



Experimental and finite element vibrational analysis of exhaust manifold heat shield

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ABSTRACT

Most of internal combustion engines have one or two heat shields that have been installed on the exhaust manifold to avoid the heat transfer to upper parts of the engine such as the valve cover. In some engines, this part fails due to the fracture and causes engine noise and other failures in the engine. In this paper, the failure of a heat shield due to vibrational loads of the engine has been investigated using the finite element method and experimental methods. Since heat shields have been made from thin shells, their analysis can be done in the vibrational field of plates. The analysis of the investigated heat shield in this field shows that two of the first resonance frequencies of heat shield are in the range of the engine speed and locations of heat shield cracks are at the maximum deflection positions.

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1) Introduction

The heat shield is one of the important parts of the internal combustion engines that can be seen in various types in most of these engines. This part be installed on the exhaust manifold and prevent the thermal radiation to upper parts of the engine such as the valve cover. Figure 1 shows a gasoline engine that has two heat shields. In this engine, the first heat shield has been installed on the exhaust manifold and the second heat shield has an air gap with the first one to decrease the heat transfer. The maximum speed of this engine is 6450 rpm.

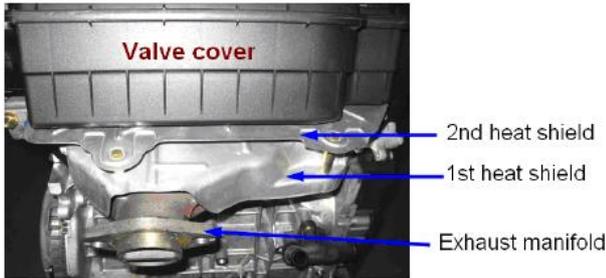


Figure 1: The heat shields on a gasoline engine

The investigation of some heat shields illustrated that this part fractures in some engines tests. For examples, the second heat shield (shown in Figure 1) has been fractured from two places after 500 hr durability test (shown in Figure 2). Cracking and the failure of this part has been reoccurred in many engine tests and it can be a cause to the noise disturbance in the vehicle and melting of the valve cover due to heat transferring. Furthermore, replacing of this part is very difficult and needs disassembling of many parts of the engine, that it imposes much costs of repairing.

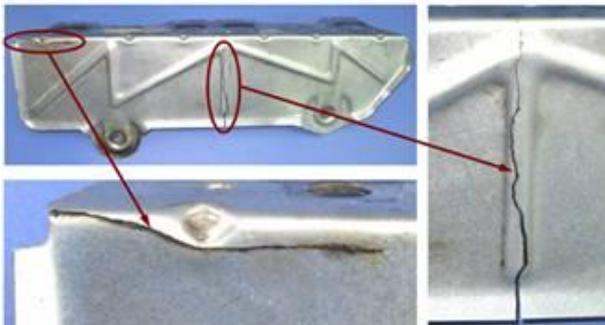


Figure 2: A cracked heat shield, the second heat shield of the engine (in Figure 1)

In order to analyze the failure of the heat shield, it is necessary to determine the cracking cause. The heat shields of internal combustion engines are subjected to thermal and vibrational loads, generally. In the engine that shown in Figure 1, the first heat shield is subjected to thermal load predominantly but in the second one, the vibrational load is dominant. Investigating of the second heat shield by the

infrared (IR) thermometer shows that the maximum temperature of this part is about 100°C.

Furthermore, the low amount of carbon and silica in the material of this part and ribs in the structure show that the vibrational load has been considered in the design of this heat shield. Therefore, the fracture of the heat shield was the result of the vibration which has to be studied by the vibration theory of plates.

In the field of plate vibrations, a great amount of researches and literatures has been presented over the past century [1]. The study of the vibration behavior of a plate with cracks is a problem of great practical interests. Only a few papers have been published on the vibrational analysis of a finite cracked plate. The presence of cracks will affect the static and dynamic characteristics of the plate, such as the static deflection and the natural frequency [2]. The heat shield that shown in Figure 2 has been clamped from one side to the exhaust manifold and pinned in the other side. Therefore, this heat shield can be considered as a rectangular plate. Vibrations of a cracked rectangular plate were investigated by Lynn and Kumbasar [3], who used the Green's function to represent the deflections of plates and to obtain a homogeneous Fredholm integral equation of the first kind. Stahl and Keer [4] studied the vibration and stability of cracked rectangular plates in terms of dual-series equations, which were then converted to a homogeneous Fredholm integral equation of the second kind. Hirano and Okazaki [5] studied a rectangular plate with cracks perpendicular to the simply supported edges [2, 6]. In the aspect of experimental studies, Maruyama and Ichinomiya [7] used a time-averaged holographic interferometry to investigate natural frequencies and corresponding mode shapes, experimentally with regard to the influence of the slit length, the position and the inclination angle of clamped rectangular plates. The determination of the location of defects in plates from measurements of natural frequencies of cracked plates was studied by Cawley and Adams [8]. To consider the vibrations of a cracked rectangular plate with arbitrary boundary conditions, a numerical method has to be used. Qian et al. [9] developed a finite element solution by deriving the stiffness matrix for an element including the crack tip from the integration of the stress intensity factor. Krawczuk [10] proposed a solution similar to that of Qian et al. [9], except that the stiffness matrix for an element including the crack tip was expressed in a closed form [6].

In this paper, natural frequencies and mode shapes of investigated heat shield (without any crack) have been obtained by the finite element method (FEM) and an experimental method using accelerometers. Then, the obtained results show in figures and tables.

2) Vibration resources of engine

In the internal combustion engines, two sources of vibration exist: the combustion excitation and reciprocating masses such as the conrod excitation. Therefore, two excitation frequencies exist in these engines that obtain from Equations 1 and 2, for the combustion excitation frequency and the reciprocating masses excitation frequency, respectively.

$$f_c (\text{Hz}) = \text{Engine speed (rpm)} \times \left(\frac{1}{60}\right) \times 2 \quad (1)$$

$$f_r (\text{Hz}) = \text{Engine speed (rpm)} \times \left(\frac{1}{60}\right) \times 4 \quad (2)$$

For examples, if the engine speed was 2400 rpm, the combustion excitation frequency is 80 Hz and the reciprocating masses excitation frequency is 160 Hz. Therefore, the excitation frequency of the engine can be obtained from the engine speed [11]. Different engine parts have various natural frequencies. If these frequencies lie in the range of zero to 430 Hz, the part will resonate by the excitation of the engine combustion or reciprocating masses.

3) FEM analysis

Resonance frequencies and mode shapes can be calculated by the FEM, firstly. Since modeling, meshing and boundary conditions of the heat shield have been done in the ABAQUS software [12]. The schematic of heat shield models after meshing and imposing boundary conditions have been shown in Figure 3. Meshing of the part has been done by a four-node-square shell element. The total number of elements is 11817. The displacement and the rotation of connection surfaces of the heat shield with the cylinder head and the exhaust manifold have been tied.

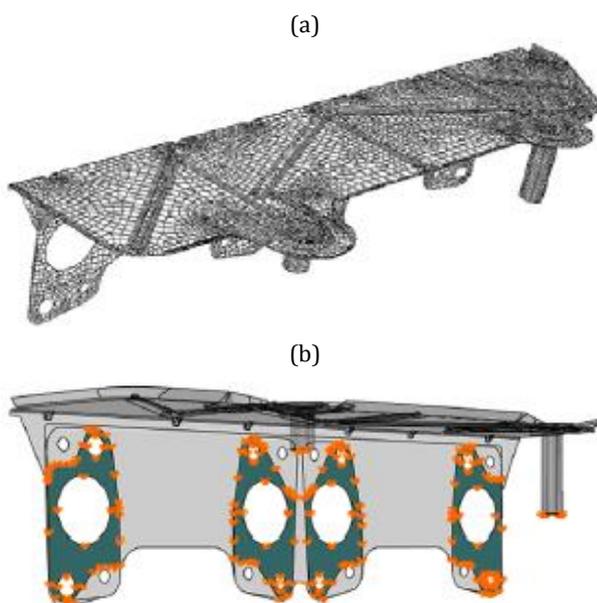


Figure 3: Schematic of heat shield in ABAQUS package; (a) meshing and (b) boundary conditions

The heat shield was made of AS100 steel [13] and its material properties are as: the mass density of 7400 kg/m³, the Young's modulus of 162 GPa and the Poisson's ratio of 0.3. The Young's modulus obtains from ASTM-E8 [14] tensile tests. The total mass of the simulated heat shield is 443 gr, that is near to the actual mass of the part, 436 gr.

Table 1 shows FEM results of heat shields including first three resonance frequencies. Since the maximum speed of the selected engine is 6450 rpm, therefore, according to Equation 1, the maximum resonance frequency of the heat shield will be 215 Hz and this part has two resonance frequencies in the range of vibrational loads.

Table 1: Results of first three resonance frequencies

| Excitation mode | Frequency (Hz) |
|-----------------|----------------|
| 1 | 125.4 |
| 2 | 171.1 |
| 3 | 280.8 |

In Figure 4, first and second mode shapes of the heat shield and corresponding crack places are shown. From this figure, it can be known that the cracks (shown in Figure 2) have been taken place during these two frequencies.

4) Engine vibration test

In order to investigate the vibrational behavior of the heat shield and the validation of FEM results, an engine vibration test has been done. For this purpose, two accelerometers on the heat shield and one accelerometer on the cylinder head of the engine have been installed. Since the accelerometer was in the vertical direction, its excitation was only in this direction. Figure 5 shows the locations of these accelerometers on the engine and Table 2 shows the specifications of the accelerometers.

Investigating of the vibrational behavior of the engine parts needs to run the engine in all work speeds. Therefore, the engine with a 60 s ramping load has been run from 1500 rpm to 5500 rpm. During this loading, the accelerometers data recording device records the acceleration of the heat shield and the cylinder head with the frequency of 4 kHz. Figure 6 shows the engine test cell and other devices that have been used in this test.

The experimental result that can be obtained from the vibration test is the acceleration in the time domain. These results have been shown in Figure 7 for three accelerometers (shown in Figure 5).

As it can be seen in Figure 7, the resonance time of the accelerometer 1 is near to the accelerometer 2. This phenomenon is rational, since the resonance of the heat shield effects on all points but according to the resonance mode, one point only has the maximum deflection. Therefore, the resonance time of the heat shield can be determined from all accelerometers time domain behaviors.

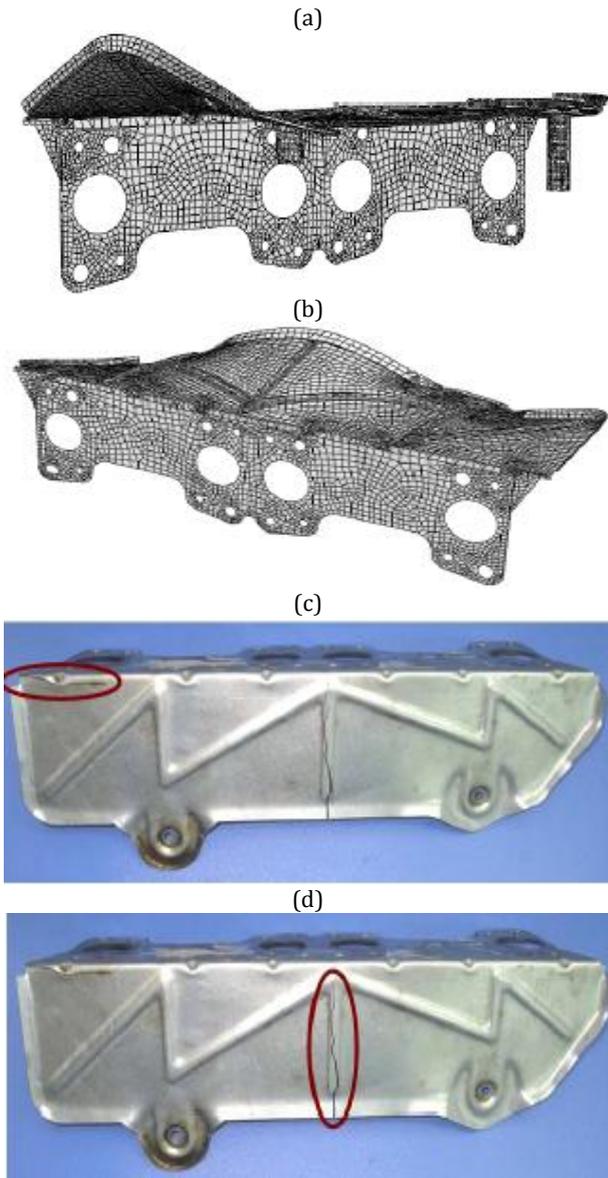


Figure 4: First and second mode shapes of the heat shields resonance and corresponding cracks; (a) mode I, (b) mode II, (c) the crack for mode I and (d) the crack for mode II

Table 2: Specifications of engine test accelerometers

| Physical parameters | Characteristics |
|---------------------|-----------------|
| Sensing element | Ceramic |
| Housing material | Titanium |
| Sealing | Welded Hermetic |
| Size (Hex × Height) | 7.1 mm × 8.4 mm |
| Weight | 2.0 gr |

In the modal analysis, the time domain results do not gain a good image from resonance frequencies of the vibrational system, but by using a Fast Furrier Transformation (FFT) of the results, the interpretation of them can be done easily and exactly. As discussed before, the time domain graph of accelerometers 1 and 2 shows the same resonance time. Therefore, the FFT of them gains the same resonance frequency that illustrates the resonance

frequency of the heat shield. Figure 8 shows the FFT of the test results for the heat shield and the cylinder head.

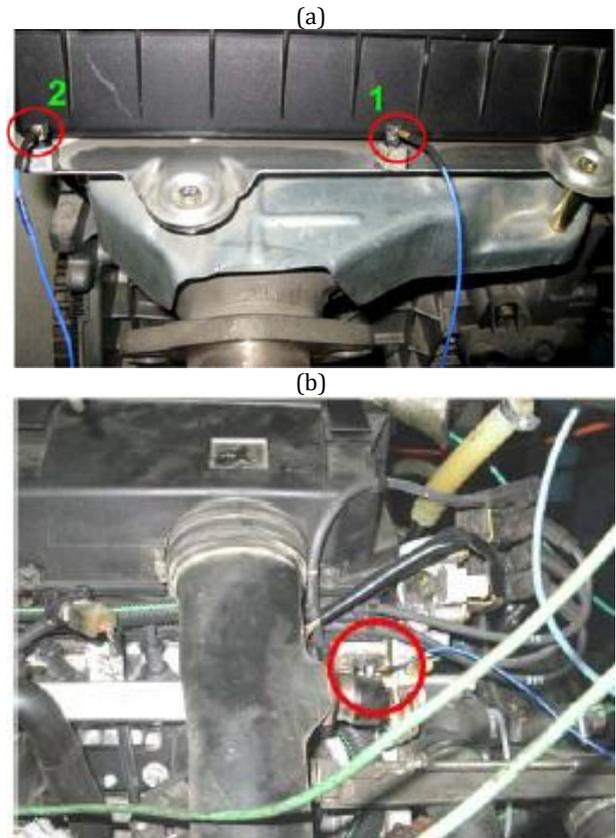


Figure 5: Locations of accelerometers on the engine; (a) the heat shield and (b) the cylinder head



Figure 6: The vibration engine test cell

In Figure 8(a), the first resonance frequency is 50 Hz. This frequency exists in Figure 8(b). Therefore, it belongs to the cylinder head. But the second frequency (125 Hz) in Figure 8(a), is not in Figure 8(b) and it is the first resonance frequency of the heat shield. This result has agreement with the FEM result. Therefore, it can be said that the first resonance frequency of the heat shield is 125 Hz and this frequency is in the range of the engine work speed (215 Hz or 6450 rpm). Also, other frequencies in Figure 8 belong to the cylinder head and have not any effects on the fracture of the heat shield.

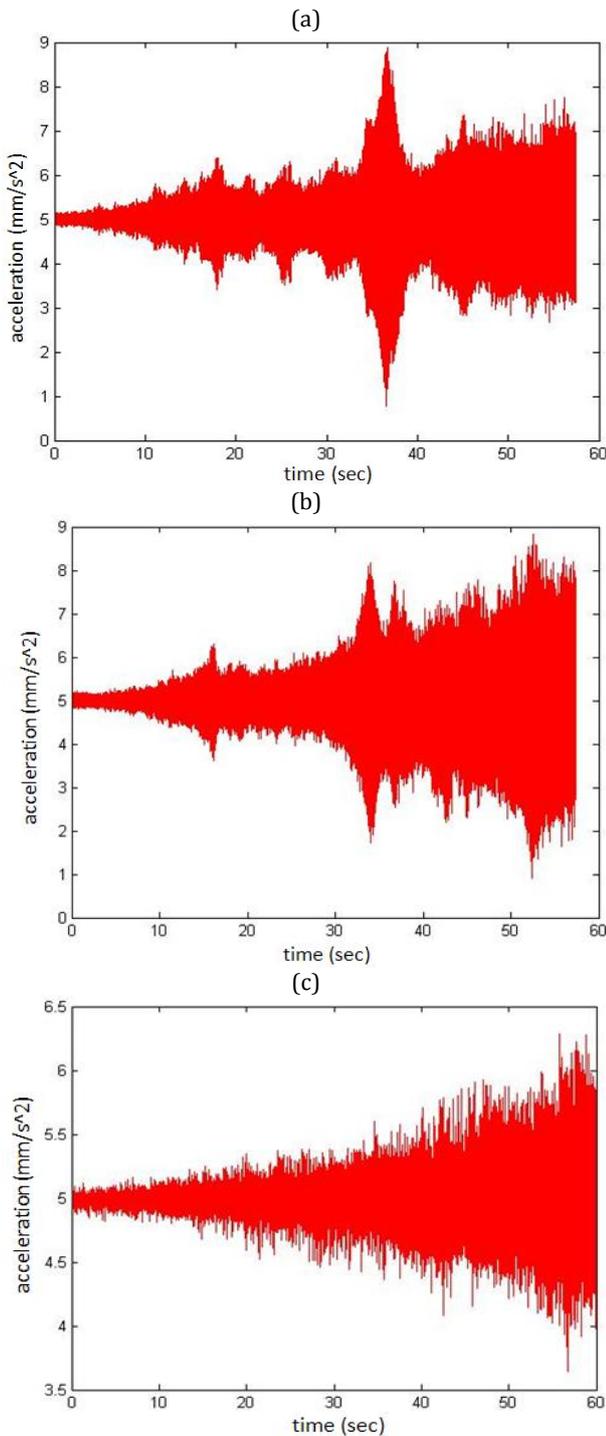


Figure 7: Experimental results in time domain; (a) the accelerometer 1, (b) the accelerometer 2 and (c) the cylinder head

5) Modification methods of heat shield

The FEM and experimental results show that the first resonance frequency of the heat shield is 125 Hz according to Equation 1. This frequency is equal to 3750 rpm of the engine speed. Since the maximum speed of the tested engine is 6450 rpm (215 Hz), therefore, the heat shield will resonate during the test. For avoiding this phenomenon, the heat shield should be modified in the manner that its first

resonance frequency increases to be upper than 215 Hz. For this purpose, some methods have been considered and the resonance frequencies of them have been calculated by the FEM. These modification methods are as: a) increasing the incline angle of the front rib from 30 deg to 80 deg; b) increasing the incline angle of all around ribs from 30 deg to 80 deg; c) increasing the number of stiffening ribs and changing the position of them; d) increasing the thickness from 0.8 mm to 1 mm; and e) increasing the thickness from 0.8 mm to 1.5 mm. Figure 9 shows these modification methods.

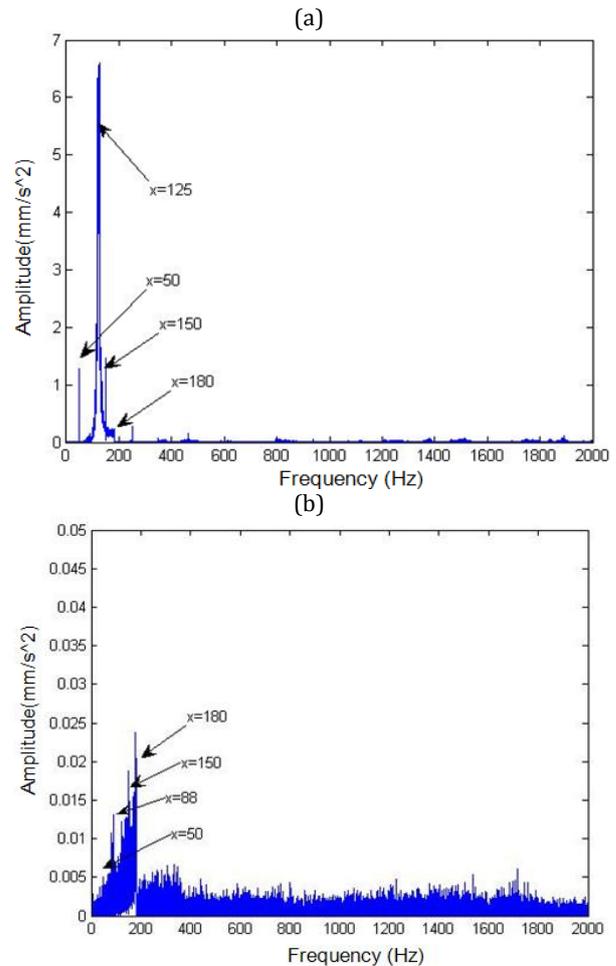


Figure 8: FFT of the experimental results; (a) heat shield and (b) cylinder head

6) Discussion

The FEM results of the modification methods have been shown in Figure 10. As it can be seen in this figure, only increasing the thickness of the heat shield to 1.5 mm can increase the first resonance frequency of the heat shield. This frequency is upper than the engine work speed which is equal to 215 Hz. In order to validate of FEM results of the heat shield with 1.5 mm thickness, an engine test has been performed for the second time. The FFT of engine test results have been shown in Figure 11.

In Figure 11, the first resonance frequency of the modified heat shield (with the thickness of 1.5 mm) is 233 Hz. It is near to the FEM results with a result of 240 Hz. Also, the resonance frequency of the cylinder head is near to the first test shown in Figure 8(b). Increasing the thickness of the heat shield from 0.8 mm to 1.5 mm causes the mass increase from 443 gr to 750 gr. This modification method is not very suitable according to the economical aspect, but at the aim of avoiding the engine noise and other part failures and also decreasing high repairing costs, it is a good method.

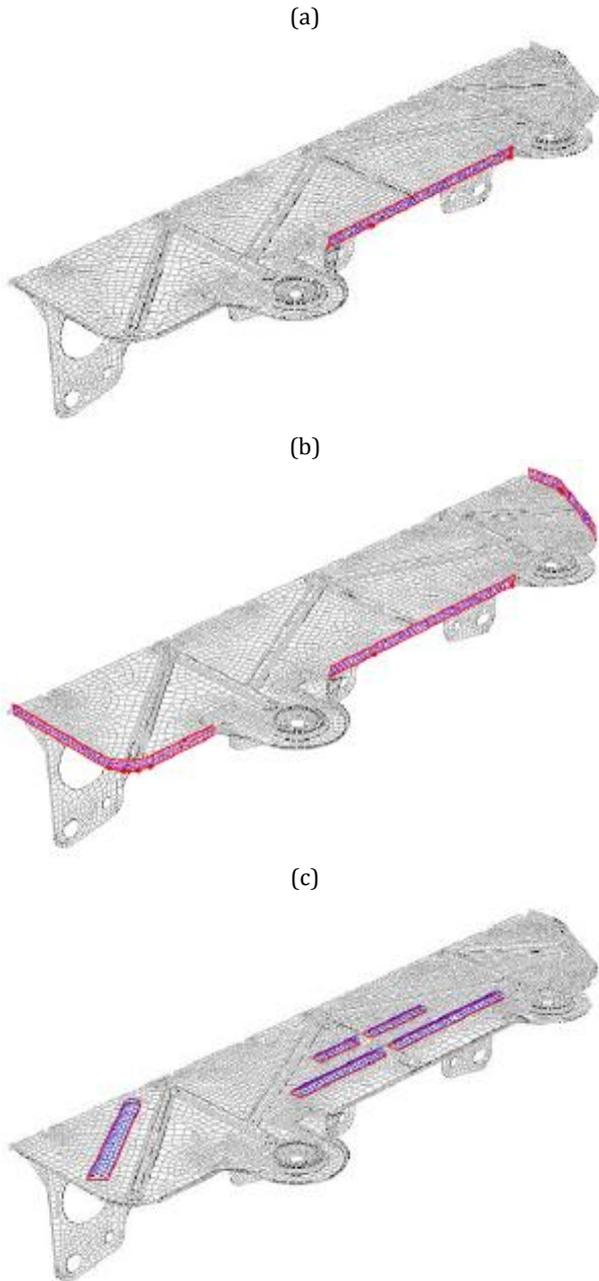


Figure 9: Modification methods of the heat shield; (a) increasing the incline angle of the front rib, (b) increasing the incline angle of all around ribs, (c) increasing and changing the stiffening ribs, (d) increasing the thickness to 1mm and (e) increasing the thickness to 1.5 mm

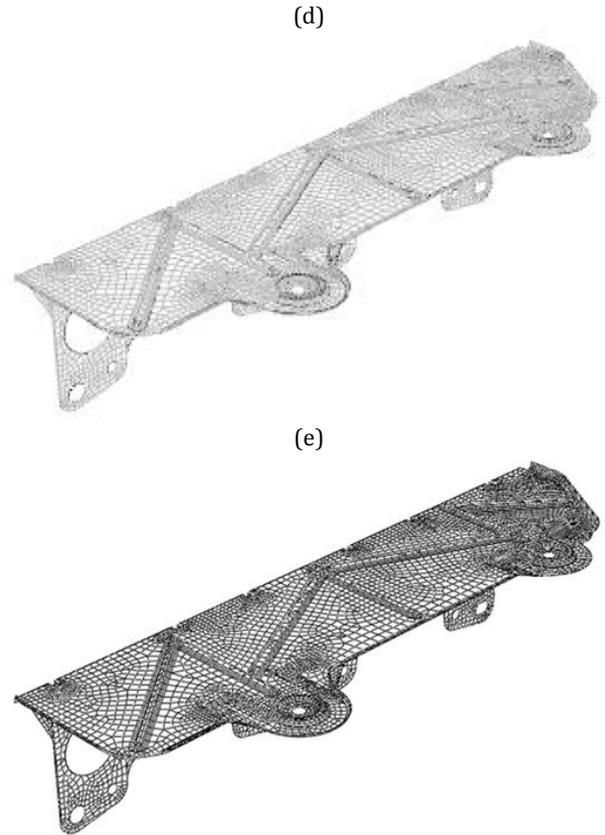


Figure 9 (Continued)

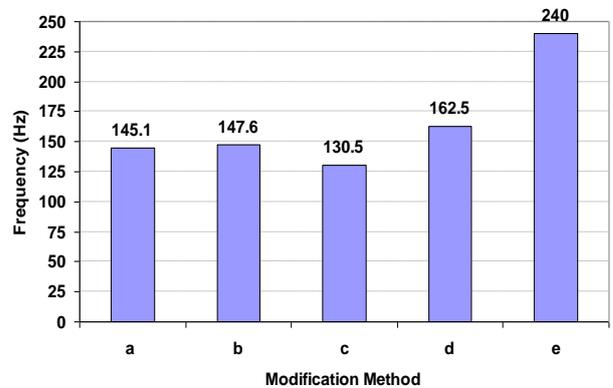


Figure 10: The FEM results of modification methods

7) Conclusion

The failure of the engine heat shield can be caused to the engine noise and melting of the cylinder head valve cover. The investigations show that the dominant load of this part is the vibrational load. In the investigated engine, the vibrational load has been considered in the design of the second heat shield, but due to some modifications on this engine such as changing the fuel system from the carburetor system to the injection system, the excitation frequencies of the engine have been changed and the heat shield has been resonated during the functional tests. The FEM and experimental results of one heat shield show that this part will resonate in two modes of the engine speed work. By increasing the thickness of the heat shield from 0.8 mm to 1.5 mm,

the failure can be avoided according to the resonance of this part, in all ranges of the engine speed.

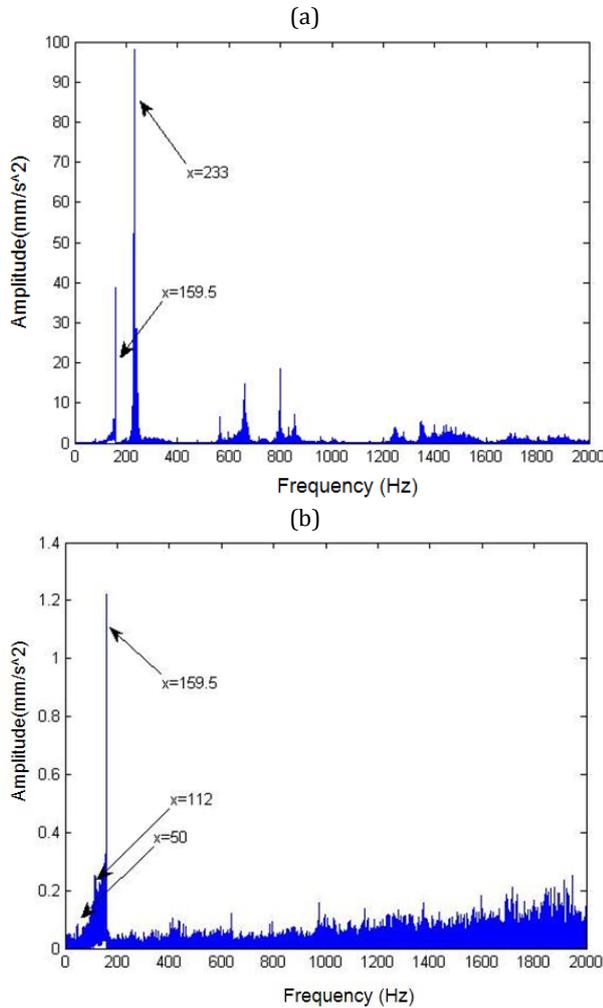


Figure 11: The FFT of the heat shield with 1.5 mm thickness for (a) the heat shield and (b) the cylinder head

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در بسیاری از خودروها بمنظور جلوگیری از انتقال حرارت از چندراهه دود به قطعات فوقانی مانند درپوش دریچه، یک یا دو سپر حرارتی نصب می‌شود. این قطعه در بعضی موتورها، ترک خورده و موجب خرابی‌های بعدی می‌گردد. در این مقاله، به تحلیل خرابی سپر حرارتی یک موتور تحت بارهای ارتعاشی به روش اجزای محدود و تجربی پرداخته شده است. از آنجا که سپرهای حرارتی از ورق نازک ساخته می‌شوند، تحلیل آنها در حوزه ارتعاش پوسته‌ها می‌باشد. بررسی انجام شده در این مقاله نشان می‌دهد که دو بسامد تشدید یک سپر حرارتی در محدوده سرعت کاری موتور می‌باشد که محل ترک‌های ایجاد شده بر روی آن، منطبق بر نقاط با بیشترین خیز قطعه، تحت این بسامدها است.

تمامی حقوق برای انجمن علمی موتور ایران محفوظ است.