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HEAT TRANSFER ANALYSIS OF TESTING BENCH COMPONENTS OF AN INTERNAL COMBUSTION ENGINE

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ABSTRACT

The purpose of the work presented in this study is related to the distributions of temperature in the various components of an internal combustion engine. We are interested in a first case to the heat transfer processes in the various components of an engine (cylinder head, piston, cylinder liner and valve) with a finned surface of a cylinder liner, and in a second case to the heat transfer in a cylinder liner of a testing bench engine with a water cooled system. In order to appreciate the influence of some parameters on heat transfer in the engine of the first case, a sensitivity study to some parameters of the problem is done. This sensitivity study approach is related to the influence of the length of the cylinder liner, the fined surface and the fined spacing. To solve partial non linear differential equations given by the mathematical modelling of these phenomena, an algorithm proposed by PATANKAR is used. Obtained results are compared to those obtained by others. A good agreement is observed between the various results.

Keywords: Heat transfer, Engine, Mathematical Modelling, Finite volume method.

1. INTRODUCTION

Internal combustion engine are subjected to gases at high temperature, following of combustion process in these engines. Hot gases, confined by engine walls, can induce disturbances in functioning sets (determined by mechanical and tribological considerations) and other complicated problems like fissuration in cylinder- head. Then, it is necessary (for the correct functioning of this device) that mechanical structures must be maintained at moderated temperatures [1, 2].

The aim of this study, is to elaborate numerical codes that can permit to understand heat transfer phenomena in internal combustion engines, particularly the determination of heat and temperature fields in all principal components of the device (piston, cylinder, cylinder- head and walls) for a given functioning data These studies must permit to recognize the different modes of heat transfer that can take place in these mechanisms, in order to control all physical processes.

Indeed the complexity of transfer process in these problems (on experimental point of view and the expensive cost to do experimental testing in the laboratory), the mathematical modelling and numerical simulation of this phenomena in a real situation, can constitute an interesting field.

Testing bench of variable compression ratio engine between 5 and 18, named TD 43, available in mechanical engineering department of our establishment (National Polytechnic School of ALGERIA), is built starting from a marine diesel engine at four times cycle and of “Farryman A30” type. It is cooled par water and can use different fuels like: gasoline, propane or natural gas. This testing bench must allow, after equipping it with data processing acquiescing systems, to valid the obtained theoretical results and to test the performance of engines using non pollutant fuels.

This paper is devoted to the distributions of temperature in the various components of an internal combustion engine. We are interested in a first case to the heat transfer processes in the various components of an engine (cylinder head, piston, cylinder liner and valve) with a finned surface of a cylinder liner, and in a second case we are interested only to heat transfer in a cylinder liner of an engine of the testing bench named previously with a water cooled system. In order to appreciate the influence of some parameters on heat transfer in the engine of the first case, a sensitivity study to some parameters of the problem is done. This sensitivity study approach is related to the influence of the length of the cylinder liner, the fined surface and the fined spacing.

2. MATHEMATICAL MODEL

The heat transfer in the various components of the engine (cylinder head, cylinder liner, piston and valve) are of conductive type. The boundary conditions at the interfaces are of convective and radiative type and, the nature of the flows In the combustion chambers is in general rather complex (Two phase or multiphase flows). Taking into account these considerations, we were interested in a first study to the purely conductive heat transfer in the various components of the engine, using coefficients of exchange resulting from the literature [2, 3, 4]. The resulting equations are given by:

$$\rho C \frac{DT}{Dt} = \nabla .(\lambda .\nabla T) \qquad (1)$$

3. INITIAL AND BOUNDARY CONDITIONS

3.1. The cylinder liner

The cylinder liner can be appeared in the form of a boring in the driving block or as a cylindrical barrel. Dimensions are imposed by the couple and the power which the engine has to provide.

With regard to our study, we have chosen two cases: the cylinder with finned surfaces of light vehicle's engine for the first case and the cylinder with water cooled system of an engine testing bench. Figures 1 and 2, show the outline of the studied cylinders.

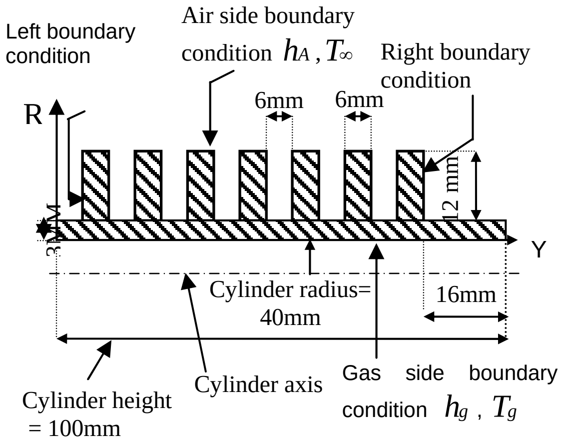

Figure 1. Cylinder liner section representing the domain to be studied (first case)

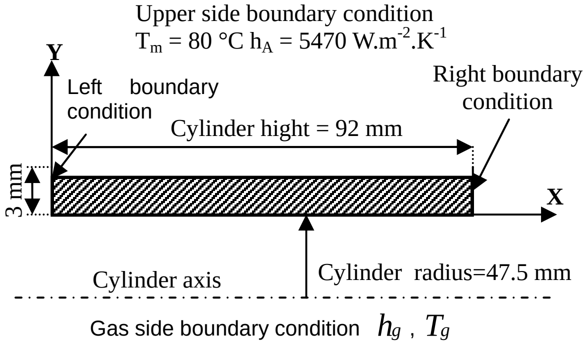

Fig 2: Cylinder liner section representing the domain to be studied (second case)

For the first case, the air side's boundary conditions, the external part of the shirt (i.e. the finned wall) is in contact with an air whose temperature, T_∞= 313K and whose coefficient of transfer h_A is constant and equal to 35 w/m².K. These conditions can be considered as a fixed parameter and of course an approximation of the reality because of their dependence of the engine speed.

The lower limit corresponds to the border with the combustion chamber, a temperature of gas, T_g and a coefficient of transfer, h_g.

For the left boundary condition, radial conduction being more significant than longitudinal conduction, we have neglected the heat flow along the cylinder liner. The cylinder liner being separated from the cylinder head by an insulating seal, the condition of insulation is thus envisaged.

For the second case, the external surface of the cylinder is in contact with cooled water whose temperature, T_∞= 353K and whose coefficient of transfer h_A is constant and equal to 5470 w/m².K. The lower and the left boundary condition are the same as the first case.

3.2. The Piston

The piston of an internal combustion engine has complex and different geometries according to the engine's design.

Figure 3, shows the outline of the studied piston, the field being rectangular and present geometrical irregularities. For reasons of symmetry, we studied only one part.

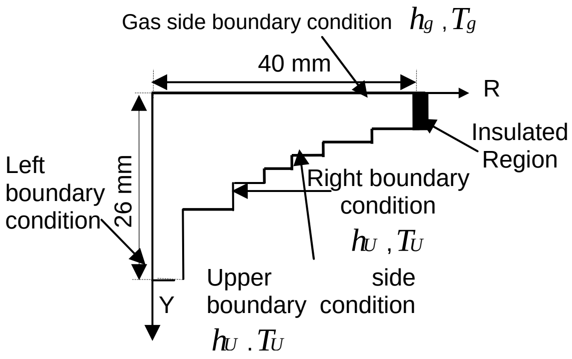

Fig 3: Piston section representing the domain to be studied.

For the initial condition, and in the first step, we have considered a linear distribution of temperature according to the axial direction and we have neglected radial and tangential conduction.

For the left boundary condition, the wall of the piston is in contact with the lubricating oil whose temperature T_u and the coefficient of transfer h_u depend on its viscosity and the mode of rotation of the engine.

The lower side boundary condition is a condition of convection with a temperature T_g and a coefficient of transfer h_g. We consider also that the axial flow is more significant than the radial, which leads us to suppose that radial flow is negligible. Taking into account the symmetry problem, the area corresponding to the right limit is considered adiabatic.

3.3. The Cylinder Head

The geometry of the cylinder head that we have chosen is simple (see figure 4), the domain to be studied is rectangular and is not presenting any geometrical irregularity.

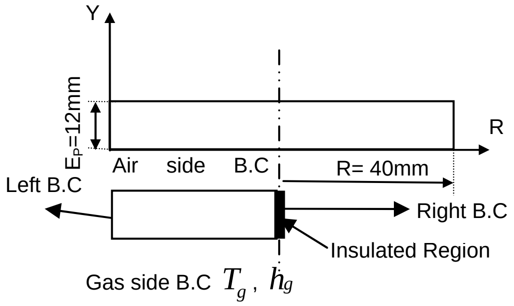

Fig 4: Cylinder head section representing the domain to be studied.

We have considered a linear distribution of temperature according to the axial direction and we have neglected radial and tangential conduction, for the initial condition.

For the boundary conditions: at the higher border, a condition of convection due to the ambient conditions caused by the surrounding air reigns.

The lower wall, corresponds to a condition of convection generated by waste gases, with a variable temperature T_g and a variable coefficient of transfer h_g. For the left limit, the longitudinal heat flow is supposed to be negligible in front of the inflow; the right limit, corresponds to an adiabatic condition, like previously, taking into account the symmetry of the problem.

3.4. The Valve

The valves of an internal combustion engine have rather complex geometries. For our study, we have chosen rectangular domain presenting certain geometrical irregularities (see figure 5).

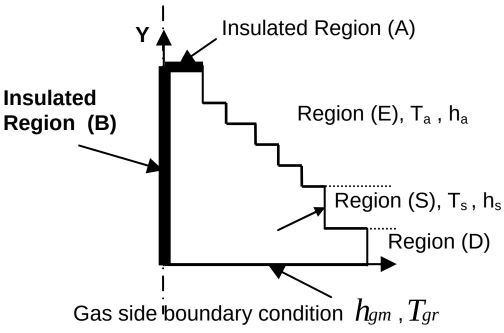

Fig 5: Valve section representing the domain to be studied.

For the valve, we have considered radial conductive heat transfer in the first part of the component in contact with gas and solid walls. A linear distribution of temperature for the initial condition is considered.

For the boundary conditions, the areas (A) and (b) are adiabatic, area (e) is characterized by a condition of convection with a temperature T_a and a coefficient of transfer.

The area (S), of contact with the seat, is determined by a condition on the temperature T_s and on the coefficient of exchange h_s.

The area (D), where the combustion gases reign, relates to a convective heat exchange with a temperature T_{gr} and a coefficient of transfer $h_{gm.}$

The lower boundary condition corresponds to a condition of convection with a temperature T_{gr} and a coefficient of transfer h_{gm}.

4. NUMERICAL SIMULATION

The problem is solved numerically by the use of the finite volume method, jointly with an algorithm proposed by PATANKAR [5].

5. RESULTS AND DISCUSSION

The presented results are related to the distributions of temperatures in the bodies of an internal combustion engine and for two engines characteristics.

The first characteristics engine related to the air cooled system with finned surfaces of the cylinder liner are as follows:

- Compression Ratio: 19.5 ;
- Operating speed: 3000rev/mn ;
- Power: 110cv ;
- Thermal conductivity: 39kcal/m.h.°C ;
- Density: 0.7800kg/m^3 ;
- Specific heat: 0.113 kcal/kg.°C.

The second characteristics engine are those of a marine diesel engine of the testing bench engine at four times cycle and of “Farryman A30” kind with water cooled system. These are as follows:

- Compression Ratio: 5 to 18 ;
- Operating speed: 2000rev/mn ;
- Thermal conductivity: 48w/m.K ;
- Density: 7800kg/m^3 ;
- Specific heat: 460 J/kg.K.

Figure 6, shows the distribution of the isotherms on the longitudinal section of the cylinder liner with an air cooled system at a given moment. We can note that the isotherms are relatively parallel between them, except for the curves to the tops of 800 °c which are slightly moved and form different geometries because we approach the wall where the temperature is of the waste gases.

Figure 7, shows the distribution of the isotherms on the longitudinal section of the cylinder liner with a water cooled system, and corresponding to the second case of the study were we can note a better heat exchange in this case.

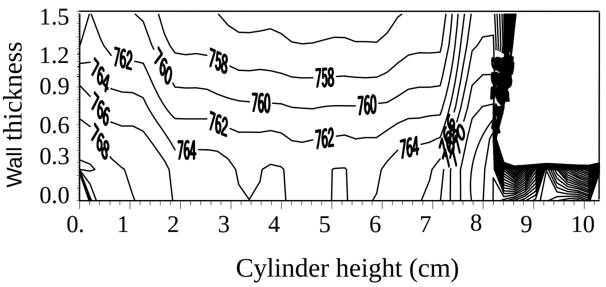

Fig 6: Temperature Distribution in the cylinder (first case) for: t = 30 s

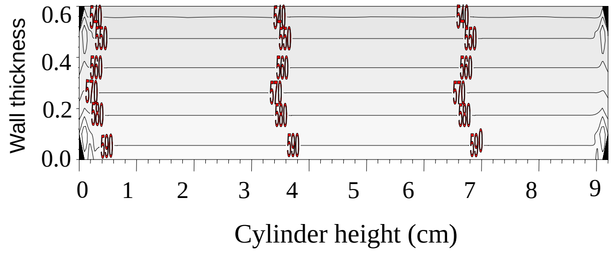

Fig 7: .Temperature Distribution
in the cylinder (second case) for: t = 30 s

Figure 8, shows the parallelism of the isotherms in the piston. On the level of the surface layers of the head of the piston, there is a non equilibrium thermal zone, the steady state mode is reached after approximately 4 minutes.

The same remarks concerning the parallelism of the isotherms can be made for figure 8, relating to the distributions of the temperature through the walls of the cylinder head.

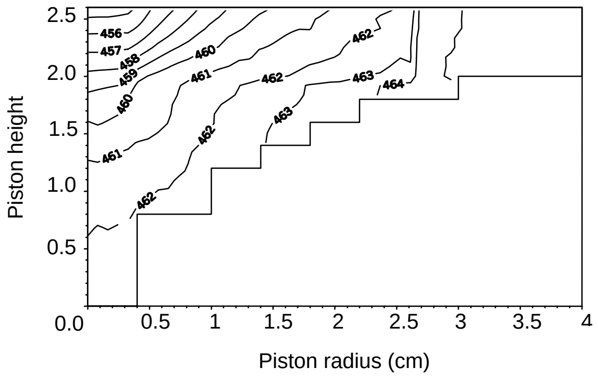

Fig 8:TemperatureDistribution
in the piston (first case) for: t = 60 s

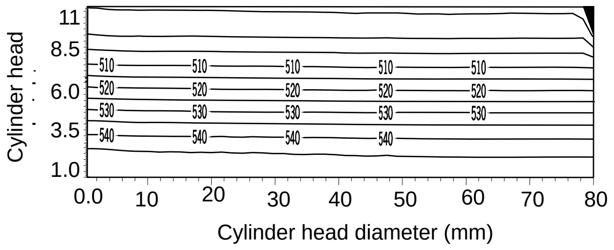

Fig 9:Temperature Distribution in the cylinder head (first case) for: t = 180 s

For figure 10, related to the temperatures in the valves, the parallelism of the isotherms, the steady state mode is reached quickly [6].

Figure 11, shows the comparison between obtained results with those of literature. We can note the good agreement between all results

Figures 12, 13 and 14, represent the distributions of temperature in a cylinder liner of 200 mm height and with 3 fines of height equal respectively to 7mm, 12mm and 17mm and for 10s of time where we can note the direct influence of the height 's fine on heat transfer.

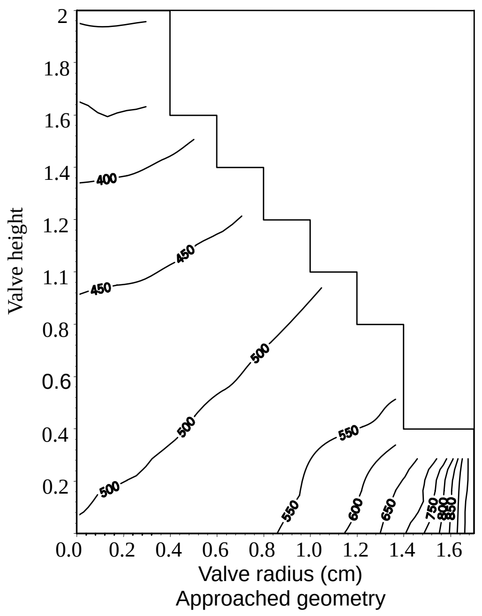

Fig 10: Temperature Distribution
in the valve (first case) for: t = 5 s

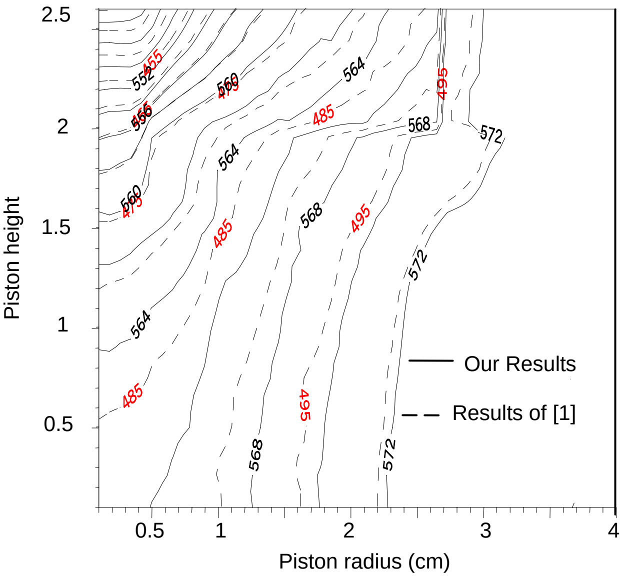

Fig 11: Comparison between results
for piston for: t= 10s

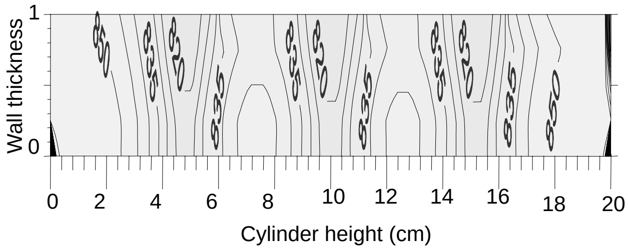

Fig 12: Temperature Distribution in the cylinder for 3
fines of 7 mm height and for a time of 10 s.

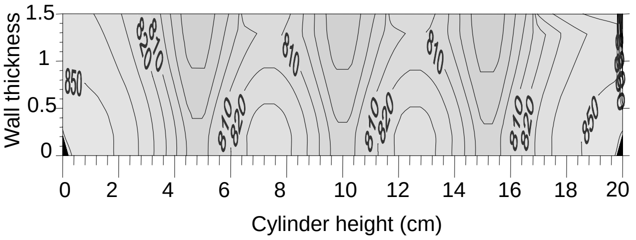

Fig 13: Temperature Distribution in the cylinder for 3 fines of 12 mm height and for a time of 10 s.

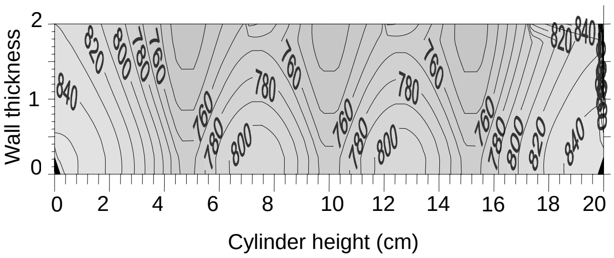

Fig 14: Temperature Distribution in the cylinder for 3 fines of 17 mm height and for a time of 10 s

Figures 15, 16 and 17, represent the distributions of temperature in a cylinder liner of 200 mm height and with 9 fines of height equal respecttively to 7mm, 12mm and 17mm and for 10s of time where we can note the direct influence of number of fines on heat transfer.

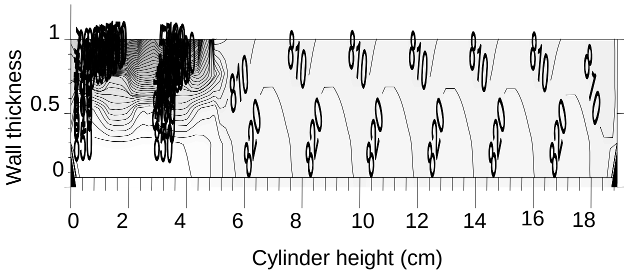

Fig 15: Temperature Distribution in the cylinder for 9 fines of 7 mm height and for a time of 10 s.

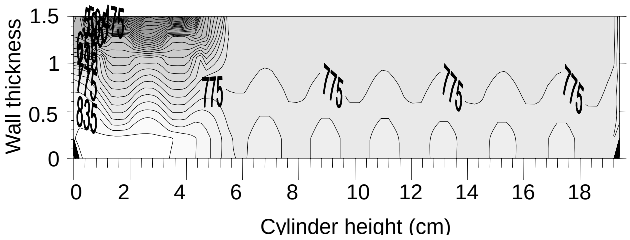

Fig 16: Temperature Distribution in the cylinder for 9 fines of 12 mm height and for a time of 10 s

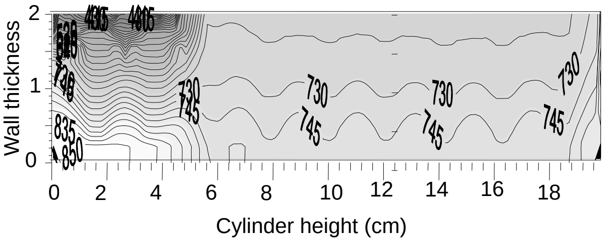

Fig 17: Temperature Distribution in the cylinder for 9 fines of 17 mm height and for a time of 10 s.

6. CONCLUSIONS

The work presented in this paper consists of the study of the thermal transfers in various bodies of an internal combustion engine.

Obtained results, relating to the distributions of temperature in the various bodies showed that, the steady state mode, in a general way is reached rather quickly.

The increase in the number of fines on the surface of the cylinder involves an increase in heat transfer exchange and the height of fines influences considerably the distribution of the temperature through the walls of the cylinder. According to the results obtained, we underline that as we increase the height of fines, we obtain better heat exchange and the steady state regime is reached more quickly for less high fines.

7. REFERENCES

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8. NOMENCLATURE

Symbol	**Meaning**	**Unit**
C	Specific heat	(J.kg $^{-1}$.K)
T	Temperature	(K)
D/Dt	Total derivative	
q	Heat flow density	(w.m $^{-2}$)
h	Coefficient of heat transfer	(w.m^{-2}K^{-2})
λ	Thermal conductivity	(w.m $^{-1}$.K^{-1})
∇	Operator of derivation	