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Simulating and analyzing the cylinder-head of a gas burner engine and comparing it with a diesel engine

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***Abstract*—** Some engine manufacturers are converting diesel engines to natural gas engines with fuel-lean conditions. Natural gas engines provide a higher amount of heat and consequently, it is essential to modify and convert different parts of the converted natural gas, particularly the cylinder head. This paper presents the results of thermomechanical stress analysis on a cylinder head of a converted natural gas engine. We first modeled the cylinder head exactly in three dimensions using SOLIDWORKS software and then meshed and analyzed it in an ANSYS software environment in this project to achieve accurate results. Multidimensional mechanical and thermal stress were completely analyzed on the cylinder head using finite element analysis (FEA). The results have been compared with the results achieved for the diesel engine to confirm the analysis (the base engine that the gas engine has been achieved by its conversion). The results explain that higher stresses have been created between the two valves. The finite element analysis (FEA) and computational fluid dynamics (CFD) analysis applied with test validation are highly influential analysis instruments to develop and design engine components.

Keywords: natural gas, engine, CFD,simulation, chassis dynamometer,engine dynamometr

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1. Introduction

Currently, various fuels are utilized in piston engines, which depends on the geographical properties and the frequency of that fuel in the region. Methane gas has covered the main part of natural gas. Natural gas includes 85% to 98% of methane [1] and other hydrocarbons such as ethane, propane, butane, etc. with amounts of nitrogen, sulfur compounds, carbon dioxide, nitrogen, and water vapor depending on the well's geographical location. [2]The properties of different natural gas compounds distinguish the thermal value of the fuel and its octane number. The plentiful natural gas resources available in Iran (apart from the issue of reducing pollutants) and the problems in providing other fuels have caused to construct gas-burner vehicles and this issue has been highly significant. [3] One of the most important activities in this field is to construct gas burner engines or convert existing engines to gas burner engines. No changes are observed in the geometrical properties of the engine in gasoline cars because of the closeness of its compression ratio with the gas burner engine (of course, the decrease in the performance properties of the conversion engine is due to a small difference in the normal compression ratio). Accordingly, the vehicle becomes a gas-burner by installing the gas-burning kit and gas capsules. [4] These engines can be run on gasoline or gas at any time when the driver wishes. [6] The conversion process is not simple for diesel engines, and a large difference in density ratio makes it unavoidable to change the geometric properties of the engine. It is not possible in these engines to change diesel fuel to gas or gas to diesel (the engine operates solely with gas or a combination of gas and diesel fuel). Recently, studies are started to conduct in order to use gas fuel by applying special mechanism and lean the engine, without changing the density ratio and geometric properties, like gasoline engine. Accordingly, it is required to examine the thermal-mechanical stress distribution of engine parts, particularly cylinder heads. In order to optimize engine parts, including cylinder heads, samples were first created and then constrained to working and laboratory conditions in ancient times. This work lasted until one of the samples achieved good results compared to the others, and it was accepted as a foundation for production. Designers required a lot of time and money to satisfy the design requirements in this method. According to the time-consuming and expensive nature of the old method, the finite element analysis method is applied to achieve designs with higher efficiency. Engineers can use the FEA finite element analysis method to predict the temperature and stress distribution in the parts and determine the heat transfer method before producing the first sample. Engineers consider the FEA method as a powerful and beneficial computer design tool. [8] It is required to start with numerical simulation and performing the necessary theories to develop or design a piece through the results of finite element analysis. The cylinder head is one of the most complex parts of the engine that the FEA plays a key role in optimizing. The achieved suggestion of this project leads the designer to evaluate and consequently, optimize the cylinder head. [5-7] The analysis objectives are as follows:

1- Confirming the results of thermal-mechanical analysis of the gas burner engine by comparing the results with the diesel engine (base engine);
2- Obtaining the values of thermal-mechanical stresses and comparing them;
3- Comparing the values of stresses with the amount of elastic stresses;
4- Recommending how to optimize the cylinder head;

Two main methods are available to design and produce a gas burner engine, one of them is to design a gas burner engine from start, which all the design operations of a complete engine must be performed. The diesel base engine is converted to a gas burner engine in the second method. This method presents information about the base engine and according to this information and the performed calculations, some engine parts are changed to produce the desired gas engine. This paper examines the second method**.**

2. Experimental Set-Ups

Experimental results are the first step for computations. It is required to measure items such as flow and cooling water temperature of the inlet to the cylinder, flow, pressure, and temperature of the inlet air to the cylinder, the flow of exhaust gases from the cylinder, flow of fuel inlet to the cylinder and the temperature of the outer surface of the cylinder head to achieve the boundary conditions and calculation inputs. Three thermometers were installed on each of the surfaces outside the cylinder and the cylinder head was installed on three samples in three rounds to measure the temperature outside the cylinder and the cylinder head. We achieved the mean temperature at the outside of the cylinder and the cylinder head at 360 ° C. (Fig.1).

Figure 1: Extracting the laboratory results

There are usually two ways to measure the required parameters, the engine dynamometer, and the chassis dynamometer. Torque and power are measured directly from the flywheel output in the engine dynamometer. The engine is attached directly to the dynamometer in this case and the necessary tests are conducted. The engine does not remove from the vehicle and the vehicle is located on the dynamometer rollers with the wheel in the second type, the chassis dynamometer. The engine was installed in the engine dynamometer and temperatures, inlet gas pressure, liquid flow, and gas were measured in this project. A six-channel temperature gauge has been applied with an accuracy of $\pm 0.1^{\circ} C$ to measure the required temperatures and devices have been used with an accuracy of $\pm 0.001 Kpa$ and $\pm 0.1 m^3 / s$ respectively to measure the pressure and flow. Tables (1) and (2) indicte the test results.

Table 1. Measured results of Natural gas engine

Parameter	Measure
Coolant flow(m^3/h)	3.76
Coolant inlet temperature (K)	350
Inlet air flow for each cylinder (kg/h)	86.97
Inlet air temperature (K)	337.1
Exhaust flow from each cylinder (kg/h)	90.5
Fuel flow for each cylinder (kg/h)	3.53
Mean temperature on the outside surfaces of cylinder head (K)	360
Inlet air pressure (Kpa)	171.35

Table 2. Measured results of Diesel engine

Parameter	Measure
Coolant flow(m^3/h)	3.76
Coolant inlet temperature (K)	348
Inlet air flow for each cylinder (kg/h)	86.97
Inlet air temperature (K)	300
Exhaust flow from each cylinder (kg/h)	90.5
Fuel flow for each cylinder (kg/h)	3.53
Mean temperature on the outside surfaces of cylinder head (K)	360
Inlet air pressure (Kpa)	100

3. Computational methods

The calculations normally begin with modeling. Modeling is an essential step because other steps depend on modeling. Meshing, using thermal boundary conditions and CFD, analysis, and validation of the results. Figure 2 shows the analysis methodology.

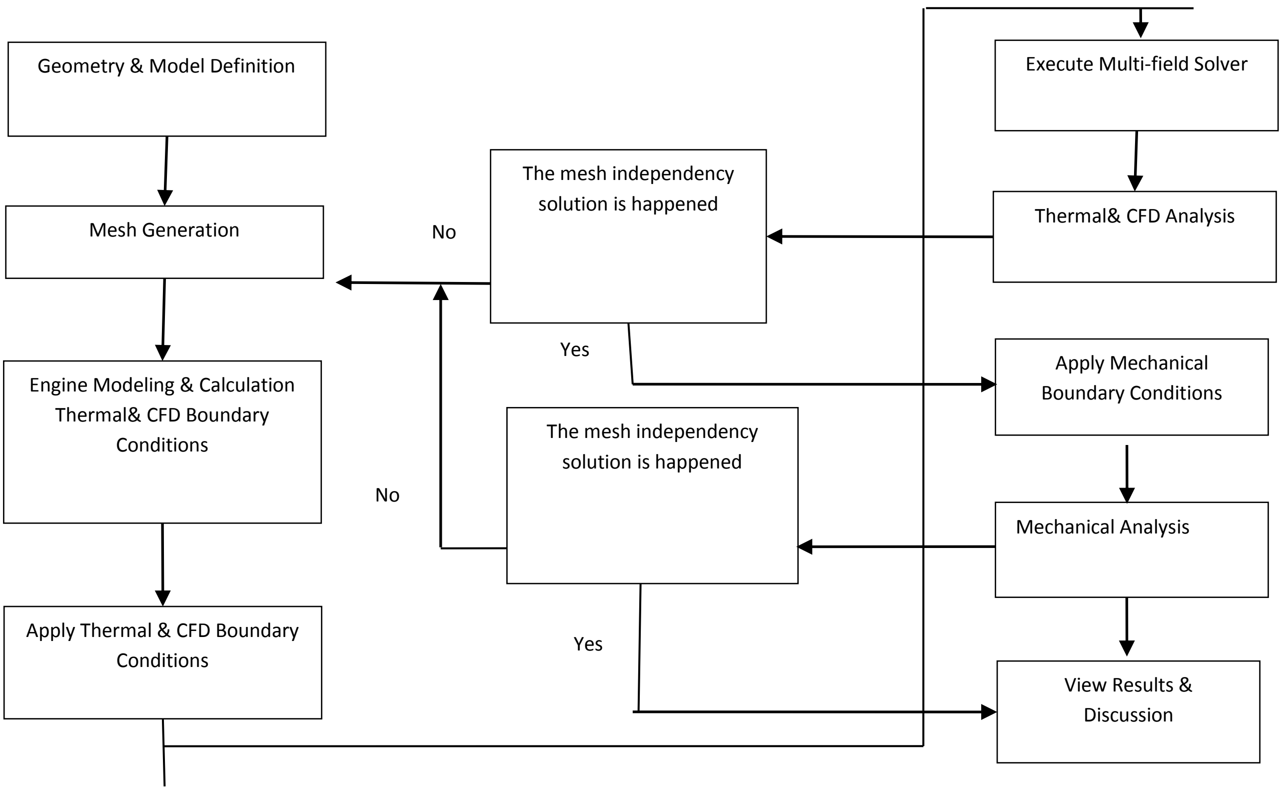


Figure 2.Flowchart of the thermomechanical analysis procedure

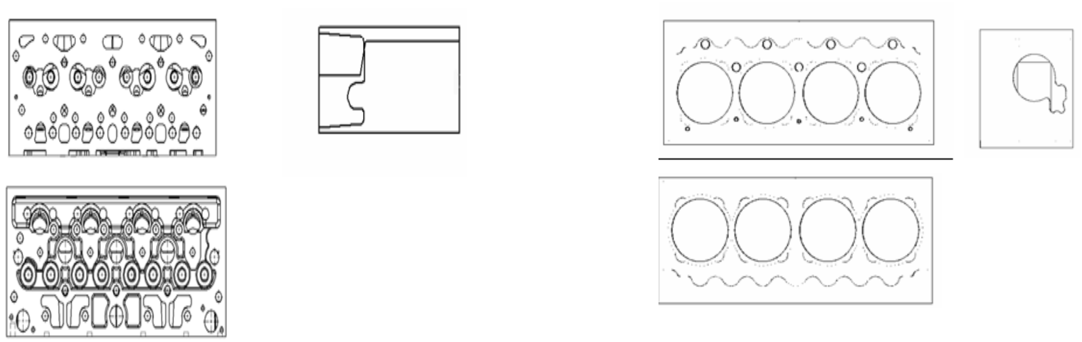
The heat-mechanical stress distribution requires drawing a geometric model of the part. Engineering drawings and 3D modeling software can help us to achieve this objective. Solid Works is one of the engineering software for geometry modeling [9, 10]. It is required to send the model created in Solid works to the Ansys software environment as a Solid output with an extension (x_t) to generate the mesh. [11, 12] It is also necessary to mesh the Solid and Fluid models together at the same time for meshing to solve Thermal-CFD through Multi-field. Accordingly, the Solid and Fluid element types have been selected. The $Solid\,70(Brick\,8node\,70)$ three-dimensional element has been used for thermal analysis and the element $Fluid\,142(3Dflotran142)$ has been used for CFD analysis. [3, 11].

The analysis may be linear or non-linear, static or dynamic, fixed or transient. According to the CFD, the results after processing achieve the proper form for engineering evaluation, which accesses the binary code of the analysis code and extracts the proper results. [13] Local properties may be a function of thermal treatment, temperature, and ... and the input is applied by the user. If there is a dependence on temperature, the user input tables will calculate the property of the local substance for the dependence on temperature [14].

The temperature distribution is determined using the energy equation. It is essential to have fluid pressure and velocity to solve the energy equation in the water passage (water velocity and temperature are measured at the input and output ports). Accordingly, the equation of motion is additionally required. The CFD method is used to obtain the temperature and velocity distribution in the waterway (Taylor, 1966), [15].

3.1 Model definition and mesh production

Figure (3, 4) shows the geometry and model of the cylinder and cylinder head, and their thermal limitation conditions have been outlined in three dimensions using SOLIDWORKS [3, 9].



a) cylinder head b)cylinder

Figure 3. Geometry of the cylinder head (a) and cylinder (b)

a)cylinder head b) cylinder

Figure 4. Model of the cylinder head (a) and cylinder (b

It is essential to model the water way to analyze the temperature in the cylinder head according to the boundary conditions on the cooling waterside. Accordingly, the water way model presented in Figure (5) was drafted. In this model, the water inlet and outlet location, water impact surfaces in the cylinder, and cylinder head have been well modeled. This model has been employed to analyze the CFD using the Multi-field method in Ansys environment with the cylinder head and cylinder model to determine the way to distribute the temperature and MATLAB software should be utilized to model the combustion chamber [16].

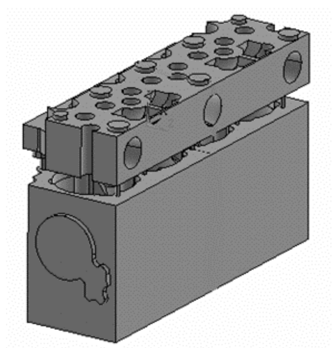


Figure 5. Model of the water Way for CFD analysis

Mesh has been modeled using ANSYS software. Four types of meshing have been made to obtain a mesh-independent response in this project, and a mesh-independent response has finally been achieved. Figures (6, 7, and 8) show the meshing of the cylinder head, cylinder, and waterway.

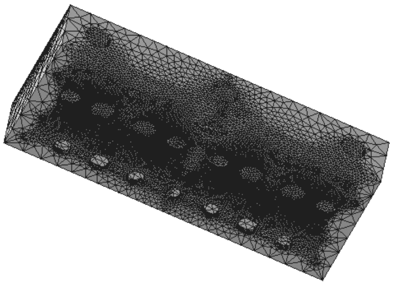


Figure 6. Cylinder head mesh

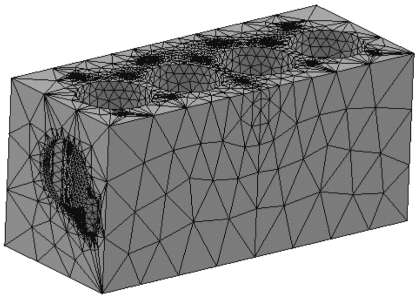


Figure 7. Cylinder mesh

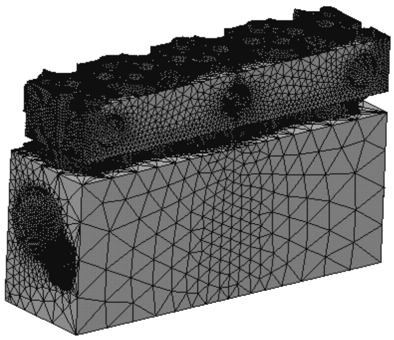


Figure 8. Waterway mesh

Mesh production plays an important role to achieve more accurate results; finer meshes have been applied than other surfaces for small and sensitive surfaces. For example, the finer mesh is used for the combustion chamber and valve seat. [3] The three-dimensional model includes 507533 elements, 50965 Solid nodes for Thermal analysis, and 733252 elements, 91244 Fluid nodes for CFD analysis. The ratio of element width to height Aspect ratio is about 2.6 in sensitive areas, including between two valves. Figure (9) shows the distribution of the node at the combustion chamber at the two cylinders of the studied engine. As it is clear, the distribution of nodes in sensitive areas is much finer than in other areas. This issue supports answering the problem accurately.

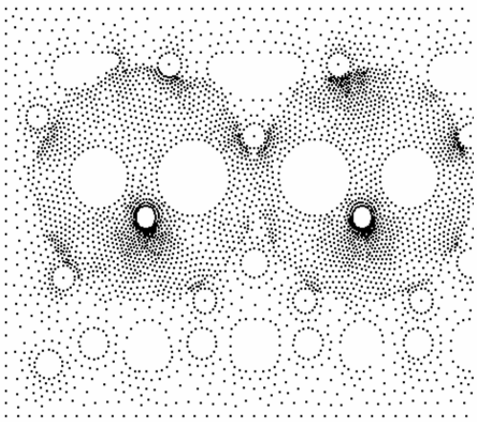


Figure 9 .Nodes distribution in combustion chamber of cylinder head

This value usually reaches about 6 away from the valve location in areas considerably away, considering that the temperature gradient and consequently the stress gradient is lower than other areas in these areas, this ratio is sufficient for the mentioned areas [16]. Larger meshes have been applied in insensitive areas, such as the top of the cylinder head. The water channel/way has meshed so that its common nodes are integrated with the Solid model (cylinder head).

3.2 Thermal boundary conditions

It is challenging to select the proper boundary conditions, especially for engine combustion chamber components whose boundary conditions may be different in terms of space and time in any thermal analysis [17, 18]. The boundary conditions of heat and pressure from the combustion chamber and the preload related to the cylinder head bolts have been regarded in this analysis. It is also necessary to calculate boundary conditions outside of the cylinder and the cylinder head. The heat transfer coefficient of displacement in five zones for the gas engine has been calculated separately to consider all thermal boundary conditions.

3-2-1 Boundary conditions of the outer surfaces of the cylinder head

It is expected to calculate the heat transfer coefficient of the outer surfaces of the cylinder head to apply the boundary conditions of the outer surfaces. Accordingly, Rayleigh equation (1) has been utilized to calculate the free displacement heat transfer coefficient [18] (1)

$$N_u = C\,(G_r P_r)^{m} \quad (1)$$

Where, m is 0.25 and C is0.52.

Rayleigh (R_a) and Grashof (G_r) numbers are obtained using Equations (2) and (3):

$$R_a = G_r P_r \quad (2)$$

$$G_r = g\beta(T_w - T_\infty)\, L^3/V^2 \quad (3)$$

Ambient temperature has been considered 293K in all stages of calculation. The g value is equal to $9.81\, m/s^2$ and according to the laboratory results, the T_w temperature value of the outer surface of the cylinder head has been achieved equal to 360K.

3-2-2 Input and output port boundary conditions

The Christopher equation (4) has been applied to calculate the heat transfer coefficient at the air inlet port to the cylinder of the gas burner engine and the combustion gases that leave the inside of the cylinder through the outlet port [19]:

$$N_u = 0.0718 R_e^{0.75} \quad (4)$$

The R_e Rayleigh number is obtained from Equation (5):

$$R_e = \dot{m}\, d / \mu A \quad (5)$$

The $\dot{m}$ value is obtained from the CFD results or test and is equal to $90.5 Kg/h$ for the output port of each cylinder. The air temperature at the inlet port is assumed to be 337.1 K (due to the presence of a turbocharger) and the gas temperature at the outlet port is obtained through equation (6) when the gas expands isotropically until it approaches the outlet gas pressure. 20]:

$$T_e = T_4\,(P_e / P_4)^{(K-1/K)} \quad (6)$$

3-2-3 Combustion chamber boundary conditions

The velocity equation (7) has been applied to calculate the heat transfer coefficient of the combustion chamber. The pressure and temperature of the combustion chamber at each angle of the crankshaft are required to apply this equation. The average value of thermal conductivity in the combustion chamber is computed using the following equation [21-23].

$$h_{(x,\,t)} = 3.2\, b^{-0.2} P^{0.8} T^{-0.55} U^{0.8} \quad (7)$$

3-2-4 Border conditions on the waterway side

The general equation of heat transfer from the combustion chamber to the waterway is defined as Equation (8). [22-23]:

$$N_u = 10.5 R_e^{0.75} \quad (8)$$

Which R_e is achieved through Equation (9):

$$R_e = \dot{m}\, d / \mu A \quad (9)$$

4. Discussion and results

According to the issues mentioned previously, it is required to ensure that the results are correct before beginning to discuss the results. In other words, the performed analysis must be confirmed. Hence, it is essential to compare the results obtained from the cylinder head of the base engine, which means diesel engine, with the results of the gas burner engine. Figure (10) indicates a number of common nodes of the second cylinder combustion chamber in both engine types.

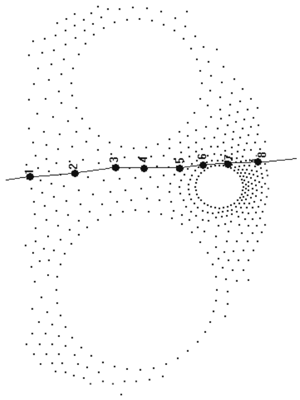


Figure 10. Nodes location of Natural gas engine and Diesel engine in the cylinder head for comparing

The second cylinder is selected due to the high Von-Mises stress and temperature distribution in the second cylinder compared to other cylinders, which has been obtained from the results of stress analysis in both engines. Figure (11) indicates Von Mises stress's comparison in the 8 selected nodes.

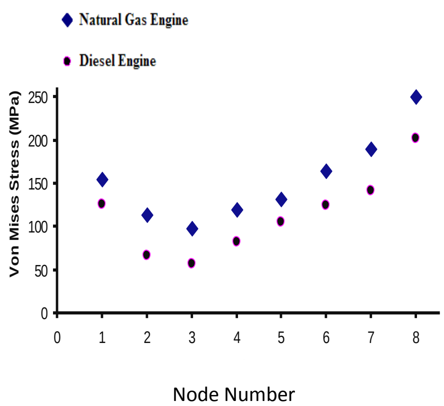
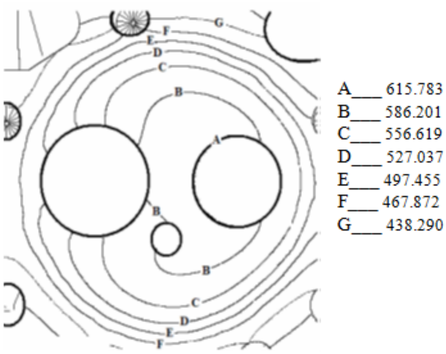


Figure 11: Comparison of Vone Mises stress in selected nodes Figure (10) for diesel and gas engine

There is the same shape of the stress change in both engines, but the amount of stress in the different nodes of the gas burner engine is nearly 1.12 times of the diesel engine, which presents highly accurate conducted calculations because it was previously predicted that according to the high temperature in the gas engine, its stress is higher than the diesel engine. After the confirmation stage, we will examine the results, which have been performed in several stages. The results of temperature distribution are first examined in both types of engines. We have tried to search for and examine high temperature and stress areas in the cylinder head instead of high values of temperature and stress, which means that it is possible to observe the high temperature and stress due to a meshing error or the way of applying a load on a node. In contrast, it is also possible to observe an area with high temperatures and stresses that have spread over a wide area and accordingly, should be considered. According to the discussed cases, most of these tests will be performed in the second cylinder.

Figures (12,13) indicate that the temperature among the valves in the combustion chamber is higher than in other places, and the temperature of the second cylinder is also higher than other cylinders.



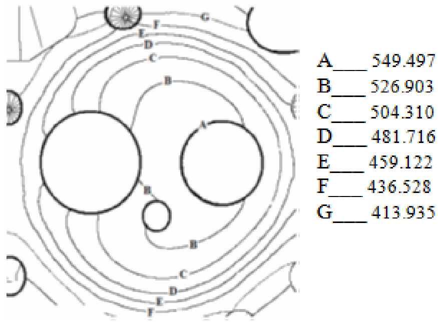


Figure 13. Temperature contours on the firedeck of the combustion surface of Diesel engine (K)

Figure 12. Temperature contours on the firedeck of the combustion surface of Natural gas engine (K)

With careful survey, we can note that the temperature distribution in all cylinders is not similar, and the temperature gradient in cylinders' center is higher than other regions. While, we move away from cylinders' center, the temperature gradient reduces. The reason is the design of waterway. Water enters from the left side of the cylinder and exits from the top of cylinder head (left and right); hence, the areas which are in contact with cooled water or inlet water are colder than other areas. It indicates that the temperature distribution in cylinders is nonlinear, thus, the cylinder one is cooler than other cylinders. Figure (14 and 15) shows the temperature distribution in cylinder head and gas engine cylinder. Based on the images, the temperature gradient in the second cylinder is larger than other cylinders, and the temperature in the cylinder centers is higher than other sections of cylinder's center, due to the high temperature gradient at the combustion chamber level in comparison with other regions. It can be predicted via FEM that there will be high stress and strain between the two valves in every cylinder, most of which is due to the high temperature in these areas. In the gas engine, the maximum temperature among the two valves is about 616 K which in the diesel engine the maximum temperature in this region is 549 K, which is one of the reasons for the high stress between the two valves. The computations indicate that the temperature gradient at cylinder head and on the gas side of the gas engine is 53 and on the cooling liquid side is 5.6. In the regions inside the cylinder head, the temperature changes from 438 K to 616 K, in the areas outside the cylinder head the temperature is low and from 293 K to 360 K.

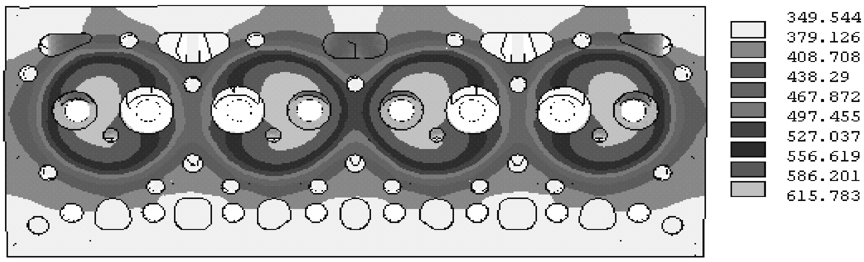
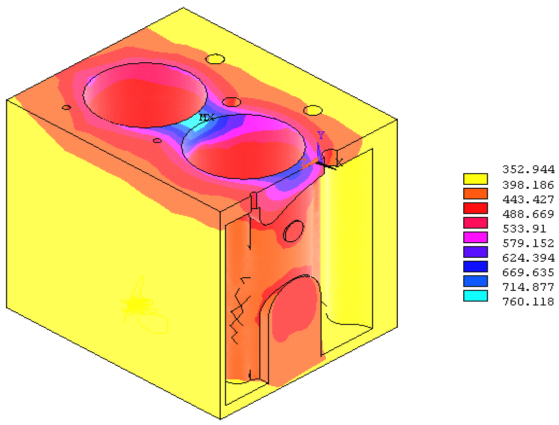


Figure 14: Temperature distribution in K in the cylinder head of a gas engine



Temperature distribution in K in the combustion engine cylinder Figure 15:

In the waterway, the temperature distribution is different from the cylinder and the cylinder head. The temperature size in the waterways is less than the cylinder head and cylinder at the similar level. Figure 16 indicates the temperature distribution in the waterways.

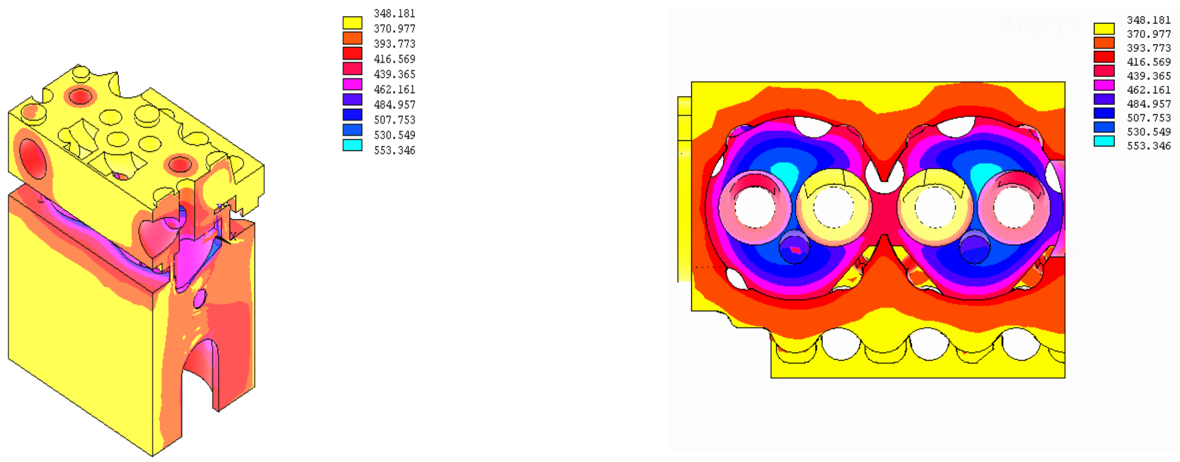


Figure 16: Temperature distribution in terms of K in the water passage

We compare the temperature and pressure diagrams for the angle of the flywheel in gas and diesel engines in this section: According to the mentioned cases in section 3-2-3, the combustion chamber boundary conditions have been adopted to calculate the heat transfer coefficient of the combustion chamber using Vashni equation (7). We require the pressure and temperature of the combustion chamber at each angle of the flywheel in order to use this equation. Hence, it is also required to model the combustion that has been performed by Matlab software, its details have not been mentioned here, and only the temperature and pressure diagrams for each flywheel angle (images) 17, 18, 19, 20 have been mentioned. It is worthwhile to mention that calculations have been conducted at 1850 rpm (maximum torque occurs at this speed). The following diagrams show that the temperature in the gas-burning engine is higher than the diesel engine.

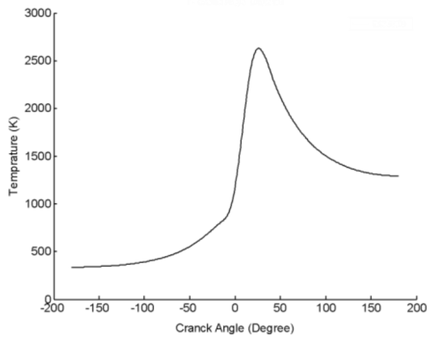


Figure 17: The temperature of the combustion in the gas engine in terms of crankshaft angles

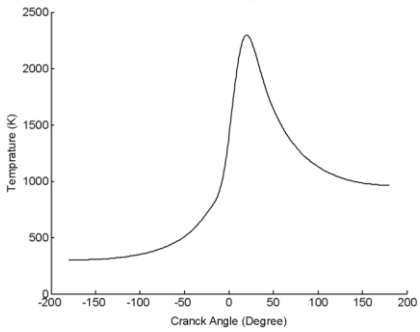


Figure 18: The temperature of the combustion in the diesel engine in terms of crankshaft angles

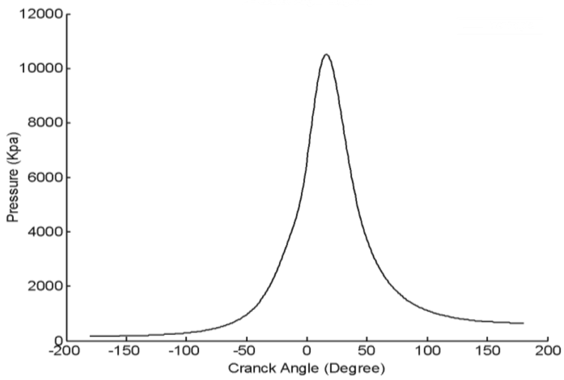


Figure 19: Pressure from combustion in th gas engine in terms of crankshaft angles

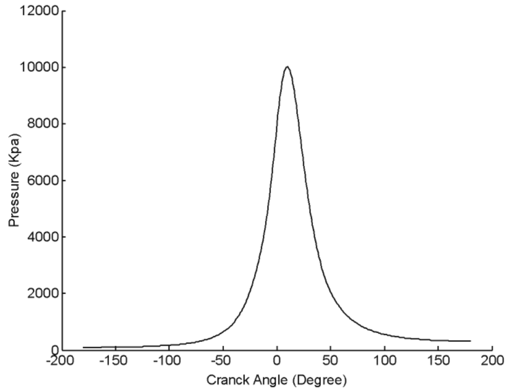


Figure 20: Pressure from combustion in the diesel engine in terms of crankcase angles

5. Conclusion

According to the explained cases, the stress density or maximum stress is located between the two-cylinder head valves (the surface of the combustion chamber of the second cylinder). Additionally, the stress values in gas-burning engines are 1.12 times of diesel engines at the same points. This additional stress results from the high temperature in the gas-burning engine compared to the diesel engine. Thermal analysis of the gas burning engine was performed independently to obtain the cause or causes of high voltage in the gas-burning engine. Examining and comparing the image stated in the previous pages lead us to conclude that 82% to 87% of the stress created in the cylinder head is related to the thermal load in a gas-burning engine and the rest is caused by the compressive load and mechanical constraints. Accordingly, it is needed to increase the heat load absorbed by the coolant to reduce the heat-mechanical stress in the cylinder head of the gas-burning engine, which will reduce the heat load on the cylinder head. According to the reference [24], EOTD (the engine outlet temperature difference) should be in the range of 55-60 degrees of centigrade. If the EOTD is not in the standard range, the amount of stress on the engine components will be increased. Changing the components of the water-cooling system (as an example of a water pump or radiator) causes the EOTD standard range will be available, otherwise, we should change the shape of the waterway to improve the heat transfer level. If the water-cooling system is improved, the designer will be required to re-access temperature and stress distribution to confirm engine components.

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