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Effect of Using EGR on In-Cylinder Irreversibility of an IDI Diesel Engine

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Abstract

Current approaches to internal combustion engine analysis are no longer just from first law of thermodynamics' point of view. Evaluation of IC engines by second law of thermodynamics (concept of exergy) helps modifying different parts of engine that cause availability dissipation. Since EGR is widely used as an effective method for decreasing NOx as a toxic pollutant in diesel engines, this experimental investigation is carried out in order to analyze different aspects of using EGR on in-cylinder irreversibility within the engine block. It was observed that use of EGR can increase in-cylinder irreversibility especially at full load.

Keywords: Diesel engine, irreversibility, EGR

Introduction

Since diesel engines have a higher thermal efficiency comparing with SI engines, diesel inefficiencies are considered as a great significance. In this way, pollution reduction policies like Euro 3 to 5 are applied to vehicle industries [1]. Experiments show that introducing a fraction of exhaust gas to cylinder can decrease nitrogen radical complexes dramatically. This can be done by lessening oxygen concentration which leads to higher combustion delay and avoids maximum temperature which is the main factor of NOx formation [2]. Using exhaust gas recirculation in diesel engines as a way of NOx suppression has been evaluated in literature in order to remove drawbacks like smoke increase or power loss.

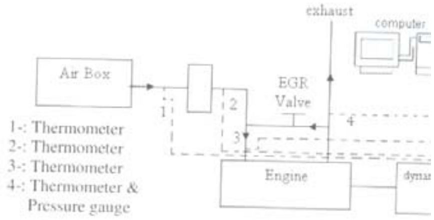
In the following research effects of using EGR on in-cylinder availability destruction during combustion are discussed and different engine experimental conditions which are involved in irreversibility appearance are tabulated.

Test Bed Setup

This experiment is done in author's lab on an IDI diesel engine in three different engine speeds of 1500, 2000 and 3000 rpm and different engine loads corresponding to 25, 50 and 75 percent of maximum achievable load in each speed. In the tests three EGR mass ratio of 0%, 10% and 20% were employed in order to investigate the effects of EGR application from lean to rich combustion conditions. In each test the EGR mass flow was being calculated via codes in MATLAB software based on constant intake volume which is drawn into the cylinder.

It should be noted that employed EGR modification is of high pressure loop type with air-cooled intercooler.

The engine test bed is demonstrated in Fig. 1.



Effects of EGR in Engine Combustion

There are three known mechanisms [3] to describe of EGR can suppress NOx in exhaust gas as a reduction of maximum cylinder temperature and pr

- The dilution effect: Decrease of in-cyli concentration, as a result of recirculated gas, yields to delay the proper mixing pr reach stoichiometric ratio fitted to start auto [2]. This delay causes bigger amount of in- gas to involve ignition and this makes th mixture cooler.
- Ignition delay: According to cylinder p diagrams of EGR equipped engines, EGR the premixed region bigger. This shifts the combustion process toward power stroke.
- Decrease of specific heat: Since exhaust gas c significant amount of H_2O and CO_2 with specific heat, the Rate of Heat Release (ROHR to decline as a consequence of heat transfer i smaller temperature difference.

Important Factors of Availability Loss within the Cy

Application of availability balance to a diesel cylinder has proved that in-cylinder irreversibi mostly because of two vital phenomenons whi combustion and heat transfer to the walls. The r irreversibility production is derived from v dissipation, turbulence, inlet valve throttling and r of the incoming air with cylinder residuals [accordance with the fact that reduction o concentration caused by using EGR, dilutes mixture in combustion chamber, local φ decrease local fuel air ratio reduces, the combustion occi lower temperature so the availability associated combustion deteriorates [5].

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Abstract
For the purpose of reducing NOx which is a serious pollutant of diesel engine caused by high temperature of diesel combustion, application of recirculated exhaust gas (EGR) is widely used in vehicle industries. Besides, it has been proven that the second law of thermodynamics (the concept of exergy) contributes to the better understanding of the engine inefficiencies and finding the sources of irreversibilities or exergy loss. In this experimental investigation, the second law analysis is applied to an EGR implemented, IDI naturally aspirated diesel engine. The aim of this study is to find how different exergy terms of the engine cylinder are affected by EGR induction. To serve this aim various exergy terms crossing the engine cylinders as a control volume are illustrated. In order to cover the main operating condition of the engine, three different speeds of 1500, 2000 and 3000 rpm and the corresponding loads of 25, 50 and 75% of maximum achievable load for each speed were tested. In each case four EGR mass ratios ranging from 0% to 20% are introduced to the engine intake to cover EGR conventions. It was concluded that using EGR mostly increases the cylinder total irreversibility mainly because of extending the premixed combustion region and consequently the decrease of maximum temperature of the cycle. This effect is more significant as engine is approaching full load.

Keywords: Diesel engine, EGR, Irreversibility.

1. Introduction

Diesel engines have inherently high thermal efficiencies, resulting from their high compression ratio and fuel lean operation [1]. The high compression ratio produces the high temperatures required to achieve auto-ignition. However, high flame temperatures predominate because locally stoichiometric air–fuel ratios prevail in such heterogeneous combustion processes [1]. Consequently, Diesel engine combustion generates large amounts of NOx because of the high flame temperature in the presence of abundant oxygen and nitrogen [2,3]. While nitric oxide (NO) and nitrogen dioxide (NO_2) are usually grouped together as NOx emissions, nitric oxide is the predominant oxide of nitrogen produced inside the engine cylinder because of the flame high temperature. The principal source of NOx is the oxidation of atmospheric (molecular) nitrogen [4]. During the past decades significant progress has been accomplished in reducing emissions of NOx and soot, but at the same time permissible emission limits from diesel engines are becoming stricter i.e. EURO-V [5]. One efficient method to control NOx in order to achieve future emissions limits is the use of rather high Exhaust Gas Recirculation (EGR) [5]. Dilution effect of EGR is a well-known mechanism which reduces the NOx in literature [6]. Due to dilution effect, heat release rates during premixed combustion, which is characterized by rapid burning and which significantly governs NOx formation, can be suppressed more efficiently [7]. Below are a couple of effects of implementation of EGR on diesel combustion which are provided in literature:

- *The dilution effect*: Decrease of inlet O_2 concentration whose principal consequences in the deceleration of the mixing between O_2 and fuel [6]. With the local O_2 concentration reduced a given amount of fuel will have to diffuse over a wider area before sufficient O_2 is encountered for a stoichiometric mixture to be formed resulting in the extension of flame region [7].Thus the gas quantity that absorbs heat release is increasing, resulting in a lower flame temperature [8,9]. Another consequence of the dilution effect is the reduction of the oxygen partial pressure and its effect on kinetics of the elementary NO formation reaction [6].
- *Ignition delay promotion:* An increase of the ignition delay (ID) with EGR rate is generally observed [10]. Higher ignition delay shifts the whole combustion toward power stroke which make the cycle retarded, leading to lower combustion temperatures [9].
- *The chemical effect:* The recirculated water vapor and CO_2 are dissociated during combustion, modifying the combustion process and the NOx formation in particular, the endothermic dissociation of H_2O results in decrease of the flame temperature [6].

- *Thermal effect:* Increase of inlet heat capacity due to higher specific of heat capacity of recirculated CO_2 and H_2O compared with O_2 and N_2 (at constant boost pressure) resulting in lower gas temperature during combustion and particularly in lower flame temperature [8,9,11].

Thermodynamic models of the real engine cycle have served as effective tools for complete analysis of engine performance and sensitivity to various operating parameters. On the other hand, it has long been understood that traditional first-law analysis, which is needed for modeling the engine processes, often fails to give the engineer the best insight into the engine’s operation [12]. In order to analyze engine performance—that is, evaluate the inefficiencies associated with the various processes—second-law analysis must be applied [12]. Kenneth Wark 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" [13]. Many studies have been published in the past few decades (the majority during the last 20 years), concerning second-law application to internal combustion engines [12]. Caton [14] carried out an investigation to study the destruction of availability or irreversibility production concerning internal combustion engines. Caton concluded that in general, the destruction of the fuel’s available energy due to the combustion process decreases for operation at higher temperatures. In addition, the effect of equivalence ratio on the destruction of availability is significant and depends on the particular operating conditions [14]. Also Dunbar et.al [15] and Som and Datta [16] conducted a detailed research about the irreversibility in combustion process. Dunbar concluded that, in almost all situations, the major source of irreversibilities is the internal thermal energy exchange associated with high temperature gradients caused by heat release in combustion reactions [15]. Som and Datta found that, the destruction of useful energy occurring in typical gaseous hydrocarbon or hydrocarbon fuel combustion is likely due to the internal thermal energy exchange (heat transfer) between the particles within the system [16]. Rakopoulos et-al [17] investigated the effect of engine speed and load on the availability balance and irreversibility production in a multi-cylinder turbocharged diesel engine. They found inlet irreversibilities and mechanical friction, in comparison, decrease with increasing engine load; the same applies to combustion and total irreversibilities and cylinder heat loss. Also the increase in speed causes an increase in mechanical friction, turbocharger, inlet and total irreversibilities, cylinder inlet air and the amount of exhaust gases available. Rakopoulos and Giakoumis's paper [12] surveys the publications available in the literature.

It is claimed in this review that the irreversibility production within the cylinder consists of combustion (dominant contribution), viscous dissipation, turbulence, inlet valve throttling and mixing of the incoming air or air–fuel mixture with the cylinder residuals [12].

2. Engine plant and testing procedure

2.1 Engine description

The following experimental tests are conducted in the author's lab on a Perkinz model 4.108 naturally aspirated, four cylinder and IDI swirl chamber type. An external EGR system is fitted to the engine in which the rate of recirculated exhaust can be controlled by a gate-valve. The engine involves an oil cooling heat exchanger in order to maintain the oil temperature constant. In the heat exchanger an in-line pump circulates the water to remove lubricating oil heat. A surge tank and orifice system is used to measure the mass flow rate of inlet air to the engine. A hydraulic DDX Henan & Frodo dynamometer is coupled to the engine. The engine specifications are given in table 1. A computer interface unit is provided to measure the temperature of inlet air at the orifice, mixture and exhaust tail utilizing K-type thermocouples while the pressure of the exhaust after EGR branch was recorded using Bourdon pressure gage. Schematic figure of the test bed is depicted in Figure 1.

Engine type	4 stroke diesel engine
Number of cylinders	4
Combustion chamber	indirect injection
Bore × stroke (mm)	79.8 × 88.9
Piston displacement (cc)	1760
Comprettion ratio	22:1
Maximum power (KW)	28
Maximum speed (rpm)	4500

Table.1-Specifications of tested diesel engine

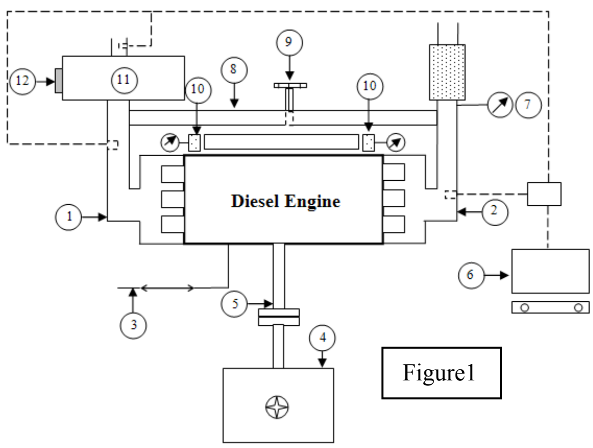

Figure1

1. Intake Manifold
2. Exhaust Manifold
3. Fuel Pipe
4. Dynamometer
5. Output Shaft
6. Computer
7. Exhaust Pressure Gage
8. EGR Pipe
9. EGR Valve
10. Flow meter
11. Air Tank
12. Orifice

2.2 Test procedure

The test is conducted in three engine speeds of 1500, 2000 and 3000 rpm. To cover the different operating conditions, four engine loads of 25, 50 and 75% of the corresponding speeds at maximum torque were considered. Once the percentage of load at the corresponding speed applied to the engine (with no EGR), the fuel pump rack position was left unchanged till the end of the test. Then the EGR ratios were set up by using the EGR valve. In each test, four EGR mass ratios of 0, 10, and 20% were applied. It should also be noted that the measured engine load usually slightly differs from the set point during EGR induction.

EGR rate is calculated as follow [7]:

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

Where m_{EGR} is the mass flow rate of EGR and m_{aEGR} is the mass flow rate of fresh air. In order to determine how far the EGR valve should be adjusted 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 trial and error. The code can estimate the orifice pressure drop at specific EGR rate in terms of the engine speed, ambient conditions and intake air properties at orifice.

2.3 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 [12]:

$$\frac{dA_{cv}}{dt} = \int_j (1-\frac{T_0}{T_j})\dot{Q}_j - \left(W_{cv} - P_0\frac{dV_{cv}}{dt}\right) + \sum_{in}\dot{m}_{in}b_{in} - \sum_{out}\dot{m}_{out}b_{out} - \dot{I} \qquad (2)$$

The above-stated terms have the following meanings:

1. $dA_{c.v.}/dt$: rate of change of control volume (i.e. cylinder, each manifold, etc) availability;

2. $\int(1-T_o/T_j)\dot{Q}_j$: availability term for heat transfer, where $(1 - T_o/T)$ is the efficiency of the ideal Carnot cycle working between the same temperature levels, as the process in study; and $\dot{Q}_j$ is the time rate of heat transfer to or from the working medium exchanged through differential surface area;

3. $(W_{c.v.} - P_0 (dV_{c.v.}/dt))$: availability term associated with work transfer;

4. b_{in}, b_{out} : availability terms associated with inflow and outflow of masses, respectively. b is defined as:

$$b = b^{tm} + b^{ch} = h - T_0 s - \sum_i x_i \mu_i^o \qquad (3)$$

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:

$$\frac{dA_{cyl}}{dt} = \dot{m}_4 b_4 - \dot{m}_5 b_5 - A_W - A_L + A_f - I \qquad (4)$$

Since the data were being recorded while the engine was working on the steady state, it is assumed that dA_{cyl}/dt=0.

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, $\dot{m}_4 b_4$ and $\dot{m}_5 b_5$ are exergy terms of intake and exhaust gas, respectively and A_f is the rate of chemical availability associated with injected fuel. Szargut and Styrylska [18], Rodriguez [19] and Stepanov [20] discuss various approximations for the chemical exergy of fossil, liqu id and gaseous fuels. One such approximation for liquid fuels of the general type $C_zH_yO_pS_q$, applicable in internal combustion engines applications can be found in Ref. [20] based on the work of Szargut and Styrylska:

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

In this paper an approximation is used for the calculation of diesel fuel chemical availability which is $a_{f.ch}$/LHV=1.069 [12]. It should be noticed that enthalpy associated with pressure of injected fuel is usually not significant and hence ignored [12].

3. Results and discussion:

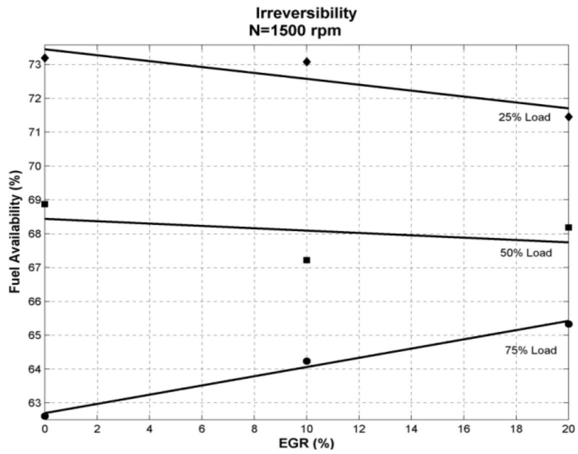

Figure2 -The response of the total irreversibility exergy term over different EGR ratios at 1500 rpm.

Fig.2 shows the response of the total irreversibility of the cylinder to the EGR induction which is introduced as the percentage of fuel exergy for different EGR mass ratios at 1500 rpm.

Because of the lean combustion during the engine operation at early stages of engine load (25% and 50% load), the CO_2 concentration in EGR is negligible and it mostly consists of extra oxygen so that EGR employment acts nearly as preheating the intake air and makes the conditions for auto-ignition more proper. Consequently, as it can be observed in fig. 2 at 25% load, the engine experiences a slight drop in the amount of produced irreversibility caused by an increase in cylinder maximum temperature associated with proper auto-ignition. But as the engine load increases, the irreversibility term tend to increase by increasing EGR. The reasons of this change which are almost prevalent in other speed and load conditions will be discussed in the following paragraphs.

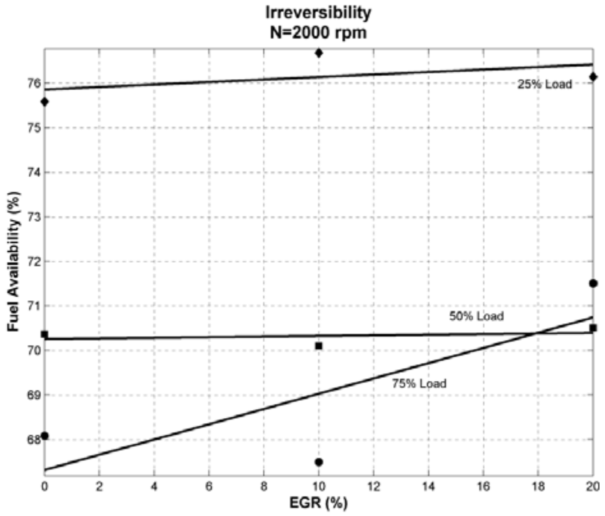

Figure3 -The response of the total irreversibility exergy term over different EGR ratios at 2000 rpm.

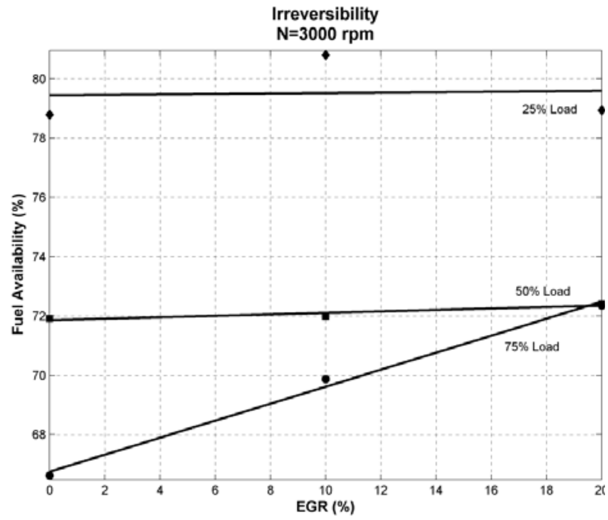

Figure4 -The response of the total irreversibility exergy term over different EGR ratios at 3000 rpm.

The reason why the effect of EGR on exergy terms differs as the engine load varies can be defined by the term, EGR effectiveness. In accordance with the fact that the amount of excess air in exhaust gas reduces as the engine approaches full load, the CO_2 is more concentrated at full rather than low or medium load. So the effect of EGR as a diluent boosts due to the higher percentage of CO_2 exists in the recirculated exhaust gas and it will enhance all the four effects of implementation of EGR on combustion that have been introduced before.

Fig. 3 and fig.4 illustrates the same terms of exergy over different EGR mass ratios. It is quite obvious in the figures that by increasing EGR ratio the irreversibility production generally increase. This change is caused by the effect of exhaust recirculation on the fuel fraction which is burnt in premixed combustion phase which was previously entitled as dilution effect. According to [12] the amount of in-cylinder irreversibility production directly increases by increasing the fuel fraction which is burnt in premixed combustion. To be more specific, longer premixed region, associated with higher EGR rate, changes the shape of heat release curve into the smoother rate of heat rejection [21] and that shall result in decreasing the maximum temperature of the cycle. It has been recognized that, in almost all situations, the major source of irreversibilities is the internal thermal energy exchange associated with high temperature gradients caused by heat release in combustion reactions [16]. It is also obvious in Figs.3 and Fig. 4 that in each speed the irreversibility increase trend usually gets sharper as the engine load increases. This behavior can be interpreted in terms of EGR effectiveness which was previously defined. The whole figures clarify the effect of engine speed and load on the irreversibility term which exactly coincides with what is reported in [17]. According to Rakopoulos paper [17], the rate of exergy loss declines as the engine load goes further due to the increase in cylinder combustion temperature (that means higher fuel-air equivalence ratio causes less degradation of the fuel chemical availability when transferred to the exhaust gases). To explain the behavior of irreversibility term when speed is considered it should be said that the reason lies in the previously mentioned dependence of equivalence ratio and irreversibility production. That means as the engine speed increases, the greater amount of air is induced thus the fuel-air equivalence ratio reduces.

4. Conclusion

An experimental study on an IDI, naturally aspirated diesel engine was carried out in order to investigate the effects of implementation of EGR on diesel performance in terms of the second law of thermodynamics criteria. Applying the governing equation of the second law to the engine cylinders as the investigated control volume revealed that employing EGR in diesel engines can affect the irreversibility production rate. It was observed that in 1500 RPM at low and medium loads, irreversibility slightly decreases by increasing EGR ratio. But when the engine is running at 2000 and 3000 RPM, using EGR mostly enhances the irreversibility production within the cylinder. Generally, it is expected that as the engine load increases, the effect of EGR on exergy terms boosts. That is caused by the increase in EGR effectiveness followed by increasing the load.

5. Nomenclature

A	availability/exergy	h	enthalpy
T	absolute temperature	s	entropy
Q	heat	a	specific availability
W	work	LHV	lower heat value
p	pressure	φ	fuel-air equivalence ratio
V	volume	c.v.	control volume
m	mass	0	restricted dead state
b	flow availability	f	fuel
I	irreversibility	ch	chemical

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