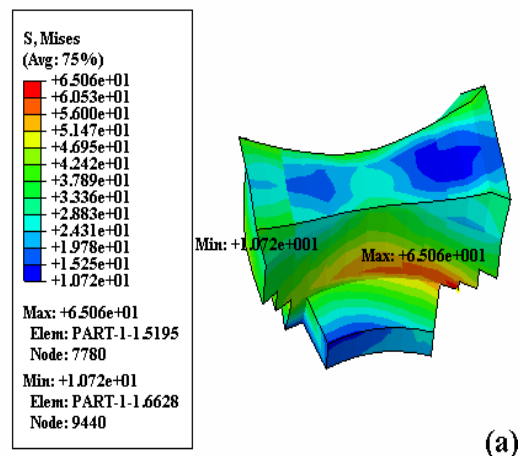


Figure. 7 The Von-Mises stress distribution at the end of the second stage of mechanical loading: (a) magnesium cylinder head and (b) aluminum cylinder head(A356.0) [22].

Maximum value of Von-Mises stress for magnesium alloy occurs in the valves bridge which is 124 MPa and for aluminum alloy is 134 MPa which is more than the value for magnesium alloy. The stress reduction value in the magnesium cylinder head is about 10 MPa (from 134 MPa to 124 MPa) which can lead to higher fatigue lifetimes in comparison to the aluminum cylinder heads. Also, for magnesium alloy but it should be considered that the weight becomes up to 40 percent lighter for magnesium alloy cylinder head in comparison to aluminum alloy cylinder heads [7,8].

The valve bridge is a crucial region [3,22]. The changes of Von-mises stress in this area in the aluminum and magnesium cylinder heads are shown in Figures 8 to 10. The Figures also include the locations of minimum and maximum Von-mises stresses. Figures 8 to 10 demonstrate that the stress in the valves bridge of the magnesium cylinder heads in tenth, twentieth and thirtieth seconds reduced respectively 22.5 MPa (from 87.5 MPa to 65 MPa), 15.6 MPa (from 128.3 MPa to 112.7 MPa) and 11.6 MPa (from 134 MPa to 122.4 MPa) compared to the aluminum cylinder heads.

The fatigue life of the cylinder heads of the engine will increase as the thermo-mechanical stress decreases [3,22]. Thus, using the magnesium alloy in the manufacture of cylinder heads of diesel engines will lead to the increase of the fatigue life compared to aluminum alloy.



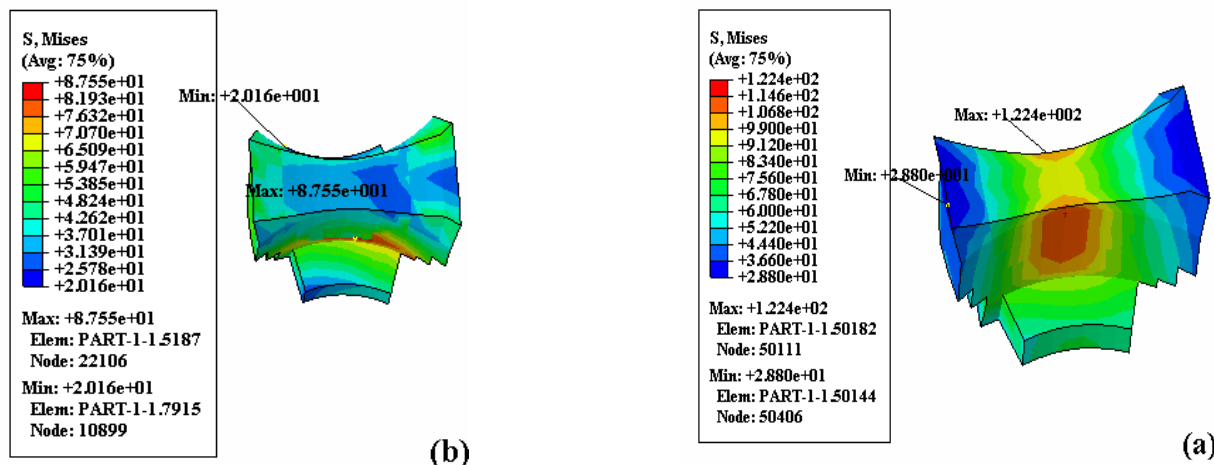


Figure. 8 The Von-Mises stress distribution in the valve bridge (in tenth second): (a) magnesium cylinder head and (b) aluminum cylinder head(A356.0) [22].

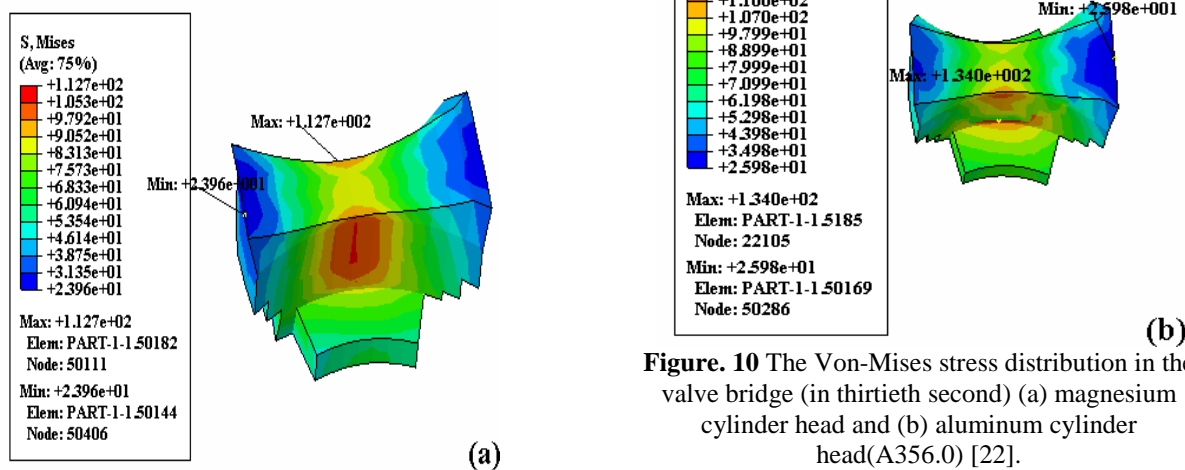


Figure. 9 The Von-Mises stress distribution in the valve bridge (in twentieth second) (a) magnesium cylinder head and (b) aluminum cylinder head(A356.0) [22].

Figure. 10 The Von-Mises stress distribution in the valve bridge (in thirtieth second) (a) magnesium cylinder head and (b) aluminum cylinder head(A356.0) [22].

As presented in Figures 8 to 10, the valve bridge is under high stress fluctuations. The location of cracks in cylinder heads is in the valves bridge. This region endures maximum stress due to the less thickness of material and high temperature caused by lack of proper cooling. Ergo, the cylinder heads will crack. Stress functions inversely to the thickness of the material. Namely, the thinnest locations withstand the highest stress. If the valves bridge becomes wider, it will be cooled better and consequently temperature gradient and thermal stress will reduce. Thus, fatigue life of cylinder heads increases [2].

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The finite element model predicts a large compressive stress field in the valves bridge as shown in Figure 7. Thermal expansion of hot spots in cylinder heads are constrained by cool regions which have less thermal expansion. As a result, the compressive stress is created in the valves bridge which corresponds to the results of the source [18]. Figure 11 displays diagram of normal stresses (S11) for aluminum and magnesium cylinder heads in the X direction for point 1 of element 50152. These elements are in the valves bridge.

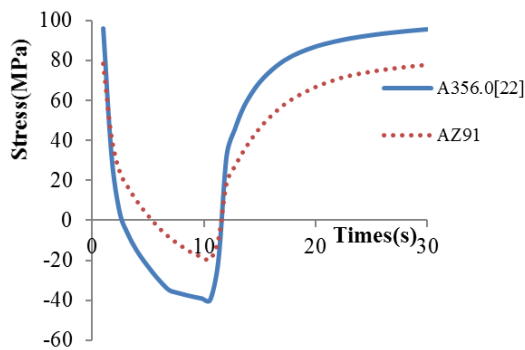


Figure. 11 The normal stresses in the X direction for point 1 of element 50152 versus time.

The engines experience maximum temperature in tenth second of running (Figure 5). Stresses are compressive because of the thermal loading and combustion pressure at the moment (Figure 11). This corresponds to the results by [3,18,22]. Maximum value of compressive stress for magnesium alloy is 19.56 MPa and for aluminum alloy is 39.88 MPa which is more than the value for magnesium alloy. The thermal loading and mechanical constraints generate a compressive stress field, which may drive to compressive yield surface. As the engine shuts off and its temperature gradually decreases to the room temperature, the stress is tensile for the sake of assembly loads (Figure 11). This corresponds to

the results by [3,18,22]. Maximum value of tensile stress for magnesium alloy is 78.45 MPa and for aluminum alloy is 96.12 MPa which is more than the value for magnesium alloy.

Figure 11 illustrates that the magnesium cylinder heads tolerate less tensile and compressive cyclic stress compared to the aluminum cylinder heads. The stress field for the yield surfaces is compressive at high temperature and turns into tensile stress at low temperature; it is correspondence to the results of sources [3,22]. The valves bridge is under the cyclic tensile and compressive stress which corresponds to the results of sources [21,22]. According to the source [22], changes in cyclic compressive and tensile stresses cause cracks in cylinder heads.

5. Conclusion

In this paper coupled thermo-mechanical analysis of magnesium alloy diesel engines cylinder heads was studied. A two-layer viscoplasticity model was used for this purpose. This model makes the cyclic stress-strain behavior of the material predictable. The temperature and stress results of magnesium alloy were compared to aluminum alloy results. The thermo-mechanical analysis proved that the maximum temperature and stress occur in the valves bridge. The obtained FEA results showed that surface temperature of the magnesium cylinder heads is about 23°C lower than the aluminum cylinder heads. Therefore, the magnesium cylinder heads endure lower temperature and fatigue life will increase in comparison to the aluminum cylinder heads. The results of finite element analysis indicated that the stress in the magnesium cylinder head decreased approximately 10 MPa than the aluminum cylinder heads. As a result, the fatigue lifetime of the magnesium cylinder heads can be improved in comparison to the aluminum cylinder heads. Also, for magnesium alloy but it should be considered that the weight becomes up to 40 percent lighter for magnesium alloy cylinder head in comparison to aluminum alloy cylinder heads [7,8]. The valve bridge is a critical region and fatigue cracks initiation in this area [3,22]. The FEA proved that the stresses in the valves bridge are compressive when the engine is running and become tensile when the engine is

shut off. The valves bridge is subjected to cyclic tensile and compressive stress. Low-cycle fatigue always occurs in this region and fatigue cracks appear after a few cycles. The thermo-mechanical analysis showed that the magnesium cylinder heads tolerate less tensile and compressive cyclic stress compared to the aluminum cylinder heads. Changes in cyclic compressive and tensile stresses cause cracks in cylinder heads [22]. Thus, using the magnesium alloy in the manufacture of cylinder heads of diesel engines will lead to the increase of the fatigue life compared to aluminum alloy.

In order to prevent them from cracking, it is recommended to modify the cooling system of engines as well as thickness and geometry of the material in crucial parts. Since thinner regions endure high stress, cylinder heads crack in these regions. TBC might also be used in the regions which not only boost the engine performance, but also increase the fatigue life of cylinder heads [37]. Materials of high thermal conductivity can be used in these regions. Materials of high thermal conductivity decrease the maximum temperature in this region, leading to an increase in the fatigue life of the cylinder heads. Cutting the valves bridge approaches the region to cooling jackets of cylinder heads. Consequently, the temperature in the region decreases and the fatigue life increases. Temperature is effective on stress-strain curves, and the thermo-mechanical analysis of cylinder heads must be coupled.

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