

Research on Optimization for the Piston Pin and the Piston Pin Boss

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Abstract: Due to the fatigue failure and the fracture injury occurs under the alternative mechanical loads, the optimal design of the piston pin and the piston pin boss is presented depending on the FEA static analysis. The optimization is carried out using the Genetic Algorithm (GA), and the piston noncircular pin hole is used to further reduce the stress concentration on the upper end of the piston pin seat.

Keywords: Genetic Algorithm (GA), piston pin, stress concentration, noncircular pin hole.

1. INTRODUCTION

While the engine developed quickly and engine performance improved greatly, the demand for engine parts is becoming higher and higher. As an important part in engine, piston endures the cyclic gas pressure and the inertial forces at work, and this working condition may cause the fatigue damage of piston, such as piston side wear, piston pin seat crack and so on. The investigations previous indicate that the greatest stress appears on the upper end of the piston pin seat and this stress concentration is one of the mainly reason for fatigue failure [1, 2]. Due to the complicated working environment for the piston, on the one hand, the FEA for the piston became more difficult; on the other hand, though there have many methods which are put forward to apply optimal design, the optimal parameters is not easy to determine.

In this study, the piston is used in a large medium-speed gas engine new developed. In order to enhance the engine dynamic and economic, it is necessary for the piston to implement optimization. The mathematical model of optimization is established firstly, and the FEA is carried out by using the ANSYS software. The GA Toolbox in MATLAB software is used to implement optimization. Based on the analysis of optimal result, the stress concentrates on the upper end of piston pin seat has become smaller, which provides a better reference for redesign of piston.

2. THE FEA OF THE PISTON PIN

2.1. Theoretical Foundation

The piston pin is used to connect the piston and the connecting rods in the internal combustion engine, and the piston pin transfers gas pressure to the connecting rods. The rubbing pair is constituted by the piston pin and piston pin boss; the gas pressure will be delivered to the rubbing pair through the piston pin and piston pin boss, and then the piston pin may occur bending. So the piston and the piston pin should have enough stiffness which can endure the pres-

sure and the friction between contacting surfaces. In addition, as an important part in engine, the working condition of piston is directly related to the reliability and durability of engine. So it is important for the piston pin and the piston pin boss to carry out structural and optimal analysis which can provide reference for design of piston.

The piston pin and the piston pin boss are the greatest parts which endure the pressure at works, and the linear velocity is lower. So the specific pressure between the piston pin and the piston pin boss should be restricted. The formula can be expressed as,

$$q = \frac{p_z - p_j}{2dl_p} \quad (1)$$

Where, p_z is the maximum gas pressure; p_j is the reciprocating inertia force; d is the reciprocating inertia force; l_p is the effective length of piston pin.

2.2. The Finite Element Analysis

In this work, due to the symmetry of structure, 1/4 model of piston and piston pin are assembled in the Pro/E software, and then the 1/4 FEM is established using ANSYS software. The 3-D 20-node solid element SOLID95 is applied to mesh the whole structure, and 102470 nodes and 48839 elements are obtained. In addition, in order to obtain the better result, the contact pair which is important to carry out the research should be established between the piston pin and piston.

The contact is highly non-linear and need more computing power. In order to compute effectively, it is important to know the physical property and establish the reasonable model. In this study, the surface-surface contact is applied to the model.

The FEA of piston pin and piston is carried out by using the ANSYS software. The equivalent stress and the deformation of the piston pin are obtained respectively, as shown in the Fig. (1 and 2).

As shown in the Fig. (1), the maximum stress appears in the middle of pin inner diameter and the value of the stress is 157MPa. As shown in the Fig. (2), the maximum deformation occurs on the inside of piston pin and the value of deformation is 0.837mm.

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The analysis of contact is carried out by using ANSYS workbench software, and the result is shown in the Fig. (3 and 4). It is correspond with the truth.

3. THE MATHEMATICAL MODEL OF OPTIMIZATION

Piston pin and piston pin seat may appear deformation at work, which usually causes crack on the upper end of piston

pin seat, as shown in the Fig. (5a, b) respectively. Due to the deformation, the greatest stress concentration is caused on the upper end of piston pin seat, as shown in the Fig. (5c), the situation becomes more serious when the stiffness of the piston is not enough, and the crack generally appeared at the point A which may gradually extend and even cause splitting along the piston vertical.

The stress distribution on the piston pin seat mainly depends on the deformation of piston pin. Therefore, in order

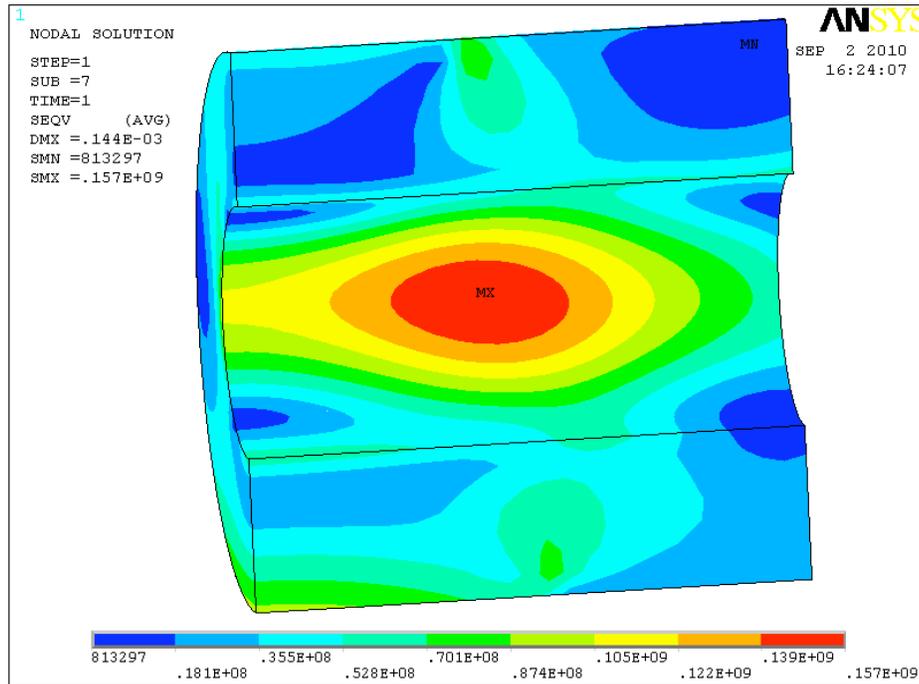


Fig. (1). The equivalent stress of piston pin.

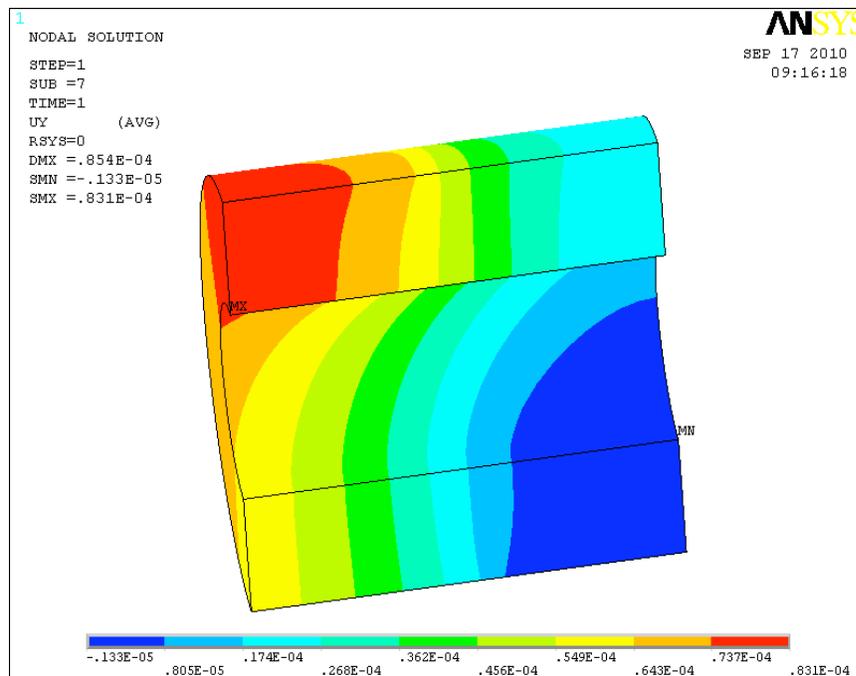


Fig. (2). The deformation of piston pin.

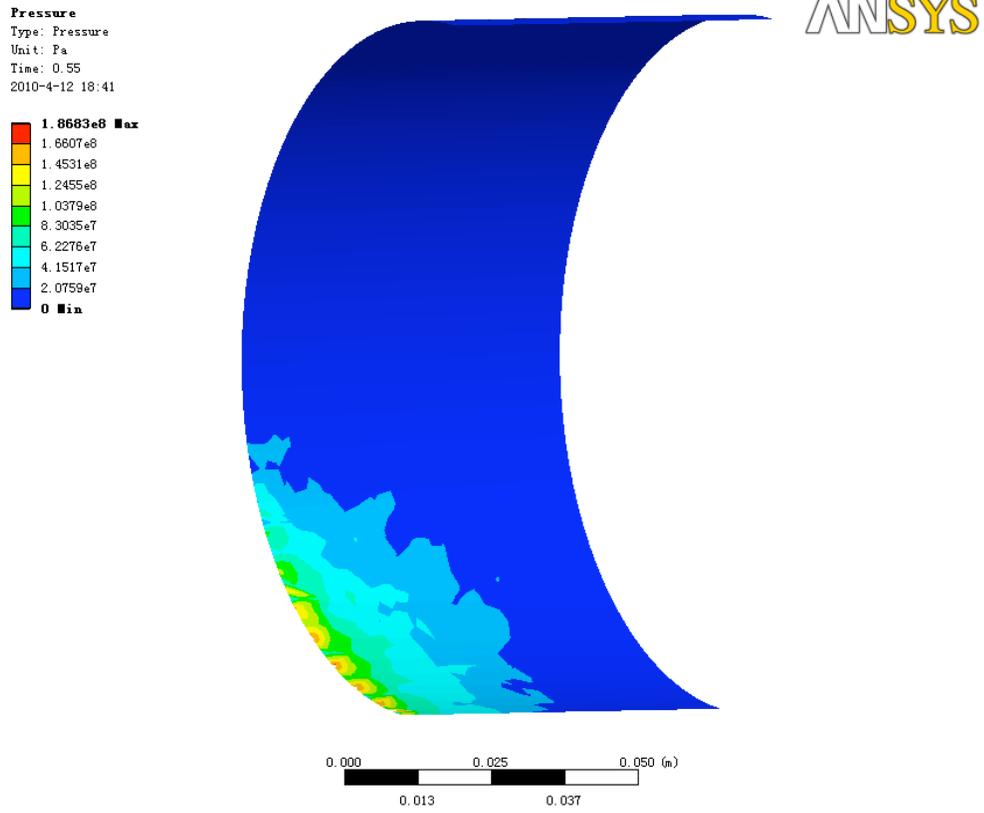


Fig. (3). The stress for the contact surface.

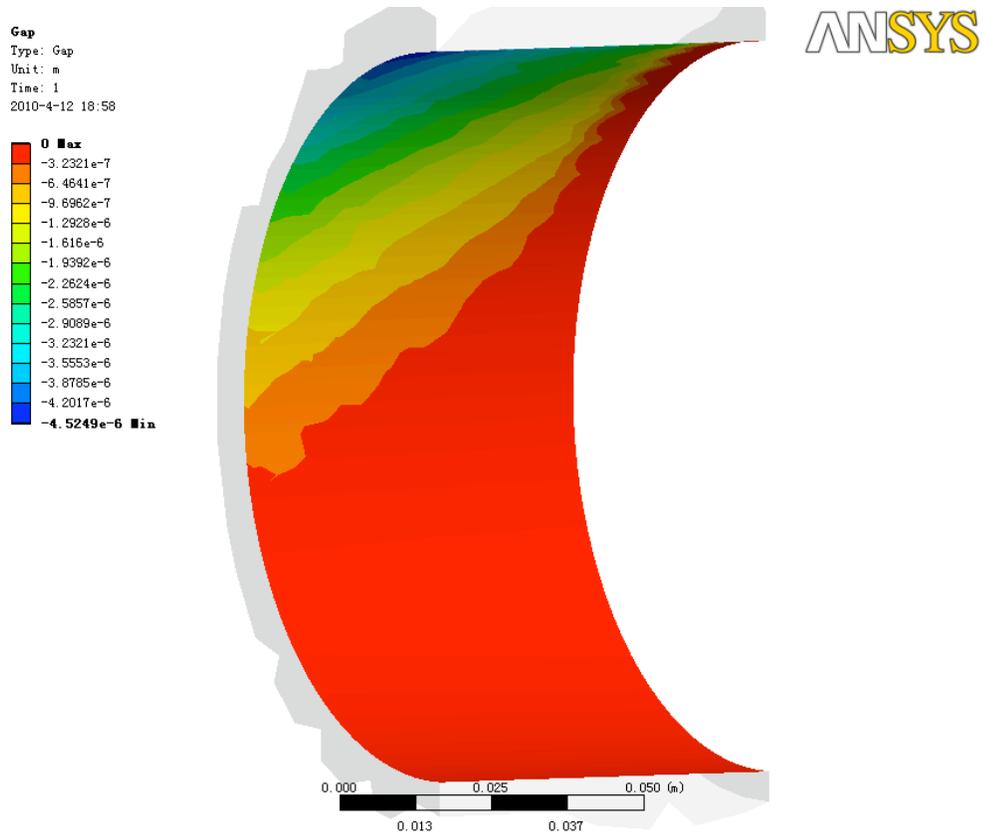


Fig. (4). The spaces between the contact surface.

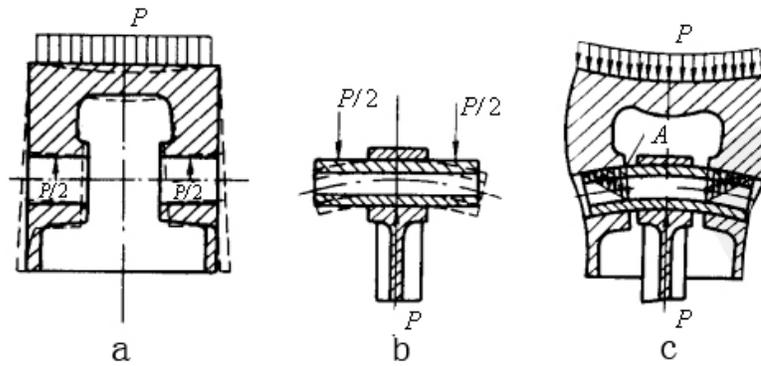


Fig. (5). The deformation sketch of piston pin and piston pin seat.

to reduce the stress concentration, the piston pin should have enough stiffness to reduce the deformation.

3.1. Piston Pin Bending

As the bending is calculated, the piston pin can be considered as a beam which is supported by two endpoints and is loaded in the middle, as shown in Fig. (6). According to the data given in the Fig. (6), the formula of piston pin bending [3] is given as:

$$f_1 = \frac{(p_z - p_j)D^2(l - 2c)^2 l}{60E(d^4 - d_0^4)} \tag{2}$$

Where, l is the length of piston pin; c is the half of the effective length of piston pin; D is the piston diameter, E is the elastic modulus; p_z is the gas pressure; p_j is the reciprocating inertia force.

3.2. Piston Pin Elliptical Deformation

The load on the piston pin is not only distributed along the axial, but also non-uniform distributed along the circumferential direction which is caused by the reacting force of piston pin seat and connecting rod. It is generally regarded as cosine distribution which leads to the piston pin elliptical

deformation. The formula of elliptical deformation [3] is given as,

$$f_2 = \frac{\pi(p_z - p_j)D^2(d + d_0)^3}{320El(d - d_0)^3} \tag{3}$$

3.3. The Quality of Piston and Piston Pin

The quality of the piston can be expressed as,

$$W_1 = c\rho_1\pi(r^2 - d^2) \tag{4}$$

And the quality of piston pin can be given as,

$$W_2 = l\rho_2\pi(d^2 - d_0^2) \tag{5}$$

Where ρ_1 is the density of the piston; r is the outer radius of piston pin; ρ_2 is the density of the piston pin.

3.4. The Mathematical Model of Optimization

According to the above analysis, the multiple objective function is established to optimize the deformation of piston pin and the quality of piston. The objective function can be obtained from Eq. 2, to Eq. 5, and it is shown in the Eq. 6, as follows:

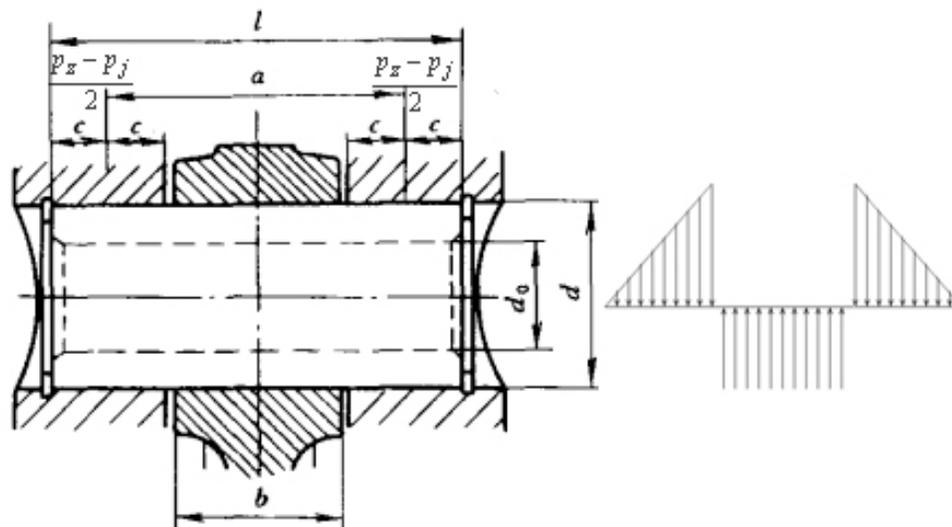


Fig. (6). The force sketch of the piston pin.

$$F = \omega_1 f_1 + \omega_2 f_2 / m_1 + \omega_3 W_1 / m_2 + \omega_4 W_2 / m_3 \quad (6)$$

Where, w_1, w_2, w_3, w_4 are weighting coefficient, m_1, m_2, m_3 are constant which is used to make the value of each objective to be at the same level.

Due to the complexity of piston group, the half of the effective length of piston pin c and the inner diameter of piston pin d_0 are set as the design variable, and the other dimensions are defined as constant.

4. OPTIMIZATION ALGORITHM

GA is applied to solve the optimization problem in this study. The optimization is carried out by using GA Toolbox in MATLAB software. GA is very effective in some ways, such as function optimization, performance evaluation and so on, and the function optimization which is nonlinear, multi-model and multi-objective can obtain better results by using GA [4]. The flow chart of GA is expressed as the Fig. (7). According to the running procedures of GA, the main function is programmed by using the GA Toolbox [5] which is operated in MATLAB software.

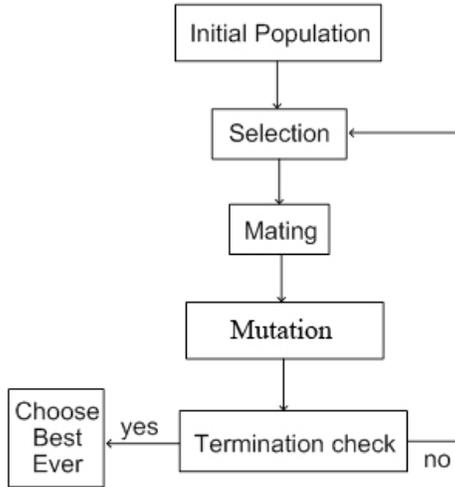


Fig. (7). The flow chart of GA.

The initial population distribution and the population distribution after 150 iterations are shown as Fig. (8a, 8b) respectively. The operation result indicated that the population distribution gathers at the convergence region after

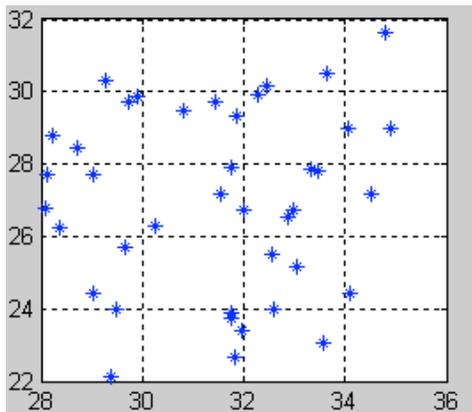


Fig. (8a). The initial population.

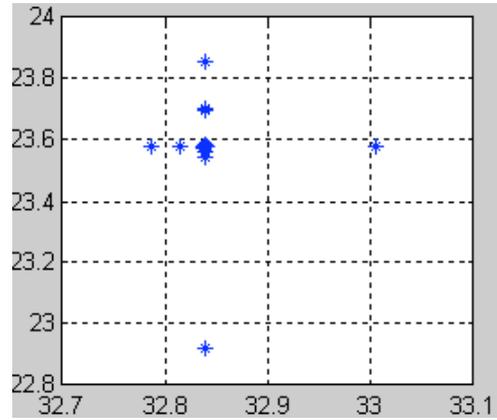


Fig. (8b). The population after 150 iterations.

150 iterations, and the optimal solution is $x_0 = [32.8408, 23.5886]$, that is the best values of the half of the effective length of piston pin c and the inner diameter of piston pin d_0 are 32.8408mm and 23.5886mm.

5. NON-CYLINDER PISTON PIN HOLE

On the basis of piston pin transmigration, the non-cylinder pin hole is applied to the piston so that the stress concentration on the upper end of piston pin seat can be further reduced. The non-cylinder pin hole is shown as the Fig. (9).

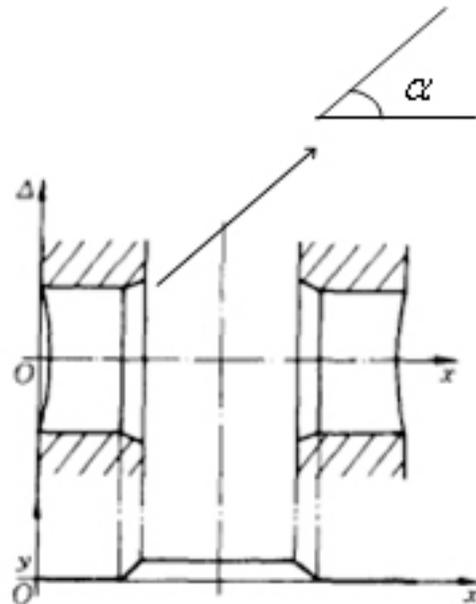


Fig. (9). The sketch of the non-cylinder pin hole.

5.1. The Starting Point

As shown in the Fig. (6), the contact length of the piston pin and piston pin seat is $l_0=2c$. In general, the starting point for non-cylinder pin hole is located in 0.5-0.6 l_0 [6], and it is placed at the point of 0.55 l_0 in this study. As known from the above optimization, the value of l_0 is 65.68mm, and the location of starting point is 36.14mm.

5.2. Taper Angle for Non-Cylinder Piston Pin Hole

According to the FEA basing on the above solutions, the deformation values of piston pin from the starting point to end of piston pin seat are obtained and are shown in the Table 1. According to the Table 1, least squares curve-fitting is carried out by using MATLAB software, and the fitted regression equation is expressed as:

$$y = 0.0001x^2 + 0.0021x + 0.8043 \tag{7}$$

The curve-fitting is shown in Fig. (10). Based on the curve-fitting equation, the taper angle [6] between curve-fitting and horizontal direction is obtained and the value of the taper angle is $\alpha = 0.2104^\circ$.

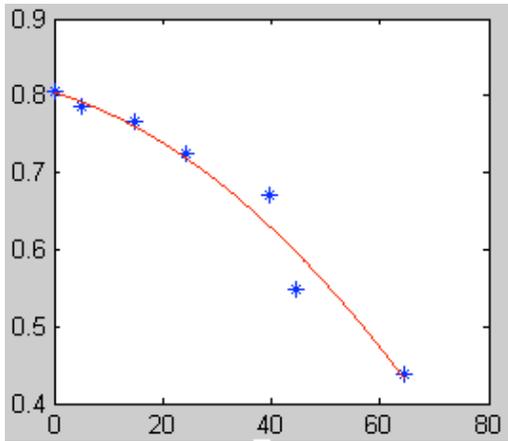


Fig. (10). The least squares curve-fitting.

effective length of piston pin c and the inner diameter of piston pin d_0 have been changed from 28.00mm and 27.00mm to 32.8408mm and 23.5886mm, the non-cylinder pin hole is applied to the optimal piston and the taper angle between curve-fitting and horizontal direction is $\alpha = 0.2104^\circ$. For the initial model of piston and the optimal one, the FEA is carried out by using ANSYS software respectively. The distribution of equivalent stress is shown as Fig. (11a) and (11b). The maximum stress of two models occurs at the upper end of piston pin seat, and the value has changed from 171MPa to 145MPa. The deformation of the piston is shown as Fig. (12a, 12b), and the biggest deformation of two models are generated at the skirt of the piston. The biggest deformation has changed from 0.359mm to 0.301mm.

Table 1. The Deformation Values of Piston Pin

No.	Coordinate (mm)	Deformation (mm)
1	0	0.805
2	4.93	0.785
3	14.81	0.766
4	24.15	0.724
5	39.54	0.671
6	44.53	0.548
7	63.54	0.439

6. OPTIMIZATION ANALYSIS

Compared with the initial piston model, the optimal piston mode is established which the values of the half of the

CONCLUSIONS

1. The optimal mathematical model which includes the deformation of piston pin and the quality of piston

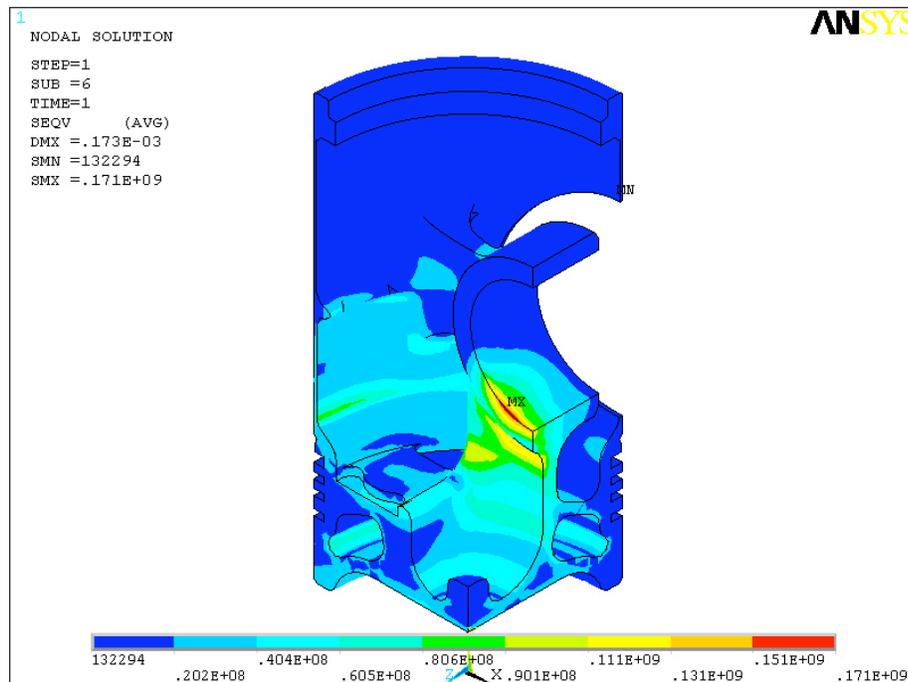


Fig. (11a). The distribution of equivalent stress of the initial model.

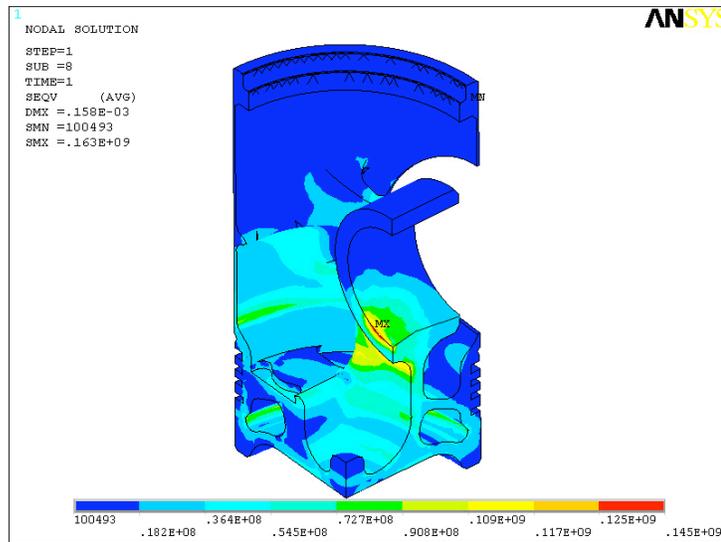


Fig. (11b). The distribution of equivalent stress of the optimal model.

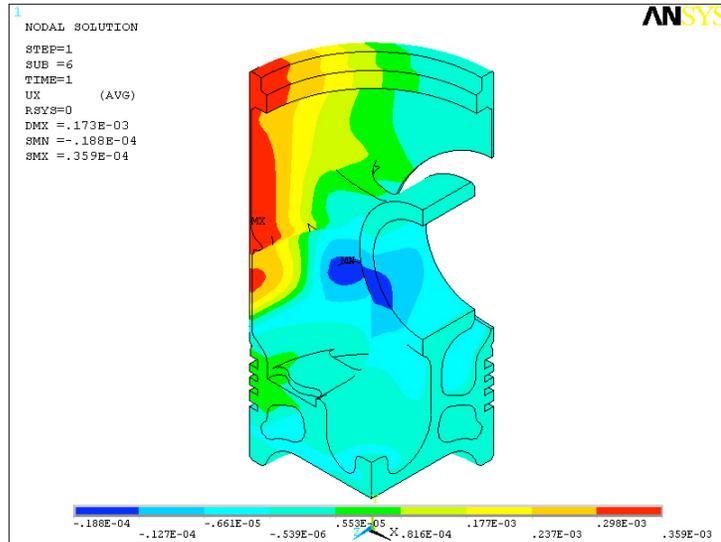


Fig. (12a). The deformation of the initial model.

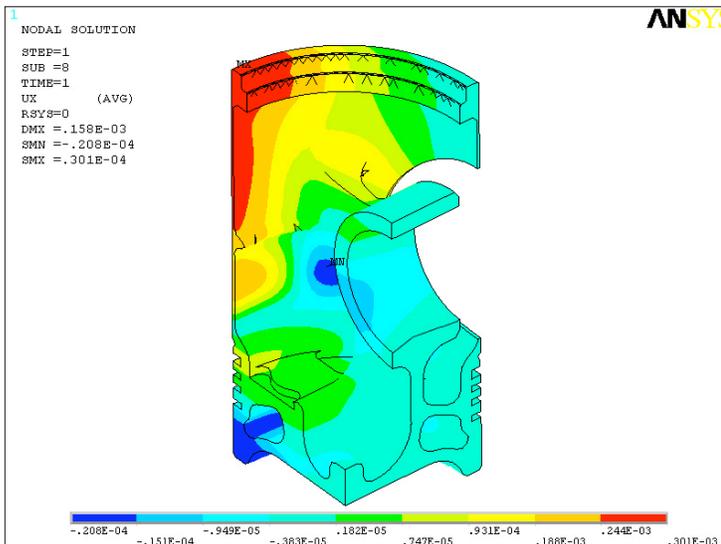


Fig. (12b). The deformation of the optimal model.

and piston pin is established, and then the GA is applied to solve the problem. The optimization values of the effective length of piston pin c and the inner diameter of piston pin d_0 are 65.6814mm and 23.5886mm.

2. The non-cylinder pin hole is applied to the piston. The best taper between curve-fitting and horizontal direction is $\alpha = 0.2104^\circ$.
3. The FEA is carried out for the initial piston model and the optimal one respectively. The results of analysis indicate that the maximum stress has changed from 171MPa to 145MPa, and the biggest deformation has been reduced from 0.359mm to 0.301mm.

ACKNOWLEDGEMENTS

This project is supported by the University science and technology Program of Shandong Province of China (No. J10LD62) and the Nature Science Foundation of Shandong Province (No. ZR2010EM070).

CONFLICT OF INTEREST

None declared.

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Received: February 14, 2011

Revised: April 06, 2011

Accepted: April 06, 2011

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