# Thermomechanical Fatigue Analysis of Diesel Engine Piston: Finite Element Simulation and Lifetime Prediction Technique

**S. M. Sivachev and L. L. Myagkov**

**Abstract** Nowadays, piston aluminum–silicon alloys are widely used for high-powered engines due to their low specific gravity, high thermal conductivity, and good castability. However, under the conditions of increase of thermomechanical loads caused by the rise in the specific power output of diesel engines, and the operating temperatures of pistons go up to 0.8–0.9 of the melting temperature resulting in a significant reduction of the Al–Si alloy high-temperature strength. In this regard, to provide a required lifetime of pistons, it is necessary to more precisely simulate their thermal and stress-strain state, taking into account two-frequency loading and inelastic deformation. In this paper, a review of existing methods for the piston life estimation is carried out; a calculation method of the piston transient temperature and strain fields for engine start-stop cycles and one operating cycle at a nominal power mode is developed. The material constants in plasticity and creep models for the Al–12Si–Cu–Ni–Mg alloy are determined. On the basis of the obtained stabilized elastoplastic hysteresis loop, the piston low-cycle fatigue is estimated using the energy criterion. According to experimental data, the piston life is corrected taking into account high-frequency load.

**Keywords** Diesel engine · Piston · Two-frequency loading · Plasticity · Creep · Low-cycle fatigue

## 1 Introduction

### *1.1 Background*

The aluminum–silicon alloys of the Al–Si–Cu–Ni–Mg system containing up to 12 different elements have high wear resistance, high thermal conductivity, low ther-

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mal expansion coefficient, low specific weight, good casting properties, and high mechanical characteristics [1–3]. The rise in specific power output of high-powered diesel engines leads to increase in mechanical and thermal loads on the combustion chamber components: the cylinder pressure can reach 200 bar, and the maximum piston temperatures represent a homologous temperatures (the ratio to the melting temperature in K) range 0.8–0.9 [3–5].

The high homologous temperatures on the piston bowl rim cause a significant decrease in piston thermal strength. Therefore, it is necessary to more precisely predict the piston operating life. The development of lifetime prediction techniques is possible using the mathematical modeling of stress-strain state of pistons, which takes into account two-frequency loads and inelastic effects of the material.

The heat-stressed components of combustion chamber (in particular, piston and cylinder head) work under two-frequency thermomechanical loading: low-frequency cycles correspond to changes in engine operating modes, and high-frequency cycles represent each operating cycle (two rotation of crankshaft for four-stroke engine). Moreover, high temperatures at the piston bowl edge lead to the appearance of plastic and creep deformations.

Belov [6] used the damage linear summation rule to account high- and low-frequency loads:

$$d_n + d_t^f + d_t^c + d_i = 1 \tag{1}$$

where $d_n$ and $d_i$ denote the damages obtained on the nominal and idle modes, $d_t^f$ and $d_t^c$ are the fatigue and creep-induced damages obtained on the transient modes, respectively.

The damages obtained on the nominal and idle modes are defined as the ratio of the number of applied cycles to the number of cycles to failure. The fatigue and creep damages are calculated using the strain-based criterion:

$$d_t^f = N \Big/ \left( \frac{\varepsilon_f(T)}{2\Delta\varepsilon_p} \right)^2; \quad d_t^c = \frac{\Delta e_p N}{\varepsilon_f(T)} \tag{2}$$

where $N$ is the number of applied cycles, $\varepsilon_f(T)$ is the true fracture ductility, $\Delta\varepsilon_p$ is the elastoplastic hysteresis loop width, $\Delta e_p$ is the one-sided translation of the elastoplastic hysteresis loop in one cycle.

Ivanchenko [7] proposed other relation:

$$A = \frac{n}{N_f} + \frac{n^*}{N_f^*} \tag{3}$$

where $A$ is the total damage, $N_f$ is the allowable number of cycles corresponding to the stress amplitude resulting from the high- and low-frequency loads, $N_f^*$ is the allowable number of cycles corresponding to the stress amplitude resulting from the thermal high-frequency load, $n$ and $n^*$ are the number of low- and high-frequency cycles to fatigue crack initiation, respectively.

Saltykov with colleagues [8] in considering the two-frequency loading spectrum of diesel engine piston crown proposed to determine the ratio of the durability of structure with two-frequency loading to the lifetime under one-frequency loading with amplitude equal to the sum of the two-frequency loading components amplitudes:

$$\frac{N_f}{N_h} = \frac{\sigma_a^t}{\sigma_a^l \sqrt{1 + \left(\frac{f_h}{f_l}\right)\left(\frac{\sigma_a^l}{\sigma_a^t}\right)^2}} \tag{4}$$

where $N_f$ is the longevity under a two-frequency loading as determined by the number of cycles of the low-frequency component, $N_h$ is the longevity under a one-frequency loading, which corresponds to a stress amplitude equal to the sum of the amplitudes of the two-frequency load components, $\sigma_a^l$ is the low-frequency stress amplitude, $\sigma_a^t$ is the total amplitude of the high- and low-frequency components, $f_h$, $f_l$ are the loading frequencies.

It should be noted that this approach is applicable in the case of elastic deformation.

The recent studies on the durability of materials for heat-stressed parts operating under two-frequency loading show that damage linear summation rule (Palmgren-Miner rule) is not confirmed [9]. This is because the crack propagation mechanisms under one- and two-frequency loadings are different [10]. Thus, it should be used the low-cycle fatigue curves representing relation between the number of low-frequency cycles of thermomechanical load and the amplitude of superimposed high-frequency mechanical load.

## *1.2 Current Work*

For this research, the following tasks have been formulated:

- Develop a method based on the commercial software ANSYS for calculating transient temperature and strain fields of the piston resulting from high- and low-frequency loads;
- Determine material constants in plasticity and creep models for the working temperature range of piston;
- Investigate the elastoviscoplastic deformation of the piston combustion bowl edge;
- Evaluate the piston fatigue lifetime.

## 2 FEM Computational Analysis

### 2.1 Object of Study

The piston of the v-type 8-cylinder YaMZ-6586 diesel engine with a rated power of 309 kW is chosen as the object of this study. The main technical characteristics of the engine are shown in Table 1.

The solid model of piston was built in the design software SolidWorks. To increase the stress concentration, the combustion bowl rim was made sharp without fillet radius.

### 2.2 Calculation Methodology

The developed method consists of performing a series of piston thermal and stress-strain state calculations (see Fig. 1). The transient temperature field for engine start-stop cycle is used for modeling of low-cycle thermal stress. The high-cycle thermomechanical stress is calculated using transient temperature field for one operating cycle and mechanical load (gas pressure and inertia).

**Table 1** Technical characteristics of YaMZ-6586 diesel engine

| | |
|---|---|
| Bore (mm) | 130 |
| Stroke (mm) | 140 |
| Rated power (kW) | 309 |
| Nominal rotational speed ($min^{-1}$) | 1900 |
| Specific effective fuel consumption (g $kW^{-1}$ $h^{-1}$) | 195 |

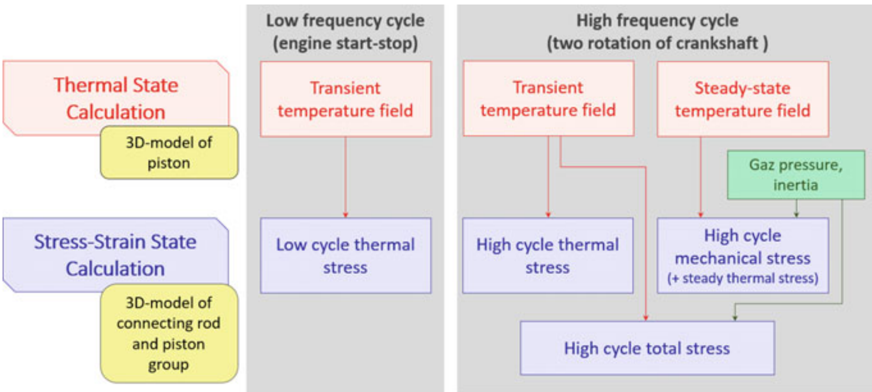


**Fig. 1** Scheme of thermal and stress-strain state calculation methodology

### 2.3 Determination of Plasticity and Creep Models Coefficients

Presently, it is necessary to use mathematical models which describe nonlinear hardening of material [11]. The curves of cyclic deformation of piston aluminum–silicon alloys show that silumin is characterized by kinematic hardening [12]. This means that the yield surface size does not change and the surface translates in the load direction. In the present work, the model proposed by Chaboche [13] is used. According to this model, the translation of the center of yield surface $\Delta\alpha$ is calculated by the following relation:

$$\Delta\alpha = \frac{2}{3} C \Delta\varepsilon_p - \gamma\alpha\Delta\hat{\varepsilon}_p \tag{5}$$

where $\alpha$ is the back stress, $C$ is the hardening modulus, $\Delta\varepsilon_p$ is the plastic strain increment, $\gamma$ is the rate of hardening modulus decrease, and $\Delta\hat{\varepsilon}_p$ is the equivalent plastic strain increment.

The coefficients $C$ and $\gamma$ were evaluated according to the stress-strain curves using the least-squares method.

The mathematical description of creep phenomena is based on the constitutive equations obtained from experiments. The Norton's law shows the relation between minimum creep strain rate $\dot{\varepsilon}^c_{\min}$, applied stress $\sigma$, and temperature $T$ [14]:

$$\dot{\varepsilon}^c_{\min} = A\sigma^n \exp\left(-\frac{Q}{RT}\right) \tag{6}$$

where $A$, $n$ are the material constants, $Q$ is the creep activation energy, and $R$ is the universal gas constant.

The experiments were done for speciments of Al–12Si–3.5Cu–2Ni–0.8Mg alloy in temperature range of 250–300 °C and stress range of 80–140 MPa show that this alloy is characterized by dislocation type creep [15]. The material constants were determined by this data.

### 2.4 Calculation of Boundary Conditions

The engine cycle at nominal power mode was calculated using the engine simulation software Diesel-RK. The thermal boundary conditions for piston surfaces were determined through ICE program. These computer tools are developed at the Bauman University. To obtain the high-cycle temperature variation for one operating cycle, the local values of heat transfer coefficient as a function of crank angle were calculated according to relation from [16]. To calculate the mechanical loading, the gas pressure and piston acceleration were applied to finite element model. The piston acceleration was calculated by the equation [17]:

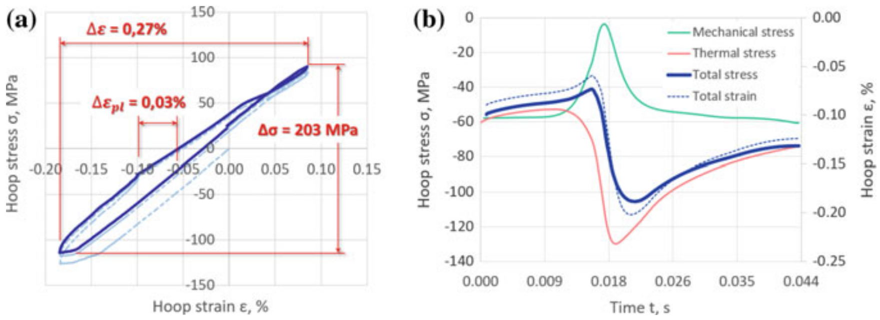


**Fig. 2** Stress-strain state of piston bowl edge pin-axle. **a** Stabilized elastoplastic hysteresis loop corresponding to three start-stop cycles and **b** High-frequency variations of stresses and strains during one operating engine cycle

$$j = R\omega^2(\cos\alpha + \lambda\cos 2\alpha) \tag{7}$$

where $j$ is the piston acceleration of the first- and second-order, $R$ is the crank radius, $\omega$ is the angular velocity of crankshaft rotation, $\alpha$ is the current value of crank angle, and $\lambda$ is the ratio of crank radius to connecting rod length.

### 2.5 *Results*

After calculating three start-stop cycles, the stabilized elastoplastic hysteresis loop for the bowl edge in the piston pin plane was obtained which is shown in Fig. 2a. The simulation of operating cycle allowed to obtain the stress variations as a result of high-frequency thermomechanical loads (see Fig. 2b).

## 3 Calculation of Piston Fatigue Lifetime

The low-cycle fatigue criteria are divided into phenomenological, cumulative damage, and crack growth models [18]. The classical phenomenological fatigue criterion represents the relation between a criterion function and the number of cycles to failure $N_f$ [19]:

$$\max_{x\in\Omega}\left[\Phi\left(\varepsilon, \varepsilon_{pl}, \sigma, \ldots\right)\right] = aN_f^b \tag{8}$$

where $a$, $b$ are two material parameters, $\varepsilon$ is the strain tensor, $\varepsilon_{pl}$ is the plastic strain tensor, and $\sigma$ is the stress tensor.

As per the energy criterion proposed by Skelton [20], the left side of "Eq. 8" is the inelastic energy determined by stabilized hysteresis loops:

$$\Delta W_p = W_f'(2N_f)^{\beta} \tag{9}$$

where $W_f'$, $\beta$ are the material constants, $\Delta W_p$ is the plastic energy dissipated per cycle.

The experimental studies suggest that the hysteresis energy approach is a better choice to evaluate the thermomechanical damage as it includes both stress and strain effects [21].

The number of start-stop cycles to failure (288,000 cycles) was obtained using the constants for Al–Si–Cu–Ni–Mg alloy from [2].

The decrease of piston lifetime due to superimposition high-frequency load on low-frequency loading was done according to experimental curves from [9]. These curves are the relations between the number of macrocycles of thermomechanical loading (with the low-frequency strain amplitude of 0.275% and the maximum cycle temperature of 300 °C) and the amplitude of applied high-frequency mechanical strain. Since the strain amplitude during the engine operating cycle is 0.07% (see Fig. 2b), the durability of sharp bowl edge will decrease by 80% to 57,600 cycles.

## 4 Conclusions

The transient temperature and strain fields of the piston corresponding to high- and low-frequency thermomechanical loading were obtained. The piston low-cycle fatigue life was evaluated using energy criteria and it is equal to 288,000 cycles. It was found that superimposed high-frequency load decreases durability of sharp combustion bowl edge to 57,600 cycles.

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