Pergamon

Energy Convers. Mgmt Vol. 38, No. 4, pp. 347–369, 1997

0196-8904/97 $17.00 + 0.00

PII: S0196-8904(96)00055-6

DEVELOPMENT OF CUMULATIVE AND AVAILABILITY RATE BALANCES IN A MULTI-CYLINDER TURBOCHARGED INDIRECT INJECTION DIESEL ENGINE

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(Received 12 July 1995)

Abstract—A multi-cylinder turbocharged Diesel engine is analysed from a second law analysis point of view via a single-zone thermodynamic model. For this purpose, a comprehensive digital computer program is developed that incorporates subroutines to simulate, among other things, combustion, heat transfer, indirect fuel injection, mass flow through valves, turbocharger and aftercooler behaviour, and real multi-cylinder engine action. This is tested favourably against relevant data from an experimental investigation conducted at the authors' laboratory. A second law analysis is performed in all parts of the engine (cylinder, compressor, turbine, aftercooler, inlet and exhaust manifolds). The analysis describes explicitly all the availability terms existing, i.e. work, heat and mass transfer, availability accumulation in every control volume and fuel flow, thus providing the proper evaluation of every component's irreversibilities, which for the present study are compressor, turbine, inlet, exhaust and combustion. The model is applied to a six-cylinder, turbocharged and aftercooled, indirect injection, four-stroke, medium-high speed diesel engine, installed at the authors' laboratory. Availability rate and cumulative availability terms, with respect to crank angle, for all the processes encountered are presented in diagrams, which show the trends of the availability accumulation and destruction in every component during an engine cycle. Separate diagrams are presented for the main chamber and the prechamber and also for the closed and open parts of the cycle. Tabulation of all second law analysis terms is given for the full load-maximum speed operation and the differences against first law assessments are discussed. Special attention is paid to the correct determination and explanation of the irreversibility quantification and second law efficiencies for every component and for the whole plant. Thus, it is demonstrated that the second law analysis offers a more spherical and comprehensive insight into the processes occurring in a diesel engine than its traditional first law counterpart. Copyright © 1996 Elsevier Science Ltd

Turbocharged diesel engine   Second law analysis   Irreversibilities

NOMENCLATURE

A = Availability (J)
a = Specific availability (J/kg)
act = Reduced activation energy (K)
b = Flow availability (J/kg)
$\bar{c}$ = Mean piston speed (m/s)
c_d = Discharge coefficient
c_p = Specific heat capacity under constant pressure (J/kg K)
c_v = Specific heat capacity under constant volume (J/kg K)
D = Cylinder diameter (m)
F = Surface (m^2)
fmep = Friction mean effective pressure (bar)
G = Gibbs free enthalpy (J)
g = Specific Gibbs free enthalpy (J/kg)
H = Enthalpy (J)
h = Specific enthalpy (J/kg)
I = Irreversibility (J)
K = Preparation constant in Whitehouse–Way model (bar$^{-0.3}$/deg)
K_1 = Reaction rate constant in Whitehouse–Way model (K$^{0.5}$/bar s deg)
LHV = Fuel lower heating value (J/kg)
M, m = Mass (kg)
$\dot{m}$ = Mass flow rate (kg/s)
N = Engine speed (rpm)
p = Pressure (N/m^2)
Q = Heat (J)

q = Heat flux (J/m^2 s)
R = Specific gas constant (J/kg K)
r = Crank radius (m)
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)
x = Molar fraction
x, y = Exponents in Whitehouse–Way model

Greek letters

γ = Specific heat capacities ratio
ε = Aftercooler effectiveness or Second law efficiency
η = First law efficiency
λ = Gas thermal conductivity (W/m K)
μ = Dynamic viscosity (kg/m s) or Chemical potential (J/kg)
ρ = Density (kg/m^3)
φ = Crank angle (degrees)
ω = Angular velocity (s^{-1})

Subscripts

ac = Aftercooler
b = Burning
C = Compressor
c = Charge air
ch = Chemical
cv = Control volume
cw = Cooling water
cyl = Cylinder
e = Exhaust, Exit
f = Fuel
g = Gas
i = Inlet, Injected (fuel)
i = Species
j = Exchanged (mass)
l = Heat loss
m = Manifold
o = Ambient (environment, atmosphere) conditions
T = Turbine
tot = Total
w = Wall, Work

Superscripts

o = Dead state
w = Water (aftercooler)
$\cdot$ = Time derivative (rate)

Abbreviations

A = After
B = Before
BDC = Bottom dead centre
CA = Crank angle (degrees)
EVC = Exhaust valve closure
EVO = Exhaust valve opening
IVC = Intake valve closure
IVO = Intake valve opening
rpm = Revolutions per minute
TDC = Top dead centre

INTRODUCTION

Diesel engine simulation modelling has long been established as an effective tool for studying engine performance and contributing towards new evaluation and development. Simple models accounting for the basic features of engine operation, treating the cylinder contents as a uniform mixture, usually termed single-zone models, have been developed and still continue to exist due to their simplicity and low computational cost combined with reasonable accuracy [1–5]. Apart from these models, the urgent need to control pollutant emissions from internal combustion engines has led to the development of other more complicated models, such as two-zone, four-zone or even multi-zone models, which furnish increased accuracy and flexibility for such complex phenomena as the formation of nitric oxide and soot in engine cylinders [6–14].

On the other hand, it has long been understood that traditional first law theory, although properly expressed and explicitly applied, often fails to give the engineer the best insight into the engine's operation; this applies certainly to both spark and compression ignition engines [15–17]. At this point comes the second law analysis, with its more careful and 'interior' study of what is happening during a process, to contribute a new way of thinking and studying thermodynamic processes. One of its fundamental points is the differentiation between work and heat. All forms of energy are no longer equivalent on an energy basis, but the ability of a certain form to convert fully into another one is the key for its superiority. Thus, work is considered as a better form of energy, since it can fully convert into heat, in contrast to heat whose ability to convert into work is dependent upon its temperature in a way that the well-known ideal Carnot cycle efficiency suggests. As a result of this, the definition of the term availability exists, which means the potential to do work. Unlike energy, availability can also be destroyed due to such phenomena as combustion, friction, mixing and throttling, which decrease the ability to produce work. The term of availability is combined with that of the surrounding environment with which the device under investigation (such as the piston internal combustion engine) exchanges work, heat and mass. The destruction of availability, otherwise termed irreversibility, in its various forms is the source for the defective exploitation of fuel into useful mechanical work in a diesel (and Otto) engine. All in all, second law analysis provides the engineer with the ability to identify those processes encountered which reduce the capability of producing work and quantify these losses so that the weight of each loss is properly taken into account [18].

Second law analysis techniques started early in the 60s, but the most important studies have been published during the last two decades. Availability balance equations were applied to a diesel engine on an overall basis in Ref. [19], but mainly extensive and detailed works, where the availability equations were coupled explicitly with theoretical simulation models of the engine cycle, have been reported in Refs [15–17, 20–25]. Works reported by various Cummins Engine Company researchers have dealt so far with applying the availability balances on the cylinder [16] as well as on the whole diesel plant [17, 20], or even the exhaust manifold alone [21]. In the previous publications, a quantification of engine irreversibilities was given, and throttling and thermal mixing losses were discussed along with the well-established combustion irreversibilities. Van Gerpen and Shapiro [15], on the other hand, performed a detailed theoretical analysis for the closed part of the cycle, bringing into focus the controversial term of chemical availability along with the thermomechanical one. More recently, the effect of limited cooled engines on the availability balances was studied; 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, as reported in a detailed analysis by Rakopoulos *et al.* [22]. Also, DI and IDI diesel engine combustion irreversibilities were brought into focus through an in-depth analysis reported by Rakopoulos and Andritsakis [23]. Various types of second law efficiencies, together with a large parametric study of engine operating parameters, have been studied in Ref. [24]. 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 [25].

This work applies the second law analysis in all processes involved in a turbocharged diesel engine, i.e. compressor, aftercooler, inlet manifold, cylinder for both the closed and open parts of the cycle, exhaust manifold and turbine. For this purpose, a single-zone thermodynamic model, following the filling and emptying modelling technique, was developed. The paper describes the method by which availability analysis is applied and how the various kinds of availability degradations (apart from the dominant part of combustion irreversibilities) are evaluated.

The present work provides detailed diagrams of rate and cumulative availability terms for every device and each process, thus revealing the way in which availability accumulation and destruction are developed. It is strongly believed that a proper application of availability analysis in every engine component and the study of the development of each term's history is a vital step towards a better understanding of engine processes and the improvement of engine efficiency. Main chamber and prechamber availability terms, as well as such terms for the closed and open parts of the cycle, are given separately for a better presentation of all second law parameters. This is an original feature, since inlet and exhaust manifold manipulation within the perspective of the second law, as well as turbocharger analysis, have been, in the past, only rarely studied [17, 20, 24], while no results and explicit diagrams have been published for the multi-cylinder engine operation case.

FIRST LAW (ENERGY) ANALYSIS

General description of thermodynamic model

In order to simulate the operation of the IDI turbocharged diesel engine under consideration for both the closed and the open parts of the engine cycle, a thermodynamic model was 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. Because of the weak mixtures involved, dissociation is neglected. The fuel is assumed to be dodecane ($C_{12}H_{26}$), which closely represents the diesel oil used for high and medium speed diesel engines, with a lower heating value of LHV = 42,000 kJ/kg.

First law of thermodynamics

The first law of thermodynamics applied to the engine cylinder for either the main chamber or the prechamber, on a degree crank angle step $d\varphi$ basis, states (see [3]) that

$$\frac{dQ_l}{d\varphi} - p\frac{dV}{d\varphi} = \frac{dU}{d\varphi} - \sum \frac{dm_j}{d\varphi} h_j, \tag{1}$$

where dQ_l is the heat loss to the walls, p is the pressure of cylinder contents, dm_j is the mass exchanged (positive when entering) in the step $d\varphi$ and h_j is the enthalpy of it. The subscript j denotes, for the main chamber, exchange with the exhaust manifold, inlet manifold and prechamber while, for the prechamber, it denotes fuel injection and mass exchange with the main chamber. dU is the change in internal energy of the cylinder contents. The internal energy is considered to be a function of temperature and equivalence ratio; relevant polynomial expressions proposed by Krieger and Borman [26] are used for each of the four species considered, i.e. O_2, N_2, CO_2 and H_2O denoted by index i (1 to 4, respectively).

Assuming that the gas medium obeys the perfect gas law $pV = mRT$, and differentiating with respect to crank angle, we get

$$p\frac{dV}{d\varphi} + V\frac{dp}{d\varphi} = mR\frac{dT}{d\varphi} + RT\frac{dm}{d\varphi}. \tag{2}$$

Equation (1) finally becomes

$$mc_v\frac{dT}{d\varphi} = \frac{dQ_l}{d\varphi} - p\frac{dV}{d\varphi} + \sum_j \frac{dm_j}{d\varphi} h_j - \sum_i u_i \frac{dm_i}{d\varphi}, \tag{3}$$

bearing in mind that

$$\frac{dU}{d\varphi} = \sum_{i=1}^{4} u_i \frac{dm_i}{d\varphi} + \sum_{i=1}^{4} m_i c_{vi} \frac{dT}{d\varphi}, \tag{3a}$$

where m_i is the mass of species i, c_v is the specific heat capacity under constant volume (a function of temperature only, $c_v = du/dT$), and m is the mass of the whole mixture. For the evaluation of the dm_j exchanged gas terms, the well-known correlation for gas mass flow through a restriction is used, accounting for one-dimensional, quasi-steady, compressible flow (see [1, 3, 4]):

$$\frac{dm_j}{dt} = c_d F \frac{p_u}{R_u T_u} \sqrt{\frac{2\gamma R_u T_u}{\gamma - 1}\left[\left(\frac{p_d}{p_u}\right)^{2/\gamma} - \left(\frac{p_d}{p_u}\right)^{(\gamma+1)/\gamma}\right]}, \tag{4a}$$

for $p_d/p_u \geqslant [2/(\gamma + 1)]^{\gamma/(\gamma - 1)}$, while for the opposite case we have

$$\frac{dm_j}{dt} = c_d F \frac{p_u}{R_u T_u} \sqrt{\gamma R_u T_u \left[\frac{2}{\gamma + 1}\right]^{(\gamma + 1)/(\gamma - 1)}}. \tag{4b}$$

In equations (4a) and (4b), c_d is the discharge coefficient, a function of the instantaneous valve lift to valve head diameter ratio [3] for mass exchange between cylinder and manifolds, while F is the geometric area through which mass is exchanged. Index d denotes downstream conditions and u upstream conditions in equations (4).

The fuel injection term is described in the next subsection through an analytical expression. During the injection period, the term dQ_l is increased due to the contribution of the amount of heat drawn by the injected fuel for its evaporation.

The manifolds are treated as plenums with time varying pressure, temperature and composition. Perfect and instantaneous mixing is assumed during the mass exchange period, and reverse flow into the inlet manifold is also taken into account. Equation (3) is valid for this case too, noting that $dV/d\varphi = 0$ and that no fuel injection term exists.

The instantaneous values for the cylinder volume and its rate of change against crank angle, needed in equations (2) and (3), are (see [1, 4]) given by

$$V = V_{cl} + \frac{\pi D^2}{4}\left[r(1 + \cos\varphi) + \lambda(1 - \sqrt{1 - \lambda^2 \sin^2\varphi})\right], \tag{5}$$

$$\frac{dV}{d\varphi} = \frac{\pi D^2}{4}\left[-r\sin\varphi(1 - \lambda\cos\varphi)/\sqrt{1 - \lambda^2\sin^2\varphi}\right], \tag{6}$$

where the term V_{cl} accounts for the cylinder clearance volume, and $\lambda = r/l$, with l the connecting rod length.

It should also be noted that $d\varphi = dt/6N$, where N is the engine speed in rpm, for the transformation of the various terms from time to °CA basis.

Fuel injection rate

For simulation of the fuel injection rate (kg/s), since no analytical model for the fuel pump was available, the following expression proposed by Ferguson [4] is used:

$$\frac{\dot{m}_{fi}}{M_{tot}} = \frac{\omega}{\varphi_d \Gamma(n)}\left(\frac{\varphi - \varphi_s}{\varphi_d}\right)^{n-1} \exp\left(\frac{(\varphi_s - \varphi)}{\varphi_d}\right), \tag{7}$$

where $\ln\Gamma(n) = (n - 0.5)\ln(n) - n + 0.5\ln(2\pi) + 1/12n - 1/360n^3 + 1/1260n^5$, with $3 < n < 5$ for a divided combustion chamber engine such as the one under study. For the present analysis, $n = 3.6$. 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. Equation (7), though not explicitly accurate, is a considerable development against the steady rate of injection relations used in most simulation models.

Combustion model

For study of the combustion process, the model proposed by Whitehouse and co-workers [1, 2, 6–8] is used for both the main chamber and the prechamber. This semi-empirical model assumes that the injected fuel mixes with the entrained air via a mixing and diffusion process, while the chemical reaction is effected using an Arrhenius type equation. Thus, the combustion process consists of two parts: a preparation limited combustion rate and a reaction limited combustion rate. The corresponding equations are

$$P = KM_i^{1-x}M_u^x p_{O_2}^y \quad (\text{kg/°CA}) \tag{8}$$

for the preparation rate, and

$$R = \frac{K_1 p_{O_2}}{N\sqrt{T}} e^{-\mathrm{act}/T} \int (P - R)\, d\varphi \quad (\mathrm{kg/^\circ CA}) \tag{9}$$

for the reaction rate. Here $M_i = \int_{\varphi_s}^{\varphi} (dm_{fi}/d\varphi)\, d\varphi$ 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 (7). Also, $M_u = M_i - \int P\, d\varphi$ 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. During the early part of the combustion process, the fuel dm_{fb} burned in a time step dt (°CA step $d\varphi$) is controlled by the reaction rate R, while after a short period of time, the fuel burned is controlled by the preparation rate P. According to Ref. [1], the constant K in the preparation rate equation is based upon the Sauter mean diameter (SMD) of the fuel droplets, being expressed by a formula of the type $K \propto (1/\mathrm{SMD})^2$. For the evaluation of SMD, the following empirical expression proposed by Hiroyasu *et al.* [3, 6, 12] is used:

$$\mathrm{SMD} = 25.1(\Delta p)^{-0.135} \rho_g^{0.12} V_{\mathrm{tot}}^{0.131} (\mu m), \tag{10}$$

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. The constants of the combustion model are derived through calibration against experimental data.

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 (see [1, 27–29]):

$$\frac{dQ_l}{dt} = F\left[a \frac{\lambda}{D} \mathrm{Re}^b (T_w - T_g) + c(T_w^4 - T_g^4) \right], \tag{11}$$

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. 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. Thermal conductivity λ and dynamic viscosity μ are calculated from the following formulae (see [3]):

$$\lambda = 3.17 \times 10^{-4} T^{0.772}, \tag{12a}$$

$$\mu = 3.3 \times 10^{-7} T^{0.7}/(1 + 0.027f), \tag{12b}$$

where f is the instantaneous fuel-air equivalence ratio. Blowby phenomena are ignored in the present analysis.

Multi-cylinder engine operation

For better simulation of engine performance, a multi-cylinder engine model has been developed, i.e. one in which all the differential and algebraic equations mentioned above are solved individually for each cylinder. Thus, a good simulation of the exhaust and inlet manifolds is achieved, accounting for small differences between the operations of each cylinder, unlike previous models which solve the governing equations for one cylinder and, thereafter, assume that all other cylinders behave the same way. This detailed simulation of multi-cylinder operation enables a correct evaluation of availability term development in the manifolds and turbocharger during an engine cycle to be made, by accounting for all engine cylinder contributions.

Turbocharger

The simulation of the turbocharger is accomplished by the use of manufacturer's data under steady state operation. For the compressor, a complete map was available, providing turbocharger speed and compressor isentropic efficiency for each pair of compressor pressure ratio and mass flow rate values. For the turbine, a single curve was available irrespective of speed, providing the correlation between mass flow rate through the turbine and pressure expansion ratio. This curve was approximated by an orifice equation (equations (4a) and (4b)) with a surface area of $F = 2610\ \mathrm{mm}^2$ (as given by the manufacturer) and a typical discharge coefficient $c_d = 0.70$ [30].

Aftercooler

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 [30]:

$$T_3 = T_2(1 - \epsilon) + \epsilon T_{cwi}, \tag{13}$$

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

$$\epsilon = 1 - c_1 \dot{m}_c^2. \tag{13a}$$

For the pressure drop across the aftercooler:

$$p_2 - p_3 = c_2 \dot{m}_c^2. \tag{13b}$$

In equations (13) and (13b), 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 following formula of Chen and Flynn [32] for turbocharged engines is used:

$$\mathrm{fmep} = 0.137 + 0.005 p_{max} + 0.162 \bar{c}, \tag{14}$$

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

Solving the equations

The differential equations stated above were solved simultaneously via the predictor–corrector method and with a user-defined step of calculation which, for the specific investigation, was 0.25°CA for the burning and expansion periods and 0.5°CA for the compression and mass exchange

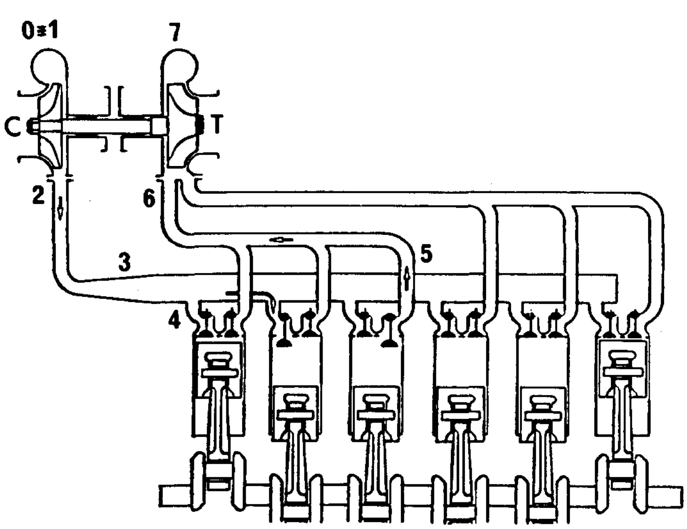


Fig. 1. General description of the engine–manifolds–turbocharger arrangement, for the MWM TbRHS 518S engine, showing strategic point locations used in the analysis.

periods. The computer program was written in FORTRAN 77 language and was executed on an IBM compatible PC.

SECOND LAW (AVAILABILITY) ANALYSIS

Basic concepts and definitions

The 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. For a closed system (experiencing heat and work interactions with the environment, where no heat interactions are permitted across the control surface of the combined system and the total volume of the control system and environment remains unaltered), the following equation holds (see [18, 33, 34]):

$$W_{max} = A = (U - U_o) + p_o(V - V_o) - T_o(S - S_o). \tag{15a}$$

Availability is an extensive property with a value greater than or equal to zero. It is obvious that availability is a property, the value of which depends not only on the state of the system but also on the ambient properties. As stated above, there is no availability in a system when thermal, mechanical and chemical equilibrium exists with the environment. It is then said that the system is at the dead state.

Thermal equilibrium is achieved when no heat transfer is possible through the boundaries of the system. This happens when the temperature of the system is equal to the temperature of the surrounding environment. In the same way, mechanical equilibrium is achieved when no work can be exchanged with the surrounding system.

As regards chemical equilibrium, there are various versions in the relevant literature for the exact definition of the term. Such an equilibrium is achieved only when there are no components of the working medium which could react with those of the environment to produce work. All the components of the working medium must be either oxidized (e.g. $C_{12}H_{26}$, CO and H) or reduced (e.g. NO and OH) in a reversible way as the system reaches the dead state. The only components of the system which cannot react chemically with the atmosphere and, therefore, constitute the components of the mixture at the dead state are O_2, N_2, CO_2 and H_2O. In addition to the work that could be obtained due to these reversible reactions, some researchers propose that, for a better definition of chemical equilibrium, one should also take into account the capacity to produce work because of the difference between the partial pressures of the components, when in thermal and mechanical equilibrium with the environment, and the partial pressures of the same components in the atmosphere [15, 18]. This work could be extracted by the use of semi-permeable membranes and devices such as the Van't Hoff's equilibrium box. Other researchers state that the extraction of work by the use of such devices is, in most cases, practically impossible and so should not be taken into account [16, 22–24]. This last point of view is followed by the present researchers, and so 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. For the purpose of this work, the atmosphere is considered to have a temperature of $T_o = 298.15$ K and a pressure of $p_o = 1.013$ bar, which closely represent normal laboratory conditions [16].

For the case of a closed system, the equation (15a) given above also suggests the following (see [18, 33]):

$$A = U + p_o V - T_o S - G_o, \tag{15b}$$

where G_o is the working medium's Gibbs free enthalpy at ambient pressure and temperature, which is calculated for the dead state composition. For the processes encountering reaction with the fuel, the dead state composition is defined as that of a perfect burning one. For other processes, such as expansion through a turbine, dead state composition equals the working medium's composition at each time step since, for the present analysis, only the four basic species also found in the environment were taken into consideration.

General availability balance equation

For an open system experiencing mass exchange with the surrounding environment, Moran [18] suggests the following equation for the availability on a time basis:

$$\frac{\mathrm{d}A_{cv}}{\mathrm{d}t} = \int_F \left(1 - \frac{T_o}{T}\right) q \,\mathrm{d}F - \left(\dot{W}_{cv} - p_o \frac{\mathrm{d}V_{cv}}{\mathrm{d}t}\right) + \sum_i \dot{m}_i b_i - \sum_e \dot{m}_e b_e - \dot{I}. \tag{16}$$

The terms here have the following meanings:

(a) $\mathrm{d}A_{cv}/\mathrm{d}t$ is the rate of change of control volume (i.e. cylinder, each manifold, etc) availability.
(b) $\int_F(1 - (T_o/T))q \,\mathrm{d}F$ is the 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 under investigation, and q is the heat flux to or from the working medium, exchanged through differential surface area $\mathrm{d}F$.
(c) $(\dot{W}_{cv} - p_o(\mathrm{d}V_{cv}/\mathrm{d}t))$ is the availability term associated with work transfer.
(d) $\Sigma_i \dot{m}_i b_i - \Sigma_e \dot{m}_e b_e$ are availability terms associated with inflow and outflow of masses, respectively.
(e) $\dot{I}$ is the 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 (d) above, the expression

$$b = h - h_o - T_o(s - s_o) \tag{17}$$

is defined as the flow availability [18].

In the following section, equation (16) is applied explicitly to each control volume of the diesel engine plant on a °CA rather than on a time basis.

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

Cylinder

Equation (16) in this case becomes

$$\frac{\mathrm{d}A_{cyl}}{\mathrm{d}\varphi} = \frac{\dot{m}_4 b_4 - \dot{m}_5 b_5}{6N} - \frac{\mathrm{d}A_w}{\mathrm{d}\varphi} + \frac{\mathrm{d}A_l}{\mathrm{d}\varphi} + \frac{\mathrm{d}A_f}{\mathrm{d}\varphi} - \frac{\mathrm{d}I}{\mathrm{d}\varphi}. \tag{18}$$

The terms on the right-hand side of this equation have the following meanings:

$$\frac{\mathrm{d}A_w}{\mathrm{d}\varphi} = (p - p_o)\frac{\mathrm{d}V}{\mathrm{d}\varphi} \tag{19}$$

is the term for work transfer, with the derivative $\mathrm{d}V/\mathrm{d}\varphi$ provided by equation (6);

$$\frac{\mathrm{d}A_l}{\mathrm{d}\varphi} = \frac{\mathrm{d}Q_l}{\mathrm{d}\varphi}(1 - T_o/T) \tag{20}$$

is the term for heat transfer to the cylinder walls with $\mathrm{d}Q_l/\mathrm{d}\varphi$ provided by Annand's correlation (see equation (11)), with T the instantaneous cylinder temperature; and

$$\frac{\mathrm{d}A_f}{\mathrm{d}\varphi} = \frac{\mathrm{d}m_{fb}}{\mathrm{d}\varphi} a_{fch} \tag{21}$$

is the flow availability associated with the burning of fuel, which for hydrocarbon liquid fuels of the general type C_mH_n is given (see [18]) by

$$a_{fch} = \mathrm{LHV}\left(1.04224 + 0.011925\frac{n}{m} - \frac{0.042}{m}\right). \tag{22}$$

For the present analysis,

$$m = 12, \quad n = 26 \text{ and } a_{\text{fch}} = 1.064\,\text{LHV}. \tag{23}$$

The fuel burning rate $\mathrm{d}m_{\text{fb}}/\mathrm{d}\varphi$ is established using the Whitehouse–Way model described in a previous section (equations (8) and (9)).

The term on the left-hand side of equation (18)

$$\frac{\mathrm{d}A_{\text{cyl}}}{\mathrm{d}\varphi} = \frac{\mathrm{d}U}{\mathrm{d}\varphi} + p_{\text{o}}\frac{\mathrm{d}V}{\mathrm{d}\varphi} - T_{\text{o}}\frac{\mathrm{d}S}{\mathrm{d}\varphi} - \frac{\mathrm{d}G_{\text{o}}}{\mathrm{d}\varphi} \tag{24}$$

is the change in the availability of the cylinder contents (see [15]) and its various terms can be described as follows. The $\mathrm{d}U/\mathrm{d}\varphi$ term, according to equation (3a), is the differential change in internal energy of the cylinder contents. The rate of entropy change of the cylinder contents term is

$$\frac{\mathrm{d}S}{\mathrm{d}\varphi} = \sum_{i=1}^{4} \frac{\mathrm{d}m_i}{\mathrm{d}\varphi} s_i(T, x_i p) + \sum_{i=1}^{4} \frac{m_i}{T} c_{pi} \frac{\mathrm{d}T}{\mathrm{d}\varphi} - \frac{V}{T}\frac{\mathrm{d}p}{\mathrm{d}\varphi}. \tag{25}$$

Here

$$s_i(T, x_i p) = s_i'(T, p_{\text{o}}) - R \ln\left(\frac{x_i p}{p_{\text{o}}}\right). \tag{26}$$

The term $s_i'(T, p_{\text{o}})$ in equation (26) is a function of temperature only, with x_i the molar fraction of species i in the mixture.

The last term in equation (24) is

$$\frac{\mathrm{d}G_{\text{o}}}{\mathrm{d}\varphi} = \sum_{i=1}^{4} \frac{\mathrm{d}m_i}{\mathrm{d}\varphi} \mu_i^{\circ}, \tag{27}$$

where $\mu_i^{\circ} = g_i(T_{\text{o}}, x_i p_{\text{o}})$ is the chemical potential of species i at ambient conditions and

$$g_i(T_{\text{o}}, x_i p_{\text{o}}) = h_i(T_{\text{o}}) - T_{\text{o}} s_i(T_{\text{o}}, x_i p_{\text{o}}) = h_i(T_{\text{o}}) - T_{\text{o}}[s_i'(T_{\text{o}}, p_{\text{o}}) - R \ln(x_i)]. \tag{28}$$

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 the first law analysis of compressible flow through a restriction (see equations (4a) and (4b)) and $b_4 = h_4 - h_{\text{o}} - T_{\text{o}}(s_4 - s_{\text{o}})$, 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. For the valve overlap period, both terms are valid simultaneously.

$\dot{I}$ is the rate of irreversibility production inside the cylinder which, for the intake process, comprises throttling at the inlet valve and mixing of the incoming air with the cylinder residuals. For the closed part of the cycle and after the fuel injection has started, the irreversibility term accounts for the availability destruction due to combustion. Exhaust valve throttling is taken into account with the exhaust manifold as will be shown in a next subsection.

Turbocharger

For the compressor and turbine, no control volume exists. Also, heat losses are usually negligible so they are not taken into account. Equation (16) for the compressor becomes

$$\dot{m}_{\text{C}}(b_2 - b_1) + \frac{\mathrm{d}I_{\text{C}}}{\mathrm{d}\varphi} 6N = -\dot{W}_{\text{C}}, \tag{29}$$

while for the turbine it becomes

$$\dot{m}_{\text{T}}(b_6 - b_7) - \frac{\mathrm{d}I_{\text{T}}}{\mathrm{d}\varphi} 6N = \dot{W}_{\text{T}}. \tag{30}$$

In these equations, the terms $\dot{W}_{\text{C}}$ and $\dot{W}_{\text{T}}$ are evaluated from the thermodynamic analysis of the turbocharger at each crank angle step via instantaneous values from the turbomachine steady-state maps. Subscripts 1 and 2 denote compressor inlet and outlet conditions, respectively, while subscripts 6 and 7 denote turbine inlet and outlet conditions, respectively, as also shown in Fig. 1.

Inlet manifold

General equation (16) states that, for the inlet manifold,

$$\frac{dA_{im}}{d\varphi} = \frac{\dot{m}_3 b_3 - \dot{m}_4 b_4}{6N} - \frac{dI_{im}}{d\varphi}, \tag{31}$$

where b_3 is the flow availability (see equation (17)) at the intake manifold inlet, evaluated at aftercooler outlet conditions, provided by equations (13), (13a) and (13b). No heat losses are taken into account. The term stated above for irreversibilities $dI_{im}/d\varphi$ accounts for the mixing of aftercooler air with intake manifold residual contents; as will be shown later, this is of minor importance.

Exhaust manifold

General equation (16) states that, for the exhaust manifold,

$$\frac{dA_{em}}{d\varphi} = \frac{\dot{m}_5 b_5 - \dot{m}_6 b_6}{6N} - \frac{dI_{em}}{d\varphi} + \frac{dA_{lem}}{d\varphi}, \tag{32}$$

where index 5 denotes exit conditions from the engine cylinder (see Fig. 1). The term

$$\frac{dA_{lem}}{d\varphi} = \frac{dQ_{lem}}{d\varphi}(1 - T_o/T_{em}) \tag{33}$$

accounts for heat losses at the exhaust manifold, where T_{em} is the instantaneous temperature of the manifold contents, calculated from an expression similar to that of equation (3) applied for the manifold. 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. [16], since the magnitude of each of these three irreversibility terms is rather small, it is not considered worthwhile to calculate each separate term; thus, throttling losses are not separated from mixing ones and those from friction ones.

For both inlet and exhaust manifolds control volume availability terms $dA_{im}/d\varphi$ and $dA_{em}/d\varphi$, respectively, similar expressions exist as for the cylinder term (see equation (24)), with the exception of the $p_o\, dV/d\varphi$ term which is zero.

For the evaluation of the inlet and exhaust manifold irreversibility rate terms, mass flow rates $\dot{m}_3$, $\dot{m}_4$, $\dot{m}_5$, $\dot{m}_6$ are taken into account from all cylinders which, at the specific time step, exchange mass with either manifold, with the help of the multi-cylinder thermodynamic model previously described.

Aftercooler

For the aftercooler, the general balance equation (16) states that

$$\dot{m}_c(b_2 - b_3) - \dot{m}_{cw}(b^w_{out} - b^w_{in}) = \frac{dI_{ac}}{d\varphi} 6N, \tag{34}$$

where b^w_{in} and b^w_{out} are the availability terms for the cooling water at aftercooler inlet and outlet positions defined by the following equation, valid for incompressible fluids (see [18]):

$$b^w = \bar{c}_{pcw}[T - T_o - T_o \ln(T/T_o)], \tag{35}$$

where $\bar{c}_{pcw}$ is the mean specific heat over the temperature range T_o to T. The term $dI_{ac}/d\varphi$ accounts for the availability destruction due to heat transfer to a cooler medium (cooling water) and can be quite large according to the temperature level of the medium to be cooled [18]. It should be noted at this point that the irreversibility term stated above is not, at first, taken into account in the whole diesel plant analysis, since availability terms associated with heat loss are considered as recoverable from the second law analysis point of view.

Cumulative availability terms

It is obvious that the cumulative terms associated with the change of availability of the control volume for the cylinder (equation (24)), inlet manifold and exhaust manifold sum to zero in a full cycle of the working medium, since initial and final conditions are the same for steady state operation.

The equations (18), (29), (30), (31), (32) and (34) for the availability balance can be solved for the rate of irreversibility term $\dot{I}$, which is the only unknown term, since pressure, temperature and composition for each control volume, at each crank angle step, are known from the thermodynamic first law analysis of the cycle described in a previous section.

EVALUATION OF SECOND LAW EFFICIENCIES

Cylinder

For the cylinder alone, the following second law efficiency is often defined (for a four-stroke engine):

$$\epsilon = \frac{\int_0^{720} (p - p_o)(\mathrm{d}V/\mathrm{d}\varphi)\,\mathrm{d}\varphi}{a_{fch}\int_0^{720} (\mathrm{d}m_{fb}/\mathrm{d}\varphi)\,\mathrm{d}\varphi = M_{tot}a_{fch}}, \tag{36}$$

which can be easily compared with the first law efficiency:

$$\eta_1 = \frac{\int_0^{720} p(\mathrm{d}V/\mathrm{d}\varphi)\,\mathrm{d}\varphi}{M_{tot}\mathrm{LHV}}. \tag{37}$$

Bearing in mind that $\int_0^{720} p_o\,\mathrm{d}V = 0$ for the (whole) cycle, equations (36) and (37) give:

$$\epsilon_1 = \eta_1 \frac{\mathrm{LHV}}{a_{fch}}, \tag{37a}$$

which, according to equation (23), becomes $\epsilon_1 = 0.94\eta_1$.

One could also take into account the differences between outgoing and incoming availability flows for the ability to produce work, and define the following second law efficiency (see [18, 24]) as follows:

$$\epsilon_2 = \frac{\int_0^{720} (p - p_o)(\mathrm{d}V/\mathrm{d}\varphi)\,\mathrm{d}\varphi + \int_0^{720} (\dot{m}_5 b_5 - \dot{m}_4 b_4)\,\mathrm{d}\varphi/6N}{M_{tot}a_{fch}}. \tag{38}$$

According to Refs [18, 19, 24], since the process through which the fuel chemical availability is converted into exhaust gas thermomechanical availability does contain heat transfer, instead of the term

$$A_1 = a_{fch}\int_0^{720} (\mathrm{d}m_{fb}/\mathrm{d}\varphi)\,\mathrm{d}\varphi = M_{tot}a_{fch}, \tag{38a}$$

one should rather use the term

$$A_2 = a_{fch}\int_0^{720} (\mathrm{d}m_{fb}/\mathrm{d}\varphi)(1 - T_o/T)\,\mathrm{d}\varphi. \tag{38b}$$

Term A_1 expresses the heat released from the burning of fuel, while term A_2 gives the heat obtained from the exhaust gases, which could be transformed into useful work. Thus, one can also define the following second law efficiency:

$$\epsilon_3 = \frac{\int_0^{720} (p - p_o)(\mathrm{d}V/\mathrm{d}\varphi)\mathrm{d}\varphi + \int_0^{720} (\dot{m}_5 b_5 - \dot{m}_4 b_4)\,\mathrm{d}\varphi/6N}{a_{\mathrm{fch}} \int_0^{720} (\mathrm{d}m_{\mathrm{fb}}/\mathrm{d}\varphi)(1 - T_o/T)\,\mathrm{d}\varphi}. \quad (39)$$

Equation (39) takes into account the fact that, for a thermal engine, the highest possible efficiency is that defined by the ideal Carnot cycle working between the same temperature levels. Thus, equation (39) reduces the useful work not on the whole fuel flow but on that amount of energy which can really produce work, since according to the second law statement, it is inevitable that a certain amount of heat will be thrown away.

Turbocharger

For the compressor, as stated, for example, in Refs [18, 33, 34],

$$\epsilon_{\mathrm{C}} = \frac{\dot{m}_{\mathrm{C}}(b_2 - b_1)}{|\dot{W}_C|} \quad (40)$$

is the second law efficiency. Similarly for the turbine, the second law efficiency is

$$\epsilon_{\mathrm{T}} = \frac{\dot{W}_{\mathrm{T}}}{\dot{m}_{\mathrm{T}}(b_6 - b_7)}. \quad (41)$$

Whole diesel engine plant

For the whole diesel plant,

$$\epsilon_{\mathrm{tot1}} = \frac{\int_0^{720} (p - p_o)(\mathrm{d}V/\mathrm{d}\varphi)\,\mathrm{d}\varphi}{A_1 \text{ or } A_2}, \quad (42)$$

or, alternatively,

$$\epsilon_{\mathrm{tot2}} = \frac{\int_0^{720} (p - p_o)(\mathrm{d}V/\mathrm{d}\varphi)\,\mathrm{d}\varphi + \int_0^{720} (\mathrm{d}m_7/\mathrm{d}\varphi) b_7\,\mathrm{d}\varphi}{A_1 \text{ or } A_2}, \quad (43)$$

where terms A_1 and A_2 are given by equations (38a) and (38b).

One could also add the increase in the availability of the cooling water (over one cycle),

$$\Delta A_{\mathrm{cw}} = \dot{m}_{\mathrm{cw}}(b^{\mathrm{w}}_{\mathrm{out}} - b^{\mathrm{w}}_{\mathrm{in}})720/6N \quad (44)$$

and so obtain

$$\epsilon_{\mathrm{tot3}} = \frac{\int_0^{720} (p - p_o)(\mathrm{d}V/\mathrm{d}\varphi)\,\mathrm{d}\varphi + \int_0^{720} (\mathrm{d}m_7/\mathrm{d}\varphi) b_7\,\mathrm{d}\varphi + \Delta A_{\mathrm{cw}}}{A_1 \text{ or } A_2}. \quad (45)$$

It is doubtful, however, if the terms added to the work term in the numerator of equations (43) and (45) are really worth exploiting, since, as will be shown in a following section, their magnitude (especially for the turbocharged case) compared with the work term is rather small. Whether the gain from these terms is worth the added plant complexity is a matter of thermo-economical study [33].

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

Type	In-line 6-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 BTDC
Intake valve closure	60°CA ABDC
Exhaust valve opening	64°CA BBDC
Exhaust valve closure	47°CA ATDC
Maximum power	320 HP (235 kW) @ 1500 rpm
Maximum torque	1500 Nm @ 1250 rpm

EXPERIMENTAL FACILITIES AND PROCEDURE

Description of the engine

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 was fitted with a Kühnle-Kopp-Kausch (KKK) turbocharger, model 4B 754/345, and a water aftercooler after the turbocharger compressor. The engine was coupled to a Schenck hydraulic dynamometer for measuring engine performance. As stated above the engine was of the prechamber type. A throat nozzle, with two holes of 5 mm diameter each, connected the cylinder prechamber with the main chamber.

The pintle type injector nozzle opening pressure was 120 bar, and the static injection timing was fixed at 15°CA BTDC within the normal engine speed operating range of 1000 to 1500 rpm (corresponding mean piston speeds of 6–9 m/s). The special characteristic of this engine is the almost constant full load torque, irrespective of speed [31].

Description of the test installation and experimental procedure

The main parts of the test installation used were:

MWM TbRHS 518S diesel engine,

Schenck U1–40 hydraulic dynamometer,

Steady state air-flow measurement apparatus consisting of a receiving chamber, two flow nozzles, a sufficiently large discharging chamber with diffusion baffle, and a Göttingen differential micromanometer of inclined alcohol column reading scale, located upstream of the turbocharger compressor,

Tank and flowmeter for diesel fuel (cetane index 48) consumption rate measurement,

Turbocharger compressor boost pressure manometer,

Turbocharger compressor inlet and exit air thermometers,

Aftercooler exit air thermometer,

Magnetic pick-up TDC marker and rpm indicator,

Kistler 7613A piezotron pressure transducer with a voltage amplifier mounted on the prechamber wall of one engine cylinder,

Kistler 7013 A1 piezoelectric pressure transducer with a charge amplifier, mounted on the cylinder head (main chamber) of the cylinder mentioned above,

IBM compatible 486 PC computer, properly interfaced for fast pressure data acquisition and recording [35, 36],

Tektronix storage oscilloscope with Polaroid camera,

Metrabyte DAS–16F high speed data acquisition card mounted on the PC bus.

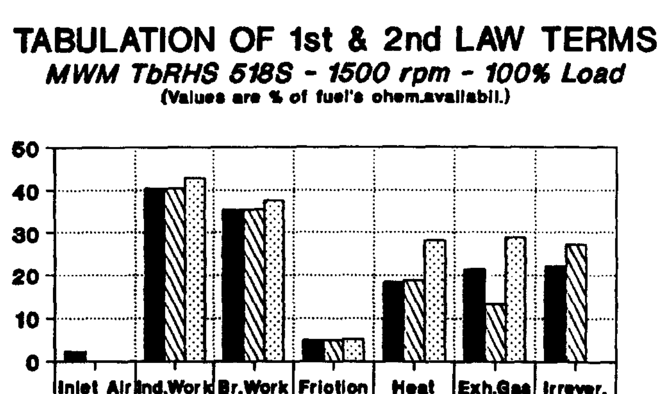
A detailed experimental investigation of the engine processes was performed for all combinations of three values of the engine load, corresponding to 50%, 75% and 100% of the engine torque at 1000 rpm, and three engine speeds of 1000, 1250 and 1500 rpm (9 runs). The experimental results obtained were used to calibrate the thermodynamic model from the first law analysis point of view. 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 (16), in its various forms for the cylinder, compressor, turbine, aftercooler and manifolds, is applied for the maximum speed-full load operation of the engine under study, so that the development of the rate and cumulative availability terms are presented and analysed in Figs 4–13. Figures 2 and 3 and Table 2 show arithmetic values of the availability terms and point out the differences from first law analysis as well as showing the quantification of various device irreversibilities.

Figure 2 provides the tabulation of energy and exergy terms for the specific operating point. It is noted that the availability terms for manifolds and turbocharger were averaged on the basis of one cylinder, so that direct comparison with in-cylinder availability terms is possible.

As can be seen from Fig. 2, values for the indicated, 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 other hand, heat losses to the cylinder walls, aftercooler and exhaust manifold walls present quite a difference between first and second law assessments, namely 28.30 versus 18.95%, as also pointed out by other researchers in the field. The term $(1 - T_o/T)$ which transforms the amount of heat rejected to possible recoverable work is responsible for this difference. For the engine being considered, this percentage lies between 65 and 70%, which is a typical value for engines with no additional measures for wall insulation. It is obvious that the higher the insulation of the cylinder walls, the greater the temperatures T and the smaller the difference between first and second law heat loss terms. Whether the provision for insulation would also account for further exploitation of exhaust gases in a bottoming cycle or a power turbine is a matter for study [22]. It is pointed out that, due to the aftercooler, the availability entering the cylinder (2.31% of fuel's availability) is rather small. Consequently, the



	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

Cylinder 2nd Law Diesel 2nd Law Diesel 1st Law

Fig. 2. Tabulation of first and second law terms for the cylinder and the whole diesel engine plant.

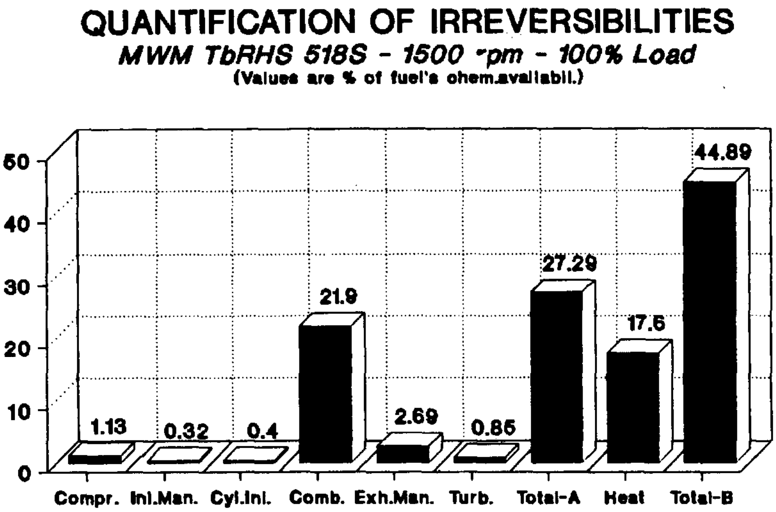


Fig. 3. Quantification of various devices and process irreversibilities.

temperatures of the cylinder contents are lower than they would be with no intermediate charge air cooling, resulting in a lower $(1 - T_o/T)$ term.

Typical values are also observed when comparing the exhaust gas terms, either that which leaves the cylinder (21.38% of the fuel's availability) or that leaving the whole plant (13.45% as compared to 28.81% of the fuel's lower heating value from the first law analysis). The difference between these terms (7.93%) has been 'consumed', either in the exhaust manifold in the form of irreversibilities and heat losses or in the turbine expansion for driving the compressor and for turbine irreversibilities. From this point of view (compressor driving), the exhaust gases have been utilized for further production of work, thus going along with the concept of second law analysis and availability. As a result, the amount of availability leaving the whole plant in the exhaust gases contains a rather small amount of available energy which, nonetheless, should not be ignored.

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%. First law analysis has certainly nothing to report about these losses which heavily reduce the capability to produce work. 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 term, which comprises as much as 10% of the total irreversibilities. Availability destruction in the inlet manifold due to the mixing of aftercooler air with manifold contents accounts for 0.32%, while inlet valve throttling and mixing with cylinder contents accounts for 0.40% of the fuel's availability. The relatively lower isentropic efficiency of the compressor is responsible for its lower second law efficiency (as will be shown below), and this leads to a higher amount of irreversibilities than the turbine (1.13 to 0.85%). Also, Fig. 3 gives the amount of irreversibilities due to heat transfer to the cooling water. This term, of the order of 17.6%, clearly shows how destructive the heat transfer from hot gases to cooler mediums can be; this fact imposes the need for exploitation of this great amount of available energy in the form of heat, which is otherwise lost. It is noted here that the amount of combustion irreversibilities reported in Fig. 3 incorporates also the availability destruction due to throttling between the main chamber and the prechamber, which, according to Ref. [23] is not greater than 0.25% of the fuel's chemical availability. The term 'Total B' in Fig. 3 corresponds to the sum of diesel plant irreversibilities ('Total A' term) plus irreversibilities due to heat transfer to the cooling medium.

Table 2 summarizes the second law efficiencies described earlier (equations (36) to (45)). For the cylinder alone, 40.31% (efficiency ϵ_1) of the fuel's availability is converted into useful work (compared with 42.89% of the fuel's lower heating value), while this amount increases to 59.38% (efficiency ϵ_2) if the difference between outgoing and incoming flows is considered for its ability to produce work. Reducing now the useful work, not over the whole amount of fuel consumed but over that part which really produces work, leads to the third efficiency, $\epsilon_3 = 51.61\%$, if only the useful work is taken into account, or even 76.03%, if the exhaust gas available energy is considered too. The term 78.10 in the third line of Table 2 corresponds to the denominator in equation (39), i.e. it amounts to 78.10% of the fuel's chemical availability, defined also as term A_2 in the second law analysis theory (equation (38b)). The compressor and turbine efficiencies are of the order of 76.38 and 87.42%, respectively. The higher isentropic efficiency of the turbine is responsible for its better second law efficiency. For the whole plant, the three efficiencies defined previously are: $\epsilon_{tot1} = 53.76\%$, $\epsilon_{tot2} = 68.83\%$ and $\epsilon_{tot3} = 70.56\%$. The small difference between the last two efficiencies is due to the low increase of the cooling water's availability (equation (44)) compared with the heat rejected from the cylinders and the aftercooler. Clearly (see Fig. 2), the lower availability of the exhaust gases leaving the whole plant (13.45%), rather than the cylinder alone (21.38 − 2.31 = 19.07%) of the fuel's availability, reflects itself in the fall of the efficiency $\epsilon_3 = 76.03\%$ to $\epsilon_{tot2} = 68.83\%$ or $\epsilon_{tot3} = 70.56\%$.

It should be noted that the choice of ignoring chemical availability terms as regards production of work due to the difference in partial pressures has affected all the results given above, since the availability at a given state for the in-cylinder contents after combustion is lower now than it would have been if the specific chemical availability term had been taken into consideration.

Figures 4 to 13 show the development of the rate and cumulative availability balances during an engine cycle for the maximum speed full load operation condition and for each process encountered. Similar trends are observed for other engine operating conditions. In all these figures, 0°CA corresponds to BDC at the end of the induction process.

Figures 4 and 5 present specifically the development of the in-cylinder processes, while Figs 6 and 7 concentrate on the mass exchange period. Until the start of combustion, the availability of the cylinder contents increases due to the work offered by the piston during the compression process. As the working medium is trapped at a temperature lower than that of the cylinder walls, availability is transferred through heat to the working medium for the early part of the cycle. Then, the heat transfer direction is reversed as the working medium temperature rises. As the availability transfer is low during compression, it is obvious that the change of the working medium availability is almost equal to the work availability transfer, the irreversibility rate being essentially zero. At the time the fuel injection starts, a small fall is observed in the control volume availability rate pattern. This is due to the ignition delay period and the simultaneous loss of heat for evaporation of the injected fuel (Fig. 4). After the start of combustion, things change drastically. The burning of fuel causes a considerable increase in pressure and temperature and, consequently, in cylinder availability and heat loss. The irreversibility rate increases due to combustion. Just after 200–220°CA, when the pressure and temperature begin to fall due to expansion, there is also a fall in the control volume availability. The rate of availability becomes negative near 220°CA. Clearly, the available energy accumulated in the cylinder contents during compression and mainly combustion is being returned with the production of work, which causes the decrease in the availability of the working medium (Fig. 5).

After the opening of the exhaust valve, as can be seen in Fig. 6, the control volume availability rate reaches a second minimum, due to the exhaust gas leaving the cylinder during the blowdown period. The cumulative availability term (Figs 5 and 7) continues to decrease so that, at the end of the cycle (720°CA), all the available energy offered during compression and combustion has been returned.

Table 2. Second law efficiencies for the MWM TbRHS 518S diesel engine plant at 1500 rpm and 100% load

Efficiency	Equation
$\epsilon_1 = 40.31\%$	(36)
$\epsilon_2 = 40.31 + 21.38 - 2.31 = 59.38\%$	(38)
$\epsilon_3 = (59.38/78.10) = 76.03\%$	(39)
$\epsilon_C = 76.38\%$	(40)
$\epsilon_T = 87.42\%$	(41)
$\epsilon_{tot1} = 40.31 + 13.45 = 53.76\%$	(42)
$\epsilon_{tot2} = (40.31 + 13.45)/78.10 = 68.83\%$	(43)
$\epsilon_{tot3} = (40.31 + 13.45 + 1.35)/78.10 = 70.56\%$	(45)

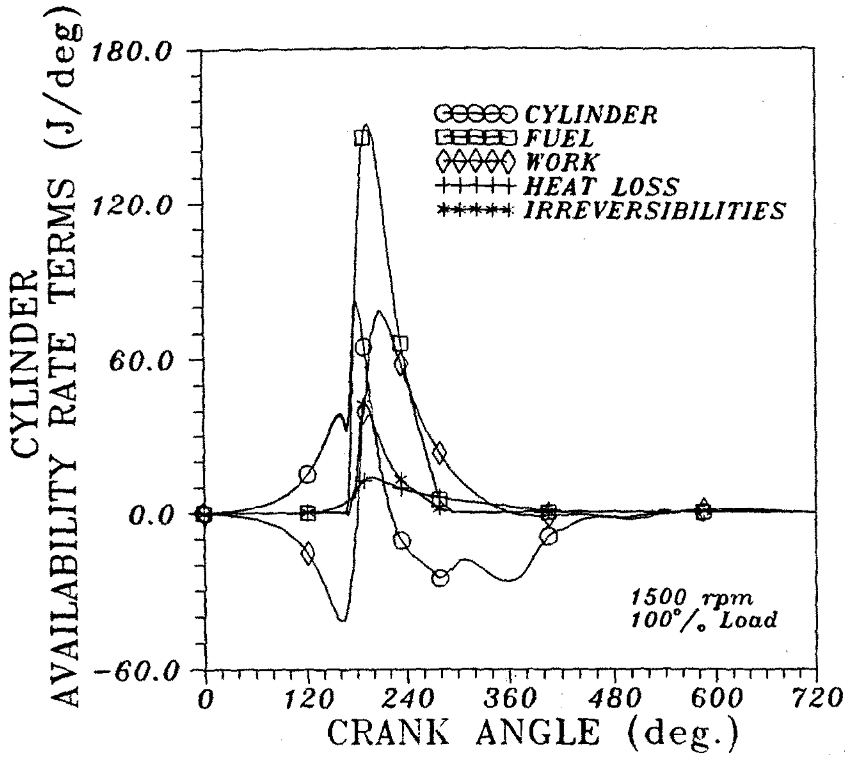


Fig. 4. Development of availability rate terms for the cylinder over the whole cycle.

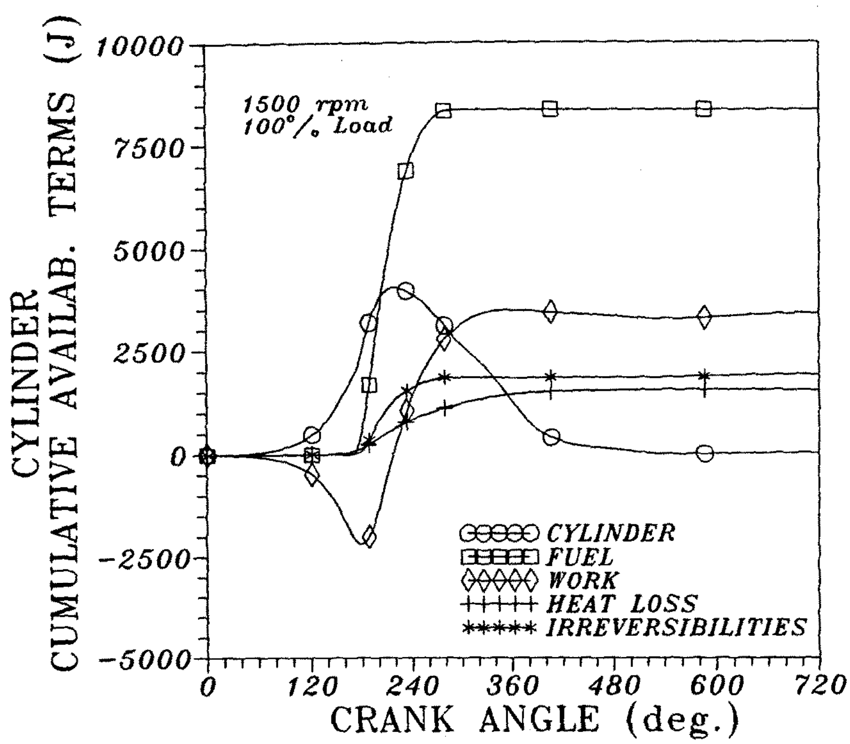


Fig. 5. Development of cumulative availability terms for the cylinder over the whole cycle.

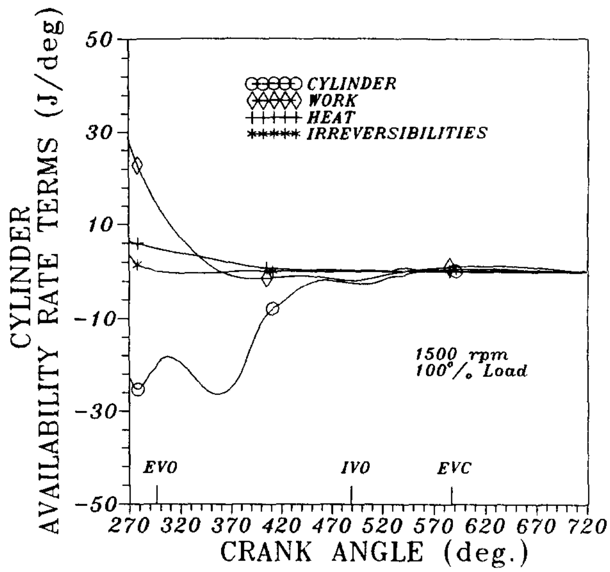


Fig. 6. Development of availability rate terms for the cylinder over the open part of the cycle.

The work and heat transfer terms are also small during the mass exchange period (Figs 6 and 7), while there is a small increase in the cylinder irreversibilities after the opening of the intake valve due to throttling across the valve and mixing of the incoming air with the cylinder residual gases (Fig. 6). During the exhaust process, the irreversibility rate is essentially zero inside the cylinder.

In Fig. 8, the main chamber and the prechamber control volume availability rate terms are shown separately. It is obvious from this figure how the main chamber incorporates the effect of mass exchange with manifolds, while for the prechamber, no important change is observed after the end of combustion. The main chamber presents a greater peak in the availability rate than the prechamber. However, it should not be forgotten that the mass of the main chamber is much greater than that of the prechamber, and this contributes greatly to the, nonetheless, small difference observed.

Figures 9 and 10 present the development of the availability rate and cumulative availability terms for the exhaust manifold, and Fig. 11 presents the cumulative terms for the inlet manifold. The effect of multi-cylinder engine operation is clearly seen, along with the fact that, for a pulse-turbocharging system, such as the one under study, considerable differences between maxima and minima exist for the rate values at the exhaust manifold (Fig. 9) as regards both control volume

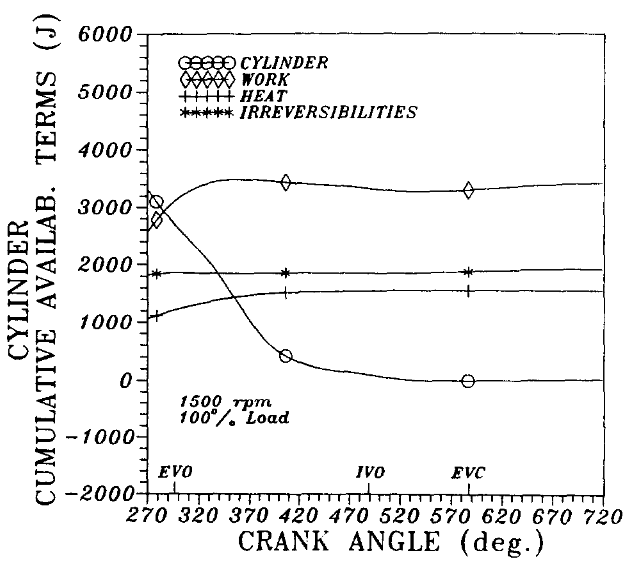


Fig. 7. Development of cumulative availability terms for the cylinder over the open part of the cycle.

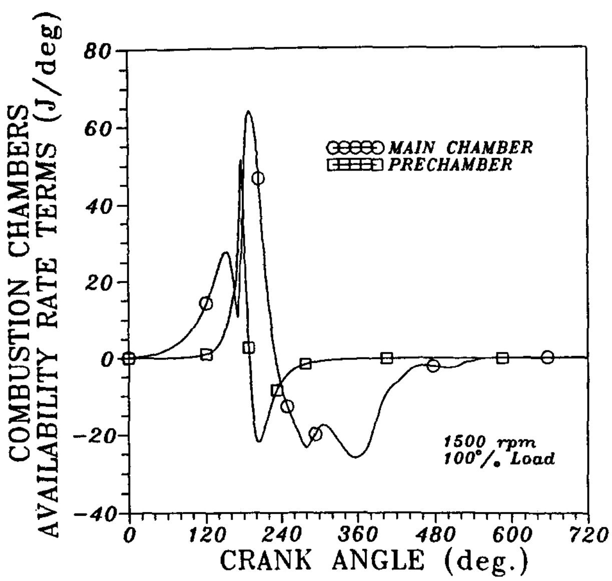


Fig. 8. Development of control volume availability rate terms for the main chamber and the prechamber over the whole cycle.

availability and exhaust gas flows. The major part of the irreversibilities is produced just after the exhaust valve opening (the blowdown period), when gas with high pressure and temperature is expanded across the exhaust valves. As much as 80% of the total irreversibilities for each cylinder exhaust process occurs during the first 60 to 80°CA after EVO.

The pulsating form of the availability terms is obvious also in the cumulative terms presented in Fig. 10 for the exhaust manifold while, for the inlet manifold, no such great pulses are expected. In fact, Fig. 11 shows how little the inlet manifold control volume availability term varies, typical for the small variations in both pressure and temperature which are observed inside the inlet manifold during an engine cycle. The fact that the engine under study has six cylinders contributes to a smooth diagram for the control volume availability of the inlet manifold.

Figure 12 presents the availability rate terms for both the compressor and turbine, where the same pulsating form is obvious, having a great magnitude only for the turbine, as expected. On the other hand, the compressor irreversibilities, due to lower second law efficiency, are higher and

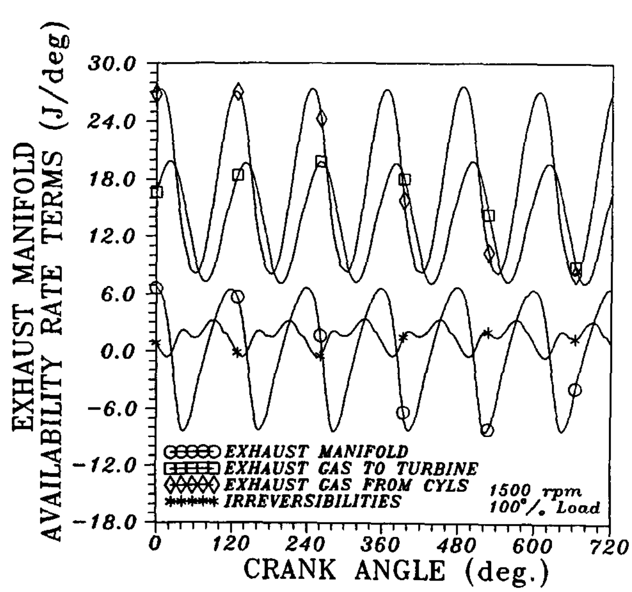


Fig. 9. Development of availability rate terms for the exhaust manifold.

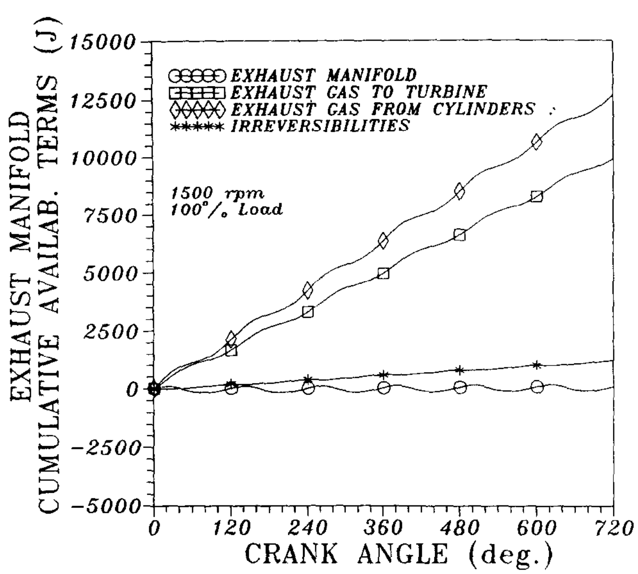


Fig. 10. Development of cumulative availability terms for the exhaust manifold.

this is mainly apparent in Fig. 13, where the corresponding cumulative availability terms for the turbocharger are presented.

CONCLUSIONS

A second law analysis was performed on a six-cylinder, turbocharged and aftercooled diesel engine. For this purpose, a single-zone thermodynamic model was used, accounting for all basic parts of engine operation, and tested favourably against experimental results. Availability equations were applied on every part of the diesel engine plant, and availability rate, as well as cumulative availability, terms for each process and each device were given and discussed. Various kinds of irreversibilities (compressor, inlet, exhaust, combustion and turbine) were identified and quantified. The following important conclusions were derived from the investigation.

Combustion irreversibilities are the main source of availability destruction, but the throttling, friction and thermal mixing losses encountered in the turbocharger and inlet-exhaust manifold

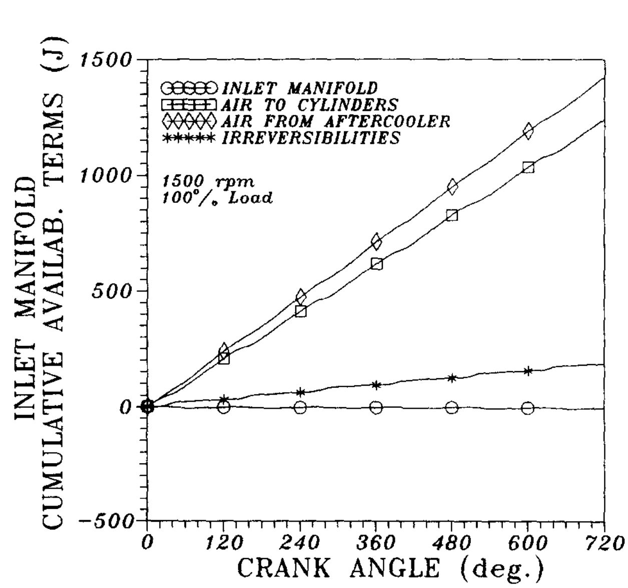


Fig. 11. Development of cumulative availability terms for the inlet manifold.

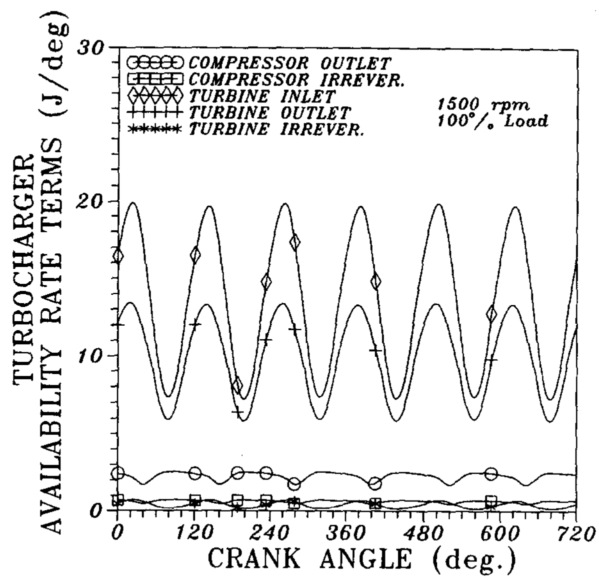


Fig. 12. Development of availability rate terms for the turbocharger.

destructions should not be ignored, since they comprise as much as 19.8% of the total irreversibilities or 13.4% of the useful work for the maximum speed full load operating conditions.

The potential to do work from the exhaust gases is limited due to turbocharging (which, on the contrary, is a very effective way of increasing power, thus agreeing with the concept of availability).

Overall second law efficiency is 40.31% (for work production), with the total irreversibilities comprising as much as 27.30%. This amount, however, increases dramatically to 44.9% if the destruction due to heat transfer to the cooling medium is considered.

Exhaust manifold irreversibilities, mainly due to exhaust valve throttling during the blowdown period, mixing of exhaust gases with manifold contents and friction, contribute as much as 10% of the total irreversibilities, thus showing one process besides combustion which the first law theory fails to describe fully.

Compressor and turbine irreversibilities, on the other hand, as well as inlet irreversibilities, possess low values: all together they amount to 10% of the full load maximum speed irreversibilities. Similar trends and quantifications are also observed for other engine operating conditions.

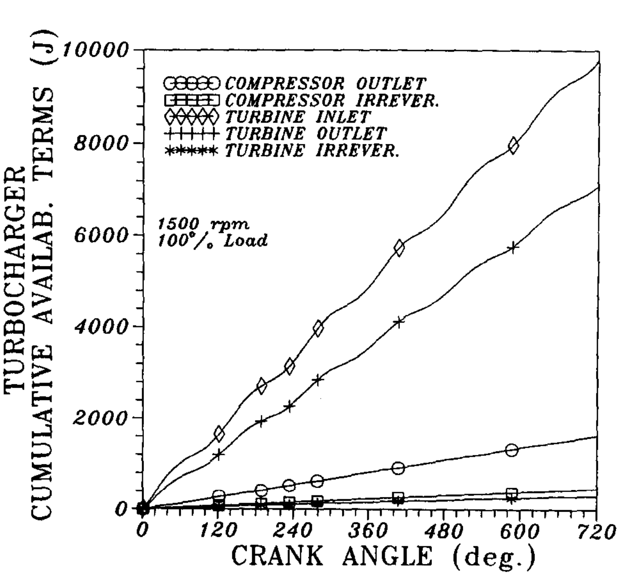


Fig. 13. Development of cumulative availability terms for the turbocharger.

Availability terms for both turbine and exhaust manifold show a great pulsating form, typical for an engine with pulse-turbocharging. On the other hand, inlet manifold terms, as well as compressor terms, do not show any extreme peaks due to the smoothing effect of the multi-cylinder operation.

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