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SPEED AND LOAD EFFECTS ON THE AVAILABILITY BALANCES AND IRREVERSIBILITIES PRODUCTION IN A MULTI-CYLINDER TURBOCHARGED DIESEL ENGINE

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Abstract—This work presents an analysis of the operation of a multi-cylinder, turbocharged, indirect injection diesel engine from a second-law analysis point of view. For this purpose, a single-zone thermodynamic model following the filling and emptying modelling technique is developed. A second-law analysis is performed in all parts of the diesel engine plant, which provides all the existing availability terms and accounts for the evaluation of every component's irreversibilities. A complete second-law terms tabulation is given for the maximum speed–full load operation case, which is compared with its first-law counterpart. A detailed parametric study is performed in all parts of the diesel engine plant comprising the effects of speed and load on the availability terms in a range that covers the whole operation of the engine under turbocharged action. Various second-law (availability) terms, such as indicated and brake work, heat transfer, inlet air, exhaust gas and friction, are presented, together with all the processes' irreversibilities, as a percentage of the fuel's chemical availability, for each engine operating condition. It is thus highlighted how the two basic engine operating parameters, i.e. speed and load, affect the operation of this engine under a second-law perspective. A compression ratio investigation on the availability terms is also performed. Copyright © 1996 Elsevier Science Ltd.

Keywords—Turbocharged diesel engine, second-law analysis, speed, load, irreversibilities.

NOMENCLATURE

A	availability (J)
a	specific availability (J/kg)
b	flow availability (J/kg)
CA	crank angle
$c_{p(v)}$	specific heat capacity under constant pressure (volume) (J/kg K)
F	surface (m^2)
f	fuel–air equivalence ratio
G	Gibbs free enthalpy (J)
g	specific Gibbs free enthalpy (J/kg)
h	specific enthalpy (J/kg)
I	irreversibility (J)
M, m	mass (kg)
N	engine speed (rpm)
p	pressure (N/m^2)
Q	heat (J)
r	crank radius (m) or compressor pressure ratio
rpm	revolutions per minute
S	entropy (J/K)
s	specific entropy (J/kg K)
T	absolute temperature (K)
t	time (s)
U	internal energy (J)
u	specific internal energy (J/kg)
V	volume (m^3)
W	work (J)
Greek letters	
ε	aftercooler effectiveness or engine compression ratio
λ	gas thermal conductivity (W/m K)
μ	dynamic viscocity (kg/m s) or chemical potential (J/kg)
ρ	density (kg/m^3)
φ	crank angle (degrees)

ω	angular velocity (per s)
Subscripts	
b	burning
C	compressor
c	charge air
ch	chemical
e	exhaust, exit
f	fuel
g	gas
i	species, inlet
l	heat loss
m	manifold
o	ambient (environment, atmosphere) conditions
T	turbine
w	wall, work
Superscripts	
o	dead state
.	time derivative (rate)

INTRODUCTION

During the last 20 years it has been understood that traditional first-law theory, although properly expressed, often fails to give the best insight into the engine's operation, whether this is of the internal combustion type or not [1–4]. In contrast, second-law analysis with its more 'interior' study of what is happening during a process contributes a new way of thinking and studying thermodynamic processes, a fact providing more flexibility and field for improvement to the engineer. Diesel engine simulation modelling, in comparison, has contributed enormously towards the aim of new evaluation and development. Thermodynamic models of the real diesel engine operation, whether of the single-zone type [5–9] or the multi-zone one [10–14], have always been the sound basis for every complete and serious study of the engine's performance and its sensitivity to various operating parameters.

Availability balance equations were applied to a diesel engine on an overall basis in ref. [15], but mainly extensive and detailed works where the second-law equations were coupled explicitly with theoretical simulation models of the engine cycle have been reported in refs [2–4, 16–22]. Works reported by various Cummins Engine Company researchers [2, 3, 16, 17] have presented a quantification of engine irreversibilities, while throttling and thermal mixing losses were brought up along with the well established combustion irreversibilities. Van Gerpen and Shapiro [4], in contrast, performed a detailed theoretical analysis for the closed part of the cycle bringing in focus the controversial term of chemical availability.

More recently, the effect of limited cooled engines on the availability balances was studied [21]; the interest for such engines emanates from their potential to do more work by utilizing the exhaust gases in a Rankine bottoming cycle or a power turbine. Also DI and IDI diesel engines' combustion irreversibilities were brought into focus through an in-depth analysis by Rakopoulos and Andritsakis [20]. A detailed analysis of all availability terms, devices' irreversibilities and development of rate and cumulative availability terms for every engine component, for a multi-cylinder, turbocharged diesel engine, have been reported by Rakopoulos and Giakoumis [22]. Finally, some important and detailed aspects of the second-law analysis techniques have been reported for the closed part of the cycle of a spark ignition engine by Rakopoulos [19].

The present paper applies the second-law analysis in all the processes involved in a turbocharged diesel engine. For this purpose, a single-zone thermodynamic model, following the filling and emptying modelling technique, is developed. It is described how availability analysis is applied and how the various kinds of availability degradations are evaluated.

The present work provides histograms of tabulated availability and irreversibility terms for the whole diesel engine plant and for the maximum speed–full load operation condition, with first- and second-law terms presented together so that the differences involved are highlighted and discussed. Its main scope is to study the effect of speed and load on all existing availability terms and especially on the various devices' irreversibilities. This is an original feature since explicit diagrams for the effect of load and speed on the availability terms of the whole diesel engine plant are not found in the relevant open literature. Moreover, a compression ratio investigation is performed in terms of minimum combustion and turbocharger irreversibilities for the particular engine.

FIRST-LAW (ENERGY) ANALYSIS

General description of thermodynamic model

In order to simulate the operation of the IDI turbocharged diesel engine under consideration a thermodynamic model is developed, the individual submodels of which are briefly described in the following subsections. There is a uniformity in space (mono-zone) of pressure, temperature and composition in the combustion chamber at each instant of time. The fuel is assumed to be dodecane ($C_{12}H_{26}$), with a lower heating value of LHV = 42,000 kJ/kg. The first law of thermodynamics is applied to the engine cylinder for both the main chamber and the prechamber, on a degree crank angle step $d\varphi$ basis, and is solved together with the perfect gas state equation [22]. Internal energy is considered to be a function of temperature and equivalence ratio; relevant polynomial expressions proposed by Krieger and Borman [23] are used for each of the four species considered, i.e. O_2, N_2, CO_2 and H_2O.

Fuel injection rate

For the simulation of the fuel injection rate (kg/s), the expression proposed by Ferguson [7] is used:

$$\frac{\dot{m}_{fi}}{M_{tot}} = \frac{\omega}{\varphi_d \Gamma(n)}\left(\frac{\varphi - \varphi_s}{\varphi_d}\right)^{n-1} \exp\left(\frac{(\varphi_s - \varphi)}{\varphi_d}\right), \tag{1}$$

where $\ln \Gamma(n) = (n - 0.5)\ln(n) - n + 0.5\ln(2\pi) + 1/12n - 1/360n^3 + 1/1260n^5$, with $n = 3.6$ for the present study. In the above expression, M_{tot} is the total mass of injected fuel per cycle and per cylinder, φ_s is the crank angle where injection begins and φ_d is the duration of injection.

Combustion model

For the study of the combustion process the model proposed by Whitehouse and co-workers [5, 6] is used for both the main chamber and the prechamber. In this model the combustion process consists of two parts; a preparation-limited combustion rate and a reaction-limited combustion rate. The corresponding equations are

$$P = K_1 M_i^{1-x} M_u^x p_{O_2}^y \text{ (kg/°CA)} \tag{2}$$

for the preparation rate, while for the reaction rate it holds

$$R = \frac{K_2 p_{O_2}}{N\sqrt{T}} e^{-act/T} \int (P - R)\, d\varphi \text{ (kg/°CA)}, \tag{3}$$

$M_i = \int_{\varphi_s}^{\varphi} (dm_{fi}/d\varphi)d\varphi$ is the total mass (kg) of injected fuel up to the time t (corresponding angle φ) considered, and $(dm_{fi}/d\varphi)$ is the known injection rate from equation (1). $M_u = M_i - \int P\, d\varphi$ is the total mass (kg) of unprepared fuel, *act* is the reduced activation energy (K) accounting also for the ignition delay and p_{O_2} is the partial pressure of oxygen (bars) in the main chamber or the prechamber. The constant K_1 in the preparation rate equation is based upon the Sauter mean diameter (SMD) of the fuel droplets [6], being expressed by a formula of the type $K_1 \propto (1/\text{SMD})^2$. For the evaluation of SMD the empirical expression proposed by Hiroyasu *et al.* [24] is used:

$$\text{SMD} = 25.1(\Delta p)^{-0.135} \rho_g^{0.12} V_{tot}^{0.131} \text{ } (\mu\text{m}), \tag{4}$$

where Δp is the mean pressure drop across the nozzle in MPa, ρ_g is the density of air at the time the injection starts in kg/m^3 and V_{tot} is the amount of fuel delivered per cycle per cylinder in mm^3/stroke.

Heat transfer model

The model of Annand is used to simulate the heat loss to the cylinder walls for both the main chamber and the prechamber [25, 26]:

$$\frac{dQ_l}{dt} = F\left[a\frac{\lambda}{D}\,\mathrm{Re}^b(T_w - T_g) + c(T_w^4 - T_g^4)\right], \tag{5}$$

where F is the surface for heat exchange $= 2\ (\pi D^2/4) + \pi DL$, with L the instantaneous cylinder height in contact with the gas and D the cylinder diameter. A single 'mean' wall temperature for the entire surface F is used constant in time. The Reynolds number $\mathrm{Re} = \rho\bar{c}D/\mu$ is calculated with a characteristic speed equal to the mean piston speed $\bar{c} = 2rN/30$. The constants a, b and c are derived from calibration against experimental data.

Turbocharger and aftercooler

The simulation of the turbocharger is accomplished by the use of manufacturer's data under steady-state operation.

The purpose of the aftercooler is to increase the density of the incoming air. By so doing it causes a pressure drop on the gas entering the cylinder. The gas temperature at the exit of the aftercooler is given by [27]

$$T_3 = T_2(1 - \varepsilon) + \varepsilon T_{cwi}\,, \tag{6}$$

where ε is the effectiveness of the aftercooler often expressed as a function of charge air mass flow rate, $\dot{m}_c$, and T_{cwi} is the cooling water inlet temperature;

$$\varepsilon = 1 - c_1\dot{m}_c^2 \tag{7}$$

For the pressure drop across the aftercooler it holds

$$p_2 - p_3 = c_2\dot{m}_c^2\,. \tag{8}$$

In equations (6) and (8) subscripts 2 and 3 denote aftercooler inlet and outlet conditions, respectively. These points, along with the other strategic points of the diesel engine plant, are shown in Fig. 1. Values for the constants c_1 and c_2 were derived during the experimental work.

Mechanical friction

For the calculation of friction inside the cylinder, the formula of Chen and Flynn [28] for turbocharged engines is used:

$$fmep = 0.137 + 0.005p_{max} + 0.162\bar{c}\,, \tag{9}$$

where *fmep* is the friction mean effective pressure (bars) and p_{max} is the maximum cylinder pressure in bars.

Solving the equations

The above-stated differential equations were solved simultaneously via the predictor–corrector method and with a user-defined step of calculation, which for the specific investigation was 1/4°CA for the burning and expansion periods and 1/2°CA for the compression and mass exchange periods. The computer program is written in FORTRAN 77 language and executed on an IBM-compatible 486 PC.

APPLICATION OF AVAILABILITY BALANCE EQUATION TO THE VARIOUS CONTROL VOLUMES OF THE WHOLE DIESEL ENGINE

Basic concepts and definitions

Availability (alternatively termed exergy) of a system, in a given state, is defined as the maximum reversible work that can be produced through the interaction of the system with its surroundings as it experiences thermal, mechanical and chemical equilibrium with its environment [1, 29]. For the purpose of this work and for reaching equilibrium, only thermal and mechanical availability terms are taken into account, while chemical availability is considered only due to reaction of the fuel to products. The atmosphere is considered to have a temperature of $T_o = 298.15$ K and a pressure of $p_o = 1.013$ bar.

General availability balance equation

For an open system experiencing mass exchange with the surrounding environment, the following equation for the availability on a time basis exists [1, 29]:

$$\frac{dA_{cv}}{dt} = \int_F \left(1 - \frac{T_o}{T}\right)\dot{q}\, dF - \left(\dot{W}_{cv} - p_o \frac{dV_{cv}}{dt}\right) + \sum_i \dot{m}_i b_i - \sum_e \dot{m}_e b_e - \dot{I}\,. \tag{10}$$

The above-stated terms have the following meaning:

1. dA_{cv}/dt: rate of change of control volume (i.e. cylinder, each manifold, etc) availability;
2. $\int_F(1 - (T_o/T)\dot{q}\, dF$: availability term for heat transfer, where $(1 - T_o/T)$ is the efficiency of the ideal Carnot cycle working between the same temperature levels, as the process in study, and $\dot{q}$ is the rate of heat flux to or from the working medium exchanged through differential surface area dF;
3. $(\dot{W}_{cv} - p_o(dV_{cv}/dt))$: availability term associated with work transfer;
4. $\Sigma_i \dot{m}_i b_i - \Sigma_e \dot{m}_e b_e$: availability terms associated with inflow and outflow of masses, respectively;
5. $\dot{I}$: rate of irreversibility production inside the control volume due to combustion, throttling, mixing, heat transfer under finite temperature difference to cooler medium, etc.

In the term (4) above, the expression

$$b = h - h_o - T_o(s - s_o) \tag{11}$$

is defined as the flow availability [1].

In the following subsections, equation (10) is applied explicitly to each control volume of the diesel engine plant, on a °CA basis.

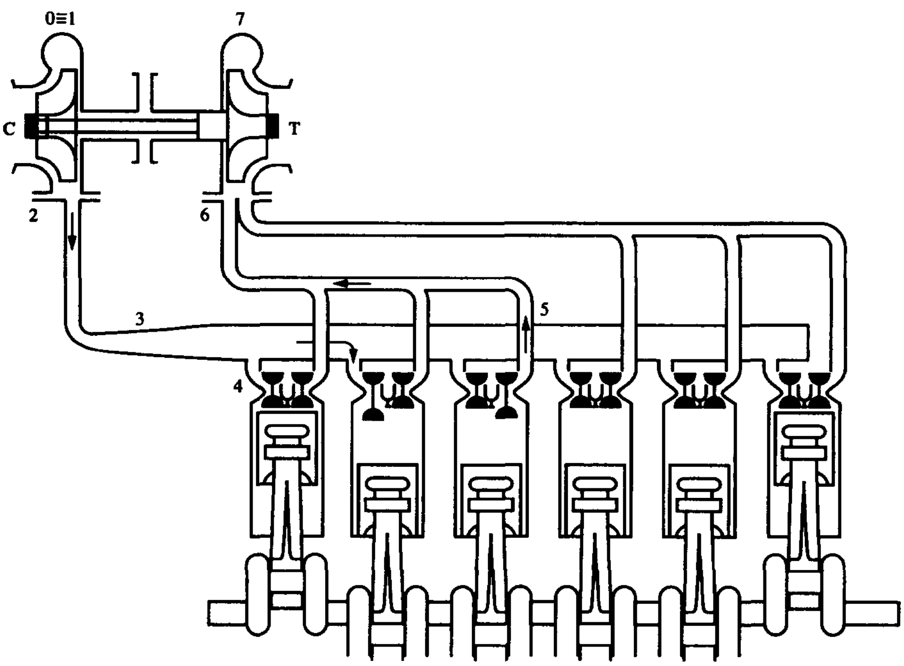


Fig. 1. General description of the engine–manifolds–turbocharger arrangement, for the MWM TbRHS 518S engine, showing strategic point locations used in the analysis.

Cylinder

Equation (10) in this case becomes

$$\frac{dA_{cyl}}{d\varphi} = \frac{\dot{m}_4 b_4 - \dot{m}_5 b_5}{6N} - \frac{dA_w}{d\varphi} + \frac{dA_l}{d\varphi} + \frac{dA_f}{d\varphi} - \frac{dI}{d\varphi}, \tag{12}$$

where the terms on the right-hand side of the above equation are

$$\frac{dA_w}{d\varphi} = (p - p_o)\frac{dV}{d\varphi} \tag{13}$$

for work transfer and

$$\frac{dA_l}{d\varphi} = \frac{dQ_l}{d\varphi}(1 - T_o/T) \tag{14}$$

for heat transfer to the cylinder walls with $dQ_l/d\varphi$ provided by Annand's correlation [equation (5)], with T the instantaneous cylinder temperature,

$$\frac{dA_f}{d\varphi} = \frac{dm_{fb}}{d\varphi} a_{fch} \tag{15}$$

is the flow availability associated with the burning of fuel, which for hydrocarbon liquid fuels of the general type $C_m H_n$ is given by [1]

$$a_{fch} = \mathrm{LHV}\left(1.04224 + 0.011925\frac{n}{m} - \frac{0.042}{m}\right). \tag{16}$$

For the present analysis

$$m = 12,\ n = 26 \text{ and } a_{fch} = 1.064\mathrm{LHV}. \tag{17}$$

The fuel-burning rate $dm_{fb}/d\varphi$ is established using the Whitehouse–Way model [equations (2) or (3)].

The term on the left-hand side of equation (12),

$$\frac{dA_{cyl}}{d\varphi} = \frac{dU}{d\varphi} + p_o\frac{dV}{d\varphi} - T_o\frac{dS}{d\varphi} - \frac{dG_o}{d\varphi}, \tag{18}$$

is the change in the availability of the cylinder contents [4]; its various terms are given below. The $dU/d\varphi$ term is the differential change in internal energy of the cylinder contents:

$$\frac{dU}{d\varphi} = \sum_{i=1}^{4} u_i\frac{dm_i}{d\varphi} + \sum_{i=1}^{4} m_i c_{vi}\frac{dT}{d\varphi}, \tag{19}$$

where m_i is the mass of species i, c_v is the specific heat under constant volume (a function of temperature only, $c_v = du/dT$) and m is the mass of the whole mixture.

The rate of entropy change of the cylinder contents term is

$$\frac{dS}{d\varphi} = \sum_{i=1}^{4}\frac{dm_i}{d\varphi}s_i(T,x_i p) + \sum_{i=1}^{4}\frac{m_i}{T}c_{pi}\frac{dT}{d\varphi} - \frac{V}{T}\frac{dp}{d\varphi}, \tag{20}$$

where $s_i(T,x_i p) = s_i'(T,p_o) - R\ln(x_i\, p/p_o)$ with R the specific gas constant. Term $s_i'(T,p_o)$ is a function of temperature only [1], and x_i is the molar fraction of species i in the mixture. The last term in equation (18) is

$$\frac{dG_o}{d\varphi} = \sum_{i=1}^{4}\frac{dm_i}{d\varphi}\mu_i^o, \tag{21}$$

where $\mu_i^o = g_i(T_o,x_i p_o)$ is the chemical potential of species i at ambient conditions:

$$g_i(T_o,x_i p_o) = h_i(T_o) - T_o s_i(T_o,x_i p_o) = h_i(T_o) - T_o[s_i'(T_o,p_o) - R\ln(x_i)]. \tag{22}$$

For the closed part of the cycle, $\dot{m}_4 = \dot{m}_5 = 0$. When the inlet valve is open, $\dot{m}_4$ is computed according to compressible flow through a restriction [7, 8] and $b_4 = h_4 - h_o - T_o (s_4 - s_o)$, where h_4 and s_4 are evaluated at conditions 4 (inlet manifold outlet state, see also Fig. 1). Similar expressions exist when the exhaust valve is open; $\dot{I}$ is the rate of irreversibility production inside the cylinder.

Turbocharger

For the compressor and turbine no control volume exists. Also, heat losses are usually negligible so that they are not taken into account. Equation (10) for the compressor becomes

$$\dot{m}_C(b_2 - b_1) + \frac{dI_C}{d\varphi} 6N = -\dot{W}_C, \tag{23}$$

while for the turbine it becomes

$$\dot{m}_T(b_6 - b_7) - \frac{dI_T}{d\varphi} 6N = \dot{W}_T. \tag{24}$$

In the last two equations the terms $\dot{W}_C$ and $\dot{W}_T$ are evaluated from the thermodynamic analysis of the turbocharger at each crank angle step via instantaneous values from the turbomachines steady-state maps. Subscripts 1, 2 and 6, 7 denote compressor and turbine inlet and outlet conditions, respectively, as shown in Fig. 1.

Inlet manifold

General equation (10) states, for the inlet manifold,

$$\frac{dA_{im}}{d\varphi} = \frac{\dot{m}_3 b_3 - \dot{m}_4 b_4}{6N} - \frac{dI_{im}}{d\varphi}, \tag{25}$$

where b_3 is the flow availability at the intake manifold inlet evaluated at aftercooler outlet conditions. No heat losses are taken into account. The above-stated term for irreversibilities $dI_{im}/d\varphi$ accounts for the mixing of aftercooler air with intake manifold residual contents.

Exhaust manifold

General equation (10) states, for the exhaust manifold,

$$\frac{dA_{em}}{d\varphi} = \frac{\dot{m}_5 b_5 - \dot{m}_6 b_6}{6N} - \frac{dI_{em}}{d\varphi} + \frac{dA_{lem}}{d\varphi}, \tag{26}$$

where index 5 denotes exit conditions from the cylinder (Fig. 1).

The term $dA_{lem}/d\varphi = (dQ_{lem}/d\varphi)(1 - T_o/T_{em})$ accounts for the heat losses at the exhaust manifold, where T_{em} is the instantaneous temperature of the manifold contents. The term $dI_{em}/d\varphi$ accounts for the irreversibility rate in the exhaust manifold, which consists of throttling across the exhaust valve, mixing of cylinder exhaust gases with manifold contents and friction along the manifold length. As stated in ref. [2], since the magnitude of each of the three above-mentioned irreversibilities terms is rather small, it is not considered worthwhile to calculate each seperate term.

For both inlet and exhaust manifolds control volume availability terms $dA_{im}/d\varphi$ and $dA_{em}/d\varphi$, respectively, similar expressions exist [equation (18)] as for the cylinder term, with the exception of the $p_o\, dV/d\varphi$ term which is zero.

Cumulative availability terms

It is obvious that the cumulative terms associated with the change of availability of control volume for the cylinder [equation (18)], inlet manifold and exhaust manifold sum up to zero in a full cycle of the working medium, since initial and final conditions are the same for the steady-state operation.

The previously stated equations (12), (23), (24), (25), (26) for the availability balance can be solved for the rate of irreversibility term $\dot{I}$, which is the only unknown term. From the irreversibility rate terms it is then easy to calculate (by integration) the corresponding cumulative terms at each

Table 1. Engine basic design data. MWM TbRHS 518S—marine duty diesel engine

Type	In-line, six-cylinder, four-stroke, IDI
Firing order	1-5-3-6-2-4
Bore	140 mm
Stroke	180 mm
Connecting rod length	350 mm
Compression ratio	17.7
Prechamber volume	43.332 cm^3
Main chamber dead volume	122.586 cm^3
Intake valve opening	51°CA before top dead centre
Intake valve closure	60°CA after bottom dead centre
Exhaust valve opening	64°CA before bottom dead centre
Exhaust valve closure	47°CA after top dead centre
Maximum power	320 HP (235 kW) at 1500 rpm
Maximum torque	1500 Nm at 1250 rpm

degree CA and at the end of the cycle, which are needed for the parametric study of the load and speed effects.

EXPERIMENTAL FACILITIES AND PROCEDURE

The experimental procedure, necessary for the calibration of the thermodynamic model, was conducted on a MWM type TbRHS 518S, six-cylinder, four-stroke, indirect injection, medium-high speed, turbocharged and aftercooled diesel engine of marine duty. The basic engine design data are given in Table 1. A schematic arrangement of the engine, showing turbocharger, manifolds and cylinder interconnections is given in Fig. 1.

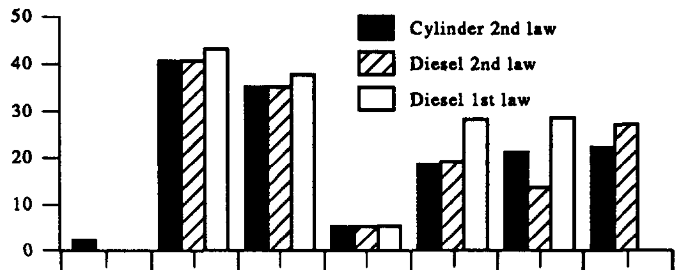
The engine is fitted with a Kühnle–Kopp–Kausch (KKK) turbocharger, a water aftercooler after the turbocharger compressor and is coupled to a Schenck hydraulic dynamometer for measuring the engine performance. A special characteristic of this engine is the almost constant full-load torque irrespective of speed. Details about the test installation can be found in refs [20] and [30].

A detailed experimental investigation of the engine processes was performed for all combinations of three values of the engine load, corresponding to 50, 75 and 100% of the engine torque at 1000 rpm, and three engine speeds of 1000, 1250 and 1500 rpm (nine runs). The experimental results obtained were used to calibrate the thermodynamic model from the first-law analysis point of view [31]. The appropriate Whitehouse–Way combustion and Annand heat transfer model constants were estimated, and the exact operating point of the turbocharger was calculated for each combination of speed and load. The computer program was then run with the availability subroutines 'turned on' for calculation of the availability balances across the engine, turbocharger and manifolds.

RESULTS AND DISCUSSION

The generalized equation (10) is applied for every process and for each one of the nine operating points of the diesel engine and the results are presented in Figs 2 – 10. Figures 2 and Fig. 3 refer to the full load–maximum speed operation case with tabulation of the various availability and irreversibility terms, while Figs 4 – 9 present the effects of speed and load, and Fig. 10 performs a compression ratio investigation from both first- and second-law perspectives. It is noted that the availability terms for manifolds, turbocharger and aftercooler were averaged on the basis of one cylinder, so that direct comparison with the in-cylinder availability terms is possible.

Figure 2 provides the tabulation of energy and availability terms for the full load–maximum speed operating point. As can be seen from this figure values for the indicated work, brake work and mechanical friction between first- and second-law analysis differ a little and this is due to the factor of 1.064, which differentiates the fuel's chemical availability from its lower heating value. On the contrary, heat losses to the cylinder walls, aftercooler and exhaust manifold walls present quite a difference between first- and second-law assessments, i.e. 28.30 versus 18.95%, as pointed out also by other researchers in the field. The term $(1 - T_0/T)$, which transforms the amount of heat rejected to possible recoverable work is responsible for this difference. Typical values are also observed when comparing the exhaust gas terms.



	Inlet air	Ind. work	Br. work	Friction	Heat	Exh. gas	Irrever.
Cylinder 2nd law	2.31	40.31	35.27	5.04	18.32	21.38	22.3
Diesel 2nd law	0	40.31	35.27	5.04	18.95	13.45	27.29
Diesel 1st law	0	42.89	37.53	5.36	28.3	28.81	0

Fig. 2. Tabulation of first- and second-law terms for the cylinder and the whole diesel engine plant for the maximum speed–full load operation.

The amount of irreversibilities inside the cylinder is 22.30% of the fuel's (chemical) availability, while for the whole plant this number increases to 27.29%.

The tabulation of the irreversibilities for the various devices and processes encountered is given in Fig. 3, which shows clearly the importance of the combustion term (21.90% of the fuel's chemical availability or 80.25% of the total irreversibilities). The other terms, though smaller in magnitude, should not be ignored, especially the exhaust manifold one, which comprises as much as 10% of the total irreversibilities. The relatively lower isentropic efficiency of the compressor is responsible for its lower second-law efficiency and this leads to a higher amount of irreversibilities than the turbine.

Figures 4 – 9 present the parametric study of the load and speed effects on the availability balances and irreversibilities production. Figure 4 in particular shows the variation of the indicated work, the brake work and the mechanical friction with speed and load, when these are reduced with respect to the fuel chemical availability.

When the variation with load is considered, one can observe that the mechanical friction term decreases with increasing load because the fuel chemical availability increases (the mass of fuel increases), while the absolute (non-reduced) value for the mechanical friction remains essentially unaltered. According to equation (9) the mechanical friction depends on both the maximum pressure (which is effected mainly by load but in a weak way) and mean piston speed, which makes the dominant contribution. The indicated efficiency shows a very slight dependence (increase) on

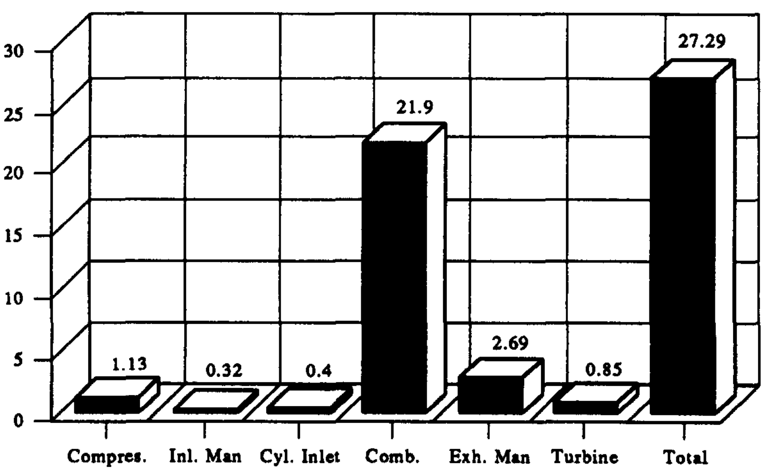


Fig. 3. Quantification of various devices and processes' irreversibilities for the maximum speed–full load operation.

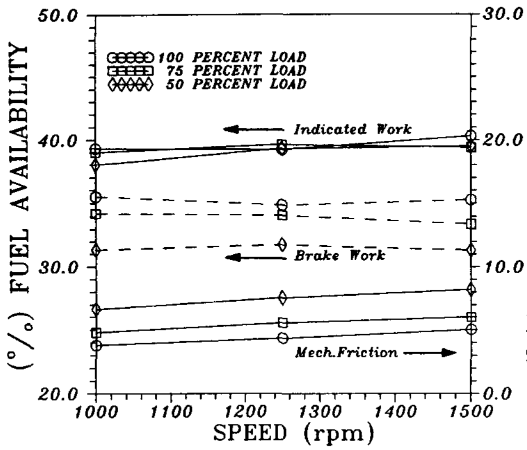


Fig. 4. The effect of speed and load on the indicated work, the brake work and the mechanical friction.

load, while the brake efficiency increases steadily with increasing load due to the decrease in the mechanical friction term.

When the variation with speed is considered, one can observe that the mechanical friction term now increases with increasing speed, also causing a slight decrease in the brake efficiency term. The term of indicated work (i.e. indicated efficiency), in contrast, presents a slight increase with increasing speed, which is mainly due to the fuel–air equivalence ratio f decreasing with increasing speed, a fact causing, in general, an improvement in fuel exploitation [8].

Figure 5 shows the variation of the combustion irreversibilities [found by integration of the term $dI/d\varphi$ in equation (12) over the period of the engine cycle when both valves are closed] and total irreversibilities with speed and load, when these are reduced with respect to the engine indicated work.

When the variation with load is considered, one can observe that the amount of irreversibilities (as a percentage of the indicated work) decreases with increasing load. This is due to the fact that combustion irreversibilities fall with increasing load. The amount of combustion irreversibilities depends heavily on the fuel–air equivalence ratio f. Greater values of f cause greater temperatures inside the cylinder (that means less degradation of the fuel chemical availability when transferred to the exhaust gases) and less mixing of exhaust gases with air during combustion and expansion, during which the dominant part of the in-cylinder irreversibilities is produced.

When the variation with speed is considered, one can observe that the combustion irreversibilities show a peak at about 1250 rpm for all engine loads examined, while total irreversibilities increase steadily with increasing speed (although at a lower rate after 1250 rpm). The explanation for the

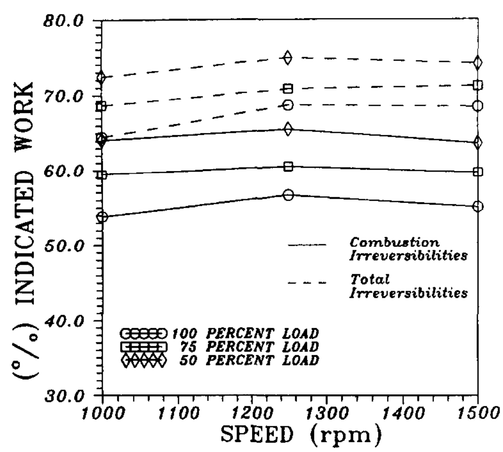


Fig. 5. The effect of speed and load on combustion and total irreversibilities.

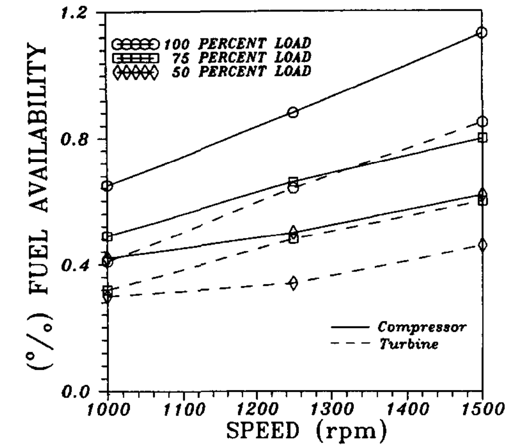


Fig. 6. The effect of speed and load on compressor and turbine irreversibilities.

combustion term behaviour lies in its previously mentioned dependence on load. For a turbocharged engine, increasing speed causes an increase in the amount of air inducted into the cylinders, thus decreasing the equivalence ratio, while simultaneously slightly increasing the indicated work (Fig. 4). The combined effects of these two factors produce the maximum observed at 1250 rpm. The total irreversibilities term, in comparison, shows an increase with increasing speed and this is due to the corresponding increase in the turbocharger's and the manifolds' availability destructions, as will be shown later.

Figure 6 shows the variation of the compressor and turbine irreversibilities with speed and load, when these are reduced with respect to the fuel chemical availability.

When the variation with load is considered, one can observe that compressor and turbine irreversibilities increase with increasing load for every engine speed. Bearing in mind that the nature of irreversibilities inside compressors and turbines is mainly connected with flow restrictions (throttling) and friction, this behaviour is explained because an increase in load causes an increase in both the amounts of air compressed and exhaust gases expanded, as well as in their pressure and temperature.

When the variation with speed is studied similar remarks can be made.

Figure 7 concentrates on the in-cylinder terms. The variation of the availability terms of cylinder inlet air [integration of term $\dot{m}_4 b_4/6N$ in equation (12) over the engine cycle], cylinder exhaust gas [integration of term $\dot{m}_5 b_5/6N$ in the same equation] and cylinder heat loss [integration of term

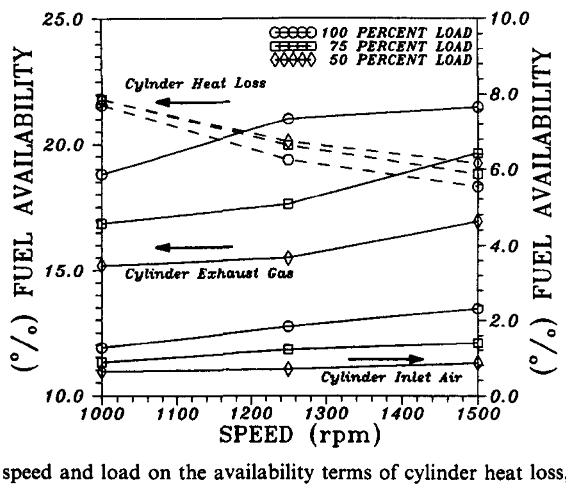


Fig. 7. The effect of speed and load on the availability terms of cylinder heat loss, inlet air and exhaust gases.

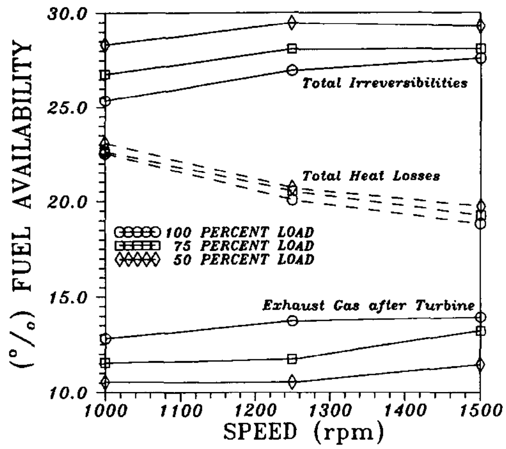


Fig. 8. The effect of speed and load on the availability terms of diesel engine plant total heat losses, exhaust gas after turbine and total irreversibilities.

$dA_l/d\varphi$ in the same equation] are given with speed and load, reduced with respect to the fuel chemical availability.

When the variation with load is considered, one can observe that the availability term for the inlet air presents an increase with increasing load; this is expected because of the larger amounts of consumed air possessing higher pressure and temperature. The same trend is observed for the exhaust gases. Heat loss to the walls shows, on the contrary, a (slight) decrease with increasing load. This is explained by the fact of increasing the in-cylinder temperatures with increasing load, which is, however, more than offset by the greater rate of increase in the fuel chemical availability.

When the variation with speed is considered, one can observe that both the inlet air and exhaust gas' availability terms show an increase with increasing speed and this is due to the increase in both the amount of air consumed and the values of its temperature and pressure, a fact which is true because of the engine under study being a turbocharged one. Conversely, increasing speed causes a decrease in available time for the processes inside the cylinder, as well as a decrease in the temperatures inside the cylinder, due to the corresponding decrease in the fuel–air equivalence ratio. These two factors prevail over the effect of increasing heat transfer coefficients [equation (5)] with increasing speed, so that finally a decrease (with increasing speed) is observed in the amount of heat lost to the walls (reduced to the fuel chemical availability).

Figure 8 shows the variation of the total irreversibilities and availability terms for the exhaust gas after turbine and total heat losses, with speed and load, when these are reduced with respect to the fuel chemical availability. The term total heat losses refers to the availability term for the

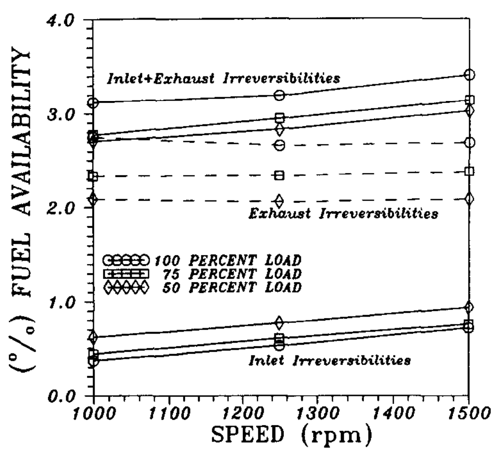


Fig. 9. The effect of speed and load on inlet and exhaust irreversibilities.

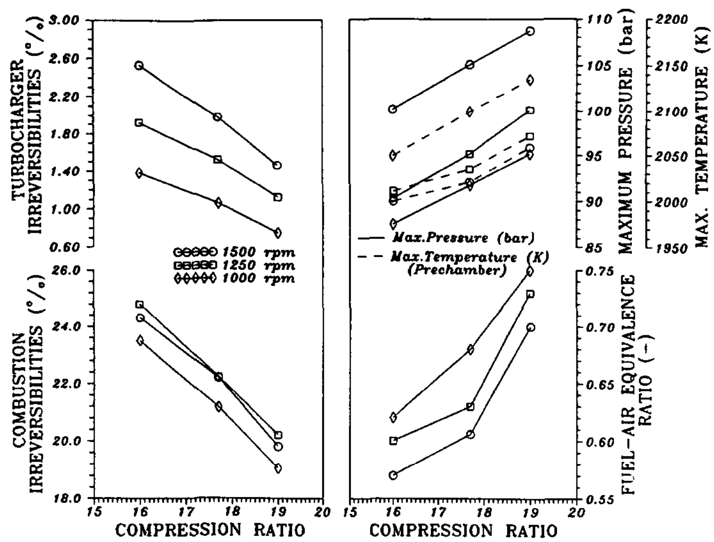


Fig. 10. The effect of compression ratio on combustion and turbocharger irreversibilities.

sum of heat loss from cylinders, aftercooler and exhaust manifold, while the exhaust gas term is defined from integration of the $(\dot{m}_T b_7)/6N$ term in equation (24) over the engine cycle.

When the effect of load is considered, one can observe that the amount of exhaust gas available from the turbine increases with increasing load, due to the increase in both the amount of gas blown out of the engine cylinders and its temperature. The total irreversibilities, conversely, decrease with increasing load, as shown in Fig. 5, with the difference now that the effect of the indicated work has been obliterated. The availability term for the total heat losses shows a similar behaviour as its in-cylinder counterpart (Fig. 7), i.e. a slight decrease with increasing load.

When the effect of speed is considered, one can observe that the total irreversibilities term increases with increasing speed (Fig. 5). The same applies to the availability term for the exhaust gas after the turbine (Fig. 7), while the availability term for the total heat losses decreases due to the decrease in available time.

A comparison between Fig. 7 and Fig. 8 shows that about one-third of the amount of exhaust gas available is lost in the exhaust manifold and in the turbine expansion for driving the compressor, while the amount of heat lost to the walls increases only by 4–5% when the aftercooler and exhaust manifold wall heat losses are also considered; this last fact is due to the relatively low temperature of the air along the aftercooler and of the gas along the exhaust manifold, which according to equation (14) produces a rather small value of the second-law term.

Figure 9 presents the variation of inlet and exhaust process irreversibilities with speed and load, when these are reduced with respect to the fuel chemical availability. The term inlet irreversibilities consists of both inlet manifold and cylinder inlet process irreversibilities, while the exhaust term consists of exhaust manifold and cylinder exhaust process ones. Exhaust irreversibilities are considerably greater than the inlet ones (since the pressures and temperatures involved are much higher). What matters here in terms of irreversibility production is high pressures (for throttling across valves), high temperature differences (for mixing) and high mass flow rates (for friction).

When the effect of load is considered, one can observe that the exhaust irreversibilities show an increase with increasing load due to the corresponding increase in both the amount of exhaust gases expanded and their pressure and temperature. The inlet irreversibilities, in contrast, decrease with increasing load for every engine speed, mainly because the fuel chemical availability increases with load at a stronger rate.

When the variation with speed is considered, one can observe that inlet, exhaust and total irreversibilities increase with increasing speed due to the corresponding increase in the amounts of air consumed and exhaust gases produced.

Figure 10 provides an investigation in terms of combustion and turbocharger irreversibilities with respect to the engine compression ratio ε, when the pressure at the start of injection as well as the injected fuel per cycle and cylinder are kept constant and correspond to 100% load of the real case, for each engine speed. This means that the compressor experiences lower pressure ratios r with increasing ε (i.e. for the 1500 rpm case: $r = 2.0$ for $\varepsilon = 16$, $r = 1.8$ for the nominal case of $\varepsilon = 17.7$, and $r = 1.6$ for $\varepsilon = 19$, etc.).

As can be concluded from Fig. 10, the combustion irreversibilities fall with increasing compression ratio, due to the corresponding decrease in the compressor pressure ratio which leads to greater fuel–air equivalence ratios (Fig. 5). The same holds for the turbocharger irreversibilities, where here the decrease in the compressor pressure ratio more than offsets the increase in the turbine pressure and temperature level [as implied by the third subdiagram of Fig. 10, which shows the maximum firing (prechamber) pressures and temperatures of each cycle]. As a result the amount of the total irreversibilities also decreases with increasing ε.

It seems that as far as the second-law analysis is concerned, higher compression ratios lead to lower amounts of combustion and total irreversibilities and are thus favourable, at least for the values of compression ratio near the real one. Nevertheless, the increase in the fuel–air equivalence ratio leads to the formation of higher amounts of soot (especially at transient conditions), while the corresponding increase in maximum firing pressures and temperatures causes greater mechanical and thermal stresses, which in turn require sturdier engine construction (and higher cost) and can lead to possible problems in the turbine blades. The decrease in the compressor pressure ratio leads also to problematic performance of the engine at lower speeds and loads decreasing the obtained work.

When keeping the fuel–air equivalence ratio constant for each engine speed examined, the combustion irreversibilities remain essentially unaltered for each value of compression ratio, as the fuel–air equivalence ratio plays the most significant role. The turbocharger irreversibilities, in contrast, present a trend as shown in Fig. 10.

CONCLUSIONS

A second-law analysis was performed on a six-cylinder, turbocharged and aftercooled, indirect injection diesel engine. For this purpose a single-zone thermodynamic model was used and tested favourably against experimental results. Availability equations were applied on every part of the diesel engine plant and tabulation of availability and irreversibility terms were given and discussed. Various kinds of irreversibilities (i.e. compressor, inlet, exhaust, combustion, turbine) were identified and quantified. The effect of speed, load and compression ratio was studied on all second-law terms involved.

Combustion irreversibilities are the main source of availability destruction for every operating point, but the throttling, friction and thermal mixing losses encountered in the turbocharger and inlet–exhaust manifolds destructions should also not be ignored, since they comprise as much as 20% (maximum) of the total irreversibilities.

Exhaust manifold irreversibilities contribute as much as 10% of the total irreversibilities, thus showing one process, besides combustion, which the first-law theory fails to describe fully.

The increase in load causes an increase in indicated efficiency, compressor and turbine irreversibilities, cylinder inlet air and exhaust gas availability terms and exhaust manifold irreversibilities. Inlet irreversibilities and mechanical friction, in comparison, decrease with increasing engine load; the same applies to combustion and total irreversibilities and cylinder heat loss.

The increase in speed causes an increase in mechanical friction, turbocharger, inlet and total irreversibilities, cylinder inlet air and the amount of exhaust gases available. Combustion irreversibilities show a peak at about 1250 rpm for every engine load. The increase in speed causes a decrease in cylinder heat loss. Finally, exhaust manifold irreversibilities remain unaltered by engine speed (for a specific load).

The compression ratio plays a significant role, affecting combustion and turbocharger irreversibilities, but its optimisation under a second-law perspective should be seen under the light of serious changes incurred in the operational, constructional (cost) and environmental behaviour of the engine.

REFERENCES

1. M. J. Moran, *Availability Analysis: A Guide to Efficient Energy Use*. Prentice Hall, New Jersey (1982).
2. P. F. Flynn, K. L. Hoag, M. M. Kamel and R. J. Primus, A new perspective on diesel engine evaluation based on second law analysis. SAE Paper 840032 (1984).
3. R. J. Primus and P. F. Flynn, The assessment of losses in diesel engines using second law analysis. ASME–WA Meeting, Annaheim, CA. *Proc. AES*, pp. 61–68 (1986).
4. J. H. VanGerpen and H. H. Shapiro, Second law analysis of diesel engine combustion. *Trans. ASME J. Engng Gas Turbines & Power* **112,** 129–137 (1990).
5. N. D. Whitehouse and R. G. B. Way, Rate of heat release in diesel engines and its correlation with fuel injection data. *Proc. Inst. Mech. Engrs* **184** (Part 3J), pp. 17–29 (1969–1970).
6. R. S. Benson and N. D. Whitehouse, *Internal Combustion Engines*. Pergamon Press, Oxford (1979).
7. C. R. Ferguson, *Internal Combustion Engines*. John Wiley, New York (1986).
8. J. B. Heywood, *Internal Combustion Engine Fundamentals*. McGraw-Hill, New York (1988).
9. C. D. Rakopoulos, Influence of ambient temperature and humidity on the performance and emissions of nitric oxide and smoke of high-speed diesel engines in the Athens/Greece region. *Energy Convers. Mgmt* **31,** 447–458 (1991).
10. D. A. Kouremenos, C. D. Rakopoulos and E. Karvounis, Thermodymanic analysis of direct injection diesel engines by multi-zone modelling. ASME–WA Meeting, Boston, MA. *Proc. AES*, Vol. 3(3), pp. 67–77 (1987).
11. D. A. Kouremenos, C. D. Rakopoulos and D. T. Hountalas, Thermodynamic analysis of indirect injection diesel engines by two-zone modelling of combustion. *Trans. ASME, J. Engng Gas Turbines & Power* **112,** 138–149 (1990).
12. J. I. Ramos, *Internal Combustion Engine Modelling*. Hemisphere, New York (1989).
13. C. D. Rakopoulos, D. T. Hountalas, G. N. Taklis and E. I. Tzanos, Analysis of combustion and pollutants formation in a direct injection diesel engine using a multi-zone model. *Energy Res.* **19,** 63–88 (1995).
14. C. D. Rakopoulos, G. N. Taklis and E. I. Tzanos, Analysis of combustion chamber insulation effects on the performance and exhaust emissions of a DI diesel engine using a multi-zone model. *Heat Recovery Systems & CHP* **15,** 691–706 (1995).
15. A. C. Alkidas, The application of availability and energy balances to a diesel engine. *Trans. ASME, J. Engng Gas Turbines & Power* **110,** 462–469 (1988).
16. R. J. Primus, A second law approach to exhaust system optimization. SAE Paper 840033 (1984).
17. T. L. McKinley and R. J. Primus, An assessment of turbocharging systems for diesel engines from first and second law perspectives. SAE Paper 880598 (1988).
18. F. Bozza, R. Nocera, A. Senatore and R. Tuccilo, Second law analysis of turbocharged engine operation. SAE Paper 910418 (1991).
19. C. D. Rakopoulos, Evaluation of a spark ignition engine cycle using first and second law analysis techniques. *Energy Convers. Mgmt* **33,** 1299–1314 (1993).
20. C. D. Rakopoulos and E. C. Andritsakis, DI and IDI diesel engines combustion irreversibility analysis. ASME–WA Meeting, New Orleans, LA. *Proc. AES*, Vol. 30, pp. 17–32 (1993).
21. C. D. Rakopoulos, E. C. Andritsakis and D. K. Kyritsis, Availability accumulation and destruction in a DI diesel engine with special reference to the limited cooled case. *Heat Recovery Systems & CHP* **13,** 261–276 (1993).
22. C. D. Rakopoulos and E. G. Giakoumis, Development of cumulative and availability rate balances in a multi-cylinder, turbocharged, indirect injection diesel engine. *Energy Convers. Mgmt* (in press).
23. R. B. Krieger and G. L. Borman, The computation of apparent heat release for internal combustion engines. ASME Paper 66 WA/DGP-4 (1966).
24. H. Hiroyasu, T. Kadota and M. Arai, Development and use of a spray combustion modelling to predict diesel engine efficiency and pollutant emissions. *Bull. JSME* **26,** 569–576 (1983).
25. W. J. D. Annand, Heat transfer in the cylinders of reciprocating internal combustion engines. *Proc. Inst. Mech. Engrs* **177,** 973–990 (1963).
26. C. D. Rakopoulos and D. T. Hountalas, Net and gross heat release rate calculations in a DI diesel engine using various heat transfer models. ASME–WA Meeting, Chicago, IL. *Proc. AES*, Vol. 33, pp. 251–262 (1994).
27. N. Watson and M. S. Janota, *Turbocharging the Internal Combustion Engine*. MacMillan, London (1982).
28. S. K. Chen and P. Flynn, Development of a compression ignition research engine. SAE Paper 650733 (1965).
29. A. Bejan, *Advanced Engineering Thermodynamics*. John Wiley, New York (1988).
30. C. D. Rakopoulos, E. C. Andritsakis and D. T. Hountalas, The influence of the exhaust system unsteady gas flow and insulation on the performance of a turbocharged diesel engine. *Heat Recovery Systems & CHP* **15,** 51–72 (1995).
31. D. A. Kouremenos, C. D. Rakopoulos and K. G. Kotsos, Phasing cylinder pressure to crank angle in a direct injection diesel engine by simulation of compression curve and elaboration of measured pressure data. *VDI Forschung im Ingenieurwesen* **57,** 87–95 (1991).