TRANSIENT HEAT CONDUCTION IN A TWO-STROKE DIESEL ENGINE PISTON

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Abstract:

In this paper we study the transient heat conduction in a piston of a diesel engine, subjected to a periodic boundary condition on the surface in contact with the combustion gases. An elliptic scheme of numerical grid generation was used, so that the irregular shaped piston in the physical domain was transformed into a cylinder in a computational domain. The timewise variations of the temperature of several points in the piston are examined for different piston materials, as well as for motored and fired engines.

Keywords: Diesel engine, finite-differences, numerical grid generation

INTRODUCTION

The solution of heat transfer problems in internal combustion engines is very complicated for several reasons, including, among others: the cyclic temperature variation of gases inside the engine; the parts involved, such as pistons, do not have a regular shape; such parts are subjected to different heat transfer coefficients from the top, bottom and lateral sides, which may vary during the cycle; and the estimation of heat transfer coefficients constitute, in itself, a problem. A review of available theoretical and experimental works on the subject was presented by Borman and Nishiwaki (1987).

In the present paper we perform a two-dimensional axially symmetric finite-difference analysis of the transient heat conduction in a piston of a two-stroke diesel engine. For such an analysis, we transformed the irregular shaped piston from the physical domain into a cylinder in a computational domain. The transient heat conduction equation was transformed into the computational domain, where it was solved with finite-differences by using the ADI (Alternating Direction Implicit) method (Peaceman and Rachford, 1955).

The computer code used in this work is an improvement over another code developed by our group in the past for the solution of a similar problem (Colaço and Orlande, 1996). The problem addressed by Colaço and Orlande (1996) also involved the transient analysis of a diesel engine piston; but considered for the computations time-averaged values for the heat transfer coefficient between the gas and the piston surface, as well as for the temperature of the gas inside the cylinder. In the present work we consider the periodic variations of the heat transfer coefficient between the gas and the piston surface and of the temperature of the gas inside the cylinder, for the computation of the transient temperature field inside the piston. In this work we used the correlation of Eichelberg (Borman and Nishiwaki, 1987; Prasad and Samria, 1990) due to its simplicity and because it does not involve many empirical constants. The temperature variation of the gas inside the cylinder was computed by using a double-Wiebe function for the heat release during combustion (Ramos, 1989).

PHYSICAL PROBLEM

The physical problem considered here is the transient heat conduction in a diesel engine piston. The piston is assumed to be axi-symmetric, so that asymmetries due to the piston pin and oil cooling channels are neglected. The piston considered in the present work is the same studied by Prasad and Samria (1990). The piston geometry with coordinates (in millimeters) relevant for this study are presented in figure 1:

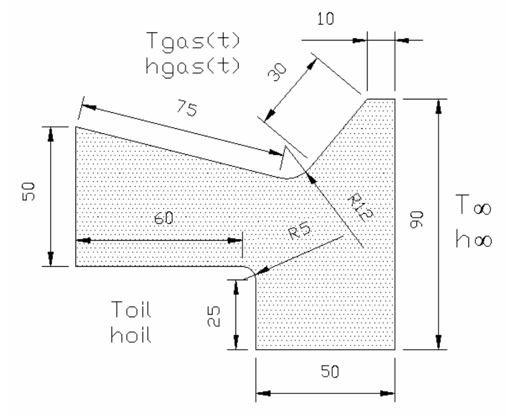


Figure 1. Geometry and coordinates

The piston is heated through its top surface by the gas inside the combustion chamber. The gas temperature (T_{gas}) and the heat transfer coefficient between gases and piston (h_{gas}) are assumed to vary within each engine cycle. The piston is cooled by oil on its bottom surfaces and by a coolant fluid flowing through passages in the cylinder wall. The oil temperature (T_{oil}), as well as the heat transfer coefficient between oil and piston (h_{oil}) are supposed to be constant. The heat transfer to the coolant fluid is taken care by using a constant overall heat transfer coefficient (h_∞), which takes into account the heat transfer from the piston to the cylinder wall, conduction through the wall, and convection from the wall to the coolant fluid. The fluid temperature (T_∞) is assumed to be constant.

The mathematical formulation of such physical problem is given by:

$$\frac{1}{\alpha^*}\frac{\partial T(\mathbf{r},t)}{\partial t} = \nabla^2 T(\mathbf{r},t)$$ in the region, for $t > 0$ (1.a)

$$-K\frac{\partial T}{\partial \mathbf{n}_i} = h_i(t)\left[T - T_i(t)\right]$$ on the boundary surface Γ_i, for $t > 0$ (1.b)

$$\partial T / \partial \mathbf{r} = 0$$ on the symmetry axis ($r = 0$), for $t > 0$ (1.c)

$$T = T_0(\mathbf{r})$$ for $t = 0$ in the region (1.d)

where h_i, T_i and $\partial T/\partial \mathbf{n}_i$ are, respectively, the heat transfer coefficient, the fluid temperature and the normal derivative of temperature at each of the boundary surfaces Γ_i. α^* and k are the thermal diffusivity and thermal conductivity, respectively.

ANALYSIS

The discretization of the piston presented in the figure 1 is difficult due to its irregular shape. In order to overcome such difficulty, we neglected the effects of the piston rings and transformed the irregular piston in the physical domain (z,r) into a cylinder in the computational domain (ξ,η), as shown in figure 2.

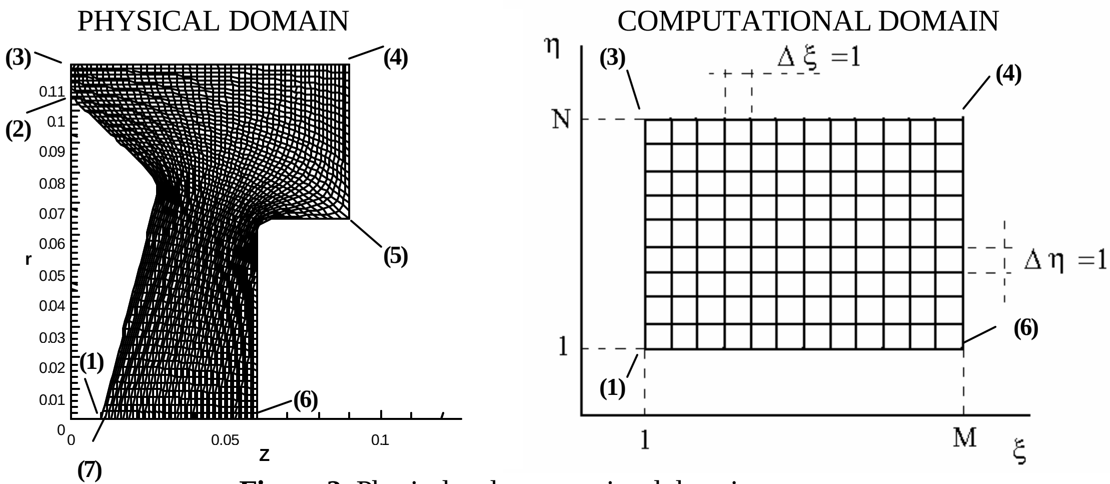


Figure 2. Physical and computational domain

In figure 2, M and N are the number of lines of ξ and η variables, respectively. The transformation above is defined by the solution of two elliptic partial differential equations, used to generate the finite-difference grid for the piston (Thompson et al, 1985; Maliska, 1995; Özisik, 1994).

The problem given by Eqs. (1) is transformed into the computational domain (ξ,η), where it is solved for the temperatures $T(\xi,\eta,t)$. In the computational domain, problem (1) takes the form:

$$\frac{1}{\alpha^*}\frac{\partial T(\xi,\eta,t)}{\partial t}=\frac{1}{J^2}\left[\alpha T_{\xi\xi}-2\beta T_{\xi\eta}+\gamma T_{\eta\eta}\right]+\left[PT_{\xi}+QT_{\eta}\right] \tag{2.a}$$

$$\frac{k}{J\sqrt{\alpha}}(\alpha T_{\xi}-\beta T_{\eta})=h_{gas}(t)[T-T_{gas}(t)] \qquad \text{at } \xi=1;1<\eta<N;t>0 \tag{2.b}$$

$$-\frac{k}{J\sqrt{\alpha}}(\alpha T_{\xi}-\beta T_{\eta})=h_{oil}(T-T_{oil}) \qquad \text{at } \xi=M;1<\eta<N;t>0 \tag{2.c}$$

$$(\gamma T_{\eta}-\beta T_{\xi})=0 \qquad \text{at } \eta=1;1<\xi<M;t>0 \tag{2.d}$$

$$-\frac{k}{J\sqrt{\gamma}}(\gamma T_{\eta}-\beta T_{\xi})=h_{\infty}(T-T_{\infty}) \qquad \text{at } \eta=N;1<\xi<M;t>0 \tag{2.e}$$

$$T=T_0(\xi,\eta) \qquad \text{for } t=0;1<\xi<M;1<\eta<N \tag{2.f}$$

where

$$\alpha=z_{\eta}^2+r_{\eta}^2\ ,\quad \beta=z_{\xi}z_{\eta}+r_{\xi}r_{\eta}\ ,\quad \gamma=z_{\xi}^2+r_{\xi}^2\ ,\quad J=z_{\xi}r_{\eta}+r_{\xi}z_{\eta} \tag{3.a,b,c,d}$$

BOUNDARY CONDITION AT THE PISTON-GAS INTERFACE

Several correlations for the heat transfer coefficient at the gas-piston surface are available in the literature (Heywood, 1988; Ramos, 1989; Borman and Nishiwaki, 1987). In this paper, we preferred to use the correlation of Eichelberg, due to its simplicity and because it does not involve many empirical constants. Such correlation was developed for naturally-aspirated large two-stroke and four-stroke diesel engines, such as the one under picture in this work. Eichelberg's correlation is given by

$$h_{gas}(t) = 7.67 \times 10^{-3} [P_i(t) T_i(t)]^{1/2} (c_m)^{1/3} \quad kW/m^2K \tag{4}$$

where c_m is the mean-piston-speed in m/s, while T and P are the instantaneous temperature in Kelvin and pressure in kPa, respectively, of the gas inside the cylinder.

For the calculation of the instantaneous temperature and pressure inside the cylinder, we assume that the compression and expansion processes are polytropic, with polytropic index of 1.3 (Ferguson, 1986). The mass of gas inside the cylinder is supposed constant and it is assumed to be at atmospheric conditions ($P = 10^5$ Pa and T = 298 K) when the piston is at the bottom-dead-center. Since we are dealing with a two-stroke engine, we assume here that exhaustion and admission take place simultaneously at the bottom-dead-center, so that, at the end of the expansion stroke, the gas inside the cylinder returns instantaneously to the initial atmospheric conditions.

During combustion, the relation between the cylinder volume, gas pressure and the rate of heat release can be expressed as (Ferguson, 1986):

$$\frac{dP}{d?} = -? \frac{P}{V}\frac{dV}{d?} + \frac{(? -1)}{V}\frac{dQ}{d?} \tag{5}$$

where θ is the crankshaft angle in degrees.

The rate of heat release during combustion can be obtained from a double-Wiebe function in the form (Ramos,1989)

$$\frac{dQ}{d\theta} = 6.9\frac{Q_p}{\theta_p}(M_p+1)\left(\frac{\theta-\theta_{ig}}{\theta_p}\right)\exp\left[-6.9\left(\frac{\theta-\theta_{ig}}{\theta_p}\right)^{M_p+1}\right] + 6.9\frac{Q_d}{\theta_d}(M_d+1)\left(\frac{\theta-\theta_{ig}}{\theta_d}\right)\exp\left[-6.9\left(\frac{\theta-\theta_{ig}}{\theta_d}\right)^{M_d+1}\right] \tag{6}$$

where the subscripts *p* and *d* refer to premixed and diffusive combustion, respectively; M_p and M_d are shape factors corresponding to premixed and diffusive combustion, respectively; q_p and q_d are the durations of the energy release in premixed and diffusive combustion, respectively; and Q_p and Q_d characterize the heat release in premixed and diffusive combustion, respectively; q_{ig} is the ignition angle. Such parameters are functions of the injection angle and can be obtained in Ramos (1989).

After the fuel is injected into the cylinder, several physical and chemical phenomena take place before combustion can start. Such phenomena result on a delay time that can be represented in the form of an Arrhenius-type expression such as

$$t_d(ms) = A\, p(atm)^{-n} \exp\left(T_a/T(K)\right) \tag{7}$$

For the case under picture in this work, we used the following values for the constants appearing in equation (7), obtained from Ramos (1989): A=53.5, n=1.23, T_a = 676.5 K.

RESULTS AND DISCUSSION

For the results presented below, the values of various parameters were chosen as follows (Prasad and Samria, 1990): *(i)* Initial temperature: T_0=20^{o}C; *(ii)* Oil temperature: T_{oil}=85^{o}C; *(iii)* Heat transfer coefficient to the oil: h_{oil}=175W/m^2 ^{o}C; *(iv)* Cooling water temperature: 85^{o}C; *(v)* Overall heat transfer coefficient to the cooling water: h_∞= 1000 W/m^{2o}C; *(vi)* Engine Speed: 850 RPM; *(vii)* Compression ratio: 17; *(viii)* Piston diameter: 0.23 m; *(ix)* Stroke: 0.3 m; *(x)* Injection angle: -20^o.

Four test-cases were examined in the present work, depending on the piston material and if the engine is motored or fired, as summarized in table 1. For the case of a motored engine, the compression and expansion processes were also assumed polytropic, with polytropic index of 1.3.

Table 1. Test-cases

Test-case	Piston material	Engine condition
1	Aluminum	Fired
2	Aluminum	Motored
3	Cast iron	Fired
4	Cast iron	Motored

The thermal conductivity and thermal diffusivity of aluminum and cast iron were taken, respectively as, 204 W/m^{o}C, 8.418 x 10^{-5} m^2/s, 54 W/m^{o}C and 0.970 x 10^{-5} m^2/s (Ozisik, 1993).

Before obtaining results for the piston transient temperature field by using the present numerical approach, a grid convergence analysis is required in order to assess the numerical error involved in the solution. Nine different grids were generated. The number M of ξ lines and N of η lines of each grid are presented in table 2, while figure 2 shows grid G5, with M=51 and N=66.

Table 2. Finite difference grids

Grid	M	N
G1	41	56
G2	51	56
G3	61	56
G4	41	66
G5	51	66
G6	61	66
G7	41	84
G8	51	84
G9	61	84

The temperatures of the first 6 points shown in figure 2 were compared for the grids presented in table 2. Such temperatures were obtained for time t=5 s and for a motored engine with an aluminum piston. The time step used was Δt=1x10^{-3}s.

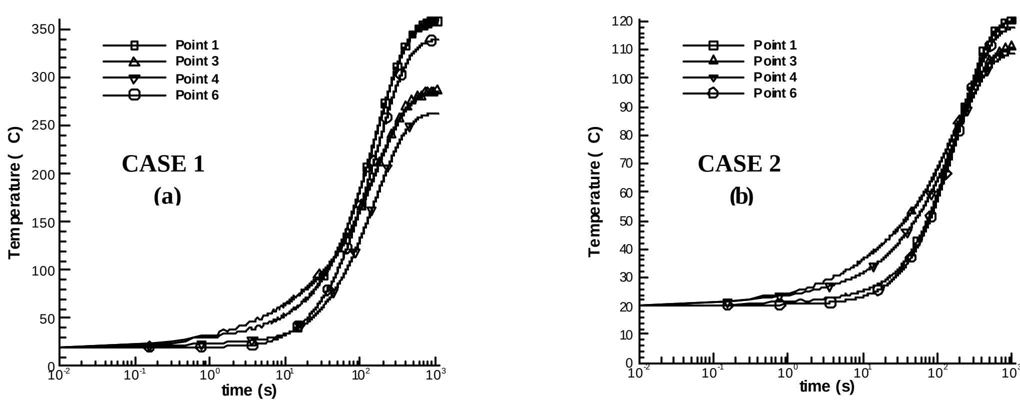
The relative differences in percent for the temperatures computed with the different grids used in this study are shown in table 3. This table shows that generally the grids are not converged with N=56, because differences of the order of 2% can be observed for point 2, when N is increased to 66, irrespective of the number of ξ lines (M) utilized. On the other, differences of less than 0.5% can be noticed when N is increased to 84, as compared to N=66. By taking into analysis now the grids with N=66 (G4, G5 and G6), we note that the grid is basically converged in the ξ direction with M=51. A maximum difference of 1.1% is observed for point 6, when M is increased from 51 to 61 (grids G5 and G6, respectively). From the examination of table 3, we decided to use grid G5 for the foregoing analysis, with M=51 and N=66. Such grid is basically converged in the ξ and η directions. Also, its CPU time was 4 min and 23 s as compared to 5 min and 7 s for grid G6, thus enabling substantial savings on computer time, without loss of accuracy. The CPU times correspond to a Pentium 200 MMX, with 64 Mb of RAM memory, running under the Microsoft Fortran PowerStation 4.0 platform.

Table 3. Relative temperature difference in percent between grids

Point	G1-G2	G2-G3	G4-G1	G5-G2	G6-G3	G4-G5	G5-G6	G7-G4	G8-G5	G9-G6	G7-G8	G8-G9
1	0.56	0.28	0.43	0.67	0.34	0.33	0.61	0.41	0.38	0.30	0.36	0.69
2	1.05	0.37	2.08	1.90	1.89	0.87	0.36	0.61	0.14	0.01	0.40	0.20
3	0.85	0.31	1.32	1.13	1.13	0.65	0.31	0.39	0.09	0.14	0.36	0.08
4	0.17	0.20	0.50	0.70	0.65	0.04	0.15	0.49	0.42	0.43	0.04	0.16
5	0.58	0.35	0.32	0.00	0.07	0.26	0.42	0.02	0.08	0.08	0.17	0.41
6	0.94	0.75	0.29	0.62	0.26	0.61	1.10	0.22	0.30	0.30	0.52	1.11

After choosing the grid, we performed an analysis of the timewise variation of the temperature in the piston for the different test-cases summarized in table 1.

Figure 3.a-d show the variation for the temperature with time, until the quasi-steady-state is reached, of selected points in the piston for cases 1-4, respectively. These figures show that, initially, the temperature of the points near the cylinder wall is larger than for the other points. Such is the case because initially the piston is assumed to be at a temperature smaller than that of the cooling fluid and of the oil. By comparing figures 3.a-b with figures 3.c-d, we note that the quasi-steady-state is reached faster for the aluminum piston than for the cast-iron piston, as a result of the larger thermal conductivity and thermal diffusivity for aluminum.



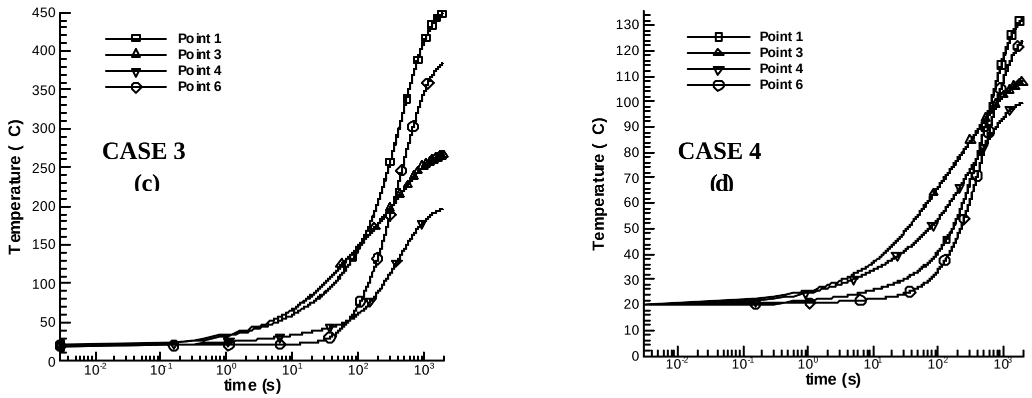


Figure 3. Temperature variation of selected points

The quasi-steady-state temperature distributions in the piston are shown in figures 4.a-d, for test-cases 1 to 4, respectively. Note in these figures the higher temperatures in the cast-iron piston than in aluminum piston. Also, note the higher temperature gradients in the radial direction in the top region of the cast-iron piston, resulting in larger thermal stresses than for the aluminum piston.

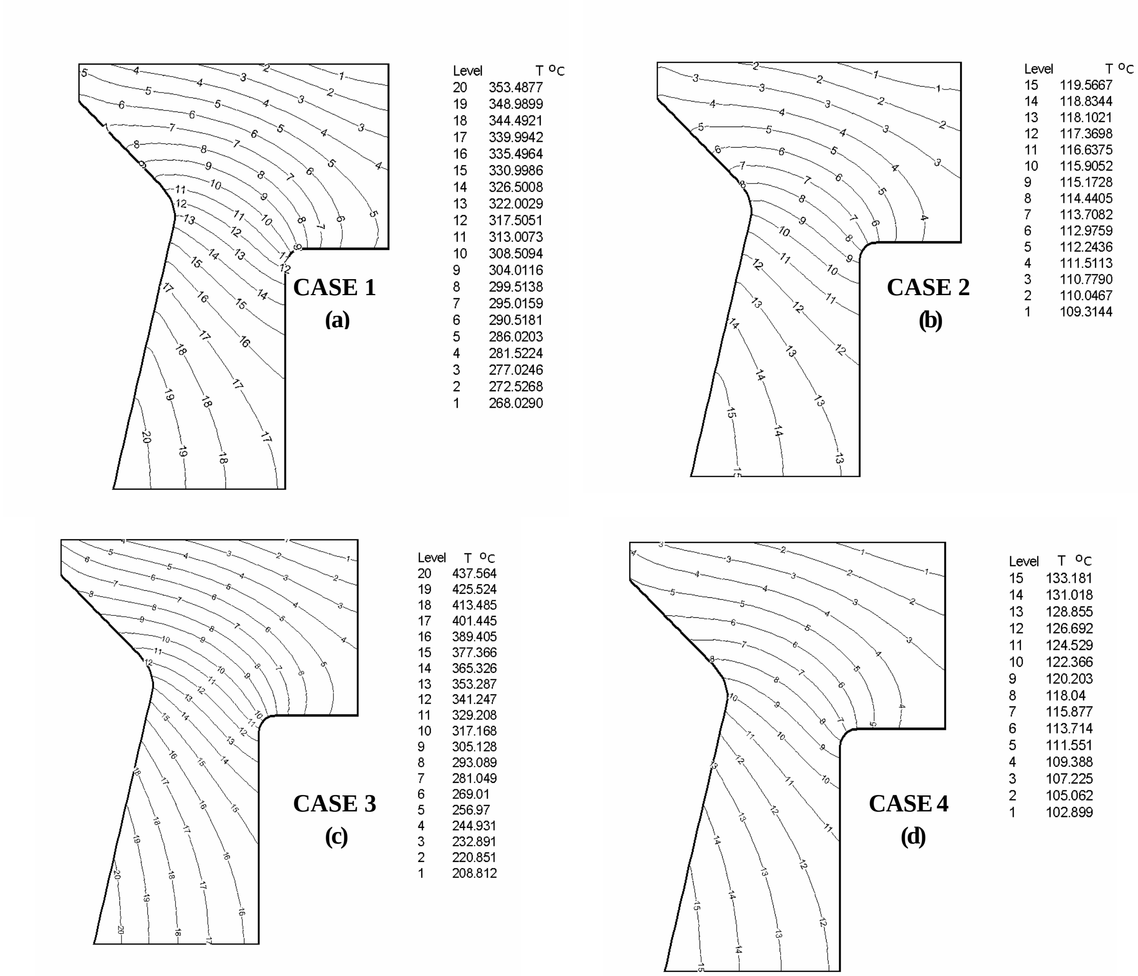


Figure 4. Quasi-steady state temperature distribution in the piston

Figure 5 presents the temperature variation of points 1 and 7 in the piston for test-case 1, involving an aluminum piston in a fired engine, when the quasi-steady-state is established. Figure 5 shows that the amplitude of variation for the temperature of point 1 is 1.94°C. For point 7, located 1.39 mm below the gas-piston interface on the piston center-line, such amplitude of variation is less than 1°C. Much smaller amplitudes of variation were observed for the temperature of the cast-iron piston, due to its smaller thermal diffusivity as compared to aluminum. Also, the amplitudes for motored engines are much smaller than for fired engines.

These results are extremely interesting because they reveal the high-level of difficulty for the solution of the inverse problem of estimating the periodic heat flux at the gas-piston interface, by using temperature measurements taken below the interface. Such is the case because the amplitudes of temperature variation are of the same order of magnitude of the expected levels for the measurement errors. Hence, it would be impossible to distinguish if the measured temperature variations result from the measurement errors or from the periodic boundary conditions, and no useful information would be recovered from the temperature measurements.

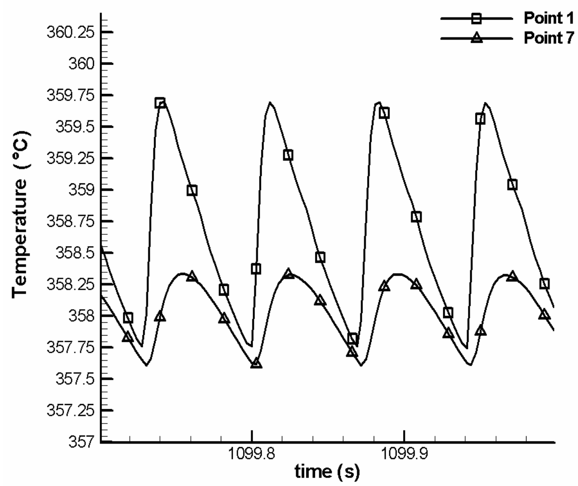


Figure 5. Temperature variations of points 1 and 7 for the test-case 1.

CONCLUSIONS

The analysis performed reveals that, for the cases studied, the steady-state was reached earlier for aluminum pistons than for cast-iron pistons. Also, cast-iron pistons are subjected to higher temperatures and larger temperature gradients, thus resulting in larger thermal stresses.

Generally, the temperature variations, resultant from the periodic boundary condition at the gas-piston interface, are largely damped within a quite small distance below the interface. Therefore, the solution of the inverse heat conduction problem of estimating the periodic boundary heat flux, by using temperature measurements below the surface, is quite difficult.

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