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Simulation of Thermal-Mechanical Strength for Marine Engine Piston Using FEA

Elijah Musango Munyao[1], Jiang Guo He[2], Yang Zhiyuan[3], Zou Xiang Yi[4]
Merchant Marine College - Shanghai Maritime University Shanghai- China.

ABSTRACT
This paper involves simulation of a 2-stroke 6S35ME marine diesel engine piston to determine its temperature field, thermal, mechanical and coupled thermal-mechanical stress. The distribution and magnitudes of the afore-mentioned strength parameters are useful in design, failure analysis and optimization of the engine piston. The piston model was developed in solid-works and imported into ANSYS for preprocessing, loading and post processing. Material model chosen was 10-node tetrahedral thermal solid 87. The simulation parameters used in this paper were piston material, combustion pressure, inertial effects and temperature. The highest calculated stress was the thermal-mechanical coupled stress and was below the yield stress of the piston material (580Mpa) at elevated temperatures hence the piston would withstand the induced stresses during work cycles.
Keywords- Coupled Thermal-Mechanical strength analysis, Finite Element Analysis (FEA); Mechanical stress, Temperature field, Thermal stress.

I. Introduction

Increased need for high power density, low emissions and high fuel efficiency impose restrictions on engine component design [1]. Hence design and analysis of engine components has become more complex. One of these components is the engine piston. The piston of a diesel engine is usually subjected to periodically changing thermal and mechanical loads [2]. The stress fields induced onto the piston due to coupled thermal-mechanical loads are difficult to analytically determine, however using finite element analysis methods it possible to study and analyze the strength of pistons and other complex components and structures [3].

The main requirements of the piston are that it should contain all the fluids above and below the piston assembly during the cycle and that it should transfer the work done during combustion process to crankshaft via the connecting rod with minimal mechanical and thermodynamic losses [4]. To meet these two major requirements, the piston should have sufficient thermal conductivity, Low thermal expansion, and high temperature strength, high strength to weight ratio and High resistance to surface abrasion. The above requirements demand for high thermal and mechanical strength designs for engine piston.

Piston simulation and strength analysis has been an important area of research which has attracted great research interests [5]. Considering only mechanical loading Swati et al [4] investigated the stress distribution a piston over one engine cycle. Transient analysis was carried out and it was concluded that inertial loads should not be ignored in strength analysis of the engine piston.

To investigate the effects of improved cooling on a piston, JI Wu et al [6] carried out a thermal mechanical stress analysis with varying distances between the cooling shaker and the piston crown top surface which come into direct contact with the hot gases. It was reported that this distance had a great influence on the thermal stresses, where the stress magnitude reduced with reducing distance since more heat could be transferred to the cooling oil; however, this had minimal effect on the mechanical stress induced on the piston. Hongyuan et al [2] reported that thermal stresses are usually higher than the mechanical stresses induced. This conclusion was arrived at after a coupled thermal mechanical stress analysis. Yanxia et al [3] through a thermal-mechanical stress analysis of a marine engine reported that the maximum temperature occurs at the piston crown top surface which gets into contact with the hot gases and the maximum deflection similarly occurs in the same region, but at the center of the surface.

Stress concentration is one of the main reasons for piston failure Praful et al [7] and therefore it still remains utterly important to carry out piston strength analyses to identify the loads that contribute to the high stresses and use the results for further design improvement and optimization. The purpose of this paper is to simulate the temperature field, the thermal-mechanical coupling stress field of the 6S35ME engine piston with the finite element analysis method to obtain results which can be used for design improvement and optimization.

II. Finite Element Model

FEA tool is the mathematical idealization of the real system. It is a computer based method that breaks geometry into elements joined by nodes and a series of equation to each element are formed, and then solved simultaneously to evaluate the behavior of the entire system. The MAN B&W's marine engine piston crown is made of heat resistant chrome molybdenum alloy steel. It is oil cooled and rigidly bolted to the piston rod to allow distortion free transmission of the firing pressure [8].The piston model is developed and analyzed to determine the magnitude and distribution of the thermal and mechanical stresses induced during operation. The following are the engine configuration for which the piston belongs [9]:

Speed = 142RPM
Maximum combustion pressure = 18MPa
Scavenging pressure = 0.265MPa
Piston stroke = 1.55m
Connecting rod length = 1.55m

2.1 Material Properties

The piston is made of heat resistant chrome –molybdenum alloy steel [8]. Chromium is effective for increasing strength and improving oxidation resistance while molybdenum increases strength at higher temperatures [10]. 42CrMo4 steel alloy is employed in modeling and simulation of the piston. The material has a constant density and Poisson's ratio of, 7800 Kg/m^3 and 0.3 respectively. The variation of the material's Young's modulus (E), Yield strength (Y.S), Thermal conductivity (k) and Coefficient of thermal expansion (CTE) with Temperature (T) are shown in TABLE 1[11].

Table 1: Variation of material properties with temperature

T (K)	E (GPa)	Y.S (MPa)	K (W/M.K)	C.T.E (10^{-6}/K)
293	212	860	44	13.2
403	210	800	43	13.2
573	208	750	40	13.7
723	202	580	37	13.7

2.2 Piston Model

Finite element modeling of any solid component consists of geometry generation, applying material properties, meshing the component, boundary constraints definition, and application of the proper load types. These steps lead to calculation of displacements and stresses in the component being analyzed. In this work, a model of the piston was developed in solid works and imported into ANSYS. The magnitudes and distribution of thermal, mechanical, thermal-mechanical coupled loads' displacements and corresponding stresses were calculated.

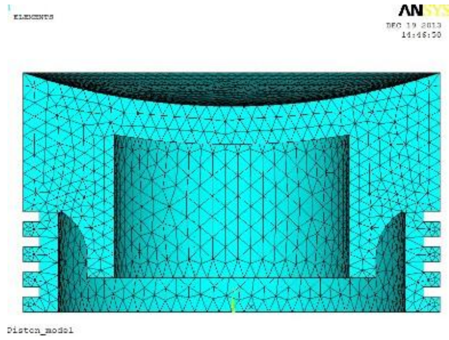

Figure 1: Piston model

III. Boundary Conditions

3.1 Thermal Boundary

The temperature of the piston surface and heat transfer through the piston body cannot be measured accurately, [2] therefore, thermal boundary conditions are used to simulate the temperature field distribution. A complete engine cycle was simulated using the known engine parameters. Mean heat transfer coefficient and the mean (bulk) temperature were determined using (3&4). The heat transfer of the piston was divided into the following 3 sections:

3.1.1 The heat transfer between the combustion gases and the piston crown.

The transient heat transfer coefficient of hot flue gases was obtained from one-dimensional thermodynamic analysis of engine cycle. There are numerous models that have been put forward to determine the heat transfer coefficient of these gases inside the engine cylinder [12]. The heat transfer model due to Woshni (1967) was used to determine the instantaneous heat transfer coefficient of the hot gases [13].

$$h_g = 0.820 D^{-0.2} P^{0.8} U^{0.8} T^{-0.53} \quad (1)$$

$$U = C_1 C_m + C_2 \frac{V_d T_r}{P_r V_r}(P - P_o) \quad (2)$$

P – Transient gas pressure (MPa),
D – Bore diameter (m),
U – Characteristic velocity (m/s)
T – Transient cylinder gas temperature (K), C_1-2.28, C2=0; during compression;
C_1=2.28, C2=0.00324; during combustion, C_m - piston mean speed,
V_d –swept volume,
T_r,V_r,P_r – Temperature, Volume and Pressure respectively, determined from a reference known position and P_o is the reference pressure.

In the analysis, the mean temperature and pressure for a cycle were used. These were determined using (3&4).

$$h_m = \frac{1}{360}\int_0^{360} h_g d\theta \quad (3)$$

$$T_m = \frac{\frac{1}{360}\int_0^{360} T_g h_g d\theta}{h_m} \quad (4)$$

h_m, T_m are the mean heat transfer coefficient and mean temperature respectively.

3.1.2 Heat transfer between the piston under-crown and the cooling oil.

The piston under crown has a complex shape; however, in this work it is modeled with a circular cross-section. The flow of the cooling oil is assumed to be fully developed turbulent and numerous correlations have been put forward to determine the turbulent heat transfer coefficient. The correlation due to Dittus and Boelter (1930) was used in this work to determine the heat transfer coefficient [14], (5&6).

$$Nu = 0.0243 \mathrm{Re}^{0.8} \mathrm{Pr}^{n} \quad (5)$$

$$h_c = \frac{Nuk}{D} \quad (6)$$

Nu, Pr, Re, k, D are Nusselt's number, Prandtl number, Reynolds's number, thermal conductivity of lubricating oil and hydraulic diameter respectively. The value of index n is 0.4, when the fluid is being heated and 0.3 when the fluid is being cooled.

$\mathrm{Pr} = \frac{\mu cp}{k}$; Where U,μ,cp,k are the piston mean speed, dynamic viscosity, specific heat capacity and thermal conductivity of lubricating oil respectively.

3.1.3 Heat Transfer at the Piston Crown top and Ring Lands

Heat is transferred to the lubrication oil film between the piston crown top and ring lands through convection and is modeled as a laminar flow between two parallel plates [14]. Hydraulic diameter D_h is determined as a function of the cross-sectional area of the plates per unit depth and the wetted perimeter. The heat transfer coefficient h_c is determined using (7).

$$D_h = \frac{4A}{P} \;;\; A = 2b*1 \; P = 2 \; D_h = 4b \;;$$

$$Nu = \frac{h_c D_h}{k} = 8.235 \quad (7)$$

Where $A,P,D_h,2b$ are Cross-sectional area of the plate per unit depth, wetted perimeter, the hydraulic diameter and the gap between the piston and the cylinder (the lubricating oil film) respectively.

3.2 Mechanical Boundary Conditions

The mechanical loads due to combustion pressure and reciprocating inertia of the piston were considered in this study. The known engine parameters and thermodynamic relations were used to determine the maximum combustion pressure. The pressure variation over one engine cycle is shown in the fig. 2. The maximum pressure was applied as a surface load on the piston model top face. The piston inertia load is applied as acceleration rather than a force in ANSYS. The maximum acceleration of the piston was determined using (8):

$$a = R\omega^2(\cos\alpha + \lambda\cos(2\alpha)) \quad (8)$$

Where $a,R,\omega,\lambda,\alpha$ are piston acceleration, crank radius, crank velocity, crank radius-connecting rod ratio and crank angular position respectively.

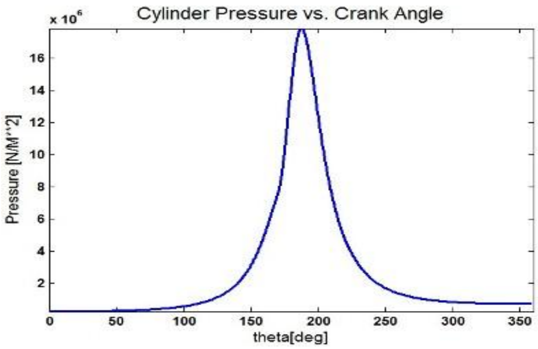

Figure 2: Pressure variation with crank angle

Results and Analyses

4.1 The temperature field of the piston

The temperature field was calculated after applying convective thermal loads determined from (1-7). The bulk temperature of the cooling oil was determined by considering the inlet and outlet temperatures of the cooling oil. The maximum temperature obtained was 660K which occurred at the piston crown top region and the minimum of 323K occurred at the lower part of the piston crown. The high temperatures can be attributed to the inadequate cooling which can be improved by considering the cooling shakers which would improve the cooling process. These were omitted in this analysis for model simplification. It can be seen that the temperature contours are smooth across the model which shows that the analysis is credible.

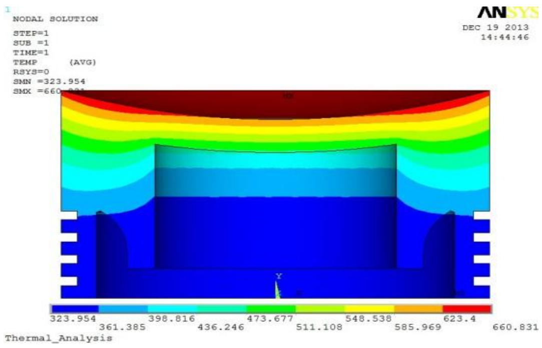

Figure 3: Temperature distribution over the piston.

4.2 Thermal Stress Analysis

The thermal effect on the piston resulted to a maximum deformation of 0.843mm which occurred at the edges of the piston top face in direct contact with the hot gases in the radial direction as shown in fig. 4. This can be attributed to the thermal expansion of the piston edges. The maximum calculated thermal stress on the piston was 495MPa as shown in fig. 5. This occurred at the inner boundary of the piston top surface. This can be attributed to high thermal strains due to the large temperature differences resulting from cooling and the irregular shape of this region. The calculated stress is below the lowest yield strength of 580MPa at high temperature as shown in table 1.This implies that at the calculated temperature the piston still has adequate strength for operating safely without failure. The piston deformation value is also within a safe margin and below the gap between the piston and the cylinder bore of 1.95mm.

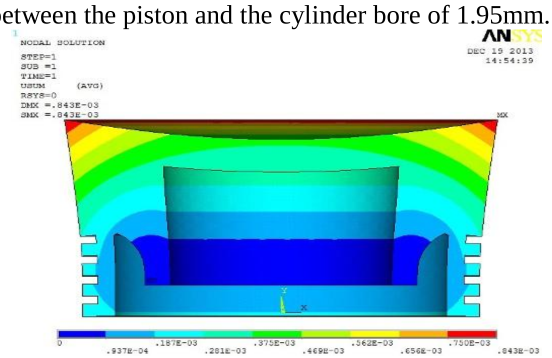

Figure 4: Piston deformation under thermal load

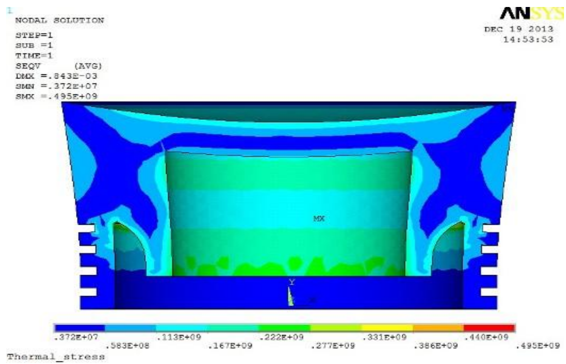

Figure 5: Thermal stress on the piston

4.3 Mechanical Stress Analysis

The maximum deformation of the piston occurred at the centre of the piston crown. This is because it is the main thrust surface of the combustion pressure load. In addition, this region has a reduced thickness compared to the other regions of the piston. The optimally reduced thickness improves heat transfer from the piston to the cooling oil which is supplied from underneath the piston crown. As shown in fig. 6&7, maximum piston deformation under mechanical load is 0.119mm with a maximum induced mechanical stress of 200MPa.

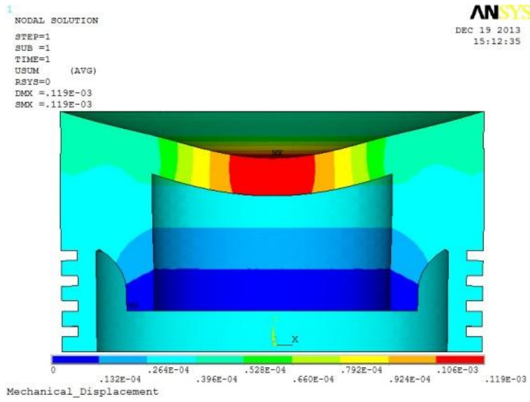

Figure 6: Piston deformation under mechanical load

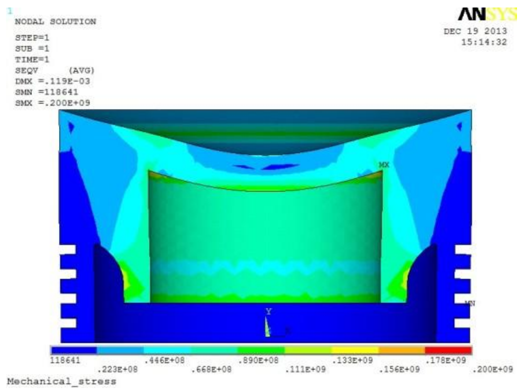

Figure 7: Mechanical stress on the piston

4.4 Coupled Thermal-Mechanical Stress Analysis

A coupled thermal-mechanical stress analysis of the piston under both thermal and mechanical loads was performed. The analysis results showed that the maximum deformation of the piston was 0.871mm and occurred at the piston top surface edges as shown in fig. 8. This deformation is within a safe limit and would not affect the safe and effective operation of the piston since the piston-cylinder gap is 1.95mm. A maximum coupled thermal-mechanical stress of 517MPa was induced onto the piston by the effect of the combined thermal and mechanical loads. The calculated stress is within a safe margin relative to the lowest yield strength value of 580MPa at elevated temperatures.

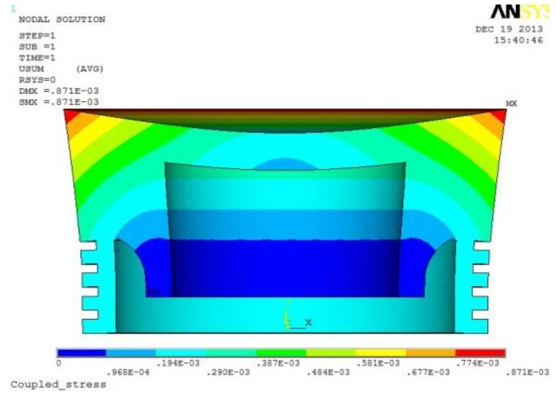

Figure 8: Piston deformation under coupled Thermal-Mechanical load

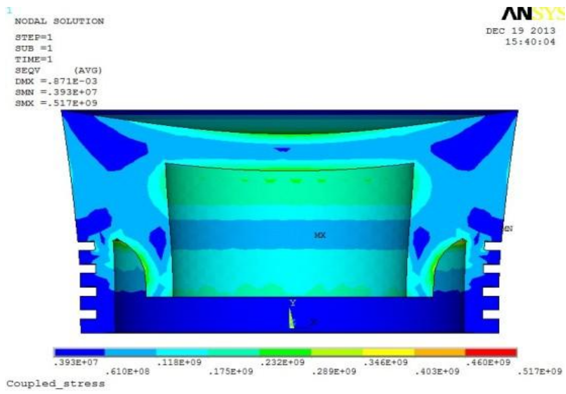

Figure 9: Coupled Thermal-Mechanical stress on the piston

Conclusion

From the temperature field analysis, the maximum temperature on the piston was 660K and the induced thermal deformation and thermal stress were 0.843mm and 495MPa respectively. The Mechanical load comprised of the combustion pressure and the inertia load and induced a mechanical deformation and mechanical stress of 0.119mm and 200MPa respectively. Under both thermal and mechanical load, the calculated deformation and coupled stress were 0.871mm and 517MPa respectively. From the analysis, it is evident that thermal stress was higher than mechanically induced stress hence it could be concluded that the piston would fail due to the thermal load rather than the mechanical load and hence during optimization design, this could be put into consideration to ensure that thermal load is reduced. It can also be deduced that individually, thermal and mechanical stress proportions have a direct influence on the coupled thermal-mechanical stress hence during design each load can be considered and reduced independently. It can be concluded that the piston can safely withstand the induced stresses during its operation.

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