Design of an Experimental EGR System for a Two Cylinder Diesel Engine

Vivek Kumar Varshney
Kumar Gaurav
Sakyasingh Tripathy
Avinash Kumar Agrawal
akag@iitk.ac.in
Department of Mechanical Engineering
I. I.T. Kanpur
Kanpur 208016
INDIA

ABSTRACT:

In diesel engines, it is highly desirable to reduce the amount of NOx in the exhaust gas. NOx formation is a highly temperature dependent phenomenon and it takes place when the temperature in the combustion chamber exceeds 2000 K. Therefore, in order to reduce NOx emissions in the exhaust, it is necessary to keep the peak combustion temperatures under control. One efficient way for ensuring this is by Exhaust Gas Re-circulation (EGR). First part of this paper covers an extensive review of the research work carried out in the area of EGR. Three types of EGR are used: *hot, partly cooled, cooled EGR*. Two configurations of EGR systems namely long route and short route system have been used in various experimental investigations. In Short Route EGR systems some modifications, such as use of nozzle or venturi mixer, have been investigated. Using EGR helps in reducing NOx, but appreciable particulate emissions are observed to increase, hence the there is a trade-off between NOx and smoke emission. To get maximum benefit from this trade-off, particulate trap is used to reduce the amount of unburned particulates in EGR, which in turn reduce the particulate emission. Some ceramic filters have been used for this purpose.

In the later portion of the paper, design of experimental set up for a ***long route partially cooled*** EGR system for two-cylinder diesel engine has been described.

INTRODUCTION

Over past 15-20 years more stringent legislation has been imposed on NOx, smoke and particulate emissions emitted form automotive diesel engines. Table 1 gives legislative limits for various pollutants in Europe and their implementation years. Table 2 gives a similar legislative limits targeted in USA. With the implementation of US EPA 2007 legislations, air pollution will be reduced by 90% compared to 2004 levels.

For reduction of vehicular emissions, baseline technologies are being used which include direct injection, turbo-charging, air-to-air inter-cooling, combustion optimization with and without swirl support, multi-valve cylinder head, advanced high pressure injection system i.e. split injection or rate shaping, electronic management system, lube oil consumption control etc. However, advanced technologies like EGR, soot trap and exhaust gas after-treatment will be essential to cater to the challenges posed by environmental emission legislations.

	CO (g/kWhr)	HC (g/kWhr)	NOx (g/kWhr)	PM (g/kWhr)	Smoke (m^{-1})	Implementation year
Euro I	4.5	1.1	9.0	0.40	--	1993
Euro II	4.0	1.1	7.0	0.15	--	1996
Euro III	2.1	0.66	5.0	0.1	0.8	2001
Euro III EEV	1.5	0.25	2.0	0.02	0.15	2003
Euro IV	1.5	0.46	3.5	0.02	0.5	2006
Euro V (proposed)	1.5	0.46	2.0	0.02	0.5	2009

Table 1: European Union Emission Legislation Limits[1]

US EPA 2002/2004:

CO (g/bhp hr)	NOx + NMHC (g/bhp hr)		PM (g / bhp hr)
15.5	2.4	2.5 with at max. 0.5 NMHC	0.1

US EPA 2007:

NOx (g/bhp-hr)	NMHC (g/bhp-hr)	HCHO (g/bhp-hr)	PM (g/bhp-hr)
0.20	0.14	0.016	0.01

Table 2: American Exhaust Emission Legislations [1]

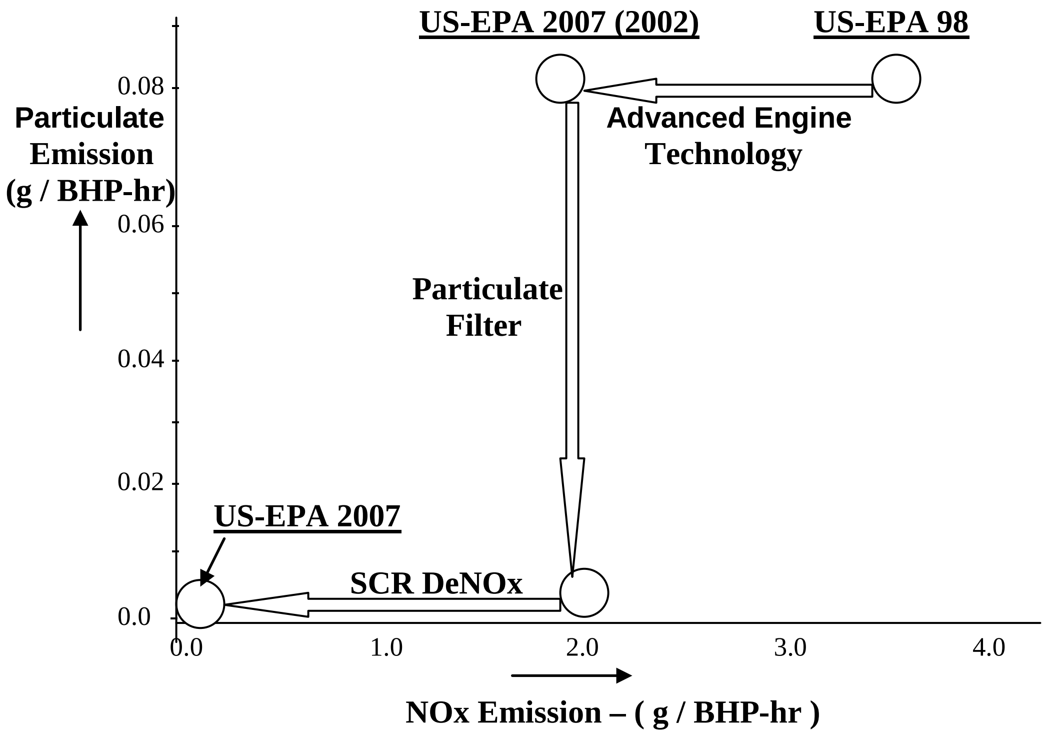


Fig 1: Targeted emission levels and potential technologies [1]

Fig 1 shows the various limits for NOx and particulate emissions and the various potential technologies. From this figure, it is glaringly clear that it is essential to control the particulate matter emissions as well as NOx emissions simultaneously in order to meet the future legislative requirements. Plenty of research work has been done in this area. NOx is produced when the temperature inside the combustion chamber rises above 2000 K, hence it is desirable to reduce the peak temperature inside the combustion chamber.

Mechanism of Formation of Nitric Oxides:

A major hurdle in understanding the mechanism of formation and controlling its emission is that the combustion is highly heterogeneous and transient in the diesel engine.

NO is formed during the post flame combustion process in a high temperature region. The most widely accepted mechanism is that of Zeldovich. The principle source of NO formation is the oxidation of the Nitrogen present in atmosphere. The Nitric oxide formation chain reactions are initiated by atomic oxygen, which forms from the dissociation of oxygen molecules at the high temperatures reached in the combustion process.

$$N_2 + O \rightarrow NO + N$$
$$N + O_2 \rightarrow NO + O$$

The local atomic oxygen concentration depends on molecular oxygen concentration as well as local temperature.

Fig 2 depicts the effect of Oxygen concentration in the intake air on the concentration of NOx emissions in the exhaust.

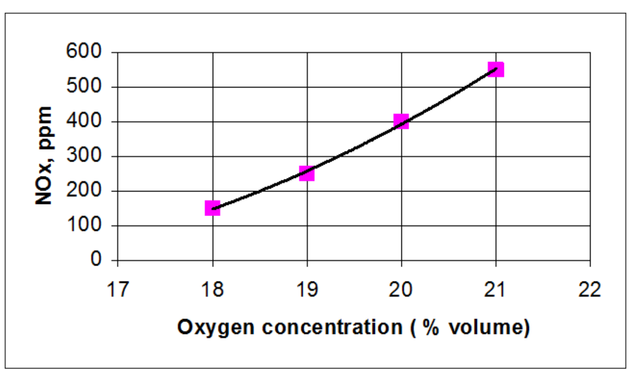


Figure 2: Effects of intake oxygen concentration on NOx formation [2]

The formation of NOx is almost absent at temperatures below 2000 K. There are various ways of lowering the combustion chamber temperature:

1. **Enriching the fuel-air mixture:** This process reduces the amount of air and hence the oxygen contained. Hence the heat generated from the combustion is not so large.
2. **Lowering the compression ratio:** This reduces the temperature of the engine after the compression stroke; hence the combustion that follows is not so intense.
3. **Spark timing control:** This also reduces the heat generated by changing the ignition timing. In diesel engine changing the fuel injection timing complements this.
4. **Exhaust Gas Re-circulation (EGR):** Re-circulating a small fraction of the exhaust gases makes the final air-fuel mixture lean and hence controls the temperature of the combustion chamber.

Exhaust Gas Re-Circulation:

EGR is a useful method for reducing NOx formation. Exhaust consists of CO_2, N_2 and water vapor mainly. When a part of this exhaust gas is re-circulated to the combustion cylinder, it acts as diluents. This also reduces the O_2 concentration in the combustion chamber. The specific heat of the EGR is much higher than fresh air hence EGR increases the heat capacity of the intake charge, thus decreasing the temperature rise for the same heat release.

$$\% \, EGR = \frac{\mathit{Volume} \text{ of EGR}}{\text{Total charge intake into the cylinder}} \times 100$$

Another way to define the EGR ratio is by the use of CO_2 concentration [3]:

$$\mathit{EGR} \text{ ratio} = \frac{[\mathrm{CO_2}]_{intake} - [\mathrm{CO_2}]_{ambient}}{[\mathrm{CO_2}]_{exhaust} - [\mathrm{CO_2}]_{ambient}}$$

The local temperature, with different EGR rates, depends on oxygen concentration and heat capacity of the intake charge.

At high loads, it is difficult to employ EGR due to deterioration in diffusion combustion and this results in an excessive increase in smoke and particulate emissions. At low loads, unburned hydrocarbons contained in the EGR would possibly re-burn in the mixture, leading to lower unburned fuel in exhaust and improved brake thermal efficiency. Apart from this, hot EGR would raise the intake charge temperature, thereby, influencing combustion and exhaust emissions.

A decrease in intake oxygen concentration decreases NO emission as shown in Fig 2. The EGR decreases local atomic oxygen concentration and local temperature, which in turn reduces the NO formation rate. Temperature in the combustion chamber is more important factor in NO formation than oxygen concentration. The increased intake charge heat capacity also influence the temperature inside and hence the NOx formation.

The higher level of exhaust smoke concentration measured for inert dilution with re-circulated exhaust gas, as compared to pure gases (Nitrogen and carbon dioxide), is due to the fact that the smoke or soot in the re-circulated exhaust gas is fed back into the intake manifold, since no soot filtration is made on the re-circulated exhaust gas.

HISTORICAL PERSPECTIVE

During last 20 years, plenty of research work has been done on EGR and its effects on the engine performance in terms of fuel efficiency, volumetric efficiency, power generated etc. These studies have been done at various loads, RPMs and variable engine parameters like temperature and pressure, compression ratio etc. Researchers have experimented with several types and configurations. A brief overview of the earlier research work is described here.

Different Types of EGR Systems

Various EGR systems have been classified on the basis of EGR temperature, configuration and pressure etc. All these classifications are discussed in the following paragraphs:

(i) Classification Based on Temperature

1. **Hot EGR:** Exhaust gas is re-circulated without being cooled, resulting in the increased charge temperature.
2. **Fully cooled EGR:** Exhaust gas is cooled before re-circulation in to the combustion chamber by the means of a water-cooled heat exchanger. In this case, condensed water enters the cylinder and produces undesirable effects.
3. **Partly cooled EGR:** To avoid the water condensation, the temperature of exhaust gas is kept just above its dew point temperature.

In case of hot EGR, thermal efficiency is found to improve due to increased intake charge temperatures and re-burning of the unburned fuel present in the re-circulated gas. Therefore it has been concluded that the use of EGR is most effective in improving exhaust emissions at low loads. In cooled EGR the condensed water is removed before mixing with fresh air. This minimizes the effect of water on soot and oxidation kinetics inside the combustion chamber. [4]

Daiso *et al* [4] carried out experimental investigations on a direct-injection diesel engine dual-fueled with natural gas by implementing all three above-mentioned EGR methods. They found that:

1. At lower loads using hot EGR, brake thermal efficiency is improved due to increased intake temperatures. Also NOx and smoke emissions come down. Cooled EGR gives relatively lower thermal efficiency but lower NOx emissions also.
2. At high loads fully cooled and partly cooled EGR tends to significant reduction in NOx and smoke emissions due to delay in dual-fueled combustion. Partly cooled EGR is found to be a better solution as it prevents the water condensation.

At lower loads, high unburnt hydrocarbon emissions are observed. Sometimes a suitable oxidation catalyst is used (Pt or Pd) in the catalytic converter for improving the combustion of after-treatment exhaust gas.

(ii) Classification Based on Configuration

1. **Long Route System (LR):** In LR system the pressure drop across the air intake and the stagnation pressure in the exhaust gas stream cause the EGR possible.
2. **Short Route System (SR):**

   Lundquist *et al* [5], tested different systems for obtaining short-route cooled EGR on turbocharged and after cooled Heavy Duty Diesel Engine. These systems differed mainly in the method used to set up a positive pressure difference across the EGR circuit. At high loads, the typical turbocharged diesel engine operates with intake pressure higher than mean exhaust pressure. This opposes the EGR flow. Two different types of short route systems were used in the experiments:

1. **Short Route System 1 (SR1):** Simple exhaust backpressure valve (EBPV) placed after the turbocharger turbine to raise EGR pressure. Therefore, EGR-inlet is placed at a point with high velocity, so that static pressure is lower than the total pressure in the EGR-line. An intercooler is required in this system to facilitate the mixing of intake fresh air and EGR. There is a valve in the EGR-line to control the EGR-rate. The configuration of SR1 is shown in Fig 3:

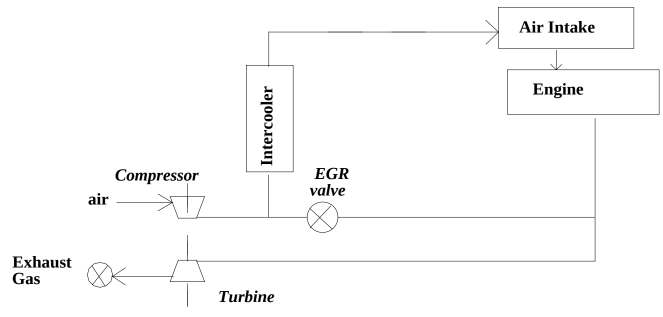


Figure 3: Schematic Layout of SR1 System [5]

2. **Short Route System 2 (SR2):** In this system, fresh air intake is passed through a venturi, which is placed downstream to the intercooler. In the venturi, low pressure is created due to high velocity, which causes the flow of EGR

from the exhaust manifold to venturi. This is shown in Fig 5.

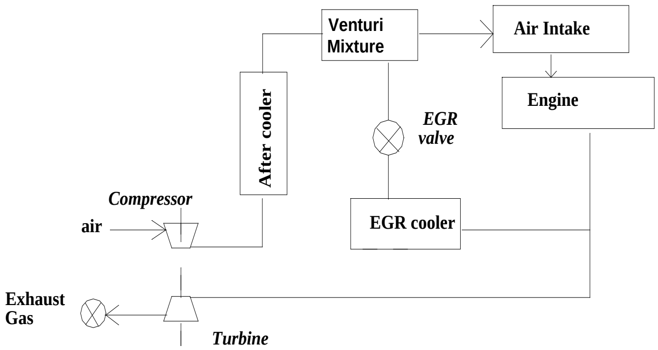


Figure 4: Turbocharged Diesel Engine with a cooled EGR system [3]

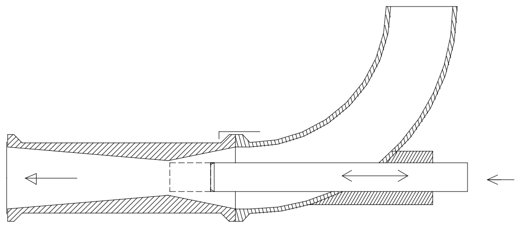


Figure 5: SR2 System with Variable Venturi [5]

Another way of controlling the EGR-rate is to use Variable Nozzle Turbine (VNT). Most of the VNT systems have single entrance, which reduces the efficiency of the system by exhaust pulse separation. Cooled EGR should be supplied effectively. Lundquist *et al* used a variable venturi, in which EGR-injector was allowed to move axially, thus varying the critical area [5].

(iii) Classification Based on Pressure

Kohketsu *et al* [6] conducted EGR experiments on a 12 liter turbocharged and intercooled Direct Injection diesel engine. They employed two different routes for EGR, namely low-pressure route system and high-pressure route system.

1. **Low Pressure Route System:** The passage for EGR was provided from downstream of the turbine to upstream side of the compressor as shown in Fig. 6. A suitable pressure difference was obtained. The *advantages* associated with this system are:
   - Reduced control complexity.
   - Fuel economy.

   But this method has some *disadvantages* too:
   - Durability and reliability problems come into the picture, since EGR is passed through compressor and intercooler.
   - Pressure loss in the intercooler increases after some time due to clogging.

   Also this system requires a durable particulate filter with reliable regeneration.

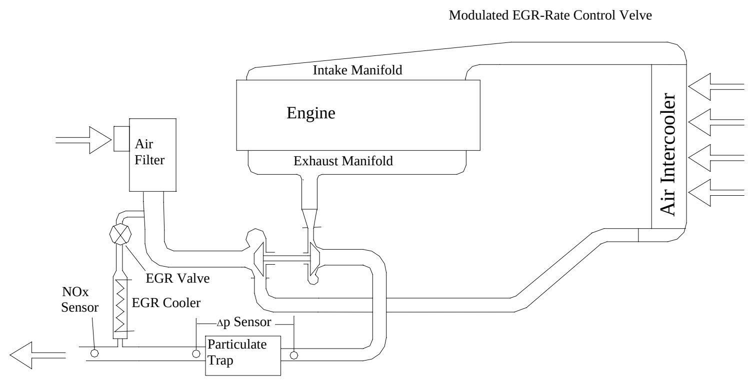


Figure 6: Low Pressure Route EGR System [6]

2. **High Pressure Route System:** The EGR is passed from upstream of the turbine to the downstream of the compressor as shown in the Fig. 7. The basic *advantages* of this system are:
   - Since EGR is not passed through compressor or intercooler, the problems of durability and reliability are not there.
   - The particulate trap is optional.

But the *problems* that may arise are:
   - System contamination.
   - Increased soot-in-oil.
   - Transport losses increase with improved TC efficiency.
   - Complicated VTG-EGR control.

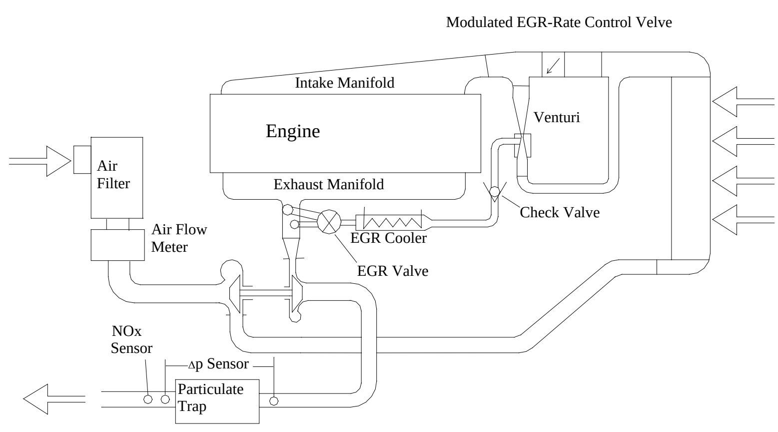


Figure 7: High Pressure Route EGR System [6]

To make EGR possible, pressure upstream of the turbine must be higher than the pressure downstream of the compressor.

$$\mathbf{P_{differential}\ (DP) = P_{upstream} - P_{downstream}}$$

$P_{differential}$: Differential Pressure

$P_{upstream}$: Pressure upstream turbine

$P_{downstream}$: Pressure downstream compressor

DP should be greater than zero for EGR systems. In light load region DP > 0, but for high load DP<0. To get the DP>0 in high load region several methods are offered, such as increasing Back Pressure and using a Variable Geometry (VG) turbocharger Variable Geometry (VG) Turbocharger: VG turbocharger achieves such geometry by changing the angle of vanes in the turbine inlet nozzle. The nozzle-area-throttling increases, as the EGR increases.

Advances and Disadvantages of Using EGR

With the use of EGR, there is a trade-off between reduction in NOx and increase in soot, CO and unburned Hydrocarbons. A large number of studies have been conducted in this area. It has been indicated that for more than 50% EGR, particulate emissions increased significantly, and therefore use of Particulate Trap was recommended. The change in oxygen concentration causes the change in the structure of the flame and hence changes the duration of combustion. It was suggested that the flame temperature suppression is the most important factor influencing the NO formation.

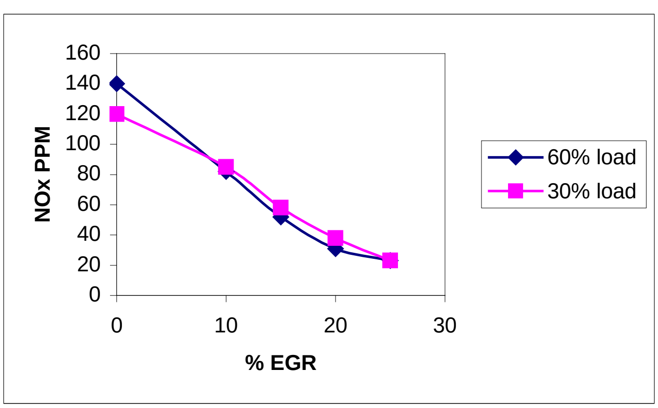


Figure 8: Effect of EGR on NOx [7]

Fig. 8 shows the reduction in NOx emission due to EGR at different loads. Implementation of EGR in Diesel engine has problems like (a) increased soot emission (b) it introduces the particulate matter into the engine cylinders. When the engine components come into contact with high velocity soot particulates, particulate abrasion occurs. Sulfuric acid and condensed water in EGR also causes corrosion. Some studies have also detected the damage on the cylinder walls due to the reduction in oil's lubrication capacity, which is hampered due to the mixing of soot carried with the particulates gas with the lubricating oil.

Use of Particulate Trap:

Studies have shown that EGR coupled with a high collection efficiency particulate trap, control the smoke, unburned hydrocarbon and NOx emissions simultaneously. Extensive research has been conducted on capturing the particulates at the exhaust of the engine using a variety of filters. Particulate collection efficiencies were obtained in the range of 50-80%. Some experimental studies on EGR in diesel engines also showed that water injection with EGR could also be used to keep the particulate emission low. Levendis *et. al.* [7] used the *Ceramem* filter (this filter was coated with a thin microporous ceramic membrane to provide the soot removal efficiencies) to make EGR possible. In their experiments 10-25% EGR were used for 30% and 60% load. Gaseous hydrocarbons were found to decrease by 80% for a 15% EGR. This is an important feature since EGR often increases unburned hydrocarbons (as a result of reduced oxygen concentration and temperature). Soot traps are used for capturing unburnt hydrocarbons and particulate matter from EGR systems. Over a period of time, these traps get clogged and their efficiency decreases. Clogged traps also offer backpressure to the engine exhaust, thus affecting engine performance. These traps need to be regenerated from time to time using thermal or aerodynamic regeneration techniques.

Thermal Regeneration is burning the soot that gets collected in the traps. Regenerations had to be performed periodically. Such devices normally operated under high backpressures since the frequency of regenerations is kept low. These particulate trap systems are unable to trap the sulphates and ash particulates.

To moderate such problems North-Eastern University has employed **aerodynamic regeneration** i.e. dislodging the soot from the trap using compressed air flowing in the opposite direction to exhaust direction. This soot is collected in a separate fiber bags and burned periodically.

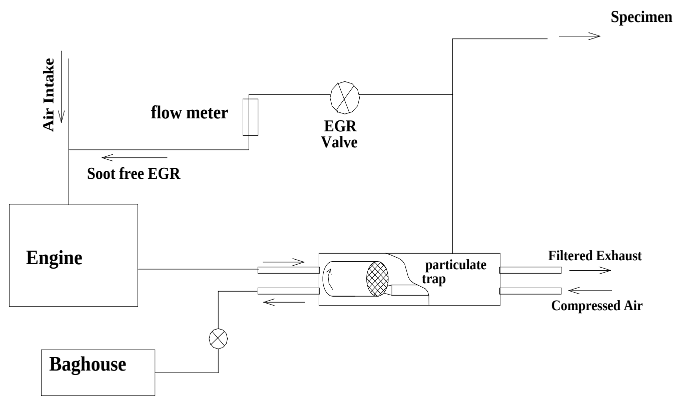


Figure 9: Schematic Diagram for Particulate Trap [7]

A complete diagram showing EGR fitted with particulate trap and regeneration mechanisms is shown in the figure 9.

Levendis *et al* [7] performed several other tests with intermittent pulsating flow. The filter was cleaned every half hour for 3 minutes. This caused a decrease in hydrocarbon retention effectiveness of the trap. The reason for this was rise in the exit temperature, as the cooling effect of the compressed air diminished. Therefore, condensation of hydrocarbons on the soot was less pronounced.

With the use of aerodynamically regenerated filters in the diesel engine, particulate emission can be reduced by as much as 99%, unburned Hydrocarbons by as much as 80% and NOx by as much as 80%.

Another method of reducing the particulate emissions for EGR engine is using multiple injection. Pierpont *et al* [8] used methanol fumigation technique to control the particulate emissions, but CO, HC and aldehyde emissions increased. Later on, experiments were conducted on the use of double and triple injections with an optimal delay between injections. This technique of multiple injection reduced particulate emissions to as much as one third and this reduction was attributed to the enhancement in the mixing, which promoted the particulate oxidation later in the cycle. The most important variables in multiple injections are the delay preceding the final pulse [8].

Effect of Supercharging on EGR:

Supercharging and increasing fuel injection pressure prevent the deterioration of smoke and unburned hydrocarbons and improve fuel economy. Daisho *et al* [9] supercharged the EGR keeping the amount of fresh air same as under naturally aspirated conditions. A root blower was utilized to compress the intake air to a higher pressure. The blower was driven by electric motor, which was separated from the engine output. A water-cooled heat exchanger was used to keep the temperature of the compressed air at room temperature.

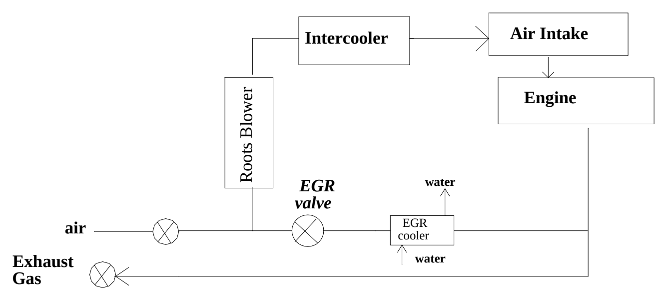


Figure 10: Layout of EGR system (with Supercharging) [9]

To determine the effect of EGR and supercharging on combustion and emission characteristics, they carried out two procedures:

Replaced EGR: The intake boost pressure was kept constant and EGR ratio was varied.

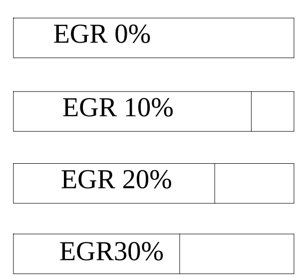


Fig 11: Replaced EGR [9]

Additional EGR: The amount of fresh air was kept constant and equal to that in naturally aspirated condition.

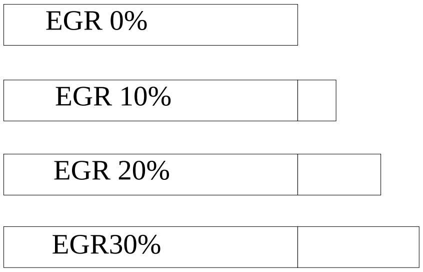


Fig 12: Additional EGR [9]

Supercharging enhances the air-fuel mixing and diffusion combustion so that smoke and THC emissions are kept under control.

Design of an Experimental EGR System for 2-Cylinder Diesel Engine

A two-cylinder constant speed diesel engine used in present investigations. It is an air-cooled, vertical, direct-injection engine. The specifications of the engine are given in table 2:

Type	PH2
No. of cylinders	2
Make / model	INDEC PH
Bore / Stroke	87.3 mm / 110 mm
Max. Power / rpm	12.5 bhp / 1500
Compression Ratio	16.5 : 1

Table 3: Specifications of the Engine

The objective of developing this experimental test setup is to investigate and demonstrate the effects of various EGR rates and other engine parameters on exhaust emissions from the engine. A long route partially cooled EGR system is chosen based on its merits mentioned earlier. Several components of this EGR system were designed. Design of each component is discussed in detail in the following paragraphs.

An air-box is designed for measurement of volumetric flow rates.

Design of Air-Box

An air box is designed to measure the volumetric flow rate of intake air to the engine. It is mounted on the inlet pipe between the air filter and the inlet manifold of the engine as shown in fig. . Air box dampens out the fluctuations of the intake air. A diaphragm is provided on the side of the air box for dampening out the local undulations effectively as shown in fig. . The air box is fitted with an orifice for volumetric flow rate measurement of air. An inclined manometer is mounted across the orifice, to measure the pressure difference inside the air box and the atmosphere.

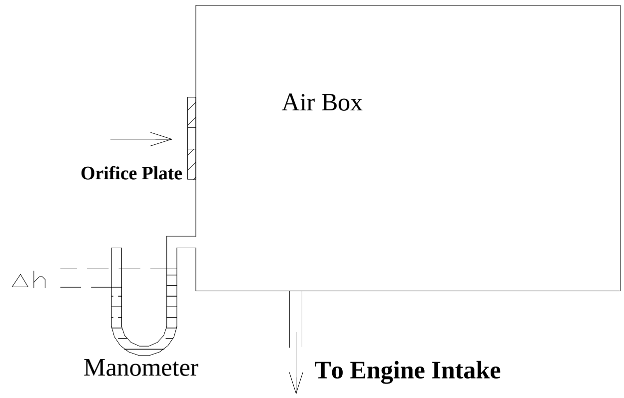


Fig 13: A Schematic Diagram [15]

Design Parameters:

$$\boxed{U = \frac{1}{40.94 * 10^5}\left(\frac{\eta_v C V N^2 n^2}{T d^4 p^2}\right)}$$

η_v = Volumetric efficiency of the engine (around 80%).
V = Total swept volume of the engine (=1.4 l)
C = Air-box volume (400 to 600 times V for single cylinder engine. Can be lesser for multi-cylinder engines).
N = Engine RPM (1500).
n = No. of cylinders (2).
p = No. of crank-strokes per induction-stroke (2 for all 4-stroke engines).
T = Ambient temperature (300 K).
d = Orifice diameter.

U = A design parameter which determines the effectiveness of the air-box in dampening out the fluctuations encountered.

The value of U should be greater than 2.5, else the air-box will not be able to dampen the violent fluctuations very effectively. This value is used in determining the orifice diameter of the air-box.

Design Calculations:

From the formula and the values already written above,

$$Ud^4 = \frac{(0.8)C(1.4*10^{-3})(2)^2(1500)^2}{(40.94*10^5)(300)(2^2)}$$

$$\boxed{Ud^4 = 2.052*10^{-6}\ C}$$

The values of U versus d were tabulated for various values of C. We note that d = 20 mm. (corresponding to C = 0.216 m^3) was the best suited value of the orifice diameter because for this case, the value of U turned out to be slightly greater than 2.5 and also from the point-of-view of manufacturing these values are nice.

Re-circulation System

Part of the exhaust gas is to be re-circulated and put back to the combustion chamber along with the intake air. The quantity of this EGR is to be measured and controlled accurately, hence a by-pass for the exhaust gas is provided along with the manually controlled EGR valve. In the first phase of the experiments the EGR valve will be controlled manually. A solenoid controlled EGR valve may automate this in the later phase of experiments.

The exhaust gas comes out of the engine during the exhaust stroke at high pressure. It is pulsating in nature. It is desirable to remove these pulses in order to make the volumetric flow rate measurements possible. For this purpose, another air box with a rubber diaphragm is installed in the EGR route. An orifice meter is designed and installed to measure the volumetric flow rate of the EGR. The design criterion used is described in the paragraph given below.

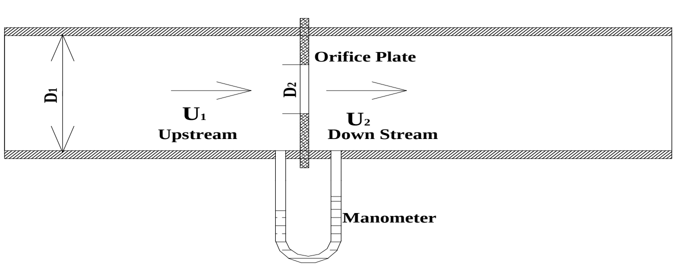


Fig 14: A Schematic Diagram [12]

Design Parameters:

Diameter ratio $\beta = D_2 / D_1$

Flow rate $W = (\rho_2 Y)\, U_2 A_2$

Let us study every quantity on the RHS of this equation one-by-one …

$$Y = 1 - \left[0.333 + 1.145\left(\beta^2 + 0.7\beta^5 + 12\beta^{12}\right)\right]\frac{\Delta P}{kP_1}$$

Y = Expansion Factor (based on absolute static pressure) which compensated for the density change of the fluid flowing in the pipe across the orifice.

$\Delta P = P_1 - P_2$ (differential pressure across the orifice)

k = $C_P / C_V = \gamma$ (for the EGR = 1.4)

$\beta = D_2/D_1$

$$U_2 = \frac{C\sqrt{2g\Delta h}}{\sqrt{1-\beta^4}}$$

… velocity of the EGR.

C : Coefficient of Discharge.

Δh : differential pressure across the orifice plate.

$A_2 = \alpha\,(\pi D_2^2 / 4)$ … area of the orifice

α : Area Multiplier (which accounts for the thermal expansion of the orifice plate as it comes in contact with the hot EGR).

D_2 : Diameter of the orifice.

The appropriate values of every quantity,

α : 1.025

D_1 : This is dia of the EGR pipe where the orifice-meter is mounted. D_1 = 1’’ = 25.4 mm

U_1 : Its determination needs a more involved calculation …

Another flow variable to be considered is the Reynolds’ Number:

$$Re_D = u_1 D_1 / \nu$$

The determination of the Reynolds’s Number is necessary for reading the value of β from the charts. For Re_D < 10 000, it is known that $0.2 < \beta < 0.5$

A set of sample calculation for various values of EGR fraction was performed while considering the constraint that we must be getting a measurable value of the quantity Δh_w in the whole range (5% to 15%) of the fraction of EGR. We rejected the results where Δh_w was greater than 50 cm or less than 2 cm.

Finally from the tabulated results, we concluded that that $\beta = 0.4$ suits the whole range the best (for this, Δh_w ranges from 1.5 cm to 13.1 cm).

Thus we propose to choose $\beta = 0.4$, for which $D_2 = 0.4*D_1 = 10$ mm.

We have proposed to use a 1'' pipe when we are doing the measurements of the amount of EGR (inside the orifice-meter), and then we will step down the pipe diameter to ½'' (i.e. 12.7 mm) using a reducer. The explanation is that while we need to use a narrow pipe (½'' dia) for the EGR system, the same pipe can not be used in the orifice-meter, hence we must step up the pipe to 1'' at the exit of the second air-box, do the relevant measurements and then step down the pipe to the original size (½'') using a reducer.

The detailed schematic design of the Experimental EGR System is shown above in fig. 13. A pressure gauge is mounted in this EGR route to measure the EGR pressure. Suitable instrumentation is provided to acquire useful data from various locations. Thermocouples are provided at the intake manifolds, exhaust manifolds and various points along the EGR route.

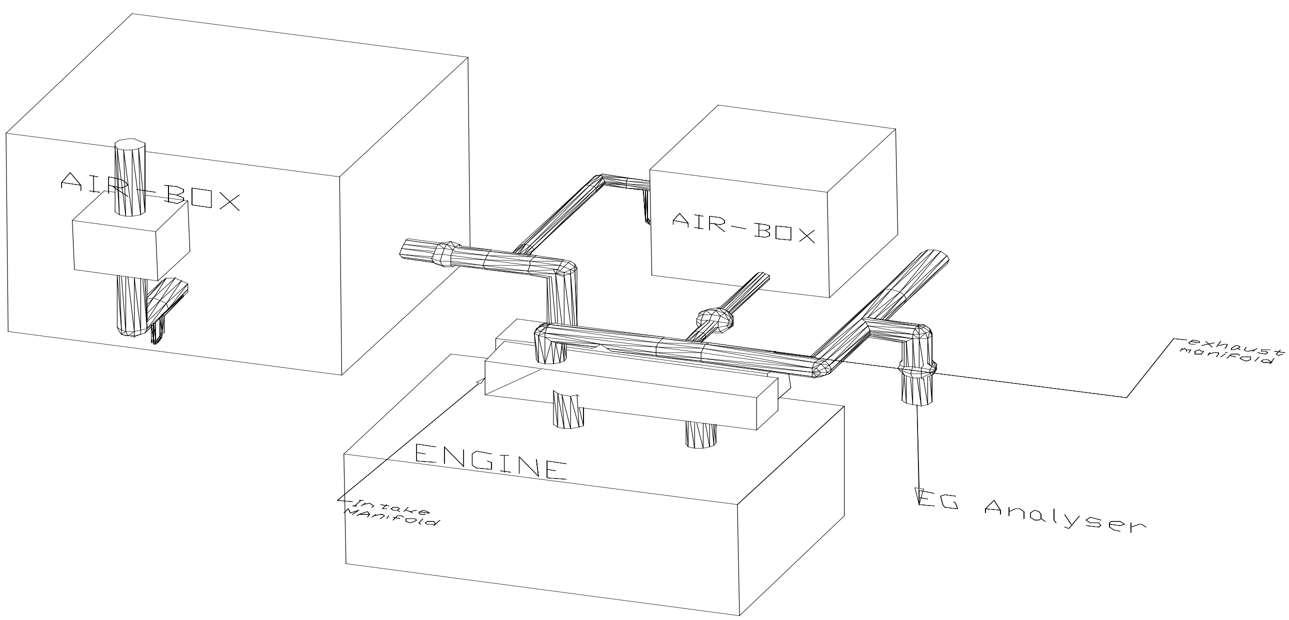
It is desirable to lower the EGR temperature before re-circulation. Circular fins are provided on the outer periphery of the pipe along the EGR route to dissipate the heat to the atmospheric air through natural convection.

'AVL-4000 light' Exhaust gas analyzer is used for the analysis of the various exhaust emissions such as CO, HC, particulate matter and NOx. This exhaust gas analyzer works on the principle of non-dispersive infrared emission (NDIR).

Proposed Experimentation

It is proposed to execute an array of experiments to study the effects of various EGR rates on the emission from the engine, primarily on soot and NOx. The ERG rates will be optimized for various engine loading conditions. Effect of EGR on various engine performance parameters such as thermal efficiency, brake specific energy consumption, smoke opacity etc. will also be investigated.

Second phase of experiments will include automation of engine for automatic control of the EGR rates using solenoid-controlled valve. Finally a catalytic converter will be used to control unburnt hydrocarbon emission and thus the objective of simultaneously controlling soot and NOx emissions.



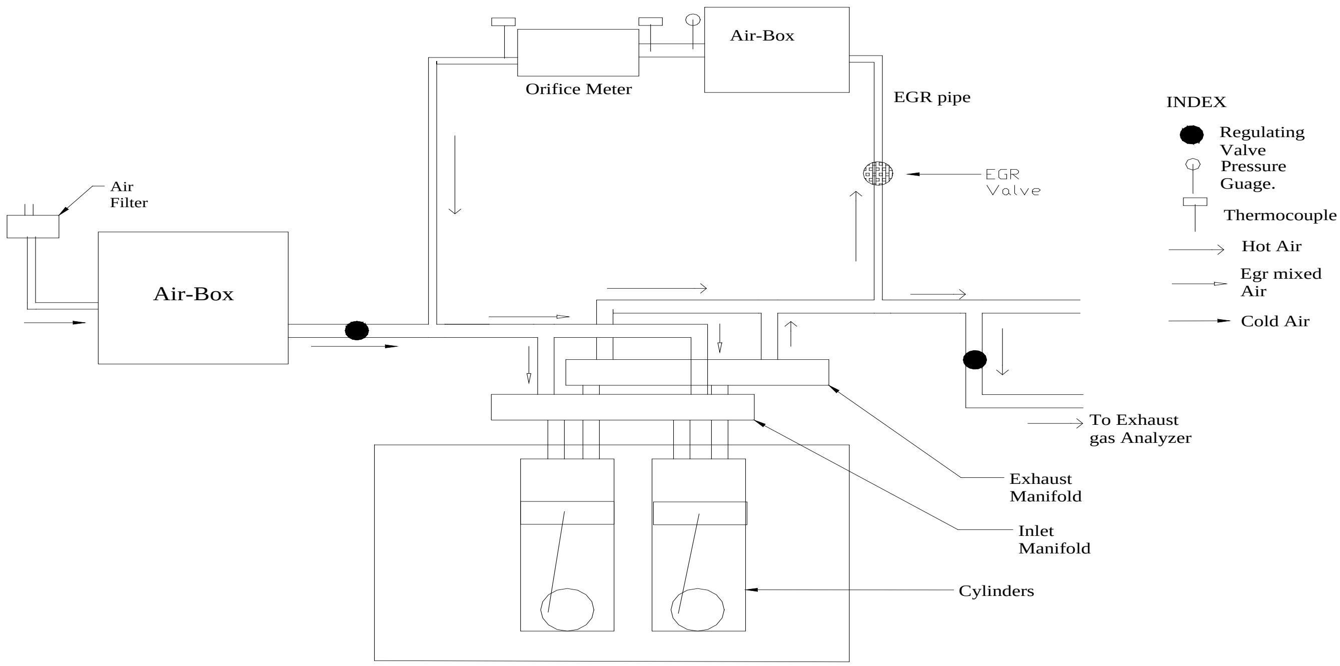


Fig. 15: Schematic Diagram of the proposed EGR System

REFERENCES

1. Johaannnes F. Kregar, AVL Emissions Test Systems, AVL LIST GmbH Graz Austria, “The Heavy Duty Diesel Engine Emission Beyond Euro III and IV”, AVL Seminar on Engine Testing on the way to Euro III and IV, October 2001 Delhi India.
2. Robert C. Yu and Syed M. Shahed, “Effects of Injection Timing and Exhaust Gas Re-circulation on Emissions from a D. I. Diesel Engine”, Society of Automotive Engineers, 1981.
3. R. S. G. Baert, D. E. Beckman, A. Veen, “Efficient EGR Technology for Future HD Diesel Engine Emission Targets”, Society of Automotive Engineers, 1999.
4. Yasuhiro Daishu, Kou Takahashi, Yuki Iwashiro, Shigeki Nakayama, Ryoji Kihara, and Takeshi Saito, “Controlling Combustion and Exhaust Emissions in a Direct Injection Diesel Engine Dual-Fueled with Natural Gas”, Society of Automotive Engineers, 1995.
5. Ulf Lundqvist, Gudmund Smedler, Per Stalhammar, “A Comparison Between Different EGR Systems for HD Diesel Engines and Their Effects on Performance, Fuel Consumption and Emissions”, Society of Automotive Engineers, 2000.
6. Susumu Kohketsu, Kazutoshi Mori, Kenji Sakai, Takazoh Hakozaki, “EGR Technologies for a Turbocharged and Inter-cooled Heavy-Duty Diesel Engine”, Society of Automotive Engineers, 1997.
7. Yiannis A. Levendis, Iraklis Pavlatos, Richard F. Abrams, “Control of Diesel Soot, Hydrocarbon and NOx Emissions with a Particulate Trap and EGR”, Society of Automotive Engineers, 1994.
8. D. A. Pierpont, D. T. Montgomery, R. D. Reitz, “Reducing Particulate and NOx Using Multiple Injections and EGR in a D. I. Diesel”, Society of Automotive Engineers, 1995.
9. Noboru Uchida, Yasuhiro Daisho, Takeshi Saito, Hideaki Sugano, “Combined Effects of EGR and Supercharging on Diesel Combustion

and Emissions", Society of Automotive Engineers, 1993.

10. M. L. Mathur and R. P. Sharma, "Internal Combustion Engine", 1994 ed. Dhanpat Rai & Sons.(998 pp)
11. John B. Heywood, "Internal Combustion Engine Fundamentals", 1998 ed. Mc Graw Hill International, Automotive Technology Series.
12. Reid F. Stearns, Russell R. Johnson, Robert M. Jackson, charles A. Larson, " Flow Measurements with Orifice Meters", 1951 ed. D. Van Nostrand Company, Inc. Princeton, New Jersy.