ISSN : 0974 - 7435
Volume 10 Issue 21
2014
BioTechnology
An Indian Journal
FULL PAPER
BTAIJ, 10(21), 2014 [13168-13176]

Strength finite element analysis of diesel engine block

Bingxia Liu[1,2], Ping Sun[1*]
[1]School of Automobile and Traffic Engineering, Jiangsu University, Zhenjiang, Jiangsu, 212013, (P.R.China)
[2]School of Energy and Power Engineering, Jiangsu University of Science and Technology, Zhenjiang, Jiangsu, 212003, (P.R.China)
Email: sunping@ujs.edu.cn

ABSTRACT

Based on diesel engine block, using the finite element analysis software ANSYS to simulate the strength. Firstly solid model and finite element model of diesel engine block are established using relevant software, and the necessary simplify operation according to the needs of finite element analysis is carried out on the block ; Then strength check is made to the block model, Stress distribution results are obtained, Which provides the basis for optimizing the block structure.

KEYWORDS

Diesel engine; Block; Finite element; Strength analysis.

INTRODUCTION

The block is one of the most important parts of the engine, and its strength and dynamic characteristics is the guarantee of the engine a prerequisite for normal work. By studying on the structural strength of engine block and show the stress and deformation distribution accurately is very important for guiding the design and improvement of the block [1-5]. Finite element analysis is the most effective and reliable method of studying the stress problem of complex structure at present[6-9]. Strength analysis of diesel engine block with ANSYS are carried out in this paper and has obtained the block's stress and strain nephogram at different working conditions, which can provide ground for structure optimization design of engine block.

ESTABLISH A MODEL AND MESH

In this thesis, solid model of engine block has been built by software Pro/E, and according to the needs of finite element analysis, we have simplified the model. The block entity model simplified mainly includes the following three aspects:

(1) Simplified the local structure of several details. Such as the block's internal horizontal clapboard grooves, local reinforcement, round corners, chamfering have been appropriately simplified. These tiny structures have little impact on the vibration of the whole dynamic characteristics.

(2) The bolt holes processing. In practice, the local stiffness will be strengthened after bolts installed. So we can ignore its groove structure while analyzing stress of the block.

(3) Ignore some of the local structure. The small bolt holes, oil holes and ducts of a fixed role can be ignored because the three-dimensional entity model of the engine block is prepared for the body's stress analysis.

In order to improve the accuracy of the stress analysis, we have used the simplified measures above and preserved some key structure, such as cam mechanism room and the coolant jacket, etc. Physical model simplified is shown in figure 1.

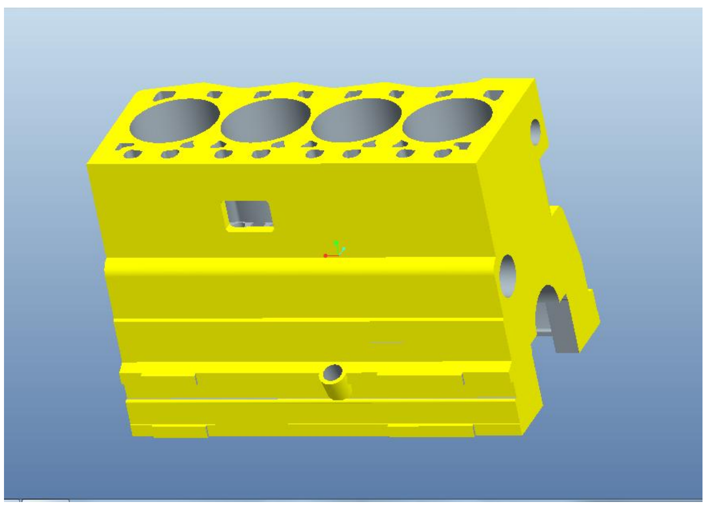


Figure 1: 3-D entity model of diesel engine block

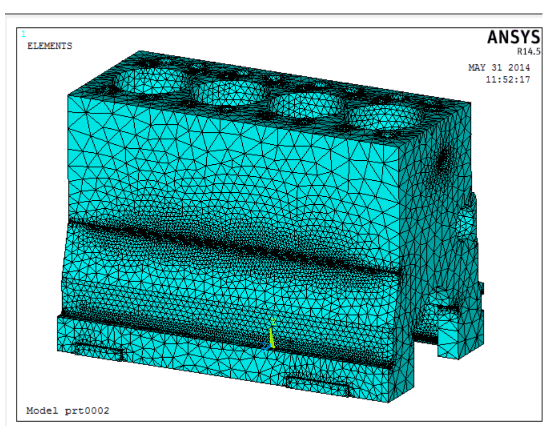


Figure 2: Diesel engine block's FEM

As we have simplified the geometric structure characteristics at the beginning of the modeling phase, the model obtained is more regular. The result is relatively reasonable because we have used intellectual finite element mesh division, which can also save time and efforts. Diesel engine block's finite element model (FEM) is shown in figure 2.

CALCULATION OF THE BODY LOAD

In order to accurately simulate the actual engine work state of the body's stress distribution, the finite element model can not only accurately reflect the characteristics of the body but also require the force applied as much as possible to simulate the actual working conditions. Under operating condition, the force on the engine block is very complex, which brings a lot of difficulties in its strength analysis. So we must simplify the force applied on the diesel engine block. On the basis of the actual response in as far as possible, we have considered the body of the gas pressure in the cylinder, the inertia force of piston connecting rod mechanism and the main bearing load, lateral force of the piston, axial load on the cylinder head bolt, which formed from itself.

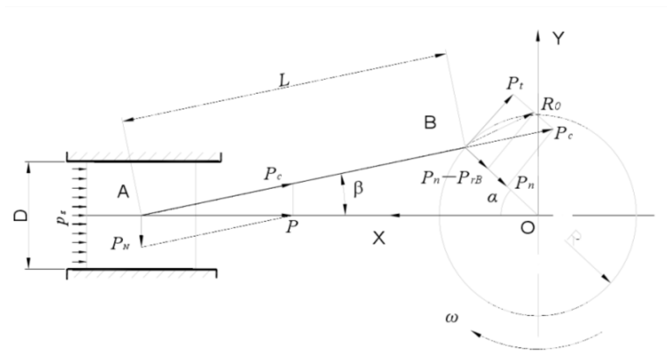


Figure 3: The connecting rod stress analysis diagram

Calculation of the lateral force of piston

The lateral force of piston is mainly composed of the inertial force formed by the reciprocating movement of the piston rod system and the gas pressure in the cylinder. The pressure of gas effect on top of the piston can be determined the size of P_{g} by the instantaneous volume and pressure of diesel engine cylinder body indicator diagram[9].

$$P_{\mathrm{g}} = p_{\mathrm{g}} \frac{\pi}{4} D^2 \tag{1}$$

Strain conditions in working process of cylinder block is shown in figure 3. Because there is a certain angle between connecting rod working and the direction of motion of the piston, it can produce certain piston lateral pressure P_H on the cylinder liner. $\lambda = r / l$, which is the ratio of the radius of the crank and the length of the connecting rod. According to the center crank connecting rod mechanism kinematic principles, we can determine the reciprocating inertial force on the piston pin:

$$P_{\mathrm{j}} = -m_{\mathrm{j}} R\omega^2 \left(\cos\alpha + \lambda\cos 2\alpha\right) \tag{2}$$

In the formula, m_{j} -reciprocating inertial mass of the piston and connecting rod head

ω -angular velocity of the rotating crank

λ -ratio of the radius of the crank and the length of the connecting rod

α -angle of the crank

So, joint force on the piston pin along the center line of the cylinder can be obtained:

$$P_{\text{合}} = P_{\mathrm{g}} + P_{\mathrm{j}} = p_{\mathrm{g}} \cdot \frac{\pi}{4} D^2 - m_{\mathrm{j}} R\omega^2 \left(\cos\alpha + \lambda\cos 2\alpha\right) \quad \ldots\ldots \tag{3}$$

P_H can be represented as:

$$P_{\mathrm{H}} = \left(P_{\mathrm{g}} + P_{\mathrm{j}}\right)\tan\beta = \left(P_{\mathrm{g}} + P_{\mathrm{j}}\right)\frac{\lambda\tan\alpha}{\sqrt{1-\lambda^2\sin^2\alpha}} \tag{4}$$

In the formula, β–swinging angle of connecting rod($\sin\beta = \lambda\sin\alpha$)
Thrust of the connecting rod:

$$P_{\mathrm{c}} = \frac{\left(P_{\mathrm{g}} + P_{\mathrm{j}}\right)}{\cos\beta} = \frac{\left(P_{\mathrm{g}} + P_{\mathrm{j}}\right)}{\sqrt{1-\lambda^2\cos^2\alpha}} \tag{5}$$

Calculation of main bearing load

The crankshaft bearing and shaft neck is influenced by load in working process, whose size and direction is variable with the time. In the process of calculating the load of the diesel engine's main bearing, we need consider the bearing operating temperature, compression area, etc. Here, we only consider the compression area of the bearing and ignore the other factors. Load of the main bearing is caused by the gas pressure in the cylinder on the piston, force through the connecting rod to the crankshaft, and then force on the main bearing at last. Main bearing load can be calculated several ways. This paper puts forward the following hypothesis:

(1) The size of the four cylinder load and change rule is exactly the same;
(2) Ignore the load in the circumferential and axial distribution of the crankshaft surface load;
(3) This article does not consider the effect of friction and elastic deformation of the materials.

Thrust of the connecting rod P_{c} can be decomposed into two mutually perpendicular force components P_t and P_n :

$$P_t = P_{\mathrm{c}}\sin\left(\alpha+\beta\right) = p_{合}\frac{\sin\left(\alpha+\beta\right)}{\cos\beta}$$

$$P_n = P_{\mathrm{c}}\cos\left(\alpha+\beta\right) = p_{合}\frac{\cos\left(\alpha+\beta\right)}{\cos\beta} \tag{6}$$

Centrifugal inertia force of the big end of the connecting rod $P_{rB} = m_{CB}R\omega^2$. The quality of the big end of the connecting rod m_{CB} .
Force on the crank pin :

$$R_B = \sqrt{P_t^2 + \left(P_n - P_{rB}\right)^2} \tag{7}$$

Force on the two main bearings of the single-cylinder engine cylinder:

$$R_1 = \frac{1}{2}R_B = \sqrt{P_t^2 + \left(P_n - P_{rB}\right)^2} \tag{8}$$

Multi-cylinder diesel engine has multiple main bearings, each main bearing is influenced by adjacent two cylinder. Diesel engine is a four-stroke and four-cylinder machine, each cylinder ignition phase angle difference of 180 degree. So we can obtain the size of the two main bearings fore and aft:

$$R_1 = R_5 = \frac{1}{2}R_B = \sqrt{P_t^2 + \left(P_n - P_{rB}\right)^2} \tag{9}$$

The direction is decided by both P_t and ($P_n - P_{rB}$).
Load of the middle main bearing:

$$R_{(\mathrm{i,i+1})} = \sqrt{P_{t(\mathrm{i,i+1})}^{2} + \left(P_{n(\mathrm{i,i+1})} - P_{\mathrm{r}}\right)^{2}} \qquad \textbf{(10)}$$

In fact, the size of the bearing load is also affected by the factors such as oil film thickness, axis path, etc. The size and direction of the force on the main bearing seat is variable with crank angle. Due to the force is surface force, we can take it as a pressure to the relevant plane.

Calculation of bolt axial load

When finite element analysis is conducted on the block, the axial force on the Cylinder head is easily neglected. The axial force and the force acting on the piston crown is a pair of action and reaction, its size is equal to P_{r}, and their directions are opposite. Fixed cylinder head bolt will be forced when any one cylinder works, but the force around the power cylinder is mainly concentrated in a few bolts. We mainly consider the axial force on the bolt around the working cylinder when finite element analysis is conducted on the block, and the size of each axial force on the bolt is equal. The changing relationship of the axial force on the bolts around the remaining three doing work cylinder is the same as the first cylinder bolts', and their phase angle differs by 180 degree. The bolts between two cylinders are acted by two cylinders at the same time, so the force on the two cylinders are added together when computing.

STRENGTH CHECK AND ANALYSIS OF THE BLOCK

When each cylinder of engine is doing work and their explosion pressure achieved the maximum, the crankshaft angle, piston displacement, piston lateral force, main bearing load and gas pressure of each cylinder is shown in TABLE 1-4.

Stress analysis of the piston's lateral force

According to the piston displacement of a moment, we can find the piston lateral force's acting surface where piston acted on the cylinder wall, and then bring pressure on the surface. These pressure means uniform distribution load. The lateral pressure on the cylinder wall is evenly distributed within the scope of the cover angle 120 degree, and then we can obtain the stress nephogram of lateral force of each cylinder under the working condition of maximum explosion pressure, which is shown in figure 4.

TABLE 1: 1st Cylinder's parameters at maximum explosion pressure condition

cylinder number	cylinder 1（inflation）	cylinder 2（exhaust）	cylinder 3（compression）	cylinder 4 （inlet）	
crankshaft angle	390°	570°	210°	30°	
piston displacement(mm)	9	95	95	9	
gas pressure(Mpa)	11.6	0.95	0.16	0.15	
piston lateral force(Mpa)	3.04	−0.4	−0.19	−0.16	
main bearing number	Mbs1	Mbs2	Mbs3	Mbs4	Mbs5
main bearing load (Mpa)	23	13	4	2.5	8

TABLE 2: 2nd Cylinder's parameters at maximum explosion pressure condition

cylinder number	cylinder 1（compression）	cylinder 2（inflation）	cylinder 3 （inlet）	cylinder 4 （exhaust）	
crankshaft angle	210°	390°	30°	570°	
piston displacement(mm)	95	9	9	95	
gas pressure(Mpa)	0.16	11.6	0.15	0.95	
piston lateral force(Mpa)	−0.19	3.04	−0.16	−0.4	
main bearing number	Mbs1	Mbs2	Mbs3	Mbs4	Mbs5
main bearing load (Mpa)	6	19	13	2.5	7

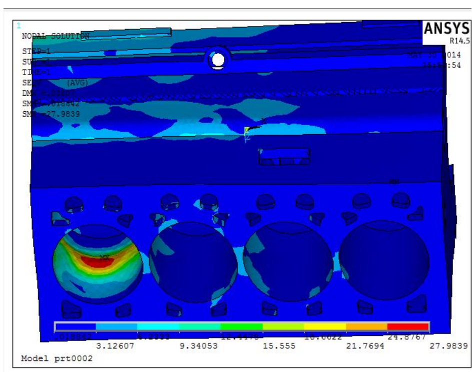
TABLE 3: 3rd Cylinder’s parameters at maximum explosion pressure condition

cylinder number	cylinder 1(exhaust)		cylinder 2 (inlet)	cylinder 3 (inflation)	cylinder 4 (compression)
crankshaft angle	570°		30°	390°	210°
piston displacement(mm)	95		9	9	95
gas pressure(Mpa)	0.95		0.15	11.6	0.16
piston lateral force(Mpa)	-0.4		-0.16	3.04	-0.19
main bearing number	Mbs1	Mbs2	Mbs3	Mbs4	Mbs5
main bearing load (Mpa)	7.5	2.5	20	12	6

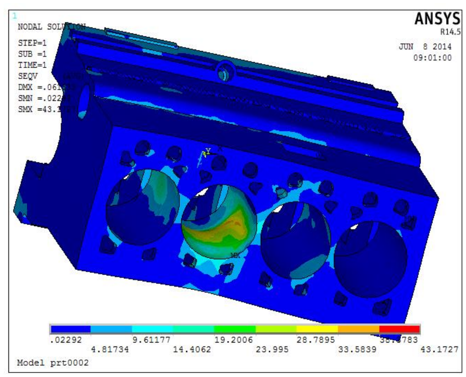
TABLE 4: 4th Cylinder’s parameters at maximum explosion pressure condition

cylinder number	cylinder 1 (inlet)		cylinder 2 (compression)	cylinder 3 (exhaust)	cylinder 4 (inflation)
crankshaft angle	30°		210°	570°	390°
piston displacement(mm)	9		95	95	9
gas pressure(Mpa)	0.15		0.16	0.95	11.6
piston lateral force(Mpa)	-0.16		-0.19	-0.4	3.04
main bearing number	Mbs1	Mbs2	Mbs3	Mbs4	Mbs5
main bearing load (Mpa)	8	4	3	19	23

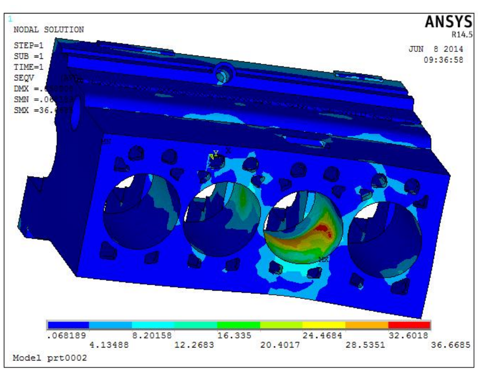
Figure 5 is a deformation figure of the first cylinder block under the working condition of maximum explosion pressure.



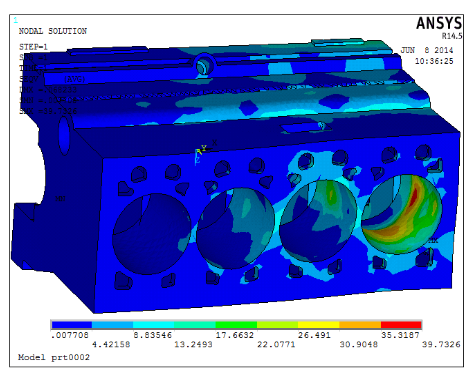
(a) cylinder 1



(b) cylinder 2



(c) cylinder 3



(d) cylinder 4

Figure 4: Stress distribution of the piston lateral force at each cylinder maximum explosion pressure condition

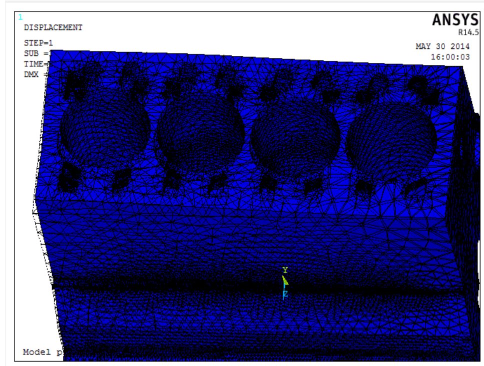
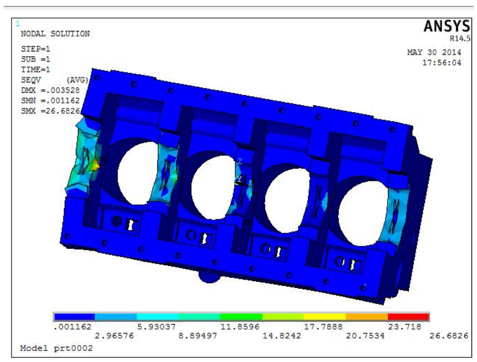
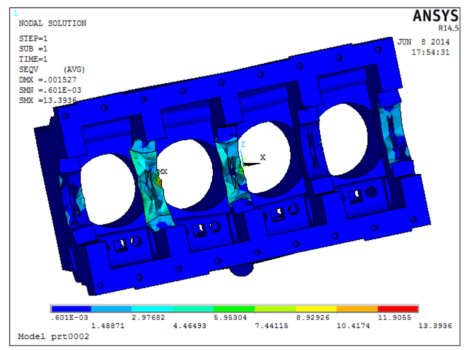


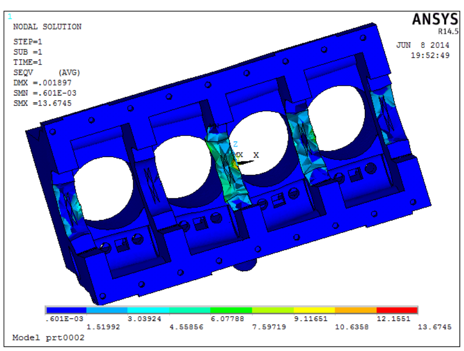
Figure 5: Deformation figure of engine block at the 1st cylinder maximum explosion pressure condition



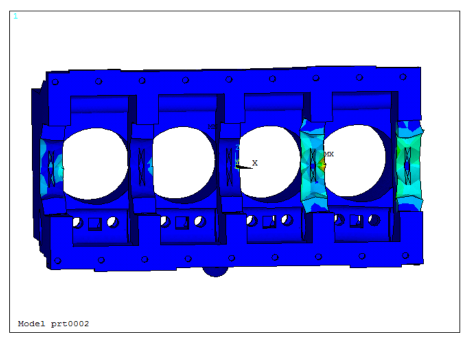
(a) cylinder 1



(b) cylinder 2



(c) cylinder 3



(d) cylinder 2

Figure 6: Stress distribution of the main bearing at each cylinder maximum explosion pressure condition

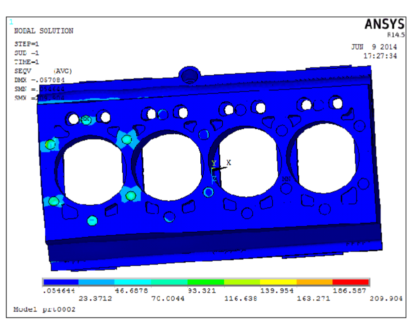


Figure 7: Stress distribution of the roof bolt hole at the 1st cylinder maximum explosion pressure condition

For example, In the first cylinder which is under the maximum explosion pressure condition, we can see that the Maximum stress appeared in the role of the piston lateral force area from figure 4. It means that the piston lateral force has more influence on the first cylinder's stress, and has less influence on the other three cylinders, which is not doing work. The maximum stress of the second cylinder appeared near one side of the first cylinder. It can be seen that the maximum stress of the cylinder is caused by the deformation from the deformation figure of the cylinder when the first cylinder's explosion pressure achieved its maximum. The third cylinder's deformation is small because it lies in the greater distances from the first cylinder. So the third cylinder's stress is mainly affected by the piston's lateral force. Therefore, we can conclude that the stress of the cylinder which is doing work is mainly affected by the piston lateral force. For the non-working cylinder, its maximum stress appeared near one side of the doing work cylinder if it is next to the doing work cylinder. If it is not next to the doing work cylinder, its stress is mainly affected by the cylinder piston's lateral force.

From figure 4, it can be seen that the maximum stress is 40MPa, which appeared at the moment when the fourth cylinder's explosion pressure achieved its maximum. It is far less than the material's ultimate strength 250MPa, so the block is safe.

Analysis of main bearing force

Because gas pressure to the top face of the piston and reciprocating movement inertial force of the piston crank mechanism transfers through connecting rod to the crankshaft journal, and then through crankshaft to the main bearing seat. The main bearing load on the main bearing seat can be handled as cosine distribution load. The cosine function is $P{=}P_{\max}\cos\varphi$. Stress distribution of the main bearing at each cylinder maximum explosion pressure condition is shown in figure 6.

For example, in the first cylinder maximum explosion pressure condition, it can be seen that the maximum stress appeared at a pair of main bearing covers just below the doing work cylinder and the stress of the block's other main bearing covers is relatively small from figure 6. It means that the load of the bearing cover is mainly caused by gas explosion pressure, gas pressure in the process of inlet and exhaust has little effect on the load of the main bearing cover. In the process of the calculation of multi-cylinder engine, we can consider to ignore the non-working cylinder's gas pressure's influence on the load of the main bearing, it can save time.

Analysis of the axial force on the bolt

The contact pressure produced by bolt pre-tightening force only have an effect on the size of stress of the cylinder body's top surface. This article did not take into account in the calculation of bolt pre-tightening force, and uniform load was applied on six threaded holes around each cylinder head. For example, in the first cylinder maximum explosion pressure condition, we have obtained axial force's stress nephogram of the bolt, which was shown in figure 7. From that, it can be seen that the maximum stress of the bolt hole is 116Mpa. It did not exceed the material's tensile strength 250Mpa, so the bolt hole is safe.

CONCLUSION

Based on diesel engine block, using the finite element analysis software ANSYS to simulate, we have obtained the stress of the block which is affected by axial force on the cylinder head bolt, lateral force of the piston and the main bearing load. Through the analysis of the strength of the block, we can conclude that in the case of a cylinder maximum explosion pressure, the cylinder piston's lateral force is the largest, and the maximum stress of the block's cylinder wall appeared at the moment of piston's site of action. The piston lateral force of the cylinder wall which is next to the doing work cylinder is greatly influenced by the doing work cylinder, and the force appeared near one side of the doing work cylinder. In a cylinder maximum explosion pressure conditions, the force on the main bearing seats which just under the doing work cylinder is the largest, the load of the rest of the cylinder's main bearing seats is smaller. The cylinder bolts around the doing work cylinder were forced evenly, the force on the surface of the piston crown was equally split among them.

REFERENCES

[1] Z.C.Zheng, Y.Gao, B.Liu; Finite Element Analysis of ZH195 Diesel Engine Block, I. C. E & Powerplant, **8**, 5-9 **(2008)**

[2] X.Jin, D.X.Xue, X.G.Song; Strength Finite Element Calculations for 6118 Diesel Engine Block,.Agriculture Equipment & Vehicle Engineering, 52,60-63 **(2014).**

[3] S.W.Chyuan; Finite element simulation of a twin-cam 16-valve cylinder structure, Finite Elements in Analysis and Design, 35, 199-212 **(2000).**

[4] M.L.Cao, A.Z.Bu, D.T.Li; A FEA on the Assembly Parts of 6110 Diesel Engine, Transactions of CSICE, **20**, 447-453 **(2002).**

[5] J.H.Wang, Z.C.Zheng, Y.Gao; Finite Element Analysis on 395 Diesel Engine Block, I. C. E & Powerplant, **30**, 29-34 **(2013).**

[6] P.T.Huthwaite; Accelerated finite element elastodynamic simulations using the GPU, Journal of Computational

Physics, **257**, 687-707 **(2014).**
[7] A.L.Braun, A.M.Awruch; Finite element simulation of the wind action over bridge sectional models: Application to the Guamá River Bridge (Pará State, Brazil), Finite Elements in Analysis and Design, **44**, 105-122 **(2008)**.
[8] H.Y.Miao, S.Larose, C.Perron; On the potential applications of a 3D random finite element model for the simulation of shot peening, Advances in Engineering Software, **40**, 1023-1038 **(2009).**
[9] W.L.Wang; Finite Element Analysis of 3100 Diesel Engine Block, Master.s Thesis, Tianjin University, Tianjin **(2007)**.