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Experimental investigation of the Exhaust Gas Recirculation effects on irreversibility and Brake Specific Fuel Consumption of indirect injection diesel engines

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ABSTRACT

An experimental study was carried out to investigate the effect of using Exhaust Gas Recirculation (EGR) on various exergy terms of an IDI diesel engine cylinder. In this paper also the effectiveness of total in-cylinder irreversibility on Brake Specific Fuel Consumption (BSFC) in a diesel engine is investigated. To serve this aim an exergy analysis is conducted on the engine cylinder which provides all the availability terms by which the evaluation of in-cylinder irreversibilities is possible. The availability terms including heat transfer, inlet and exhaust gases and work output are presented during the engine operation at different load and speeds. To clarify the effect of using EGR in each case, EGR is introduced to the cylinder at various ratios during the tests. Finally, the dependence of total in-cylinder irreversibility and engine BSFC at particular engine operating conditions is introduced and the variations are compared. The results show that using EGR mostly increases the total in-cylinder irreversibility mainly due to extension of the flame region which reduces maximum combustion temperature. Also, the results revealed that the variations of the total in-cylinder irreversibility and engine BSFC follow the same trend especially at high load conditions.

1. Introduction

Diesel engines naturally benefit from high thermal efficiencies as a consequence of lean combustion and rather high compression ratio. Their high compression ratio can provide appropriate conditions required for auto-ignition. High flame temperature is a predominant issue in diesel engines which originates from the non-homogeneous nature of diesel combustion caused by locally stoichiometric air to fuel ratios [1]. As a consequence of high temperature of diesel combustion, available oxygen and nitrogen from inlet combine and form nitrogen oxides like Nitric Oxide (NO) and Nitrogen dioxide (NO_2) which are categorized as NO_x emissions [2]. Also abundant oxygen presented during diesel combustion greatly contributes to the NO_x formation [3]. In the last decades, reduction of soot and NO_x emissions from diesel engines was extensively perused by the researchers and also new stricter emission control policies like EURO-V highlight the importance of NO_x and soot emission level control. In order to meet future emission regulations, use of high EGR is known as an efficient method [4]. Increasing the EGR ratio can change the shape of heat release rate during premixed combustion which can greatly suppress NO_x formation [5]. With the increase of EGR rate in diesel engines the following effects on performance and combustion are observed.

1.1. Dilution effect

Since the EGR application decreases the concentration of oxygen in the cylinder, the fuel spray has to diffuse further to encounter sufficient O_2 to form a stoichiometric mixture suitable for combustion. This extended flammable region contains not only stoichiometric mixture but also additional amount of CO_2, H_2O and N_2. The additional quantity of these components absorbs the released heat from the combustion and combustion temperature will be lowered [5]. Also the reduced amount of oxygen decreases the oxygen partial pressure and it affects the kinetics of the elementary NO_x formation reaction [6].

1.2. Ignition delay effect

The existence of diluents such as CO_2 and H_2O causes an increase in ignition delay and changes the location of start of combustion. As a consequence, the whole combustion process shifts further toward the expansion stroke. This causes in the products of combustion to be less exposed to high temperature conditions and accordingly less nitrogen oxides formation [7].

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Nomenclature

A	availability/exergy
T	absolute temperature
Q	heat
W	work
P	pressure
V	volume
m	mass
b	flow availability
I	irreversibility
h	enthalpy
s	entropy
a	specific availability
LHV	lower heat value
Φ	fuel–air equivalence ratio
μ	chemical potential

Subscripts

cv	control volume
0	restricted dead state
in	intake
ex	exhaust
f	fuel
L	loss
W	work
ch	chemical
tm	thermo mechanical
fch	fuel chemical availability
cyl	cylinder

1.3. Chemical effect

The recirculated CO_2 and water vapor from exhaust gases dissociate at the presence of high temperature during combustion period which can modify combustion temperature and NO_x formation. Particularly, the endothermic dissociation process of H_2O and CO_2 reduces the flame temperature [6].

1.4. Thermal effect

According to the higher heat capacity of CO_2 and H_2O contained in EGR in comparison with O_2 and N_2 which are normally a part of inlet air, the overall heat capacity of the in-cylinder mixture will be increased which means less flame temperature [6].

Thermodynamic analysis of the real engine cycles can be used for examining the engine performance and the engine response to various parameters. On the other side, it is quite clear that the traditional first law of thermodynamics cannot solely give the engineers best insight to the processes. For a better analysis of the engine performance and detecting the inefficiencies associated with different processes, second law analysis can be applied. The fundamental concept in the second law-based analysis is 'availability' (or exergy) [8]. Kenneth Wark [9] defined potential work (availability) as followed; "The work potential of a given quantity of energy is defined as the maximum possible useful work that can be obtained from the energy in a given environment". Many investigations in literature in the past decades (especially during last 20 years) are concerned with second law analysis of internal combustion engines [8]. Dunbar and Lior [10] and Som and Datta [11] conducted a detailed research about the irreversibility in combustion process. They found that, chemical reactions and transport phenomena of the species are the source of irreversibility in combustion process. They implied that the most significant role in the exergy destruction in combustion is owned by internal thermal energy exchange [11]. Som and Datta claimed that the most important way of limiting the exergy destruction in combustion is reducing irreversibility during heat conduction due to internal mixing. They also expressed that to reduce heat conduction because of internal mixing, the combustion process should be controlled by using air preheating so that less temperature gradient within flammable region occurs [11]. Application of the second law analysis to the internal combustion engines is widely discussed in the literature. For instance, Caton [12] carried out an analytical examination to study the exergy destruction regarding internal combustion engines and he investigated the effects of temperature, pressure, and equivalence ratio on availability destruction considering octane–air mixtures. Caton concluded that generally the operation of the engine under high temperature conditions can restrict the combustion irreversibility due to more reaction rate of available fuel and higher fuel conversion efficiency. Also Rakopoulos et al. [13] applied an exergy analysis to investigate the effect of speed and load on the availability balance and irreversibility production on a heavy duty diesel engine. They concluded that the irreversibility of combustion and total irreversibility decrease with increasing engine load. Also, they declared that increasing the engine speed at constant load leads to increasing total irreversibility due to higher amount of air flow which results in lower equivalence ratio. The combustion cyclic variation in internal combustion engines has an important role in restricting the thermal efficiency especially in lean burn engines [14]. Because of the significance of cycle to cycle combustion variation (CCV), this phenomenon is also considered in analyzing the engine irreversibilities. For example Green et al. [15] conducted an experimental investigation on CCV parameter to study the effect of heat release rate on time irreversibility in a lean burn SI engine. They employed linear Gaussian random process and noisy nonlinear dynamical process and found out that nonlinear dynamical process can describe time irreversibility more properly. Rakopoulos and Giakoumis [8] surveyed the publications available in the literature concerning the application of the second law of thermodynamics analysis on internal combustion engines. They claimed in the review that the most important factor which produces irreversibility in internal combustion engines is the combustion process. Other irreversibility producer factors are viscous dissipation, turbulence, inlet valve throttling and mixing of the incoming air or air–fuel mixture with the cylinder residuals.

In this experimental study the engine cylinders are considered as a control volume and the effects of EGR on various exergy terms of an IDI naturally aspirated diesel engine are discussed. Also the influence of total in-cylinder irreversibility on engine BSFC is studied.

2. Test rig and experimental procedure

2.1. Engine specifications

The following tests are conducted in the author's laboratory on a Perkins model 4.108 naturally aspirated, four cylinders and IDI swirl pre-chamber type diesel engine. An external EGR system is fitted to the engine in which the rate of recirculated exhaust can be controlled by a gate-valve. An oil cooling heat exchanger has been utilized in order to prevent the lubricating oil from exceeding critical temperature and the cooling water is also circulated with an in-line pump. A surge tank and orifice system is used to measure the mass flow rate of inlet air to the engine. The power output of the

test engine was measured by a hydraulic DDX Henan & Frodo dynamometer. The engine specifications are given in Table 1. A computer interface unit is provided to measure the temperature of inlet air at the orifice, intake mixture and the exhaust gases by utilizing K-type thermocouples. Also the pressure of the exhaust after EGR branch was recorded using Bourdon pressure gauge. The accuracy of measurement devices is provided in Table 2. Scheme of the test bed is shown in Fig. 1.

2.2. Test procedure

The test is conducted in three engine speeds of 1500, 2000 and 3000 rpm. Four engine loads of 25, 50, 75 and 100 percent of maximum achievable torque in each speed were considered. Once the percentage of load at each speed without using EGR is applied to the engine, the fuel pump rack position was kept unchanged and then the EGR ratios were set up by using the EGR valve. In each test, four EGR mass ratios of 0, 10, 20 and 30 percent were investigated. It should also be noted that when the EGR is introduced, the engine load was slightly readjusted by the dynamometer to achieve the specified engine speed. For each case, the results of similar tests were compared and showed a good agreement which can guarantee the repeatability of experiments.

EGR rate is calculated as follows [5]:

$$\mathrm{EGR}(\%) = \frac{\dot{m}_{\mathrm{EGR}}}{\dot{m}_{\mathrm{EGR}} + \dot{m}_{\mathrm{aEGR}} \times 100} \tag{1}$$

where $\dot{m}_{\mathrm{EGR}}$ is the mass flow rate of EGR and $\dot{m}_{\mathrm{aEGR}}$ is the mass flow rate of fresh air. In order to determine how far the EGR valve should be opened to achieve a desirable EGR mass ratio, different EGR rates were extracted from a simple computer code based on the equation of gas state and the method of trial and error. The code can estimate the air tank orifice pressure drop at a specified EGR rate by taking the engine speed, ambient conditions and intake air properties at orifice into consideration. The determination of the code inaccuracy was performed and it was lower than 5%. The program flowchart is illustrated in Appendix A.

3. General availability balance equation

For an open system experiencing mass exchange with the surrounding environment, the following equation for the availability on a time basis exists [6]:

$$\frac{\mathrm{d}A_{\mathrm{cv}}}{\mathrm{d}t} = \int_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j - \left(W_{\mathrm{cv}} - P_0\frac{\mathrm{d}V_{\mathrm{cv}}}{\mathrm{d}t}\right) + \sum_{\mathrm{in}} \dot{m}_{\mathrm{in}} b_{\mathrm{in}} - \sum_{\mathrm{ex}} \dot{m}_{\mathrm{ex}} b_{\mathrm{ex}} - \dot{I} \tag{2}$$

The above-stated terms have the following meanings:

1. $\mathrm{d}A_{\mathrm{cv}}/\mathrm{d}t$: rate of change of non-flow exergy of control volume (i.e. cylinder, each manifold, etc.) availability;
2. $\int(1 - T_0/T_j)\dot{Q}_j$: availability term for heat transfer, where $(1 - T_0/T)$ is the efficiency of the ideal Carnot cycle working between the same temperature levels, as the process in study; and $\dot{Q}_j$ is the time rate of heat transfer to or from the heat source.
3. $(W_{\mathrm{cv}} - P_0(\mathrm{d}V_{\mathrm{cv}}/\mathrm{d}t))$: availability term associated with work transfer;
4. b_{in} and b_{ex}: availability terms associated with intake and exhaust of masses, respectively. b is defined as:

$$b = b^{\mathrm{tm}} + b^{\mathrm{ch}} = h - T_0 s - \sum_i x_i \mu_i^{\mathrm{o}} \tag{3}$$

5. $\dot{I}$: rate of irreversibility production inside the control volume due to combustion, throttling, mixing, heat transfer under finite temperature difference to cooler medium, etc.

Applying equation (2) to the whole engine cylinders will yield to the following equation:

$$\frac{\mathrm{d}A_{\mathrm{cyl}}}{\mathrm{d}t} = \dot{m}_{\mathrm{in}} b_{\mathrm{in}} - \dot{m}_{\mathrm{ex}} b_{\mathrm{ex}} - A_{\mathrm{W}} - A_{\mathrm{L}} + A_{\mathrm{f}} - I \tag{4}$$

Since the data were being recorded while the engine was working on the steady state, the non-flow exergy $(A_{\mathrm{cyl}}/\mathrm{d}t)$ is zero.

In equation (4) A_{w} is the rate of work shaft availability, A_{L} is the rate of heat loss availability to the cylinder walls, $\dot{m}_{\mathrm{in}} b_{\mathrm{in}}$ and $\dot{m}_{\mathrm{ex}} b_{\mathrm{ex}}$ are exergy terms of intake and exhaust gas, respectively and A_{f} is the rate of chemical availability associated with injected fuel.

Stepanov investigated available methods for estimating chemical energies and exergies of fuels are described [16]. One approximation for liquid fuels of the general type $C_zH_yO_pS_q$, applicable in internal combustion engines applications can be found in Ref. [16]:

Table 1
Specifications of diesel engine.

Engine type	4 Stroke diesel engine
Number of cylinders	4
Injection type	Indirect injection
Bore × Stroke (mm)	79.8 × 88.9
Piston displacement (cc)	1760
Compression ratio	22:1
Maximum power (kW)	28
Maximum speed (rpm)	4500

Table 2
Resolution of the measurement devices.

Device	Error (%)	Unit
Pressure gauge	1	mmHg
Thermocouple	0.01	°C
Dynamometer	1	Nm
Speed meter	0.1	rpm
Flow meter	1	L/min

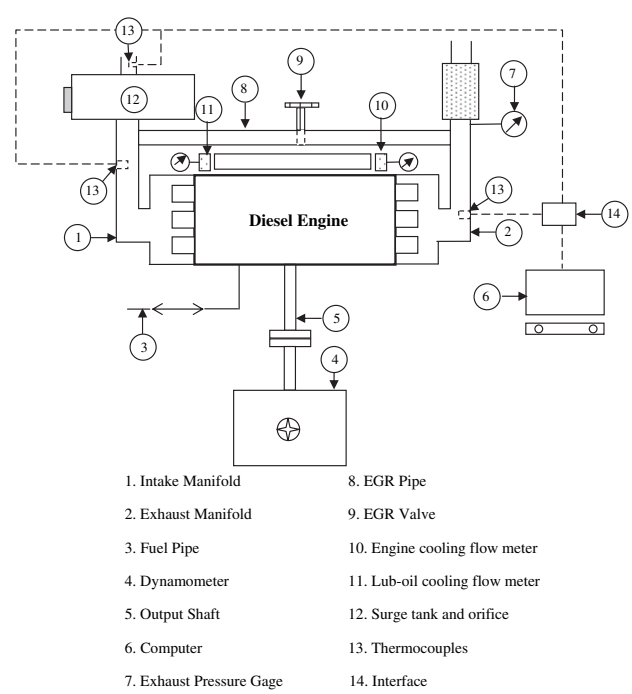


Fig. 1. Test bed components.

$$a_{fch} = LHV\left[1.0401 + 0.01728\frac{y}{z} + 0.0432\frac{p}{z} + 0.2196\frac{q}{z}\left(1 - 2.0628\frac{y}{z}\right)\right] \quad (5)$$

In this paper an approximation based on the equation (5) is used for the calculation of diesel fuel chemical availability. It should be noticed that enthalpy associated with pressure of injected fuel is usually not significant and hence ignored [6].

4. Results and discussion

Fig. 2 shows the exhaust and heat transfer exergy terms, output work and total irreversibility of the cylinder control volume for different EGR mass ratios at 1500 rpm as a percentage of fuel exergy. Because of the lean combustion during the engine operation at early stages of engine loads (Fig. 2a and b), CO_2 and other components of EGR are less concentrated. Therefore, exhaust gases mostly consist of extra air, so EGR employment acts almost like a preheater of the intake air. As a result, the fuel–air mixture at start of combustion meets the required conditions for lower ignition delay. Consequently, as it is depicted in Fig. 2a the engine experiences a slight growth in the output shaft work and exergy term of dissipated heat to cooling medium because of an increase in cylinder maximum temperature associated with shorter ignition delay. The slight growth of heat transfer exergy term and work inevitably decreases the total in-cylinder irreversibility as it is apparent in Fig. 2a. According to Table 3 at 25 percent of load, the BSFC also decreases when EGR is increasing. But at higher engine loads (Fig. 2c and d) as the equivalence ratios go up, the irreversibility terms tend to increase whereas the shaft work drops gradually when EGR increases. This behavior is also apparent in BSFC variations (Table 3 at 50, 75 and 100 percent) when BSFC increases by increasing EGR. As Figs. 3 and 4 declare at low load conditions (25 and 50 percent load) which are proportional to lower equivalence ratios, the negative effects of EGR on combustion which reduce the in-cylinder temperature, e.g. dilution effect, are almost neutralized with the effect of EGR as a preheater of the intake mixture which can increase the cycle temperature, so small changes in the output work and the total in-cylinder irreversibility are observed. Also regarding Table 3 at 2000 and 3000 rpm, the BSFC variations against EGR at low load conditions (25 and 50 percent load) are less in comparison with high load conditions.

Unlike the minor changes of irreversibility in low load conditions, this can be observed in Figs. 2–4 that in high load conditions, the in-cylinder irreversibility changes noticeably by increasing EGR ratio. The reason of this change lies within the fact that CO_2 and other EGR components are more concentrated as the engine approaches full load. Therefore, by increasing the load or equivalence ratio at constant speed (Table 3), the effect of EGR gets more significant. That means all the four effects of implementation of EGR on combustion, i.e. thermal effect, chemical effect, ignition delay promotion and dilution effect are boosted.

The increase in the in-cylinder irreversibility by increasing the EGR ratio at high loads can be explained in terms of heat transfer exergy term and work. Since the exhaust exergy term is almost unvaried, the effects of this on irreversibility change are neglected. In Figs. 2–4 it is evident that by increasing the EGR ratio, the output work declines so the output exergy regarding shaft work is reduced

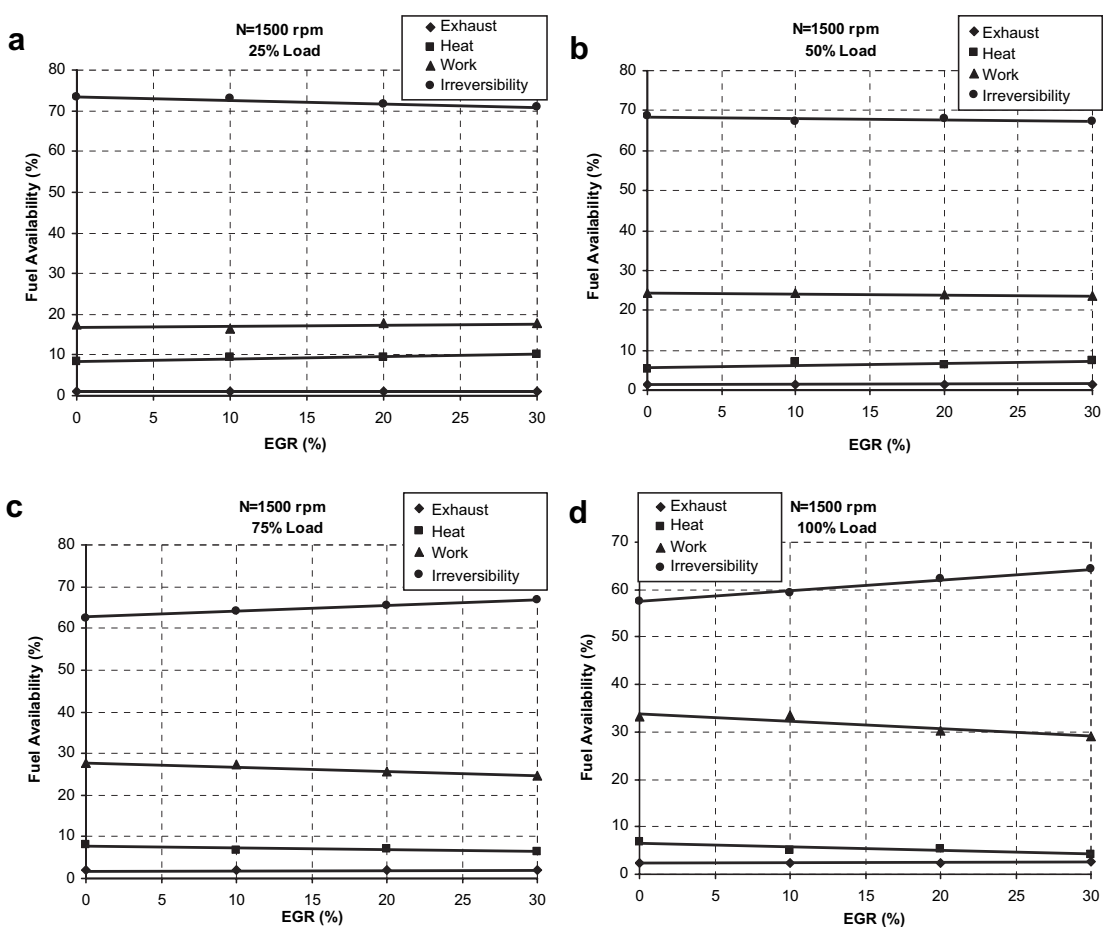


Fig. 2. Development of various exergy terms at 1500 rpm at four different loads.

Table 3
Variations of BSFC, equivalence ratio (Φ) and irreversibility in different loads and speeds.

Load (%)	EGR (%)	1500 rpm			2000 rpm			3000 rpm		
		BSFC (g/kWh)	Φ	Irrev (%fuel)	BSFC (g/kWh)	Φ	Irrev. (%fuel)	BSFC (g/kWh)	Φ	Irrev. (%fuel)
25% Load	0	448.16	0.36	73.21	443.65	0.38	75.58	525.48	0.41	78.79
	10	461.93	0.38	73.04	457.31	0.41	76.68	527.05	0.47	80.81
	20	431.51	0.39	71.45	466.34	0.42	76.14	530.20	0.50	78.94
	30	430.55	0.41	70.83	500.12	0.45	76.69	532.31	0.55	79.66
50% Load	0	317.59	0.52	68.87	320.80	0.54	70.36	353.15	0.56	71.91
	10	314.19	0.56	67.21	321.04	0.59	70.10	370.92	0.64	71.98
	20	323.27	0.59	68.17	348.61	0.62	70.50	386.71	0.69	72.41
	30	324.65	0.63	67.39	408.38	0.66	70.46	403.83	0.75	72.63
75% Load	0	282.90	0.75	62.60	275.80	0.74	68.08	314.49	0.67	66.61
	10	286.54	0.79	64.23	306.83	0.81	67.49	341.82	0.82	69.87
	20	304.90	0.84	65.33	363.85	0.87	71.50	379.66	0.89	72.35
	30	313.46	0.88	66.78	469.72	0.93	72.45	410.52	1.01	75.35
100% Load	0	184.29	0.94	57.64	178.90	0.92	63.84	186.72	0.80	60.27
	10	178.83	0.98	59.34	211.25	1.00	62.24	228.04	0.99	63.29
	20	226.62	1.06	62.21	290.44	1.08	68.08	281.65	1.08	67.98
	30	239.13	1.11	64.28	429.01	1.16	68.96	327.08	1.23	71.55

and the total irreversibility changes as well. This work reduction mainly occurs as a result of the reduction of in-cylinder temperature and simultaneously the cylinder pressure of the cycle. The drop in the temperature of the cycle can be interpreted through three effects of EGR on combustion which are dilution effect, thermal effect and chemical effect. When the EGR is introduced to the cylinder, the flame diffuses further in the combustion chamber which is due to the dilution effect, so the extra amount of gases is involved in the pre-mixed combustion process which will result in less combustion temperature. On the other hand, the excessive H_2O and CO_2 which are results of EGR implementation increase the overall specific heat of the mixture and lead to temperature drop caused by thermal effect. Meanwhile, the endothermic dissociation of the components in the cylinder which is recognized as chemical effect reduces the temperature of combustion additionally. Moreover, it is understood from Table 3 that by increasing the EGR at constant speed and load, the equivalence ratio increases. Considering the unchanged amount of injected fuel at each load, it is revealed that the intake air mass

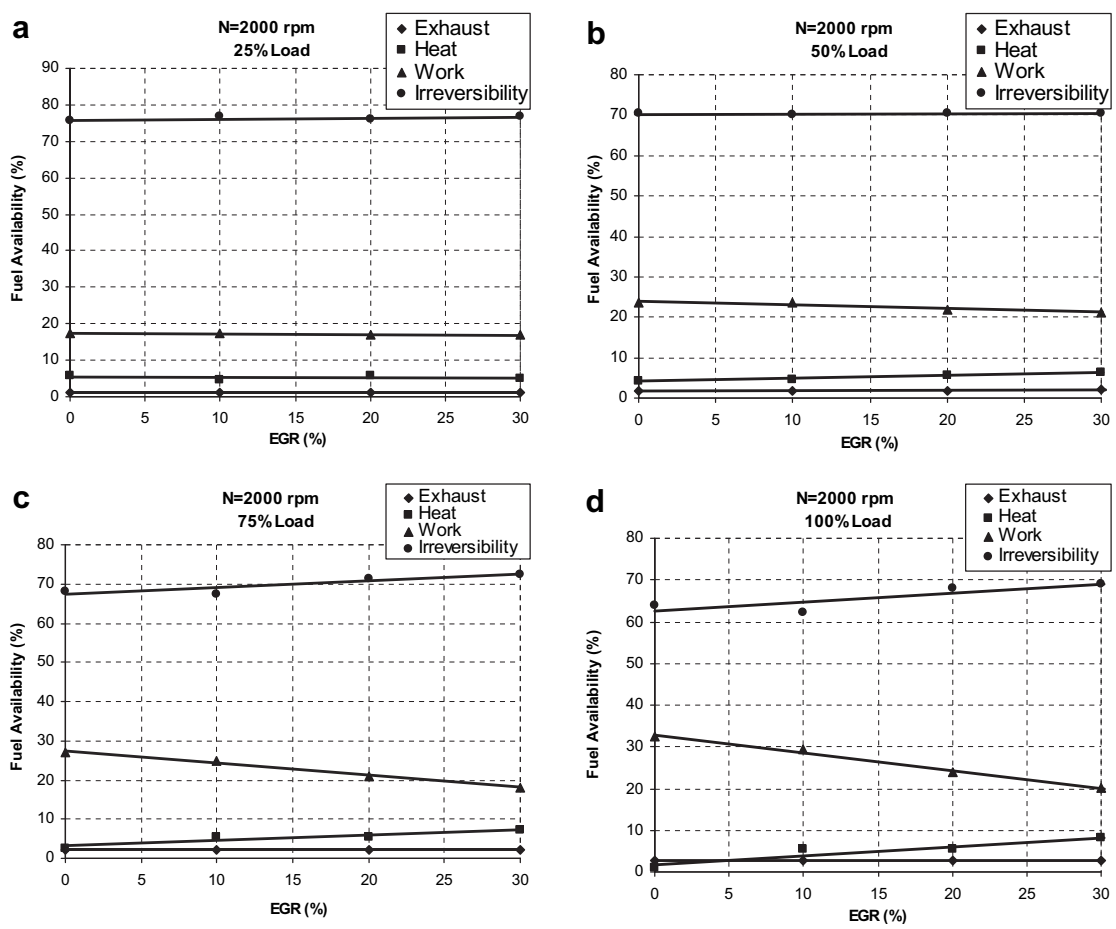


Fig. 3. Development of various exergy terms at 2000 rpm at four different loads.

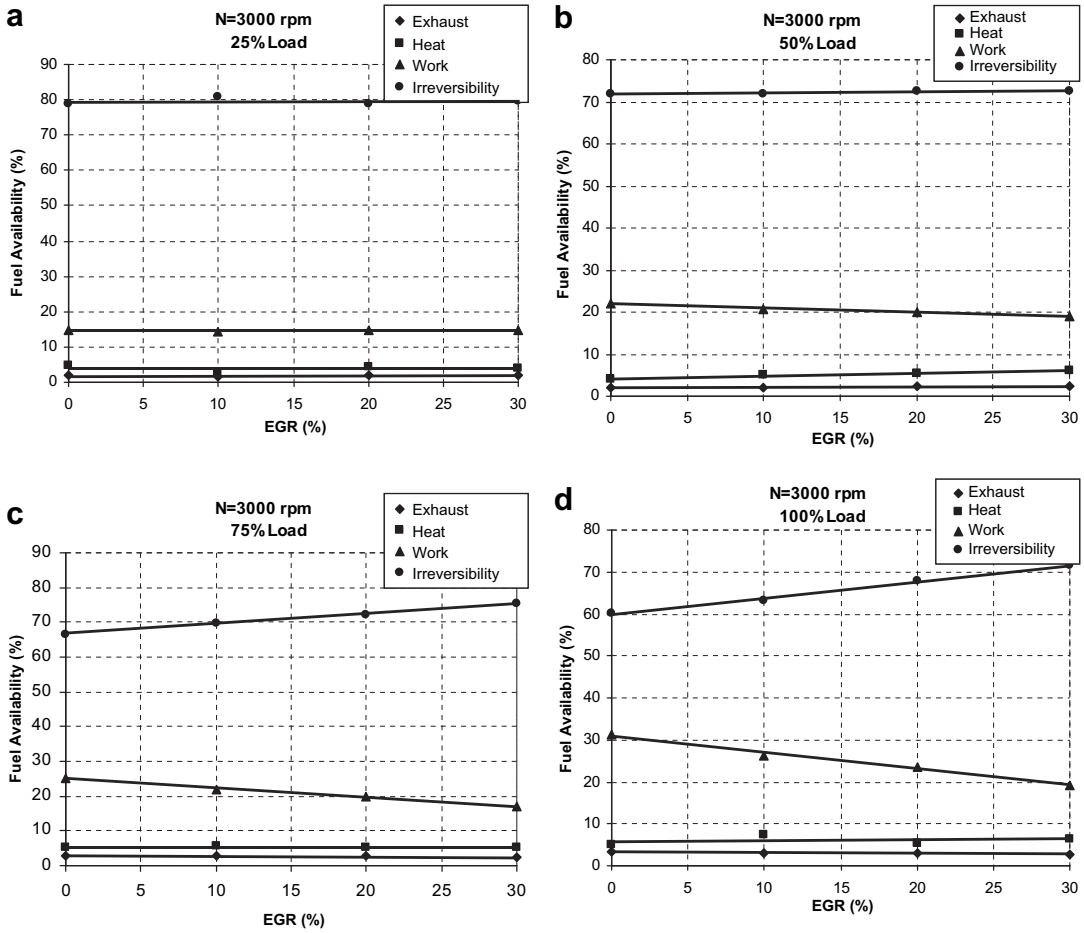


Fig. 4. Development of various exergy terms at 3000 rpm at four different loads.

flow reduces because of EGR implementation and this means that the volumetric efficiency drops. So the reduced volumetric efficiency can also be known as a factor of output work decrease. Table 3 also confirms the increase in BSFC by increasing EGR ratio while the engine is operating at high load in all three speeds.

When the change of heat transfer exergy terms by EGR induction is concerned, it should be explained that the heat transfer exergy is particularly influenced by two factors, reduction of in-cylinder temperature and promotion of combustion retard. The former reduces the heat rejection rate from the cylinder and the latter increases it. The interactions between these two specify the variation of heat transfer exergy term as the EGR ratio changes. It is observed in Table 3 that at constant EGR ratio and percentage of load, as the engine speed is increasing, the fuel–air equivalence ratio increases, so the effect of EGR on decreasing the maximum temperature is more distinct. On the other side, by increasing the engine speed, the crank angles which are involved in combustion retard are increasing. These two opposite behaviors make the heat exergy terms not to change remarkably at Figs. 2c, d and 4c, d. But in 2000 rpm at high loads (Fig. 3c and d), there is a significant growth in heat exergy terms which implies that one factor of heat rejection overcomes the other one. Since the brake mean effective pressure (BMEP) at 2000 rpm is higher in comparison with 1500 and 3000 rpm (as Fig. 5 shows), the temperature of the cycle is less reduced by EGR in this speed. So the heat rejection term at 2000 rpm increases as the EGR is induced more.

Overall effects of heat transfer exergy terms and output shaft work show that by increasing EGR at high loads in each speed, the amount of the total in-cylinder irreversibility increases. Also, it is perceived from Table 3 that by increasing EGR at high load conditions, the engine BSFC increases similarly. According to earlier discussions, it can be inferred that when the BSFC of diesel engine increases by introducing more EGR, the total in-cylinder irreversibility increases too.

Table 3 clarifies the effect of engine load on the total irreversibility term which approves what has been reported by Rakopoulos and Giakoumis [13]. According to their work, when the variation with load is considered, one can observe that the amount of irreversibilities decreases with increasing load. This is due to the fact that

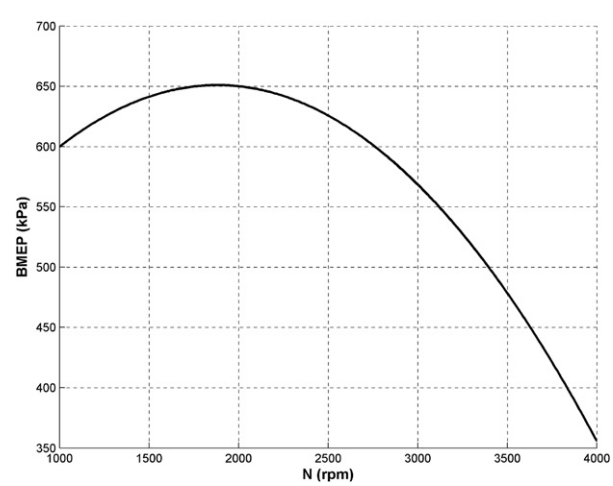


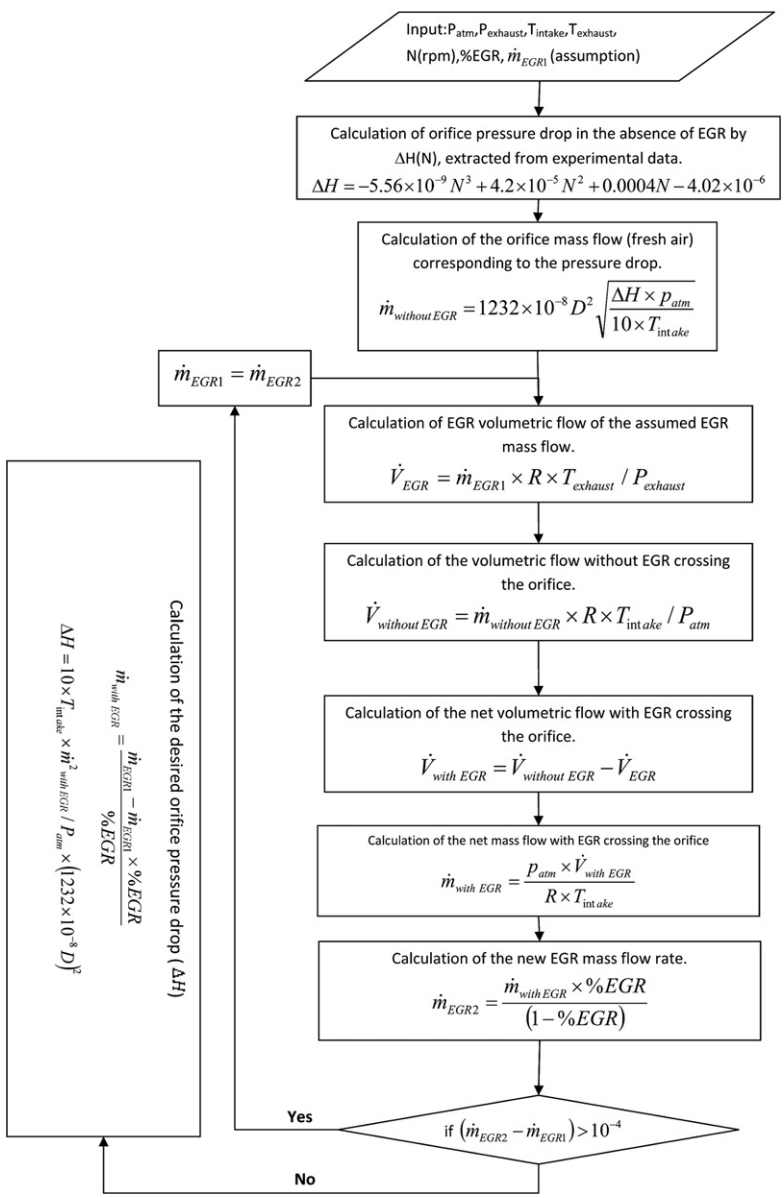
Fig. 5. Engine characteristic curve of BMEP.

combustion irreversibilities fall with increasing load. The amount of combustion irreversibilities significantly depends on the fuel–air equivalence ratio. Higher values of fuel–air equivalence ratio cause higher temperatures in the cylinder, i.e. less degradation of the fuel chemical availability when transferred to the products and less mixing of the products with air during combustion and expansion. This is the dominant part of the in-cylinder irreversibility [13].

5. Conclusion

An availability study was performed in order to investigate the effects of EGR on various exergy terms in an IDI, four cylinders, four strokes naturally aspirated diesel engine of diesel engine cylinder. Also the influence of total in-cylinder irreversibility on engine BSFC was discussed. The test was conducted in three engine speeds of 1500, 2000 and 3000 rpm and in four loads of 25, 50, 75 and 100 percent of maximum achievable load. In the tests, four EGR mass ratio percentage of 0, 10, 20 and 30 were employed. Results show that at lower engine load and speed, there is a considerable decrease in total in-cylinder irreversibility due to higher output shaft work when EGR is induced. This is proved from the fact that EGR at low load conditions acts as intake preheater which increases the cycle temperature. At higher speeds and low load conditions, the positive effect of EGR as intake preheater is vanished by the negative effects of EGR which decline the cycle temperature, and total in-cylinder irreversibility almost remains unchanged. But when the EGR is implemented in high load conditions, a significant increase in total irreversibility is observed in all speeds. This is mainly caused by the work deterioration which implies the dominant effects of EGR in decreasing cycle temperature and more ignition delay. The comparison of the irreversibility and engine BSFC variations shows that, at all operating conditions, when the EGR is employed there are similarities between the trends of the cylinder total irreversibility and BSFC.

Appendix A. The flowchart of EGR ratio predictor program



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