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Finite Element Analysis of Combustion Chamber of A four Stroke Diesel Engine

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ABSTRACT

The world diesel engine market share in 2010 was almost 46% of petro fuel at $142.5 billion. World demand of diesel is 6.7% annually through 2015 to $197.5 billion [1]. The diesel engine has the highest thermal efficiency among any reciprocating engines. This is on account of the very high compression ratio (22 to 26:1), due to which it can have a thermal efficiency exceeding 50%. CI engines are indispensible in the transport sector, agriculture machineries and marine engines. The performance enhancement of CI engines through efficiency improvement is the call of the day. One of the most effective ways of improving thermal efficiency of CI engine is to enhance the compression ratio. A higher compression ratio can be achieved only through a more robust and sturdy engine structure, as it will induce more stress requiring stronger materials and better design. This work therefore, focuses on the Better thermal efficiency, through improved design of combustion chamber. Finite Element Method (FEM) has been a facilitating tool.

KEYWORDS : combustion chamber, piston, FEM, isotherms, stresses.

1.1 INTRODUCTION OF THE PROBLEM

Compression ignition engines are the major source of mechanical power, contributing to about 8% of world energy consumption. For optimum design of an I.C. Engine, the knowledge of heat transfer rate from the working gases to the engine is important to obtain the points of highest thermal stresses. Exact analysis of the heat transfer is complex due to variations in temperature, pressure and the irregular shapes involved. Also, the engine shouldn't lose a large amount of heat in cooling [2,3]. Accurate heat flow and thermal deformation analyses would help in guarding against excessive stresses, fatigue, corrosion, etc.

Although the temperature of hot gases (T_g) and the coefficient of heat transfer (h_g) between gases and piston top surfaces vary cyclically, the problem is simplified by giving them constant values [4]. The temperature of inlet cooling water and air at starting condition is maintained at 20°C and 25°C respectively.

The temperature of piston can be estimated in different ways like direct measurements using thermocouples involving drilling of narrow holes for passing the thermocouples wires [5]. The other method is numerical method, which consists of finite difference method, and finite element analysis method. Here the analysis has been done by using finite FEM. Using the governing equation and the appropriate boundary conditions; the mathematical variation statement of the problem is obtained.

1.2 STATEMENT OF THE PROBLEMS

For achieving a robust structure of CI engine we are concerned with temperature distribution and thermal stress analysis in the piston [6,7]. The finite element approach is applied for the analysis. The piston selected for the above objective is having a diameter of 195 mm. Although temperature of hot gases (Tg) and coefficient of heat transfer (h_g) between hot gases and piston top surface very cyclically, but the problem has been simplified by giving them constant values.

The instantaneous values of heat transfer coefficient H_g on the gases face at crank angle (φ) is given by Eichelberg [8] as:

$$H_g(\varphi)=2.44(S)^{1/3}[P_g(\varphi)T_g(\varphi)]^{0.5}\ \mathrm{W/m^2\,K}$$

Where,

S= the mean piston speed m/s
$P_g(\varphi)$= gas pressure in N/m²
$T_g(\varphi)$= gas temperature in °C
Φ= angle covered

Where φ denotes the crank angle measured from TDC position as '0'.

The mean heat transfer coefficient over the cycle is given by:

$$H_{gm}= (1/\varphi_0) \int H_g(\varphi)d\varphi$$

The resultant gas temperature is given by:

$$T_{gr}= (1/h_{gm})\int H_g(\varphi)T_g(\varphi)d\varphi$$

The value of the H_{gm} and T_{gm} are function of the break mean effective pressure and the load on the engine.

The constant heat transfer coefficient between the piston and ring is given as:

h_{C1}= 35000 W/m²K, h_{C2}=20 W/m²K, h_{C3} = 5810 W/m²K, h_{C4} = 5810 W/m²K, h_{C5} = 1745 W/m²K

The present analysis has been carried out at four different loading given as:

(i) T_g = 1000°c, h_g= 290 W/m²K (ii) T_g = 800°c, h_g= 262 W/m²K

(iii) T_g= 600°c, h_g= 232 W/m²K (iv) T_g= 400°c, h_g= 230 W/m²K

1.3 METHOD OF SOLUTION

The problem is solved by Finite Element Analysis method. For solution of the problem, the piston along with the rings and wall has been divided into 196 tri-angular elements having 164 nodes as given in fig.1.3.1. The breakup of the elements and nodes for different part is as given below:

Part	**Element**		**Nodes**	
	From	To	From	To
Piston body	1	151	1	107
Piston ring	152	161	108	127
Cylinder wall	162	196	128	164

The construction of finite element approach starts from the variation statement of the problem and then using proper shape function a number of algebraic equations are developed which equal the number of nodal elements in the problem domain. The necessary boundary conditions imposed are convective on all the three sides' i. e. air, water and gas side of the piston and the contact boundary condition between piston ring clearances. Then the principle of minimization of variational integral is carried out to get a set of simultaneous equations. A FORTRAN program code is developed to solve these equations in order to find the unknown parameters i. e. temperature at different nodal points of the piston.

Heat balance of the problem is checked for accuracy and satisfied with result by observing that the amount of heat supplied at gas side of the piston is equal to the amount of heat loss to both water and air side of the piston. Isotherms at four different loads are plotted in order to study the thermal analysis i.e. how the temperature is distributed in piston cross-section.

Using these temperature fields, first the thermal stresses are calculated at the centre of gravity of the element. From this, the principal stresses and the stresses acting at the each nodal point are calculated. These stresses obtained above are then analyzed at three different cross-section of the piston body.

1.4 PROPERTIES OF MATERIALS

Properties of the materials	Aluminium	Cast iron	Cast steel
Density (Kg/m^3)	2800	7200	7850
Thermal conductivity (W/mK)	175.0	54.0	45.0
Modulus of elasticity(GPa)	72	95	200
Coefficient of Thermal Expansion(/°C)	23x10^{-6}	12x10^{-6}	12 x 10^{-6}
Poission ratio	0.33	0.25	0.30
Specific heat(J/Kg K)	920	586	460

1.5 INFERENCES FROM FEM ANALYSIS

Isotherms and deformations of various node have been plotted for different loads, viz.- (i) 100% of capacity(L_1), (ii) 75% of the capacity(L_2), (iii) 50% of capacity(L_3), and (iv) 25% of capacity.

The FEM analyses facilitate isotherms for piston and the resulting thermal stresses at different nodes for various loads.

(i) The isotherms show an upward trend towards the periphery of piston due to faster cooling as given in fig.4.1.1.

(ii) As the load decreases the thermal gradient reduces due to lower heat energy input as given in fig.4.1.2.

(iii) The deformation causes proportional stress when the piston is confined by the cylinder whose boundary also experiences deformation. Thus the stress caused is proportional to the relative confinement as given in fig.4.1.3. This is computed in the program using generalized Hooke's law.

(iv) It is found that as the load on engine decreases, the radial deformation reduces, as given in fig.4.1.4.

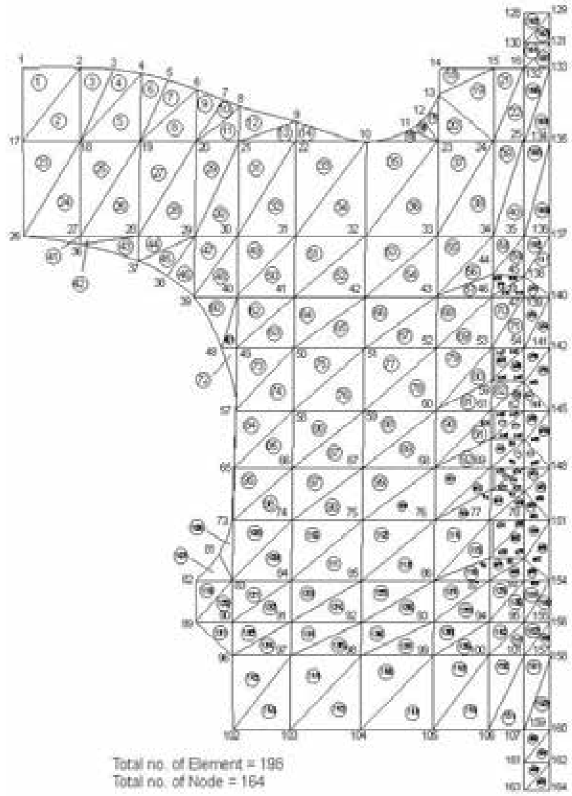

Fig 1.3.1 Piston showing the Node Number and Element Number

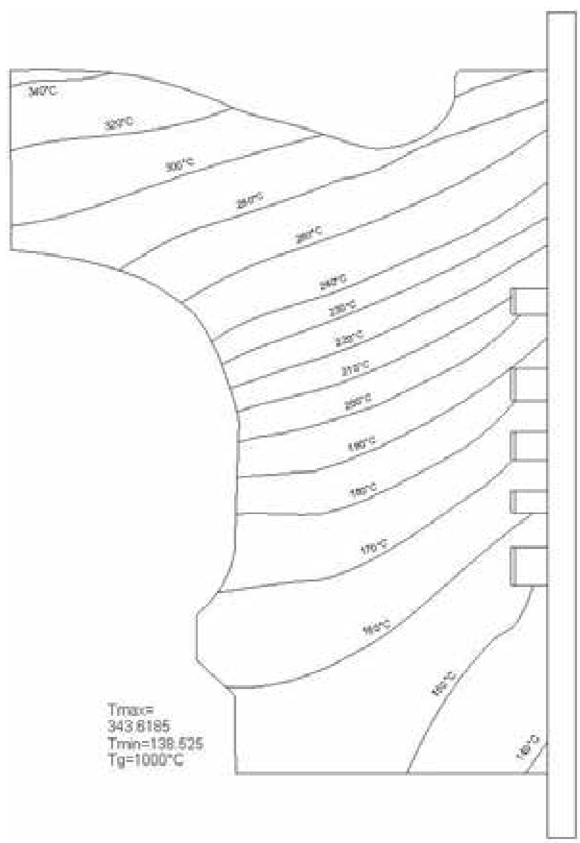

Fig 1.5.1 Isotherm Distribution in the Piston body under first load

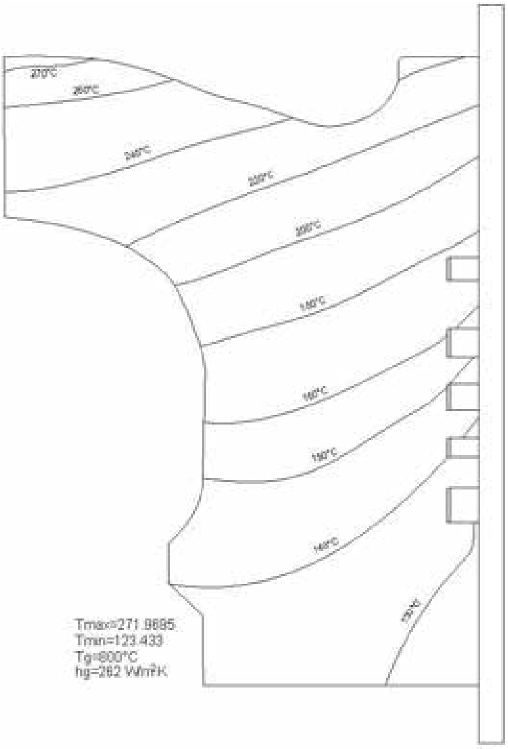

Fig 1.5.2 Isotherm Distribution in the Piston body under second load

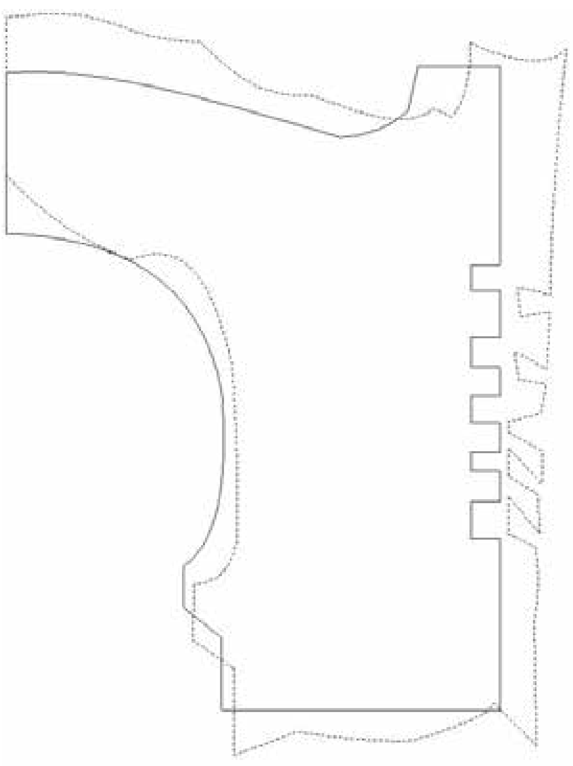

Fig.1.5.3 Deformed shape of the Piston body under first load

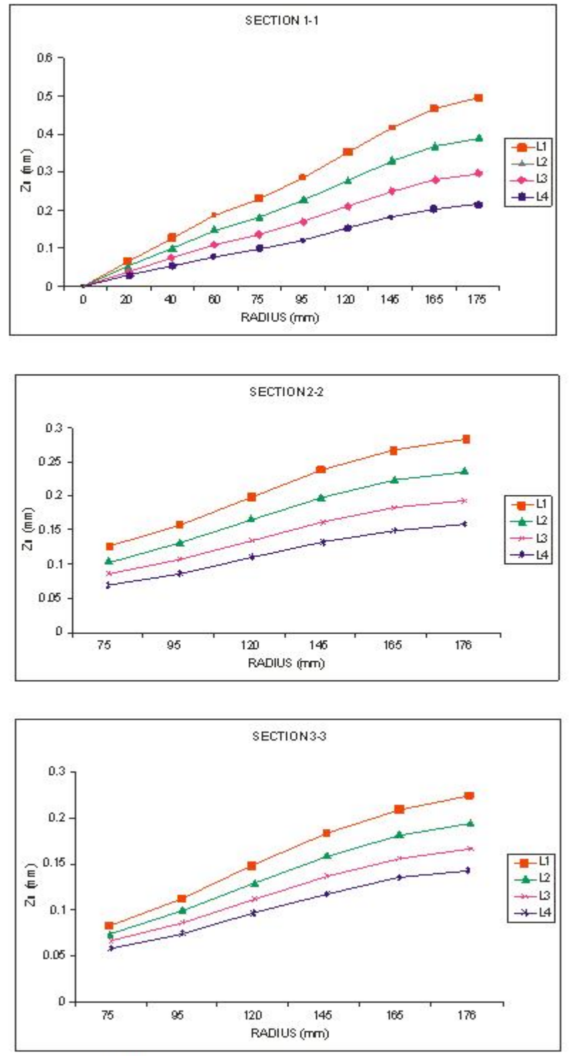

Fig.1.5.4 Radial displacement (Zu) vs Radius

1.6 CONCLUSION

This paper lists the conclusions drawn from the FEM analysis. The analytical method for prediction of temperature and heat balance is used to determine the thermal stress at different nodal points on piston body. The non-uniform temperature variation between the node at the face of the piston to the node at the center of piston increases with time. Thus it increases the probability of failure of piston.

The isothermic plots show that the temperature decreases in the radial direction at any cross section away from the piston center. Also temperature decreases from top face to the bottom face of the piston. This is to be expected as heat must flow from hottest side i.e. piston crown towards water and airsides, which are relatively at lower temperature. From the isotherms, it is clear that the temperature is highest at center point of the top of the piston. So it is necessary to provide some cooling arrangement either by insulating the piston or by circulating coolant in side cylinder.

1.7 SCOPE FOR FUTURE WORK

1. Now a day the designers are interested to make the engine adiabatic by insulating various parts like cylinder wall, valve and piston surface etc. Thus this analysis can be further extended to study heat flow pattern and thermal stress behavior inadiabatic engine.

2. Problems can be considered tor radiative heat exchange in combination with convective heat exchange in case of air-cooled engine.

3. The program developed in this thesis work can be used to analyze transient case also.

4. The friction occurring between surface of piston ring and cylinder wall is being neglected here but its effect can be incorporated to solve the problem more realistically.

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