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Thermal Modelling of a Marine Diesel Engine Piston

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Abstract: ***This work presents a computational thermal analysis of a 4-ring articulated piston of a marine diesel engine for determining the temperature distribution of piston head. The temperature distribution is the basis of calculating thermal stresses and deformations and subsequent identification of their largest value locations. The piston model is developed in CATIA V5R19 and exported to ANSYS for finite element analysis. The input parameters are gas temperature, cooling water temperature and oil temperature, whereas the boundary conditions given in the analysis are coefficients of heat transfer (CoHTs) and heat flux. COHTs vary radially on the piston head and these values are calculated using Seal's formula. The piston top surface is divided into ten sub-regions of equal intervals and the thermal analysis results show that there are temperature variations on piston top surface where the 5-6-7-8 sub-regions have the highest temperature range of 603.59-632.91K. The temperature of the first four regions gradually increases and their temperature range value is 486.31-603.59 K. In the remaining two sub-regions the temperatures were found to be in the range of 544.95-603.59K. The piston skirt has the minimum temperature range of 369.03-398.35Ksince it is far from combustion. The total heatflux pattern closely follows that of the temperature. The results obtained will help in enhancing the piston design in the future for more safety and longer life.***

Keywords: Marine, Piston, Thermal analysis, Finite Element Analysis (FEA), and COHT

1. Introduction

A pistonis considered one of the most important components of IC engine which is subjected to a variety of loadings such as thermal, mechanical and inertial loading. Due to high temperature gases produced during combustion, the piston top surface is subjected to a tremendous thermal loading. The amounts of heat transferred are unequal; some parts will be exposed to higher combustion temperatures than others resulting in radially changing heat transfer coefficients (CoHTs) along the piston top surface.

Because of high combustion temperatures, Pistons are designed with a cooling gallery to reduce the thermal stresses and protect it from failure. The oil enters the cooling gallery inlet, flows through it and then exits from the outlet. A large portion of heat is also transferred from piston to cylinder liner through piston ring assembly which consists of four ring grooves where the rings are installed.The area between piston top surface and first ring groove is called the piston top land which is of great importance when thermal analysis is studied because it is subjected to large thermal and mechanical loads.

The main objective of this work is to determine the temperature variation through Finite Element Analysis (FEA) to give information about high thermal stress locations to be controlled in better future designs for safer and longer life pistons. The piston considered in the analysis has a radius of 140 mm and a height of 302.4 mm. Its top surface has a bowl shape and it is divided into ten sub-regions of equal intervals of 14 mm. The temperatures of cooling water and oil are respectively 368 K and 400 K.

2. Literature Survey

The first studies related to thermal piston analysis began in the 1940s with the development of the electrical analogy method [1] and simple analytical method [2,3] which are used to calculate the field temperature in the piston in order to obtain cooling of the piston. At the beginning of 1960s rheological models were developed, making it possible to take into account the geometric complexity of the piston [4,5]. With the development of the finite element method, numerical models appeared from the beginning of the 1980s.

Li [6] used a three dimensional finite element model of an aluminum diesel engine piston to calculateoperating temperatures and thermal and mechanicalde formations due to thermomechanical loads. He showed that skirt contours play an important part in the reduction of scuffing and friction. Finite element models of pistons for high-speed direct injection (HSDI) diesel engines were developed and showed the influence of the thermal interaction on the design and development of pistons [7,8].MTahar Abbes et al. [9] described thethermomechanical behaviour of a direct injection diesel engine piston by presenting a three dimensional finite element full analysis.

3. Finite Element Analysis

The idea behind finite element analysis is to divide a model into a fixed finite number of elements where each element expresses the physical and geometrical properties of the body. The piston is modelled using 3D CAD software (CATIA V5R19) as shown in figure1. A half part geometric model of piston can be also seen in figure 2. The model is then exported to FEA software for processing (ANSYS). To obtain accurate calculation results, a reasonable mesh size must be determined. The smaller the cell, the higher the accuracy and the longer the calculation time. However, the cell size can be modified according to the specific conditions. For example, the cell size can be smaller where the shape is complicated or the temperature changes dramatically, while at the rest of occasion it can be larger. In this way, the calculation accuracy can be improved without increasing the number of cells and nodes [10]. Figure 3 shows meshing of half part piston where the model has 147051 elements and 237916 nodes.

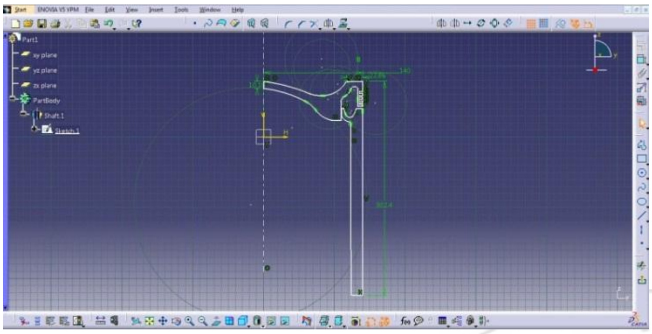

Figure1: Piston sketch in CATIA V5R19

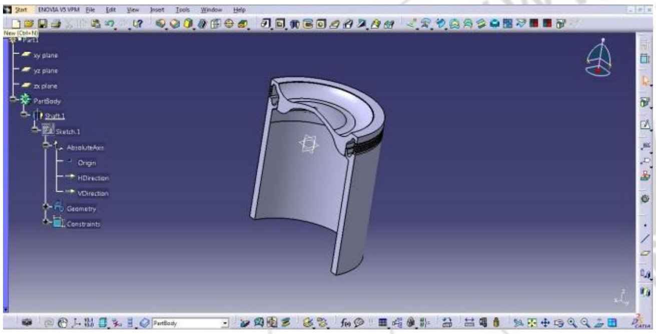

Figure 2: Geometric model of piston half part

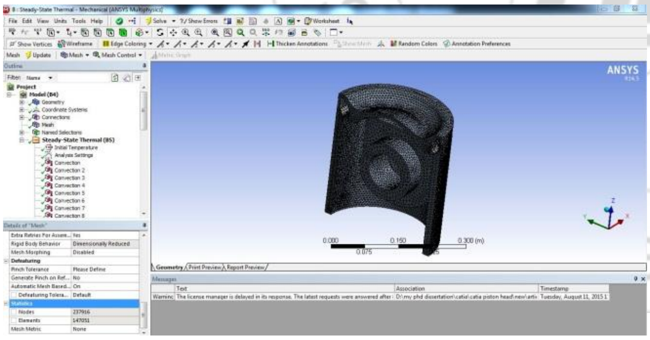

Figure 3: Meshing of the piston half part

3.1 Thermal and Geometric Properties of Piston Materials

The piston investigated is a four ring articulated piston of a marine diesel enginemanufactured of a structural steel crown and aluminum alloy skirt. A crown of a high strength material is required to withstand the tremendous thermal and pressure loads of combustion. At the same time an aluminum alloy skirt has the advantage of light weight with lower inertia. However, because aluminum has higher coefficient of expansion than structural steel, larger clearance with cylinder liner should be considered during manufacturing.

The lubricating oil film fills the clearance between piston ring assembly and cylinder liner bore. This clearance is supposed to be small enough to prevent blow-by of combustion gases from combustion chamber to crank case and large enough to make sure that piston moves smoothly in the cylinder. Lubricating oil is also applied to the cylinder liner surface in order to reduce friction between rings and cylinder liner [11].

Table 1: piston material properties

Properties	*Aluminum alloy*	*Structural steel*
Density	$2770\ kg/m^3$	$7850\ kg/m^3$
Young's Modulus	$71 \times 10^3 Mpa$	$2.0 \times 10^5 Mpa$
Poisson Ratio	0.33	0.3
Thermal conductivity	Tabular	$60.5\ w/mk$
Specific heat	$875\ J/kg.k$	$434\ J/\ kg.k$

3.2 The Applied Boundary Conditions

Firstly coefficients of heat transfer (CoHTs) and temperature of the surrounding medium in different parts of the piston body should be obtained. The surrounding medium heat is transferred in different ways. At the top surface of piston, combustion heat is transferred mainly by convection. Heat flow from piston rings to cylinder liner is not a simple process because of large number of circumstances affecting the process such as oil film thickness varying, rings twisting, piston titling, thermal expansion, etc. However, a common assumption regarding heat flow from piston ring assembly to cylinder liner that it occurs through convection only [12]. Therefore, determining the heat transfer boundary condition is mainly to determine heat transfer coefficient and the medium temperature between the piston boundary and combustion gases, cooling water, and oil. Generally these boundary conditions are based on empirical or semi-empirical methodology.

4. Thermal Numerical Analysis

The thermal analysis of the piston was conducted based on the assumption of steady working state of engine. Based on this assumption, bothpiston top surface temperature and the instant CoHT of gas were considered time averaged to fit the need of steady thermal analysis in this paper and can be calculated by the following equations:

$$h_m = \frac{1}{720}\int_0^{720} h_g \, d\varphi \qquad (1)$$

$$T_{res} = \frac{1}{720.h_m}\int_0^{720} h_g T_g \, d\varphi \qquad (2)$$

Where h_g, T_g are respectively the instant CoHT of gas at piston top surface and instant gas temperature.H_m was defined as the average CoHT of gas at the top surface of piston head and the average piston surface temperature is denoted as T_{res}.

For more precise and detailed thermal analysis, the spatial effect of CoHTs on the top surface of piston should be considered. For this reason Seal's formula was usedwhich assumes that the CoHTs change along the radial direction. The equations are as below followed:

$$If\ r<N,\ h_r = \frac{2h_m}{1+e^{0.1N^{1.5}}} \qquad (3)$$

$$If\ r>N,\ h_r = \frac{2h_m}{1+e^{0.1N^{1.5}}} e^{0.1(2N-r)^{1.5}} \qquad (4)$$

Where r is the distance from local point to the center axis of piston head, N is the distance from the point where maximum temperature occurs to the center axis of piston.

For numerical simulation the piston top surface has been radially equally divided into 10 sub-regions with concentric circles each has a specific CoHT obtained by equations 3 or 4. So the boundary conditions of convection are specified according to the assumption of the distribution of the couple (h_r,T_s) on the surfaces. The temperatures T_s are represented by T_{res} which is the average temperature of combustion gases in the combustion chamber through engine cycle (913 K),T_w is the temperature of cooling water in the cylinder water jacket(368 k) and T_h is the temperature of the cooling oil (400 k). We have CoHTvaluesof ten sub-regionson the top surface of piston as it is shown in figure 4.Regarding the calculation of CoHTs of other remaining parts such as piston ring region and skirt, it is done by empirical equations.

The surface of the piston model has been divided into 36 sub-regions and each sub-region was assigned with a specific initial CoHT which was obtained with previous equations. The applied boundary conditions are coefficients of heat transfer and heat flux of 1MW/m^2.

5. Results and Discussion

The thermal numerical analysis results of piston are shown in figure [5] and figure [6]. The piston top surface temperature gets increased from piston center to the eighth sub-region with a temperature range value of 456.99-632.91K to start getting reduced to 544.95K. Red colored area of piston top surface (5-6-7-8 sub-regions) have the highest temperature range value of603.59-632.91K.

As we can see in figure 7, the piston skirt has the minimum temperature range of 369.03-398.35 K since it is far from combustion. Regarding the total heat flux, it coincides with temperature whereas same region has the highest total heat flux.

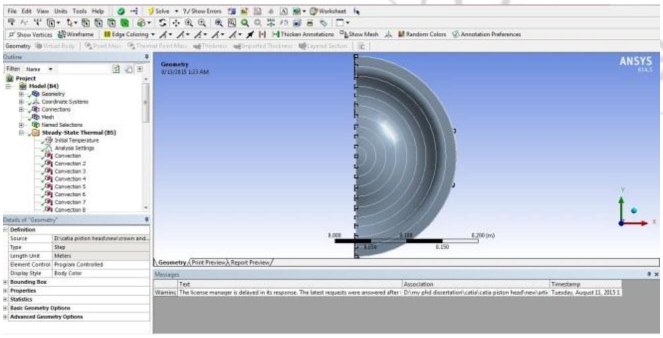

Figure 4: sub-regions of piston top surface

6. Conclusions and Future Work

Thermal analysis of a marine diesel engine piston is presented in this paper to investigate the thermal field distribution which indicates the maximum stress locations to be considered in a future safer design. Coefficients of heat transfer COHTs play a sensitive rule in determining the temperature distribution since they are considered radially changing on piston top surface according to Seal's formula.

For future suggestions on how to control the temperature distribution, adjusting the cooling gallery location on the crown has a good effect on the temperature distribution. Another good suggestion is to change the combustion chamber design. Also for further research, the mechanical loads can be studied to investigate stress analysis along with thermal analysis.

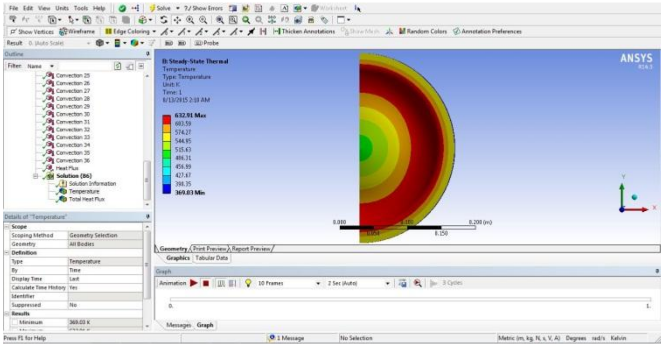

Figure 5: Thermal distribution on piston top surface

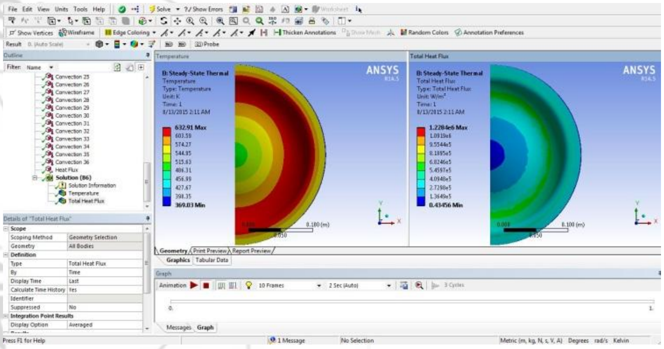

Figure 6: Temperature field distribution on piston top surface with total heat flux on the right

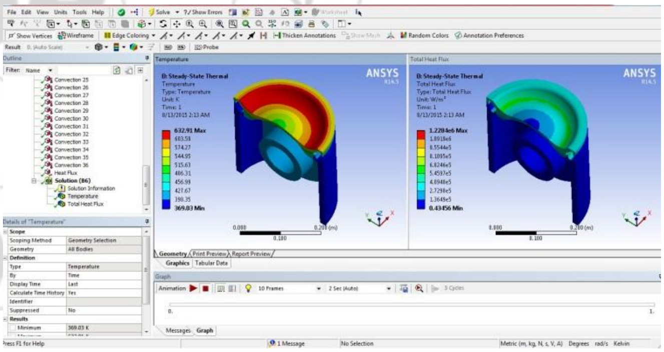

Figure 7: Temperature field distribution on piston body with total heat flux on the right

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