Research on the Coordination and Matching of Cylinder Head Gasket Strength and Sealing Performance

WANG Longa, ZUO Zhengxingb*, CHENG Yingc, ZHU Xinrand
School of Mechanical Engineering, Beijing Institute of Technology, Beijing, China
a email: WANGLONG9876@foxmail.com
b email: zxzuo@bit.edu.cn

Abstract: In this paper, the cylinder head gasket of a certain type of diesel engine is taken as the research object. Under the condition of changing the mechanical load parameters, the changing trend of the strength and sealing performance of the cylinder head gasket is analyzed, and then its reliability and durability are evaluated. Through the analysis, we can see that the changing trend of the strength and sealing performance of the cylinder head gasket is the opposite. Therefore, in this paper, the fatigue life and sealing performance of the cylinder head gasket are regarded as factors with equal weight, and the total coordination coefficient is introduced to measure the change of the strength and sealing performance of the cylinder head gasket after changing the explosion pressure and bolt preload, the best loading scheme is selected.

1. Introduction
Cylinder head gasket is a seemingly simple component, but it plays an irreplaceable role in internal combustion engines [1]. Although its volume is small, it is used as a tight-sealing part to connect the cylinder head and block (or cylinder liner) of the diesel engine, and it plays a sealing role in the combustion chamber, cooling water channel, lubricating oil channel, etc. If the cylinder head gasket fails due to damage or poor installation, it will cause water leakage, gas channeling, and other problems, which will affect the working efficiency of the diesel engine, and even lead to serious problems such as ablation.

To improve the reliability and fatigue life of cylinder head gasket of heavy-duty diesel engine, Dong Yi et al. used orthogonal experiment, hybrid neural network and other methods and algorithms to optimize the operation coefficient of cylinder head gasket, the accuracy of the algorithm, and the effectiveness of the improvement was proved by finite element analysis results[2].

Sk kandreegula et al. studied the influence of the bolt preload of the compression bolt on the sealing efficiency of the cylinder head gasket by using the finite element numerical simulation method, and obtained that the thermal load will affect the position of the maximum contact pressure on the cylinder head gasket[3].

O Aizawa et al. studied the influence of circulating water temperature on the sealing performance of diesel engine cylinder head gasket[4].

In the combustion chamber assembly, there are much researches on the fatigue life of cylinder head, engine block, cylinder liner, and other components, but there are little researches on the fatigue life of cylinder head gasket because most of the cylinder head gaskets are the multi-layer composite structure, there are complex structures such as ribs, and materials such as asbestos are used, which makes it more complicated to calculate their fatigue life. However, in this article, the material of the cylinder head...
gasket is steel, and its structural form is a ring of equal height, so the relevant fatigue life calculation formula can be used to calculate its life.

This paper studies the influence of bolt preload and explosion pressure on the sealing performance and fatigue life of cylinder head gaskets.

2. Establishment of finite element model

In this paper, each cylinder has an independent cylinder head and cylinder head gasket. To save the amount of calculation, this paper does not use the overall model of diesel engine, but cuts the engine block, and assembles it with the cylinder head gasket and other parts in the form of one whole cylinder and two half-cylinders.

When analyzing the cylinder head gasket, the finite element model needs to include cylinder head, engine block, cylinder head bolt, cylinder head gasket, and other components [10]. The model studied in this paper also needs to add a cylinder liner and cylinder liner support ring.

2.1 Meshing

Considering the calculation scale and calculation accuracy, the cylinder block and cylinder head adopt subdivision quadratic element mesh, and other parts adopt hexahedral linear element mesh. The mesh generation of the cylinder head gasket is shown in Fig 1 below. 120 nodes are arranged in the circumferential direction (that is, the nodes are separated by 3°), and 6 nodes are arranged in the radial direction and thickness direction respectively. The finite element model of combustion chamber assembly is shown in Fig 2 below:

![Figure 1. Finite element model of cylinder head gasket](image1)

![Figure 2. Finite element model of the combustion chamber assembly](image2)

2.2 Boundary conditions

2.2.1 Displacement boundary conditions

The assembly displacement boundary conditions include fixed constraint and symmetry constraint. In this paper, the fixed constraint is set at the crankshaft mounting hole on the cylinder block, the partition plane of the cylinder block adopts symmetric constraint.

2.2.2 Mechanical load

In this paper, the mechanical load on the combined structure includes two parts: the bolt preload of the cylinder head bolt and the gas explosion pressure in the cylinder. The bolt preload of each cylinder head bolt is 130kN, and the cylinder head is connected to the engine block by four pre-tightening bolts. The gas explosion pressure in the cylinder is $p = 22\text{MPa}$.

2.2.3 Thermal boundary conditions

In this combined structure, the cylinder head and the cylinder liner are directly in contact with the high-
temperature gas, so it is necessary to apply thermal boundary conditions to these two components, and then determine the temperature field of the cylinder head gasket through the heat transfer between the components, to analyze its stress state and fatigue form under the thermal-mechanical coupling.

3. Calculation results and analysis

3.1 Analysis of Cylinder Head Gasket Temperature Field

The temperature field distribution of the cylinder head gasket is shown in Fig 3. In this view, the positive direction of the Z-axis is the exhaust valve side.

![Figure 3. Cloud map of cylinder head gasket temperature field distribution](image)

From the calculation results of the cylinder head gasket temperature field, it can be seen that the temperature value of the cylinder head gasket near the exhaust side of the cylinder head is the highest, which is 173.1°C (446.25K). This is because the temperature of the gas on the exhaust side of the cylinder head is high. In other positions, the temperature on the exhaust side of the cylinder head is significantly higher than in other positions, which indirectly affects the temperature distribution at the cylinder head gasket.

3.2 Fatigue Life Analysis of Cylinder head gasket

The equivalent stress distribution nephogram of cylinder head gasket in the preloaded state and the thermal-mechanical coupling state is shown in Fig.4.

![Figure 4. Equivalent stress distribution nephogram of cylinder head gasket](image)

From Fig 4, it can be seen that the Mises stress value of the cylinder head gasket under the action of bolt preload + cylinder explosion pressure is smaller than that in the pre-tightened state, which indicates that the action direction of the explosion pressure on the cylinder head gasket is opposite to that of the cylinder head bolt on the cylinder head gasket. And the stress value of the cylinder head gasket under the thermal-mechanical coupling condition is higher than that under the other two conditions, which indicates that the effect of the thermal load on the cylinder head gasket is the same as the effect of the cylinder head bolt on the cylinder head gasket and is greater than the effect of the explosion pressure in the cylinder.

In this paper, the yield limit of the material used for the cylinder head gasket is $\sigma_y = 175\text{MPa}$, under the preloading conditions and the thermal-mechanical coupling conditions, plastic deformation occurs at the positions where the cylinder head gasket stress is higher, so the fatigue form of the cylinder head gasket is low cycle fatigue.

The position of the cylinder head gasket close to the exhaust side of the cylinder head is selected as the inspection point, as shown in the black circle in Fig 4(c).
When calculating the fatigue life, a multi-axis fatigue life assessment model based on the critical plane theory is adopted here. The determination of the critical plane is determined by programming, the specific process is shown in Fig 5 below.

In this paper, the SWT model [11] modified by Socie [12] is adopted, assuming that the critical plane is the plane with the maximum principal strain, it is considered that the crack propagation perpendicular to the direction of the maximum tensile stress is the main stage of fatigue life, and the maximum normal strain amplitude and the maximum normal stress on the plane where the maximum normal strain amplitude is located are taken as the damage parameter, the expression is as follows:

$$\frac{\Delta \varepsilon_{n}^{\max}}{2} \sigma_{n}^{\max} = \frac{\sigma_{f}^{f}}{E} (2N_{f})^{2b} + \sigma_{f}^{f} \varepsilon_{f}^{f} (2N_{f})^{b+c}$$

(1)

Where:
- $\sigma_{f}^{f}$—Fatigue strength coefficient;
- $b$—Fatigue strength index;
- $\varepsilon_{f}^{f}$—Fatigue ductility coefficient;
- $c$—Fatigue ductility index;
- $N_{f}$—Fatigue life;
- $\Delta \varepsilon_{n}^{\max}$—Maximum normal strain range;
- $\sigma_{n}^{\max}$—The maximum normal stress on the plane where $\Delta \varepsilon_{n}^{\max}$ is located;
- $E$—Young’s modulus.

The material parameters of the cylinder head gasket are as follows: $\sigma_{f}^{f} = 896MPa$, $b=-0.12$, $\varepsilon_{f}^{f} = 0.41$, $c=-0.51$, $E=219000MPa$. 

Figure 4. Equivalent stress cloud diagram of cylinder head gasket
Determine the stress and strain time history of the survey location

Determine damage parameters

Determine the damage parameters on any plane

θ, φ change from 0° to 180°

Is it the maximum?

Yes

No

Maximum damage plane

Critical plane and damage parameters

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Figure 5. Flow chart of determining the critical plane

Figure 6. Variation curve of normal strain amplitude with azimuth angle at the inspection point in the cylinder head gasket

The variation trend of normal strain amplitude with azimuth θ and φ at the inspection point of the cylinder head gasket is shown in Fig 6, when θ = 76° and φ = 138°, the normal strain amplitude is the maximum value of Δε₁₀₁₀₁ = 0.02596 in each plane.

In a loading cycle, only one plane has the maximum normal strain amplitude, and this plane is regarded as the critical plane. On this plane, the maximum normal stress is σ₁₀₁₀₁ = 30.95MPa.

By substituting the above data into formula (1), the fatigue life at the inspection point of the cylinder head gasket is calculated to be \( N = 14,660 \) times.

3.3 Cylinder head gasket sealing performance evaluation

By establishing a contact surface with the cylinder head and cylinder liner, the cylinder head gasket can seal compressed air and gas in the combustion chamber. For the sealing performance of cylinder head gasket, the commonly used evaluation indicators include the contact pressure extremum \( σ_{pmax} \) and \( σ_{pmin} \), the contact pressure difference \( σ_{pmax} - σ_{pmin} \), the contact pressure ratio \( σ_{pmax}/σ_{pmin} \). In this paper, the maximum contact pressure \( σ_{pmax} \), the minimum contact pressure \( σ_{pmin} \) and the contact pressure difference \( σ_{pmax} - σ_{pmin} \) are used to evaluate the sealing performance of the cylinder head gasket.

The cylinder head gasket seal is required to be circumferentially continuous, that is, the contact pressure of each point in the circumferential direction is within the required range. On the premise of meeting this requirement, the cylinder head gasket seal coordination evaluation is carried out, and the seal evaluation is required to be the uniformity of contact pressure. The smaller the extreme value of the contact pressure difference of cylinder head gasket, the better the sealing performance. The evaluation factor of cylinder head gasket seal is introduced here, which is defined as:
\[ \xi = \begin{cases} 
1 - \frac{\sigma_{p_{\text{max}}}-\sigma_{p_{\text{min}}}}{\sigma_{b}-5*P_f}, & \sigma_{p_{\text{min}}} \leq 5*P_f \land \sigma_{p_{\text{max}}} < \sigma_{b} \\
0, & \sigma_{p_{\text{min}}} > 5*P_f \lor \sigma_{b} < \sigma_{p_{\text{max}}}
\end{cases} \]

Where:
- \( \xi \) — The evaluation factor of cylinder head gasket seal;
- \( \sigma_{p_{\text{max}}} \) — The maximum contact pressure;
- \( \sigma_{p_{\text{min}}} \) — The minimum contact pressure;
- \( \sigma_{b} \) — The strength limit of the cylinder head gasket material;
- \( P_f \) — Maximum value of gas explosion pressure.

To ensure that the cylinder head gasket has reliable sealing performance, it is necessary to ensure that the contact pressure on the cylinder head gasket is greater than 5 times the maximum explosion pressure. When the minimum contact pressure is less than 5 times of the maximum explosion pressure, the gas is easy to leak and the sealing performance will fail, at this time, \( \xi = 0 \); when the maximum explosion pressure is greater than the strength limit of the cylinder head gasket, the irreversible deformation of the cylinder head gasket will be caused, and then the gas will leak and the seal fails, at this time, \( \xi = 0 \). At the same time, the smaller the maximum difference of contact pressure, the more uniform the contact pressure distribution, and the larger the sealing evaluation factor.

According to the above definition, it is necessary to calculate the contact pressure curve of the cylinder head gasket under the preload condition. The sealing of the cylinder head gasket is realized by the contact ring surface of the cylinder head and the cylinder head gasket and the contact ring surface of the cylinder liner and the cylinder head gasket. Now, the contact area between the cylinder head and the cylinder head gasket is divided and the nodes are numbered. As shown in Fig 7, positions 2, 4, 6, and 8 are the action positions of the bolts, and positions 1, 3, 5, and 7 are the positions far away from the bolts. There are 120 nodes in the circumferential direction of the contact annulus and 6 nodes in the radial direction, that is, the circumferential direction of the contact ring is 360 degrees, divided every 3 degrees, and there are 6 nodes in the same radial direction. The contact pressure values of six nodes in the same radial direction are calculated and extracted, and the average value is obtained, which is regarded as the contact pressure value at the angle of the contact area. The contact pressure at each angle in the contact ring surface is calculated and plotted as shown in Fig 8.

It can be seen from Fig 8 that on the contact pressure curve, the contact pressure value near the bolt acting position is higher, and the value on the exhaust side is higher than the intake side, and the highest value of the contact pressure on the cylinder head gasket appears on the exhaust side (It is shown at 2 and 8 in Fig 8), which is \( 227.699 \text{MPa} \). The value of contact pressure on the cylinder head gasket far away from the bolt is relatively low, and the lowest value of contact pressure appears between the two bolts (It is shown at 3 and 7 in Fig 8), which is \( 175.06 \text{MPa} \). The main reason for the uneven distribution of contact pressure on the cylinder head gasket is that the distance between different areas and bolts is different. At the same time, due to the irregular internal structure of the cylinder head, the force acting on the cylinder head gasket is different, which affects the stress distribution on the cylinder head gasket.

The maximum value \( \sigma_{p_{\text{max}}} = 227.699 \text{MPa} \) and the minimum value \( \sigma_{p_{\text{min}}} = 175.06 \text{MPa} \) on the contact pressure curve, the ultimate strength of the cylinder head gasket material in the combustor combination structure is \( \sigma_{b} = 295 \text{MPa} \), and the maximum gas explosion pressure \( P_f = 22 \text{MPa} \) are substituted into the calculation formula of cylinder head gasket sealing evaluation factor, the sealing evaluation factor of the combined structure of combustion chamber is obtained as \( \xi = 0.7155 \).
4. Analysis of the influence of load parameters on the strength and sealing performance of the cylinder head gasket

4.1 Analysis of the influence of bolt preload on the strength and sealing of the cylinder head gasket

This section studies the influence of the change of the bolt preload on the fatigue life and sealing performance of the cylinder head gasket. The initial value of the bolt preload of the model used in this paper is 130kN. To analyze the change rule of cylinder head gasket strength and sealing performance under different bolt preload levels, the explosion pressure is set at 22Mpa. The life, contact pressure distribution, and sealing evaluation factor of cylinder head gasket under five working conditions of bolt preload of 125kN, 130kN, 135KN, 140KN, and 145KN are calculated respectively. The calculation results are shown in Fig 9, Fig 10, and Fig 11.

Figure 7. Distribution of contact pressure nodes

Figure 8. Contact pressure curve of cylinder head gasket

Figure 9. The curve of fatigue life of cylinder head gasket with different bolt preload
It can be seen from Fig 9 that with the increase of bolt preload, the lifetime value of the cylinder head gasket decreases continuously, which indicates that the increase of bolt preload will not only increase the stress value of cylinder head gasket under mechanical load condition and thermal-mechanical coupling conditions but also increase the difference of stress value under these two conditions, that is, increase the stress amplitude and reduce the fatigue life of cylinder head gasket.

It can be seen from Fig 10 that with the increase of bolt preload, the pressure value in the contact area of cylinder head gasket increases uniformly; while it can be seen from Fig 11 that the value of seal evaluation factor increases linearly with the increase of bolt preload. Therefore, with the increase of bolt preload, not only the contact pressure value of the cylinder head gasket is increased, but also the uneven distribution of the contact pressure value of the cylinder head gasket is reduced, the sealing coordination is improved, and the sealing performance of the cylinder head gasket is enhanced.

4.2 Analysis of the influence of explosion pressure on the strength and sealing of cylinder head gasket

In this section, we study the impact of explosion pressure change on the fatigue life and sealing performance of cylinder head gasket. The initial value of explosion pressure of the model used in this paper is 22Mpa. To analyze the change law of cylinder head gasket strength and sealing performance under different explosion pressure levels, the bolt preload is set to 130kN, and the fatigue life and sealing evaluation factors of cylinder head gasket under five working conditions of explosion pressure of 20MPa, 21MPa, 22MPa, 23MPa, and 24MPa are calculated respectively. The calculation results are shown in Fig 12 and Fig 13.
It can be seen from Fig 12 that with the increase of explosion pressure, the fatigue life of the cylinder head gasket increases continuously, which indicates that the increase of explosion pressure will reduce the difference of stress value of cylinder head gasket under mechanical load condition and thermal-mechanical coupling condition, it means the stress amplitude is reduced.

It can be seen from Fig 13 that the sealing evaluation factor of the cylinder head gasket decreases with the increase of the explosion pressure because when the bolt preload is determined, the contact pressure difference of the cylinder head gasket is determined. According to formula (2), the larger the explosion pressure is, the smaller the calculated value is, so the sealing performance decreases.

4.3 Comparison of results of different design schemes
To measure the influence of bolt preload and explosion pressure on the fatigue life and sealing performance of the cylinder head gasket, the strength and sealing coordination performance of the cylinder head gasket under different mechanical loads are evaluated. The concept of total coordination coefficient is proposed here, which unifies the variables, that is, the fatigue life and sealing performance of cylinder head gasket are regarded as factors with equal weight, based on the calculation results of the initial value of the mechanical load (the explosion pressure is 22MPa, and the bolt preload is 130KN). After changing the explosion pressure and the bolt preload, the strength of the cylinder head gasket, and the comprehensive performance of the seal change. If the result is a positive value, it indicates that the comprehensive performance is improved, otherwise, the comprehensive performance is decreased.

The definition formula of the total coordination coefficient is as follows:

\[
\alpha_i = \frac{1}{2} \left( \frac{N_i - N_0}{N_0} + \frac{\xi_i - \xi_0}{\xi_0} \right)
\]  

(3)

Where: \(\alpha_i\)—The total coordination coefficient of cylinder head gasket in the i-th design scheme;
\(N_0\)—The fatigue life value of the cylinder head gasket when the bolt preload is 130KN and
the explosion pressure is 22MPa;

\( \xi_0 \) — The sealing evaluation factor of the cylinder head gasket when the bolt preload is 130KN and the explosion pressure is 22MPa;

\( N_i \) — The fatigue life value of the cylinder head gasket in the i-th design scheme;

\( \xi_i \) — The sealing evaluation factor of the cylinder head gasket for the i-th design scheme.

The two load parameters of bolt preload and explosion pressure have five design levels respectively. In this section, the life value and seal evaluation factor of cylinder head gasket under all 25 combination design schemes are calculated, and then the total coordination coefficient is calculated. The calculation results are shown in Table 1.

| Experiment | Bolt preload (KN) | Explosion pressure (MPa) | Fatigue life | The evaluation factor of cylinder head gasket seal | The total coordination coefficient |
|------------|-------------------|--------------------------|--------------|-----------------------------------------------|----------------------------------|
| Experiment 1 | 125 | 20 | 23050 | 0.72937 | -0.03112 |
| Experiment 2 | 125 | 21 | 24660 | 0.72225 | -0.00403 |
| Experiment 3 | 125 | 22 | 26800 | 0.71474 | 0.03335 |
| Experiment 4 | 125 | 23 | 28360 | 0.70682 | 0.05889 |
| Experiment 5 | 125 | 24 | 28750 | 0.69844 | 0.06080 |
| Experiment 6 | 130 | 20 | 22030 | 0.73006 | -0.05096 |
| Experiment 7 | 130 | 21 | 23520 | 0.72296 | -0.02624 |
| Experiment 8 | 130 | 22 | 25100 | 0.71547 | 0 |
| Experiment 9 | 130 | 23 | 26840 | 0.70757 | 0.02914 |
| Experiment 10 | 130 | 24 | 28750 | 0.69921 | 0.06134 |
| Experiment 11 | 135 | 20 | 19930 | 0.73138 | -0.09187 |
| Experiment 12 | 135 | 21 | 21170 | 0.72431 | -0.07211 |
| Experiment 13 | 135 | 22 | 22500 | 0.71686 | -0.05082 |
| Experiment 14 | 135 | 23 | 25300 | 0.70900 | -0.00054 |
| Experiment 15 | 135 | 24 | 26910 | 0.70068 | 0.02572 |
| Experiment 16 | 140 | 20 | 18580 | 0.73331 | -0.11741 |
| Experiment 17 | 140 | 21 | 19640 | 0.72629 | -0.1012 |
| Experiment 18 | 140 | 22 | 20760 | 0.71889 | -0.08416 |
| Experiment 19 | 140 | 23 | 21940 | 0.71108 | -0.06602 |
| Experiment 20 | 140 | 24 | 23190 | 0.70283 | -0.04688 |
| Experiment 21 | 145 | 20 | 16940 | 0.73515 | -0.1488 |
| Experiment 22 | 145 | 21 | 17810 | 0.72819 | -0.13633 |
| Experiment 23 | 145 | 22 | 18750 | 0.72084 | -0.12274 |
| Experiment 24 | 145 | 23 | 19810 | 0.71308 | -0.10705 |
| Experiment 25 | 145 | 24 | 20900 | 0.70489 | -0.09106 |

It can be concluded from Table 1 that under the same level of bolt preload, the greater the explosion pressure is, the higher the fatigue life of the cylinder head gasket is, and the worse the sealing performance is. Under the same level of explosion pressure, the higher the bolt preload is, the lower the fatigue life of the cylinder head gasket is, and the better the sealing performance is. The variation rules under different levels of bolt preload and explosion pressure are consistent with the conclusions in Section 4.1 and Section 4.2.

Compared with the calculation results of 25 different design schemes, the total coordination coefficient of scheme 10 (bolt preload is 130kN, explosion pressure is 24MPa) is the largest. Compared with the default scheme (bolt preload is 130kN, explosion pressure is 22MPa), the fatigue life of this scheme is increased by 14.542%, the sealing performance is decreased by 2.273%, and the total
coordination coefficient is increased by 6.134%. The results show that the comprehensive performance of the strength and sealing of the cylinder head gasket under this scheme is better than other design schemes. Under this scheme, the sealing performance is reduced, but the minimum contact pressure on the cylinder head gasket is still greater than the maximum value of 5 times the explosion pressure, and the sealing coordination factor is not 0, which can ensure better sealing performance without air leakage. Therefore, if only considering the strength and sealing performance of the cylinder head gasket, the scheme is feasible.

5. Conclusion
In this paper, by changing the bolt preload and cylinder explosion pressure (each factor has five levels), the changing trend of cylinder head gasket strength and sealing performance is analyzed. When the bolt preload increases, the thermal-mechanical coupling fatigue life of the cylinder head gasket decreases, and the sealing performance increases; when the cylinder explosion pressure increases, the thermal-mechanical coupling fatigue life of the cylinder head gasket increases, and the sealing performance is reduced. It can be found that the changing trend of strength and sealing is the opposite. The comprehensive performance of strength and sealing is measured by the total coordination coefficient. Comparing each design scheme with the initial scheme, the scheme with the highest total coordination coefficient is the optimal scheme. The research of this paper can provide some reference for the mechanical load design of the combined structure of the internal combustion chamber.

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