

Experimental Assessment of Instantaneous Heat Transfer in the Combustion Chamber and Exhaust Manifold Walls of Air-Cooled Direct Injection Diesel Engine

G. C. Mavropoulos, C. D. Rakopoulos and D. T. Hountalas
National Technical University of Athens

Copyright © 2008 SAE International

ABSTRACT

An experimental analysis is carried out to investigate several heat transfer characteristics during the engine cycle, in the combustion chamber and exhaust manifold walls of a direct injection (DI), air-cooled, diesel engine. For this purpose, a novel experimental installation has been developed, which separates the engine transient temperature signals into two groups, namely the long- and the short- term response ones, processing the respective signals in two independent data acquisition systems. Furthermore, a new pre-amplification unit for fast response thermocouples, appropriate heat flux sensors and an innovative, object-oriented, control code for fast data acquisition have been designed and applied. Experimentally obtained cylinder pressure diagrams together with semi-empirical equations for instantaneous heat transfer were used as basis for the calculation of overall heat transfer coefficient. On the other hand, one-dimensional heat conduction theory with Fourier analysis techniques combined with an iterative procedure between calculated and measured temperature data are implemented, in order to calculate the instantaneous local engine cylinder heat transfer coefficients in the cylinder head surfaces. For the exhaust manifold, the local gas temperature was calculated based on a thermal-zone model approach developed. Analysis of experimental results revealed significant differences between the overall and local peak heat transfer coefficient values in the cylinder head surface. The effect of engine speed and load as well as the effect of air-swirling motion on cylinder head heat transfer coefficient variation are presented and quantified, revealing the significant influence of turbulence on local heat transfer conditions. The effect of valve motion on instantaneous heat transfer is also depicted and quantified. Analysis of the results for the instantaneous wall temperatures and heat transfer coefficient on the exhaust manifold reveals interesting details regarding the flow of exhaust gases, both in the blowdown and displacement phases of the engine exhaust stroke. From the results it is concluded that the

instantaneous heat transfer coefficient variation for engine combustion chamber is highly non-uniform, unlike its values calculated from standard correlations that assume spatial uniformity. This is very important, since especially for air-cooled diesel engines limited relevant information seems to exist in the literature.

INTRODUCTION

Heat transfer to the combustion chamber walls of internal combustion engines is one of the most important aspects influencing greatly both engine design and operation. This importance has been well recognized from the early stages of engine development [1*-3]. Results from various theoretical and experimental research efforts on this field have been presented during the last decades in the literature by several researchers [4-7]. It is generally recognized that the problem is quite difficult and many of its aspects still remain to be analyzed and elucidated.

During the recent years, research efforts concerning heat transfer phenomena in reciprocating internal combustion engines have been intensified, because of their major importance, among other aspects, on successful thermodynamic cycle simulations [8-10] and investigations of thermal loading aiming at securing structural integrity of critical combustion chamber components [11-13]. The complexity of heat transfer in internal combustion engines is very involved, since the relevant phenomena are transient, three-dimensional, and subject to rapid swings in cylinder gas pressure and temperatures [14,15], while the combustion chamber itself with its moving boundaries adds further to this complexity. It was successfully written that: "the problem is a modeler's nightmare and an experimenter's agony, but is far too important to ignore" [16].

Concerning diesel (compression-ignition) engines, and more specifically the air-cooled ones, it appears that

* Numbers in brackets designate references at the end of the paper

limited reports exist in the literature dealing with them [17-19]. At the same time, new demands are added in today's diesel engine design mainly towards the areas of controlled ignition of new and alternative fuels [20,21], reduction of tailpipe emissions [8,10,22] and application of ceramic materials in the combustion chamber surfaces of low heat rejection (LHR) engines [23-25]; all of these are directly related to the details of the transient cycle heat transfer. Especially, the development and marketing of low heat rejection (or 'adiabatic') diesel engines has altered dramatically the way the engine heat transfer has been considered up to this moment. Thus, these engines require proper selection or development of new materials with extremely high thermal resistance, with the result for the size of thermal gradients introduced in the combustion chamber surface layers to have currently been increased almost by an order of magnitude.

Other new engine development areas in which recently engine heat transfer studies are gaining an increased importance are resulting from engine downsizing that has enforced higher operating temperatures of engine parts and also from HCCI engines due to the importance of local heat flux variation on mixture stratification. Especially, the stringent pollutant emission levels imposed by upcoming regulations (EURO V and VI) [26,27] point out the need for extensive heat transfer studies, specifically on the short-term transient variations introduced during an engine cycle [28].

A special part of engine heat transfer studies concerns the intake and exhaust engine manifolds. The importance of gas exchange subsystem in the engine development and operation has already been recognized many years ago. In the most recent years, a series of developments in this system have been performed with the aid of advanced modeling methods and detailed experimental data. The phenomena of transient heat transfer in intake and exhaust engine manifold are of special difficulty due to the complex dynamic nature of gas flow inside them. The detailed knowledge of heat transfer distribution would help utilizing pressure pulses more effectively in tuning the exhaust manifold. Many interesting developments have been recently reported by Alkidas et al. [29,30] and other researchers [31].

The present authors have initiated some years ago a general research program aiming at the investigation of the basic heat transfer mechanism as well as the thermal loading parameters in diesel engines, focusing on problems associated with today's technological innovations. Within this framework, they have already reported detailed structural and thermodynamic analysis models [32-34], which are also capable to take account of both the operating and design engine characteristics influencing the heat transfer behavior of the insulated engine [35]. They have also reported in detail the short-term variation of instantaneous diesel engine heat flux and temperature during an engine cycle [36-38], and presented interesting phenomena concerning both the engine cylinder and exhaust manifold.

The experimental analysis for the local heat transfer coefficient provided in the present work for the cylinder head and exhaust manifold promote further the investigation of heat transfer characteristics inside the internal combustion engine. Thus, in similar studies existing in today's literature, theoretical models (mainly Computational Fluid Dynamics (CFD)) are used to derive the spatial and temporal variation of heat transfer coefficient inside the combustion chamber and exhaust manifold [39-41]. On the other hand, the experimental heat transfer data provided are related almost exclusively to heat flux and temperature variations, since these are the primary experimental data.

The experimental heat transfer coefficients provided in the present work unveil, additionally, a lot of interesting phenomena, like the effect of engine speed and load as well as of the intake air swirl on the mechanism of local cylinder heat transfer. For the exhaust manifold heat transfer, the analysis reveals an essential difference between the heat flux and heat transfer coefficient distributions. The primary dependence of exhaust manifold heat transfer coefficient on gas velocity is verified from the analysis of the present experimental data.

THE SIMULATION MODEL

A model for the processing and analysis of experimental data has been developed by the authors. Some of its aspects have already been reported in previous publications [32,34,36]. The procedure applies the basic thermodynamic and heat transfer principles to satisfy the processing demands of the experimental investigation.

The procedure consists of the following four main sub-models:

- Analysis of cylinder pressure data
- Time periodic heat conduction problem
- Instantaneous heat transfer coefficient (including spatial distribution) in the cylinder head, and
- Instantaneous heat transfer coefficient in the exhaust manifold.

An outline of their basic characteristics is provided in the next subsections.

ANALYSIS OF CYLINDER PRESSURE DATA - The procedure for the analysis of cylinder pressure data has been developed by the authors in the past [34] where it was used for the analysis of experimental measurements taken during an engine transient, i.e. long-term response that contains variations of engine speed and/or load. In the present work, a simpler form of this method is used for the calculation of the (spatially) overall value of heat transfer coefficient during an engine cycle, i.e. with reference to a short-term response transient. Thus, in the present investigation engine speed and load values remain constant during each measurement and when the latter is finished their values are changed to the next operating point to cover adequately the whole engine cycle of operation.

The method is based on the processing of experimental cylinder pressure diagrams, which computes the estimation of heat release rate and boundary conditions (gas temperature, heat transfer coefficient) variations during an engine cycle. As a result, the power and fuel engine consumption values can also be determined (not used in the present case). A short description of its basic steps is given hereafter. More details can be found in [34].

Top dead center, engine speed, angular acceleration -

The experimental results for the cylinder pressures for several (5 to 6) engine cycles are recorded in a contiguous file. Initially, the measured pressure values are corrected for any drifting of the signal obtained from the charge amplifier. The position of TDC is indicated by a sign obtained from a magnetic pick-up. This sign presents a sudden change from its maximum (positive) to its minimum (negative) value. By taking the bisection of these two values, the exact TDC position is determined. By considering also the TDC position of any two consecutive cycles and knowing the sampling rate of the measurements (ksamples/sec), it is apparent that the engine speed of the (four-stroke) engine can be easily calculated. Taking the first derivative of speed with respect to time, the corresponding angular acceleration curve is also calculated, which obviously is zero in the present case (due to steady-state operation). Owing to the high signal resolution used in the present experimental investigation (0.25 deg CA in most operating cases), the determination of TDC position and exact engine speed was performed independently for every one of the operating cases examined. Thus, a high degree of accuracy was obtained by excluding error sources introduced from engine speed variations around its mean (nominal) value.

Gas temperature and heat transfer coefficient - The gas temperature can be determined using the ideal gas state equation:

$$p(\varphi)V(\varphi) = m(\varphi)RT_g(\varphi) \Rightarrow T_g(\varphi) = \frac{p(\varphi)V(\varphi)}{m(\varphi)R} \quad (1)$$

For the estimation of the gas temperature from Eq. (1), the instantaneous mass must be known with accuracy. For the gas exchange period, the mass flow to or from the engine cylinder is calculated using a detailed simulation model [8-10]. An initial estimate of mass is accomplished at the point of IVC, where the corresponding conditions (pressure and temperature) for the specific engine are calculated from a constants determination procedure developed by the authors [9]. During the compression and expansion periods (closed cycle), the instantaneous mass at any crank angle φ is calculated from

$$m(\varphi) = m(\varphi_{IVC}) - \int_{\varphi_{IVC}}^{\varphi} \frac{dm_e}{d\varphi} d\varphi \quad (2)$$

where φ_{IVC} is the crank angle at inlet valve closure and $dm_e/d\varphi$ the mass flow rate from the cylinder due to blowby expressed in kg/deg, given by

$$\frac{dm_e}{d\varphi} = \frac{(dm/dt)}{6n} \quad (3)$$

The heat transfer model of Annand is used for the calculation of instantaneous overall (spatially) heat transfer coefficient h_g [4]

$$\begin{aligned} \frac{dQ_w}{d\varphi} &= \frac{F}{6n} \left[a \frac{k_g}{D} Re^b (T_w - T_g) + c_r (T_w^4 - T_g^4) \right] = \\ &= F \frac{h_g}{6n} (T_w - T_g) \end{aligned} \quad (4)$$

where surface $F = \pi D^2/4 + \pi D x$ with x the instantaneous cylinder height in contact with the gas, k_g is the gas thermal conductivity, and the Reynolds number 'Re' is calculated with a characteristic dimension and speed equal to the piston diameter and mean piston speed, respectively. For the estimation of wall temperature T_w , the same constants determination procedure, as previously referred to, was used.

TIME PERIODIC HEAT CONDUCTION PROBLEM - To calculate the heat flux through a certain location of the combustion chamber wall, during a complete engine cycle, the time periodic (unsteady) heat conduction model is used. The unsteady heat conduction equation is solved with boundary conditions derived from the experimental temperature measurements. Assuming that heat flow through the component is one-dimensional [42] and that material properties remain constant, the corresponding expression of the unsteady heat conduction equation is given by

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \quad (5)$$

where x is in this case the distance from the wall surface, $\alpha = k_w/\rho_w c_w$ is the wall thermal diffusivity, with ρ_w the density and c_w the specific heat capacity.

The assumption for one-dimensional heat flux is very well justified for this case, since it is well known that the corresponding temperature transients penetrate only within a very small distance from the combustion chamber surface [38]. On the other hand, this assumption could not be used for the steady state component of heat flux, since it is also well-known that as the distance from the combustion chamber surface increases, high temperature gradients are developed in three dimensions inside each component. A solution in this case is to use a measuring probe carefully insulated to ensure one-dimensional heat flow. The boundary conditions for Eq. (5) can then be written as

$$\left. \begin{aligned} T(0,t) &= T_w(t), & \text{at } x = 0 \\ T(\delta,t) &= T_\delta = \text{const}, & \text{at } x = \delta \end{aligned} \right\} \quad (6)$$

where δ is the distance from the wall surface of the in-depth thermocouple, which provides constant temperature readings.

Using Fourier analysis, the measured wall surface temperature is expressed in the following form [36,37]:

$$T_w(t) = T_m + \sum_{n=1}^N [A_n \cos(n\omega t) + B_n \sin(n\omega t)] \quad (7)$$

where T_m is the time averaged value of T_w , A_n and B_n are the Fourier coefficients, n is the harmonic number, N is the total number of harmonics, and ω (in rad/s) is the angular frequency of temperature variation, which for a four-stroke engine is half the engine angular speed.

The solution of heat conduction Eq. (5), subjected to boundary conditions of Eqs (6), takes then the form:

$$T(x,t) = T_m - (T_m - T_\delta) \frac{x}{\delta} + \sum_{n=1}^N \exp(-\phi_n \cdot x) G_n(x,t) \quad (8)$$

where

$$G_n(x,t) = A_n \cos(n\omega t - \phi_n x) + B_n \sin(n\omega t - \phi_n x) \quad \text{and} \\ \phi_n = (n\omega / 2\alpha)^{1/2}.$$

Finally, using the Fourier's law, the instantaneous heat flux on the combustion chamber surfaces is calculated by differentiating Eq. (8) with respect to x at position $x=0$:

$$q_w(t) = -k_w \left. \frac{\partial T}{\partial x} \right|_{x=0} = \frac{k_w}{\delta} (T_m - T_\delta) + \\ + k_w \sum_{n=1}^N \phi_n [(A_n + B_n) \cos(n\omega t) + (B_n - A_n) \sin(n\omega t)] \quad (9)$$

INSTANTANEOUS HEAT TRANSFER COEFFICIENT-CYLINDER HEAD - In order to investigate the local distribution of heat transfer coefficient in the internal combustion chamber surfaces, it is necessary to consider both spatial distribution as well as time variation of the instantaneous heat flux. In that case, the instantaneous heat flux equation takes the form:

$$q_w(t,F) = h_g(t,F) [T_g(t,F) - T_w(t,F)] \quad (10)$$

so that it allows also for spatial distribution of all variables inside the combustion chamber surfaces.

The theoretical aspects of this problem have been analyzed in detail by Morel and Keribar [40]. They have developed a model providing the spatial distribution of heat transfer coefficient in the internal surfaces of the

combustion chamber, which was proved simple and quite effective. In that model they have divided the combustion chamber area into three zones, thus calculating a different h_g value for each one of them. Their model considers the fluid motion as the dominating factor and in the end allows for the estimation of h_g in each area.

In the present work, the main interest is on the results of experimental investigation concerning local distribution of heat transfer coefficient. Thus, the following much simpler procedure for the calculation of local instantaneous gas heat transfer coefficient was used based solely on the processing of experimental data.

Initially, in Eq. (10) the variation of heat flux with respect to space and time is calculated using the procedure described already in the previous subsection concerning the time periodic heat conduction. Additionally, the wall surface temperature T_w is recorded during the experimental procedure. We notice that in order to calculate the local instantaneous value of h_g from Eq. (10), the gas temperature variation is the only variable that has to be determined. An initial distribution of the (spatially) mean cylinder gas temperature is already calculated following the analysis described previously in the corresponding subsection.

To arrive at a respective 'local' gas temperature variation, the previous procedure has to be modified only during combustion (closed cycle) according to the assumptions of Morel [40] for the combustion chamber zone under investigation. This is done in the present study using an iterative procedure developed by the authors for the mean value variations of gas temperature and heat transfer coefficient during the 720 deg CA of the four-stroke engine cycle, given from the following relations:

$$h_{mg} = \frac{1}{4\pi} \int_0^{4\pi} h_g d\phi, \quad T_{mg} = \frac{1}{4\pi h_{mg}} \int_0^{4\pi} T_g h_g d\phi \quad (11)$$

These values are used as inputs for the boundary conditions of a FEM simulation model of the combustion chamber component in question (cylinder head in the present case) and the engine steady-state condition under consideration. The solution for the temperature field using the FEM method is repeated until a reasonable agreement between measured and calculated temperature distributions is reached for the combustion chamber area of interest. The procedure has already been applied with success by the present authors in the past [33], both for steady-state and transient engine operations. Based on the up-to-date experience, three to four repetitions of this procedure are sufficient to provide satisfactory agreement of temperature values for most of the cases. This procedure is currently automated and performed via a respective software program developed by the authors. Examples from its application can be found in [33,34].

INSTANTANEOUS HEAT TRANSFER COEFFICIENT-EXHAUST MANIFOLD - For the calculation of instantaneous heat transfer coefficient variation during an engine cycle in the internal surface of the exhaust manifold (in contact with the exhaust gas), a simple model based on the processing of the experimental results has been developed by the authors.

In the case of the exhaust manifold a one-dimensional quasi-steady heat transfer simulation model was adopted, where the temperature distribution in the exhaust pipe wall was considered uniform in the circumferential direction. Therefore, exhaust gas and pipe wall temperatures are varying only with the distance from the exhaust valve and also with time. This assumption of circumferential symmetry of thermal field is close to reality, permitting the processing of the derived experimental data rather effectively. The development of a more detailed 3D theoretical model based on the principles of fluid dynamics was beyond the scope of the present investigation.

Similar to the case of combustion chamber surfaces referred to in the previous subsection, also for the case of the exhaust manifold the only problem is to reach the gas temperature variation during the engine cycle in the position where the experimental measurements were taken and, consequently, the heat transfer calculations will be applied. To accomplish this task, it is initially considered the corresponding (spatially) mean gas temperature distribution inside the engine cylinder derived from the engine simulation model using the procedure already described in the corresponding subsection. This is considered then as the gas temperature variation in the entrance of exhaust manifold, immediately after the exhaust valve. The corresponding temperature variation at the point of interest inside the exhaust pipe is derived afterwards as a sequence of independent temperatures initiated from the distribution of temperature in the point of entrance, taking also into account the corresponding time delay and the heat losses that occur between the two points. Thus, the idea finally is to break the exhaust gas stream after the exhaust valve in a sum of independent "thermal zones", each one carrying its own individual temperature and history, and follow them from the exhaust valve until the point of measurement (Fig. 1). This simple approach, of course, did not take into account some important aspects of the actual fluid flow, like the intense mixing process of the combustion gases inside the exhaust manifold and the subsequent energy distraction accompanying it. However, it provides a sufficient first approach to reach the instantaneous heat transfer coefficient variation inside the context of the present experimental investigation. Different approaches on gas temperature prediction constitute the models presented by Alkidas et al. in [43] and also by Ranganathan et al. in [44].

The time delay in the flow of gases introduced between the entrance of manifold and the measurement point is determined from the measured variation of

instantaneous manifold wall surface temperatures. The latter presents a characteristic point of minimum after the event of EVO. In the present experimental investigation, this point appears 15 deg CA after the EVO and this (as will be shown later) was found to be depended only on the measuring position and not on engine speed or load. Based on previous experience from experimental and simulation data, the corresponding minimum on exhaust gas variation will appear for the same location with a phase difference of 6 deg CA (advanced from the corresponding minimum point of wall temperature). This is a result of the thermal resistance to the transfer of heat through the boundary layer in the internal manifold wall.

Having estimated the angle shift for the gas temperature variation between the two points of interest, the estimation of the gas temperature in the measurement position is divided afterwards in two separate phases corresponding to the state of flow inside the exhaust manifold. For the part of engine cycle between the events of EVO and EVC, the flow inside the exhaust manifold is considered turbulent (which is the actual case). Thus, the estimation of gas temperature variation in the point of measurement is accomplished after accounting for the heat losses using one of the well-known relations providing the heat transfer coefficient for turbulent flow inside ducts [42]:

$$Nu = 0.023(Re)^{0.8} Pr^{1/3} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} \quad (12)$$

where the gas properties are evaluated at the bulk 'b' temperature, which is the arithmetic mean of the inlet and outlet temperatures, while subscript 's' refers to the surface temperature.

In the second phase, after the event of EVC the gas velocity inside the exhaust manifold is reduced significantly. Therefore, the gas flow is transformed to a laminar one and the corresponding heat transfer equation applied for the thermal zones considered in this stage, is [42]:

$$Nu = 1.86 \left[Re Pr \frac{D_1}{L} \right]^{1/3} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} \quad (13)$$

where L is the length of the flow path and D_1 is the equivalent hydraulic diameter, $D_1 = 4A/f$, where A is the flow surface area and f its internal (cooling) perimeter.

Finally, after the last thermal zone has passed the exhaust manifold point of measurement, a linear temperature reduction is considered for the rest of the part of the engine cycle until the next event of EVO occurs. This is used to simulate the exhaust gas cooling inside the manifold for the respective part where minimum gas velocity occurs.

After applying the above considerations, the final result is checked by comparing the mean exhaust gas

temperature values calculated using the method described above, with the actual gas temperature measurements taken during the experiments from a thermocouple placed at the point of interest. The result is displayed in Fig. 2, where a good coincidence between calculated and experimental results is shown.

Therefore, it is concluded that the method can be used as a first estimation to derive results concerning the instantaneous gas-wall temperature for a specific point inside the exhaust manifold. The heat transfer coefficient variation is finally calculated from Eq. (10).

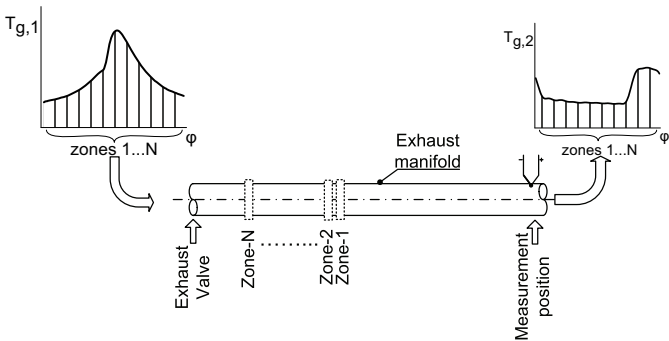


Fig.1. Graphical representation of the thermal zone model used for the exhaust gas temperature estimation in the position of measurement.

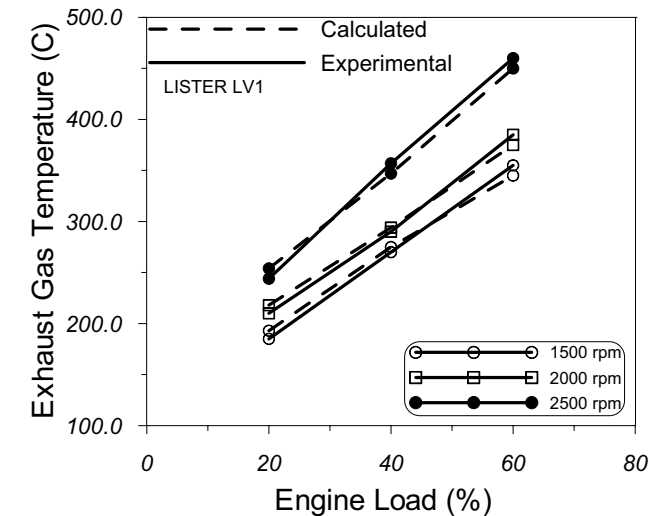


Fig.2. Comparison between calculated and experimental exhaust gas temperatures vs. engine load for the three engine speeds examined.

TEST ENGINE DESCRIPTION

The experimental investigation was conducted on a single cylinder, Lister LV1, direct injection, diesel engine. The technical data of the engine are given in Table 1. This is a naturally aspirated, air-cooled, four-stroke engine, with a bowl-in-piston combustion chamber. All the combustion chamber components (head, piston, liner etc.) are made from aluminum. The normal speed range is 1000-3000 rpm. The engine is equipped with a PLN fuel injection system. A three-hole injector nozzle (each hole having a diameter of 0.25 mm) is located in the middle of the combustion chamber head. The injector nozzle opening pressure is 190 bar. The engine is fitted

with a Bryce-Berger fuel injection pump, coupled to a variable-speed mechanical governor. The engine is permanently coupled to a Heenan & Froude hydraulic dynamometer.

Table 1: Engine basic design data of Lister LV1 diesel engine

Engine type	Single cylinder, 4-stroke, air-cooled, DI
Bore/Stroke	85.73 mm/82.55 mm
Connecting rod length	148.59 mm
Compression ratio	18:1
Speed range	1000-3000 rpm
Cylinder dead volume	28.03 cm ³
Maximum power	6.7kW @ 3000 rpm
Maximum torque	25.0 Nm @ 2000 rpm
Inlet valve opening/closing	15°CA before TDC /41°CA after BDC
Exhaust valve opening /closing	41°CA before BDC /15°CA after TDC
Inlet / Exhaust valve diameter	34.5mm / 31.5mm
Fuel pump	Bryce-Berger with variable-speed mechanical governor
Injector	Bryce- Berger
Injector nozzle opening pressure	190 bar
Static injection timing	28°CA before TDC
Specific fuel consumption	259 g/kWh (full load @ 2000 rpm)

The engine and dynamometer include the following general purpose equipment:

- Rotary displacement air-flow meter for engine air flow rate measurement
- Tank and flow-meter for diesel fuel consumption rate measurement
- Mechanical rpm indicator for approximate engine speed readings
- Hydraulic brake water pressure manometer, and
- Hydraulic brake water temperature thermometer.

EXPERIMENTAL MEASURING INSTALLATION

GENERAL - The whole measuring installation was developed in the authors' laboratory and is specially designed for addressing internal combustion engine thermal transient applications. Its configuration is based on the separation of the acquired engine signals into two main categories:

1. Long-term response ones, where the signal presents a non-periodic variation (or remains essentially steady) over a large number of engine cycles, and
2. Short-term response ones, where the corresponding signal period is one engine cycle.

The two signal categories, taken from appropriate transducers, are recorded separately via two independent data acquisition systems, appropriate

configured for each one of them, so that basically two measuring set-ups are used.

LONG-TERM RESPONSE INSTALLATION - The long term response set-up [33,34] comprises 'OMEGA' J- and K-type fine thermocouples (14 in total), installed at various positions in the cylinder head and liner in order to record the corresponding metal temperatures. These are common fine wire, shielded thermocouples, with a response time of about 5 ms. Thermocouples of the same type were also used for measuring the temperatures of the exhaust gas, cooling air inlet, and engine lubricating oil.

The extensions of all thermocouple wires were connected to a sub-multiplexer KEITHLEY EXP-16 unit for signal amplification. Their conversion to digital values is performed afterwards with the aid of a KEITHLEY DAS16-F A/D board installed on a separate PC computer. A software code was written by the authors in order to accomplish the above tasks.

SHORT-TERM RESPONSE INSTALLATION - The short-term response installation [36-38] presents the higher interest and, in general, it is the most important as far as the periodic thermal phenomena inside the engine cycle are concerned. It is evident that an experimental installation of this kind presents special difficulties due to the combination of low voltage level with the high frequency of the signals involved. The following subsections provide a description of the main aspects of this installation and the innovations developed by the authors to overcome the above mentioned problems.

Transducers and heat flux probes - The "short-term" response installation comprises the following transducers:

- "Tektronix" TDC marker (magnetic pick-up) and electronic 'rpm' counter and indicator
- "Kistler" 6001 miniature piezoelectric transducer (carefully calibrated) for measuring the cylinder pressure, which is flush mounted to the cylinder head. Its output signal is connected to a "Kistler" 5007 charge amplifier
- Four heat flux probes installed in the engine cylinder head and the exhaust manifold, for measuring the heat flux losses at the respective positions. The exact locations of these probes (HT#1 to 4) and of the piezoelectric transducer (PR#1), are shown in the layout graph of Fig. 3a and also in the image of Fig. 3b.

The design and construction of the above heat flux probes was performed at the authors' laboratory. They are customized especially for this application as shown in Fig. 4a and also in the image of Fig. 4b (for the type used on cylinder head-pos. HT#1 to HT#3). They belong to two different types as described below:

- For the engine cylinder head, the corresponding heat flux probes (HT#1-3 in Fig. 3a) consist of a fast response, K-type, flat ribbon, eroding thermocouple,

which was custom designed and manufactured for the needs of the present experimental installation [45], in combination with a common K-type, in-depth thermocouple. The response time of the eroding thermocouples, according to the manufacturer and as also confirmed by other reports in the literature [29,46], is less than 10 μ s, a fact making them very suitable for cycle instantaneous temperature measurements. Each one of them is afterwards fixed inside a corresponding compression fitting, together with the in-depth one that is placed at a distance of 6 mm apart, inside the metal volume as is shown in Fig. 4 (a and b). The heat flux sensors formed in this way are finally flush-mounted at the engine cylinder head, with the exception of HT#3 that was fixed inside the cylinder metal wall, 0.5 mm away from the gas side of cylinder head surface.

- In the case of the exhaust manifold, the heat flux sensor (HT#4 in Fig. 3a) has the same external form, except that the fast response thermocouple used is a J-type, 'coaxial' one, having a response time of the same magnitude as the eroding ones (10 μ s). This type was again customized concerning its design and manufacturing, to fulfill the needs of the present experimental installation. It is accompanied with a common J-type, in-depth thermocouple, located inside the compression fitting at a distance of 6 mm behind it. The sensor was flush-mounted on the exhaust manifold at a distance of 100 mm (when considered in a straight line) from the exhaust valve.

In the case of the K-type, flat ribbon, eroding thermocouples, the thermal junction is formed by an abrasive action across the sensing surface, with a medium-grit abrasive paper. This action causes thousands of microscopic hot-weld junctions, which join one ribbon to the other ribbon. On the other hand, in the case of the J-type, coaxial thermocouple, the thermal junction is formed with the aid of a few micrometers thickness chromium disk, vacuum-deposited at the face of the hollow iron tube and in contact with the internal constantan wire.

Both heat flux sensor types were designed and constructed in the authors' laboratory based on the one hand on the experience of other investigators in this field [28,29], but mostly on the know-how developed in the Internal Combustion Engines Laboratory of the National Technical University of Athens, throughout extensive preliminary testing procedures performed during a period of several years before reaching the final mature state for the present application.

Before being placed to their position in the cylinder head, all heat flux sensors were extensively tested and calibrated through a long series of experiments conducted in different engines, under motoring and firing operating conditions.

Signal pre-amplification and data acquisition system - In order to obtain a clear thermocouple signal, while acquiring temperature data, it was found necessary that

all fast response thermocouple signals should pass through an initial pre-amplification stage. It is again mentioned at this point that this kind of measurements are perhaps the most difficult ones to acquire, owing to the low voltage level of a thermocouple output signal combined, in the present case, with an unusual high frequency. The highest frequency value used for temperature signals acquisition during the present series of experiments was 48 ksamples/sec/channel for the case of 2000 rpm engine speed, with a signal resolution of 0.25 deg CA. This frequency magnitude is obviously quite high when compared to that for a common thermocouple signal.

To overcome this problem, a special preamplifier and signal display device was designed and constructed in the authors' laboratory, based on the 'Analog Devices' Monolithic Thermocouple Amplifiers type AD594 and AD595 for the J- and K-type thermocouples, respectively. Ten of the above amplifiers were installed on a common chassis together with necessary selectors and displays, forming a flexible device that can route the independent heat flux sensor signals either in the input of an oscilloscope for display and observation, or in the data acquisition system for recording and storage. The output of each amplifier is a signal with a magnitude of 10 mV/K which presents remarkable stability with temperature, i.e. small drift error. This pre-amplification stage involves only the fast response thermocouple signals unlike the corresponding steady-state ones from the in-depth thermocouples which, due to their low frequency, do not present this need.

The output signals from the thermocouple pre-amplifier unit, together with the magnetic TDC pick-up and piezoelectric transducer signals, are connected to the input of a 'Keithley' DAS-1801ST A/D board installed on a second PC computer, which is thus separated from that of the long-term response installation. This board can acquire input data at a total throughput rate of 312.5 ksamples/s from up to 8 differential analog inputs, by utilizing also dual-channel Direct Memory Access (DMA) operation. Using this board we obtain a high signal resolution, which corresponds to much less than 1.0 deg CA depending also on the number of simultaneously recorded signals. This high acquisition rate capability also secures the, almost, simultaneous sampling of all input signals, a fact that plays an important role in the case of high frequency transient signals recording. A graphical layout of the short-term response experimental installation is shown in Fig. 5.

For the control of this high-speed data acquisition system, a computer code was developed using the tools and features of the 'TestPoint' control software. The resultant program, written by the authors, is specially designed for transient internal combustion engine applications. It comprises an object-oriented functional code, allowing for direct on-screen selection of the desired values of all acquisition parameters, namely: engine type (two- or four-stroke); engine speed (in rpm); desired number of recorded cycles; selection of the acquired channels; sampling resolution (in deg CA).

EXPERIMENTAL DATA SELECTION AND PROCESSING

The experimental measurements covered a combination of three engine speeds, namely 1500, 2000 and 2500 rpm, with three different loads, namely 20%, 40%, and 60% of full engine load. During the design of the experiments, it was decided not to increase the load above the level of 60% in order to protect the function of sensitive fast response thermocouples and avoid their exposure to extremely high temperatures. Twenty five consecutive engine cycles were recorded for every measurement, in order to take into account the cycle-by-cycle variation of the signals. Sampling interval for all cases was 0.25 deg CA, except the 2500 rpm measurement cases where the corresponding interval was 0.5 deg CA. For the long-term response set-up where the recorded temperatures remain essentially steady during every measurement, an approximate duration of 2 sec for their acquisition was adequate. An 'experimental matrix' including several of the most important engine operating parameters as measured during the investigation, is provided in Table 2 in Appendix.

The recorded signals from the piezoelectric pressure transducer were initially adjusted to their zero level, in order to compensate for the drift introduced, and then converted to the corresponding pressure value by using the amplification constant of the 'Kistler' amplifier (for the present case this was set at 20 bar/V).

Concerning the output voltage signals from the AD594 and AD595 pre-amplifiers, they were initially converted to their corresponding thermocouple voltage values (in mV) by using appropriate transfer functions provided by the manufacturer. These transfer functions take into consideration the corresponding amplification gain for the two thermocouple types imposed by the amplifiers and an input offset induced by the amplifier characteristic. The final temperature values are calculated using the International Thermocouple Reference Tables. The resulting temperature curves were found to possess very little noise, which was an indication of the satisfactory design and setting up of the present experimental installation.

The stored data were analyzed in order to calculate the final results for heat fluxes and heat transfer coefficients. For the computation of instantaneous heat fluxes from the temperature measurements provided by the fast response thermocouples, a suitable computer program was written implementing the theory of instantaneous heat flux measurements exposed in the corresponding subsection. In the final stage of calculations, the local instantaneous heat transfer coefficient h_g is estimated via the corresponding procedures described previously for the case of cylinder head or exhaust manifold.