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FACTORS AFFECTING EXERGY TERMS IN NATURALLY ASPIRATED DIESEL ENGINES

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Abstract: An exergy model for naturally aspirated compression ignition engines has been validated with experimental data from a four-cylinder 3.92-litre engine testing on a dynamometric test bed. For full load, in two engine operation modes at rated speed and maximum torque speed, the influence of the atmospheric pressure and temperature factors on the main exergy terms was investigated, as well as that of the compression ratio and excess air coefficient. The results indicate the paths to reduce exergy destruction by means of engine design and operation parameters.

Keywords: Naturally-Aspirated Diesel Engine, Efficiency, Irreversibilities, Second-law analysis

1 INTRODUCTION

Nowadays the demands on energy aim not only to its quantity, but also on its quality of producing useful work. So, in the field of internal combustion engines, a combination of first and second laws of thermodynamics is needed to maximise the work output, using the measure called exergy (or availability), a form of energy related both to the system and its environment.

Exergy analysis has been reported in literature and review articles [1-3] in order to identify and reduce the exergy losses caused mainly by combustion, heat transfer, friction and throttling.

Some previous research works on second law analysis of naturally aspirated diesel engines included the simulations on standard thermodynamic cycles [4], finding relation between experimental fuel burning rate and availability destruction, assessing the energy recovery potential and evaluating the influence of engine speed on exhaust gas irreversibilities [5-7].

The present paper aims to address the influence of dead state parameters and functional parameters on the energy parameters and exergy terms specific to naturally aspirated diesel engines.

The main instruments of the research work are the thermodynamic cycle numerical code and the set of experimental data measured on the engine test bench.

2 EXPERIMENTAL PROCEDURE

The engine code is 392-L4-D, having the main technical characteristics presented in Table 1 [8]. The engine used for the present investigations was manufactured by ROMAN Truck Factory to power road vehicles, being a four-stroke, compression ignition, liquid cooled engine, with an omega-shaped combustion chamber in the piston. The engine was tested on a 220 kW eddy-current dynamometric engine test (Ono-Sokki) on the facilities of Road Vehicle Institute (INAR, Brasov), instrumented for the measurement of speed, load, air and fuel consumption, exhaust gas flow rates, coolant and lubricant temperatures and pressures. The power declaration in Table 1 was corrected according to ISO 1585 standard, the engine test code for net power.

Table 1

Main engine parameters 392-L4-D

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Engine type	Direct injection
Cylinder configuration	4-cylinder
Bore x Stroke [mm]	102 x120
Displacement [L]	3.92
Compression ratio, ε	18.5:1
Rated power [kW]	60.5
Rated speed [rpm]	2800
Maximum torque [N·m]	248
Maximum torque speed [rpm]	1600

For the purpose of this study, engine speed characteristics were performed at full load, being defined two representative operation modes, the first one at rated speed, and the second one at maximum torque speed. The atmospheric conditions were p_0=712 mm Hg =94924 Pa and T_0=290 K. So, the experimental measurements formed a base line for calibration of the measures required in the exergy analysis.

3 MATHEMATICAL MODEL

The general mathematical model is based on hypothesis, assumptions and equations from reference [9] and it was already used by article authors in [10] to report irreversibilities of a four stroke dual combustion, turbocharged and intercooled, diesel engine. The model was both extended with an in-house computational code and simplified to naturally induction case, due to the lack of turbocharging and intercooling. In the hypotheses of an ideal conventional dual combustion diesel engine cycle, the thermal agent is a perfect gas, with the properties of air, starting from ambient pressure p_0 and temperature T_0 . The main processes are: constant pressure induction starting from the opening of the intake valve r`-a, at pressure p_a, lower than p_0 starting from the opening of intake valve; adiabatic reversible compression a-c; isochoric combustion c-y; isobaric combustion y-z; adiabatic reversible expansion z-d; isochoric free exhaust d-d′ with lowering p_d to p_r; forced exhaust d′-r at constant p_r and adiabatic reversible expansion r-r′ of the residual gas.

The main cycle parameters are $\varepsilon=V_a/V_c$, $\lambda=p_y/p_c$, $\rho=V_z/V_c$ and the relative pressure losses of induction, $\psi_a=\Delta p_a/p_0$ and exhaust $\psi_e=\Delta p_r/p_r$. During the cycle, the thermal engine continuously evolves in steady state operation modes, the mass and energy were conserved, neglecting the variation of specific heats with temperature, the blow-by gas loss and the chemical energy of incomplete combustion products.

As input data, the following values were used: the gas constant R=287 J/kgK, the mean adiabatic exponent k=1.3, the lower heating value of the fuel, H_i=42.5 MJ/kg and the minimum stoichiometric mass of air for total combustion of 1kg of fuel, L_0=14.5 kg air/kg fuel.

A source of irreversibility is pressure loss due to gas throttling at the passing through small area sections of valves and ports within cylinder head. It was considered the global pressure loss coefficient ψ corresponding to gas change processes, induction and exhaust.

$$\psi = \frac{p_0 - \Delta p_a}{p_r} = \left(1-\psi_a\right)\frac{p_0}{p_r} = \left(1-\psi_a\right)\left(1-\psi_e\right) \quad (1)$$

The engine volumetric efficiency, η_V can be expressed in function of pressure loss coefficients:

$$\eta_V = \left(1-\psi_a\right)\frac{\varepsilon-\left(\frac{1}{\psi}\right)^{1/k}}{\varepsilon-1} \quad (2)$$

Pressure loss coefficient ψ was calculated assuming that $\psi_a=\psi_e$ and for η_V known from previous calculations validated by measurements. In order to define the cycle parameters, it was used q_{ar} , the sum of combustion heats reported to cylinder displacement, then it could be calculated corresponding to pressure ratio in isochoric combustion, λ, and volume ratio in isobaric combustion, ρ, according to assumptions from reference [9].

$$q_{ar} = \eta_V \frac{\rho_0}{\alpha L_0} H_i \quad (3)$$

with intake air density:

$$\rho_0 = \frac{p_0}{RT_0} \quad (4)$$

$$\lambda=\frac{p_z}{p_0\varepsilon^k\left(1-\psi_a\right)} \tag{5}$$

$$\rho = 1+\frac{k-1}{k}\left(\frac{\varepsilon-1}{p_{max}}q_{ar}-\frac{1}{k-1}\cdot\frac{\lambda-1}{\lambda}\right) \tag{6}$$

So, the cycle temperatures could be calculated starting from T_0 and using only five parameters, with induction and exhaust temperatures yielding from equation (7) and (8).

$$\frac{T_a}{T_0}=\frac{\varepsilon-\left(\frac{1}{\rho}\right)\left(\frac{1}{\psi\lambda}\right)^{\frac{1}{k}}}{\varepsilon-\left(\frac{1}{\psi}\right)^{\frac{1}{k}}} \tag{7}$$

$$\frac{T_r}{T_a}=\frac{\rho(\psi\lambda)^{\frac{1}{k}}}{\psi} \tag{8}$$

Residual gas fraction γ_R is considered, as well as the exhaust duct temperature at the end of exhaust, T_e, which intermediates T_d and T_r.

$$\gamma_R=\frac{1}{\varepsilon\rho(\psi\lambda)^{\frac{1}{k}}-1} \tag{9}$$

$$\frac{T_e}{T_0}=\frac{1}{k}\cdot\frac{\lambda\rho^k+\left(\frac{1}{\psi}\right)\left(k\frac{\varepsilon-1}{\varepsilon}-1\right)}{1-\left(\frac{1}{\varepsilon}\right)\left(\frac{1}{\psi}\right)^{\frac{1}{k}}} \tag{10}$$

In order to describe the dual combustion process, a mean temperature of combustion processes was defined, T_{mar} :

$$T_{mar}=T_a\varepsilon^{k-1}\frac{(\lambda-1)+k\lambda(\rho-1))}{\ln(\lambda\rho^k)} \tag{11}$$

The input data for the two operation modes are presented in Table 2.

The mean indicated pressure is an indicator of engine efficiency and can be calculated as a sum of indicated mechanical works of each cycle process, divided by the volume of the stroke, V_s.

Table 2

Main input data

Speed [rpm]	**2800**	**1600**
Excess air coef. α	1.72	1.6
p_{max} [bar]	70	67
η_V	0.8	0.86
$\psi_a=\psi_e$	0.18	0.12
γ_R	0.029	0.025

The formulas for each transformation will generate the total sum, mentioning that the terms in eq. (16) and (17) are negative.

$$\frac{L_{rr'}}{V_s}=p_0\frac{1-\psi^{(1-\frac{1}{k})}}{\left(1-\psi_e\right)(k-1)(\varepsilon-1)} \tag{12}$$

$$\frac{L_{r'a}}{V_s}=p_0\left(1-\psi_a\right)\frac{\varepsilon-\left(\frac{1}{\psi}\right)^{\frac{1}{k}}}{(\varepsilon-1)} \tag{13}$$

$$\frac{L_{yz}}{V_s}=p_0\left(1-\psi_a\right)\frac{\varepsilon^k}{\varepsilon-1}\lambda(\rho-1) \tag{14}$$

$$\frac{L_{zd}}{V_s}=p_0\left(1-\psi_a\right)\frac{\varepsilon^k}{\varepsilon-1}\lambda\rho\frac{1-\left(\frac{\rho}{\varepsilon}\right)^{k-1}}{k-1} \tag{15}$$

$$\frac{L_{ac}}{V_s}=p_0\left(1-\psi_a\right)\frac{\varepsilon^k-\varepsilon}{\varepsilon-1}\cdot\frac{1}{k-1} \tag{16}$$

$$\frac{L_{dr}}{V_s}=p_0\frac{1}{\left(1-\psi_e\right)} \tag{17}$$

Finally, the total exhaust heat reported to fuel energy should be considered:

$$q_{ea}=\frac{Q_{ea}}{H_i}=\frac{k}{k-1}(1+\alpha L_0)R(T_e-T_a) \tag{18}$$

The main engine irreversibilities are those associated with the combustion process, π_{irar} exergy, loss of heat released towards environment, π_{Qea}, irreversibilities associated with exhaust process due to finite temperature difference, π_{ire}, and irreversibilities of induction and exhaust throttling, π_{la}, π_{le}.

There are also other irreversibilities which are neglected in the present work such as those of incomplete combustion, mechanical losses caused by friction, mixing of fresh air and residual gas and heat transfer in compression, expansion and gas exchange processes.

The irreversibilities have been reported to the energy of 1 kg of diesel fuel, with lower heating value H_i = 42.5 MJ/kg :

The combustion process irreversibility, π_{irar}

$$\pi_{irar}=\left(1+(1+\gamma_R)\alpha L_0\right)RT_0\frac{1}{k-1}\ln\lambda\,\rho^k \qquad (19)$$

The exhaust heat exergy loss to environment, π_{Qea}

$$\pi_{Qea}=\frac{k}{k-1}(1+\alpha L_0)RT_0\left(\frac{T_e}{T_0}-\frac{T_a}{T_0}-\ln\frac{T_e}{T_a}\right) \qquad (20)$$

The exhaust gas irreversibility, π_{ire}

$$\pi_{ire}=\frac{k}{k-1}(1+\alpha L_0)RT_0\ln\frac{T_e}{T_{d'}} \qquad (21)$$

The induction and exhaust throttling, π_{la} , π_{le}

$$\pi_{la}=\pi_{le}=(1+\alpha L_0)RT_0\ln\frac{1}{1-\psi_e} \qquad (22)$$

There is also a custom of reporting the irreversibilities to the chemical fuel exergy or availability, A_f , which is higher than H_i, thus affecting the magnitude of percentages. Their ratio depends, for hydrocarbons, on the atoms carbon number, z, and hydrogen, y, [2], according to equation (23):

$$\frac{A_f}{H_i}=(1.04224+0.01925\frac{y}{z}-\frac{0.042}{z}) \qquad (23)$$

In case of diesel fuel with L_0=14.5 kg air/kg fuel, the ratio A_f/H_i is approximately 1.06. So the following irreversibilities terms which are reported to H_i could be easily reported to A_f.

4 INTERPRETATION OF RESULTS

Based on the numerical model, several simulations have been done for rather small variations of the following parameters: T_0, p_0, combined case of T_0 and p_0, compression ratio, ε, and excess air coefficient, α, for two distinct engine speeds.

In this way, the influence of aforementioned parameters was investigated based on mean indicated pressure p_{mi} , exhaust gas heat Q_{ea} , indicated efficiency η_i , and exergy terms π_{irar} , π_{Qea} , π_{ire}.

The variations of throttling exergy terms π_{la}, π_{le}, the smallest terms, around 1% each, were not investigated because the model did not include experimental data on induction and exhaust throttling losses.

4.1 Dead state influence

Any thermodynamic system could evolve spontaneously up to the limit of its environment, to which it may reach the thermal, mechanical and chemical equilibrium or the dead state, thus meaning that it has zero available energy or exergy.

In many exergy studies, the influence of the chemical reactions between engine and environment is neglected, so the dead state is associated only with the environment temperature T_0 and pressure p_0, usually considered to be T_0 = 298.15 K and p_0 = 1.01325 bar [2].

In this work, the reference parameters are those registered during experiments. As both measures are variable, the exergy analysis can be influenced.

4.1.1. Temperature variation T_0

The reference dead state temperature within the study was 290 K, then by using the numerical model, T_0 was varied from 273 K to 310 K, keeping p_0 constant, based on experimental data.

Fig. 1 shows a very small increase of exhaust gas heat Q_{ea}, which can be directly related to the higher combustion temperatures.

There is a quasi-constant behaviour of mean indicated pressure p_{mi} , and indicated efficiency, η_i , valid for both engine speeds.

Engine exergy losses, illustrated in Fig. 2, show a slight increase in combustion irreversibilities π_{irar} and exhaust gas heat exergy loss π_{Qea} , confirmed by references [9, 11] along with a slight decrease of exhaust gas irreversibilities π_{ire} reported also in a dedicated study on dead state [12].

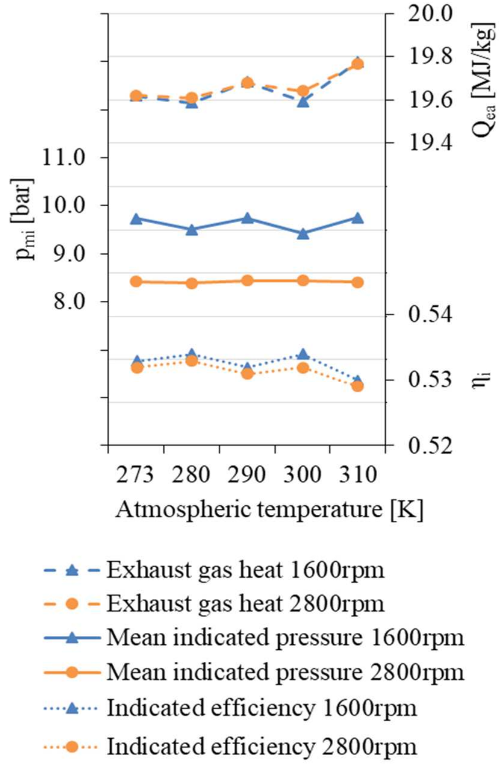

Fig. 1. Intake temperature influence over exhaust gas heat, mean indicated pressure and indicated efficiency.

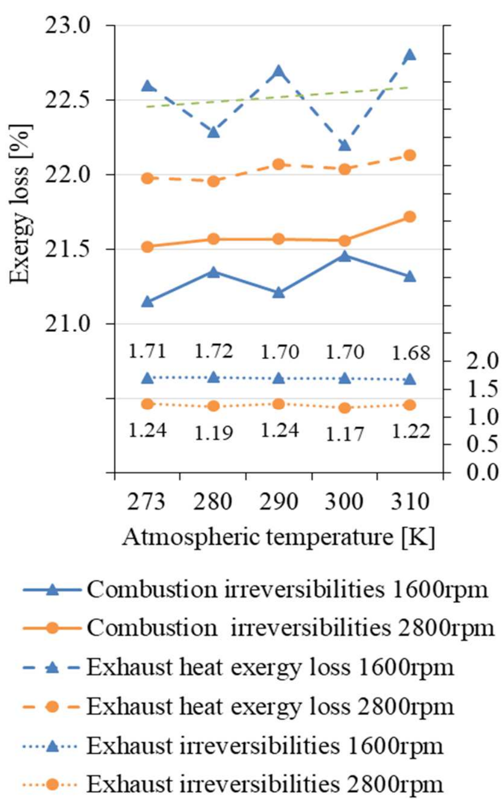

Fig. 2. Intake temperature influence over engine exergy losses.

The effect of higher speeds can be explained based on the characteristics of the heat transfer rate, which is increased by more intense gas motion (higher Reynolds numbers) and shorter heat transfer time available.

This conducts to the increase of combustion irreversibilities π_{irar} and to the decrease of exhaust gas heat exergy loss π_{Qea} [13], along with a negligible reduction of π_{ire}, similar to findings in [2].

4.1.2. Pressure variation, p_0

The atmospheric pressure was varied in the numerical model between 712 – 760 mmHg (94924 – 101325 Pa), keeping a constant T_0 from experiment.

The lower values correspond to the higher altitude at the location of the test bench.

The 1 atm point was used to evaluate the behaviour of the engine at zero altitude.

As illustrated in Fig. 3, the effect of pressure variation could yield a minimum of exhaust gas heat Q_{ea}, which corresponds to a slight increase of efficiency, especially at low speed.

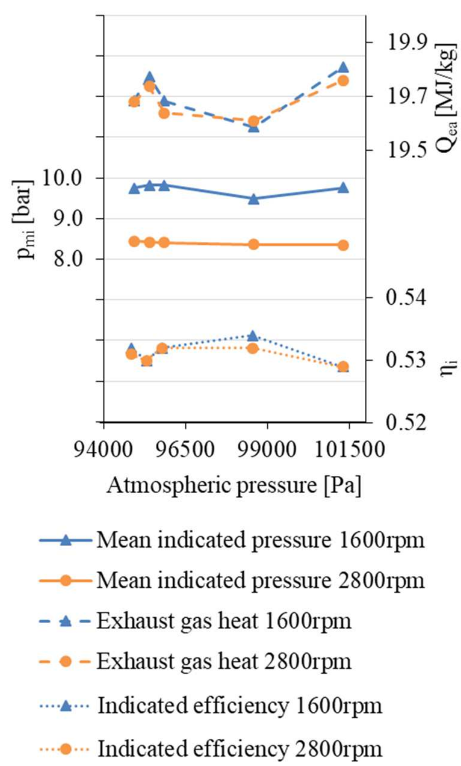

Fig. 3. Atmospheric pressure influence over exhaust gas heat, mean indicated pressure and indicated efficiency.

The influence of speed is obvious just for the mean effective pressure, being rather low for Q_{ea} and indicated efficiency η_i. The exergy terms variation presented in Fig. 4 indicates that combustion irreversibilities π_{irar} and π_{ire} have the same tendency as in case of temperature variation, with a lack of sensitivity, especially for π_{ire}. For the rated speed case, π_{Qea} drops continuously to 21.5%, while for the maximum torque speed, there is a minimum of 22% at an intermediate pressure value of 98590 Pa.

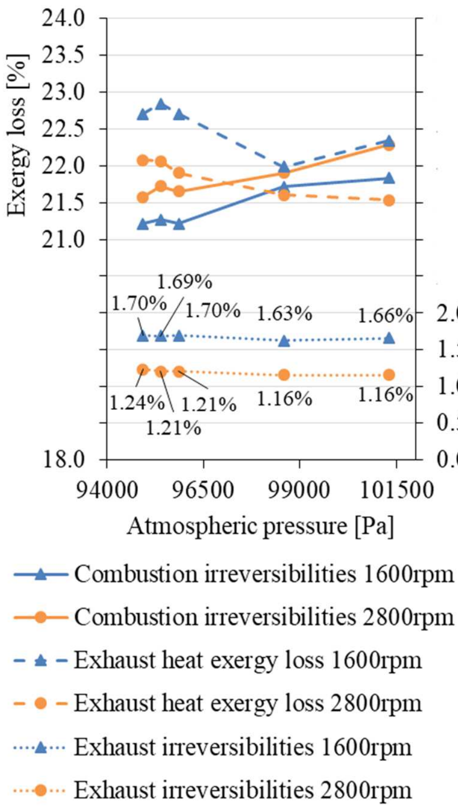

Fig. 4. Atmospheric pressure influence over engine exergy losses.

4.1.3. Simultaneous T_0 and p_0 variation

The summing effect of both factors, T_0 and p_0 can be assessed through the ratio (p_0/T_0), which is proportional to induction air density according to eq. (4). Extreme values result from (p_0, T_0) couples, such as (p_{min}, T_{max}) and (p_{max}, T_{min}), defining the air density range.

Fig. 5 depicts the influence of air density, having a milder profile for Q_{ea} and similar ones for p_{mi} and η_i, as those presented in Fig. 3.

In Fig. 6, the trends of the exergy terms π_{irar} and π_{Qea} seem to complement one each other.

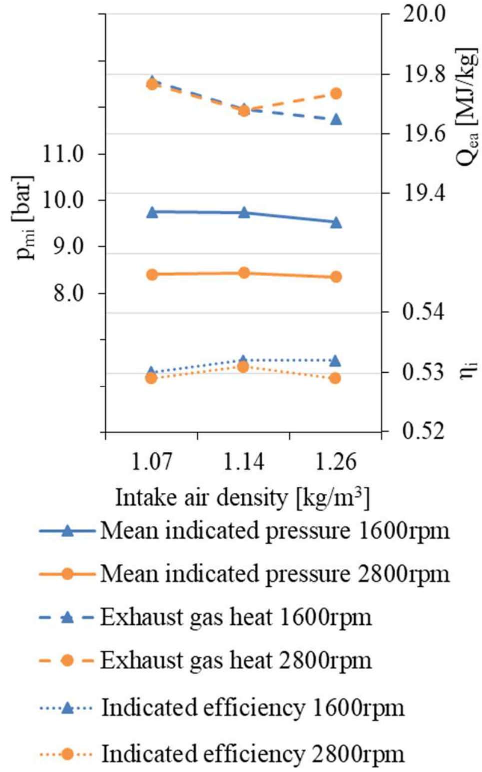

Fig. 5. Intake air density influence over exhaust gas heat, mean indicated pressure and indicated efficiency.

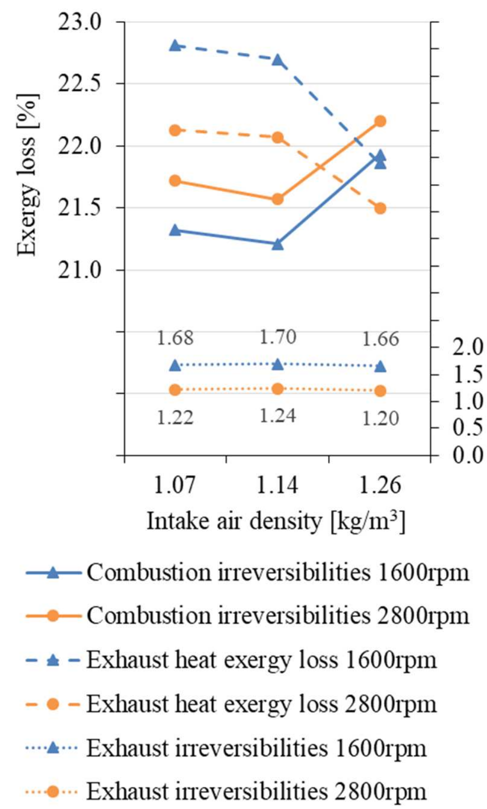

Fig. 6. Intake air density influence over engine exergy losses.

4.2 Compression ratio variation

The engine compression ratio 18.5 has been varied within 17-21.5 range in the numerical model. Fig. 7 illustrates that by compression ratio increase, there is a reduction of exhaust heat at both speeds, a rise in indicated efficiency and a slight increase of mean indicated pressure.

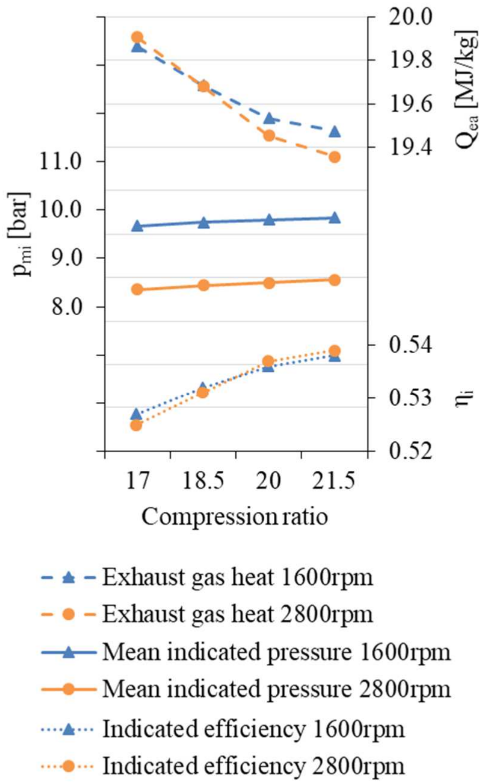

Fig. 7. Compression ratio influence over exhaust gas heat, mean indicated pressure and indicated efficiency.

According to [2], the rise of compression ratio leads to higher levels of temperature and pressure, which will reduce combustion losses, fact proven in the present work by the combustion irreversibilities profile in Fig. 8.

The exhaust heat loss and exhaust irreversibilities have a slight decrease also mentioned in reference [11].

4.3 Excess air coefficient variation

In a compression ignition engine, the excess air coefficient α is a reversed measure of load.

A higher value results in a higher concentration of oxygen in the combustion chamber, which leads to a higher thermal energy availability.

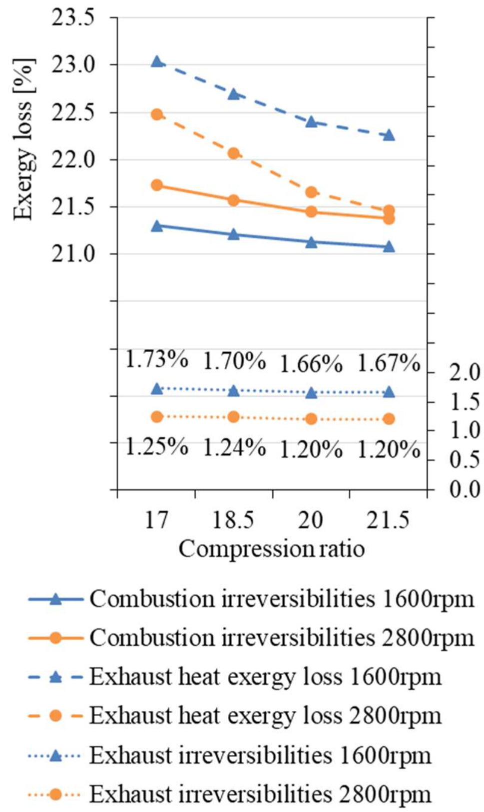

Fig. 8. Compression ratio influence over exergy losses

In the current study, the excess air coefficient α was varied between 1.3 and 2.2, whilst keeping intake temperature and pressure constant. The compression ratio has been kept as the reference, at the value of 18.5.

Fig 9 shows the influence of the increase of excess air coefficient which generates a dramatic reduction of the mean indicated pressure, with no obvious difference from one engine speed to another; it also significantly reduces exhaust gas heat, whilst increasing the engine speed leads to a lower rate of exhaust gas heat variation. The engine efficiency is directly proportional to the excess air coefficient with no significantly distinct pattern from one engine speed to another.

Fig. 10 shows the influence of the excess air coefficient over exergy losses. When excess air is raised, there is a substantial growth of combustion irreversibilities, π_{irar} , with minor differences between engine speeds, while the evolution of exhaust heat exergy loss, π_{Qea} , is reversed, being considerably reduced.

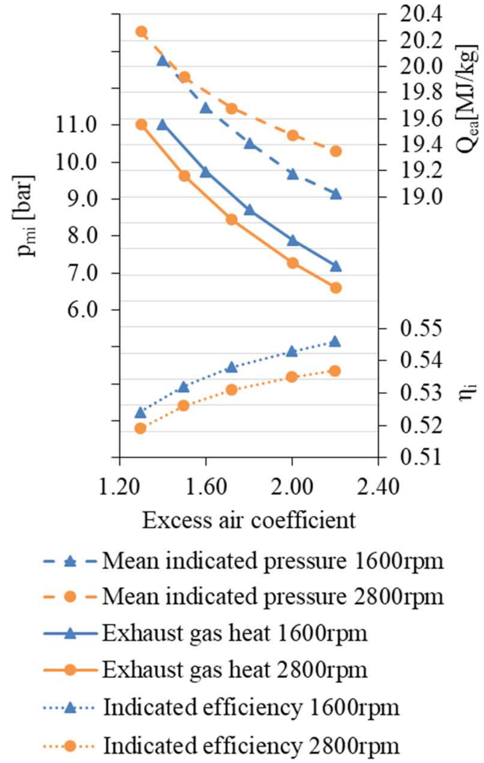

Fig. 9. Excess air coefficient influence over exhaust gas heat, mean indicated pressure and indicated efficiency.

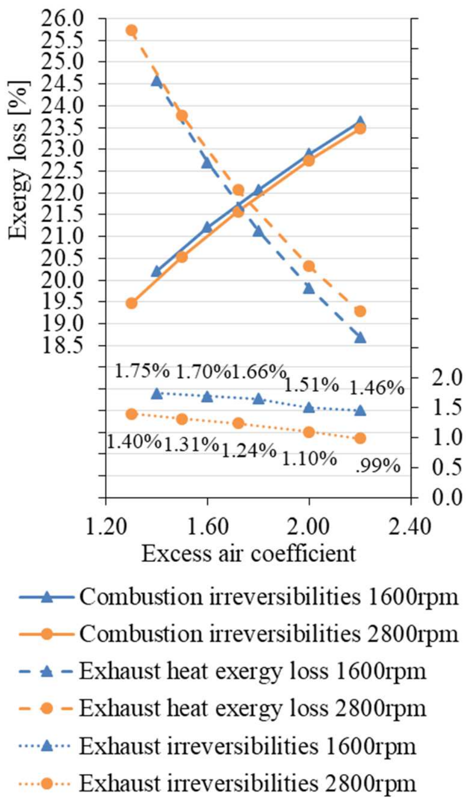

Fig. 10. Excess air coefficient influence over engine exergy losses.

The exhaust process irreversibilities are higher for lower engine speeds and present a slight decrease with increasing excess air coefficient. Several research works specifically performed on naturally-aspirated diesel engine [9, 11, 13] confirm the same trends and magnitudes of these parameters.

4.4 Opportunities and limitations

In the theoretical exergy analysis presented here, a special attention should be paid to the π_{Qea} term which has a practical value for predicting energy harvesting from the exhaust gas.

The increase of energy harvesting potential could be done by mitigation of π_{Qea} . The most efficient way to do so is by running the engine at higher excess air coefficients, which lowers π_{Qea} by 5% (Fig.10).

Other methods, such as decreasing the atmospheric temperature or increasing the air density have a more limited influence of approximately 1% reduction.

The numerical model applied has limitations due to classical assumptions on the working fluid: to behave like an ideal gas, to have a composition similar to air and to neglect the variation of gas properties with temperature.

Although the internal irreversibility of the heat transfer between the exhaust gas and atmosphere is considered, there is a limitation of the model related to neglecting of external irreversibilities caused by heat transfer during the processes, excepting combustion, which is evaluated at max. 5% of the exergy balance [11].

5 CONCLUSIONS

According to the present work, it has been confirmed that the dead state parameters, but also the excess air coefficient and compression ratio play an important role in naturally aspirated diesel engine exergy efficiency.

In a decreasing order, the exergy analysis terms π_{irar} and π_{Qea} were sensitive to excess air coefficient, p_0, compression ratio and temperature T_0 , while π_{ire} prove to be less sensitive to all aforementioned factors, excepting the engine speed.

The main contributions of the paper could be summarized as follows:

- the validation of the thermal performance model with engine experimental data;

- the study of the influence of dead state factors (temperature and pressure), compression ratio and excess air coefficient on engine efficiency and exergy terms;

- the identification of several paths to raise the potential of exhaust waste heat recovery.

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Factori care afectează termenii exergiei în motoarele diesel aspirate natural

Rezumat: Un model exergetic pentru motoare diesel aspirate natural a fost validat utilizând date experimentale din testarea pe bancul dinamometric a unui motor cu o capacitate cilindrică de 3,92 litri și patru cilindri. Pentru două moduri de funcționare la sarcină totală, prin rularea motorului la turația corespunzătoare puterii nominale, respectiv la turația cuplului maxim, a fost investigată influența presiunii atmosferice, a temperaturii atmosferice, inclusiv a raportului de comprimare şi a coeficientului excesului de aer asupra principalilor parametri ai bilanțului exergetic. Rezultatele indică metodele prin care se poate reduce distrugerea exergiei optimizând parametrii de funcționare, cât și soluțiile de proiectare a motoarelor termice.

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