Abstract

Current research in IC engines mainly focus on various methods to achieve higher efficiency and high specific power. As a single design parameter, combustion chamber peak ring pressure has increased more than before. Apart from the structural aspects of withstanding these loads, designers face challenges of resolving thermal aspects of cylinder head. Methods to enhance heat transfer without compromising load withstanding capability are being constantly explored. Conventional cylinder heads have got at inner surface. In this paper we have suggested a modification in inner surface to enhance the heat transfer capability. To increase the heat transfer rate, inner flame deck surface is configured as a curved and stepped surface instead of at. We have reported the effectiveness of extend of curvature in the inner flame deck surface in a different technical paper. Here, we are making a direct comparison between stepped and curved surface only. From this analysis it has been observed that curved surface reduces the flame deck temperature considerably without compromising the structural strength factors compared to step and at surface.

I. Introduction

In present scenario, competitive automotive segment is in demand for higher power output and the demand is yet growing stronger. From the future engine trends [2], it is observed that specific power of the engine is likely to grow from 50 kW/l to 65 kW/l for CI engines and corresponding operating pressure and temperature also increases from 160 bar - 200 bar and temperature from 260°C to 280°C at the bridge between valve ports. In IC engines, cylinder head encounters end to end effect of combustion cycle i.e. from intake of combustion gas to exhaust of combustion products. It comprises of passage for intake, exhaust, coolant flow, valve and fuel injection arrangements. Since it acts as the load path, cylinder heads experience high operating temperature. Pulsating high peak pressure and temperature loads demand a rugged heavy cylinder head, whereas high specific power (kW/kg) needs a lightweight design. This conflicting technical requirement stresses the designer with limited options in terms of material, space, coolant flow design. Cast iron in form of lamellar or Compacted Graphite Iron (CGI) are the most commonly used cylinder head materials. As material property is a function of body temperature. Hence, controlling the body temperature ensures longer service life of the component. The possibility of enhancing the heat transfer rate over the flame deck is studied by varying the thickness and shape of the inner flame deck surface.

The operational temperature range of cast iron cylinder heads could be specified up to approximately 420°C, 450°C, and 480°C for gray cast iron, compacted graphite iron, and spheroidal graphite iron. When operated at higher temperatures crack is likely to occur at nose bridge and ultimately lead to fatigue damage. Thus, effective coolant flow to maintain low body temperature, methods of transferring heat from combustion chamber reducing the thermal loads are explored by engine designers.

This paper reports the effect of introducing a curved and stepped inner surface to enhance heat transfer rate of the cylinder head and thereby reducing the body temperature. Thus, FEM is considered as starting step in development of new product and experimenting with new modifications. A hypothetical high performance engine cylinder head with step and curved surface is designed to study the effect of inner profile on heat reduction and thus, the best suited configuration is reported.
Basic 3D model is created in CREO 2.0 and is shown in fig.1. Tetrahedral elements were used to mesh the model and it took 1950663 nodes and 10154632 elements to arrive at a considerable mesh quality. The material properties considered for analysis is given in table 1. CFD analysis is carried out as per the standard analysis method using ANSYS FLUENT. Coolant flow in the coolant passage was assumed to be 3D flow, steady state, incompressible turbulent flow, and the viscosity in the near wall region was also taken into account. CFD analysis was carried out using coolant at a constant inlet temperature of 120°C. Body Temperature is obtained as output from CFD. For ThermoMechanical analysis, load due to the combustion gas peak pressure is applied on the flame deck surface considering the body temperature into account.

II. BOUNDARY CONDITIONS

Energy and turbulence model was considered for CFD analysis. Cast Iron is considered as Solid body and Water as the coolant domain. 7 m/s water inlet velocity and 393°C K is taken as input parameter for water. At outlet 1bar abs. pressure is taken as boundary condition. Simplex algorithm was used as solution method and first order up wind scheme was used. Boundary surfaces are shown in fig.2.
Table 2: HEAT TRANSFER COEFFICIENTS

| PART (COMBUSTION GAS CONTACT) | TEMPERATURE OF COMBUSTION PRODUCT (K) | HEAT TRANSFER COEFFICIENT (W/m²K) |
|-------------------------------|---------------------------------------|----------------------------------|
| FLAME FACE                    | 1262                                  | 858                              |
| EXHAUST PORT                  | 1024                                  | 530                              |
| INTAKE PORT                   | 379                                   | 230                              |
| OIL SURFACE                   | 425                                   | 200                              |

III. Heat Transfer Coefficient

Heat transfer coefficient is not a material property and depends on a number of factors such as the material conductivity, velocity of the fluid, temperature of the fluid, turbulence, surface finish etc. These complications have made it difficult to find the value of heat transfer coefficient (HTC) by experimental methods. Woschni empirical equation is used for calculation of HTC. Input values for this equation come from basic system specification. Value of the heat transfer coefficient can be found from Nusselt number - Reynolds number type correlation. Woschni model is preferred over other model and the assumptions made are given below:
1. Universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine.
2. Calculates the amount of heat transferred to and from the charge.
3. It is the most commonly used heat transfer model and applied to all cylinder elements.

Assumptions:
1. On all surfaces of the cylinder heat transfer coefficient and velocity of charge is uniform.
2. This model accounts the increase in gas velocity in the cylinder during combustion.
3. Woschni’s proposal that the average gas velocity should be proportional to the mean piston speed.
4. Air is the working medium.
5. Heat transfer by conduction through the walls is one-dimensional
Thermo dynamic simulation model is also considered for the HTC calculation and the values obtained are given in Table 2.

These heat transfer coefficient are validated using mathematical calculation based upon Woschni equation [2]. The equation is given as below:

\[ h = 0.820 \ast D^{0.2} \ast P^{0.8} \ast W^{0.8} \ast T^{-0.53} \] (1)

where,

\[ h \] = Heat Transfer Coefficient
\[ D \] = Bore dia in m
\[ L \] = Stroke in m
\[ N \] = Speed in rpm
\[ P \] = Combustion Pressure in MPa
\[ P_0 \] = Motor Pressure
\[ W \] = Characteristic Speed in mps
\[ W = C_1 \cdot S_p + C_2 \cdot V_s \cdot \frac{T_1 \cdot (P - P_0)}{(P_1 \cdot V_1)} \]  \hspace{1cm} (4)

where,

\[ C_1 = 6.18; \quad C_2 = 0; \]

\[ C_1 = 2.28; \quad C_2 = 0; \]

\[ C_1 = 2.28; \quad C_2 = 0.00324 \cdot P_0; \]

\[ S_p = \text{Mean piston Speed} \ [\text{m/s}]; \]

\[ T_g = \text{Average gas temperature} \ [\text{K}]; \]

\[ P_t = \text{Pressure at start of combustion in MPa} \]

\[ V_t = \text{Volume at start of combustion in m}^3 \]

\[ T_i = \text{Temperature at start of combustion in m} \]

\[ V_s = \text{Swept volume of cylinder in m}^3 \]

The heat transfer coefficient on the gas side, \( h \cdot V_s \cdot \theta \) for the engine can be represented by the Fig 3. Average of the heat transfer coefficient calculated is found matching with values obtained through thermodynamics simulation. Average heat transfer coefficient considered on the gas side is 800.87 W/m²K.

![Figure 3](image)

**IV. Steady State Thermal Analysis**

In IC engines heat from the combustion fluctuates periodically. This high rpm makes the temperature fluctuations to penetrate about a millimeter. Depth of heat penetration is governed by the Thermal diffusivity \( \alpha \). Considering rated speed engine as 2600 rpm calculation for depth of heat penetration is shown in Fig 4.

\[ Engine \text{ rated speed} = 2600 \text{ rpm} \]

\[ \text{corresponding time scale} = 0.023 \text{ sec} \]

\[ \text{Thermal diffusivity of cast iron } \alpha = 1 \times 10^{-5} \text{m}^2/\text{sec} \]

\[ \text{Penetration Depth } X = \sqrt{\alpha \cdot t} \]  \hspace{1cm} (6)

\[ X = 4.87 \times 10^{-4} \text{m} \]

Since the depth of heat penetration is small it is assumed that the temperature profile does not change with reference to time. Hence steady state thermal analysis is carried out. The heat distribution is shown Table 3. Internal heat distribution is shown for each case.
One of the essential aspects of CFD simulation study is to check the grid sensitivity. The effect of grid size on the results has been studied for case-2 to prove the grid independent results. Number of grid has been reduced from 2751925 to 305030 as shown in Fig. 5. Velocity and Temperature were compared for both fine and coarse mesh. It can be inferred from the figures 6 & 7 that there is no significant change in the values of velocity, temperature, which shows that the results are not sensitive to the grid.

CFD and thermo mechanical analysis was carried out for three different configurations. Coolant inlet velocity is taken as 7m/sec. Boundary conditions are maintained same for all the three cases. From the structural analysis it is found to withstand the peak firing pressure. Summary of CFD results for each case; velocity profile and temperature an stress distribution is given in Table 3. Configuration 3 performs better compared to other two cases in terms of low body temperature and stress distribution.
Figure 7: TEMPERATURE PROFILE COMPARISON FOR FINE AND COARSE GRID

REFERENCES

[1] Prof. V. Ganesan, -INTERNAL COMBUSTION ENGINE

[2] Amit Paratwar, -SURFACE TEMPERATURE PREDICTION AND THERMAL ANALYSIS OF CYLINDER HEAD OF DIESEL ENGINE. IJERA, Vol 3. Aug 2013, pp. 892-902.

[3] Stefan Trampert, -THERMO MECHANICAL FATIGUE LIFE PREDICTION OF CYLINDER HEADS IN COMBUSTION ENGINES. INTEL., JOURNAL OF GAS TURBINE AND POWER, Vol 130. JAN 2008, 012806-1 TO 10.

[4] Dr.-Ing Franz J Massen, -LIGHT WEIGHT DIESEL COMPONENTS

[5] Metals Databook, Alok Nayar.

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### Table 3: Summary of CFD Results for 7 m/s Average Inlet Velocity

| Configuration | Stress [MPa] | Temperature [°C] | Velocity Profile [m/s] |
|---------------|--------------|------------------|------------------------|
|               | 2.4 MPa      | 190-200 °C       | Average Velocity: 0.125 m/s |
|               | 3.0 MPa      | 195-210 °C       | Average Velocity: 0.145 m/s |
|               | 3.5 MPa      | 307-324 °C       | Average Velocity: 0.165 m/s |