TRANSIENT HEAT TRANSFER ANALYSIS OF A DIESEL ENGINE PISTON

Marcelo J. Colaço and Helcio R.B. Orlande
Programa de Engenharia Mecânica
EE / COPPE / UFRJ - Caixa Postal 68503
21945-970 - Rio de Janeiro - RJ - Brasil

SUMMARY

In this paper we study the transient heat transfer in a piston of a diesel engine, by using a two dimensional finite difference analysis. An elliptic scheme of numerical grid generation is used, so that the irregular shaped piston in the physical domain is transformed into a cylinder in a computational domain. The timewise variation of the temperature of several points in the piston is examined for different thermal loads and cooling regimes.

INTRODUCTION

The in-cylinder heat transfer has been generally recognized as a major factor influencing internal combustion engines efficiency and exhaust emissions. Therefore, the analysis of in-cylinder heat transfer is of importance for fuel economy, as well as for environmental preservation.

The solution of heat transfer problems in internal combustion engines is very complicated for different 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).

Different expressions for the time-varying heat transfer coefficient between gases and piston have been suggested in the literature (Borman and Nishiwaki, 1987, Heywood, 1988, Kornhouser and Smith, 1994). However, stress computations do not require the knowledge of the cyclic variation of temperature in the piston and are based on time-averaged heat transfer coefficients (Borman and Nishiwaki, 1987, Singh et al, 1986, Prasad and Samria, 1990). Such is the case because the temperature only varies in a very small depth below the surface during each cycle.

In this paper, we perform a two-dimensional axi-symmetric finite-difference analysis of the transient heat conduction in a diesel engine piston. 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 solutions of the resulting tri-diagonal systems were obtained with a vector version of Thomas algorithm (Ortega, 1988) in order to take advantage of the vector capabilities of the Cray supercomputer at COPPE. The computer code used was thoroughly tested by comparing its results with analytical solutions available for regular geometries in cartesian, polar and cylindrical coordinates (Colaço et al, 1995a,b). A grid convergence analysis was performed by testing meshes, generated with Thompson's elliptic scheme of numerical grid generation (Thompson et al, 1985, Maliska, 1995, Özisik, 1994). The influence of the time step on the solution was also examined. For sake of simplicity, we have used time-averaged heat transfer coefficients at the piston surfaces.

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 geometry and coordinates (in millimeters) relevant for this study are presented in figure 1 (Singh et al, 1986):

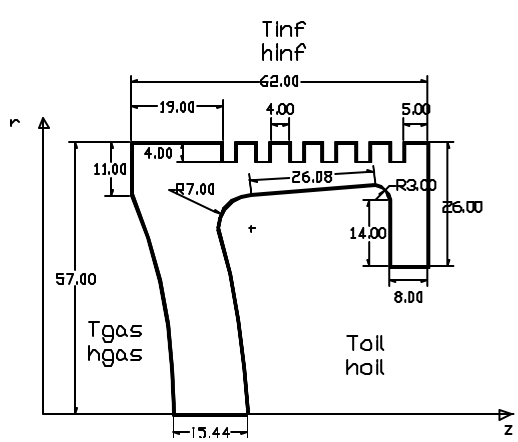

Figure 1 - Geometry and coordinates

The piston is supposed to be initially (t=0) at the temperature distribution $T_o(\vec{r})$. For times t>0, the piston is subjected to convection (third-kind) boundary conditions at all boundaries. The piston is heated through its top surface by the gases inside the combustion chamber. The gases temperature (T_{gas}) and the heat transfer coefficient between gases and piston (h_{gas}) are assumed to be constant. 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, the conduction through the wall, and the 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:

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

$$-k\frac{\partial T}{\partial \vec{n}_i} = h_i(T - T_i) \quad \text{on the boundary surface } \Gamma_i, \qquad (1.b)$$

$$\text{for } t > 0$$

$$\frac{\partial T}{\partial r} = 0 \quad \text{on the symmetry axis } (r = 0), \text{ for } t > 0 \qquad (1.c)$$

$$T = T_0(\vec{r}) \quad \text{for } t = 0 \text{ in the region} \qquad (1.d)$$

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

For the solution of the problem given by Eqs. (1) we have used the finite-difference method as described next.

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 have transformed the irregular piston in the physical domain (z,r) into a cylinder in the computational domain (ξ,η), as shown in figure 2.

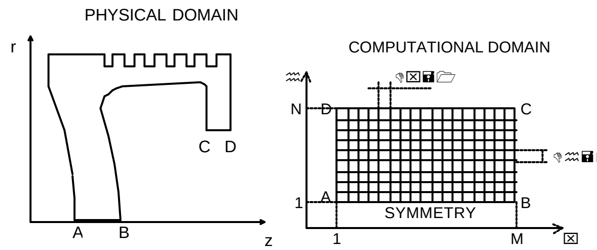

Figure 2 - Physical and computational domain

In figure 2, M and N are the number of lines of the 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). In the computational domain, such equations are given by:

$$\alpha z_{\xi\xi} - 2\beta z_{\xi\eta} + \gamma z_{\eta\eta} + J^2[P(\xi,\eta)z_\xi + Q(\xi,\eta)z_\eta] = 0 \qquad (2.a)$$

$$\alpha r_{\xi\xi} - 2\beta r_{\xi\eta} + \gamma r_{\eta\eta} + J^2[P(\xi,\eta)r_\xi + Q(\xi,\eta)r_\eta - \frac{1}{r}] = 0 \qquad (2.b)$$

where the subscripts denote partial derivatives and

$$\alpha = z_\eta^2 + r_\eta^2 \qquad \beta = z_\xi z_\eta + r_\xi r_\eta \qquad (3.a,b)$$

$$\gamma = z_\xi^2 + r_\xi^2 \qquad J = z_\xi r_\eta + r_\xi z_\eta \qquad (3.c,d)$$

P(ξ,η) and Q(ξ,η) are control functions, which can be used to refine the grid in regions of large temperature gradients (Thompson et al, 1985, Maliska, 1995, Özisik, 1994).

The problem given by Eqs. (1) is also transformed into the computational domain (ξ,η), where it is solved for the temperatures T(ξ,η,t). Since the transformation given by Eq. (2) is one-to-one for a non-vanishing jacobian J, the solution T(z,r,t) in the physical domain can be obtained by knowing T(ξ,η,t). In the computational domain, problem (1) takes the form:

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

$$\text{in } 1 < \xi < M;\ 1 < \eta < N,\ t > 0 \qquad (4.a)$$

$$\frac{k}{J\sqrt{\alpha}}(\alpha T_\xi - \beta T_\eta) = h_1(T - T_1) \quad \text{at } \xi = 1;\ 1 < \eta < N;\ t > 0 \qquad (4.b)$$

$$-\frac{k}{J\sqrt{\alpha}}(\alpha T_\xi - \beta T_\eta) = h_2(T - T_2) \quad \text{at } \xi = M;\ 1 < \eta < N;\ t > 0 \qquad (4.c)$$

$$(g T_\eta - b T_\xi) = 0 \quad \text{at } \eta = 1;\ 1 < \xi < M;\ t > 0 \qquad (4.d)$$

$$-\frac{k}{J\sqrt{g}}(g\ T_\eta - b\ T_\xi) = h_3 (T - T_3) \text{ at } \eta = N;\ 1 < \xi < M;\ t > 0 \qquad (4.e)$$

$$T = T_0(\xi, \eta) \quad \text{for } t = 0 ; 1 < \xi < M;\ 1 < \eta < N \qquad (4.f)$$

where $h_i \equiv h_i(\eta)$ and $T_i \equiv T_i(\eta)$, appearing in Eqs. (4.b-c) for i=1,2, are the heat transfer coefficients and fluid temperatures, respectively, at each piston surface which transforms into the boundaries ξ=1 (i=1) and ξ=M (i=2). Similarly, $h_3 \equiv h_3(\xi)$ and $T_3 \equiv T_3(\xi)$ are the heat transfer coefficient and fluid temperature, respectively, at each piston surface that maps into the boundary η=N.

Eqs. (2) subjected to appropriate boundary conditions were discretized by finite-differences in order to generate the grid. The program 2DGRID (Colaço et al, 1995a) was used in this paper. Such program can treat boundaries composed by line or circle segments. The program also permits the choice of either first-kind (fixed points) or second-kind homogeneous (orthogonality) boundary conditions at each surface.

The solution of Eqs. (4) for the transient temperature field T(ξ,η,t) was also obtained by finite-differences, by using the ADI (Alternating Direction Implicit) method of Peaceman and Rachford (1955). The resulting tri-diagonal systems were solved with a vector version of Thomas' Algorithm (Ortega, 1988) in order to reduce the CPU time in the Cray J90 at COPPE. We have used the program 2DHEAT based on such vectrorizable ADI method (Colaço et al, 1995b). This is a quite general program for the solution of transient linear diffusion problems in two-dimensional geometries. It is capable of treating boundary conditions of first, second or third kinds, with constants or variable coefficients, as well as source terms in the region. The program 2DHEAT was validated by comparing its results for regular geometries in rectangular, cylindrical and polar coordinates, with known analytical solutions (Özisik, 1993). We performed more than 75 tests and the relative error between numerical and analytical solutions was always less than 1% (Colaço et al, 1995b). The application of the programs 2DGRID and 2DHEAT to the transient analysis of the diesel engine piston shown in figure 1 is described next.

RESULTS AND DISCUSSION

For the results presented below, the values of various parameters were chosen as follows (Singh et al, 1986):

(i) Initial temperature: T_o=120°C;
(ii) Oil temperature: T_{oil}=50°C;
(iii) Heat transfer coefficient to the oil: h_{oil}=175W/m^2°C;
(iv) Piston thermal diffusivity: α*=0.97x10^{-5}m^2/s;
(v) Piston thermal conductivity: k=54W/m^2°C;
(vi) Heat transfer coefficient to the gases in the combustion chamber: h_{gas}=290 W/m^2°C.

Other parameters of interest for the analysis were varied for the study of several test-cases. The values of such parameters appear below when required.

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. For the same reason, an analysis of the time step used in the ADI method shall be performed.

Five different grids were generated. The number M of ξ lines and N of η lines of each grid are presented in table 1, while figure 3 shows grid G4, with M=20 and N=195.

Table 1 - Finite difference grids

Grid	M	N
G1	30	195
G2	30	163
G3	20	163
G4	20	195
G5	20	210

The temperatures of the 18 points shown in figure 4 were compared for the grids presented in table 1. Such temperatures were obtained for time t=10 s and for a water cooled engine with h_∞=1400W/m^2°C and T_∞=85°C. The temperature of the gases in the combustion chamber was taken as T_{gas}=800°C. The time step used was Δt=1x10^{-3}s for grids G1 to G4; but such time step resulted in a unstable solution for grid G5. Therefore, we have used Δt=0.5x10^{-3}s for grid G5 due to stability requirements.

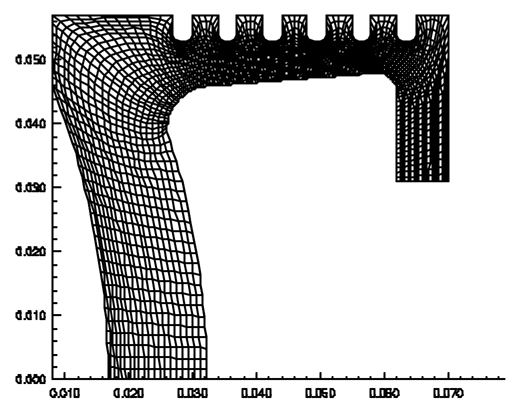

Figure 3 - Grid G4, M=20 and N=195

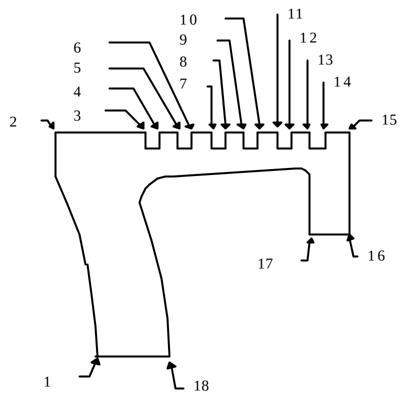

Figure 4 - Points to test the grid convergence

Table 2-Relative temperature difference in percent between grids

Point	G1-G2	G2-G3	G1-G4	G3-G4	G4-G5
1	0.024	0.002	0.002	0.027	0.001
2	0.138	0.207	0.002	0.346	1.117
3	0.348	0.061	0.124	0.412	0.941
4	1.556	0.110	0.461	1.937	1.003
5	1.600	0.115	0.483	1.999	1.022
6	1.585	0.093	0.531	2.055	1.082
7	1.642	0.094	0.556	2.139	1.086
8	1.484	0.020	0.522	2.056	1.094
9	1.511	0.031	0.534	2.108	1.109
10	1.194	0.194	0.466	1.877	1.064
11	1.220	0.220	0.450	1.915	1.082
12	0.815	0.266	0.346	1.440	0.972
13	0.959	0.304	0.285	1.565	0.965
14	0.274	0.123	0.122	0.520	0.827
15	0.445	0.224	0.074	0.747	0.753
16	0.045	0.095	0.100	0.040	0.079
17	0.056	0.159	0.244	0.029	0.080
18	0.015	0.009	0.008	0.015	0.003

The relative difference in percent for the temperatures computed with the different grids used in this study are shown in table 2. It can be noticed in columns 3 and 4 of table 2, that the temperatures vary by less than 0.56 % by using M=30 instead of M=20, either for N=163 or N=195. Thus, it appears that the grid is sufficiently converged in the ξ direction for M=20. On the other hand, the temperatures vary by more than 1.5 %, for several points by using N=195 instead of N=163, as shown by columns 2 and 5 of table 2. A relative difference of 1.1% is observed for few points in column 6 of table 2, by increasing N from 195 to 210 with M=20; but the CPU time increased by a factor of 2.2 in such case. Based on the foregoing analysis, we decided to use grid G4 with M=20 and N=195 in this study. The use of grid G5 with M=20 and N=210 was avoided so that the CPU time would not be excessively large.

The effect of the time step on the solution was also examined. The solution changed by less than 0.002% by reducing the time step from 10^{-3}s to 10^{-4}s. On the other hand, the solution became unstable when the time step was increased to 10^{-2}s. Therefore, we have used in the examples shown below the time step $\Delta t=10^{-3}$s.

After choosing the grid and the time step, we performed an analysis of the timewise variation of the temperature in the piston for different cooling techniques and thermal loads. The test-cases studied in this paper are summarized in table 3.

Table 3 - Test-cases

Case	Coolant	T_∞(°C)	h_∞(W/m²°C)	T_{gas} (°C)
1	water	85	1400	800
2	water	85	1400	1000
3	water	85	1200	800
4	air	25	123	800
5	air	25	123	1000

Figures 5-9 show the variation for the temperature with time, until the steady-state is reached of selected points in the piston for cases 1-5, respectively. These figures show a temperature reduction for some of the points studied, due to the initial temperature of 120°C used in the analysis, which is larger than the oil and coolant temperatures. Such is the case for a warm engine start-up. By comparing figures 5 and 6 for a water-cooled engine, we notice an increase on the temperatures of points 1-4, when the temperature of the gases inside the cylinder increase from 800°C to 1000°C, which correspond to a larger engine speed or load. However, the temperature of points 10, 15 and 17 are very little affected by such increase on the gases temperature. This is not the case for an air-cooled engine, as can be noticed by comparing figures 8 and 9, for T_{gas}=800°C and T_{gas}=1000°C, respectively, which shows a general increase of the temperatures in the whole piston when the gases temperature is increased. A comparison of figures 5 and 7 shows that for a water-cooled engine, the temperatures of the piston are very little affected by reducing the overall heat transfer coefficient to the cooling fluid from h_∞=1400 W/m²°C to h_∞=1200 W/m²°C.

By comparing figures 5 and 8, we notice a general increase on the temperature levels inside the piston, when the engine cooling system uses air instead of water. This fact shows that the piston rings in the air-cooled engine are more likely to degrade due to thermal creep, than those in the water-cooled engine. Also, note that the radial temperature difference between points 1 and 2 is much larger for a water-cooled than for an air-cooled engine. This fact shows that the piston in the water-cooled engine is subjected to higher thermal stresses than in the air-cooled engine. The conclusion above can also be drawn by comparing figures 6 and 9. Figures 5-9 also show that, for the test-cases studied, the steady-state is reached earlier for water-cooled engines than for air-cooled engines. The steady-state was reached after approximately 600s for a water cooled engine and after approximately 1000s for an air-cooled engine.

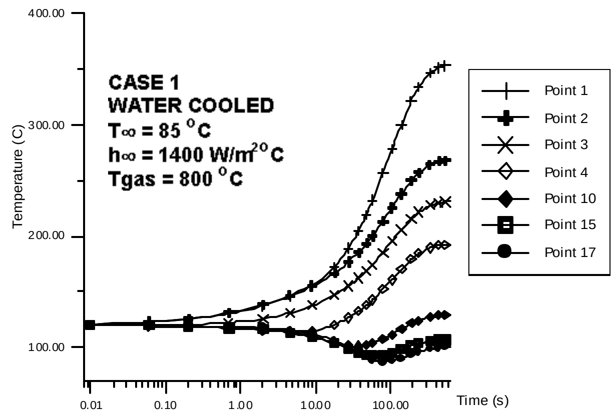

Figure 5 - Temperature variation with time for case 1

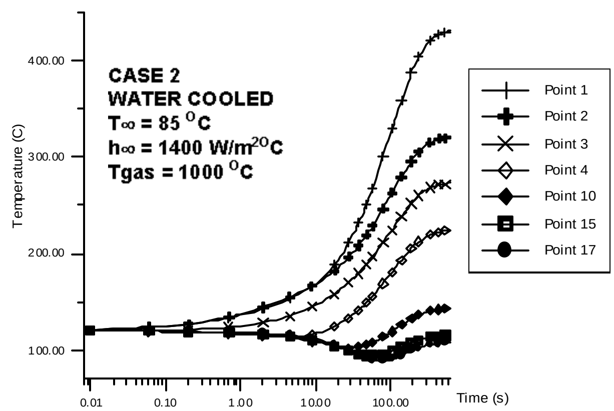

Figure 6 - Temperature variation with time for case 2

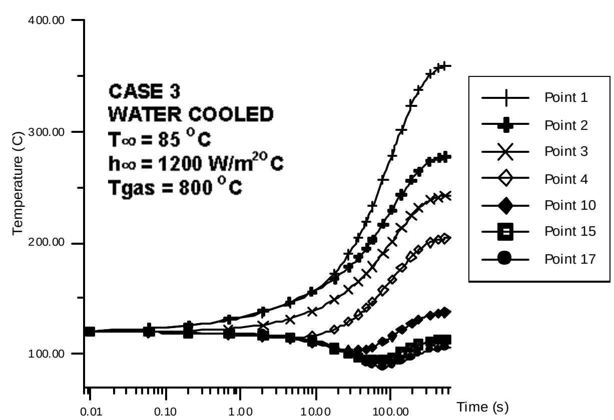

Figure 7 - Temperature variation with time for case 3

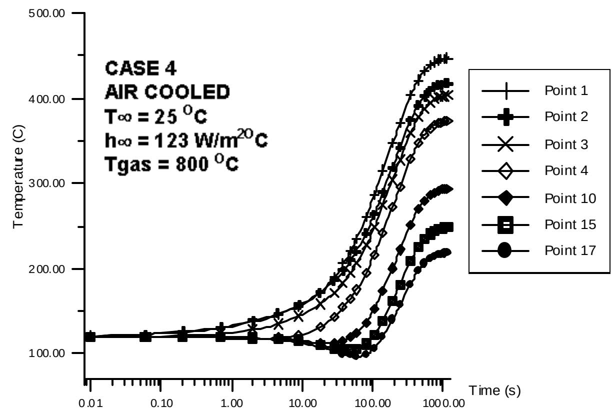

Figure 8 - Temperature variation with time for case 4

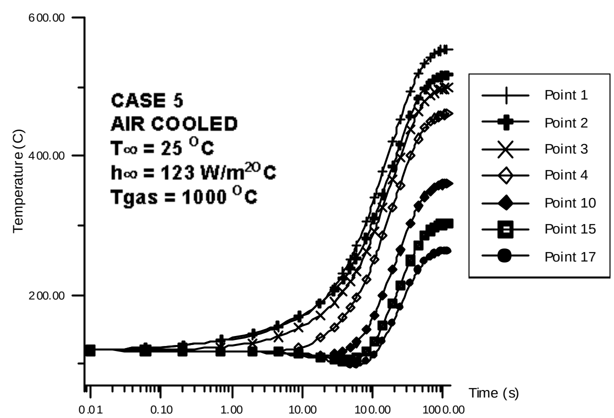

Figure 9 - Temperature variation with time for case 5

For illustration, the steady-state temperature distribution in the piston is shown in figures 10-14 for test-cases 1 to 5, respectively. Note in these figures the higher temperatures in the air-cooled engines than in the water-cooled engines. Also, note the higher temperature gradients in the radial direction in the top region of the piston for the water-cooled engines.

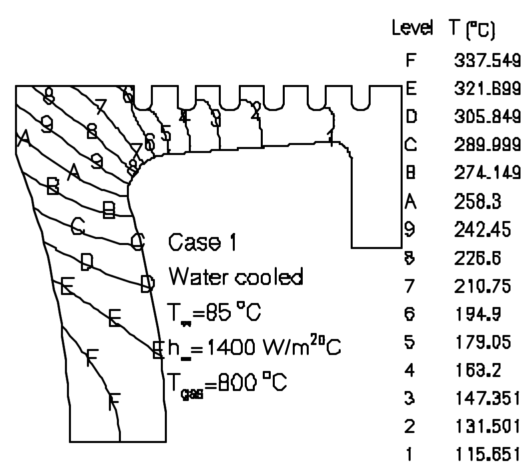

Figure 10 - Steady state temperature distribution for case 1

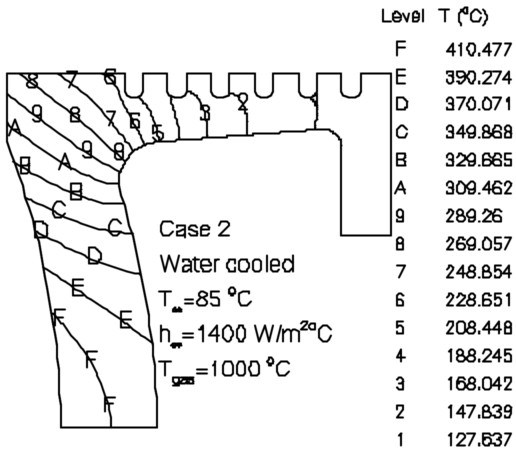

Figure 11 - Steady state temperature distribution for case 2

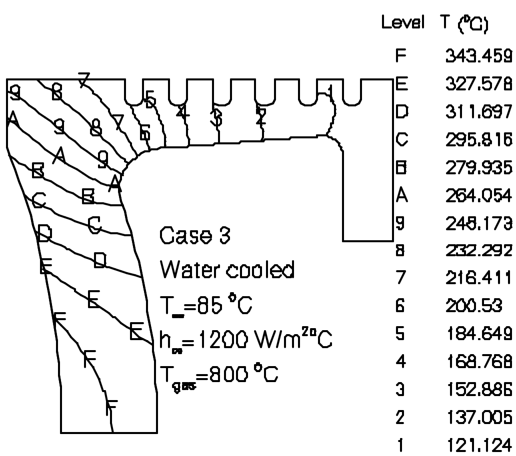

Figure 12 - Steady state temperature distribution for case 3

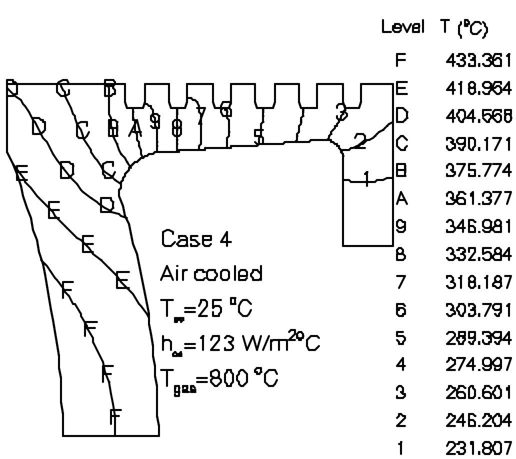

Figure 13 - Steady state temperature distribution for case 4

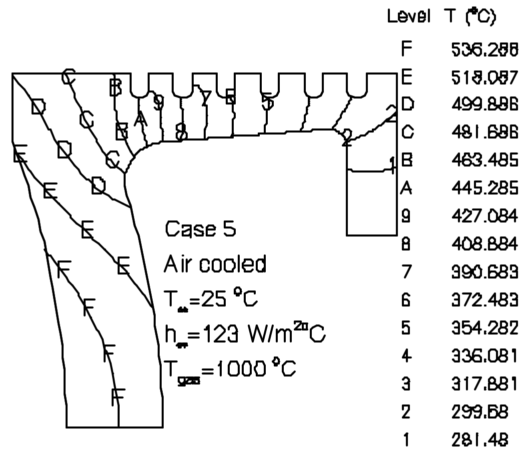

Figure 14 - Steady state temperature distribution for case 5

We have also compared the CPU times for the vectorized version of our code and for the same code with vectorization inhibited. The speed-up of the vectorized code over the non vectorized one was 5.5, on the Cray J90 at COPPE.

CONCLUSIONS

We have applied an elliptic scheme of numerical grid generation in order to solve the transient heat conduction problem in a piston of a diesel engine, subjected to time-averaged boundary conditions.

The analysis performed reveals that, for the cases studied, the steady-state was reached earlier for water-cooled engines than for air-cooled engines. The temperature levels are higher in the air-cooled engine piston; but the water-cooled engine piston is subjected to larger radial temperature gradients, and consequently, to larger thermal stresses.

The use of a vector version of Thomas algorithm to solve the tri-diagonal systems resulting from the discretization with the ADI method, resulted in a speed-up of 5.5 over a non-vector version of the same code.

Next steps to be taken as a continuation of the present work are the study of the piston subjected to time-varying boundary conditions; and the solution of the inverse problem of estimating such boundary conditions.

ACKNOWLEDGEMENT

The CPU time for this work has been provided by NACAD-COPPE-UFRJ.

REFERENCES

Borman, Gary and Nishiwaki, Kazuie, 1987, "Internal Combustion Engine Heat Transfer", Prog. Energy Combust. Sci., Vol. 13, pp. 1-46.

Colaço, M.J., Orlande, H.R.B. and Cotta, R.M., 1995a, "Software para geração de malhas bidimensionais em geometria irregular pelo método de diferenças finitas - 2DGRID", Anais do II Congresso Regional de Estudantes de Engenharia Mecânica, UFRJ, pp.117-121.

Colaço, M.J., Orlande, H.R.B. and Cotta, R.M., 1995b, "Software para estudo de difusão de calor em geometrias irregulares usando diferenças finitas", Anais do II Congresso Regional de Estudantes de Engenharia Mecânica, UFRJ, pp. 317-322.

Heywood, J.B., 1988, "Internal Combustion Engine - Fundamentals", MC Graw -Hill, New York.

Kornhauser, A.A. and Smith, J.L., 1994, "Application of a Complex Nusselt Number to Heat Transfer During Compression and Expansion", ASME J. Heat Transfer, Vol. 116, pp.536-542.

Maliska, C.R., 1995, "Transferência de Calor e Mecânica dos Fluidos Computacional", LTC, Rio de Janeiro.

Ortega, J.M., 1988, "Introduction to Parallel and Vector Solution of Linear Systems", Plennum Press, New York.

Özisik, M.N., 1994, "Finite Difference Methods in Heat Transfer", CRC Press, Boca Raton.

Özisik, M.N., 1993 "Heat Conduction", 2nd edition, McGraw-Hill International Editions, New York.

Peaceman, D.W. and Rachford, H.H., 1955, "The Numerical Solution for Parabolic and Elliptic Differential Equations", J. Soc. Ind. Appl. Math, Vol.3, pp. 28-41.

Prasad, R. and Samria, N.K., "Transient Heat Transfer in an Internal Combustion Engine Piston", 1990, Computers & Structures, Vol. 34, no. 5, pp. 787-793.

Singh, V.P., Upadhyay, P.C. and Samria, N.K., 1986, "Some heat transfer studies on a diesel engine piston", Int. J. Heat Mass Transfer, Vol. 29, No. 5, pp. 812-814.

Thompson, J.F., Warsi, Z.U.A. and Mastin, C.W., 1987, "Numerical Grid Generation", North-Holland, New York.