Structure Design and Simulation of Titanium Engine Piston Based on Thermal-Mechanical Coupling Model

Yaochen Xu
Mo Yang

Department of Mechanical Engineering
Blekinge Institute of Technology
Karlskrona, Sweden
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Abstract:
Piston is the ‘heart’ of the automobile engine. It’s one of the key components of the engine and it’s working the hard condition which accelerated the piston wear and broken. A good design of the piston in this thesis is compared with existing piston to extend the Mean Time Between Maintenance. In order to achieve the deformation, thermal and stress distribution of the piston, ANASYS software is used to analyze the piston under the thermal loads and mechanical loads. The results are shown that the temperature distribution occurs on the top of the piston when the piston under the thermal load and the greatest stress occurs on the piston pin when the piston under the thermal-structure coupling. The temperature distribution is conformed to the facts, but the greatest stress is a bit large when coupling.

Keywords
Acknowledgements

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*Mo Yang*

*Yaochen Xu*
6. Finite Element Analysis ................................................................. 41
   6.1 Software introduction ................................................................. 41
   6.2 Materials ................................................................................. 43
   6.3 The Thermal-Mechanical Coupling analysis .............................. 44
       6.3.1 Meshing ......................................................................... 44
       6.3.2 Static forces boundary conditions .................................... 45
       6.3.3 Thermal boundary conditions ......................................... 46
       6.3.4 Results of temperature distribution ................................. 48
       6.3.5 Results of deformation distribution ................................. 50
       6.3.6 Results of stress distribution ........................................... 51
7. Comparison ............................................................................... 53
8. Discussion and Conclusions ....................................................... 55
9. Future works ............................................................................ 56
10. Reference ............................................................................... 57
11. Appendices ............................................................................. 59
1. Notation

1.1 List of symbols

\( \alpha \)  
Excess air ratio

\( B \)  
Pin seat interval

\( d \)  
Piston pin diameter

\( H \)  
The height of the piston

\( H_1 \)  
The height of the compress

\( H_2 \)  
Skirt length

\( H_{\text{paar.cm}} \)  
The calorific value of the working mixed gas

\( n_1 \)  
Polytropic index of compression

\( n_2 \)  
Polytropic index of expansion

\( M_1 \)  
Fresh air volume

\( T_a \)  
The end of the intake temperature

\( T_b \)  
The temperature of the end expansion process

\( T_r \)  
The temperature of the exhaust gas

\( T_z \)  
Combustion terminal temperature

\( \Delta T \)  
The change of the fresh air temperature

\( p_a \)  
Inlet end of the pressure

\( p_b \)  
The pressure of the end expansion process

\( p_c \)  
Compression terminal pressure

\( p_M \)  
The average pressure of the mechanical loss

\( p_r \)  
Residual gas pressure

\( p_z \)  
Actual combustion pressure

\( p_i \)  
The theory of the mean indicated pressure

\( \gamma_y \)  
Coefficient of residual gas

\( \mu_0 \)  
Theoretical molecular changes coefficient

\( \mu_2 \)  
Actual molecular changes coefficient

\( \rho \)  
Initial expansion ratio

\( \delta \)  
The thickness of the top of piston

\( \delta_g \)  
Skirt wall thickness

\( \zeta_z \)  
Heat utilization factor

\( \eta_e \)  
Effective efficiency

\( \eta_v \)  
Coefficient of charge
### 1.2 Abbreviations

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Full Form</th>
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<tbody>
<tr>
<td>FEM</td>
<td>Finite Element Method</td>
</tr>
<tr>
<td>FEMA</td>
<td>Failure modes and Effects Analysis</td>
</tr>
<tr>
<td>LHV</td>
<td>Low heating value</td>
</tr>
<tr>
<td>MTBM</td>
<td>Mean Time between Maintenance</td>
</tr>
<tr>
<td>PM</td>
<td>preventive maintenance</td>
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</table>
2. Introduction

Piston is the ‘heart’ of the automobile engine. It’s one of the key components of the engine and it’s working the hard condition. The function of the piston is bearing the gas pressure and making the crankshaft rotation through the piston pin.

Piston works in high temperature, high pressure, high speed and poor lubrication conditions. Piston contact with high temperature gas directly, the instantaneous temperature can be up to 2500K. Because of the high temperature and the poor cooling condition, the temperature of the top of the piston can be reach 600~700K when the piston working in the engine. And the temperature distribution is uneven. The top of the piston bears the gas pressure, in particular the work pressure. Gasoline engine can be up to 3~5Mpa and diesel engine can be up to 6~9Mpa. It makes the piston produce the impact and bear the side pressure [1]. The piston works in high speed (8~12m/s) reciprocating motion, and the speed is changing, so it makes a large inertial force, which makes the piston bear a great additional load. Working in these bad conditions, the piston accelerated wears, meanwhile produces the additional load, thermal stress and chemical corrosion of the gas.

2.1 The current situation of the piston design

General piston is cylinder, but according to the different working conditions and requirements, the construction of the piston can be various. Generally the piston is divided into three parts: head, the skirt and piston pin.

The head is the part of the piston top and ring groove. The piston top is completely depending on the requirements of the combustion chamber, the top is designed with flat or nearly flat to be conducive to reduce the area contacting with the high temperature gas, so that the stress can be distributed uniformity. Most gasoline engine uses flat top piston. Piston is designed to be a complex shape, with a depth pit as a part of the combustion chamber, to meet the need of mixture gas in order to improve the combustion efficiency and reduce the deflagration to the minimum extent. The recess of the piston is for mounting the piston ring. The piston ring is sealed to prevent air leakage and prevent the oil entering the combustion chamber.
The skirt is the lower part of the piston. It keeps the piston working in the reciprocating movement of the vertical posture. That is, it’s the guide portion of the piston. The shape of the piston skirt is very particular, especially like the light passenger cars, the designers consider the skirt from the engine structure and performance to make the engine’s structure compact and smooth operation. The piston pin is the supporting portion via a piston pin and connecting rod, its located above the piston skirt.

2.2 New material and design of piston

Europe’s largest piston manufacturer Mahler, by optimizing the design of the diesel engine piston and meticulous improvements of the piston materials, it reaches the Euro V emissions standards.

Mahler has developed two higher grader of alloy, which can withstand higher thermal load and pressure load, the new alloy is not only increase 60% in friction strength, but also reduce 15% in wear rate.

With increasingly stringent requirements of the vehicle’s engine power, economic, environmental and reliability, the piston has developed into a set of lightweight high strength new material, deformed cylindrical composite surface and non-circular piston pin hole. As high-tech products to ensure the piston’s heat resistance, wear resistance, smooth oriented and good sealing function, also reduce the engine friction power loss, the fuel consumption, noise and emissions. In order to meet the above functional requirements, it is usually designed the piston’s outer circumference to be deformed cylindrical, that is perpendicular to the axis of the piston cross section or amendment elliptical. Ovality’s accuracy can be up to 0.005mm and its changes in a rule along the axis direction.

Based on the related researches, aluminum alloy is mostly used material in making car pistons, and experiments using other material such as cast iron, cast steel, ceramics and carbon as piston material of diesel engines done at Zhongnan University [2].

The advantage of the aluminum alloy is its density leading it to reduce the mass of a
piston and the inertia of reciprocal motion. Due to the insufficiency of inherent thermal strength of aluminum alloy, the use of aluminum alloy in a diesel engine is restricted [2].

Ceramic is another material which can be used in automobile engines manufacturing. The advantages of the ceramic are its lightness, good abrasion, high resistance, heat insulation properties and high-temperature strength. The modular ceramic piston has been used in some specific engines, but full ceramic piston has no examples of successful application. The main obstacle hindering the use of ceramic is the brittleness of ceramics resulting to low reliability [2].

Cast steel pistons have high mechanical strength, and their heat resistance, corrosion resistance and abrasion resistance are superior to aluminum alloys and cast irons, while having a stable high temperature performance and low coefficient of linear expansion. The drawback is that the density of cast steel is too large that leads to wear and tear occurring in the cylinder liner [2].

3. Background

Vehicle maintenance describes the act of inspecting or to testing the vehicle’s components and replacing the fluids. Preventive maintenance is to ensure the vehicle’s safety, longevity and drivability. In preventive maintenance (PM), a lot of components to be replace to get more safety drive. [3]

Common car’s maintenance tasks include:
- Car wash
- Check and replace the engine oil and replace oil filters.
- Replace fuel filters
- Tires for pressure and wear
- Replace brake pads
- And so on……

Preventive maintenance can be a valuable investment, increasing fuel efficiency and saving the expense of emergency repair service and the inconvenience of a breakdown [4]. PM is conducted to keep equipment working and extend the life of the components.
3.1 MTBM

MTBM means mean time between maintenance, the mean time between unscheduled on-equipment maintenance actions caused by design or manufacturing defects [5]. This measure includes:
• chargeable inherent maintenance actions
• unscheduled maintenance
• on-equipment maintenance (line or organizational level)

3.2 Comparison of the maintenance project per 10000km

The section illustrated which is the most important part of the cars. Here shows some tables of the parts which have been maintained frequently per 10000km. Then compare with the tables and find the key parts of the cars.

Table 3.1.2012 corolla 1.8GL manual [6]

<table>
<thead>
<tr>
<th></th>
<th>10000</th>
<th>20000</th>
<th>30000</th>
<th>40000</th>
<th>50000</th>
<th>60000</th>
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<tbody>
<tr>
<td>Engine oil</td>
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<td>•</td>
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<td>•</td>
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<tr>
<td>Engine filter</td>
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<td>•</td>
<td>•</td>
<td>•</td>
<td>•</td>
<td>•</td>
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<tr>
<td>Air filter</td>
<td>•</td>
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<td>•</td>
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<tr>
<td>Air filtration</td>
<td></td>
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</tbody>
</table>

Table 3.2.2011 Teana 2.5L CVT XE [7]

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<thead>
<tr>
<th></th>
<th>10000</th>
<th>20000</th>
<th>30000</th>
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<td>Engine filter</td>
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<tr>
<td>Air filter</td>
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</tr>
</tbody>
</table>
From the table 3.1, table 3.2 and table 3.3, the engine oil and air filter are maintained frequently. And in engine working, the amount of the engine oil determined the operating frequency of the machine filter. So in the maintenance project, the engine oil can be said that it is the most important one. In engine working, engine oil can be direct effect on the piston. So this thesis is considered to design the piston model and analyze the thermal-mechanical coupling model. Then compare the material of the common piston with another material of piston.
4. Structure design on piston

Before making a design of piston, the type of engine should be chosen and some of calculation works need to be done. The aim of this preparatory work is to get the parameters of the engine including temperature range, pressure range and Strength Check. This will facilitate the designers to determine the main dimensions of the 490 engine piston and structural details. After that, it will pave the way for simulation modeling on thermal-mechanical coupling.

4.1 The original parameters and structure of the engine

The selected engine is the type 490. The main parameters show as Table 4.1.

<table>
<thead>
<tr>
<th>Table 4.1. The parameters set by the engine factory</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
</tr>
<tr>
<td>Number of cylinders</td>
</tr>
<tr>
<td>Diameter of the cylinder</td>
</tr>
<tr>
<td>Stroke</td>
</tr>
<tr>
<td>The form of air jacket</td>
</tr>
<tr>
<td>Combustion chamber</td>
</tr>
<tr>
<td>Minimum steady speed of idling(r/min)</td>
</tr>
<tr>
<td>Rated power/speed</td>
</tr>
<tr>
<td>Maximum speed(r/min)</td>
</tr>
<tr>
<td>Maximum torque(N.m/rpm)</td>
</tr>
<tr>
<td>Minimum fuel consumption(g/kw.h)</td>
</tr>
</tbody>
</table>
### 4.2 Introduction of the main parameters of the engine working process [9]

#### 4.2.1 Excess air ratio $\alpha$

Installing EFI systems on modern engines, which can be guaranteed to get the almost the ideal composition of the mixture gas by the speed characteristic. In order to make the engine as far as possible to get enough economy and sought to reduce the combustion product’s hazard, when $\alpha = 1.2 \sim 1.8$ that minor hazard can be reached. For diesel engines, $\alpha$ is always greater than one to ensure the diesel fuel which injected into the cylinder can be completely burned. When the diesel engine sucked a certain amount of air, if $\alpha$ is small that means it can be inject the fuel to the cylinder, also means that the suction air of the cylinder with high utilization and make a big power. Thus, $\alpha$ is a reflection of an indicator about the formation of the mixture gas, the perfect degree of the combustion and the performance of the engine.

General range of values $\alpha$ when diesel engine works at full load: supercharged diesel engine: $\alpha = 1.8 \sim 2.2$; non-supercharged diesel engine: $\alpha = 1.2 \sim 1.8$

#### 4.2.2 Heat utilization factor $\xi_z$

$\xi_z$ reflects the part which can be improved the gas internal energy and successful conversion in fuel LHV. It can be considered by the structure of the engine, the working condition, the cooling system, the shape of combustion chamber, the coefficient of the excess air and the speed of the engine crankshaft... It can be
determined on the basis of the experimental data. According to the experimental data, the value of non-supercharged engine is between 0.8~0.95.

### 4.2.3 Residual gas pressure $p_r$

$P_r$ can be determined by the number of valve, the layout of valve, timing phase, supercharging characteristic, the load condition, the cooling system and many other factors. Non-supercharged engine $p_r = 1.08p_0$, supercharged engine $p_r = 0.95p_k (P_0 = 0.1MP, P_k = 0.17MP)$.

### 4.2.4 Polytropic index of compression $n_1$

According to the rotational speed of the crankshaft, the compression ratio, the cylinder dimensions, the piston and the cylinder’s material and other factors, the experiment can be done to get the value $n_1$. Taking into account the compression process is very fast (0.015s~0.005s), it can be used to estimate the $n_1$ by the average adiabatic index.

### 4.2.5 Polytropic index of expansion $n_2$

Polytropic index of expansion $n_2$ can be selected by experimental data. With the heat utilization coefficient increases, the ratio of the piston stroke $S$ and cylinder diameter $D$ increases and the cooling intensity increases, the value of $n_2$ is also increased. With the load increases and the cylinder linear dimension increases ($S/D$ is constant), $n_2$ is decreased. And when improve the engine high-speed, the value of $n_2$ usually decreases.

### 4.3 The calculation of the engine working process

All the equations in 4.3 come from the reference [9].
4.3.1 Operating parameters

a) Lower calorific value of fuel
According carbon proportion is 87%, hydrogen proportion is 12.6% and oxygen proportion is 0.4%. Then it can be calculated as below.

\[
H_u = 33.91C + 125.6H - 10.89(O - S) - 2.51(9H + W)
\]

\[
H_u = 42500-44400MJ/kg
\]

Where C is the proportion of carbon, H is the proportion of hydrogen, O - S is smaller enough can be negligible, W is unit mass or unit volume of the steam quantity when fuel burning.

b) The amount of air required for combustion
Calculated the amount of air \(L_0\) required for burning 1 kg fuel by thousand mole theory and the amount of air \(l_0\) required for burning 1 kg fuel by kilogram. Then the calculation is shown as below.

\[
L_0 = \frac{1}{0.21(\frac{C}{12} + \frac{H}{4} - \frac{O}{32})}
\]

\[
L_0 = \frac{1}{0.21(0.87 + \frac{0.123}{4} + \frac{0.04}{32})} = 0.495 \text{ (kmol/kg)}
\]

\[
l_0 = \frac{1}{0.23(\frac{n}{3C} + 8H - O)}
\]

\[
l_0 = 14.45 \text{ air } \frac{kg}{fuel kg}
\]

c) Excess air coefficient \(\alpha\)
When rated engine speed, \(\alpha\) is 1.3

d) Fresh air volume \(M_1\)
According to the reference, the amount of the burning mixture gas \(M_1\) is calculated as below.
That
\[ M_1 = \alpha L_0 = 0.636 \left( \frac{kmol}{kg} \right) \]  \hspace{1cm} (4-4)

e) **The temperature of the exhaust end gas** \( T_r \) **and terminal pressure** \( p_r \)

According to the reference, the calculation is shown.

\[ T_r = \frac{350}{\text{log}_n + 0.005(\varepsilon - 1) + 0.01(\alpha - 1)} \]  \hspace{1cm} (4-5)

\[ p_r = 1.08 \ast P_0 \]  \hspace{1cm} (4-6)

Where \( n \) is the speed of the engine, \( P_0 \) is the atmospheric conditions and \( P_0 = 0.1 \)

That \( T_r = 849 \) (K)  and  \( p_r = 0.108 \) (Mpa)

### 4.3.2 Intake process

a) **The change of the fresh air temperature** \( \Delta T \)

The value of the temperature \( \Delta T \) is related with the structure of intake manifold and the arrangement, also related with the high speed of the engine and supercharger factors. When it’s supercharger, that \( \Delta T = 5 \sim 10 \). When it’s non-supercharger, that \( \Delta T = 10 \sim 40 \).

Then

\[ \Delta T = \frac{\Delta T_n (110 - 0.0125n)}{110 - 0.0125n \varepsilon} \]  \hspace{1cm} (4-7)

\[ \Delta T_n = 20 \) (c)\]

After calculated the result is \( \Delta T = 20 \) (k)

b) **Inlet end of the pressure** \( p_a \)

From the equation as below

\[ p_a = p_0 \ast (1 - \frac{\varepsilon - \delta_1}{2.4} \ast n \varepsilon \ast \phi \ast (\varepsilon - 1) \ast 10^{3}))^2 / 520)^{\frac{k}{k-1}} \]  \hspace{1cm} (4-8)
Where $p_0=0.1\text{Mpa}$, $\delta_1 = 0.5$ is the residual waste shrinkage coefficient, $\phi = 0.7$. The $P_a=0.99\text{ (Mpa)}$

c) Coefficient of residual gas $\gamma_Y$

$$\gamma_Y = \frac{T_0+\Delta T}{T_r} \times \frac{p_r}{\varepsilon^* p_a - p_r}$$

(4-9)

That $T_0 = 288\text{(k)}$, $\gamma_Y = 0.04$

d) The end of the intake temperature $T_a$ and coefficient of charge $\eta_v$

$$T_a = \frac{T_0+\Delta T+\gamma_Y T_r}{1+\gamma_Y}$$

(4-10)

$$\eta_v = \varepsilon * p_a * \frac{T_0}{\varepsilon-1} * P_0 * T_a * (1 + \gamma)$$

(4-11)

Where $\varepsilon$ is the ratio of compression. That $T_a = 325.93\text{ (k)}$ and $\eta_v = 0.84$

4.3.3 The process of compression

a) Polytropic index of compression $n$

From the reference, it is easily to get the value of $K_i$, and it is easily to get the $n_i$ from $k_i$

That $n_i=1.36$

b) Compression terminal pressure $P_c$

The equation is

$$p_c = p_a \varepsilon^{-n_i}$$

(4-12)

Then the result is $P_c=5.091\text{ (Mpa)}$
c) Compression terminal temperature
The equation is.

\[ T_c = T_a e^{ni-1} \]  \( (4-13) \)

The result is \( T_c = 922.62 \) (k)

4.3.4 Combustion process

a) The theoretical molecular changes coefficient \( \mu_0 \) and the actual molecular changes coefficient \( \mu_2 \)

\[ \mu_0 = \frac{M_2}{M_i} \]  \( (4-14) \)

\[ \mu_2 = \frac{\mu_0 + \gamma \gamma}{1 + \gamma \gamma} \]  \( (4-15) \)

Then the result is \( \mu_0 = 1.051 \) and \( \mu_2 = 1.049 \)

b) The calorific value of the working mixed gas \( H_{pa\sigma.cM} \)
The equation is

\[ H_{pa\sigma.cM} = \frac{H_a}{[M_i(1+\gamma \gamma)]} \]  \( (4-16) \)

c) Combustion terminal temperature \( T_z \)
The equation is

\[ T_z = t_z + 273 \]  \( (4-17) \)

\[ (m c_v^c)_t = \left( \frac{1}{M_2} \right) [M_{CO_2} (mc_{CO_2})_t + M_{CO} (mc_{CO})_t + M_{H_2O} (mc_{H_2O})_t + M_{H_2} (mc_{H_2})_t] \]  \( (4-18) \)
\[ \zeta_Z H_{p a s x M} + (m c_v)'_{t_0} = \mu (m c_v)'_{t_0} \quad (4-19) \]

\[ \mu = \frac{\mu_0 + \gamma}{1 + \gamma} \quad (4-20) \]

where \( \mu_0 \) is the coefficient of variation of the mixture gas \( \mu_0 = \frac{M_2}{M_i} \), \( \mu \) is the variant coefficient of the working mixture gas molecular, \( (m c_v)'_{t_0} \) means the average molar ratio of the products of the combustion.

\( \zeta_Z \) means heat utilization factor, the value is related with the structure of the engine, cooling system, the shape of the combustor, the coefficient of the excess air and the speed of the engine crankshaft. From the research, it shows that when the engine under full load, the value of \( \zeta_Z \) is 0.750~0.95. And for the engine, the result is calculated:

\[ \zeta_Z = 0.75 \quad \text{and} \quad T_Z = 1854.4 (k) \]

**d) Actual combustion pressure** \( p_z \)

The equation is

\[ p_z = \lambda P_a \quad (4-21) \]

For the supercharged diesel engine, \( \lambda = 1.5 \). And for the non-supercharged diesel engine that \( \lambda = 2.0 \). Then the result of \( p_z \) is 7.5 (Mpa).

**e) Initial expansion ratio** \( \rho \)

From the reference, the equation of initial expansion ratio is

\[ \rho = \mu * \frac{T_Z}{\lambda} * T_c \quad (4-22) \]
4.3.5 The process of expansion and exhaust

a) Polytropic index of expansion $n_2$

For the equation

$$n_2 = 1.14 + 0.035 \times \frac{n_e}{n}$$  \hspace{1cm} (4-23)

The result is calculated be $n_2 = 1.19$

b) The temperature $T_b$ and the pressure $p_b$ of the end expansion process [9]

From the equation

$$p_b = p_z \delta^{n_2} \hspace{1cm} (4-24)$$

$$T_b = \frac{T_z}{\delta^{n_2-1}} \hspace{1cm} (4-25)$$

So the result is

$$p_b = 0.241 \text{ (Mpa)} \text{ and } T_b = 1070 \text{ (k)}$$

c) The theory of the mean indicated pressure $p_i$

From the reference, equation shown as

$$\dot{p}_i = \frac{p_c}{\varepsilon} - 1 \left[ \lambda \left( 1 - \frac{1}{\varepsilon^{n_2-1}} \right) - \frac{1}{n_1-1} \times (1 - \varepsilon^{n_1-1}) \right]$$  \hspace{1cm} (4-26)

4.3.6 Effective index of engine

a) Mean effective index pressure $p_i$

The equation is

$$P_i = \varphi_u P_i$$  \hspace{1cm} (4-27)
Where $\varphi_u$ is the coefficient of the index diagram and $\varphi_u = 0.96$. That the result is

$$P_i = 0.832 \ (Mpa)$$

b) The index of thermal efficiency and the index of the fuel consumption efficiency

$$\eta_i = \frac{p_l c \alpha}{H_u \rho_0 \eta_v} \quad (4-28)$$

$$g_i = \frac{3600000}{H_u \eta_i} \quad (4-29)$$

Then the result of calculation is

$$\eta_i = 0.506 \quad \text{and} \quad g_i = 239.32 \left( \frac{g}{\text{KW} \cdot \text{h}} \right)$$

c) The average pressure of the mechanical loss $p_m$

For the equation

$$p_m = 0.1 \sqrt{\gamma} \left( 1 + \frac{n}{1000} \right) \quad (4-30)$$

Where the kinematic viscosity of lubricants $\gamma = 25$

d) Effective pressure and mechanical efficiency

For the equation

$$\eta_m = 1 - \frac{p_m}{p_i} \quad (4-31)$$

$$p_e = p_i \eta_m \quad (4-32)$$

Then get the result as
\[ \eta_m = 0.80 \quad \text{and} \quad p_e = 0.64 \ (Mpa) \]

e) **Effective efficiency** \( \eta_e \)

From the equation

\[ \eta_e = \eta_i \eta_M \quad (4-33) \]

Then

\[ \eta_e = 0.404 \]

f) **The main parameters of the engine**

When calculate the efficient fuel consumption rate \( g_e \), used the equation

\[ g_e = \frac{3600}{H_u \eta_e} \quad (4-34) \]

For calculate the engine displacement \( V_\pi \), used the equation

\[ V_\pi = \frac{\pi D^2 S_i}{4 \times 10^6} \quad (4-35) \]

That \( V_\pi = 2.8 \ (L) \)

Where \( D \) is the diameter of the cylinder, \( S \) is the stroke of the cylinder and \( i \) is the number of the cylinder

For effective power of engine \( N_e \), that

\[ N_e = \frac{p_e V_\pi n}{30 \tau} \quad (4-36) \]

For the effective torque of engine \( M_e \), that

\[ M_e = \left( 3 \times \frac{10^4}{\pi} \right) \times \frac{N_e}{n} \quad (4-37) \]
For hourly fuel consumption of engine $G_r$, that

$$G_r = N_e g_e \times 10^{-3} \quad (4-38)$$

After calculation, the result is shown as a table as below

<table>
<thead>
<tr>
<th>Table 4.2. The result of the main parameters of engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>$n$</td>
</tr>
<tr>
<td>$N_e \ (kW)$</td>
</tr>
<tr>
<td>$M_e \ (N.m)$</td>
</tr>
<tr>
<td>$G_r \ (\frac{kg}{h})$</td>
</tr>
<tr>
<td>$g_e \ (\frac{g}{kWh})$</td>
</tr>
</tbody>
</table>

### 4.4 The selected parameters of the diesel engine’s piston

#### 4.4.1 Working conditions and design requirements of the piston

1) Because of the bad condition the piston works in, it will be chosen the material with good hot strength, wear-resistant, small proportion, lower coefficient of thermal expansion, good thermal conductivity.

2) With the reasonable shape and wall thickness to make it with good heat dissipation, strength, required stiffness and reduce weight to avoid the stress concentration.

3) Ensure the air tightness of the combustion chamber and less gas channeling and oil channeling, meanwhile without increasing the friction loss of the piston group.

4) Let the piston and cylinder keep the best fit in different operating conditions.

5) Reduce the heat of piston from the gas absorb so that the absorbed heat can be smooth diffuse.

6) Ensure that the sliding surface has enough lubricant when lower oil consumption.

7) Improve the working conditions of the top of piston and the first ring to prevent the top cracking, the ring bonded and stuck and excessive wear.

8) Improve the actual carrying capacity of the piston pin and pin seat to reduce wear and prevent rupture.
9) Determine the right shape of skirt and the measure of the thermal expansion controlling to improve the carrying capacity and reduce the gap of cylinder, both to improve the wear and make the running smoothly.

### 4.4.2 Choices of the material of piston

According to the requirement of the piston design, the material of piston should meet the following requirements:

1) High-intensity heat. When the temperature is 300~400 degree, it has also enough mechanical properties to prevent the parts damaged.
2) Good thermal conductivity and poor heat absorptivity. Not only reduce the temperature of the top and ring, but also reduce the thermal stress.
3) The expansion coefficient is small. Keep the small gap with the piston and cylinder.
4) The specific gravity is small. Reduce the reciprocating inertia force of the piston group to reduce the mechanical load of the crankshaft and connecting rod.
5) Good wear properties.
6) Good manufacturability and cheap.

Based on the related researches, aluminum alloy is mostly used material in making car pistons, and experiments using other material such as cast iron, cast steel, ceramics and carbon. Here we consider a high-performance material, titanium alloy, which has very high tensile strength and toughness. It is light in weight, have extraordinary corrosion resistance and the ability to withstand extreme temperatures. Then analyze if it is feasible or not to expand the service life in the normal design way.

The main performance parameters of titanium alloy are shown as Table.4.3. The whole thickness of titanium alloy piston is large, but due to the small density of the titanium alloy, the mass is lighter than the cast iron piston, so it’s good for the high speed engine. At the same time, due to the high thermal conductivity of the titanium alloy, it make the water four stroke engines keep the good temperature when the engine injects the oil to cooling the temperature of inner wall. But the expansion coefficient of titanium is big, it is broken the gap of the cylinder and increase the friction loss.
Table 4.3. The main performance parameters of eutectic titanium alloy

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>4.62e-009 tonne mm^-3</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion</td>
<td>9.4e-006 C^-1</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>5.22e+008 mJ tonne^-1 C^-1</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>2.19e-002 W mm^-1 C^-1</td>
</tr>
<tr>
<td>Resistivity</td>
<td>2.19e-002 W mm^-1 C^-1</td>
</tr>
<tr>
<td>Young’s Modulus [MPa]</td>
<td>96000</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.36</td>
</tr>
<tr>
<td>Bulk Modulus [MPa]</td>
<td>1.1429e+005</td>
</tr>
<tr>
<td>Shear Modulus [MPa]</td>
<td>35294</td>
</tr>
</tbody>
</table>

4.4.3 Determinations of the dimension of piston

The main dimension of the structure is shown as Figure 4.1.

*Figure 4.1. The main measurement of the piston*
a) The height of the piston $H$

The high of the piston depends on the following factors:

- The requirements of the diesel engine height dimension.
- Speed $n$
- The shape and the dimension of the combustion chamber.
- Bearing area of the piston skirt.
- Try to choose the small height when it can keep the structural arrangements reasonable and under the given bearing area.

The ratio of the piston of the small and medium high-speed diesel engine and the cylinder is $H/D$, general the range is 1.0~1.3 and the recommended values is about 1.1. When speed is high, the $H$ will be small. To reduce the quality to control the inertia force increases by the speed increases.

As the factors above, $H/D$ is determined to be 1.04, so $H=94$.

b) The height of the compress $H_1$

The height of the compress $H_1$ is determined by the position of the piston pin. And $H_1$ is also determined by the distance $h$ from the first piston ring to the top, the height of the girdle $H_5$ ($H_5$ is determined by the number of the piston ring and height) and the height of the up skirt. Try to reduce $H_1$ to reduce the height of the engine when keeping the gas ring is good. For the small high-speed engine ($D < 105\text{mm}$), general range of $H_1/D$ is 0.5~0.7 [9].

So trying to find the available $H_1$ to meet the needs for the good engine working and less height of engine, that $H_1/D$ is 0.605 and $H_1=54.5$.

c) The distance $h$ from the first ring slot to the top of piston

- When $h$ is small, the thermal load of the first ring is high. So the $h$ is determined by the thermal load and the cooling condition that the temperature is not large than the permissible limit, general about 180~220.
- When keeping the working reliability, try to reduce the $h$ to reduce the height and weight of the piston.
- General $h/D$ range of the titanium piston is 0.14~0.20 [9].

In summary, $h/D=0.16$ and $h=15$. 
d) The number of the piston rings and arrangements

- The number of the piston rings [9]:
  High-speed engine: Two gaseous rings and one oil ring.
- The arrangement of the oil ring: General use one oil ring and assemble above the pin hole.

e) The measurement of the ring slot

The axial height of the ring slot is equal to the axial height of the piston ring b. The diameter of the ring slot D is determined by the gap of the back of piston ring, the size of the gap and the thermal expansion of the piston, and also impact on the back pressure of the piston ring. So that the D [9] can be estimated:

Gaseous ring:
\[
D = [D - (2t + KD) + 0.5]^{+0.25}_{-0.25} \text{ (mm)}
\]  
(3-39)

Oil ring:
\[
D = [D - (2t + KD) + 1.5]^{+0.125}_{-1.25} \text{ (mm)}
\]  
(3-40)

Where D is the diameter of the piston, t is the thickness of the piston ring, K is the coefficient of the titanium piston and K=0.006

Generally, the excessive fillet on the button of the ring slot is 0.2~0.5mm.

f) The height of the ring land

- Because of the temperature of the ring land is high, gas pressure bearing is maximum and easily to broken when impact by the piston ring, so the height of the first ring land is larger than others.
- The range of the height of the ring land shown as below [9].

<table>
<thead>
<tr>
<th>Type</th>
<th>The ratio of the height of ring land and the diameter of piston</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>First ring land h1/D</td>
</tr>
<tr>
<td>Titanium piston</td>
<td>0.04~0.08</td>
</tr>
<tr>
<td>High-speed engine</td>
<td>0.04~0.08</td>
</tr>
</tbody>
</table>
Determined that
h1/D1 = 0.08, h2/d = 0.05, h3/D = 0.45
h1 = 7.4, h2 = 5, h3 = 4.

g) The thickness of the top of piston  δ
δ is determined by the stress of the top of piston, stiffness and the cooling requirements. For the titanium piston of the small high-speed diesel engine, if it has enough cross-section to transfer the temperature on the top of the piston, that the mechanical strength is generally sufficient. Thermal stress increases with the thickness of piston top, so that choosing the thickness which can bear the gas pressure. Generally, when using the titanium piston that \( \frac{\delta}{D} \) is about 0.07~0.15 [9]. The thin thickness of top can reduce the thermal stress.

h) Skirt length H2
1. Select H2 should make the pressure of skirt portion than within the permitted range, the pressure of skirt portion according to the

\[
q_1 = \frac{N_{max}}{DH_2}
\]  

(4-41)

2. The general ranges of H2 / D as follows [9].
High-speed diesel 0.60 to 0.88
3. Upper and lower skirt length should be appropriate to the proportion, upper skirt length the H4 is too small, easy to produce peak load, caused the piston surface galling and abrasions. General the following proportions

\[
H3 = (0.6 \sim 0.75) \times H2
\]  

(4-42)

The results are

\[
\frac{H2}{D} = 0.6
\]

\[
H2 = 54
\]
\[ \frac{H3}{H2} = 0.73 \]

\[ H3 = 39.5 \]

I) Skirt wall thickness $\delta_g$

The minimum thickness of the titanium piston skirt wall is generally (0.03 ~ 0.06) D. Thin-walled skirt portion can reduce the weight of the piston advantageously, but it is also ensure that the skirt portion has sufficient rigidity, may be provided to strengthen the ribs. Stiffeners can be set to strengthen tendons. That the results are chosen as

\[ \frac{\delta_g}{D} = 0.04 \]

\[ \delta_g = 3.7 \]

J) Piston pin diameter $d$ and Pin seat interval $B$

For high-speed machine ($D < 100\text{mm}$), the generally ranges of $d/D$ is from 0.28 to 0.38 [9]. For middle and small size of high-speed diesel engines, the range is $d/D < 0.4$. If $d / D$ too large, the distance between pin surface and piston top surface is too small, this will bring difficulty to the piston rod group design. Final value is

\[ \frac{d}{D} = 0.28 \]

\[ d = 29 \]

\[ \frac{B}{d} = 1.56 \]

\[ B = 35 \]
4.5 The design of the piston head

The shape of the top of the piston is mainly based on the design requirements of the combustion system. The thermal load is one of the important evidence for choosing the combustion system. The sectional shape of the head affects the heat flow and temperature distribution in the piston.

Titanium piston head designed as a good thermal conductivity of the "heat flux", i.e., according to the heat flow passage of the piston, the large arc transition, in order to increase the heat transfer section of the skirt portion from the top to the head, thus heat rapidly spread, so that the temperature of the piston head can be reduced. Temperature decreases, while also helping to eliminate the stress concentration, so that the carrying capacity of the piston can be increased.

Piston head bear a greater load, and fatigue cracks often happened at ‘Valve pits’, ‘the combustion chamber Throat edge’ and ‘Roots join at the top of the piston wall and the pin seat’. So the measurements to solve this problem are

1. Design the shape of the head reasonably to reduce the mechanical stress of the top surface of the piston.
2. Avoid processing the cusp, use large fillet, elimination of stress concentration;
3. Reduce the heat load of the piston, and increase the titanium alloy fatigue limit;

The position of the first piston ring is one of the important factors determining the structure of the head of the piston. In order to reduce the height and weight of the piston, the first ring can be higher and closer to the top of the piston. But it would be increasing the temperature of the first ring. The engine allowed thermal load depends largely on the first track of the temperature of the ring, and proved, the first ring wear is the largest and ring groove is most likely to be broken and killed. Therefore, the engine overhaul interval of the piston group depends largely on the life of the first ring. This shows that, as much as possible to improve the reliability and life of first ring is of great significance. First ring on the piston position should be like this: I.e., when the piston is in the upper dead point, the first ring outer surface should not go beyond the outside of the cooling water jacket, In the occasion of the cylinder liner, cylinder liner sudden shoulder will affect the first ring up the degree of improvement. The distance between the top of the piston and the first ring groove can be given as:

\[ h = (0.14 \sim 0.2)D \]  (4-43)

In order to reduce the temperature of the first rings, taking the following measures: At the top of the piston for hard anodic oxidation treatment, Can improve the heat resistance and hardness of the top surface of the piston, and to increase the thermal resistance, so that the top cooling. Improve the processing quality and the correct choice of the ring groove of the piston ring groove side clearance reliability and durability is very important for the ring groove and the ring. Because piston rings reliable work to be snapped to the outer surface of the ring and the cylinder wall, ring up and down both sides of the ring groove corresponding plane snapping the premise. The gap of ring and the ring groove is too big which will exacerbate the impact of the ring on the ring groove. The gap is too small, easy to make the ring adhesive ring groove and lose sealing effect. In the high-speed engine, it is often to set the first ring backlash increases to 0.1 \sim 0.2mm.

Piston ring groove wear has a great impact on the use of performance, In order to extend the service life of the piston, it is necessary to pay special attention to improve the wear resistance of the first ring groove. In addition to pay attention to the quality of the ring groove machining, the correct choice of the backlash of the piston ring and the ring groove, to reduce the impact of the ring groove wear, from the structure to pay attention to improve the first ring wear resistance and its
measures are as follows:

1. The Ringed seat (that is wearable ring), Cast the austenite ring seat into the first ring groove, the ring seat and the piston material rely inter diffuse to form a metal molecular binding, the intermediate layer is a variety of compounds, can be larger to improve the life of the annular groove.

2. Setting the cross-sectional shape of the ring seat to become trapezoidal, makes titanium alloy cooling along the radial contraction, clamping the ring seat.

When determine allocation compression height and ministries dimensions, first, fix the position of the first ring. According to the formula of the first ring shore strength check. The number of gas ring is determined according to the gas pressure, engine speed and engine mode. Leak and increased with the increase of the gas pressure and the diameter of the cylinder, with the engine speed increase and decrease, from the theory, when the gas ring and the piston and the cylinder wall is close fit, a gas ring enough. Recently there has been a high-speed engine with only one gas ring and one oil ring abroad. This is because in a high speed and high load environment, to reduce the number of rings and ring height is very significance. The main advantage is to reduce the number of rings: to reduce friction and wear, to reduce the reciprocating load, to increase the reliability of the engine. Taking into account the increase in gas pressure units, startup tightness and improve the thermal conditions by the piston ring to the cylinder wall, more engines use 3-ring, this design in a 3-ring, two gas rings and an oil ring.

![Figure 4.3. The ring of piston](image)

In addition to the number of rings, to reduce the height of the ring with a part by
proceeding to reduce the height of the ring groove and ring shore. Start from reducing the friction of the piston and the cylinder liner power, the smaller ring height is also conducive to the reduce weight, to shorten the running time, while the degree of adaptability of the cylinder no parallelism is also good. But this will increase the heat intensity through the ring, and is easily broken in the machining and assembly. The height of the ring groove depends on the height of the ring, shore determine the height of the ring, so the pressure acting on the ring will not cause deformation of the ring shore. Taking into account the high temperature than the other ring shore, first ring shore by the shock pressure is also large, cracks easily generated at the root of the ring shore, so the first ring shore of titanium piston is thick, generally take

\[ h_1 = (0.04\sim0.08)D \]  \hspace{1cm} (4-44)

The thickness of the remaining ring shore takes

\[ h_n = (0.03\sim0.045)D \]  \hspace{1cm} (4-45)

### 4.6 The design of the skirt portion of the piston

The role of the skirt portion of the piston is a piston within the cylinder for reciprocating movement guide and to withstand the side pressure. The long skirt is beneficial to reduce the pressure per unit area and reduce wear and tear, also not easy to cause the strain damage of piston and cylinder liner. However, from the perspective of reducing piston height, it hopes that the skirt portion is kept as short as possible. Short of the skirt portion does not easily collide with the link. Vehicle engine piston skirt length is generally taken to be:

\[ H_2 = (0.4\sim0.8)D \]  \hspace{1cm} (4-46)

When consider the skirt length, it must take care of the position of the piston pin hole relative to the piston skirt. Reasonable allocation of the length of the upper skirt portion and a lower skirt portion, prevent piston work highly skewed, causing localized strongly wear.

The piston pin hole should be in the correct position of the skirt portion that makes the side pressure load generated wholly uniform distribution along the piston. When
the force on the side of the cylinder wall is \( N \), and when the downward movement of the piston in the power stroke, piston encounter frictional resistance \( \mu N \) (\( \mu \) is the friction coefficient) the formation of torque \( \mu N \times D / 2 \), will enable the tilting of the piston in a clockwise direction. If the side pressure on the cylinder walls are uniformly distributed, the reaction force of the cylinder wall of the piston will through the midpoint of the piston, Therefore, the center line of the piston pin should be arranged above the midpoint of the skirt portion, in order to form a torque opposite to the direction of the moment \( \mu N \times D / 2 \) so that the piston without tilting clockwise direction. According to the moment equilibrium conditions, the distance \( Y \) between the centerline of piston pin and the midpoint of the skirt portion can be obtained.

\[
Y \times N = \mu N \times D / 2 \quad \text{(4-47)}
\]

If using the same method to analysis the compression stroke, piston pin centerline should below the midpoint of the skirt, but in this case the side pressure will not be considered, because the side pressure in the power stroke is much smaller. So generally take:

\[
H3 = (0.6 \sim 0.7)H2 \quad \text{(4-48)}
\]

When the piston working, the combustion gas pressure evenly distributed on top of the piston, and the piston pin to give the reaction force acting on the piston head at the pin seat, the resulting deformation of the diameter of the skirt portion is increased along the axial direction of the piston pin. The role of the side pressure makes the skirt portion of the piston increase in the same direction. In addition, piston pin seat near area has the metal accumulation, heated expansion becomes larger, and it will result in the increment of diameter along the axial direction of the piston pin is larger than any other direction when thermal deformation happened. Therefore, when the working piston generate the mechanical deformation and thermal deformation, its skirt portion sectional to becomes long axis of the ellipse in the direction of the piston pin.

In view of the above circumstances, in order to make a uniform gap between piston and cylinder wall when in normal operation so as not to be stuck in the cylinder or cause localized wear, piston must be processed into the cross-section of the skirt portion for the oval long axis perpendicular to the direction of the piston pin in the cold state. In order to reduce the amount of thermal deformation of the vicinity of the pin seat, some piston pin and the outer surface is formed in the skirt portion of
the seat near subsidence of 0.5 ~ 1.0 mm. Since the temperature distribution and the mass distribution of the piston in the axial direction are uneven. Therefore the amount of thermal expansion of the individual cross-section is large small. This feature of titanium alloy pistons is particularly obvious. In order to make the work state of titanium alloy pistons close to a cylindrical, it is necessary to made prior to the piston diameter small great approximate conical.

In the present, the parabolic shaped skirt portion or convex skirt portion of the piston to obtain a very wide range of applications. The reasons why skirt portion is made convex are as follows:

When the engine working, the piston of each cross-section of radial deformation can be regarded as composed of two parts. Part is a section of free expansion, the diameter increment is:

\[
\Delta D = \alpha D(t - 20{\degree}C) \tag{4-49}
\]

Wherein: \(\alpha\) is piston material linear expansion coefficient, \(D\) is cylinder diameter, 20\(^\circ\)C is calculation of the cross section of the working temperature. The other part is the deformation due to thermal stress. This part is relatively small, so the actual calculation of the amount of thermal expansion of the diameter of the piston can be ignored when the elastic deformation. The actual wear scar of the skirt portion shows that, only in the upper and lower edges with polished signs, sometimes there are abrasions. So under working conditions in order to obtain line-shaped piston skirt, the skirt portion should be made in the convex type, i.e. at a length \(X\), the diameter of the piston skirt having a straight line to form a line in the cold state increase in 2YT value. Even better, in the cold state, to make the skirt portion form a certain shape, this will keep the convex shape of skirt when it is heated. The main advantages of the convex skirt piston are that the surface in contact with the cylinder liner is about double than before and the thermal stress was reduced by approximately 20%. And also decrease the engine noise, oil consumption. When the piston inside the cylinder and lateral displacement occurs due to the change of direction of the side pressure, for general linear shape of piston skirt portion, the contact is inevitable to take place on the upper and lower edge of the cylinder. At this time, most of the skirt surface without lateral pressure, so that the pressure ratio of the contact part is larger. At the same time, the edge of the piston is not easy to maintain lubrication, likely to cause abrasions, and even cylinder scoring. The surface of the skirt portion which without lateral pressure, because the interval with the cylinder wall with a layer of oil or
emulsion, so this lead to the reduction of the outgoing heat. However, the convex skirt portion of the piston does not have this drawback. It is conducive to reduce the wear of the skirt portion.

4.7 The design of the piston pin boss

The piston pin stress distribution depends on the pin seat deformation of the piston pin both adapt to each other. If the piston pin is relatively large stiffness, however, piston pin seat stiffness is smaller, both deformation cannot adapt to each other. The results lead to the edge of the upper side of the inner bore of the pin boss, etc. produce severe stress concentration, resulting in the pin boss cracked. Therefore, the design of the piston pin boss and piston pin should be considered unified. This demands that the piston pin has a higher stiffness, to reduce the bending deformation of the piston pin. Piston pin boss should withstand high pressure, but also has a certain degree of flexibility to adapt to the deformation of the piston pin. Generally pin seat outer diameter taken

\[ d = (0.32\sim0.42)D \]  \hspace{1cm} (4-50)

Internal diameter

\[ d_0 = (0.25\sim0.60)d \]  \hspace{1cm} (4-51)

The design of the piston pin seat using a trapezoid structure, it has the advantage in that:

1. Increase in the length of the pin boss and the connecting rod small end bearing surface, thereby reducing the liner than the pressure of the pin holder and the small head.
2. The supporting surface has the overlap along the axial length, thus reducing the bending deformation of the piston pin.
3. Interval of the pin seat portion is reduced, thereby also reducing the stress of fillet at roots of the pin boss and the top and bottom.

Comparison with the direct selling seat, tilted 10 ° bevel pin seat, have declined in the stress of the dangerous point, such as the fillet at the top of the piston pin seat,
maximum 43% reduction. Cooling oil channel edge stress is reduced by 25% to 29%. The top of the pin seat stress is reduced to 16%. This indicates that the beveled pin seat piston can withstand more than 15 ~ 20% of the load, which is equivalent to increase the mean effective pressure 3kgf/cm^2.

4.8 The design of the gap between piston and cylinder

The gap of the piston and the cylinder wall affect the oil consumption, noise, gas leak quantity, wear and piston cooling. Gap should be selected so that the piston and the cylinder wall has the smallest gap in the hot state, the gap is consistent throughout the piston height, in order to increase the piston life. The cylinder diameter and piston material should also be considered when determining the gap, so that neither the gap is too large to percussion, nor gap is too small to stuck piston. Due to the requirements of the piston side surface shape and elliptical, along the height and the circumferential direction of the piston gap has different values, Which is the top of the piston clearance $\Delta_o$, and vertical direction of the pin hole, the skirt portion of the gap $\Delta_{\perp}$. Reduced $\Delta_o$ can reduce the thermal load of the piston head, reducing the $\Delta_{\perp}$ may be weakened heeling swing piston commutation with percussion Liner phenomenon, which can greatly reduce cavitation of the liner, but if the piston gap is too small, will also easily cause the piston damage and cylinder scoring. Skirt portion does not withstand the thrust load in the direction of the axis of the pin hole, the effect of clearance $\Delta_{\parallel}$ can be ignored, so the choice range of $\Delta_{\parallel}$ in the design is larger. The values of piston gap is related to such factors as the degree of enhancement of the engine, piston cooling method, materials, heat treatment specification and piston shape. When select value, full load state must be taken into account to avoid cylinder scoring, preliminary selected with reference to the relevant empirical data or press a rough calculation:

$$\Delta = \Delta_{min} + D(a_2\Delta t_2 - a_1\Delta t_1)$$  \hspace{1cm} (4-52)

Where $\Delta$ is piston assembly gap, $\Delta_{min}$ minimum operating clearance, $\Delta_{min}$ factors to take into consideration the amount of shrinkage of the cylinder due to the uneven temperature generated for each group in the piston and cylinder optional packet clearance tolerance, as well as the film thickness or insurance gap. The $a_1$ and $a_2$, respectively, are the coefficient of linear expansion of the piston and the
cylinder liner, titanium piston is

\[(18\sim21) \times 10^{-6} \text{mm/mm} \cdot \text{deg}\]

\(\Delta t_1, \Delta t_2\) respectively, are the amount of change of the temperature of the piston and the cylinder liner, \(\Delta t_1, \Delta t_2\) preferably based on the test data, the temperature at the middle part of the water-cooled four-stroke high-speed diesel engine cylinder liner is about 110 °C.

### 4.9 Final dimensions of piston

The final dimensions of the piston are shown in Table 4.5.

<table>
<thead>
<tr>
<th>Table 4.5. The final dimension of the piston</th>
</tr>
</thead>
<tbody>
<tr>
<td>piston diameter D</td>
</tr>
<tr>
<td>Compressed height H1</td>
</tr>
<tr>
<td>Piston height H</td>
</tr>
<tr>
<td>Skirt height H2</td>
</tr>
<tr>
<td>The top land height h</td>
</tr>
<tr>
<td>Upper skirt height H4</td>
</tr>
<tr>
<td>Down skirt height H3</td>
</tr>
<tr>
<td>pin hole diameter d</td>
</tr>
<tr>
<td>First ring shore height h1</td>
</tr>
</tbody>
</table>

### 4.10 Piston strength check

1. Piston top

   \[\sigma_u \leq [\sigma_u] = 50 \text{ Mpa}\]

   Mechanical stress

   \[\sigma_u = 0.68p_z \left(\frac{D_1}{2\delta}\right)^2 = 30.47 \text{ Mpa}\]

2. The first ring shore
\[ [\sigma] = 29.4\text{~}39.2 \text{ Mpa} \]

**Bending stress**

\[ \sigma_w = 4.5p_z \left( \frac{D}{h_1} \right)^2 \times 10^{-3} = 4.992 \text{ Mpa} \]

**Shear stress**

\[ \tau = 3.14p_z \left( \frac{D}{h_1} \right) \times 10^{-2} = 2.86 \text{ Mpa} \]

**Total stress**

\[ \sigma = \sqrt{\sigma_w^2 + 3 \tau^2} = 7.033 \text{ Mpa} \]

**Specific pressure**

\[ [q_1] = 0.5\text{~}0.9 \text{ Mpa} \]

\[ q_1 = \frac{N_{\text{max}}}{DH_2} = 0.91 \text{ Mpa} \]

According to the theory, all kinds of stress are within safety limits. The Geometric dimensions now can be created in CAD Inventor.
5. Geometry modeling

Analyzed herein, the piston combustion is located in departing from the piston axis with dimple-shaped. Section of the skirt portion is ellipse which the long axis is perpendicular to the direction of the piston pin. Due to the complexity of the internal shape of the piston and the deformation is asymmetric, the complete three-dimensional model of piston should be created.

Use Autodesk Inventor CAD software to establish the piston parts model. Then import the piston model to the finite element software. Some small details of model is ignored, in order to reduce the number of units and the differential of the cell size, such as piston oil hole and oil guide slot.

The geometry of piston was created using Autodesk Inventor. The part and is shown in figure 5.1.

![Figure 5.1.Model of piston](image)
6. Finite Element Analysis

After a while of working in bad conditions, pistons show some cracks and wears leading to increase of emission pollution and reduction of engine efficiency. This issue can be problematic in some areas which care about quality and the lifetime of their production.

It is hard to expand the service life of piston in the normal design way. One way is to replace the piston material. Here we will replace aluminum alloy piston with titanium alloy piston and analyze if it is feasible.

Our main purpose is to make a comparison of titanium with aluminum as manufacturing material for internal combustion engine piston. Finite Element Analysis approach showed that titanium has a more desirable working temperature and subject to less stresses.

6.1 Software introduction

ANSYS is developed by the United States ANSYS, Inc. And ANSYS is a powerful finite analysis software, integrate financial structure, thermal fluid, electromagnetic, acoustic analysis in one. With a friendly interface, efficient and accurate solver and perfect post-processing function, it has been widely used in industrial production and scientific research. ANSYS software is effectively combined with the techniques of finite element numerical analysis and CAD and CAE, it can make the user get the problems intuitively and accurately, saving the development costs. Meanwhile ANSYS is the first finite analysis software to get the ISO9000 authenticate. It has the following 3 characteristics:

1) Powerful and widely used: it can be used in the multi-physical field and multi-field analysis of linear and non-linear problem, like structure, thermal, fluid, electromagnetic, acoustic and so on.

2) Integrated processing technology: including geometric modeling, automatic meshing, solving and post-processing, optimization of design and other
functions and tools.

3) Extensive product and open system: Different products can be used in various industrial fields, such as aviation, aerospace, shipbuilding, cars, weapons, railway, electronic, mechanical, nuclear industry, energy, construction, medical and so on.

The advantages of ANSYS software is reflected in the following points:
1) Seamlessly integrated with CAD to meet the needs of the requirements of engineers to solve complex engineering problems quickly.

2) Powerful grid processing capabilities: Complex models require very accurate hexahedral mesh to get effective results. In many solving process of engineering problem, an area of the model will produce a great strain, if not do the re-division of the grid which will lead to solving suspend and get the incorrect results. With its precise handling capacity and unit mesh, it makes ANSYS has a lot of advantages, so it’s more and more welcomed by users.

3) Precision non-linear problem solving: With the development of science and technology, linear theory can’t meet the requirements of the design. A lot of engineer problem, as material damage and failure and crack propagation, can’t be solved by only using the linear theory. And for analysis the materials, such as plastic, rubber, ceramics, concrete and rock, it must be considered with the material nonlinearity. As we all know, solving the non-linear problem is very complex, it involves not only the mathematical problems, also acquire amount of theoretical knowledge and solving skills, it’s difficult to learn it. So the ANSYS companies spend a lot of manpower and material resources to develop the solver applied to nonlinear solution to meet the needs of users want to get the high-precision nonlinear analysis.

4) Strong coupling field solver capability: The finite element method was first used in the aerospace filed, mainly used to solve the linear structural problems. Now the software and the finite element method to solve the linear structural is relatively mature, the direction of development is the solving of the problem of the non-linear structural, fluid dynamics and coupling field. For examples, the heat generated due to frictional contact and plastic work when metal forming, the solution needs the result of structure filed and temperature field to be cross iterative, that is ‘thermal coupling’. Due to using the finite element more and
more deeply and more and more complex of people’s attention, the coupling filed solving is urgent needed by the users. ANSYS software is the only one can do the coupled field analysis.

The process of simulation diagram is shown as figure 6.1.

![Figure 6.1. The process of simulation](image)

6.2 Materials

The titanium alloy is chosen to be the piston’s material. The material properties
such as constants, thermal conductivity and elasticity are shown from table 6.1 to table 6.2.

**Table. 6.1. Constants of Titanium Alloy**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>4.62e-009 tonne mm^-3</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion</td>
<td>9.4e-006 C^-1</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>5.22e+008 mJ tonne^1 C^-1</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>2.19e-002 W mm^-1 C^-1</td>
</tr>
<tr>
<td>Resistivity</td>
<td>1.7e-003 ohm mm</td>
</tr>
</tbody>
</table>

**Table. 6.2. Elasticity of Titanium alloy**

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Young's Modulus MPa</th>
<th>Poisson's Ratio</th>
<th>Bulk Modulus MPa</th>
<th>Shear Modulus MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>96000</td>
<td>0.36</td>
<td>1.1429e+005</td>
<td>35294</td>
</tr>
</tbody>
</table>

6.3 The Thermal-Mechanical Coupling analysis

6.3.1 Meshing

Since of the symmetry, the half part of the piston model is sufficient to do the analysis. The piston shape is irregular, especially in the presence of various curved surfaces of inner cavity. Firstly, Automatic meshing method is used to mesh the model. The mesh grid is shown as figure 6.2. The model has a total of 37471 nodes and 23128 elements.
6.3.2 Static forces boundary conditions

Since this is a static analysis of the piston, to consider a number of reasons, ignored the reciprocating inertia force, only consider the gas pressure on the piston side and thrust force. For diesel engines, due to the throttling effect of the gap between the piston head and the cylinder, in the first ring shore around the gas pressure was \(0.9P_g\), around the second ring shore gas pressure was \(0.2P_g\), the surrounding gas pressure of the other ring shore can be negligible.

Gas force

\[ P_g = (P_g - P_0) \cdot \pi D^2 / 4 \]  \hspace{1cm} (6-1)

Wherein, \(P_g\) is the absolute pressure of the cylinder gas, \(P_0\) is the absolute pressure of the crankcase gases. \(P_g\) is positive value when the force direction is the downlink direction of the piston, the negative value indicates the direction of its
force is the direction of the piston upward. During mechanical stress finite element analysis to take $P_g = 7.5 \text{ Mpa}$, $P_0 = 0.1 \text{ Mpa}$. The gas force calculated from the top of the piston $P_g = 77\,052 \text{ N}$. Therefore, the surrounding gas pressure in the first ring shore as 6.75MPa, the gas pressure of the second ring shore around is 1.5MPa.

Side thrust

$$N = (P_g - P_j)\tan \beta$$  \hspace{1cm} (6-2)

Where, $P_j$ is crank linkage reciprocating inertia force, $\beta$ is the swing angle of connecting rod, take $\beta = 2.4^\circ$. Here ignores reciprocating inertia force, so $P_j = 0 \text{ N}$. Obtained by calculation of the skirt portion of the side thrust, $N = 3229 \text{ N}$. Table 6.3 is the location and size of each load in the finite element model loaded [1].

<table>
<thead>
<tr>
<th>Load types</th>
<th>Role location</th>
<th>Effect size [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>The top of the piston</td>
<td>7.4</td>
<td></td>
</tr>
<tr>
<td>Gas force</td>
<td>The first ring of the shore around</td>
<td>6.75</td>
</tr>
<tr>
<td>Side thrust</td>
<td>The skirt portion</td>
<td>0.26</td>
</tr>
<tr>
<td>The second ring of the shore around</td>
<td>1.5</td>
<td></td>
</tr>
</tbody>
</table>

6.3.3 Thermal boundary conditions

When the internal combustion engine works, the value of the coefficient of radiation heat transfer is much smaller than the value of the convection of heat transfer, so ignored the thermal radiation. When calculate the temperature field of piston, determined the boundary conditions of the piston reasonably, this can be based on the value of the temperature of the piston’s surface point and use it to identify and revise. The measure method as hardness plug method, fusible alloy method, hardness recovery method and temperature measurement with thermocouple, and only the temperature measurement with thermocouple can measure the temperature when piston changes, so this method is reliable and high accuracy. The third kind boundary condition is used to analyze the temperature distribution. Whatever the theoretical analysis and experimental studies, it is not possible to get the actual and
experimental operation of piston by the first and second kind of boundary conditions. According to the reference, the average value of the heat transfer coefficient between the gas and the top surface is 260~465w/m$^2$k. In fact, the heat transfer coefficients will be changed along the radius direction of the piston, the changes is due to the radius and structure of the piston. Since most of the heat came out through the piston ring, the coefficient of the heat transfer on both sides of the ring grooves is significantly greater than the surface of the skirt. The coefficient of the heat transfer out of the surface of the skirt portion is 115~465w/m$^2$k. The temperature of the piston top can be determined by average gas temperature of the working cycle, the temperature of inside piston can be determined by the average gas temperature of crankcase and the temperature of outer circumferential surface can be determined by the average temperature of the cooling water and cooling air. The piston material of 490 is considered to use titanium, modulus of elasticity is 96000Mpa, Poisson ratio is 0.36, the coefficient of linear expansion is 9.4e-006 C$^{-1}$, the thermal conductivity is 2.19e-002 W mm$^{-1}$ C$^{-1}$ [10].

<table>
<thead>
<tr>
<th>Table 6.4.Piston convection boundary conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston</td>
</tr>
<tr>
<td>Top</td>
</tr>
<tr>
<td>Side of combustor</td>
</tr>
<tr>
<td>Combustor</td>
</tr>
<tr>
<td>Fire shore</td>
</tr>
<tr>
<td>Upper surface of first ring</td>
</tr>
<tr>
<td>Side surface of first ring</td>
</tr>
<tr>
<td>Lower surface of first ring</td>
</tr>
<tr>
<td>Between the first and second ring</td>
</tr>
<tr>
<td>Upper surface of second ring</td>
</tr>
<tr>
<td>Side surface of second ring</td>
</tr>
<tr>
<td>Lower surface of second ring</td>
</tr>
<tr>
<td>Between the second and third ring</td>
</tr>
<tr>
<td>Upper surface of third ring</td>
</tr>
<tr>
<td>Side surface of third ring</td>
</tr>
<tr>
<td>Lower surface of third ring</td>
</tr>
</tbody>
</table>
6.3.4 Results of temperature distribution

Figure 6.3 and figure 6.3 show the temperature distribution of piston.

![Temperature Distribution of Piston](image)

*Figure 6.3. The temperature field of piston in the thermal load acts (Internal)*
The temperature distribution of the piston is uneven, with the maximum value of 445.5 °C and the minimum value of 101.5 °C. There is a big range of temperature distribution on the top surface of the piston area. The temperature is higher at the combustion chamber side of the deviation from the center of the piston. Highest temperature appears in the throat of the exhaust port of the combustion chamber adjacent side, the temperature reached 445.5 °C. The lowest temperature of the top surface of the piston edge is on the inlet side with 292 °C. The temperature was gradually reduced with the increase of the piston radius. The D-value between maximum temperature and minimum temperature on the top surface is 38 °C. The maximum temperature of the first piston ring groove zone appears in the exhaust side of the annular groove end surface reaches 300 °C. The maximum temperature on skirt portion is 177 °C, while a minimum of 101.5 °C, and the temperature difference is 70 °C. The maximum temperature on the inner cavity of piston is 254.5 °C which at the back of the combustion chamber.
6.3.5 Results of deformation distribution

Figure 6.5. Temperature distribution in the thermal and force coupling

Figure 6.6. Temperature distribution in the thermal and force coupling

The figure 6.14 and figure 6.15 are the deformation contours of the piston under the
mechanical and thermal loads. From the figure, it’s obvious to see that the edge of the top of the piston and fire shore have the biggest deformation. The value is between 0.16~0.18mm. For overall analysis of the piston, from top to bottom of the piston cylindrical, deformation decreases gradually and then gradually increase.

6.3.6 Results of stress distribution

![Stress distribution in the thermal and force coupling (External)](image)

*Figure 6.7. Stress distribution in the thermal and force coupling (External)*
Figure 6.8 Stress distribution in the thermal and force coupling (Internal)

Figure 6.7 and figure 6.8 show the stress map when the piston under the joint action of the temperature and the side thrust. The largest stress produces on the top of the piston pin boss, the value is 465Mpa. The result shows that the temperature contributed a great deal on the piston stress. The values of the most sections of the piston are under 100Mpa.
7. Comparison

The aluminum alloy of 490 piston was chosen to compare with titanium alloy 490 piston, the results of stress distribution, temperature distribution and coupling stress distribution of these two pistons are shown as table 7.1.

**Table 7.1. Results comparison**

<table>
<thead>
<tr>
<th>Type</th>
<th>Aluminum alloy</th>
<th>Titanium alloy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress</td>
<td>The maximum stress occurs on the up edge of the piston pin, the value is 860 Mpa.</td>
<td>The maximum stress occurs on the up edge of the piston pin, the value is 465 Mpa.</td>
</tr>
<tr>
<td>Temperature</td>
<td>The maximum temperature occurs on the junction of the top surface of the piston and combustion chamber, the value is 267 °C.</td>
<td>The maximum temperature occurs on the junction of the top surface of the piston and combustion chamber, the value is 445 °C.</td>
</tr>
</tbody>
</table>
The maximum stress occurs on the skirt of the piston, the value is 0.33mm.

The maximum stress occurs on the pin boss of the piston, the value is 0.18mm.

The table 7.1 shows that the titanium alloy piston has better performance in stress field and deformation field. And the temperature of titanium alloy is more than aluminum alloy in working area. Considering that the melting point of aluminum is 500 °C and for titanium is 1700 °C, regarding to its melting point, we improved it by 25%. A conclusion can be drawn that titanium has better thermal property than aluminum.
8. Discussion and Conclusions

The result showed that titanium alloy piston has a better performance in stress and deformation in comparison with aluminum alloy. Considering that the melting point of aluminum is 500 °C and for titanium is 1700 °C, regarding to its melting point, we improved it by 25%. A conclusion can be drawn that titanium has better thermal property than aluminum.

Besides it can be seen that titanium can help us to improve piston qualities. Although titanium is expensive and maybe it is uneconomical for large-scale applications, it can be used in some special cases.

In this work, the MTBM has been improved by the quality increasing of piston.

Combined CAD and ANSYS, get the results of stress and deformation and temperature when the piston under the mechanical loads, thermal loads and assembly the mechanical and thermal load. And get the discussion as below:

1) The temperature is higher at the combustion chamber side of the deviation from the center of the piston. Highest temperature appears in the throat of the exhaust port of the combustion chamber adjacent side, the temperature reached 445 °C. The temperature of the piston ring area is extremely important for the reliability of the engine, if the temperature of the ring zone is too high, it will make the lubrication oil to be deterioration even carbonization. It causes the piston ring bonded, loss of activity to make the piston rapid wear, deformation.

2) The stress under the mechanical action, the maximum stress value of the piston is 465Mpa, and the most stress of other parts below 100Mpa. For the tensile strength of the piston, it’s having a enough strength margin.

3) When under the assembly of mechanical and thermal loads, the value of the largest displacement is 0.16~0.18mm, causing at the edge of the piston top and the fire shore. The stress of the top of the piston is mainly caused by the temperature load and the deformation of the piston is caused by the thermal expansion.
9. Future works

Because of the time and ability, the work in thesis has some shortages.

1) When modeling the piston, ignore the first ring lined with iron ring, it will impact the stress on the first ring.

2) It’s not ideal when all DOF the piston pin, it will produce more stress and impact the result of analysis.

3) Analyze the temperature field with the calculation of empirical formula instead of the experiment measurement; it may be some influences on the result.

4) When mesh the grid, using the default to get the analysis result is not accuracy on some parts of piston.

For the further work, the constraint condition of the piston pin can be improved much more better, hoped that it can be closer to the real situation of piston. And in the conditions allow, the experimental approach can be used to determine the convection coefficient of the piston to get the realistic temperature field.
10. Reference

11. Appendices

1) Material Properties

*Table.11.1. Constants of Titanium Alloy*

<table>
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</tr>
<tr>
<td>Resistivity</td>
<td>1.7e-003 ohm mm</td>
</tr>
</tbody>
</table>

*Table.11.2. Elasticity of Titanium Alloy*

<table>
<thead>
<tr>
<th>Temperature C</th>
<th>Young's Modulus MPa</th>
<th>Poisson's Ratio</th>
<th>Bulk Modulus MPa</th>
<th>Shear Modulus MPa</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>96000</td>
<td>0.36</td>
<td>1.1429e+005</td>
<td>35294</td>
</tr>
</tbody>
</table>

2) The piston under the thermal loads

*Table.11.3. Temperature Result*

<table>
<thead>
<tr>
<th>Object Name</th>
<th>Temperature</th>
<th>Total Heat Flux</th>
</tr>
</thead>
<tbody>
<tr>
<td>State</td>
<td>Solved</td>
<td></td>
</tr>
</tbody>
</table>

**Scope**
- Scoping Method: Geometry Selection
- Geometry: All Bodies

**Definition**
- Type: Temperature Total Heat Flux
- By: Time
- Display Time: Last
- Calculate Time History: Yes
- Identifier: Results

**Results**
- Minimum: 131.56 °C 2.0153e-003 W/mm²
- Maximum: 266.73 °C 0.8477 W/mm²

**Information**
- Time: 1. s
- Load Step: 1
- Substep: 1
3) The piston under mechanical loads

*Table.11.4.Results*

<table>
<thead>
<tr>
<th>Object Name</th>
<th>Total Deformation</th>
<th>Equivalent Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>State</td>
<td>Solved</td>
<td></td>
</tr>
</tbody>
</table>

**Scope**

<table>
<thead>
<tr>
<th>Scoping Method</th>
<th>Geometry Selection</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geometry</td>
<td>All Bodies</td>
</tr>
</tbody>
</table>

**Definition**

<table>
<thead>
<tr>
<th>Type</th>
<th>Total Deformation</th>
<th>Equivalent (von-Mises) Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>By</td>
<td>Time</td>
<td></td>
</tr>
<tr>
<td>Display Time</td>
<td>Last</td>
<td></td>
</tr>
<tr>
<td>Calculate Time History</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Identifier</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Results**

| Minimum  | 0. mm  | 2.7426e-002 MPa |
| Maximum  | 1.0802e-002 mm | 134.57 MPa |

**Information**

<table>
<thead>
<tr>
<th>Time</th>
<th>1. s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Step</td>
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</tr>
<tr>
<td>Substep</td>
<td>1</td>
</tr>
<tr>
<td>Iteration Number</td>
<td>1</td>
</tr>
</tbody>
</table>

4) The piston under assembly thermal and mechanical loads

*Table.11.5.Result*

<table>
<thead>
<tr>
<th>Object Name</th>
<th>Total Deformation</th>
<th>Equivalent Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>State</td>
<td>Solved</td>
<td></td>
</tr>
</tbody>
</table>

**Scope**

<table>
<thead>
<tr>
<th>Identifier</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Scoping Method</td>
<td>Geometry Selection</td>
<td></td>
</tr>
<tr>
<td>----------------</td>
<td>-------------------</td>
<td></td>
</tr>
<tr>
<td>Geometry</td>
<td>All Bodies</td>
<td></td>
</tr>
</tbody>
</table>

**Definition**

<table>
<thead>
<tr>
<th>Type</th>
<th>Total Deformation</th>
<th>Equivalent (von-Mises) Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>By</td>
<td>Time</td>
<td></td>
</tr>
<tr>
<td>Display Time</td>
<td>Last</td>
<td></td>
</tr>
<tr>
<td>Calculate Time History</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Identifier</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Results**

| Minimum   | 0. mm | 0.68575 MPa |
| Maximum   | 0.33263 mm | 859.71 MPa |

**Information**

| Time       | 1. s |
| Load Step  | 1    |
| Substep    | 1    |
| Iteration Number | 1    |

**Integration Point Results**

<table>
<thead>
<tr>
<th>Display Option</th>
<th>Averaged</th>
</tr>
</thead>
</table>