Experimental evaluation of the thermal characteristics of internal combustion engine cylinder liners

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Abstract

The thermal characteristics of internal combustion engine cylinder liners have been investigated by means of an original test apparatus. Both single alloy (aluminium alloy and cast iron) and compound liners have been tested. The thermal conductivity has been measured by means of steady state heat transfer tests, in identical conditions for each liner considered. The influence of periodically varying heat transfer conditions on the liner thermal behaviour is shown and the deviation caused by a steady state measurement is evaluated.

Riassunto

Sono state valutate le caratteristiche termiche di canne cilindro di motori a combustione interna per mezzo di una macchina di prova originale. Sono stati sottoposti a prova sia cilindri monolega (ghisa e lega di alluminio), sia cilindri composti. La conducibilità termica è stata misurata mediante prove in condizioni di scambio termico in regime stazionario, identiche per ciascuna canna considerata. È mostrata l'influenza di condizioni di scambio in regime periodico sul comportamento termico delle canne ed è calcolato lo scostamento introdotto dalla misurazione in condizioni di regime stazionario.

Introduction

The purpose of the experimental test described here is to evaluate the thermal characteristics of cylinder liners differing in material and production technology. In effect, the wall temperature depends on the thermal conductivity of the cylinder liner and influences the tribological conditions of the liner-ring-piston coupling and therefore engine life.

The following liners — either coming from dissected crankcases or made on purpose — have been tested:

— Aluminium alloy (AlSi7) crankcase cast around a cast iron liner: S (sample n° 1).

— Aluminium alloy (AlSi7) crankcase cast around a cast iron liner: D (sample n° 2).

— Aluminium alloy (AlSi7) crankcase cast around a cast iron liner: C (sample n° 3).

— Aluminium alloy (AlSi7) crankcase cast around a liner of Al-Si hypereutectic alloy (sample n° 4).

— Cast iron liner pressed into an aluminium alloy (AlSi7) crankcase (sample n° 5).

— Cast iron liner bound to an aluminium alloy (AlSi7) crankcase (sample n° 6).

— Single-piece aluminium alloy liner (AlSi7) (sample n° 7).

— Single-piece cast iron liner (sample n° 8).

They were machined to the same dimensions (within narrow tolerances) and roughness.

Experimental Method

The most suitable testing apparatus proved to be the following (Figs. 1-4):

— The liner is thermally insulated at both ends and immersed in a container filled with water — in equilibrium with melting ice — at a constant temperature of 0°C.

— The liner is full of distilled water that can be heated to 80°C by a 1500 W resistor.

— The resistor is connected to a Variac ^{TM} in order to control the electrical power.

— A high precision digital wattmeter measures the active power supplied; the steady state heat flow through the liner wall can thus be determined.

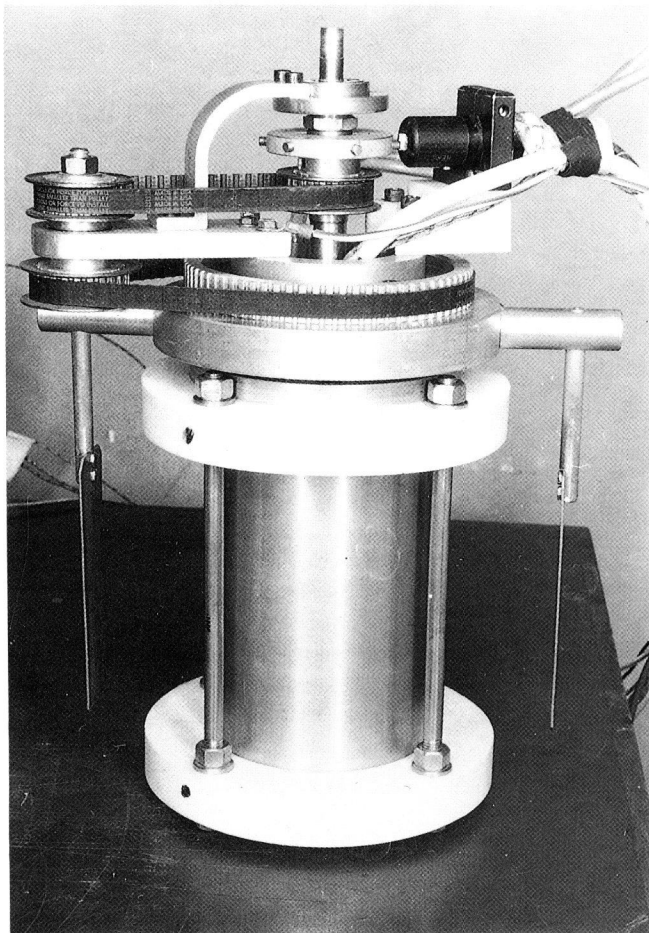Fig. 1 - Test apparatus and measurement equipment.
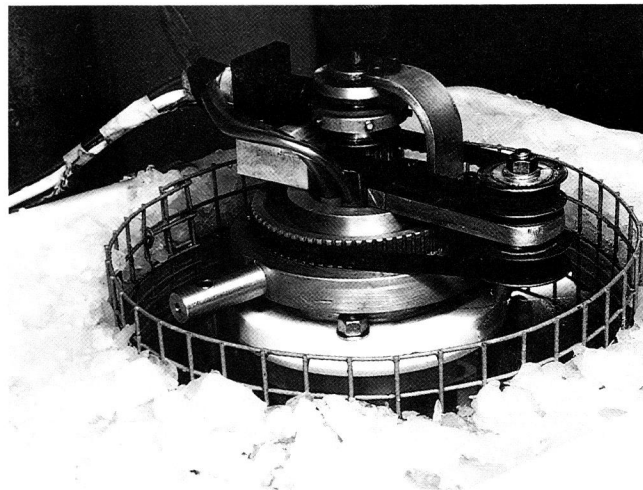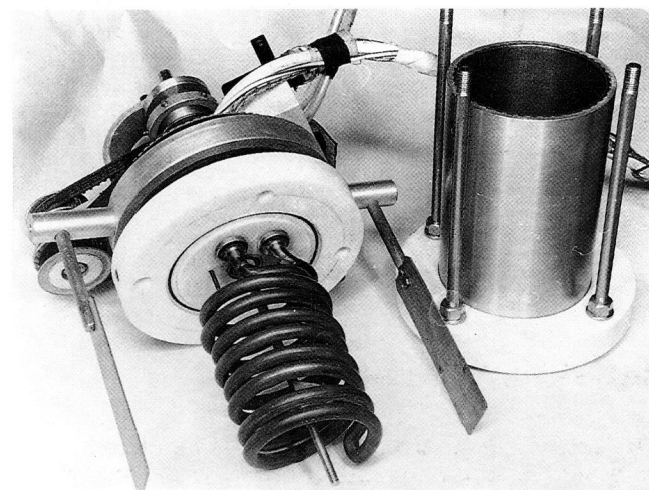
Fig. 2 - Test apparatus.

Fig. 3 - Test apparatus.

Fig. 4 - Test apparatus.

— Both inside and outside the liner, forced convection thermal exchange between resistor and water and between water and liner is granted by means of two stirrers. Upward currents of warm particles and a consequent irregular temperature distribution are therefore prevented: experimental results can only be accurate if any fluctuations of heat transfer conditions are avoided.

The upward currents become negligible by setting a sufficiently high angular speed of the stirrers so that the water temperature can be considered uniform: using various appropriately positioned thermocouples, the most suitable speed proved to be equal to 250 revolutions per minute for the inside stirrer with a 1:3 ratio between the internal and external shafts. Cavitation phenomena have been observed at higher speeds.

— Some thermocouples are positioned inside and outside the liner in order to control, with the approximation of 1°C, the temperature of the water and its uniformity.

— A metal cylindrical grid, having appropriate diameter and mesh, permanently delimits the working space of the external stirrer to prevent pieces of ice from entering it, and affecting the correct heat transfer conditions.

Test Procedure

The liners are mounted so that they can be easily interchanged. Moreover, the geometrical and physical conditions of the system are the same for any liner tested. Due to the mentioned identical conditions, the repeatability of the measurements is achieved and the

Fig. 5 - Single alloy liner.
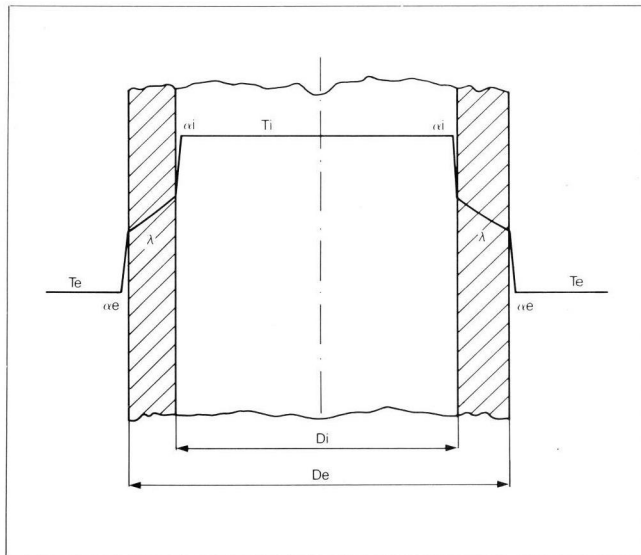
results obtained for the different liners can be compared.

Since the inevitable thermal losses are negligible values, and are anyway identical for any liner considered, the active power dissipated by the resistance, measured by the digital wattmeter, equals, at steady state conditions, the heat flow. The test must be repeated at different power values to determine the relationship between the thermal gradient and the power supplied.

For a single alloy cylinder liner (Fig. 5), the steady state heat transfer equation is:

$$\dot{Q} = 2\pi l \, K \, \Delta T \tag{1}$$

Fig. 6 - Compound cylinder liner.
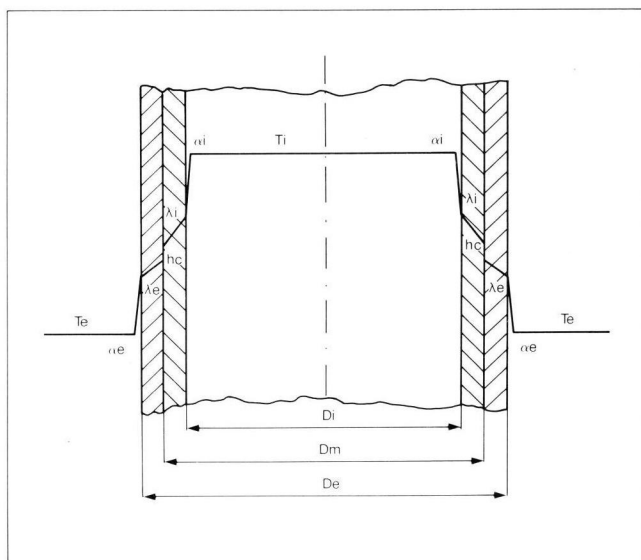
where

$$\frac{1}{K} = \frac{1}{\alpha_i R_i} + \frac{1}{\lambda} \ln \frac{R_e}{R_i} + \frac{1}{\alpha_e R_e}$$

with the following symbols:

$\dot{Q}$ = heat flow
K = total coefficient of heat transfer
l = liner height
α_i, α_e = convection coefficients
R_i, R_e = internal and external radii
λ = thermal conductivity.

For a compound liner (Fig. 6), it is:

$$\frac{1}{K} = \frac{1}{\alpha_i R_i} + \frac{1}{\lambda_i} \ln \frac{R_m}{R_i} + \frac{1}{h_c R_m} + \frac{1}{\lambda_e} \ln \frac{R_e}{R_m} + \frac{1}{\alpha_e R_e}$$

where h_c is the interface thermal conductivity. By defining an equivalent thermal conductivity as:

$$\lambda_{eq} = \frac{\ln \dfrac{R_e}{R_i}}{\dfrac{1}{\lambda_i} \ln \dfrac{R_m}{R_i} + \dfrac{1}{h_c R_m} + \dfrac{1}{\lambda_e} \ln \dfrac{R_e}{R_m}}$$

for a compound liner and $\lambda_{eq} = \lambda$, for a single alloy liner, it is possible to write:

$$\frac{1}{K} = \frac{1}{\alpha_i R_i} + \frac{1}{\lambda_{eq}} \ln \frac{R_e}{R_i} + \frac{1}{\alpha_e R_e}$$

The term $[(1/\alpha_i R_i) + (1/\alpha_e R_e)]$, for a given value of the thermal gradient, can be calculated by testing, as previously described, a cylinder liner for which the thermal conductivity, λ, is known. If both internal and external temperatures, stirring speed and surface roughness are kept constant for any tested liner, the term $[(1/\alpha_i R_i) + (1/\alpha_e R_e)]$ remains constant as well; therefore the equivalent thermal conductivity is given by the equation:

$$\lambda_{eq} = \frac{\ln \dfrac{R_e}{R_i}}{\dfrac{1}{K} - \left(\dfrac{1}{\alpha_i R_i} + \dfrac{1}{\alpha_e R_e} \right)}$$

Interface Resistance

An interface thermal resistance occurs when two similar or dissimilar materials are held together. When

Fig. 7 - Interface resistance.
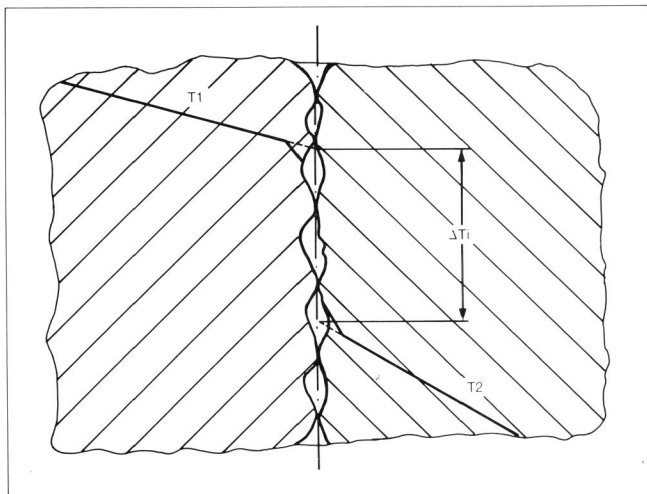
the materials are bonded, its value depends on the resistance of the adhesive, whereas for a pressure connection, the resistance is caused by imperfect contact due to the presence of macroscopic and microscopic imperfections (roughness) on the surfaces. Since in the latter case heat is transferred by conduction through the points of real contact, convection in the interstitial fluid and radiation, many parameters influence the interface resistance. These are: the contact pressure, which magnifies the extension of the true contact area, the materials considered, the surface conditions (roughness, flatness, presence of an oxide film), the mean temperature at the surface, the heat flux and the characteristics of the interstitial fluid. One way to reduce the interface resistance is to introduce a high conductivity film between the two materials.

The reciprocal of the interface resistance, i.e. the interface conductance, is defined as:

$$h_c = \dot{q}/\Delta T_i$$

where ΔT_i is the temperature gradient at the interface and $\dot{q}$ is the heat flux.

Test Results

The eight liners described have been tested according to this procedure. They all had a height of 137 mm, an internal diameter of 82.7 mm and an external diameter of 94.5 mm. The mean values of the test results, assuming a thermal gradient of 40°C for every liner and the AlSi7 liner, with a thermal conductivity of 138 W ·

$m^{-1} \cdot K^{-1}$, as reference, are listed below.

Liner	Description	λ_{eq} [$W \cdot m^{-1} \cdot K^{-1}$]
1	AlSi7 cast around a c.i. liner: S	36,7
2	AlSi7 cast around a c.i. liner: D	42,6
3	AlSi7 cast around a c.i. liner: C	23,6
4	AlSi7 cast around Al – Si hypereutectic alloy:	84,7
5	C.i. liner pressed into AlSi7	30,9
6	C.i. liner bonded in AlSi7	16,2
7	AlSi7 single alloy liner (reference)	138
8	Cast iron liner	46,3

A certain degree of dispersion has been noticed in the experimental results — the tests have been repeated a few times — of the liners number 2 and 4. This may be due to their anisotropy causing different conditions of thermal exchange when the angular position is changed with respect to the apparatus. Actually the resistance, due to its helicoidal shape, is far from having a perfect axial symmetry and its axis is not exactly coincident with the liner axis. Moreover, the geometry of the apparatus (grid, basin, etc.) may be affected by errors.

The influence of the interface thermal resistance is not negligible, as we can desume by comparing the single alloy liner with the compound ones.

The interface conductivity, on the other hand, is vastly affected by the liner production technology, as shown by the differences among the results of the three liners made of AlSi7 cast around cast iron and between the pressed and the bonded one. The latter is very interesting because of the simplified production procedures but shows a great influence of the interface adhesive on the thermal characteristics.

Appendix. Extension of the Results: Periodic Conditions

The different cylinder liners have been compared referring to steady state tests, while, as it is well

known, in their real use the liners undergo periodically varying conditions, if the engine speed is kept constant. If we consider periodic heat transfer conditions we have to take into account, in addition to the thermal conductivity, the specific heat and the density of the materials. It is therefore essential to estimate the influence of those parameters on the liner behaviour.

A comparison between the steady state and the periodic behaviour, even though the real working conditions of the engine are not considered and simplifying hypotheses are introduced, can be made referring to a single alloy liner. In a condition of periodically varying heat transfer the temperature of the inside wall of the liner can be represented by a periodic function and it is possible to write.

$$T_1 = B_1 + \sum_{i=1}^{\infty} (K_i \cos i\omega t + G_i \sin i\omega t) \quad (3)$$

where B_1 is the mean temperature of the wall.
Since the thickness of the liner is small if compared to the diameter we can refer to the flat wall theory without introducing a large error. The one-dimensional unsteady temperature distribution is therefore ruled by Fourier's equation:

$$\frac{\partial T}{\partial t} = \frac{\lambda}{\varrho c} \frac{\partial^2 T}{\partial x^2} = a \frac{\partial^2 T}{\partial x^2} \quad (4)$$

where:

$\lambda =$ thermal conductivity
$\varrho =$ density
$c =$ specifc heat
$X =$ wall thickness
$x =$ distance from the internal wall

The solution of the equation (4), assuming a periodically varying temperature at the internal wall, as described by the (3), is

$$T = B_1 - \frac{B_1 - B_2}{X} x +$$

$$+ \sum_{i=1}^{\infty} e^{-Fx} [K_i \cos (i\omega t - Fx) +$$

$$+ G_i \sin (i\omega t - Fx)]$$

where B_2 is the temperature of the outside wall and

$$F = \sqrt{\frac{i\omega}{2a}}$$

The specific heat flux $\dot q$ can be calculated by multiplying the opposite of the temperature gradient by the

thermal conductivity

$$\dot q = - \lambda \left(\frac{\partial T}{\partial x} \right)_{x=0}$$

The thermal gradient is:

$$\frac{\partial T}{\partial x} = - \frac{B_1 - B_2}{X} -$$

$$- \sum_{i=1}^{\infty} Fe^{-Fx} [(K_i + G_i) \cos (i\omega t - Fx) + (G_i - K_i) \cdot$$

$$\cdot \sin (i\omega t - Fx)]$$

Then

$$\dot q = \lambda \frac{B_1 - B_2}{X} + \lambda \cdot$$

$$\cdot \sum_{i=1}^{\infty} F[(K_i + G_i) \cos i\omega t + (G_i - K_i) \sin i\omega t] \quad (5)$$

Whatever the expression of $\dot q$, provided it is periodically varying in a period equal to $2\pi/\omega$, its mean value in the period, $\dot Q$, must be

$$\dot Q = \lambda \frac{B_1 - B_2}{X}$$

Then the mean temperature gradient is

$$B_1 - B_2 = \frac{X \dot Q}{\lambda}$$

and it depends on the conductivity but not on the density and specific heat. It is easy to calculate the expression of the instantaneous temperature gradient by considering, as a condition of the second kind (given flux), the simple case of a sinusoidally pulsating flux (angular frequency equal to ω)

$$\dot q = \dot Q (1 - \cos \omega t) \quad (6)$$

By combining equations (5) and (6), it follows that

$$T_1 = B_2 + \frac{\dot Q X}{\lambda} + \frac{\dot Q}{2\lambda F} (\cos \omega t + \sin \omega t)$$

and

$$T_1 = B_2 + \frac{\dot Q X}{\lambda} + \frac{\dot Q}{\sqrt 2 \lambda F} \cos \left(\omega t - \frac{\pi}{4} \right)$$

By remembering that

$$F = \sqrt{\frac{\omega}{2a}} = \sqrt{\frac{\omega \varrho c}{2\lambda}}$$

it is possible to express the temperature gradient as

$$T_1 - B_2 = \frac{\dot{Q}X}{\lambda} + \frac{\dot{Q}}{\sqrt{\omega \varrho c \lambda}} \cos\left(\omega t - \frac{\pi}{4}\right)$$

and its maximum value is

$$T_{1max} - B_2 = \frac{\dot{Q}X}{\lambda} + \frac{\dot{Q}}{\sqrt{\omega \varrho c \lambda}}$$

The rate between the maximum and mean values of the temperature wall gradient is therefore given by

$$\frac{\Delta T_{max}}{\Delta T_{med}} = \frac{T_{1max} - B_2}{B_1 - B_2} = 1 + \frac{1}{X}\sqrt{\frac{\lambda}{\omega \varrho c}}$$

and since the thermal diffusivity is

$$a = \frac{\lambda}{\varrho c}$$

it results that

$$\frac{\Delta T_{max}}{\Delta T_{med}} = 1 + \frac{\sqrt{a}}{X\sqrt{\omega}}$$

The equation (7), even though referred to simplified flux conditions, is helpful in evaluating the order of the influence of the angular frequency, ω, on the real maximum wall temperature.

Let us evaluate the rate given by equation (7) by introducing the following values:

$X = .01$ m
$\omega = 100$ rad $\cdot$ s^{-1}

and let us consider two different materials:

1. Cast iron

$\lambda = 50$ W $\cdot$ m^{-1} $\cdot$ K^{-1}
$c = 586$ J $\cdot$ kg^{-1} $\cdot$ K^{-1}
$\varrho = 7.2 \cdot 10^3$ kg $\cdot$ m^{-3}

we have

$$\frac{\Delta T_{max}}{\Delta T_{med}} = 1.034$$

2. Hypereutectic alloy AlSi21

$\lambda = 126$ W $\cdot$ m^{-1} $\cdot$ K^{-1}
$c = 920$ J $\cdot$ kg^{-1} $\cdot$ K^{-1}
$\varrho = 2.68 \cdot 10^3$ kg $\cdot$ m^{-3}

we have

$$\frac{\Delta T_{max}}{\Delta T_{med}} = 1.071$$

By considering steady state conditions we therefore introduce a small error.

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