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Thermo–Mechanical Analysis of AV1 Diesel Engine Piston using FEM

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ABSTRACT

The heat transfer processes in diesel engine piston can be modeled with a variety of methods. In this paper, an inverse heat transfer method is employed to conduct thermo-mechanical FE analysis. Considering the thermal boundary condition for numerical simulation; thermal stresses, mechanical stresses and the distortion of the piston have been calculated at various section in the piston body under the thermal loading thus providing reference for design improvement. Contours of displacement of nodal points and stresses introduce were shown as well. Results show that, the main cause of the piston safety, piston deformation and the great stress is the temperature, so it is possible to further decline the piston temperature with structure optimization. Measuring the stresses in different parts of the cylinder head and of the piston, we can adjust the cooling, or we can improve the materials or even we can improve the properties of the fuels.

Keywords - FE analysis, diesel engine piston, thermal expansion coefficient, thermal analysis

1. INTRODUCTION

In the combustion chamber engines, some of the parts like cylinder head, cylinder liner, piston and valve are most thermal loaded parts because they are in direct contact with the flem. Due to this they losses their strength and slightly deform from its original state. So it becomes important to calculate the piston stress distribution in order to control the deformations within acceptable levels. The stress distribution enables the designer to optimize the thermal aspects of the piston design at lower cost, prior to the first prototype is constructed. The mechanical stress and thermal stress level depends on the distribution of temperature in the parts, thermal expansion coefficient, young modulus of elasticity, thermal load, design of the parts and cooling conditions.

Most of the internal combustion (IC) engine pistons are made of aluminium alloy which has a thermal expansion coefficient and young modulus of elasticity much higher than the cylinder bore material made of cast iron. This leads to some differences between working and the design clearances. Therefore, thermo-mechanical analysis of the piston is extremely crucial in designing more efficient engines.

2. LITERATURE REVIEW

During working process, the piston affected by the high-pressure gas pressure, the inertia force caused by high-speed reciprocating motions, friction forces and effect of the thermal load caused by high-temperature gases. Although the thermal load and mechanical load are two kinds of different loads acting on the piston, they will both affect the reliability and endurance of the operation of the piston. Deformation occurs to the piston under the effect of the thermal load and the piston deformation will affect the transfer of heat, the thermal stress and the mechanical stress, so it is necessary to integrate the dual function of the thermal stress and the mechanical stress of the piston to carry out coupling analysis and solve so as to better reflect the stress field distribution and deformation condition of the piston in the operation condition. For better functioning the highest temperature of any point on piston should not exceed 65% of the melting point temperature of the alloy. According to Sharma, S.K. et al. (2013) The limiting temperature for the engine piston alloy is about 370 oC. This temperature level can be increased in ceramic coating diesel engines piston.

Most of the early efforts were focused on empirical analysis due to the limitation of the means of research (Borman, G. et al. (1987), Woschni, G. A. (1967)). Since the beginning of the 1980s, numerical approaches have been used to analyze the temperature distribution and to evaluate the thermo-elastic behaviour of engine pistons. Prasad, R. et al. (1991) formulated the set of equations with the help of isotherms and based on finite difference approximation of an aluminium alloy piston and valves of a diesel engine revels that temperature increases with insulation coating and hence the thermal stresses developed also increases. Li, C. H. (1986) used a three-dimensional finite element model of an aluminum diesel engine piston to calculate operating temperatures. He showed that skirt contours played an important part in the reduction of scuffing and friction. Rakopoulos and Mavropoulos (1996) used a piston model for the calculation of the temperature field and heat flow field under steady and transient engine operating conditions. Three-dimensional finite-element analyses were implemented for the representation of the complex geometry metal components and found a satisfactory degree of agreement between theoretical predictions and experimental measurements.

Liu and Reitz (1998) developed an axis symmetric transient heat conduction model to predict the combustion chamber wall temperatures. Jenkin et al. (1998) studied the near-wall temperature field in the burned and unburned gases, incorporating a turbulence model into the engine cycle simulation. Bin Zhao (2102) investigated temperature and thermal stress field of the ceramic coated piston of diesel engine by using the wavelet finite–element method and ANSYS software. It was observed by Cerit, M. (2011) that the temperature of coating surface was increased with increases in the thickness in a decreasing rate. During thermal cycling from room temperature to 1150 oC, the thermal conductivity and diffusivity of TBC coating increase. As may be seen from the literature, most of the studies have been concerned with thermal modelling of piston. The present paper is concerned with thermo-mechanical FE analysis to analyze the temperature and stresses under the effect of the thermal and mechanical load respectively.

3. METHODOLOGY

A geometrical model of the piston was developed based on dimensions obtained from of the actual object. A final CAD model is presented in Fig. 1. In this model, a certain geometric simplification was assumed – including missed bends with a small radius on the edge of crown and lateral surface of the head of the piston. Finally, the geometrical model was discretized into 3-node triangular finite elements. Such elements had to be applied due to a complex shape of the piston and the lack of the axial symmetry. The size of finite elements was different in respective sections of the piston – larger elements were employed for the piston crown and skirt, whereas the smaller ones were used close to the oil channels. Meshing of the piston, were built with 203 solid element having 155 nodes in such a way that it can tolerate irregular shapes without losing much of accuracy.

Moreover, surface to surface contact elements were defined between the piston ring and ring grove. Fig. 1 shows the FE discrete model of the piston. Uniform shapes and forms of elements play an important role to obtain precise results. Therefore, Steady state thermal and structural analyses were carried out to investigate the temperature, radial thermal stresses and mechanical stresses in the diesel engine pistons at two section line 1-1 and 2-2 as shown in Figure 1 and distortion in the piston body surface. 3D thermo mechanical FE analyses were performed by using self generating computational code which was based on the finite element methods (FEM). The specifications of the test engine are tabulated in Table 1. Piston and piston rings are made of AlSi alloy and cast iron respectively. These materials were assumed to be linear elastic and isotropic. Thermo mechanical material properties of these materials are tabulated in Table 2.

Fig.1 FE discrete model, actual and 2-D cut view of the piston model

*Table 1 Engine and their **specification***

Specification	Type	Specification	Type
Cooling	Water-Cooled Engine	Governing	Class"B1"
Model	AV1	Power rating	5 hp
No. of Cylinders	1	Fuel injection	Direct Injection
Cubic Capacity (ltr)	0.553	Rated Speed (rpm)	1500
Overall Dimensions of the standard engine		617 X 504 X 843 (L X B X H)	

Table 2 Thermo physical properties of metal

Sr.No.	Properties	Units	Aluminium	Cast iron	Steel
1	Thermal conductivity	$Wm^{-1} k^{-1}$	175	70	50
2	Density	$Kg\ m^{-3}$	2700	7200	7850
3	Thermal diffusivity	$m^2\ hr^{-1}$	0.259	0.04563	0.044
4	Specific Heat	$W\ Kg^{-1}\ K^{-1}$	0.248	0.16	0.131

Table 3 Heat transfer parameter for four different cases of engine loading

Parameter	Case 4 (Full Load)	Case 3 (3/4 Load)	Case 2 (Half Load)	Case 1 (No Load)
Tg (Combustion side) ^oC	1000	800	600	400
Hg (Combustion side) w/m^2k	290.5	232.4	174.3	116.2

The adopted heat transfer coefficient on the contact surfaces are-
H_a (heat transfer coefficient at piston under crown surface) =174.3 w/m^2k , H_1 (heat transfer coefficient at Ring lands and Piston skirt upper and lower side) =2905.4 w/m^2k, H_2 (heat transfer coefficient at Ring lands and Piston skirt contact surfaces) = 20 w/m^2k, H_3 (heat transfer coefficient between piston rings and cylinder wall contact surfaces) = 38346 w/m^2k, H_4 (heat transfer coefficient between piston and cylinder wall contact surfaces) = 2324 w/m^2k, H_w (heat transfer coefficient through cylinder wall to water) = 1859.2 w/m^2k and temperature on water side (T_w) was 120 ^oC and on crank case side (T_a) was 80 ^oC. Heat transfer parameter for four different cases of engine loading is represented in Table 3.

The analysis presented in this paper is divided into two sections, the temperature field distribution and the thermal and mechanical stresses distribution. The finite element technique with triangular element is used to reduce the variational formulation to a set of algebraic equations. The expressions to calculate nodal temperatures and the corresponding stresses at every element are derived. The construction of finite element approach starts from the variational statement of the problem and then using proper shape function a number of algebraic equations are developed which equal to the number of nodal elements in the problem domain. Then by minimizing the approximate function a set of governing equation is developed for whole of the piston body. Computer algorithm and a FORTRAN program code are developed to solve these equations in order to find the unknown parameters i.e. temperature, distortion and stresses at different nodal points of the piston. Computer program is based on heat transfer through conduction, convection, matrix multiplication, matrix inversion, heat flow and stiffness. By using these subroutine and main program, temperatures and heat flow field were calculated.

4. RESULTS

FE analyses were carried out using fortran program in two stages. In the first stage, the piston FE model was heated from the initial temperature of 25°C to the maximum temperature resulted from the thermal boundary conditions. The maximum temperature and minimum temperature reaches up to 261.11 °C and 243.44 °C respectively at section 1-1 and 262.89 °C and 209.42 °C respectively at section 2-2 at full loading condition. Temperature range at section 1-1 and 2-2 at all loading condition are presented in Table 4. Selected results of the temperature analysis in the form of contour band are presented for section line 1-1 in Fig. 2 and for section line 2-2 in Fig. 3. From Fig. 2 and Fig. 3, it shows that the temperature is maximum at the inner side because that nodal points are directly exposed to the combustion gasses as well as the radius increase the temperature fall down. Corresponding thermal and mechanical stresses are found in the piston body due to the temperature variations from inner to outer radius. These radial thermal stresses and mechanical stress for section line 1-1 and 2-2 are shown in Table 4 and Table 5. Selected results of the radial thermal stresses analysis and mechanical stresses analysis in the form of contour band are presented in Fig. 4 - Fig. 7 for section line 1-1 and section line 2-2 respectively. After analysis it shows that the mechanical stress is very less as compared to radial thermal stress, which is shown in Fig. 6 and Fig. 7. so the main cause of failure of the piston is thermal loading. Each figure shows a sharp variation in magnitude and direction. It's because every element tends to maintain the constant strain property. If a node expends for a certain element then it must compress for another element. Thus undergoes a simultaneous expansion and compression for different and vice-versa. However by considering a very large number of elements these sharp variations in stress can be eliminated and a more continuous curve can be obtained.

Table 4 Temperature, Radial thermal stress and Mechanical stress at Section 1-1

	Temperature at Section 1-1				Radial thermal stress at Section 1-1				Mechanical stress at Section1-1			
Radius (mm)	Full load	3/4 load	Half load	No load	Full load	3/4 load	Half load	No load	Full load	3/4 load	Half load	No load
20	261.11	208.77	166.64	136.09	835.91	739.78	670.88	597.03	25.61	21.54	15.44	10.78
25	255.83	205.42	164.86	135.44	955.83	537.16	551.69	407.55	24.59	19.31	14.77	9.66
30	250.76	202.22	163.14	134.81	970.19	746.74	497.38	354.26	8.28	6.06	5.01	3.03
32.5	248.60	200.85	162.43	134.56	594.38	637.54	384.72	278.46	6.46	5.36	4.51	2.68
36.5	245.83	199.12	161.53	134.28	739.90	623.95	473.53	231.98	0.89	1.68	1.41	0.84
40	243.44	197.63	160.78	134.06	865.21	630.34	611.35	332.59	-4.80	-2.77	-2.27	-1.39

Table 5 Temperature, Radial thermal stress and Mechanical stress at Section 2-2

	Temperature at Section 2-2				Radial thermal stress at Section 2-2				Mechanical stress at Section 2-2			
Radius (mm)	Full load	3/4 load	Half load	No load	Full load	3/4 load	Half load	No load	Full load	3/4 load	Half load	No load
0	262.89	209.55	166.62	135.48	465.55	563.61	664.58	286.64	-7.29	-6.01	-3.69	-3.00
5	261.24	208.51	166.07	135.29	772.98	689.42	439.65	383	5.87	5.07	4.09	2.53
10	257.97	206.44	164.98	134.90	986.84	763.56	588.38	458.84	3.16	2.78	1.60	1.38
15	253.72	203.76	163.55	134.39	756.57	716.61	383.10	394.37	-2.5	-2.24	-1.49	-1.12
20	248.98	200.76	161.96	133.81	647.11	694.51	333.58	213.27	-7.06	-5.17	-3.99	-2.58
25	243.22	197.11	160.00	133.09	324.28	385.69	499.55	343.15	-11.6	-8.54	-6.16	-4.27
32.5	215.33	179.30	150.31	129.29	771.49	481.27	427.59	362.26	4.49	4.52	2.79	2.26
36.5	212.27	177.4	149.33	128.99	545.69	598.77	378.32	403.13	-2.4	-1.91	-1.05	-0.96
40	209.42	175.62	148.42	128.71	644.98	569.32	409.6	236.62	-0.03	-0.09	-0.28	-0.04

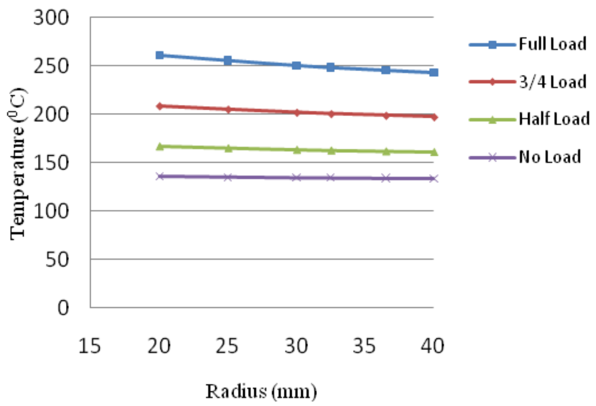

Fig. 2 Temperature distribution at section 1-1

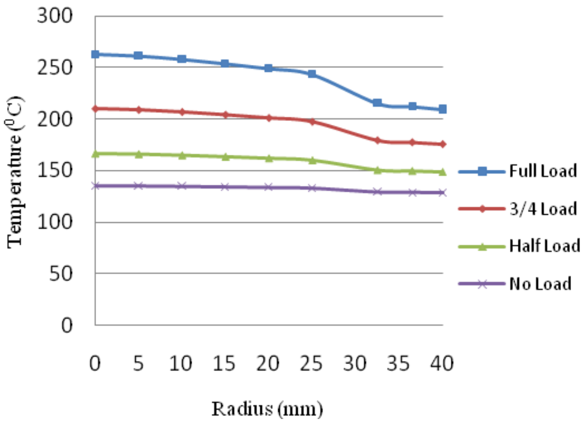

Fig. 3 Temperature distribution at section 2-2

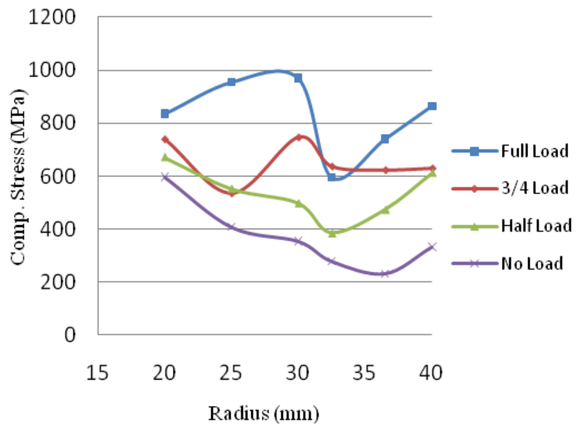

Fig. 4 Radial thermal stress at section 1-1

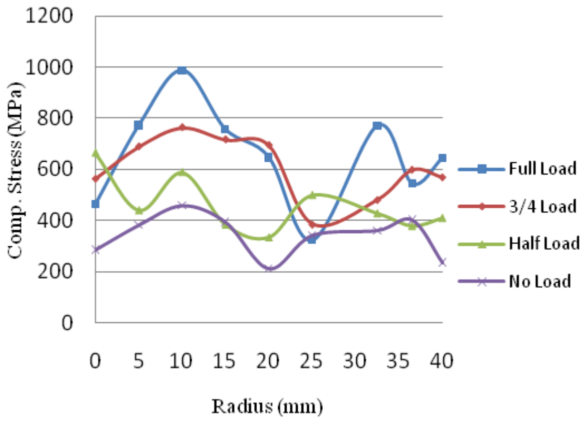

Fig. 5 Radial thermal stress at section 2-2

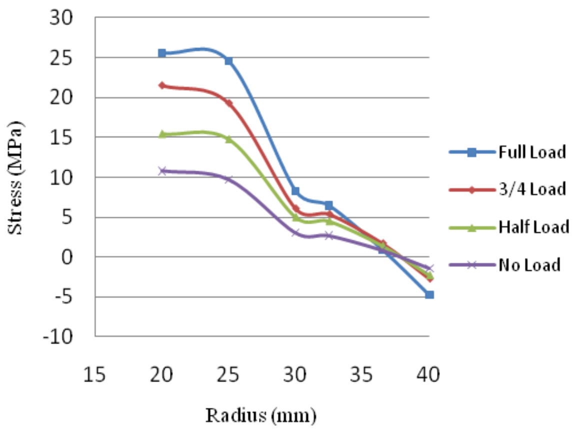

Fig. 6 Mechanical stress at section 1-1

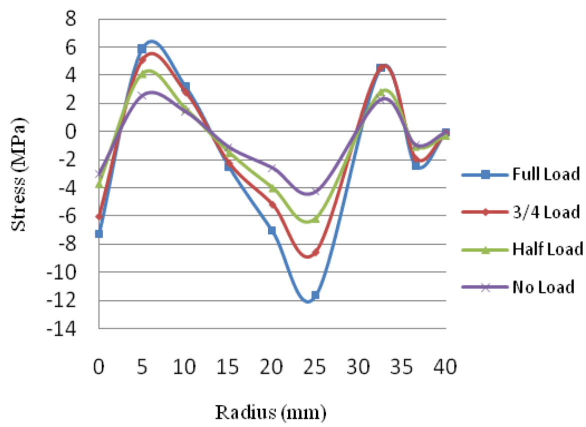

Fig. 7 Mechanical stress at section 2-2

5. CONCLUSION

The purpose was to compare behavior of the piston under thermal load and mechanical load. The obtained results shows that the thermal stresses induced in the piston body are much more as compare to mechanical stresses. Through the analysis, it is concluded that the main factor influencing the piston intensity is the temperature, thus providing basis for the optimization design of the piston. The deformation and the stress of the piston are mainly determined by the temperature, so it is necessary to decrease the piston temperature through structure improvement.

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