Assessment of Internal Combustion Engine Exergy Based on Theoretical Cycles and Experimental Data

Venetia Sandu, Adrian Mazilu

”Transilvania” University, 29 Eroilor blvd, Brasov, Romania

Abstract **– This work presents an experimental energy and exergy analysis for the operation of a multi-cylinder four-stroke direct injection turbocharged and intercooled commercial diesel engine. A simplified model of an extended thermodynamic cycle over compressor-intercooler-engine and turbine was adapted, and then being calibrated with engine dynamometric bench measurements. For two representative engine operation modes - rated power and maximum torque - at full load, second law terms were assessed in function of chemical diesel fuel availability. The result was an integrated method for a reliable assessment of availability losses, which becomes a useful tool to evaluate the further improvement of process efficiency and of the energy harvesting potential.**

Keywords **– Diesel engine, second-law analysis, irreversibilities.**

1. Introduction

Today, in spite of environmental concerns, the internal combustion engine (ICE) is internationally used in many fields, and it is the dominant power plant in transportation. The energy density of the fuel, low costs, maturity of emission treatments and reasonable efficiency are several gains which keep ICEs as part of the next 15 years [1], at least.

DOI: 10.18421/TEM84-25
https://dx.doi.org/10.18421/TEM84-25

Corresponding author: Venetia Sandu,
”Transilvania” University, Brasov, Romania
Email: *venetia.sandu@unitbv.ro*

Received: 25 August 2019.
Revised: 25 October 2019.
Accepted: 31 October 2019.
Published: 30 November 2019.

The efforts of improving ICE thermal efficiency are related to classical thermodynamics, the first law shows a quantitative energy approach, while the second law is focused on the quality of energy to produce work, known as available energy or exergy.

Energy availability of a system is defined, according to [2], as the „maximum reversible work which can be produced through the interaction of the system with its surroundings as it experiences thermal, mechanical and chemical equilibrium with its environment”. Since the real thermodynamic processes in ICE are irreversible, availability is destroyed in a certain extent during combustion, heat transfer at finite temperature difference, mechanical work at finite pressure difference, friction and throttling [3]. Application of second law analyses to the ICE allows researchers to find the paths to mitigate exergy destruction, to assess losses of availability with design and engine variables and to develop techniques for further improving efficiency of processes and devices for energy harvesting.

Among ICEs, turbocharged intercooled Diesel engines exhibit higher thermal efficiencies and durability, in high power range applications, and it is chosen as object of study in the present paper.

Previous research works on second law analysis of turbocharged intercooled Diesel engines included the simulations on standard thermodynamic cycles, finding the relation between equivalence ratio and availability destruction [4],[5] or assessing the advantages of turbocharging, charge air intercooling, turbo-compounding or cylinder insulation [6]. Alkidas [7] calculated the availability destruction based on test measurements on brake power and heat released in coolant and lubricant, finding combustion and heat transfer as the main sources of irreversibility. Similarly, Rakopoulos and coworkers applied second law analysis on a naturally aspirated diesel engine based on experiments [8], and later developed a thermodynamic model applied on a multi-cylinder turbocharged diesel engine, using experimental data to calibrate the model for the first law analysis; the results indicated combustion as main source of irreversibility, evaluating also the effect of speed, load and compression ratio on second-law balance [9].

As in most of the previous research work on ICEs second-law was based both on thermodynamic cycle theory and on experimental tests on real engines, the present paper provides data on energy and exergy balance, with irreversibility terms calculated for each process and device within the structure of a turbocharged and intercooled diesel engine.

The purpose of the paper is to prepare an instrument for engine irreversibility evaluation in order to select and develop energy optimization and waste heat recovery techniques.

2. Theoretical Approach

2.1. The Cycle

In order to analyze the main engine energy losses, the paper applies both the theory of ideal gas cycles in four components – compressor-intercooler- engine and turbine and some dynamometric test bench measurements of a real engine.

The simplified model of thermodynamic cycle described in [10] is applied to a four stroke dual combustion diesel engine, in the hypothesis of an open system, with the thermal agent having the properties of air, which evolves continuously in ICE in stationary flow.

The basic thermodynamic processes are associated to each component and are summarized in Table 1.

The ambient air parameters are pressure p_0 and temperature T_0. In the process 0-1 air is inducted in the centrifugal compressor of the turbocharger at pressure p_1, lower than p_0 , suffering a throttling pressure loss ψ_1, then it is compressed in the 1-s process up to pressure p_s, its temperature being increased to T_s.

The heated air is then cooled in a special purpose heat exchanger, known as intercooler, in the process s-2 and its pressure is reduced by throttling with pressure loss ψ_2 due to section variations and frictions along intercooler passages.

The air enters the engine in the induction process 2-a, filling the cylinder initially at constant volume and then at constant pressure p_a, lower than p_2, due to throttling loss ψ_a across intake port and valve.

During the 2-a intake process there is an overlapping with another process known as scavenging when, for a short interval, both engine valves are simultaneously opened, and fresh compressed air passes from intake manifold into exhaust manifold, pushing out the exhaust gas; in terms of irreversibility the scavenging is considered to be a mixing process of air, residual gas and exhaust gas accompanied by heat exchange.

Table 1. Thermodynamic processes in the components

Process	**Compressor**	Specific parameters
0-1	Air aspiration at $p_1<p_0$	$\psi_1= (p_0-p_1)/p_0$
1-s	Irreversible adiabatic air compression	k, p_s, T_s
	Intercooler	
s-s'	Isobaric cooling of charge air	$\tau=\Delta T_R/\Delta T_{Rmax}$
s'-2	Isothermal charge air throttling	$\psi_2= (p_s-p_2)/p_s$
	Engine	
2-a'	Isothermal charge air throttling at cylinder intake	$\psi_a= (p_2-p_a)/p_2$
a'-a	Irreversible cylinder scavenging between intake valve opening and exhaust valve closing	β- scavenging ratio
a-c	Adiabatic air compression	ε
c-y	Isochoric combustion c-y	λ
y-z	Isobaric combustion y-z	ρ
z-d	Adiabatic gas expansion z-d	(λ/ρ)
d-e	Exhaust d-e	p_d , p_r
e-t	Isothermal exhaust gas throttling	$\psi_e= (p_r-p_t)/p_r$
	Turbine	
t-3	Irreversible adiabatic gas expansion	k
3-e_T	Exhaust gas pressure loss to p_0	$\psi_3= (p_3-p_0)/p_3$

Quantitatively, the scavenging is measured by the scavenging ratio, β, defined as fresh air mass flow rate released in the exhaust manifold reported to fresh air mass flow rate retained in the cylinder.

In the a-c process the air charge is compressed in the engine cylinder, then two combustion processes are initiated: isochoric c-y and isobaric y-z.

The products of combustion expand in the z-d phase producing mechanic work and being freely expelled.

During the exhaust stroke the piston forcedly releases the rest of exhaust gas found at p_r and $T_e > T_r$ due to scavenging.

The gas flow pressure drops due to section variations in the exhaust valve and port with pressure loss coefficient ψ_e.

The exhaust gas is expanded further in the centrifugal turbine, in the t-3 process, and then in the process 3-eT towards the environment with throttling pressure loss ψ_3.

The processes are illustrated in Figures 1-3 and the input data are given in Table 2.

Table 2. Input data for the calculation

Input data	
p_0	Ambient pressure [Pa]
T_0	Ambient temperature [K]
R	Gas constant [J/kgK]
c_p, c_v	Specific heat at constant pressure or volume [J/kgK]
H_i	Lower heating value of the fuel [MJ/kg]
ε	Engine compression ratio
L_0	Stoichiometric mass of air for total combustion of 1kg of fuel [kg air/kg fuel]
$\psi_a=\Delta p_a/p_2$	Pressure loss in engine intake
$\psi_e=\Delta p_r/p_r$	Pressure loss in engine exhaust
$\psi_3=\Delta p_3/p_3$	Pressure loss in turbine
V_c	Volume of combustion chamber [l]
V_S	Volume of the stroke [l] or cylinder displacement
p_{max}	Maximum pressure in the cylinder [bar]
p_s	Air compression pressure [Pa]

For each transformation within the cycle there were determined thermodynamic parameters, mainly pressures and temperatures, as follows:

Pressure at compressor intake, p_1

$$p_1 = \left(1-\psi_1\right) p_0 \tag{1}$$

Pressure at intercooler exit, p_2

$$p_2 = \left(1-\psi_2\right) p_s \tag{2}$$

Pressure at engine intake, p_a

$$p_a = \left(1-\psi_a\right) p_2 \tag{3}$$

Temperature at compressor exit T_s is measured on the test bench, knowing the type of the turbocharger and its internal efficiency of the compressor η_s .

$$T_s = T_0 \cdot \left[\; 1+\frac{\left(\frac{p_s}{p_1}\right)^{\frac{k-1}{k}}-1}{\eta_s} \;\right] \tag{4}$$

with $k = c_p/c_v$

Temperature at the intercooler exit T_2 is calculated based on effectiveness of the air-to-air heat exchanger, τ, which is defined as the ratio of the real charge air temperature drop across the cooler core to the temperature differential available for cooling [11]:

$$T_2 = T_0\left[\tau+\left(1-\tau\right)\frac{T_s}{T_0}\right] \tag{5}$$

The density of charge air at the of intercooler exit, ρ_2 :

$$\rho_2 = \frac{p_2}{RT_2} \tag{6}$$

Temperature of the charge air at engine intake, T_a , is calculated taking into account the mixing with residual gas fraction, γ_R, existent in the cylinder, at temperature T_R :

$$T_a = \frac{T_2+\gamma_R T_R}{1+\gamma_R} \tag{7}$$

$$\varphi_a = \frac{T_a}{T_2} \tag{8}$$

The coefficient φ_a indicates the increase of intake temperature as result of mixing with residual gas.

Figure 1 represents the transformations of the working fluid taken place in compressor, in temperature-entropy coordinates, thus being used for evaluation of irreversibility.

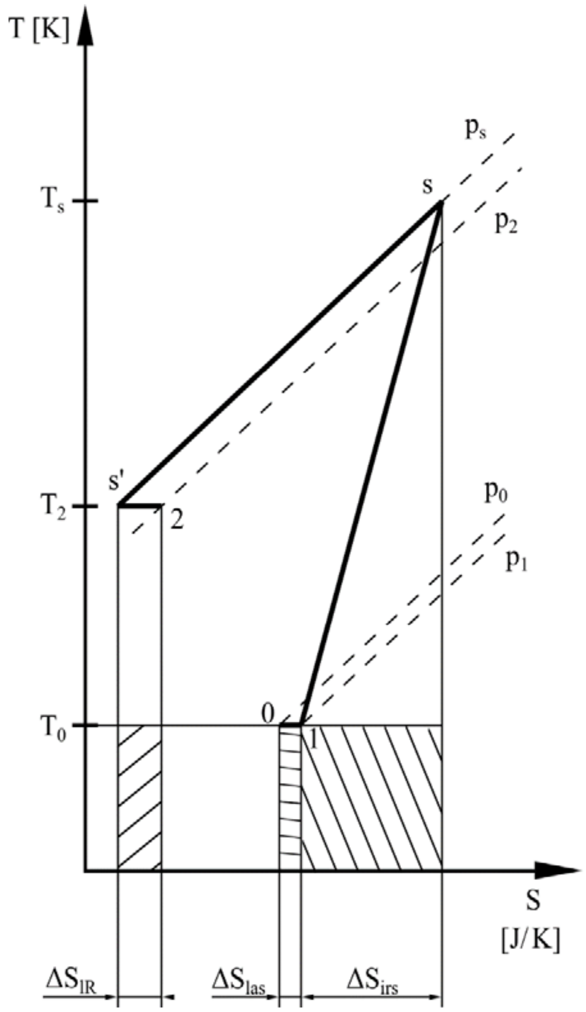

Figure 1. Compressor-thermodynamic processes

The compression stroke rises the pressure to value p_c and temperature T_c :

$$p_c = p_a \varepsilon^k \tag{9}$$

$$T_c = T_a \varepsilon^{k-1} \tag{10}$$

The pressure rise in isochoric combustion, λ:

$$\lambda = \frac{p_z}{p_c} \tag{11}$$

That parameter can be calculated knowing the maximum value of the cylinder pressure, p_z.

$$\rho = \frac{V_z}{V_c} \tag{12}$$

In order to find the parameter ρ, the thermal energy balance of the combustion process is applied:

$$Q_{ar} = Q_v + Q_p = m_c c_v \left(T_y - T_c\right) + m_c c_p \left(T_z - T_y\right) \tag{13}$$

The total heat Q_{ar} released during combustion consists of isochoric heat Q_v and isobaric heat Q_p in which m_c represents the mass of working fluid.

$$T_z = \rho T_y = \rho \lambda T_c \tag{14}$$

$$Q_{ar} = m_c R T_c \cdot \frac{\lambda - 1 + k\lambda\left(\rho - 1\right)}{k-1} \tag{15}$$

By reporting Q_{ar} to the engine displacement unit, V_s , it can be extracted ρ:

$$q_{ar} = \frac{Q_{ar}}{V_s} = \frac{m_c H_i}{V_s} = p_c \cdot \frac{\lambda - 1 + k\lambda\left(\rho - 1\right)}{\left(k-1\right)\left(\varepsilon - 1\right)} \tag{16}$$

$$\rho = 1 + \frac{k-1}{k}\left(\frac{\varepsilon - 1}{p_z} q_{ar} - \frac{1}{k-1} \cdot \frac{\lambda - 1}{\lambda}\right) \tag{17}$$

The maximum temperature in the cycle, T_z:

$$T_z = \rho T_y \tag{18}$$

The temperature T_d and pressure p_d at the end of expansion stroke :

$$T_d = \frac{T_z}{\left(\dfrac{\varepsilon}{\rho}\right)^{k-1}} \tag{19}$$

$$p_d = \frac{p_z}{\left(\dfrac{\varepsilon}{\rho}\right)^{k}} \tag{20}$$

The exhaust gas pressure p_r can be calculated in function of pressure of the gas entering the turbine p_t to which is added the fraction ψ_e to compensate the throttling during flow on exhaust valve, port and manifold.

$$p_r = \frac{p_t}{1 - \psi_e} \tag{21}$$

The transformations performed in the engine are illustrated in Figure 2.

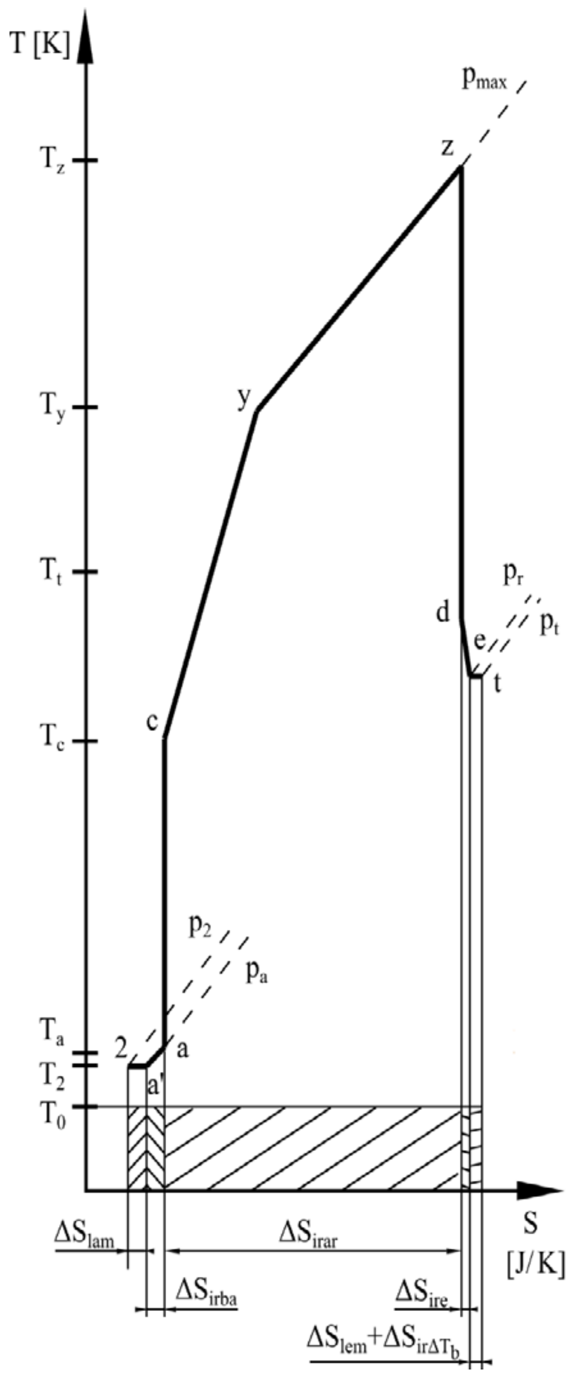

Figure 2. Diesel engine- thermodynamic processes

The ratio between exhaust pressure p_r and p_d expansion stroke pressure was used to evaluate the temperature of the gas in exhaust manifold, T_e , and to validate the residual gas fraction γ_R :

$$\frac{T_e}{T_d} = \frac{1}{k} \cdot \frac{1+\left(k\frac{\varepsilon-1}{\varepsilon}-1\right)\frac{p_r}{p_d}}{1-\frac{1}{\varepsilon}\left(\frac{p_r}{p_d}\right)^{\frac{1}{k}}} \tag{22}$$

$$\gamma_r = \frac{1}{\varepsilon} \cdot \left(\frac{p_r}{p_d}\right)^{\frac{1}{k}} \tag{23}$$

The exhaust gas temperature T_r was calculated based on the hypothesis that during free exhaust the expansion of the gas is adiabatic [10].

$$T_r = T_d \cdot \left(\frac{p_r}{p_d}\right)^{\frac{k-1}{k}} \tag{24}$$

During the scavenging process taking place between induction valve opening and exhaust valve closing, there is an isobaric mixing process of exhaust gases at temperature T_e and scavenged air at temperature T_b and with conservation of gas enthalpies. As a result scavenging temperature T_b can be calculated:

$$\frac{T_b}{T_a} = \frac{k\left(\frac{\beta}{\varphi_a}(1-\gamma_r)-1\right)+\frac{1}{\varepsilon}\left(k+\frac{p_r}{p_a}-1\right)}{k(\beta-1)(1-\gamma_r)} \tag{25}$$

Finally, the gas temperature upwards turbine T_t depends both on the characteristics of exhaust gas and scavenged air, calculated based on the scavenging factor β :

$$T_t = \frac{(\beta-1)T_b + T_e}{\beta} \tag{26}$$

The temperature of exhaust gas at turbine exit, T_3, is based on turbine efficiency, η_T:

$$T_3 = \left\{1-\eta_T\left[1-\left(\frac{p_3}{p_t}\right)^{\frac{k-1}{k}}\right]\right\}T_t \tag{27}$$

The pressure p_3 is related to ambient pressure p_0 with the pressure loss factor ψ_3 :

$$p_3 = \frac{p_0}{1-\psi_3} \tag{28}$$

The transformations of the working fluid performed in turbine are presented in Figure 3.

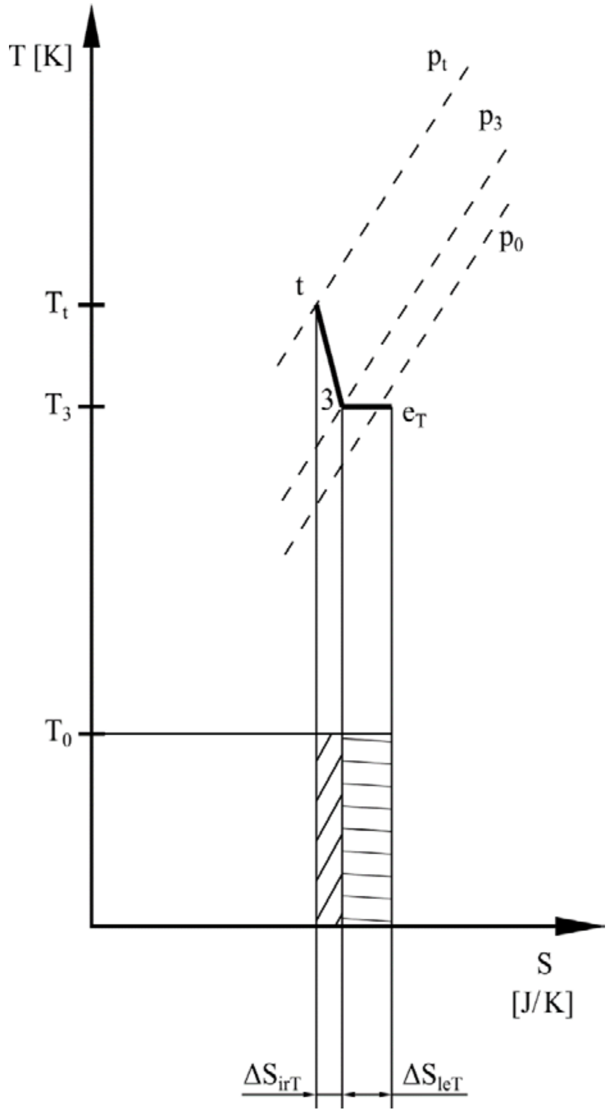

Figure 3. Turbine - thermodynamic processes

Being known that the mechanical works of each transformation, during intake, compression, isobaric combustion, expansion and exhaust, it can be calculated the mean pressure of the cycle p_{mc} and further the thermodynamic cycle efficiency, η_t .

$$p_{mc} = \frac{W_{in} + W_{comb} + W_{\exp} - |W_{comp}| - |W_{ex}|}{V_S} \tag{29}$$

$$\eta_t = \frac{p_{mc}}{q_{ar}} \tag{30}$$

2.2. Irreversibility Losses

For the control volume enclosing the engine and its auxiliaries, at steady state operation, there can be written exergy equation based on exergy transfer in form of work, heat transfer, working fluid change flow in and out the control volume and exergy destruction [12].

The exergy balance applied in the diesel engine, written for sections 0 (air intake in the compressor) and e_T (exhaust gas released from turbine) states that the exergies of the heat exchanges (combustion, heat released through exhaust gas and heat released through intercooler) equalize the variation of mechanic works within the system and losses through internal irreversibilities of the processes [10].

By marking $\Pi_{ir} = T_o \Delta S_{ir}$ (with ΔS_{ir} -entropy variation) as losses caused by internal irreversibility of a thermodynamic process, and assuming that the mechanical work produced in turbine W_T is totally consumed for the driving of the compressor, W_S, then the exergy of combustion process is equal only with the variation of engine mechanical work, W_e , and sum of irreversibilities of the heat released by exhaust gas to environment , $\Pi_{Q_{eT}}$, heat released in the intercooler, Π_{QR}, and losses caused by internal irreversibility of thermodynamic processes $\sum_j \Pi_{irj}$.

$$Q_{ar} - \Pi_{irar} = W_e + \Pi_{QR} + \Pi_{Q_{eT}} + \sum_j \Pi_{irj} \quad (31)$$

The calculation of irreversibilities as energy losses Π_{ir} [MJ], are reported to mass of fuel, thus resulting π_{ir} [MJ/kg].

Combustion losses

$$\Pi_{irar} = T_o \Delta S_{irar} = \left(m_a + m_f\right) T_0 \left(\Delta s_{qv} + \Delta s_{qp}\right) \quad (32)$$

m_a - mass of air, m_f - mass of fuel

$$\Pi_{irar} = m_f \left(1 + \alpha L_0\right) T_0 \left(c_v \ln \frac{T_y}{T_c} + c_p \ln \frac{T_z}{T_y}\right) \quad (33)$$

α - air fuel equivalence ratio,

$$\pi_{irar} = \left(1 + \alpha L_0\right) R T_0 \frac{1}{k-1} \ln \lambda \rho^k \quad (34)$$

Heat transfer losses

Π_{QR}- loss due to irreversibility of heat transfer between intercooler air and ambient air.

$$\Pi_{QR} = \left|Q_R\right| - T_0 \left|\Delta S_{QR}\right| = m'_a c_p \left(T_S - T_2 - T_0 \ln \frac{T_S}{T_2}\right) \quad (35)$$

with $m'_a = \beta \alpha L_0$ - mass of air including scavenging air.

$$\pi_{QR} = \beta \alpha L_0 R \frac{k}{k-1} \left(T_S - T_2 - T_0 \ln \frac{T_S}{T_2}\right) \quad (36)$$

$\Pi_{Q_{eT}}$ - loss due to irreversibility of heat transfer between exhaust gas and ambient air, at turbine exit.

$$\Pi_{Q_{eT}} = \left|Q_{eT}\right| - T_0 \left|\Delta S_{QeT}\right| = $$
$$= (m'_a + m_f) \frac{k}{k-1} R T_0 \left(\frac{T_{eT}}{T_o} - 1 - \ln \frac{T_{eT}}{T_o}\right) \quad (37)$$

$$\pi_{QeT} = \left(1 + \beta \alpha L_0\right) R T_0 \frac{k}{k-1} \left(\frac{T_{eT}}{T_o} - 1 - \ln \frac{T_{eT}}{T_o}\right) \quad (38)$$

Π_{irS} - loss due to irreversibility of air compression in compressor

$$\Pi_{irS} = T_0 \left|\Delta S_{irS}\right| = m'_a T_0 \Delta s_{irS} \quad (39)$$

$$\pi_{irS} = \beta \alpha L_0 R T_0 \left(\frac{k}{k-1} \ln \frac{T_S}{T_1} - \ln \frac{p_s}{p_1}\right) \quad (40)$$

Π_{irT} - loss due to irreversibility of gas expansion in the turbine

$$\Pi_{irT} = T_0 \left|\Delta S_{irT}\right| = \left(m'_a + m_f\right) T_0 \left(c_p \ln \frac{T_3}{T_t} - R \ln \frac{p_3}{p_t}\right)$$

$$\pi_{irT} = \left(1 + \beta \alpha L_0\right) R T_0 \left(\ln \frac{p_t}{p_3} - \frac{k}{k-1} \ln \frac{T_3}{T_t}\right) \quad (41)$$

Π_{irba}- loss due to irreversibility of intake- scavenging process

$$\Pi_{irba} = T_0 \Delta S_{irba} = m'_a T_0 \Delta s_{irT} = \left(\beta m_a\right) T_0 c_p \ln \frac{T_a}{T_2}$$

$$\pi_{irba} = \left(\beta \alpha L_0\right) R T_0 \frac{k}{k-1} \left(\ln \frac{T_a}{T_2}\right) \quad (42)$$

Π_{ire} - loss due to irreversibility of exhaust gas process

$$\Pi_{ire} = T_0 \Delta S_{ire} = m_f \left(1 + \alpha L_0\right) RT_0 \frac{k}{k-1} \ln \frac{T_e}{T_r}$$

$$\pi_{ire} = \left(1 + \alpha L_0\right) RT_0 \frac{k}{k-1} \ln \frac{T_e}{T_r} \quad (43)$$

$\Pi_{ir\Delta Teb}$ - loss caused by irreversibility of heat transfer between exhaust gas and scavenging air

$$\Pi_{ir\Delta Teb} = T_0 \left(\Delta S_{Qb} - \left| \Delta S_{Qe} \right| \right) =$$
$$= T_0 \left(m_b c_p \ln \frac{T_t}{T_b} - \left(m_a + m_f \right) c_p \ln \frac{T_e}{T_t} \right) \quad (44)$$

with $m_b = m'_a - m_a$ mass of scavenging air,

$$\pi_{ir\Delta Teb} = RT_0 \frac{k}{k-1} \left(\left(\beta - 1\right) \alpha L_0 \ln \frac{T_t}{T_b} - \left(1 + \alpha L_0\right) \ln \frac{T_e}{T_t} \right)$$

Throttling losses

Π_l - losses caused by irreversibility of pressure drop of the working fluid in compressor, intercooling heat exchanger, engine intake, engine exhaust and turbine:
-in compressor:

$$\Pi_{laS} = T_0 \cdot \Delta S_{laS} = m'_a T_0 \Delta s_{laS} = \beta m_a RT_0 \ln \frac{p_0}{p_1} \quad (45)$$

$$\pi_{laS} = \beta \alpha L_0 RT_0 \ln \frac{1}{1 - \psi_1} \quad (46)$$

-in the intercooler:

$$\Pi_{lR} = T_0 \cdot \Delta S_{lR} = m'_a T_0 \Delta s_{lR} = \beta m_a RT_0 \ln \frac{p_s}{p_2} \quad (47)$$

$$\pi_{lR} = \beta \alpha L_0 RT_0 \ln \frac{1}{1 - \psi_2} \quad (48)$$

-at engine intake:

$$\Pi_{laM} = T_0 \cdot \Delta S_{laM} = m'_a T_0 \Delta s_{laM} = \beta m_a RT_0 \ln \frac{p_2}{p_a}$$

$$\pi_{laM} = \beta \alpha L_0 RT_0 \ln \frac{1}{1 - \psi_a} \quad (49)$$

-at engine exhaust:

$$\Pi_{leM} = T_0 \cdot \Delta S_{leM} = \left(m'_a + m_f \right) RT_0 \ln \frac{p_r}{p_t} \quad (50)$$

$$\pi_{leM} = (1 + \beta \alpha L_0) RT_0 \ln \frac{1}{1 - \psi_e} \quad (51)$$

-turbine exhaust

$$\Pi_{leT} = T_0 \cdot \Delta S_{leT} = \left(m'_a + m_f \right) RT_0 \ln \frac{p_3}{p_0} \quad (52)$$

$$\pi_{leT} = (1 + \beta \alpha L_0) RT_0 \ln \frac{1}{1 - \psi_3} \quad (53)$$

3. Numerical case study

3.1. Experimental data collection

The theoretic model was applied to a four-stroke turbocharged and intercooled diesel engine for commercial vehicles whose main characteristics are presented in Table 3:

Table 3. Engine technical data

Engine type	Direct injection
Cylinder configuration	6-cylinder, in line
Bore x Stroke [mm]	102 x112
Displacement [L]	5.5
Compression ratio	17:1
Rated power [kW]	118
Rated speed [rpm]	2600
Max. torque [N·m]	460
Max. torque speed [rpm]	1700

The tests were performed at facilities of Road Vehicle Institute on a special purpose 300kW eddy-current dynamometric test bench, instrumented for measurements of ambient air, charge air and exhaust gas temperatures, pressures and flowrates; the acquired data were processed in the associated National Instruments LabVIEW environment.

The measurements have been performed upstream and downstream compressor, engine and turbine, resulting temperatures and pressures in the cycle (eq. 1-28) and pressure loss coefficients ψ, as defined in Table 2.

During the engine testing the ambient temperature t_0 was 20°C and atmospheric pressure p_0, 95190 Pa.

Three full load engine speed tests were performed at total load on the speed range 1200-2600 rpm, being measured effective power, torque, hourly fuel consumption, specific fuel consumption; test data were averaged and the performances were corrected according to ISO 1585 standard.

3.2. Numerical-experimental correlations

For engine and turbine compressions and expansions, the adiabatic exponents were taken k = 1.3 and gas constant R=287 J/kg K, neglecting the change in gas composition, specific heat variation with temperature and blow-by flowrate.

The scavenging ratio was adopted as β = 1.15

according to recommendations from reference [13]; experimental measures of pressure losses, air fuel equivalence ratios, boost pressures and compressor and turbine efficiencies from turbo machine steady-state map (Holset H1S type turbocharger) were used to calculate cycle data.

The maximum pressures in the cycles were measured with the indicator diagram in order to calculate the pressure increase ratio.

Mean pressures of the cycles were calculated with equation (29), and thermal efficiencies were calibrated using the indicated efficiency; mechanical power losses were measured with the Willan line method, thus yielding the effective efficiencies.

A synopsis of the most important values in the calculations is presented in Table 4, for rated speed and maximum torque runs.

Table 4. Thermodynamic cycles data

Parameter	Rated speed	Maximum torque
ψ_1	0.013	0.0036
ψ_2	0.034	0.0182
ψ_a	0.02	0.01
ψ_e	0.05	0.04
ψ_3	0.047	0.0122
γ_r	0.03	0.02
α	1.9	1.5
τ	0.68	0.60

3.3. Energy and exergy balance

First law balance

By applying energy conservation in the time unit on a control volume around the engine, in a steady state operation, it yields that chemical energy of the diesel fuel $\dot{m}_f H_i$ is distributed on three main energy paths; the first one is the effective work reported to time, generating engine brake power, P_e, the second one is heat rate released through exhaust gas which is equivalent to $\dot{Q}_{eT}$ and the third one is the heat exchange rate from cylinder to coolant, oil and ambient air, $\dot{Q}_c$.

In most of engine balances [14] there is a fourth term named miscellanea losses $\dot{Q}_{mis}$ which are difficult to measure (heat transfer of turbocharger to the oil, blow-by gas enthalpy, incomplete combustion, experimental unbalance), evaluated to 2% of fuel energy which was not considered separately.

For this case study, the energy balance was done by measuring engine brake power P_e, and heat rate released through exhaust gas $\dot{Q}_{eT}$, the difference up to unit was considered to be $\dot{Q}_c$ which in this case contains also the miscellanea losses.

The engine tests include also data on effective power and mechanical efficiency which are used to calibrate indicated efficiency.

Second law balance

The diesel fuel chemical availability, A_f , can be calculated in function of lower heating value, H_i, and chemical composition, according to [15], [16], [17], neglecting the sulfur content:

$$A_f = H_i \left(1.0401 + 0.01728 \frac{y}{z} + 0.0432 \frac{p}{z} \right) \qquad (54)$$

with y, z, p – hydrogen, carbon and oxygen atoms in molecule. As nowadays diesel fuel can be a blend with up to 7% biodiesel, fuel which contains roughly 10% oxygen, it was evaluated the oxygen content in the fuel based on keeping the same molar mass of 197.7 kg/kmol attributed to petroleum diesel fuel $C_{14.4}H_{24.9}$ [2], on the expense of a slight reduction of carbon content. The equivalent of 2% mass of O_2 in the fuel, introduced with the biodiesel, leads to the formula $C_{14.07}H_{24.9}O_{0.12}$ and the previous formula of A_f becomes $A_f = H_i \cdot 1.071$.

Also, that fuel composition requires a minimum mass of oxygen for combustion of 1 kg of fuel, L_0=14.2 (kg air/kg fuel).

The lower heating value of the diesel fuel was taken H_i = 42.5 MJ/kg and, consequently, A_f = 45.5 MJ/kg.

4. Results and discussions

From the full load engine speed characteristic there were selected two representative operation points of a vehicle engine, at rated speed (2600 rpm) and to maximum torque speed (1700 rpm). The first one represents the maximum vehicle speed capacity and the second the maximum traction capacity.

By applying the irreversibility formulas from section 2.2 there were calculated the availability losses for both operation points.

Only the irreversibilities of the processes and the throttling losses are highlighted in the hatched areas of Figures 1-3.

Both compressor and turbine losses include irreversible and throttling losses, and they are defined by equations (40, 45), respectively (41, 53). Intercooler losses are throttling losses, as described in equation (48).

The inlet loss is a throttling one (eq.49) and exhaust losses included irreversibilities of the processes (eq.43) and throttling (eq.51).

The distribution of the losses in the diesel engine can be evaluated for each engine subsystem, as working agent passes through them.

Table 5 presents the assessment of first and second law terms for the rated speed point.

Table 5. Percentages of first and second law terms

Rated speed 2600 rpm	**First law**, (% of fuel energy)	**Second law**, (% of fuel availability)
Indicated work	43	40.15
Exhaust gas	33	13.96
Heat transfer	24	16.51
Combustion loss	-	19.24
Compressor	-	2.5
Intercooler	-	0.90
Engine intake	-	0.89
Engine exhaust	-	1.78
Turbine	-	1.29
Thermal mixing	-	0.90
Unaccounted losses	-	1.88
Total	100	100

The difference between the indicated works expressed according to first and second laws in Table 5 appears due to the factor 1.071 which multiplies fuel lower heating value, increasing chemical diesel fuel availability.

The second law values in Table 5 were computed by reporting all the □□$_{\text{ir}}$ terms to A_f , the results being expressed as percentages of fuel availability [18].

The interpretation of the data from Table 5 is highly relevant for the energy usability: while 24% of fuel energy is released through heat transfer from combustion chamber, it is usable further in form of heat transfer only as 16.51% of fuel availability.

Similarly, 33% of fuel energy is expelled in exhaust gas, but just 13.96% of fuel availability, their percentage ratio, roughly the percentage ratio of those figures (40%) can be converted backwards into work in ideal thermodynamic devices.

The term combustion loss represents the deviation of the real combustion from an ideal, reversible process. It represents the highest availability destruction within the engine, evaluated for this point at 70% of the total internal irreversible losses, excepting unaccounted losses.

The compressor losses (9%) are higher than the turbine ones (4.7%), the fact which was found out by other authors and could be explained by lower isentropic efficiency of this aggregate [9]. Engine exhaust contributed with an important share of 8.1%, while inlet and intercooler have 4.9%, and, respectively, 3.3%.

The results from Table 5 were analyzed by comparison with similar second-law analysis performed in references [5], [6], [9]. The synthesis of the compared values is illustrated in Table 6.

Table 6. Second law terms comparison

(%) Fuel availability	Present paper	[5]	[9]	[6]
Power (kW) / Intercooled	118 / I	220	235 / I	224 / I
Indicated work	40.15	43.9	40.31	39.21
Exhaust gas	13.96	14.01	13.45	12.73
Heat transfer	16.51	17.6	18.95	13.98
Combustion loss	19.24	19.2	21.9	21.2
Compressor loss	2.5	1.4	1.13	1.64
Total inlet loss (only throttling)	0.89	- (0.7)	0.72	- (0.58)
Total exhaust loss (only throttling)	2.23	- (2.3)	2.69	- (1.66)
Turbine loss	1.29	0.8	0.85	1.69
Thermal mixing	0.9	-	-	0.81

Both for heat rejected from the engine and for heat in the exhaust gas it is confirmed a Carnot cycle dependency of availability of heat in the form:

$$A_Q = Q\left(1 - \frac{T_0}{T}\right) \qquad (55)$$

with Q- heat in the first law, A_Q- availability of heat Q according to second law.

Regarding the structure of availabilities mentioned in Table 6, it can be noticed that the present paper provides distinct data on availability loss in intercooler which was not reported in the aforementioned references.

By analyzing the second point calculations at a lower speed, 1700 rpm, it can be emphasized the changes on irreversibilities within the engine in comparison to the first operation point (2600 rpm), as displayed in Figure 4.

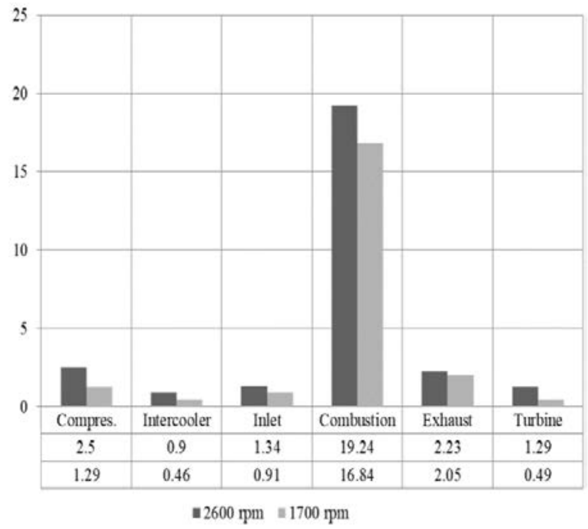

Figure 4. Distribution of component losses (% of fuel availability) at rated power and maximum torque speeds

At the maximum torque speed 1700 rpm, the dominant irreversibility is, as expected in the combustion process, ranked at 16.84% of fuel availability (75% of the total internal irreversibilities) thus indicating that exergy destruction is smaller than at the rated speed (19.24% of the fuel availability).

As a general observation, the internal irreversibilities in engine subsystems were distributed similarly as in previous case. The compressor losses (5.7%) are higher than the turbine ones (2.1%); engine exhaust contributed with a important share of 9% , while inlet and intercooler have 4%, and, respectively, 2% ; there is a notable change of order in the hierarchy of components, on the second position, after combustion irreversibilities, the exhaust line losses replaced the compressor ones. The reason is the higher level of temperatures in the exhaust duct than in rated speed situation.

A result of present assessment is the comparison between exhaust gas availability in maximum speed and maximum torque full load operation modes.

For 1700 rpm exhaust gas has 15.31% from fuel availability, higher than 13.96 % for 2600 rpm.

This is a valuable information related to the waste heat recovery technologies which could select the appropriate engine operation mode for a better exhaust energy harvesting such as organic Rankine bottoming cycle or thermoelectric generators.

Similarly, assuming that the diesel engine could drive a Combined Heat and Power generator set, for the same engine indicated efficiency, it may be demonstrated that heat transfer availability in the coolant is higher at lower speeds, thus determining the appropriate selection of the engine constant speed.

The influence of speed on computed irreversibilities was in accordance with results reported in [3], [18]; by increasing speed, the total irreversibilities increase from 22.8% to 27.5%, exhaust gas availability increases from 13.96% to 15.31%, turbocharger irreversibility increases for both aggregates from 1.78% to 3.79%.

The limitations of classical thermodynamic model which involved assumptions like:

- the working fluid is a perfect gas,
- gas properties have negligible variation with temperature,
- gas composition is similar to air, were partly compensated by experimental data.

The theoretic model is flexible because it can cover both compression ignition and spark ignition engines with dissimilar air supply systems, easily adapted numerically; for turbocharged version without intercooling (τ=0), is eliminated the effect of the heat exchanger. For the naturally aspirated version, if coordinates corresponding to point s are replaced with point a, the effect of turbocharger is totally released. For the spark engine assessment, the isobaric combustion is eliminated if $\rho = 1$.

5. Conclusions

The first-law energy balance applied to assess the engine performances is limited in explaining unrecoverable degradations, so it is needed a second-law analysis to evaluate the availability balance over each process. The theoretical model calibrated with experimental data generated irreversibility values was proved to be rapid and accurate (engine energy losses and exergy being associated to each energy path), in good agreement with other research works. The main findings are 70-75% availability destruction in combustion process and the decrease of irreversibilities with speed reduction. The computed exergy losses for two representative operation modes demonstrated a gain in capacity of performing useful work of the maximum torque point of 6.05% of fuel availability (4.7% difference of internal irreversibilities + 1.35% difference from exhaust gas heat) equivalent to 6.5% of fuel energy saving.

References

[1] Caton J. (2018). The Thermodynamics of Internal Combustion Engines: Examples of Insights. *Inventions*, *3*(2), 33.

[2] Moran, M. J. (1989). *Availability analysis: a guide to efficient energy use corrected edition*. New York: ASME press.

[3] Rakopoulos, C. D., & Giakoumis, E. G. (2006). Second-law analyses applied to internal combustion engines operation. *Progress in energy and combustion science*, *32*(1), 2-47.

[4] Flynn, P. F., Hoag, K. L., Kamel, M. M., & Primus, R. J. (1984). *A new perspective on diesel engine evaluation based on second law analysis* (No. 840032). SAE Technical Paper.

[5] Heywood, J. B. (1988). Internal combustion engine fundamentals. McGraw Hill Book Co. *New York*.

[6] Primus, R. J., & Flynn, P. F. (1986). The assessment of losses in diesel engines using second law analysis *Computer-Aided Engineering and Energy Systems*, *3*, 61-68.

[7] Alkidas, A. C. (1989). *The use of availability and energy balances in diesel engines* (No. 890822). SAE Technical Paper.

[8] Rakopoulos, C. D., Andritsakis, E. C., & Kyritsis, D. K. (1993). Availability accumulation and destruction in a DI diesel engine with special reference to the limited cooled case. *Heat Recovery Systems and CHP*, *13*(3), 261-276.

[9] Rakopoulos, C. D., & Giakoumis, E. G. (1997). Speed and load effects on the availability balances and irreversibilities production in a multi-cylinder turbocharged diesel engine. *Applied Thermal Engineering*, *17*(3), 299-313.

[10] Radcenco, V. (1976). Termodinamica tehnica si masini termice. Procese ireversibile. *EDP, Bucuresti*.

[11] Challen, B., & Baranescu, R. (Eds.). (1999). *Diesel engine reference book*. McFarland.

[12] Moran, M. J., & Tsatsaronis, G. (2017). Engineering thermodynamics. In *CRC Handbook of Thermal Engineering* (pp. 1-112). CRC Press.

[13] Grunwald, B. (1980). Teoria, calculul și construcția motoarelor pentru autovehicule rutiere, *EDP, Bucuresti*.

[14] Payri, F., Olmeda, P., Martín, J., & Carreño, R. (2015). Experimental analysis of the global energy balance in a DI diesel engine. *Applied Thermal Engineering*, *89*, 545-557.

[15] Stepanov, V. S. (1995). Chemical energies and exergies of fuels. *Energy*, *20*(3), 235-242.

[16] Rodriguez, L. (1980). Calculation of available energy quantities. *Thermodynamics: second law analysis*, 39-60.

[17] Szargut, J., Styrylska, T. (1964). Angenäherte Bestimmung der Exergie von Brennstoffen. *Brennst.-Wärme-Kraft 16*(12), 589–596.

[18] Kyritsis, D. C., & Rakopoulos, C. D. (2001). *Parametric study of the availability balance in an internal combustion engine cylinder* (No. 2001-01-1263). SAE Technical Paper.