TEMPERATURE AND POLLUTION CONTROL BY GAS RECIRCULATION IN DIESEL ENGINE

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ABSTRACT

Experimental investigations on the effect of exhaust Gas recirculation (EGR) on temperature reduction in a diesel engine were embarked upon as an emission reduction technique for NO_x formation. A fraction of the engine exhaust gases was recycled and mixed with the fresh intake charge when engine was operated at elevated temperatures. A decrease in exhaust gas temperature was observed. The effect of the recyclying on other engine performance characteristics was also studied.

1.0 INTRODUCTION

The ideal combustion products from a mixture of fuel and air in the Internal combustion Engine are water, Carbondioxide and Nitrogen which are formed from a purely stochiometric mixture ratio. However, in a practical combustor reminiscent of the common internal combustion engine, more compounds are formed during combustion in the engine, some of which are toxic and hence should be eliminated almost completely if possible. These are exhausted as pollutants which may arise both as products of combustion and also as impurities in the fuel which are not burnt during the process of combustion. Examples of these are carbonmonixide (CO) resulting from insufficient air to promote complete combustion, unburnt hydrocarbon (HC) caused by quenching of flame at cylinder walls of the engine and Oxides of Nitrogen (NO_x) resulting mainly from increase in Peak temperature or overheating of the engine which

itself is as a result of poor cooling, spark advance and increased engine load. The most important engine operating variables which affect NO_x emissions are speak temperature and charge dilution with residual gas. A number of emission control techniques have been devised in an attempt to reduce emission of all three pollutants viz HC, CO and NO_x from internal combustion engine. Among the devices are stratified charge combustion, [1] prechamber combustion [2], catalytic converter and exhaust gas recirculation.

In this work, an experimental investigation is carried out on the effect of exhaust gas recirculation (EGR) on NO_x formation in a desiel engine as an emission reduction technique with respect to other varying engine performance characteristic.
A fraction of the engine exhaust (not more than 2.2%) was recycled and mixed with the intake gases. Dilution of the intake mixture with some combustion products acting as heat sink, decrease combustion temperature culminating in reduction of energy density and flame speed thus resulting in useful reduction in temperature. Unfortunately many controls which reduce the concentration of the incomplete combustion products such as unburnt hydrocarbon, carbonmonoxide and smoke cause an increase in NO_x emission without increasing other pollutants, the amount of E. G. R. should be optimised and decreased as the engine speed increases.

The problem is more difficult for diesel engine because many of these controls (particularly for NO_x) reduces the fuel economy which is a major attraction for diesel engine. However, if EGR is combined with turbocharging NO_x is reduced and more power is produced without increasing the hydrocarbon unburnt, carbonmonoxide and reduced fuel economy.

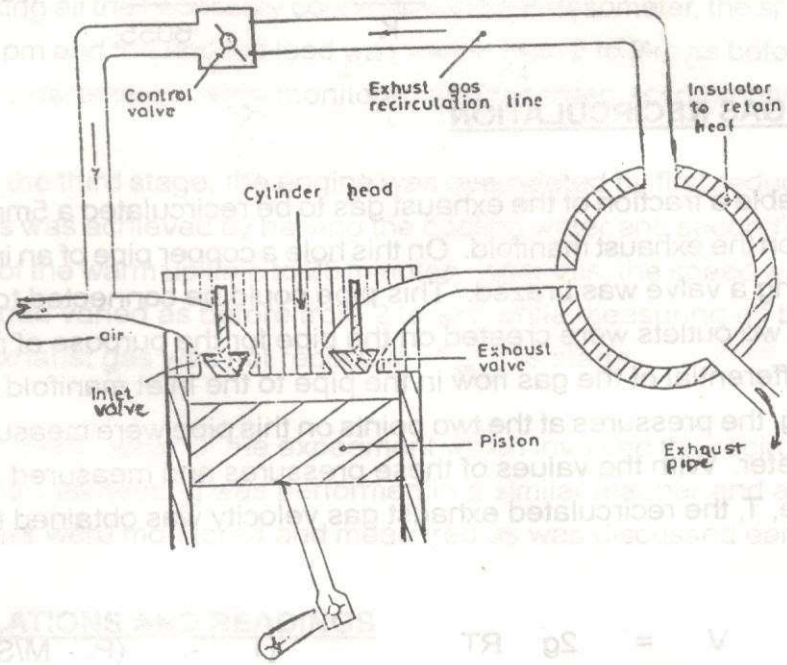
A 12% recirculation to the inlet will result in about 80% reduction NO_x formation at low load and 60% reduction at high load. [4]

The general aims of maximising power output from Internal Combustion Engine and minimizing pollutants are conflicting since combustion efficiencies are maximised at, or close to stoichiometric mixing conditions where the highest temperature is achieved.
This high temperature leads to formation of high level of pollutants. Optimisation can only be achieved by very careful control of air - fuel ratio and temperature levels throughout the system. This process of temperature moderation can easily be achieved using the EGR technique. The amount of exhaust gas recirculated is expressed as a percentage of the total mass of charge introduced into the cylinder, that is, as a percentage of the sum of the exhaust gas recirculated and fresh air-fuel mixture fed into the engine. The exhaust gas is assumed to be cooled to the inlet manifold temperature.

2.0 EXPERIMENTAL RIG

2.1 THE GARDNER DIESEL ENGINE

The garderner diesel engine is a single - cylinder four-stroke compression ignition engine. The fuel is supplied from an external tank from where it is linked to the engine by a pipe. The engine is cooled by water which circulates internally round the engine block before passing through an outlet pipe to the atmosphere. Provisions are made for the insertion of thermometers to measure both water inlet and outlet temperatures as well as the exhaust gas temperature.



The cooling water flow rate was determined by measuring the quantity of water that was collected in a bucket within a period of time. The fuel consumption rate was measured by timing the consumption of a constant fuel volume. The brake - power of the engine was obtained from the brake load applied to a hydraulic dynamometer and the engine speed. The engine speed is controlled by a hydraulic governor which is provided with the fuel injection system.

2.2 ABSORPTION HYDRAULIC DYNAMOMETER

The dynamometer is coupled to the engine crank shaft and internal series of semi-elliptical cups seperated by vanes. The rotor runs between these cups and the shafts gland are provided so that the casing forms water tight compartment. Rotation of the rotor shaft causes the water casing to circulate by centrifugal force round the elliptical orbit formed by each rotor cup, thus absorbing the entire power reaching the rotor from the engine.

A dynamometer is essentially a torque measuring device which must, by its definition, be capable of providing an adjustable and measurable torque opposing that of the engine. The absorption dynamometer converts the work done upon it by the engine into heat. To remove the heat thus generated, fresh cooling water is constantly admitted to the casing and discharged after heating. The casing is carried on tunnion bearings in such a way that it can respond freely to the torgue being absorbed and infact, is prevented from being carried round with the rotor by being connected to lever and weights. With a given speed power B. P. is obtained thus:

$$B.P = \frac{W.N}{K} = \frac{W.N}{6055} \quad (KN) \text{ -(1)}$$

2.3 EXHAUST GAS RECIRCULATION

To enable a fraction of the exhaust gas to be recirculated a 5mm diameter hole was drilled on the exhaust manifold. On this hole a copper pipe of an internal diameter 5mm carrying a valve was brazed. This pipe could be connected to the intake manifold. Two outlets were created on the pipe for the purpose of measuring the pressure differential of the gas flow in the pipe to the inlet manifold. As the engine was running, the pressures at the two points on this pipe were measured with the aid of a desometer. With the values of these pressures and measured exhaust gas temperature, T, the recirculated exhaust gas velocity was obtained thus:

$$V = 2g \quad RT \quad [1 - (P_2 \quad M/S - \quad (2)$$
$$P_1$$

3.0 EXPERIMENTAL INVESTIGATION

The engine was started and allowed to warm up for ten minutes at full speed, this warm-up time was necessary to stabilise both speed and temperature while ensuring also that the dynamometer arm was at horizontal position. The test was carried out in four stages viz.

1 Normal running of engine.
11 Normal running with exhaust gas recirculation
111 Overheating of the engine
1V Overheating of the engine with exhaust gas recirculations. Delete

In the first stage the engine was run at a constant speed of 1000 Rpm and the load applied at the dynamometer side was varied from 2 - 9kg. For each load applied, the engine was allowed to run for ten minutes at the end of which the water inlet and

outlet temperatures together with the exhaust temperatures were recorded. While the engine was running, the fuel consumption was closely monitored and time for consumption of a constant fuel volume for each load applied was also recorded.

Before embarking on the second stage, that is, exhaust gas recirculation tests, the engine was stopped and allowed to cool down for about three-hours in order to achieve the same initial conditions as the first stage run. Thereafter, the engine was started again and allowed to warm up for ten minutes, as usual while connecting the copper pipe to the inlet manifold.
After making all the necessary connections to the desometer, the speed was stabilised at 1000Rpm and the applied load was varied from 2 to 9kg as before during which all necessary parameters were monitored and recorded accordingly.

For the third stage, the engine was overheated by first reducing the water flow rate. This was achieved by halving the cooling water and secondly by extending the duration of the warm up time to 25 minutes. After this, the speed was set at 1000Rpm and the load varied as before from 2 to 9kg while measuring all the parameters as usual. Exhaust gas was not recirculated at this stage.

The fourth stage of the experiment which involved the recirculation of exhaust gas during overheating was performed in a similar manner and all the relevant parameters were monitored and measured as was discussed earlier.

4.0 CALCULATIONS AND READINGS

It is essential to illustrate by equations how some typical parameter values were obtained.

a BRAKE POWER (B. P.)

$$B.P = \frac{W \times N}{6035} \quad KW \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 3$$

W - Load applied

N - Engine speed

6035 - Conversion factor and dynamometer constant.

b TORQUE, Tq

$$Tq = \frac{B.P.}{2n\,N} \quad NM \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 4$$

c MASS OF FUEL CONSUMED PER SECOND, M_f

$$M_f = \frac{V_f \rho}{t} \quad \text{.....} \quad 5$$

V_f = Volume of fuel consumed

ρ = Density of fuel

t = Time for fuel consumption

d SPECIFIC FUEL CONSUMPTION, S.F.C.

$$S.F.C. = \frac{M_f}{B.P} \quad \text{.....} \quad 6$$

e COMBUSTION ENERGY, E_f

$$E_f = Q_f \times M_f \quad \text{.....} \quad 7$$

f BRAKE THERMAL EFFICIENCY

$$B = \frac{B.P}{E_f} \quad \text{.....} \quad 8$$

g RECIRCULATED EXHAUST GAS MASS FLOW RATE, M_e

$$M_e = \rho_e \cdot VA \quad \text{.....} \quad 9$$

where

$$V = \frac{2g \gamma RT_e}{\gamma - 1} \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

γ = 1.4

R = 287J/Kgk

A = $1.59 \times 10^{-5} m^2$

at T_e = 365K

ρ air = 0.9682 kg/m^3

h Percentage of exhaust gas recirculated was calculated to be 2.2%

This percentage of recirculated gas was left constant thro... experiment duration, since it was the maximum we could get with the copper pipe.

5.0 DISCUSSION OF RESULTS

Graphs of Temperature against Brake power shown in figures 1 to 4 have one thing in common and this is the reduction in peak temperatures at each load level on the application of the exhaust gas recirculation (EGR). A closer look reveals that the temperature reduction is more remarkable when the engine is overheated. This can be exemplified in figures 1 to 4. At a constant speed of 1000Rpm, Fig. 1, the maximum temperature recorded was 135°C and this dropped to 131°C giving a 3% reduction on EGR application. Also at the same speed when the engine was overheated, Fig. 2, the maximum peak temperature reached was 140°C and this dropped to 133°C corresponding to a 5.3% reduction in temperature on EGR application. For a constant speed of 1200 RPM, the maximum recorded temperature was 145°C which dropped to 140°C as shown in fig. 3, giving a 3.4% reduction in temperature. Whereas, with the overheating condition in Fig. 4, the maximum temperature was 150°C and this also dropped to 142°C giving a 6% reduction on EGR introduction.

Figures 5 to 8 depict that the brake thermal efficiencies were also affected by the application of EGR. For a constant speed of 1000 RPM under normal running operation, figs. 5 to 6, the highest recorded brake thermal efficiency was 25.6% and on application of EGR this dropped to 22.44%.
On overheating the maximum brake thermal efficiencies was 27.84% which also dropped to 27.00% with EGR application.
At a constant speed of 1200 RPM the maximum recorded values were 26.8% and 26.69% and these dropped to 24.36% and 24.95% for normal and overheating conditions respectively on application of EGR. These results depict that EGR is a disadvantage to the brake thermal efficiency of the engine.

In figures11 to 14 are plotted the variations of engine specific fuel consumption against the brake power for the conditions of normal engine running and EGR application. From all the curves, it is observed that the specific fuel consumption increased with EGR application. For a constant speed of 1000 RPM the maximum and minimum specific fuel consumption were 0.712kg/Kwh and 0.346 kg/Kwh and these increased to 0.788 kg/kqh NS 0.409 KG/Kwh respectively with EGR application as shown in fig. 11. The changes correspond to 11% and 18% for the same normal operation. On overheating at the same speed, fig. 12, the maximum and minimum specific fuel consumption were 0.783kg/kwwh and 0.3024kg/kwh and these increased to 0.822kg/kwh and 0.309kg/kwh corresponding to a change of 5% and 2.3%, on the

application of EGR. When the speed was increased to 1200 RPM, fig. 13, the values for maximum and minimum S.F.C. were 0.77kg/kwh and 0.3kg/kwh which increased to 0.834kg/kwh and 0.334kg/kwh corresponding to a change of 8% and 7.1% respectively on EGR application.

Whereas, when overheated the recorded values for maximum and minimum S.F.C. were 0.744kg/kwh and 0.315kg/kwh, these increased to 0.799kg/kwh and 0.343kg/kwh respectively, thus corresponding to a change of 7.4% and 9% respectively with EGR application.

Figures 9 and 10 depict similar characteristics with figs. 1 to 4 in which the peak temperatures were reduced on the application of EGR. Here, not only were temperatures reduced, the brake thermal efficiencies show corresponding reductions too.

The differences in the parameters measured on the introduction of EGR were small due to the percentage of the exhaust gas recirculated (i.e 2.2%). A greater difference should be expected if this percentage was made larger.

6.0 CONCLUSION

Introduction of a fraction of the exhaust gas to the inlet manifold leads to increase in the residual gas in the cylinder which constitutes a heat sink, thus slowing down the flame speed and prolonging the combustion process. This increase in combustion duration invariably causes more fuel to be consumed (increase in specific fuel consumption). The system, as observed, does not favour fuel economy, and causes a reduction in thermal brake efficiency of the engine.

It can therefore be concluded that EGR results in:

1 Significant reduction in temperature, hence reduction in pollution.

2 Increase in fuel consumption rate

3 Decrease in brake thermal efficiency

4 Effective change in fuel - air raito, hence reduction in fuel economy.

Finally, since the control of NO_X should be made during the combustion process and also because reduction in NO_X formation during the actual combustion process can be achieved by reduction in peak temperatures, EGR is therefore a very effective means of controlling this pollutant without increasing the other pollutants.

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NOMENCLATURE

1	B. P	-	Brake power (KN)
2	W	-	Load (Newtons)
3	N	-	Engine speed (Revolutions per minute)
4	V	-	recirculated exhaust gas velocity, M/s
5	T	-	exhaust gas temperature, ^oK
6	g	-	acceleration due to gravity, M/S^2
7	γ	-	ration of specific heats, (C_p/C_v)
8	R	-	Gas constant, KJ/Kg K
9	P	-	pressure, KN/M^2
10	Tq	-	Torque, Nm
11	M	-	Mass flow rate Kg/S
12	V_f	-	Vol of fuel consumed, $M^3/5$
13	ρ	-	density, kg/M^3
14	t	-	time, seconds
15	S.F.C.	-	specific fuel consumption, Kg/kwh
16	E_f	-	Combustion energy, KW
17	Q_f	-	Calorific Value of fuel, KJ/kg
18	B	-	Brake thermal efficiency
19	A	-	Recirculation pump area