
RESEARCH NOTE

INVESTIGATION OF THE EFFECT OF EXHAUST GAS RECIRCULATION (EGR) ON POLLUTANTS EMISSION FROM SI ENGINE EXHAUST GASES

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Abstract This paper reports the results of an experimental investigation on the effect of the EGR on the pollutants emission, especially on oxides of nitrogen, from the exhaust gases and performance of the spark ignition engine. Also along with experimental work, a mathematical model for the determination of the effect of EGR on the peak cycle temperature and for the prediction of its effect on the NO_x emission is constructed. The results of theoretical calculation show how peak temperature of theoretical cycle decreased with increasing of the recycling gas percentage. Tests were conducted on a four stroke petrol engine used in passenger cars. The experimental investigations show that NO_x emission decreases with increasing of the percentage of the recycling gas. At 9% recycling, 73% of NO_x emissions decreased at EGR valve opening 360 degrees. Brake specific fuel consumption and brake power are changed negligibly.

Key Words Emission Control, EGR, SI Engines, Pollutants

چکیده این مقاله نتایج تحقیقات تجربی را در مورد تأثیر EGR بر روی نشر آلاینده ها، بویژه اکسیدهای ازت از گازهای اگزوز عملکرد موتور اشتعال جرقه بیان می نماید. همچنین به موازات کار تجربی فوق، یک مدل ریاضی جهت تعیین تأثیر EGR در دمای نوک سیکل و برای پیش بینی تأثیر در نشر آلاینده NO_x ساخته شده است. نتایج محاسبات تئوری نشان می دهد که چگونه دمای نوک سیکل با افزایش درصد گاز بازخورانی شده، کاهش می یابد. تست ها بر روی یک موتور بنزینی چهار زمانه خودرو انجام گرفته اند. نتایج کار تجربی نشان می دهد که نشر آلاینده NO_x با افزایش درصد گاز بازخورانی شده کاهش می یابد. بعنوان مثال با ۹ درصد بازخورانی، ۷۳ درصد آلاینده NO_x کاهش می یابد. در این حالت شیر EGR، ۳۶۰ درجه باز بوده است. در این شرایط مصرف سوخت ویژه ترمزی و توان خروجی موتور به مقدار قابل اغماضی تغییر می یابند.

INTRODUCTION

In the internal combustion engines decreasing of the pollutants emission from the exhaust gases is always regarded an important problem.

IC engine produces several pollutants in the form

of NO_x , CO and HC. Among these pollutants, oxides of nitrogen (NO_x) are highly toxic. NO_x is emitted from petrol engines, diesel engines and gas turbines. The peak combustion temperature in an SI engine is of the order of 2800K and is ideal for the formation of NO_x . About 90% of the oxides of nitrogen is in the

form NO. Later, it gets oxidized to NO₂ in the atmosphere.

It is found that the contribution of NO_x from petrol engines (spark ignition) is much more compared with the compression ignition engines.

Therefore, decreased emissions from the exhaust gases of SI engines, especially NO_x emissions is a more important problem. There are two ways to control the NO_x present in an IC engine exhaust gases: by controlling the formation of NO_x by changing parameters which reduce the peak combustion temperature and by using a catalytic reactor in the exhaust system. NO_x emission can be reduced by changing the parameters such as excess air coefficient, ignition timing, compression ratio, etc.

However, exhaust gas recycling was found to be a rational method of controlling the NO_x emission at the source [1,2]. So far, the effect of EGR on the working process of the engines has been investigated mostly experimentally.

Theoretical investigation of the effect of EGR on the SI engine working cycle, especially its effect on the peak temperature is considered an important issue. In this work both theoretical and experimental investigations are considered.

THE EXPERIMENTAL SET-UP

The engine used for comparative experimental

investigation was a four stroke SI engine, typical data of which are given in Table 1.

Figure 1 shows the schematic diagram of the exhaust gas recycling system of the engine.

The test bench was equipped to measure the brake horse power generated, speed, fuel consumption, air consumption and exhaust temperature.

An EGR system comprises an exhaust accumulator, two flexible tubes, exhaust surge tank, EGR valves, and a heat exchanger. An orifice meter designed to meter the exhaust gas and connecting pipes.

A portion of the exhaust gas from the exhaust accumulator passes through the flexible tube. Then the exhaust gas enters a surge tank to dampen the pulsations of the exhaust gas that is being recycled so that the quantity of the recycling gas may be measured. Exhaust gas can be cooled in the heat exchanger for condensation of the water vapour. From surge tank the gas passes through an orifice meter. The gas flow can be controlled by an EGR valve provided on the upstream side of the orifice meter. The pressure drop across the orifice meter is measured by micro manometer. From the orifice meter, the gas enters a flexible tube under the carburetor. Tests were conducted without cooling the exhaust gas with various EGR valve openings. The speed characteristics of the engine were drawn at 100% and 50% load with EGR valve openings of 180 and 360 degrees.

TABLE 1. Typical Data of the SI Engine.

Type	Gasoline engine, four-stroke, overhead camshaft
Cylinder number	4 in line
Diameter of cylinder (mm)	89
Piston stroke (mm)	96
Displacement (Cm ³)	2389
Max Power (hp/rpm)	105/4800
Max torque (kgm/rpm)	19.1/2400
Compression ratio	8.3 to 1
Fuel system	Down draft type two barrel single Carburetor, mechanical fuel pump

MATHEMATICAL MODEL OF THE THEORETICAL CYCLE OF THE ENGINE WITH EGR

Theoretical investigations on the effect of EGR on the working cycle of the SI engine were conducted by a mathematical model which is based on the first law of thermodynamics, equation of state of ideal gas and the volume of cylinder; namely.

$$dQ = dU + pdV \quad (1)$$

$$d(pV) = d(MRT) \quad (2)$$

$$V = f(\theta) \quad (3)$$

Assumptions:

1. Compression process is adiabatic and index of adiabatic is variable.
2. Combustion process takes place at a constant

volume and heat loss is ignored.

3. Quantity of air-fuel mixture decreases with increase in percent recycling and excess air coefficient remains almost constant.

The temperature of the mixture in the cylinder at the end of intake process can be determined by the equation of total internal energy of mixture.

$$U_{\text{mix}} = U_{\text{fa}} + U_{\text{EGR}} + U_{\text{r}} \quad (4)$$

In the final form the expression for the temperature of mixture can be written as:

$$T_1 = (\alpha_{\text{fa}} c_{\text{vfa}} T_{\text{fa}} + \alpha_{\text{EGR}} c_{\text{vEGR}} T_{\text{EGR}} + \alpha_{\text{r}} c_{\text{vr}} T_{\text{r}}) / (1 + \alpha_{\text{r}}) c_{\text{vmix}} \quad (5)$$

Where

$$\alpha_{\text{fa}} = M_{\text{fa}} / M_{\text{mix}}$$

$$\alpha_{\text{EGR}} = M_{\text{EGR}} / M_{\text{mix}}$$

$$\alpha_{\text{r}} = M_{\text{r}} / M_{\text{mix}}$$

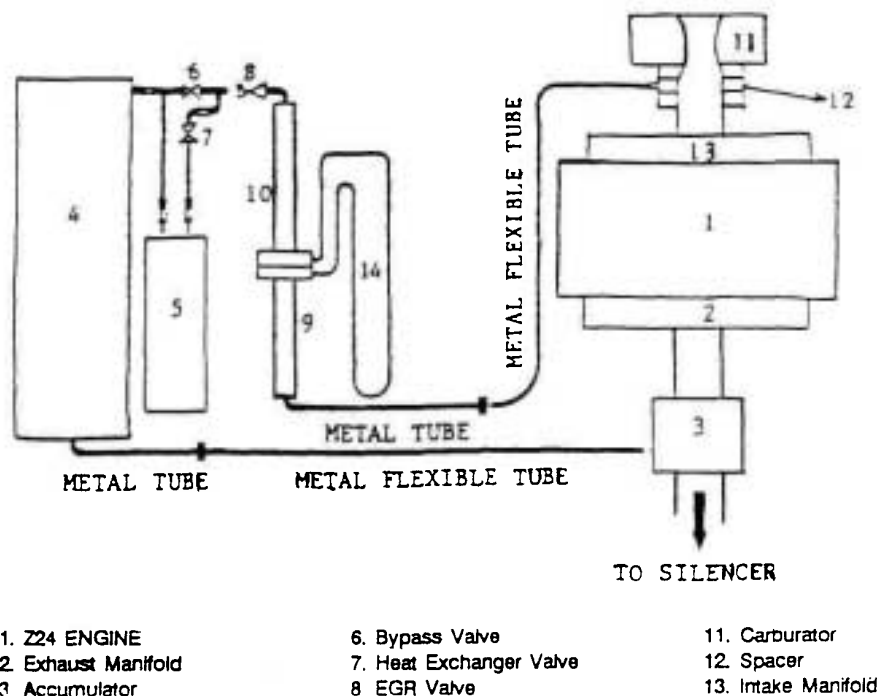


Figure 1. Schematic diagram of the exhaust gas recycling system of the engine.

The temperature of the mixture in the cylinder at the end of the compression process was determined by the equation of adiabatic process

$$TV^{k-1} = \text{const} \quad (6)$$

The peak temperature of the theoretical cycle was determined by the equation (1). In the final form, the expression for the cycle peak temperature can be written as:

$$(H - \Delta H)g_{fc}/M_{\text{mix}} + c_{v\text{mix}}T_2 = \mu c_{v\text{EG}}T_3 \quad (7)$$

The mathematical model also gives possibility of calculation of another parameters of the theoretical cycle.

DISCUSSION OF THE RESULTS

Figure 2 shows the results of the calculation of the effect of recycling on the temperature at the end of compression stroke, the cycle peak temperature and the adiabatic index.

It is seen from this figure that, with increasing the percentages of the EGR, the temperature at the end

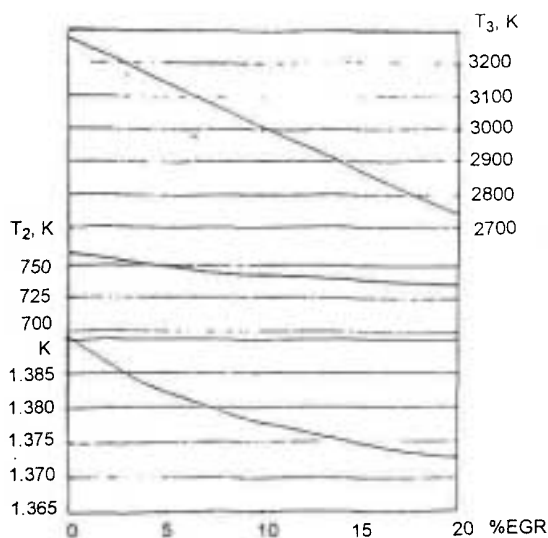


Figure 2. Results of calculation of the effect of recycling gas on the temperatures of the theoretical cycle and adiabatic index.

of compression stroke and the maximum temperature of the theoretical cycle are decreased.

For example at 10% of the recycling gas about 6.5% peak temperature is decreased.

Decreasing the maximum temperature of the cycle is caused by the decreasing of concentration of the fuel in the mixture, also by increasing specific heat with the increase in recycling gas.

Decreasing the temperature at the end of compression process is caused by decreasing adiabatic index which is due to the increasing of specific heat with the increasing of recycling gas.

In Figure 3, at 100% load brake specific fuel consumption (bsfc) increases with increase of the recycling gas. At 7% when recycling, bsfc increases to 13.3% at 4500 rpm, there is some loss of power as well.

Loss of power increases with increase in percentage of recycling, as a result of increased dilution of the mixture, which leads to improper combustion. At 9% recycling power decreases to 8.2% at 4500 rpm.

Figure 4 shows the effect of EGR on the NO_x, CO and HC emissions.

It can be seen that NO_x emission decreases with increase in percentage of recycling gas.

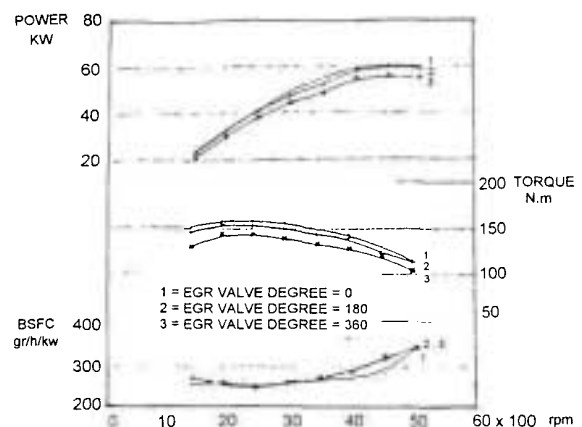


Figure 3. The effect of recycling gas on the engine performance at 100% load.

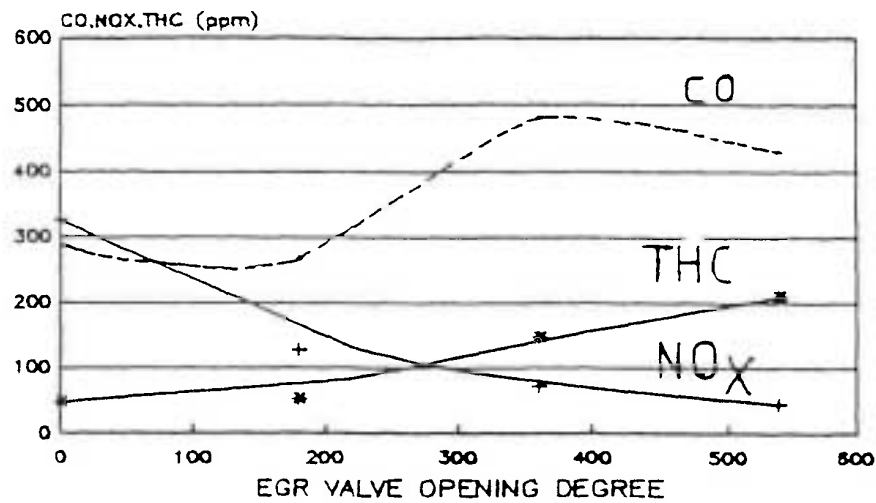


Figure 4. The effect of recycling gas on the NO_x, CO and HC emissions at 2000 rpm.

Decreasing of the NO_x emission is caused by decreasing peak combustion temperature due to late burning of the mixture, and decreasing of the oxygen combustion. At 9% recycling about 73% reduction of the NO_x emission was achieved at 4500 rpm.

Figure 4 indicates that with increasing the recycling gas, CO and HC emissions are increased. This is obviously caused by incomplete combustion of the charge due to dilution with exhaust gas.

CONCLUSION

1. Results of the calculation using mathematical model indicate that, with increasing percentage of the recycling gas, the maximum temperature of the theoretical cycle of the engine is decreased. At 10% of the recycling gas about 6.5% peak temperature is decreased.

2. EGR system gives a possibility of decreasing NO_x emission. EGR controlled 73% of oxides of nitrogen at 9% recycling.

3. Decreasing of the NO_x emission is obviously caused mainly by decreasing peak combustion temperature and decreasing of the oxygen

concentration.

4. Brake specific fuel consumption increases and power decreases with increasing of the percent of the recycling gas.

5. NO_x emission decreases with decrease in speed.

6. The excess air coefficient approximately remains constant with increasing of the EGR.

7. EGR is a promising method of decreasing the NO_x emission.

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NOMENCLATURE

Q = heat, J
 g_{fc} = cyclic mass of fuel, kg
 U = Internal energy, J
 T_3 = peak temperature, K
 P = Pressure, MPa

μ = coef. of change of moles

V = volume, m^3

M_r = mole quantity of residual gas

R = gas constants, J/mol K

M_{mix} = total mole quantity

k = adiabatic index of mixture,

T_1 = temperature of mixture, K

H = lower calorific value of fuel, J/kg

θ = crank angle, degrees

ΔH = heat in combustions, J/kg

α_{fa} = portion mole of air-fuel mixture

c_{vmix} = specific heat of mixture, J/mol K

α_r = portion mole of residual gas,

c_{vaf} = specific heat of air-fuel mixture, J/mol K

α_{EG} = portion mole of exhaust gases

c_{vr} = specific heat of residual gas, J/mol K

T_{fs} = temperature of air-fuel mix. K.

c_{vEG} = specific heat of exhaust gases, J/mol K

T_r = temperature of residual gas, K

M_{fa} = mole quantity of air-fuel mixture

T_{EGR} = temperature of exhaust gases, K

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