Irreversibility production during transient operation of a turbocharged diesel engine

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Abstract: A computer model has been developed for studying the first- and second-law balances of a turbocharged diesel engine under transient conditions. Special attention is paid to the identification and quantification of the irreversibilities of all processes and devices after a ramp increase in load. The model includes a detailed analysis of mechanical friction, a separate consideration for the processes in each cylinder during a cycle ('multi-cylinder' model) and a mathematical simulation of the fuel pump. Experimental data taken from a turbocharged diesel engine are used for the evaluation of the model's predictive capabilities. The contribution of combustion, manifolds, aftercooler and turbocharger irreversibility production is analysed using detailed diagrams. It is revealed that transient in-cylinder irreversibilities develop in a different manner compared to the respective steady-state. Combustion has always a dominant contribution but the exhaust manifold irreversibilities cannot be ignored, whereas those attributed to the inlet manifold, turbocharger and aftercooler are always of lesser importance.

Keywords: turbocharged diesel engine; transient operation; second-law; exergy; availability; irreversibilities.

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1 Introduction

The turbocharged compression ignition (diesel) engine is nowadays the most preferred prime mover in medium and medium-large units applications (truck driving, land traction, ship propulsion and electrical generation), owing to its reliability that is combined with excellent fuel efficiency. However, its transient operation is often linked with off-design (e.g. turbocharger lag) and consequently non-optimum performance. This often leads to unacceptable exhaust emissions and poor speed response and, on the other hand, points out the significance of proper interconnection between the various engine components (governor, fuel pump, turbocharger and load).

During the last decades, diesel engine modelling and experimental investigation has helped enormously the study and optimisation of transient operation. Several works have been published that deal, among other things, with fundamental or parametric studies of transient operation (Rakopoulos and Giakoumis, 1998; Rakopoulos et al., 2004; Watson, 1981; Winterbone, 1986), with experimental study of transient emissions (Arcoumanis et al., 1994) and with effect of variable geometry turbines (Bartsch et al., 1998), etc. for both load and speed changes schedules.

On the other hand, it has long been understood that traditional first-law analysis, which is needed for modelling the engine processes, often fails to give the engineer the best insight into the engine's operation. In order to analyse engine performance – that is, evaluate the inefficiencies associated with the various processes – second-law analysis must be applied (Moran, 1982; Obert and Gaggioli, 1963; Rakopoulos and Giakoumis, 2006b). For second-law analysis, the key concept is ‘availability’ (or exergy). The availability content of a material represents its potential to do useful work. Unlike energy, availability can be destroyed, which is a result of such phenomena as combustion, friction, mixing or throttling. The destruction of availability – usually termed irreversibility – is the source for the defective exploitation of fuel into useful mechanical work in an internal combustion engine. The reduction of irreversibilities can lead to better engine performance through a more efficient exploitation of fuel. To reduce the irreversibilities, we need first to quantify them. That is, we need to evaluate the availability destructions using the second-law balance.

The application of second-law analysis to the internal combustion engine operation has been the subject of various works since the mid-1980s. A thorough review can be found in Rakopoulos and Giakoumis (2006b). Many of the researchers involved have investigated the irreversibilities in production mechanism (Alkidas, 1989; Primus and Flynn, 1986; Rakopoulos and Andritsakis, 1993; Rakopoulos et al., 1993; Van Gerpen and Shapiro, 1990; Velasquez and Milanez, 1994), while Kyritsis and Rakopoulos (2001) and Rakopoulos and Kyritsis (2006) have focused on the irreversibility production mechanism for the interesting case of alternative fuels. However, all the above works have focused on the steady-state diesel engine operation and, with the notable exception of the work by Primus (1984), on the (dominant) combustion irreversibilities contribution. It seems, therefore, logical to expand on the investigation of irreversibility production mechanism so as to cover the transient diesel engine operation and include also the exergy losses of the other subsystems of a diesel engine plant.

To this aim, an experimentally validated transient diesel engine simulation code has been expanded so as to include the second-law balance (Rakopoulos and Giakoumis, 2004a,b, 2005). The code used incorporates some important features to account for the peculiarities of the transient operations. Improved relations concerning fuel injection, combustion, dynamic analysis, friction and heat transfer to the cylinder walls during transient response have been developed, which contribute to an in-depth modelling (Rakopoulos and Giakoumis, 1998, 2006a).

The analysis carried out will be given in a series of diagrams, which depict the interesting second-law values with special reference to the various irreversibility contributors of all engine processes and subsystems (inlet manifold, exhaust manifold, compressor, turbine and aftercooler). Owing to the narrow speed range of the engine in hand, mainly load increases under constant governor setting are investigated, which, nonetheless, play a significant role in the European Transient Cycles of heavy duty vehicles. The contribution and development of each process or device's exergy losses during transient operation will be studied and discussed.

2 Background on irreversibilities production during steady-state operation of internal combustion engines

Any system undergoing a chemical reaction experiences destruction of availability due to the (inherent) irreversibility of the reaction process. For an internal combustion engine cylinder, this irreversibility production arises due to combustion, viscous dissipation, turbulence, inlet-valve throttling and mixing of the incoming air (compression ignition engine) or air–fuel mixture (spark ignition engine) with the cylinder residuals. Typical values of in-cylinder irreversibilities are in the order of 20–25% for full load, four-stroke, turbocharged, diesel engine operation. Greater values are expected for compression ignition engine operating at low loads.

The contribution of combustion to the total in-cylinder irreversibilities is more than 95% as will be analysed in a later section. Some fundamental aspects of the mechanism of availability destruction due to combustion in an internal combustion engine can be summarised as follows:

1. About 80% of the combustion irreversibilities occur during the heat transfer process between the reacting gas and the yet unburned mixture (Dunbar and Lior, 1994).
2. An increasing combustion temperature – as, for example, is the case with increasing fuel–air equivalence ratio 'Φ' (in lean operation) of compression ignition engines – decreases the combustion irreversibilities when these are reduced to the fuel availability. This conclusion is interrelated to the previous one, since an increasing gas temperature decreases the relative amount of heat transfer from the reacting gas to the yet unburned mixture.
3. The effect of pressure changes during combustion on the availability is modest.

4 The amount of combustion irreversibilities can be correlated to the differential change in mixture composition, and notably nothing else. This was the result from the work conducted by Rakopoulos and Andritsakis (1993), who calculated the combustion irreversibility production rate as a function of fuel reaction rate only, revealing that, after all, both heat transfer and work production inside the cylinder only indirectly influence the irreversibilities accumulation.

5 Recent research in the field of alternative fuels has indicated some very promising results. For example, a decrease in combustion irreversibility can be achieved when using lighter (methane) or oxygenated (methanol) fuels (Kyritsis and Rakopoulos, 2001), whereas hydrogen enriched combustion can also help towards a decrease of combustion irreversibilities (Rakopoulos and Kyritsis, 2006) and thus the potential for increased piston work.

On the other hand, the irreversibilities in the turbocharger of an internal combustion engine are mainly fluid flow losses, owing to fluid shear and throttling assuming around a 10% of the total engine irreversibilities.

The irreversibilities in the aftercooler account for the loss of availability due to the heat transfer to a cooler medium; they depend largely on the temperature level of the medium to be cooled. The transfer of heat to a cooler medium is a procedure not desirable from the second-law of thermodynamics point of view. The particular one is responsible for the loss of around 0.5–1% of the fuel's chemical availability in internal combustion engine applications.

Inlet manifold irreversibilities account mainly for mixing of incoming air with the intake manifold contents, and are, usually, less than 1% of the fuel's chemical availability.

The irreversibility rate in the exhaust manifold arises from throttling across the exhaust valve, mixing of cylinder exhaust gases with manifold contents and gas friction along the manifold length. It assumes values of around 1.5–3% of the fuel's chemical availability (greater values correspond to turbocharged engines). Primus (1984) concluded that an optimal exhaust manifold diameter exists as regards frictional and throttling losses, while increasing the engine speed or engine load or turbine efficiency, or decreasing the compression ratio or turbine power, results in an increase of the exhaust manifold losses during steady-state operation.

3 First-law analysis

3.1 In-cylinder modelling

Since the present analysis does not, at the moment, include prediction of exhaust emissions and, on the other hand, deals with transient operation calculations on a Degree Crank Angle (°CA) basis, a single-zone model is used for the evaluation of thermodynamic processes. This approach combines satisfactory accuracy with limited PC program execution time. Polynomial expressions from Heywood (1988), with a 298 K reference datum, are used for the thermodynamic properties of each of the four species (O_2, N_2, CO_2 and H_2O) considered. They concern evaluation of internal energy and specific heat capacities for first-law application to the cylinder contents, using the filling and emptying modelling technique (Heywood, 1988; Rakopoulos and Giakoumis, 1998; Winterbone, 1986).

For heat release rate predictions, the fundamental model proposed by Whitehouse and Way (1969–1970) is used. In this model the combustion process consists of two parts, that is, a preparation limited and a reaction limited (Arrhenius type) combustion rate. It is vital for a proper simulation of transient response that combustion modelling takes into consideration the continuously changing nature of operating conditions. Thus, constant K in the (dominant) preparation rate equation of the Whitehouse–Way model is correlated with the Sauter Mean Diameter (SMD) of the fuel droplets through a formula of the type $K \propto (1/\mathrm{SMD})^c$ (Benson and Whitehouse, 1979).

The improved model of Annand (Annand and Ma, 1970/1971) is used to simulate heat loss Q_L to the cylinder walls,

$$\frac{dQ_L}{dt} = F\left\{\frac{k_g}{D}\mathrm{Re}^b\left[a\left(T_g - T_w\right) + \frac{a'}{\omega}\frac{dT_g}{dt}\right] + c\left(T_g^4 - T_w^4\right)\right\} \tag{1}$$

where a, a', b and c are constants evaluated after experimental matching at steady-state conditions, F is the piston area, k_g is the gas thermal conductivity (W/mK), and the Reynolds number Re is calculated with a characteristic speed derived from a k–ε turbulence model and a characteristic length equal to the piston diameter.

During transient operation, the thermal inertia of the cylinder wall is taken into account using a detailed heat transfer scheme, which models the temperature distribution from the gas to the cylinder wall up to the coolant (convection from gas to internal wall surface and from external wall surface to coolant, and conduction across the cylinder wall).

Various sophisticated submodels have been incorporated in the main code, which have been analysed in previous publications (Rakopoulos and Giakoumis, 1998, 2004a, 2006a). These deal with:

1 *Multicylinder engine modelling*: At steady-state operation the performance of each cylinder is essentially the same, due to the constant position of the governor clutch resulting in the same amount of fuel being injected per cycle.

   At transient operation, on the other hand, each cylinder experiences different fuellings during the same engine cycle due to the continuous movement of the fuel pump rack, initiated by a load or speed change. These differentiations in fuelling can result in significant differentiations in torque response and finally speed, so affecting significantly the whole engine operation. As regards speed changes only the first cycles are practically affected, but when load changes are investigated significant variations can be experienced throughout the whole transient cycle.

   Contrary to the usual approach, that is, solution of the governing equations for one cylinder and subsequent use of suitable phasing images of this cylinder's behaviour, a true multicylinder engine model is developed. Here, all the governing differential and algebraic equations are solved individually for every one cylinder of the six-cylinder engine under study. The dominant variable is here the fuel pump rack position, which moves continuously during the transient event due to the respective governor clutch movement, thus differentiating the amount of injected fuel per cylinder even in the same cycle.

   Apart from this differentiation in fuelling, each cylinder experiences also different air mass flow-rates during the same transient engine cycle, due to the transient operation of both turbocharger compressor and inlet manifold. These are taken into account, as both inlet and exhaust manifolds are modelled in the PC code to exchange mass only with that cylinder which experiences inlet or exhaust at the *particular* computational step. Consequently, no phasing image of one cylinder's inlet and exhaust process through a 'mean' inlet and exhaust manifold simulation is applied, while at the same time interactions between exhausting cylinders, which can under certain circumstances lead to backflow, are also taken into account. As a consequence, the use of the multicylinder engine model results in different individual cylinder air–fuel ratios during an engine transient cycle, due to the respective differentiation in both air mass flow and fuelling rates.

This 'multicylinder' approach has, of course, the drawback of increasing the computational time almost linearly to the number of cylinders involved, but it is capable of providing more accuracy with the transient operation phenomena. Moreover, it can offer better results as regards manifolds simulation even at steady-state conditions. The engine under study, being a six-cylinder one, has a twin-entry turbine. This means that its exhaust manifold consists of two parts, one of which communicates with cylinders 1, 2 and 3 and the other with cylinders 4, 5 and 6. This particular configuration has also been taken under consideration in the simulation results described below.

2 *Fuel pump operation*: a mathematical fuel injection model, experimentally validated at steady-state conditions, is applied to simulate the fuel pump-injector lift mechanism (Rakopoulos and Hountalas, 1996), taking into account the delivery valve and injector needle motion. The unsteady gas flow equations are solved using the method of characteristics, providing the dynamic injection timing as well as the duration and the rate of injection for each cylinder at each transient cycle. The obvious advantage, here, is that the transient operation of the fuel pump is also taken into account, mainly through the fuel pump residual pressure value, which is built up together with the other variables during the transient event. This individual fuel injection 'subroutine' is called upon once for every cylinder at each cycle with the values of angular velocity, fuel pump rack position and pump residual pressure existing at the point of the individual cylinder's static injection timing.

3 *Friction*: for the calculation of friction inside the cylinder, the method proposed by Taraza et al. (2000) is adopted. It describes the non-steady profile of friction torque during each cycle, based on fundamental friction analysis. In this method, the total amount of friction is divided into four parts, that is, piston rings assembly, loaded bearings, valve train and auxiliaries. Total friction torque at each °CA is the sum of the above terms and it varies continuously during the engine cycle, unlike the usually applied 'mean' friction mean effective pressure (fmep) equations where friction torque remains constant throughout each cycle.

The turbocharger compressor performance is analysed using its steady flow characteristics that are valid under steady-state conditions. Knowing the compressor mass flow rate and speed, the compressor pressure ratio and isentropic efficiency can be calculated using a two-dimensional data interpolation scheme. For this purpose, digital representations of the pressure ratio and isentropic efficiency curves are needed, and this is achieved by fitting second order polynomial expressions to the available compressor map curves. Likewise, turbine modelling is accomplished using the turbine map as provided by the manufacturer.

3.2 Engine dynamics

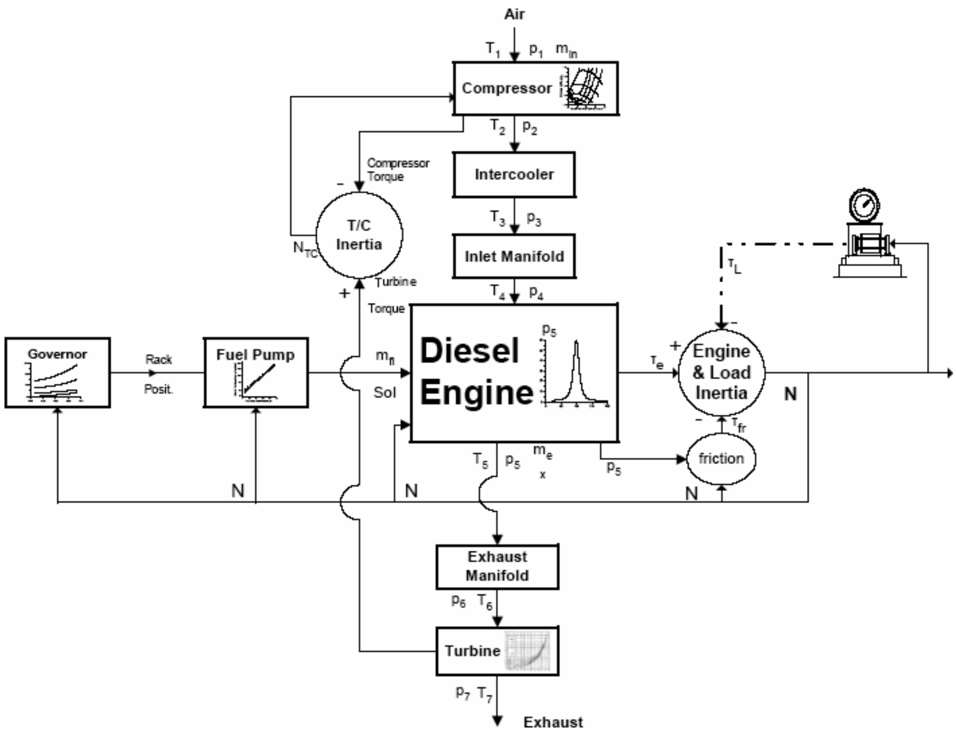
The conservation of angular momentum applied to the total system (engine plus load) yields:

$$\tau_e(\varphi,\omega) - \tau_{fr}(\varphi,\omega)_{trans} - \tau_{Load}(\omega) = G_{tot}\frac{d\omega}{dt} \tag{2}$$

where G_{tot} is the engine-flywheel-load mass moment of inertia and $\tau_e(\varphi, \omega)$ stands for the instantaneous value of the engine torque. The connecting rod is modelled as a rigid body experiencing reciprocating and rotating movement at the same time (Rakopoulos and Giakoumis, 1998). Also, $\tau_{Load}(\omega)$ is the load torque, which, for the hydraulic brake coupled to the engine examined, is $\propto\omega^2$. Lastly, $\tau_{fr}(\varphi, \omega)_{trans}$ stands for the friction torque derived using the Taraza et al. (2000) model described in the previous subsection.

The block diagram of the engine simulation code valid under transient conditions is depicted in Figure 1, that shows all the interconnections between engine subsystems and between diesel engine and load.

Figure 1 Block diagram of transient simulation code



4 Second-law analysis

In the following paragraphs the availability balance will be applied to all diesel engine subsystems, on a °CA basis (Moran, 1982; Rakopoulos and Giakoumis, 2004a,b, 2006b). Indices 1–7 refer to the strategic points locations indicated in Figure 1.

4.1 Availability balance of engine cylinder

For the jth engine cylinder, on a °CA basis, we have:

$$\frac{\mathrm{d}A_{\mathrm{j}}}{\mathrm{d}\varphi}=\frac{\dot{m}_{4\mathrm{j}}b_4-\dot{m}_5 b_{5\mathrm{j}}}{6N}-\frac{\mathrm{d}A_{\mathrm{w}}}{\mathrm{d}\varphi}-\frac{\mathrm{d}A_{\mathrm{L}}}{\mathrm{d}\varphi}+\frac{\mathrm{d}A_{\mathrm{f}}}{\mathrm{d}\varphi}-\frac{\mathrm{d}I_{\mathrm{cyl}}}{\mathrm{d}\varphi} \tag{3}$$

In the above equation, $\dot{m}_{4\mathrm{j}}$ is the incoming flow rate from the inlet manifold to the jth cylinder, whereas $\dot{m}_{5\mathrm{j}}$ is the outgoing one from the jth cylinder to the exhaust manifold. The terms b_4 and b_5 refer to the flow or stream availability (or exergy) of the incoming and the outgoing cylinder mass flow rates, respectively, given by (neglecting kinetic and potential energy contribution),

$$b=h-T_{\mathrm{o}}s-\sum_i x_i\mu_i^{\mathrm{o}} \tag{4}$$

with x_i the mole fraction of species i and μ_i its chemical potential. The term on the left-hand side of Equation (3) is expressed explicitly as:

$$\frac{\mathrm{d}A_j}{\mathrm{d}\varphi}=\frac{\mathrm{d}U}{\mathrm{d}\varphi}+p_{\mathrm{o}}\frac{\mathrm{d}V}{\mathrm{d}\varphi}-T_{\mathrm{o}}\frac{\mathrm{d}S}{\mathrm{d}\varphi}-\sum_i\frac{\mathrm{d}m_i}{\mathrm{d}\varphi}\mu_i^{\mathrm{o}} \tag{5}$$

providing the rate of change in the total availability of the cylinder contents.

Also,

$$\frac{\mathrm{d}A_{\mathrm{W}}}{\mathrm{d}\varphi}=\left(p_{5\mathrm{j}}-p_{\mathrm{o}}\right)\frac{\mathrm{d}V}{\mathrm{d}\varphi} \tag{6}$$

is the (indicated) work transfer, where $\mathrm{d}V/\mathrm{d}\varphi$ is the rate of change of cylinder volume with crank angle and $p_{5\mathrm{j}}$ the instantaneous cylinder pressure found from the first-law analysis of the engine processes.

$$\frac{\mathrm{d}A_{\mathrm{L}}}{\mathrm{d}\varphi}=\frac{\mathrm{d}Q_{\mathrm{L}}}{\mathrm{d}\varphi}\left(1-\frac{T_{\mathrm{o}}}{T_{5\mathrm{j}}}\right) \tag{7}$$

is the heat transfer availability to the cylinder walls, with $\mathrm{d}Q_{\mathrm{L}}/\mathrm{d}\varphi$ found from Equation (1) and $T_{5\mathrm{j}}$ is the instantaneous cylinder gas temperature. This availability loss is considered here as external to the cylinder control volume, but there are some researchers, as for example Alkidas (1989), who treat the heat losses as another source of irreversibility by defining an individual open thermodynamic system for the water-cooling circuit. By so doing, they usually sum up the combustion irreversibilities and the availability term for heat transfer (Equation (7)), and also calculate the availability increase in both the water and oil coolant circuits. In fact, the heat loss from the gas to cylinder walls contains a significant amount of availability, which is almost completely destroyed only after this has been transferred to the cooling medium.

$$\frac{\mathrm{d}A_{\mathrm{f}}}{\mathrm{d}\varphi}=\frac{\mathrm{d}m_{\mathrm{fb}}}{\mathrm{d}\varphi}a_{\mathrm{fch}} \tag{8}$$

is the burned fuel availability with a_{fch} being the fuel (chemical) availability. For the present study, the following approximation is applied for evaluating the fuel (C_zH_y) chemical availability (Moran, 1982)

$$a_{\mathrm{fch}} = \mathrm{LHV}\left(1.04224 + 0.011925\frac{y}{z} - \frac{0.042}{z}\right) \tag{9}$$

with LHV the fuel lower heating value. The fuel burning rate $\mathrm{d}m_{\mathrm{fb}}/d\varphi$ is calculated for each computational step using the Whitehouse–Way combustion model.

4.2 Availability balance of the engine subsystems

4.2.1 Turbocharger

For the compressor, steady-state operation is assumed so that there is no accumulation term; the availability balance is then,

$$\frac{\dot{m}_1 b_1 - \dot{m}_2 b_2}{6N} + \frac{\dot{W}_{\mathrm{C}}}{6N} = \frac{\mathrm{d}I_{\mathrm{C}}}{\mathrm{d}\varphi} \tag{10a}$$

With $\dot{m}_1 = \dot{m}_2$ the charge air flow rate (Figure 1).

For the turbine, accordingly,

$$\frac{\dot{m}_6 b_6 - \dot{m}_7 \mathrm{b}_7}{6N} - \dot{W}_{\mathrm{T}} = \frac{\mathrm{d}I_{\mathrm{T}}}{\mathrm{d}\varphi} \tag{10b}$$

The associated heat losses are considered negligible. In Equations (10a) and (10b), the terms $\dot{W}_{\mathrm{C}}$ and $\dot{W}_{\mathrm{T}}$ are evaluated from the thermodynamic analysis of the turbocharger at each °CA step, via instantaneous values 'picked up' from the turbo-machinery steady-state maps.

4.2.2 Aftercooler or intercooler

For the aftercooler, the availability balance is

$$\frac{\dot{m}_2 b_2 - \dot{m}_3 b_3}{6N} - \Delta A_{\mathrm{cw}} = \frac{\mathrm{d}I_{\mathrm{AC}}}{\mathrm{d}\varphi} \tag{11}$$

where b_2 is the flow availability at the compressor outlet – aftercooler inlet, b_3 the flow availability at the aftercooler outlet – inlet manifold inlet, and

$$\Delta A_{\mathrm{cw}} = \frac{1}{6N}\dot{m}_{cw}\left(b_{\mathrm{out}}^{\mathrm{cw}} - b_{\mathrm{in}}^{\mathrm{cw}}\right) \tag{12}$$

where

$$b_{\mathrm{out}}^{\mathrm{cw}} - b_{\mathrm{in}}^{\mathrm{cw}} = c_{\mathrm{pcw}}\left[T_{\mathrm{cw-out}} - T_{\mathrm{cw-in}} - T_{\mathrm{o}}\ln\left(\frac{T_{\mathrm{cw-out}}}{T_{\mathrm{cw-in}}}\right)\right] \tag{13}$$

is the increase in the availability of the cooling medium having a mass flow rate $\dot{m}_{\mathrm{cw}}$, specific (mass) heat c_{pcw}, initial temperature entering the aftercooler $T_{\mathrm{cw-in}}$ and final temperature leaving the aftercooler $T_{\mathrm{cw-out}}$.

4.2.3 Inlet manifold

For the inlet manifold, the availability balance reads:

$$\frac{dA_{im}}{d\varphi} = \frac{\dot{m}_3 b_3 - \sum_{j=1}^{z} \dot{m}_{4j} b_4}{6N} - \frac{dI_{im}}{d\varphi} \tag{14}$$

where b_4 is the flow availability at the intake manifold and $j = 1, \ldots, z$ is the cylinder exchanging mass with the inlet manifold found from the energy analysis at each °CA step. No heat losses are taken into account.

4.2.4 Exhaust manifold

For the exhaust manifold, the availability balance is:

$$\frac{dA_{em}}{d\varphi} = \frac{\sum_{j=1}^{z} \dot{m}_{5j} b_{5j} - \dot{m}_6 b_6}{6N} - \frac{dI_{em}}{d\varphi} + \frac{dA_{Lem}}{d\varphi} \tag{15}$$

where index 6 identifies the exhaust manifold state. The term

$$\frac{dA_{Lem}}{d\varphi} = \frac{dQ_{Lem}}{d\varphi}\left(1 - \frac{T_o}{T_6}\right) \tag{16}$$

accounts for the heat loss across the exhaust manifold (considered as external to the manifold control volume and, thus, not included in the respective irreversibilities), with T_6 the instantaneous temperature of the manifold contents.

5 Experimental study

The objective of the experimental test bed developed was to validate the performance of the engine simulation by setting up a comprehensive instrumentation. The experimental investigation was conducted on an MWM TbRHS 518S, six-cylinder, turbocharged and aftercooled, Indirect Injection (IDI), medium-high speed diesel engine. The engine is fitted with a 'KKK' turbocharger, a water aftercooler after the turbocharger compressor and is permanently coupled to a 'Schenck' hydraulic dynamometer. Details about the experimental set-up can be found by Rakopoulos et al. (1998). The basic data for the engine, turbocharger, brake and data processing system are given in Table 1.

Since the particular engine is one with a relatively small speed range, mainly load changes (increases) with constant governor setting were examined. For the transient tests conducted, the initial speed was 1180 or 1380 rpm and the initial load 10% of the engine's full load. The final conditions for the transient events varied from 47% to 95% of the engine's full load.

Table 1 Basic data for the engine and turbocharger

	MWM TbRHS 518S
Engine model and type	In-line, six-cylinder, four-stroke, compression ignition, indirect injection, turbocharged, aftercooled, heavy-duty
Speed range	1000 ÷ 1500 rpm
Bore/stroke/swept volume	140 mm/180 mm/2.77 lt per cylinder
Compression ratio	17.7:1
Maximum power	320 HP (236 kW) @ 1500 rpm
Maximum torque	1520 Nm @ 1250 rpm
Fuel pump	Bosch PE-P series, in-line, six-cylinder with mechanical governor Bosch RSUV 300/900
	KKK M4B 754/345
Turbocharger	Single-stage, centrifugal compressor,
	Single-stage, twin entry, axial turbine
Moment of inertia	Engine and brake: 15.60 kg m^2
	Turbocharger: 7.5×10^{-4} kg m^2

A typical example of a conducted transient experiment is given in Figure 2, showing the response of some important first-law values. Here, the initial load was 10% of the full engine load at 1180 rpm. The final load applied was approximately 50% of the full engine load (400% relative load-change). The overall matching between experimental and predicted transient responses seems satisfactory for both engine and turbocharger variables as regards trends and final operating conditions of the engine. The boost pressure is notably delayed compared to the speed profile owing to the well-known turbocharger lag effect.

6 Irreversibilities production during transient diesel engine operation

Figure 3 shows the response of the in-cylinder availability terms, that is, work, heat loss to the walls, exhaust gas and irreversibilities as a function of the engine cycles for a 10–75% load-change of the engine commencing from 1180 rpm. All of these terms are cumulative values (in Joules) over each cycle (for all six cylinders of the engine). The availability term for work and heat loss to the walls increase with fuelling as a function of the engine cycles, because of increases in the charge temperature resulting from increases of the injected fuel quantity and accompanying fuel–air equivalence ratios. Similar results hold for the exhaust gas from the cylinder term and for the irreversibilities term. In-cylinder irreversibilities consist of combustion, inlet and exhaust ones. All availability values are characterised by a time delay compared to the engine speed (cf. Figure 2), while the in-cylinder irreversibilities follow closely the profile of the injected fuel availability.

Figure 2 Experimental and predicted engine properties response to an increase in load

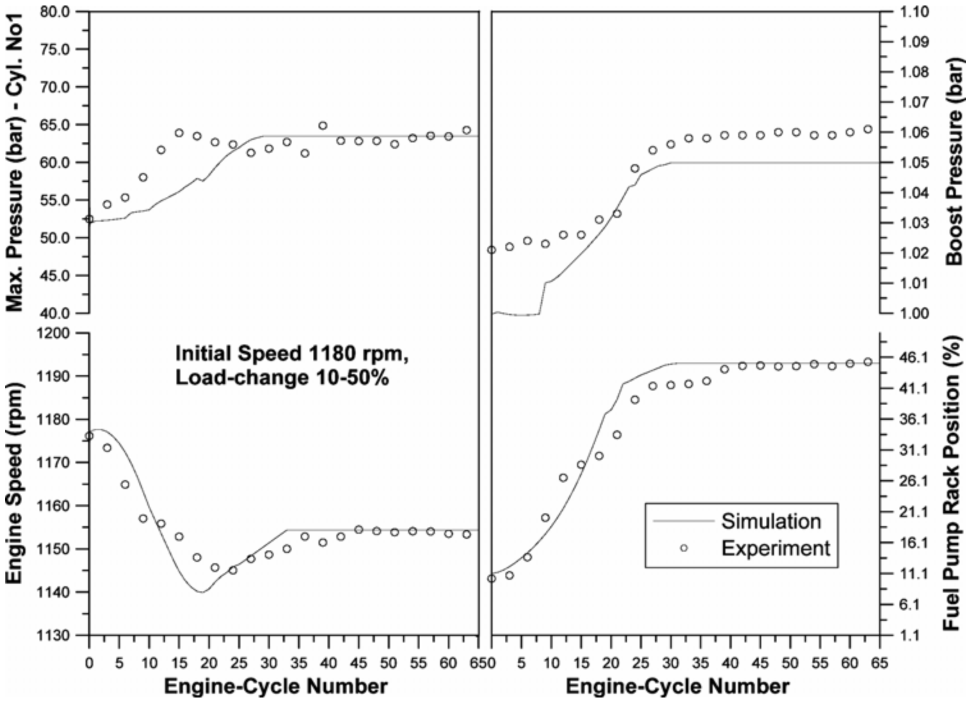
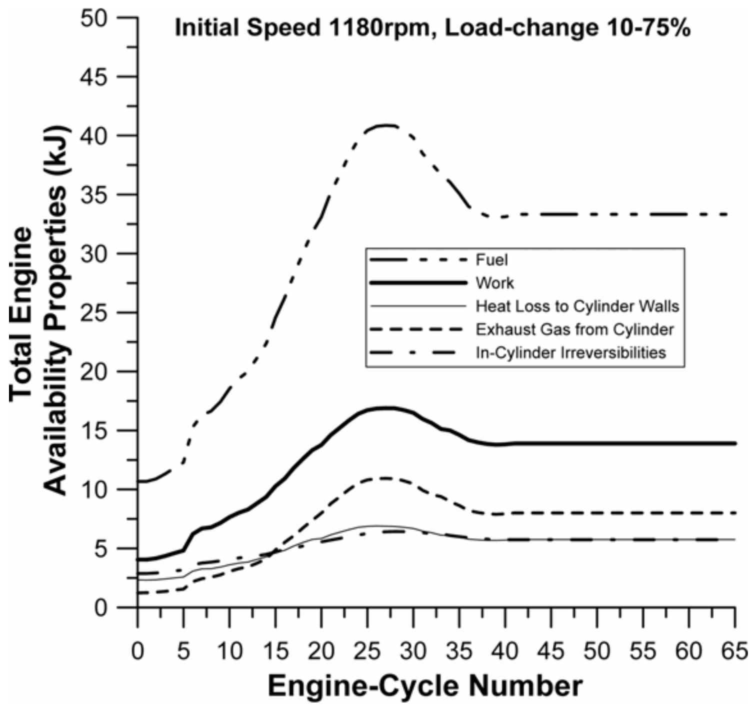


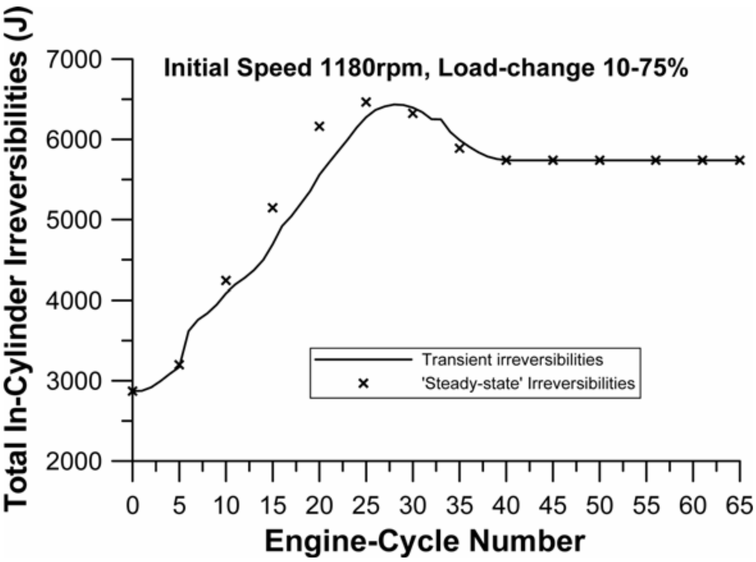
Figure 3 Response of total in-cylinder availability terms (from all six cylinders) to an increase in load



In Figure 4 a comparison is illustrated between transient and some intermediate 'steady-state' in-cylinder irreversibilities values (i.e. steady-state irreversibilities for all six cylinders of the engine studied, at the engine speed and fuelling of the corresponding transient cycles). The work by Van Gerpen and Shapiro (1990) revealed that during steady-state operation the effect of both combustion duration and shape of heat release curve is modest as regards the total amount of irreversibilities, although the rate of irreversibility production is greatly affected. An attempt to expand this finding into the transient operation too, would result in the misleading assumption that transient cumulative combustion irreversibilities per cycle should practically be affected *only* by the amount of injected fuel and the respective equivalence ratio. However, during transients various off-design phenomena occur mainly during the early cycles, for example:

1. an influence in mixture formation and ignition delay due to air-deficiency caused by turbocharger lag
2. a change in the average fuel droplet diameter caused by lower density and swirl, which leads to increased jet penetration in comparison to the respective steady-state operating conditions
3. the lower end gas and wall temperatures in combination with a higher amount of end gas result in an increased ignition delay and hard combustion course, during the early cycles where the turbocharger lag is more prominent (Harndorf and Kuhnt, 1995).

Figure 4 Comparison between total in-cylinder irreversibilities, during a load increase, with intermediate 'steady-state' irreversibilities



In Figure 4, a difference up to 11% at the 20th cycle – this is indeed an 'early' cycle of the transient event, as the particular engine-brake configuration has a very high mass moment of inertia – is observed when comparing the transient with the respective 'steady-state' irreversibilities. This best highlights the different evolution profile of the transient combustion irreversibilities compared to steady-state operation. The difference is mainly attributed to the differentiated fuel–air equivalence ratio experienced during transients owing to the turbocharger lag, which significantly affects the air–mass flow rate. It is also enhanced by the fact that, during transients, integration of Equation (5) over an engine cycle does not sum up to zero, as the initial conditions of the new cycle differ from the initial conditions of the previous one.

Figure 5 examines the same in-cylinder properties studied in Figure 3, but now their values are reduced to the fuel availability over each cycle. Two points are illustrated here:

1 The reduced in-cylinder irreversibilities decrease as the transient event develops. This happens due to the fact that combustion irreversibilities fall with increasing load. Greater loads, that is, fuelling, result in less degradation of fuel chemical availability when transferred to the ('hotter' and hence possessing greater work potential) exhaust gases.

2 The amount of availability included in the exhaust gas to ambient is not small (almost 18% of the fuel's chemical availability at the end of the transient event), although a part of the work potential of the exhaust gases leaving the cylinders has already been exploited by driving the turbine.

Figure 5 Response of in-cylinder availability terms, reduced to the fuel availability, to an increase in load

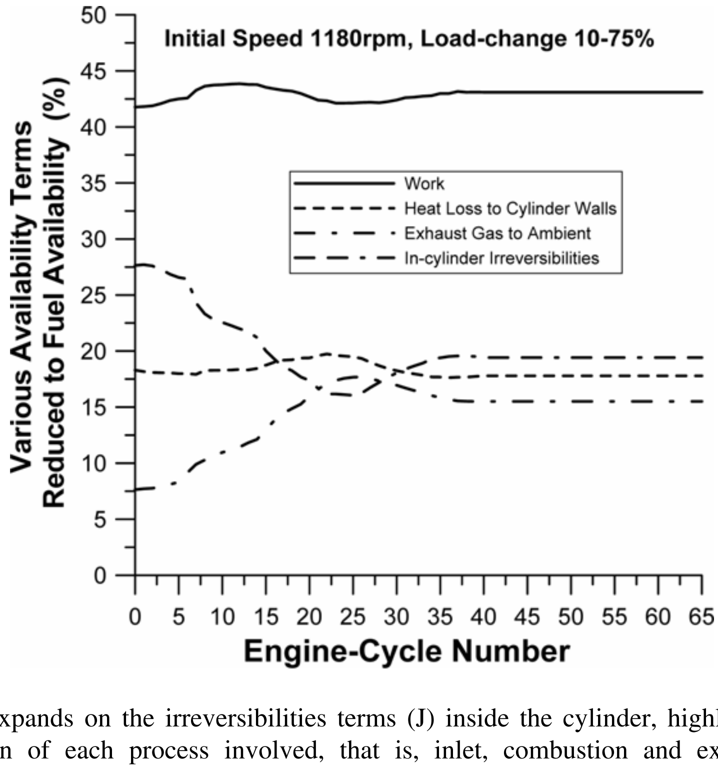
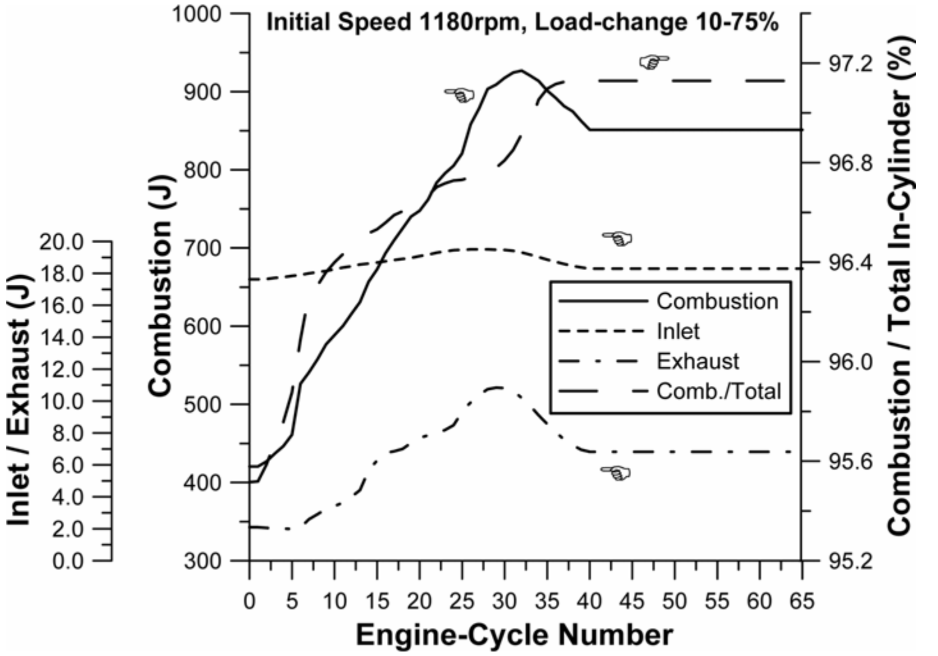


Figure 6 expands on the irreversibilities terms (J) inside the cylinder, highlighting the contribution of each process involved, that is, inlet, combustion and exhaust. The dominant role of the combustion irreversibilities is obvious here (at least 95% of the total cylinder ones), and with increasing importance as the transient event develops. Their greatest part is produced at the early stage of combustion where the gas temperature is still low, as it will be shown later in this section. The percentage of combustion irreversibilities is in agreement with the results reached by Primus and Flynn (1986) and Alkidas (1988) for steady-state operation. The latter researcher working on a single-cylinder, naturally aspirated, DI diesel engine and using a simplified mixing model, estimated the air–fuel mixing irreversibilities to be 3% of the total ones. Primus and Flynn (1986) calculated in-cylinder, non-combustion irreversibilities as 4.96% of the total ones, for an engine configuration similar to the one under study here.

Figure 6 Response of various in-cylinder irreversibilities to an increase in load



Flynn et al. (1984) argued that only that part of the combustion irreversibilities, which is associated with the heat release placement and shape, can be affected by engine development and thus can be improved (highlighting the 'inevitable' of the combustion irreversibility). A key remark here is that increasing the level of combustion temperatures, as for example when increasing the equivalence ratio or compression ratio or insulating the cylinder walls, results in a relative decrease in the combustion irreversibilities, since combustion becomes less irreversible as the fuel chemical availability is transferred to 'hotter' exhaust gases. This fact denotes that such a process is, in principle, a favourable one from the second-law perspective, highlighting a part of the path that has to be followed for improving engine performance.

Figure 7 focuses on the main chamber and prechamber fuel and irreversibilities response during the transient event. The slope of the irreversibilities profile for the main chamber is considerably lower than the corresponding fuel one, owing to the subsequent increase in the pressures and temperatures associated with the increase in fuelling. The relative prechamber contribution is higher at low loading and decreases as long as the fuelling increases, reaching a minimum at cycle 24. This happens due to the fact that the lower the engine loading the greater the fraction of the fuel burnt in the prechamber as is also depicted in this figure (bold line). This leads to a considerable increase in the prechamber irreversibilities, which for the 10% load case are as high as 25% of the total ones, while for the 100% load case they drop to 4%. The prechamber irreversibilities include also the availability loss at the two narrow throats connecting the main chamber to the prechamber of the engine. In a previous paper by Rakopoulos and Andritsakis (1993), this amount of throttling losses was calculated at, maximum, 0.23% of the fuel's chemical availability during steady-state operation of the engine in hand.

Figure 7 Development of main chamber and prechamber cumulative burned fuel availability and irreversibilities terms after an increase in load

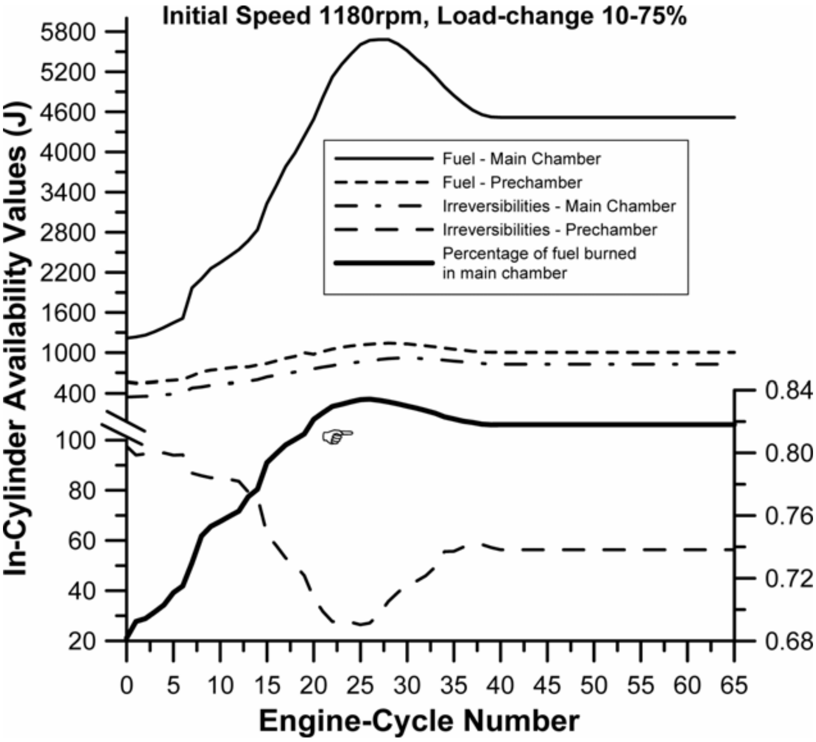


Figure 8 concentrates on the first and the last cycle of the transient event, showing the development of the rate of the various irreversibilities (J/°CA), that is, in-cylinder (combustion plus inlet and exhaust), compressor, turbine, inlet manifold and exhaust manifold. Both the fuel and the combustion irreversibility profiles develop in a similar way, with the combustion irreversibility term reaching a maximum during the early stages of the (still premixed) combustion phase. On the other hand, the manifold's and turbocharger's irreversibilities (as well as their other second-law values) incorporate the pulsating nature of the six cylinders operation. As expected, this is more pronounced for the exhaust manifold and the turbine operation, owing to the pulse turbocharging scheme involved. For the turbine and for the final cycle of the transient event, the ratio of the maximum to minimum irreversibility rate is almost 20.

Figure 9 illustrates the irreversibilities terms of all the subsystems, reduced now to the total irreversibilities. Here, we show how cylinder as well as inlet manifold, exhaust manifold, compressor, turbine and aftercooler irreversibilities develop during the transient event. The relative importance of the in-cylinder irreversibilities decrease as the transient event develops due to the increase in loading and thus fuelling (cf. Figure 5). The inlet manifold irreversibilities constitute a very small percentage of the total ones (not greater than 3% and with decreasing importance during the transient event), whereas those of the exhaust manifold increase substantially from cycle 10 where the main increase in the injected fuel quantity (and thus gas pressures and temperatures) occurs, reaching as much as 15% of the total irreversibilities at cycle 31. Turbocharger irreversibilities increase during the transient event due to the increase in both compressor and turbine pressures and temperatures, accounting for 4.7% of the total irreversibilities at cycle 28. The compressor irreversibilities outweigh the turbine ones, except for the first cycles where the turbine isentropic efficiency is lower. Aftercooler irreversibilities never exceed 0.47% of the total ones (cycle 30), revealing the low importance of this process as well as the very low potential for work recovery. It is also interesting to note, that the cycle where the maximum (or minimum) percentage occurs differs for each subsystem. This is attributed to the different 'inertia' of each subsystem, which affects its transient response.

Figure 8 Development in the rate (J/°CA) of irreversibility terms of all diesel engine subsystems, at the initial and final steady-state conditions of a transient event

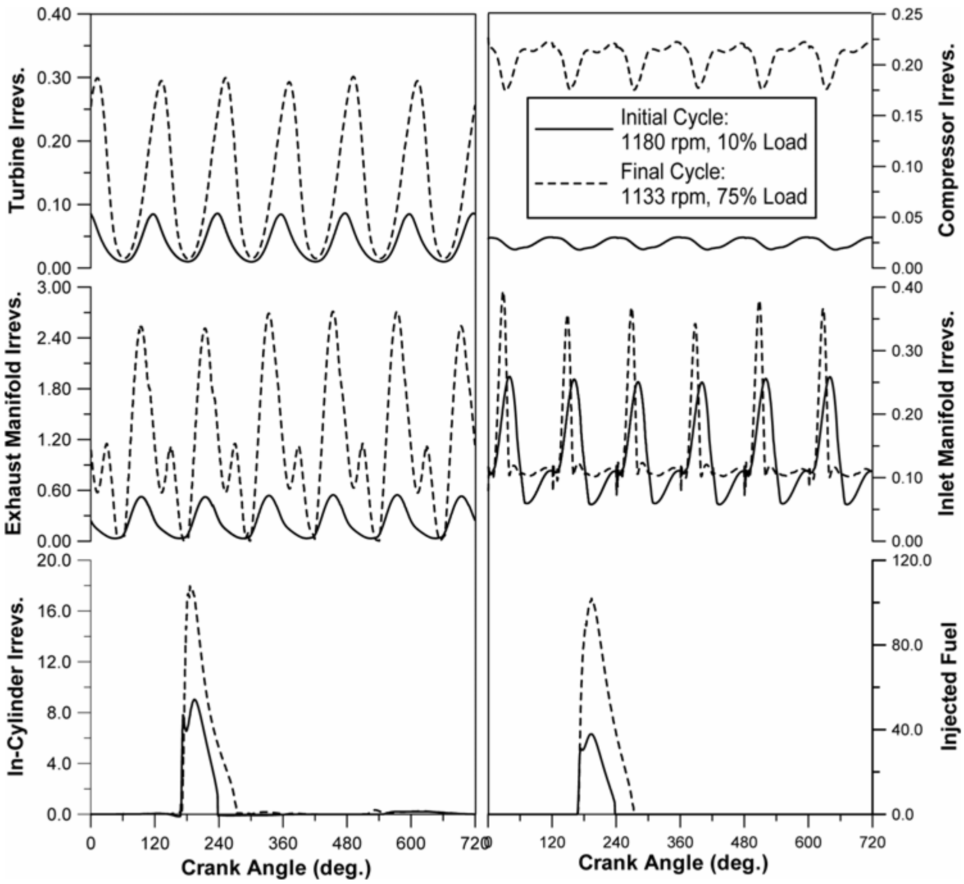
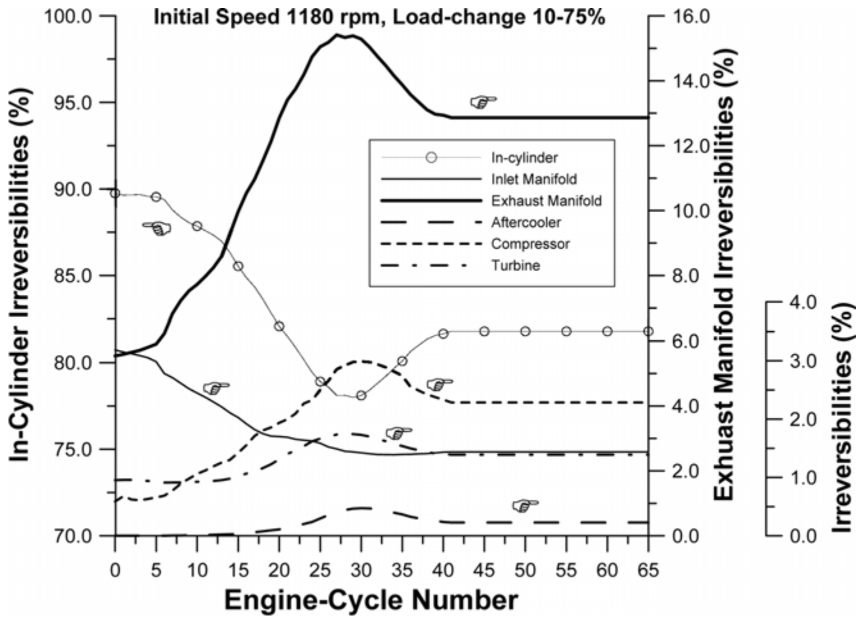


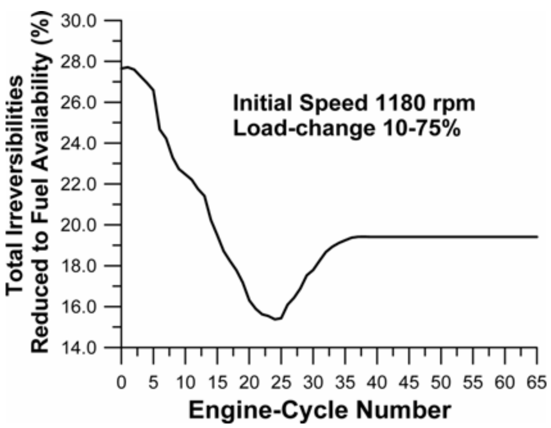
Figure 9 Response of various diesel engine and its sub-systems irreversibilities terms, reduced to the total irreversibilities, to an increase in load



All these results go along with the findings reached by Primus and Flynn (1986), who calculated exhaust manifold losses at 2.5%, and turbine irreversibilities at 1.69% of the fuel availability when operating at steady-state conditions and for a similar engine configuration.

Finally, Figure 10 illustrates the response of the total diesel engine plant irreversibilities, that is, in-cylinder plus manifolds plus aftercooler plus turbocharger, reduced to the total injected fuel availability. Owing to the great contribution of the combustion irreversibilities, the total ones show a similar transient profile with those of the in-cylinder ones depicted in Figure 4, that is, they decrease as the loading increases with the minimum value observed at cycle 23.

Figure 10 Response of total irreversibilities, reduced to the fuel availability, to an increase in load



7 Conclusions

A detailed first- and second-law analysis has been carried out on a six-cylinder, turbocharged and aftercooled diesel engine in order to study the first- and second-law balances of the engine during transient operation after a load increase. The model's first-law results were confirmed with experimental tests. The following results were reached from the analysis:

- In-cylinder irreversibilities account for a significant percentage of the injected fuel availability during a transient event, as is also the case at steady-state conditions. Their relative importance compared to other irreversibilities is always dominant during each transient. Their evolution profile and absolute value, however, differs from the respective 'steady-state' irreversibilities, mainly during the early cycles of the transient event.
- Other second-law values, such as exhaust gas from the cylinder as well as heat loss to the cylinder walls are of important magnitude, the recovery of which can notably improve engine performance.
- The main chamber (of an indirect injection diesel engine fitted with a prechamber) contributes mostly to the total combustion irreversibilities, ranging from 75% at low loading to almost 96% at full loading, aided by the higher level of pressures and temperatures in the prechamber during combustion.
- Cylinder irreversibilities decrease, proportionally, after a ramp increase in load due to the subsequent increase in fuelling, with the combustion irreversibilities accounting for at least 95% of the total cylinder ones. As a general rule, every operating parameter that can decrease the amount of combustion irreversibilities (e.g. greater cylinder wall temperature) is favourable according to the second-law and is capable, theoretically, to lead to increased piston work.
- Exhaust manifold irreversibilities increase significantly during a load increase, reaching as high as 15% of the total ones, highlighting another process which needs to be studied for possible efficiency improvement. This increased amount of irreversibilities arises mainly from the greater pressures and temperatures due to turbocharging, which has already lowered the magnitude of combustion irreversibilities. The inlet manifold irreversibilities, on the other hand, are of lesser and decreasing importance during a transient event.
- Turbocharger irreversibilities, although only a fraction of the (dominant) combustion ones, are not negligible, whereas the intercooler irreversibilities steadily remain of lesser importance (less than 0.5% of the total ones) during a load change.
- The recovery period and the general profile of the various irreversibilities terms depend on the respective first-law values, since the second-law balance is based on first-law data. Moreover, the minima or maxima observed vary as they depend on each term's inertia.

References

Alkidas, A.C. (1988) 'The application of availability and energy balances to a diesel engine', *ASME Transactions, Journal of Engineering for Gas Turbines and Power*, Vol. 110, pp.462–469.

Alkidas, A.C. (1989) *The Use of Availability and Energy Balances in Diesel Engines*, SAE Paper No. 890822.

Annand, W.J.D. and Ma, T.H. (1970/1971) 'Instantaneous heat transfer rates to the cylinder head surface of a small compression-ignition engine', *Proceedings of the Institution of Mechanical Engineers*, Vol. 185, pp.976–987.

Arcoumanis, C., Megaritis, A. and Bazari, Z. (1994) 'Analysis of transient exhaust emissions in a turbocharged vehicle diesel engine', *Institution of Mechanical Engineers, Conference on "Turbocharging and Turbochargers",* Paper C484/038, London, pp.71–81.

Bartsch, P., Prenninger, P. and Allmer, I. (1998) 'Transient performance optimization of turbocharged engines by means of gas exchange simulations' *Institution of Mechanical Engineers, International Conference on "Turbocharging and Air Management Systems"*, Paper C554/016, London, pp.237–251.

Benson, R.S. and Whitehouse, N.D. (1979) *Internal Combustion Engines*, Oxford: Pergamon Press.

Dunbar, W.R. and Lior, N. (1994) 'Sources of combustion irreversibility', *Combustion Science and Technology*, Vol. 103, pp.41–61.

Flynn, P.F., Hoag, K.L., Kamel, M.M. and Primus, R.J. (1984) *A New Perspective on Diesel Engine Evaluation Based on Second Law Analysis*, SAE Paper No. 840032.

Harndorf, H. and Kuhnt, H-W. (1995) 'Improvement of transient behaviour of turbocharged diesel engines through additional air injection in the turbocharger', *Motortechnische Zeitschrift*, Vol. 56, No. 1, pp.20–28 (in German).

Heywood, J.B. (1988) *Internal Combustion Engine Fundamentals*, New York: McGraw Hill.

Kyritsis, D.C. and Rakopoulos, C.D. (2001) *Parametric Study of the Availability Balance in An Internal Combustion Engine Cylinder*, SAE Paper No. 2001-01-1263.

Moran, M.J. (1982) *Availability Analysis: A Guide to Efficient Energy Use*, New Jersey: Prentice Hall.

Obert, E.F. and Gaggioli, R.A. (1963) *Thermodynamics*, New York: McGraw-Hill.

Primus, R.J. (1984) *A Second Law Approach to Exhaust System Optimization*, SAE Paper No. 840033.

Primus, R.J. and Flynn, P.F. (1986) 'The assessment of losses in diesel engines using second law analysis', *ASME WA-Meeting, Anaheim CA, Proceedings of AES*, pp. 61–68.

Rakopoulos, C.D. and Andritsakis, E.C. (1993) 'DI and IDI diesel engines combustion irreversibility analysis', *ASME-WA Meeting, New Orleans LA, Proceedings of AES*, Vol. 30, pp.17–32.

Rakopoulos, C.D., Andritsakis, E.C. and Kyritsis, D.C. (1993) 'Availability accumulation and destruction in a DI diesel engine with special reference to the limited cooled case', *Heat Recovery Systems and CHP*, Vol. 13, pp.261–276.

Rakopoulos, C.D. and Giakoumis, E.G. (1998) 'Simulation and analysis of a naturally aspirated IDI diesel engine under transient conditions comprising the effect of various dynamic and thermodynamic parameters', *Energy Conversion and Management,* Vol. 39, pp.465–484.

Rakopoulos, C.D. and Giakoumis, E.G. (2004a) 'Availability analysis of a turbocharged diesel engine operating under transient load conditions', *Energy-The International Journal*, Vol. 29, pp.1085–1104.

Rakopoulos, C.D. and Giakoumis, E.G. (2004b) *Parametric Study of Transient Turbocharged Diesel Engine Operation from the Second-Law Perspective*, SAE Paper No. 2004-01-1679.

Rakopoulos, C.D. and Giakoumis, E.G. (2005) *Second-Law Analysis of Indirect Injection Turbocharged Diesel Engine Operation Under Steady-State and Transient Conditions*, SAE Paper No. 2005-01-1131.

Rakopoulos, C.D. and Giakoumis, E.G. (2006a) 'Sensitivity analysis of transient diesel engine simulation', *Proceedings of the Institution of Mechanical Engineers – Part D, Journal of Automobile Engineering*, Vol. 220, pp.89–101.

Rakopoulos, C.D. and Giakoumis, E.G. (2006b) 'Second-law analyses applied to internal combustion engines operation', *Progress in Energy and Combustion Science*, Vol. 32, pp.2–47.

Rakopoulos, C.D., Giakoumis, E.G. and Hountalas, D.T. (1998) 'Experimental and simulation analysis of the transient operation of a turbocharged, multi-cylinder, IDI diesel engine', *Energy Research*, Vol. 22, pp.317–331.

Rakopoulos, C.D., Giakoumis, E.G., Hountalas, D.T. and Rakopoulos, D.C. (2004) *The Effect of Various Dynamic, Thermodynamic and Design Parameters on the Performance of a Turbocharged Diesel Engine Operating Under Transient Load Conditions*, SAE Paper No. 2004-01-0926.

Rakopoulos, C.D. and Hountalas, D.T. (1996) 'A simulation analysis of a DI diesel engine fuel injection system fitted with a constant pressure valve', *Energy Conversion and Management*, Vol. 37, pp.135–150.

Rakopoulos, C.D. and Kyritsis, D.C. (2006) 'Hydrogen enrichment effects on the second-law analysis of natural and landfill gas in engine cylinders', *Hydrogen Energy*, Vol. 31, No. 10, pp.1384–1393.

Taraza, D., Henein, N. and Bryzik, W. (2000) *Friction Losses in Multi-Cylinder Diesel Engines*, SAE Paper No. 2000-01-0921.

Van Gerpen, J.H. and Shapiro, H.N. (1990) 'Second-law analysis of diesel engine combustion', *ASME Transactions, Journal of Engineering for Gas Turbines and Power*, Vol. 112, pp.129–137.

Velasquez, J.A. and Milanez, L.F. (1994) *Analysis of the Irreversibilities in Diesel Engines*, SAE Paper No. 940673.

Watson, N. (1981) *Transient Performance Simulation and Analysis of Turbocharged Diesel Engines*, SAE Paper No. 810338.

Whitehouse, N.D. and Way, R.G.B. (1969–1970) 'Rate of heat release in diesel engines and its correlation with fuel injection data', *Proceedings of the Institution of Mechanical Engineers – Part 3J,* Vol. 184, pp.17–27.

Winterbone, D.E. (1986) 'Transient performance', in J.H. Horlock and D.E. Winterbone (Eds). *The Thermodynamics and Gas Dynamics of Internal Combustion Engines*, Vol. II, Oxford: Clarendon Press.

Nomenclature

A	availability (exergy), J
b	flow availability (exergy), J/kg
D	cylinder bore, m
G	mass moment of inertia, kg m^2
h	specific enthalpy, J/kg
k	thermal conductivity, W/(m K)
I	irreversibilities, J
m	mass, kg
N	engine speed, rpm

p	pressure, bar
Q	heat loss, J
s	specific entropy, J/kg K
T	temperature, K
t	time, s
V	volume, m^3

Greek symbols

μ	chemical potential, J/kg
τ	torque, Nm
φ	crank angle measured from the BDC position, deg
ω	angular velocity, rad/s

Subscripts

o	initial/atmospheric conditions
AC	aftercooler
C	compressor
ch	chemical
cw	cooling water
em	exhaust manifold
f	fuel
fr	friction
g	gas
im	inlet manifold
L	load or loss
T	turbine
TC	turbocharger
w	wall or work

Abbreviations

°CA	degrees crank angle
BDC	bottom dead centre
fmep	friction mean effective pressure, bar
LHV	lower heating value, J/kg
rpm	revolutions per minute
SMD	Sauter mean diameter, μm
TDC	top dead centre