

Computational Modeling for Thermal Analysis of AV1 Diesel Engine Valve using FEM

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Abstract

Under steady-state, a thermal investigation has been taken to study operating temperatures and heat flow rate in the valves of an AV1 diesel engine. Temperatures, temperature fields, and heat flow rate were measured under all four thermal loading conditions (full, third-fourth, half, and no load) using FORCE-2 FE (finite element) software. Appropriate averaged thermal boundary conditions were set on different surfaces for the FE model. Results obtained in the engine valves revealed that in addition to heat transfer by convection and radiation from combustion gases, the temperature and heat flux distributions are considerably affected by heat conduction from the valve seat. Contours of temperature fields introduced were shown as well. Results show that the main cause of valve safety is valve deformation and great thermal stress. So it is feasible to further decrease the valve temperature with structure optimization. Measuring the temperature in different parts of the diesel engine, we can adjust the cooling, or we can improve the materials, or even we can improve the properties of the fuels. The FEA result provides effective theoretical evidence for further improving the valves' performance. The evaluation confirmed the significant variation previously observed between the various methods.

Keywords: Valve, Temperatures field, Heat flux, Temperature, Finite elements

1. Introduction

In the combustion chamber engines, some of the parts like cylinder head, cylinder liner, piston, and valve are the most thermal loaded parts because they are in direct contact with the flame. Due to this they lose their strength and slightly deform from their original state. So it becomes important to calculate the stress distribution to control the deformations within acceptable levels. The thermal stress and mechanical stress level depend on the distribution of temperature in the parts, thermal load, coefficient of thermal expansion, young modulus of elasticity, design of the parts, and cooling conditions. For the thermal design of internal combustion diesel engine valves, an accurate estimation of heat transfer for various locations

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is of vital importance. Heat transfer affects the performance, efficiency, and emissions, as well as the life of the engine components, such as pistons, rings, and valves. To control the thermal stresses and deformations within acceptable levels it is important to calculate the valve temperature distribution. Because of exposure to hot exhaust gases; the exhaust valve of an internal combustion engine is one of the most critical parts. The design of valves depends on many parameters, such as fluid dynamics of the exhaust gas, the fatigue strength of the valve material, oxidation characteristics of the valve material, exhaust gas behavior of the material at high temperature, the configuration of the cylinder head, the coolant flow, the shape of the exhaust port, etc. To obtain an optimum condition and accurate estimations of heat flow rates, the temperature distribution in valves is needed. Thermal analysis guides us to design and optimize the engine parts to estimate heat flow accurately and consequently prevents failure of the parts due to excessive stresses, fatigue, corrosion, etc. Numerous works have been carried out in the study of heat transfer correlation in internal combustion engines. Prasad and Samria (1990) have formulated a set of equations based on finite difference approximation of an aluminum alloy piston of a diesel engine. They have resulted with the help of isotherms, that with an increase in insulation coating, the temperature increases. E-Adu-Nada et al. (2006) introduced thermodynamic analysis of SI Engines by a theoretical model of air standard otto cycle by implementing temperature depending on specific heat. Singh, V.P. et al. (1986) proposed a model to study the heat transfer in the cylinder and piston assembly. Sakhrich, A. et al. (2010) suggested that early injection timings lead to a higher level of pressure and temperature in the cylinder and found out the optimum equivalence ratio for different engines. Sharma, S.K. et al. (2013) introduced an air-cavity in a diesel engine valve to prevent heat loss without affecting the thermal stress. By making these cavity temperatures at all the nodes were dropped down without doing any thermal barrier coating on it. Kandil, A. et al. (1995) presents a complete evaluation of temperature and stress distribution, in a non-steady state, is obtained by a numerical model. In this model, the temperature of the inside surface of the thick-walled cylinder is considered to change according to certain boundary conditions. Prasad et al. (1990) carried out to assess the transient thermal response of the inlet and exhaust valves of a tata Mercedes, six cylinders, 4 strokes, water-cooled diesel engine with numerical calculations based on finite difference approximation. A detailed analysis has been given for estimating the boundary conditions of the inlet and exhaust valves of an IC engine.

The specific objective of this research paper was to develop an enhanced method to predict the temperature field and heat transfer from engine valves. This method should be supported by a research methodology comprising the application of thermodynamic principles and the fundamental equations of heat transfer. The justification for undertaking this research on heat transfer in engines is based on the potential benefits of this kind of investigation.

2. Statement of the problem

This thermal investigation is concerned with the diesel engine valve of the AV1 diesel engine, as shown in [Figure 1]. The engine is a single-cylinder, water-cooled engine. The compression ratios and - consequently - the power and the torque, are different. Selected technical data for the AV1 engine is provided in [Table 1]. In [Figure 1] the major diameter of the valve stem is 8 mm, height is 110 mm and base diameter is 30 mm. valves are made of alloy steel. The thermophysical materials properties and heat transfer parameters for four different cases of engine loading of the valves are given in [Table 2] and [Table 3]. In [Table 3], T and H represent the temperature and heat transfer coefficient, and subscript g , s , ex , b ,

and f represent the boundaries of the valve on the gas side (combustion chamber), valve seat side, exhaust gas side, bush contact surface side and fresh airside. With the help of these boundary conditions, temperature distribution will be analyzed in the valve.

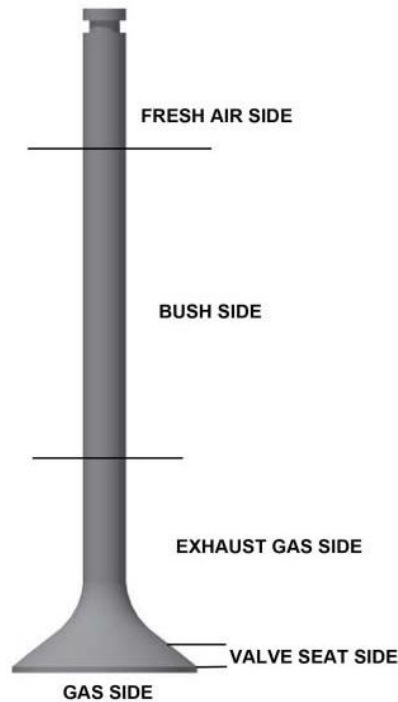


Figure 1. Diesel engine valve

3. Development of a geometrical FE model

A geometrical model of the valve was developed based on dimensions obtained from the actual object. In this model, a certain geometric simplification was assumed – including missed bends with a small radius on the edge of the lateral surface of the valve. Finally, the geometrical model was discretized into 3-node triangular finite elements. Such elements had to be applied due to the simple shape of the valve and good agreement with the axial symmetry. The size of finite elements was different in the respective sections of the valve.

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. Thermophysical properties of metal

S.N.	Properties	Units	valve material (Alloy Steel)
1	Thermal conductivity	W/m ² k	50
2	Density	Kg/m ³	7850
3	Thermal diffusivity	m ² /hr	0.044
4	Specific Heat	Kcal/kg ⁰ c	0.113

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

Parameter	Full Load	3/4 Load	Half Load	No Load
Temperature in °C				
Tg(Gas side)	1000	800	600	400
Ts(Valve Seat side)	300	120	120	120
Ta(Air side)	25	25	25	25
Tb(Bush Side)	80	80	80	80
Tex(Exhaust Gas side)	290	270	250	230
Heat transfer coefficients (W/m ² K)				
Hg(Gas side)	290	232.5	203.48	174.4
Hw(water side)	1859.2	1859.2	1859.2	1859.2
Ha(Air side)	23	23	23	23
Hb(Bush side)	1745	1745	1745	1745
Hex(Exhaust Gas side)	175	175	175	175

4. Thermal FE analysis

To determine the temperatures, temperature field, and heat flow field due to the temperature variation in the valve, Finite Element Analyses (FEA) were carried out using FORTRAN force 2.0 versions with self-generating computational code. For this paper, use was made of the force 2.0 program which contains various subroutine programs based on heat transfer through conduction, convection, matrix multiplication, matrix inversion, heat flow, and stiffness. By using these subroutine and main programs, temperatures and heat flow field were calculated. Mathematical modeling of conduction equation, convection equation, and contact heat transfer equation has been done which are shown below. In the theory of finite element study, first, the proper variation principle is selected and then the function involved is expressed in terms of approximate assumed displacements, which satisfies the given boundary conditions. Then by minimizing the approximate function a set of governing equations is developed.

4.1. Heat transfer equation for conduction

The generalized governing differential equation for heat conduction can be represented as [1, 4]:

$$\frac{\partial}{\partial x} \left[K_x \left(\frac{\partial T}{\partial x} \right) \right] + \frac{\partial}{\partial y} \left[K_y \left(\frac{\partial T}{\partial y} \right) \right] + \frac{\partial}{\partial z} \left[K_z \left(\frac{\partial T}{\partial z} \right) \right] + q_E = \rho C \frac{\partial T}{\partial \tau} \quad (1)$$

Where,
 K_X, K_Y, K_Z – Thermal conductivity in X, Y, and Z path respectively

q_E – Heat conduction per unit volume

ρ – Density of the material

C - Heat capacity of the material

For isotropic material, the thermal conductivity on all three directions are constant, i.e.

$K_X, K_Y, K_Z=K$

And in cylindrical coordinates;

$$\nabla^2 T = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \left(\frac{\partial T}{\partial r} \right) + \frac{1}{r} \left(\frac{\partial^2 T}{\partial \theta^2} \right) + \frac{\partial^2 T}{\partial z^2} \quad (2)$$

The variation integral in the axis-symmetric coordinate system can be represented as;

$$\chi_K^{(e)} = \frac{1}{2} (2\pi K) \iint_A \left[\left(\frac{\partial T}{\partial r} \right)^2 + \left(\frac{\partial T}{\partial z} \right)^2 \right] r dr dz \quad (3)$$

The Polynomial selected in terms of shape function is given by

$$t(r, z) = [N_I \quad N_J \quad N_K] \begin{Bmatrix} t_i \\ t_j \\ t_k \end{Bmatrix} = [N]^{(e)} \{t\}^{(e)} \quad (4)$$

Where N is the shape function. Differentiating the above equation w.r.t coordinates (r, z)

$$\frac{\partial t^{(e)}}{\partial r} = [b_i \quad b_j \quad b_k] \{t\}^{(e)} = [b]^{(e)} \{t\}^{(e)} \quad (5)$$

With the help of equation (5), the conductive matrix ($\chi_K^{(e)}$) becomes-

$$\frac{\partial \chi_K^{(e)}}{\partial \{t\}^{(e)}} = (2\pi K A^{(e)} r_c) \left[[b]^{(e)t} [b]^{(e)} + [c]^{(e)t} [c]^{(e)} \right] \{t\}^{(e)} \quad (6)$$

Where

$$\iint_A r dr dz = A^{(e)} r_c, \quad r_c = \frac{r_i + r_j + r_k}{3} \quad (7)$$

Let $A^{(e)} r_c = V^{(e)}$; where $V^{(e)}$ is the volume of an element.

$$\frac{\partial \chi_K^{(e)}}{\partial \{t\}^{(e)}} = \begin{bmatrix} k_{11} & k_{12} & k_{13} \\ k_{21} & k_{22} & k_{23} \\ k_{31} & k_{32} & k_{33} \end{bmatrix} \begin{Bmatrix} t_i \\ t_j \\ t_k \end{Bmatrix} = [k]^{(e)} \{t\}^{(e)} \quad (8)$$

Where $[k]^{(e)} =$ stiffness matrix

4.2. Heat transfer equation for contact boundary

The universal governing differential equation for contact boundary can be represented as:

$$q_c = K_1 \left[\frac{\partial T}{\partial n} \right]^1 = -K_2 \left[\frac{\partial T}{\partial n} \right]^2 \quad (9)$$

$$\text{Where, } q_c = h_c(T^1 - T^2) \quad (10)$$

Variation formulation for contact state line between 2 elements (e) and (p) can be written as-

$$\chi_{bcont.} = \frac{h_c}{2} \int_{s_i}^{s_j} [\{t\}^e - \{t\}^p]^2 2\pi r ds \quad (11)$$

$$\frac{\partial \chi_{bcont.}}{\partial \{t_s\}^e} = \frac{2\pi h_c r_m r_{ij}}{6 \cos \theta} \begin{bmatrix} 2 - \frac{\varepsilon}{2} & 1 \\ 1 & 2 + \frac{\varepsilon}{2} \end{bmatrix} \left\{ \begin{array}{l} \{\{t_s\}^e - \{t_s\}^p\}_1 \\ \{\{t_s\}^e - \{t_s\}^p\}_2 \end{array} \right\} \quad (12)$$

4.3. Heat transfer equation for convective boundary

The universal governing differential equation for heat convection can be represented as:

$$-K \left(\frac{\partial T}{\partial n} \right) = h(T - T_\infty) \quad (13)$$

The variational formulation for convective boundary can be represented as;

$$\delta \chi_{bconv.} = \int_A -K \left(\frac{\partial T}{\partial n} \right) \delta T dS = \int_A h(T - T_\infty) \delta T dS \quad (14)$$

$$\delta \chi_{bconv.} = \int_i^j -K r \left(\frac{\partial T}{\partial n} \right) \delta T dS \quad (15)$$

Where i and j are the nodal points of the element of side s.
Let $\frac{r}{s} = \cos \theta$, $\left(\frac{\partial T}{\partial n} \right) = h(t - t_\infty)$ and $ds = \frac{dr}{\cos \theta}$

$$T(s) = N_{s_i} t_i + N_{s_j} t_j = [N_s] \{t\} \quad (16)$$

Where N_{s_i} and N_{s_j} are the shape factor.

$$N_{s_i} = \frac{s_j - s}{s_{ij}} = \frac{(r_j - r)}{\frac{r_{ij}}{\cos \theta}} = \frac{(r_j - r)}{r_{ij}}$$

$$N_{s_j} = \frac{s - s_i}{s_{ij}} = \frac{\frac{(r - r_i)}{\cos \theta}}{\frac{r_{ij}}{\cos \theta}} = \frac{(r - r_i)}{r_{ij}} \quad (17)$$

With the help of these equations (15, 16 and 17) variational integral for convective boundary can be represent as

$$\frac{\partial \chi_{bconv.}}{\partial \{t_s\}} = \frac{2\pi h r_m r_{ij}}{6 \cos \theta} \begin{bmatrix} 2 - \frac{\varepsilon}{2} & 1 \\ 1 & 2 + \frac{\varepsilon}{2} \end{bmatrix} \left\{ \begin{array}{l} \{t_{s_i}\} \\ \{t_{s_j}\} \end{array} \right\} - \left\{ \begin{array}{l} (ht_\infty)_1 \\ (ht_\infty)_2 \end{array} \right\} \quad (18)$$

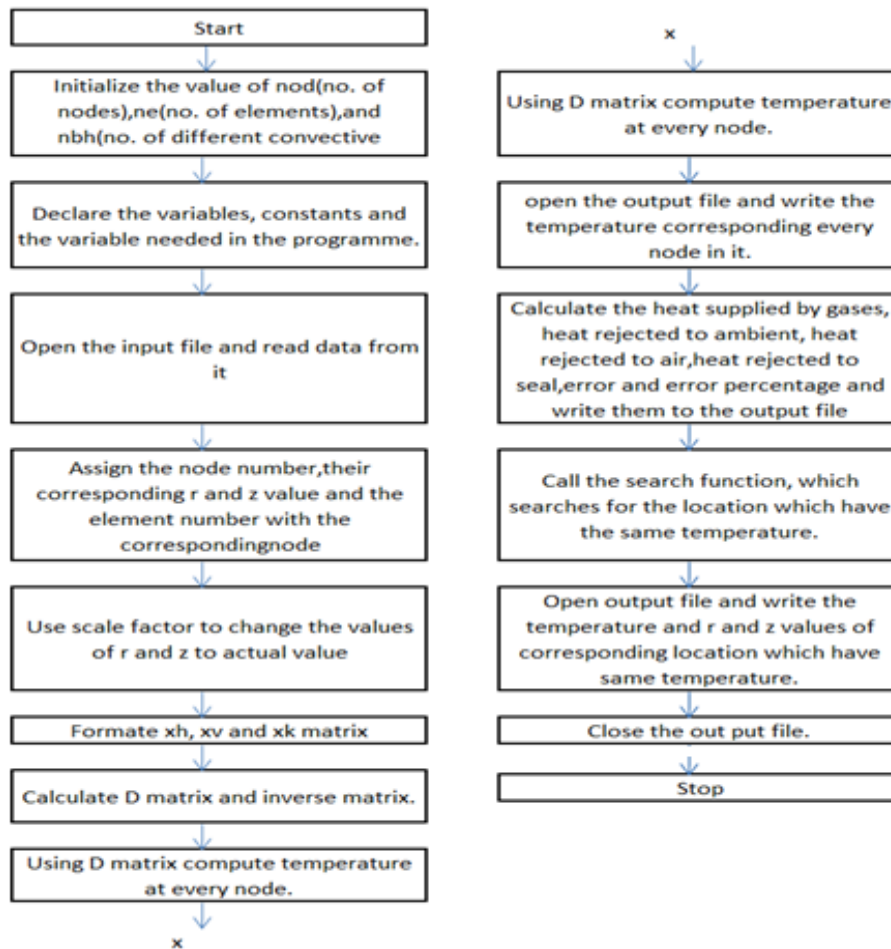
Where,

$$r_m = \frac{r_i + r_j}{2}, r_{ij} = r_j - r_i, r_j = r_m - \frac{r_{ij}}{2}, \varepsilon = \frac{r_{ij}}{r_m}$$

$$\frac{\partial X_{bconv.}}{\partial \{t_s\}} = [H]_s \{t\}_s - \{ht_\infty\} \quad (19)$$

5. Flow chart

The following process is followed to examine the temperature and heat flow rate in the valve through the finite element method.



Step 1: Initially supply the value of nodes, elements, constants, different convective boundaries, and variables desired in the program.

Step2: Initialize the input data.

Step3: Use scale factor to modify the model node value to real node value.

Step4: Make the conductive, convective, and contact matrix and then develop D and DI matrix (D inverse matrix) with the help of input data (value of nodes, elements, constants, different convective boundaries, and variables)

Step5: calculate the temperature at every nodal point with the use of the DI matrix.

Step6: Calculate the heat supplied by the gases to the valve (Hg), heat rejects to air (Ha), heat rejects to water (Hw).

Step7: Search the nodal points which have the same temperature value, then plot the constant temperature (isotherms) line in the model.

6. Temperature field calculation

The prediction of the temperature distribution in the valves involves the resolution of the heat conduction and convection equation with the appropriate boundary conditions. For this purpose two-dimensional finite-element model for the valves modeling case was considered for steady-state engine operation. These models give satisfactory results with significant computer time economy.

Now from equations (8), (12) and (19), we can find the conduction, convection and contact boundary variational integral of heat transfer globally(for all the elements).

$$\frac{\partial \{X\}^g}{\partial \{t\}^g} = [K]^g \{t\}^g + [H]^g \{t\}^g - \{ht_{\infty}\}^g = 0 \quad (20)$$

$$\text{Let } [K]^{(g)} + [H]^g = [D]^g \quad (21)$$

$$\{ht_{\infty}\}^{(g)} = \{V\}^{(g)} \quad (22)$$

$$\{t\}^{(g)} = [DI]^{(g)} \{V\}^{(g)} \quad (23)$$

where $[DI]^{(g)} = [D]^{(g)^{-1}}$, $\{t\}^{(g)}$ global temperature.

The solution of the reduced steady-state heat conduction problem in the r, z coordinate system is found by sub-dividing the quarter part of the valves into 117 elements and 89 nodes. Every element exists in thermal equilibrium with its neighboring elements. Generate computational code to solve mathematical model through FORTRAN language and find the temperature at all the nodes afterward draw temperature field (isothermal distribution curve) in the diesel piston model.

7. Results

The results of calculations carried out and have been presented in [Figure 2], [Figure 3], [Figure 4] and [Figure 5] show the temperature field (isothermal distribution) through the valves for four different engine loading conditions having resulting gas temperatures of 1000oC, 800oC, 600oC, and 400oC respectively. The solid lines show the temperature field of valves. The result shows that there is a tendency to decrease the temperature fields in the valves as the thermal load decreases. The maximum and minimum temperatures in the valves occur at the bottom surface and the top surface, respectively, as shown in [Table 4]. From [Figure 2], [Figure 3], [Figure 4], and [Figure 5], it can be seen that valves receive heat from hot gases, which increases with an increase in engine loads. likewise, the heat gone to water, and heat gone to air, also increases with an increase in engine loads. [Table 5] shows the minimum and maximum value of heat transfer rate during the valve. [Figure 6] shows the variation of heat increase (Qg) by the valve from the hot gases, heat gone to valve seat (Qs), the heat lost to incoming air (Qa), and heat lost to valve bush (Qb) at the four different thermal loading conditions. It seems that heat received from the hot gas is increased with an increase in the engine combustion temperature (Tg). Similarly, the heat lost to the seat, the

heat lost to the bush, and heat lost to air increases with engine combustion temperature. The temperature and heat transfer are obtained by indirect evaluation of the boundary conditions of the valve. The method presented may be used for other parts of I.C. engines such as the piston, cylinder heads, and cylinder wall, etc.

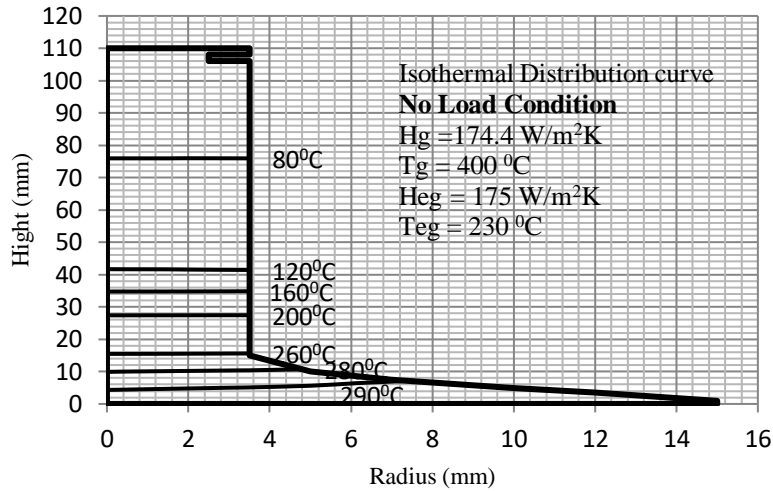


Figure 2. Isothermal curves at no load condition

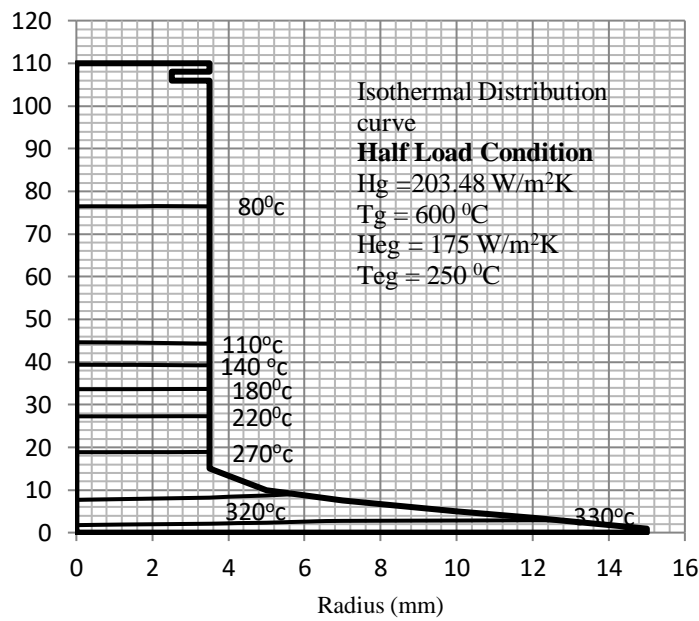


Figure 3. Isothermal curves at half load condition

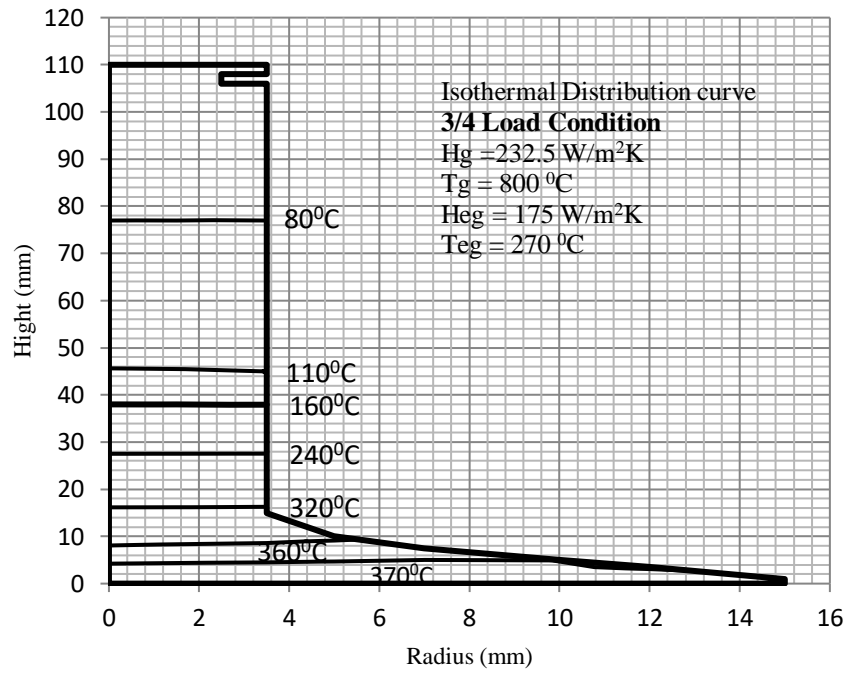


Figure 4. Isothermal curves at 3/4 load condition

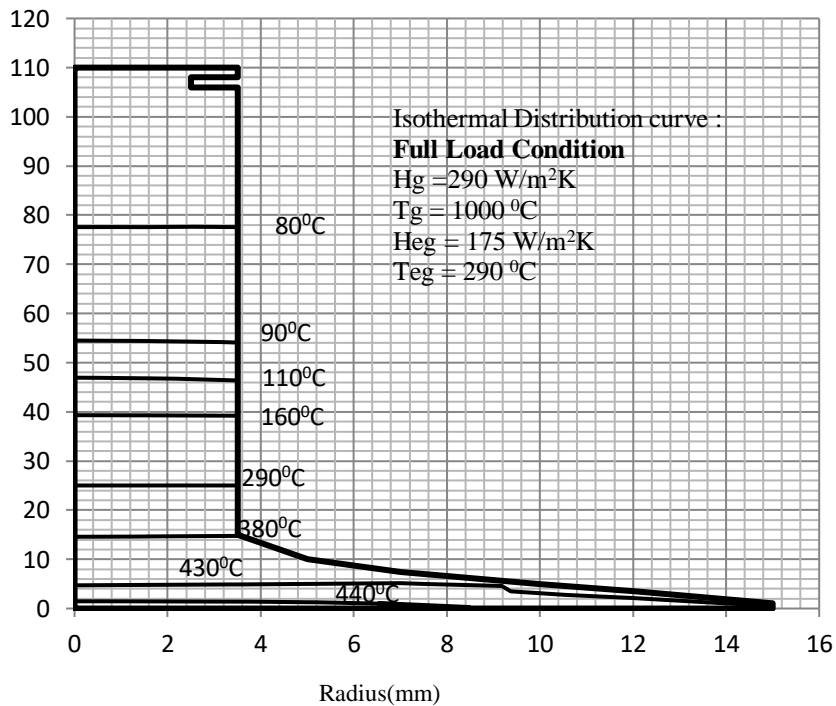


Figure 5. Isothermal curves at Full load condition

Table 4. The maximum and minimum temperature in the valves

Engine loading condition	Max. Temperature (°C)	Min. Temperature (°C)
Case 1 (No Load)	290	80
Case 2 (Half Load)	330	80
Case 3 (3/4 Load)	370	80
Case 4 (Full Load)	440	80

Table 5. Maximum and minimum value of heat flow rate (kW/hr) in the valve

Particulars	heat flow rate (kW/hr)	
	Maximum	Minimum
Qg (Gas Side)	130.60616	14.41556
Qs (Seat Side)	98.04006	0
Qb (Bush side)	19.54745	12.43298
Qa (Air Side)	12.43071	3.414

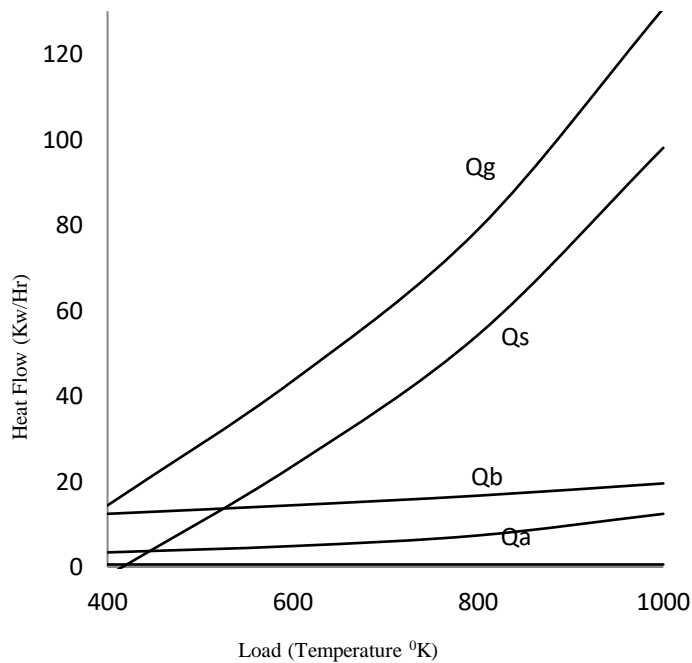


Figure 6. Heat flow pattern for four different load cases

8. Nomenclature

K_X, K_Y, K_Z = Thermal conductivity in X, Y and Z direction respectively ($\text{Wm}^{-1}\text{K}^{-1}$)
 q_E = Heat conduction per unit volume (J m^{-3})
 ρ = Density of the material (kg m^{-3})
 $A^{(e)}$ = Area of the Element (e) (m^2)
 C = Specific Heat
 \emptyset = engine crank angle ($^\circ\text{CA}$)
(e) = Element Number (e)
 E = Young's Modulus of Elasticity (N m^{-2})
 $\{F\}^{(e)}$ = Force on the Element (e)
 $\{F\}_P$ = Force Due To Distributed Load
 $\{F\}_{\epsilon_0}$ = Force Due To Initial Strain
 $\{F\}_{\sigma_0}$ = Force Due To Initial Stress
i, j, k = Nodal Point Number of an element
 q_g = Heat Generation per Unit Volume
 r_{ij} = $r_j - r_i$, Difference Between r Co-ordinates Of Nodal Point i & j
 s_{ij} = Distance between Nodal Points i & j
 T = Temperature Variable
 T_α = Surrounding Temperature
 T_s = Surface Temperature
 α = Thermal Expansion
 $\delta_i, \delta_j, \delta_k$ = Displacement of Nodal Points i, j, k
 $\{\delta\}^{(e)}$ = Displacement of Element
 χ = Variational Integral
 χ_b = Boundary Term of Variational Integral
 $[k]^{(e)}$ = stiffness matrix
 h_c = Convective heat transfer coefficient

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