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Combined Effect of EGR and Inlet Air Preheating on Engine Performance in Diesel Engine

M.M.Z. Shahadat[*1], M.N. Nabi[*], M.S. Akhter[*] and M.S.H.K. Tushar[*]

Abstract – *Concern of environmental pollution and energy crisis all over the world have caused the research attention on reduction of IC engine exhaust emissions and saving of energy simultaneously. This investigation mainly focuses on reducing exhaust emission and energy saving by investigating diesel combustion with neat diesel fuel and a new attachment of inlet air preheater with Exhaust Gas Recirculation (EGR) system. Experiment was conducted in a four stroke naturally aspirated (NA) diesel engine with inlet air preheating attachment and EGR system. Inlet air preheating and EGR were applied to the experimental engine separately and also together to observe their effects on engine performance. In this study, engine exhaust gas temperature was used to preheat the inlet air passing through a newly designed air preheating system. In inlet air preheating attachment, a counter flow heat exchanger was used to transfer heat from exhaust gases to inlet air. It was found that combined effect of inlet air preheating attachment and EGR system provided better result on engine performance than individual effect. It was found that at medium load conditions, oxides of nitrogen (NO_x), carbon monoxide (CO), engine noise, and brake specific fuel consumption decreased when inlet air preheating and EGR were applied together as compared to those during normal operations of the engine. Thus the modified engine provides clean atmosphere and better fuel economy without reducing useful characteristics (brake power, brake thermal efficiency etc) of the engine.*

Keywords – Air preheating, EGR, exhaust emission, ignition delay.

1. INTRODUCTION

In the present civilization the CI (compression ignition) engines play an important role on prime mover, being used in buses, trucks, locomotives, tractors, pumping sets, and other stationary industrial applications, small and medium electric power generation and marine propulsion.

Compressed air of earlier CI engine was capable to ignite few particular gasoline fuels [1], [2]. Earlier CI combustion was sporadic, due to the defective fuel injection system. The injection system and the engine construction were improved day by day and finally the efficiency of the CI engine was raised to 36% after working with the design of the combustion space in 1901 [3]. The forces coming on the various parts of the engine are greater due to the compression ratio 12:1 to 22:1 and heterogeneous mixture, lean mixture (large air-fuel ratio), and therefore heavier parts are necessary in CI engine. Both the factors heavier engine as well as varied applications and are manufactured in a large range of sizes, speeds, and power output [3], [4]. The smoke and odor are the result of the nature of diesel combustion phenomenon, i.e. incomplete combustion of a heterogeneous mixture, and droplet combustion.

Fuel is injected by the fuel-injection system into the engine cylinder toward the end of the compression stroke, just before the desired start of combustion. The liquid fuel usually injected at high velocity as one or more jets through small orifices or nozzles in the injector tip, atomizes into small drops and penetrates into the combustion chamber [5]. The fuel vaporizes and mixes with the cylinder air of high-temperature and pressure. Since the air temperature and pressure are above the fuel's ignition point, spontaneous ignition of portion of the already-mixed fuel and air occurs after a delay period of a few crank angle degrees. The cylinder pressure increases as combustion of the fuel-air mixture occurs. The consequent compression of the unburned portion of the charge shortens the delay before ignition for the fuel and air, which has mixed to within combustible limits, which then burns rapidly. It also reduces the evaporation time of the remaining liquid fuel. Injection continues until the desired amount of fuel has entered the cylinder. Atomization, vaporization, fuel-air mixing, and combustion continue until essentially all the fuel passed through each process. In addition, mixing of the remaining air in the cylinder with burning and already burned gases continues throughout the combustion and expansion processes [6].

The ignition delay is defined as the time (or crank angle) interval between the start of injection and the start of combustion. The start of injection is usually taken, as the time when the injector needle lifts off its seat (determined by a needle-lift indicator). The start of combustion is more difficult to determine precisely. It is best identified from the change in slope of the heat release rate. Depending on the character of the three combustion processes, the pressure data alone may indicate when pressure change due to combustion first occurs; in DI (direct injection) engines under normal conditions ignition is well defined, but in IDI (indirect ignition) engines the ignition point is harder to identify [7], [8]. Flame luminosity detectors are also used to determine the first appearance of the flame.

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The chemical delay of the ignition delay is controlled by the pre-combustion reactions of the fuel. Though ignition occurs in vapor phase regions, oxidation reactions can precede in the liquid phase as well between the fuel molecules and the oxygen dissolved in the fuel droplets. In addition, cracking of large hydrocarbon molecules to smaller molecules is occurred. These chemical processes depend on the combustion of the fuel and the cylinder charge temperature and pressure, as well as the physical processes described above which govern the distribution of fuel throughout the air charge [9].

Since the ignition characteristics of the fuel affects the ignition delay, this property of a fuel is very important to determine diesel-operating characteristics such as fuel conversion efficiency, smoothness of operation, misfire, smoke emissions, noise, and ease of starting. The ignition quality of a fuel is defined by its cetane number. For low cetane fuels ignition delay is too long, most of the fuel is injected before ignition occurs, which results in very rapid burning rates once combustion starts with high rates of pressure rise and high peak pressures. Under extreme conditions, auto ignition of most of the injected fuel occurs, this produces an audible knocking sound, often referred to as "diesel knock" [10]. For fuels with very low cetane numbers, with an exceptionally long delay, ignition may occur sufficiently late in the expansion process for the burning, process to be quenched, resulting in incomplete combustion, reduced power output, and poor fuel conversion efficiency. For higher cetane number fuels, with shorter ignition delays, ignition occurs before the injection of the most of fuel [11], [12]. The rate of heat release and pressure rise is then controlled primarily by the rate of injection and fuel-air mixing, and smoother engine operation results.

Afify [13] found the effect of intake air heating on engine performance with emulsified fuel. He reported that with preheating of intake air the exhaust emissions like unburnt hydrocarbon or CO emissions reduced. He also reported that BSFC is improved with preheating of intake air.

Azpiazu *et al.* [14] conducted research to observe the effect of preheating the intake air on engine performance. He found that CO emission is reduced significantly when the intake air is preheated. Engine performance was improved when air is preheated, authors added in their report.

Anand *et al.* [15] reported that simultaneous reduction of NOx and smoke from a direct-injection diesel engine with EGR and diethyl ether (DEE) as fuel. The authors conducted their research with the blend of diesel and DEE as fuel together with EGR. Optimum results of engine exhaust emissions were experienced by the authors when the blend of DEE-diesel with the combination of EGR were applied to the engine.

The diesel combustion reaction consists of hydrocarbon chains being oxidized in an explosive reaction to form carbon dioxide (CO_2) and water (H_2O). However, the reaction is not one hundred percent efficient and the constituent are not pure. The air used to supply the oxygen (O_2) contains about 80% nitrogen and diesel fuel contains small percentage of sulfur. The result is that trace amount of other chemicals are found in the reaction. All of the trace constituents are increasing an objectionable concentration to the environment with the increase of utilization of IC engine and finally posses a health risk at higher concentration.

Injected system and time also significantly affect the NO_x formation as well as the variation in fuel characteristics such as cetane number, viscosity, and rate of burning etc. Carbon monoxide and particulate matter are also the product of incomplete combustion due to insufficient amount of air in the air-fuel mixture. These trace productions are not formed in large quantities due to the availability of the excess amount of oxygen during combustion in CI engine and generally are not of concerned [16].

The aim of this paper is to investigate the effect of EGR and inlet air preheating on diesel combustion and exhaust emissions with conventional diesel fuel. One of the targets of this work is to reduce the diesel emissions with the newly designed system without deteriorating engine performance. The results of diesel emissions with the newly designed system were compared with that of without using the system.

2. EXPERIMENTAL SETUP AND PROCEDURE OF EXPERIMENTATION

The experiment has been conducted to a four-stroke diesel engine. The specification of the tested engine is shown in Table 1. Conventional diesel fuel is used as fuel. The properties of the tested fuels are shown in Table 2.

Table 1. Test engine specifications

Items	Specification
Model	S 195
Type	Single cylinder
Bore × stroke	95 × 115 mm
Rated output	9.8 kW / 2000 rpm
Compression ratio	20
Type of cooling	Water evaporative
Injection pressure	13.5 MPa

Table 2. Properties of tested fuel

Test property	Diesel fuel
Viscosity 25°C (cP)	6.8
Density @25°C (g/cc)	0.80
Heating value (MJ/kg)	44.5
Cetane number	49.0
Carbon mass (wt%)	86.8
Hydrogen (wt%)	13.1
Oxygen (wt%)	0.00
C/H ratio	6.63
Sulfur (wt%)	0.042
Total glycerin (%)	-
Free glycerin (%)	-
Distillation (°C)	
10%	232
50%	275
90%	334

Figure 1 shows a schematic diagram of the attachment of inlet air preheating system by exhaust gas and EGR system to the diesel engine. The rpm was measured directly from the tachometer attached with the engine shaft. The outlet temperature of cooling water and exhaust gas temperature were measured directly by using thermometers attached to the engine. Thermometer-1 was used for measuring inlet air temperature with air preheating and EGR system. Thermometer-2 and thermometer-3 were used for measuring exhaust gas temperature with and without air preheating system respectively. A counter flow heat low heat exchanger was used to heat the inlet air utilizing the heat of exhaust gases.

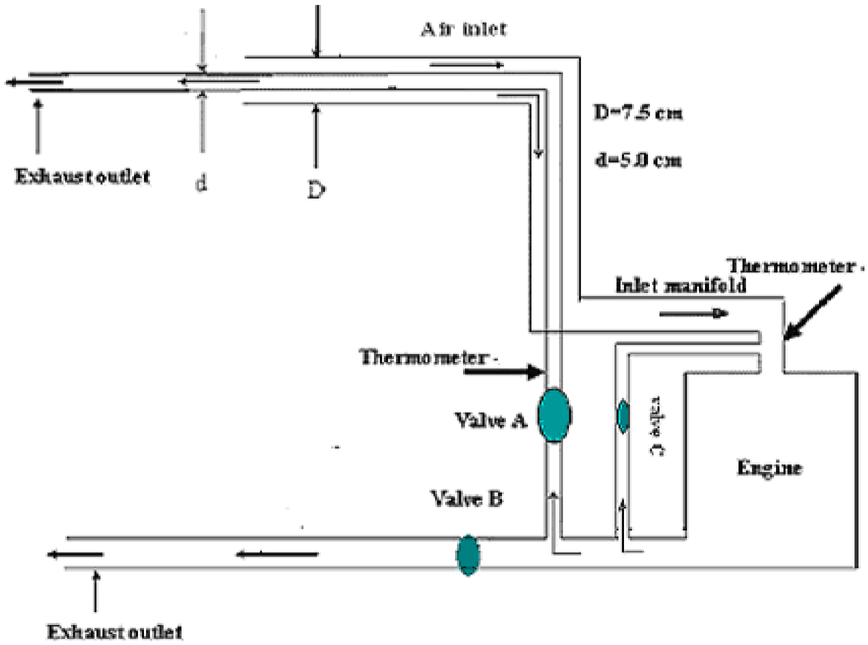

Fig. 1. Experimental setup

Prolonged exhaust pipe was surrounded with cylindrical inlet air passage in air preheating system. For extracting maximum heat from exhaust gases, both inlet and exhaust pipes were arranged concentric. To minimize heat lost to the atmosphere, the inlet passage was insulated by plaster of Paris, whose heat flow resistivity is comparatively higher [17]. In preheating attachment the exhaust gas outlet pipe and inlet air outlet pipe were 5.08 cm and 7.62 cm in diameter, respectively. These pipes were made of GP sheet having thickness of 0.45 mm. Three gate valves were used in the experimental setup. Gate valve A was used for alternative passage of exhaust gases while the gate valve B is used to control the direct passage of exhaust gases. Gate valve C was used to control the EGR system.

In case of without preheating of inlet air system, gate valve B was closed and gate valve A was opened. The inlet air (suction air) comes to engine cylinder through inlet and exhaust gases go out through the alternative exhaust passes. Again, to preheat the inlet air, gate valve B is opened and the gate valve A is closed. As a result, heated inlet air (suction air) becomes more heated as it comes to the engine cylinder. Air is heated through a long passage of inlet pipe surrounded by also a long exhaust pipe which conducts the heat transfer. By controlling the opening of gate valves A and B, the amount of preheating of inlet air was controlled. On the other hand for the combined effect of inlet air preheating and EGR system both valves A and valve C were opened and valve B was closed.

Figure 2 shows the heat exchanger. Overall heat transfer from exhaust gas to inlet air was calculated by the following equations [18]:

$$q = UA\Delta T_{overall} \tag{1}$$

where, A = area of heat flow,
U = overall heat transfer co-efficient and
ΔT = difference between average exhaust gases temperature (T_E) and average inlet air temperature (T_A)

U can be evaluated by the following equation:

$$U = \frac{1}{R_1 + R_2 + R_3}, \quad R_1 = \frac{1}{h_i A_i}, \quad R_2 = \frac{\ln(r_0 - r_i)}{2\pi kl},$$

$$\text{and } R_3 = \frac{1}{h_o A_o} \tag{2}$$

The overall heat transfer coefficient is calculated based on the inside area of the exhaust pipe. Setting the value of U in the above equation we have the following equation for q.

$$q = \frac{T_E - T_A}{1/h_i A_i + \ln(r_0/r_i)/2\pi kL + 1/h_0 A_0} \tag{3}$$

Where, h_i = convection heat transfer co-efficient of exhaust gases; h_0 = convection heat transfer co-efficient of inlet air; k = thermal conductivity of exhaust pipe material; A_o, A_i = outlet and inlet heat transfer area, respectively; r_0, r_i = outer and inner radius of the exhaust pipe, respectively; and L = length of heat transfer area.

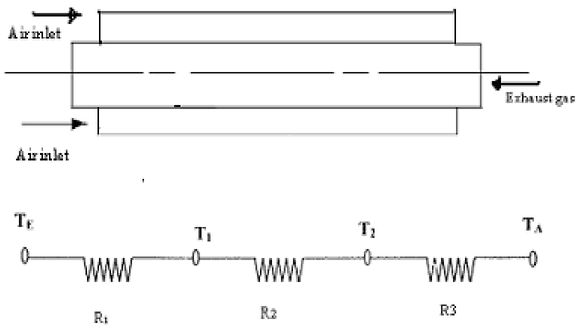

Fig. 2. Heat exchanger; T_A= Inlet air temp, T_E = Exhaust temp

The experiment was conducted without preheating and EGR attachment at first (normal condition) while inlet air temperature was recorded as 32°C. After warming period of the engine inlet air temperature was set at different elevated temperature (50°C, 55°C, 60°C) by controlling the amount of exhaust gas passing through the valve A and B, experiments were carried out individually with inlet air preheating arrangement and EGR system. Finally they were applied together to the engine to observe combine effects on engine performance. Only 25% EGR was applied to the test engine and this percentage of EGR has been fixed over the experiment as it was found that 25% EGR shows better result [19]. A digital exhaust gas analyzer (Table 3) was used to measure the exhaust emission parameters during the experimentation.

Table 3. Specification of gas analyzer

Items	Specification
Type	IMR 1400 Digital, portable
Calibration	Automatic zero point calibration after switch on calibration time 1 minute
Fuels	Oil light, natural gas, town gas, coal gas, liquid gas, coal and wood dry.
Gas probe	Heated probe with PTC resistor temperature 65ºc (Thermocouple Ni-Cr)
Gas hose	3-Way-Hose, Length 3.5 m
Air-probe	Integrated current sensor
Dust filter	Cellpor-filter, 4 micron
Operating temperature	-10 ºC to + 40 ºC
Power supply	Main 230V/50-60 Hz

3. COMPUTATION PROCEDURE FOR ADIABATIC FLAME TEMPERATURE AND NOx EMISSION

The combustion products of the fuels with C-H-O components were derived from the following equation for whole chemical reactions, which was confirmed by considering chemical equilibrium as will be mentioned later.

$$\begin{aligned} C_\alpha H_\beta O_\gamma + \varepsilon \frac{1}{\varphi}\left(O_2+\frac{0.79}{0.21}N_2\right) \rightarrow\ & y_1CO + y_2CO_2 \\ & + y_3O_2 + y_4H_2 + y_5H_2O + y_6OH \\ & + y_7H + y_8O + y_9NO + y_{10}N_2 \end{aligned} \quad (4)$$

Where, α: Mole number of carbon in unit mole of fuel,
β: Mole number of hydrogen in unit mole of fuel,
γ: Mole number of oxygen in unit mole of fuel.

$$\varepsilon = \frac{1}{\alpha + \beta/4 - \gamma/2} \quad (5)$$

Where, ϕ : Equivalence ratio,
Yi: mole fraction of each species in combustion products.

Mole fraction (y_i) and adiabatic flame temperature (T_b) were computed by considering the following chemical equilibrium and enthalpy balance in combustion reaction. NOx emissions were derived from y_9 in the forgoing equation for whole chemical reaction. These computations were carried out under the standard atmospheric condition.

$CO_2 \Leftrightarrow CO+1/2O_2$	(equilibrium constant K_1)
$H_2O \Leftrightarrow H_2+1/2O_2$	(equilibrium constant K_2)
$H_2O \Leftrightarrow 1/2H_2+OH$	(equilibrium constant K_3)
$1/2H_2 \Leftrightarrow H$	(equilibrium constant K_4)
$1/2O_2 \Leftrightarrow O$	(equilibrium constant K_5)
$1/2O_2+1/2N_2 \Leftrightarrow NO$	(equilibrium constant K_6)

The equilibrium constants were treated as a function of gas temperatures T. Adiabatic flame temperatures were computed by considering the enthalpy balance before and after the combustion. The gas mole enthalpies before combustion (H_u) and after combustion (H_b) were described by the following equations respectively.

$$Hu=H^0{}_{fuel}(T)+\frac{1}{\varepsilon\phi}\left[H^0{}_{O_2}(T)+\frac{0.79}{0.21}H^0{}_{N_2}(T)\right](J/mole) \quad (6)$$

$$Hb=\sum_{i=1}^{10}\left[y_i H^0{}_i(T_b)\right](J/mole) \quad (7)$$

$$H^0{}_i(T)=\int_{T_0}^{T} C_P{}^0 dT+\Delta_f H^0{}_i(T_0)\ (J/mole) \quad (8)$$

Where, $H^0{}_i(T)$: Mole enthalpy of each species (J/mole)
CP^0 : Specific heat at constant pressure of each species (J/mole/K)
T_0 : Initial atmospheric temperatures (K)

Thermodynamic properties of combustion gases in JANAF table [20] and the programs of Mizutani [21] and Ferguson [22] were used for the computations.

4. RESULTS AND DISCUSSIONS

Figure 3 shows the individual and combines effects of inlet air preheating and EGR to the engine on NO_x emission at different engine speeds with constant load of 10kg. In this case inlet air temperature was raised to 55ºC by controlling the quantity of exhaust gases passing through the heat exchanger. It is clear that NO_x emission increased with the increase of engine speed for all experimental conditions. It was observed that when both inlet air preheating and EGR are applied together to the engine, NO_x reduction was at lowest compared to other experimental conditions.. It may be due to shortened ignition delay and premixed combustion duration due to preheating inlet air while the EGR system reduced the temperature of the combustion chamber due to the higher specific heat of CO_2. This result is almost identical with the result in [23], where it was found that there is a significant difference on NOx emission when 20% EGR was applied to the engine.

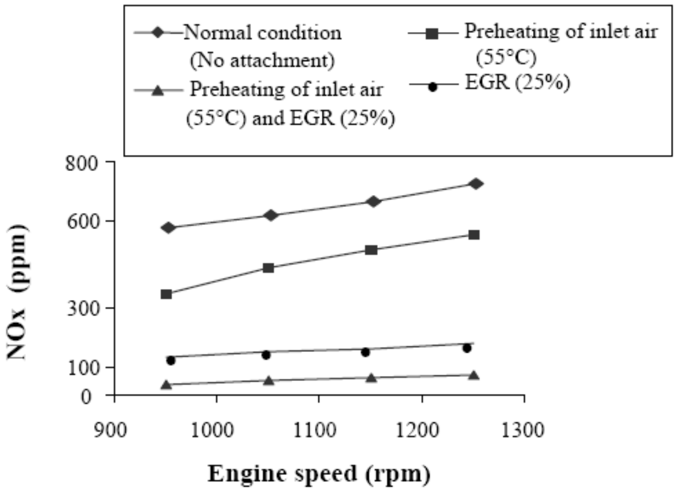

Fig. 3. Effect of inlet air preheating and EGR system on NOx Emission at different engine speed (Load 10kg).

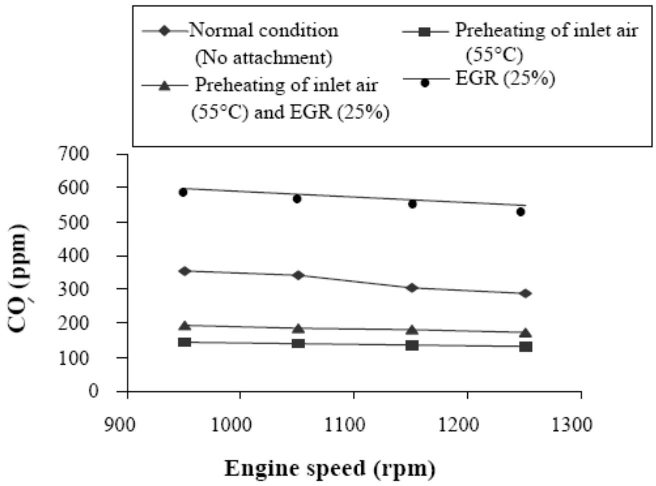

Fig. 4. Effect of inlet air preheating and EGR system on CO Emission at different engine speed (Load 10kg).

Figure 4 illustrates the effect of experimental attachments (inlet air preheating and EGR) on CO emission at different engine speeds with constant load (10kg) individually and together. In this case inlet air temperature was raised at 55°C as described above. It was observed that the lowest amount of CO was produced during the lonely effect of inlet air preheating system in engine combustion phenomena among the other conditions of experimental attachment, because inlet air preheating system provided better oxidation of CO. On the other hand, individual effect EGR system maximize the CO production because certain percentage of exhaust gases enter into the combustion chamber during suction stroke, so the combined effect of air preheating and EGR systems are not better in case of CO emission though, which is better for NOx emission. Now it is required to optimize the combination of the above systems to achieve lower NOx and CO emission simultaneously. Entropy Technology and Environmental Consultants, Inc [24] conducted research on effect of air preheating with the combination of flue gas recirculation (FGR). It showed that NOx emission reduced significantly with the combination of intake air preheating and FGR system.

Figure 5 depicts the effects of experimental attachments (inlet air preheating and EGR) on brake thermal efficiency at different engine speed with constant load (10kg). Here, the experiment was conducted at 55°C temperature of inlet air. Higher break thermal efficiency of the engine was found for the case of preheated inlet air compared to non-preheated inlet air. Brake thermal efficiency depends on BP (brake power), m_f (fuel consumption) and CV (calorific value of fuel). In the experiment a significant increase of brake thermal efficiency was found when inlet air preheating attachment was attached to the engine. Heated inlet air supplied larger amount of heat energy and provided better combustion of fuel. Inlet air was heated by exhaust gases energy, which ensured the overall lower amount of heat loss in heat balance sheet. As a result inlet air preheating system demonstrated some percentage of energy saving in diesel engine for the same power output of the normal condition. It was also evident from this figure that brake thermal efficiency increased with the increase of engine speed and beyond the limit of the optimum speed, engine efficiency decreased with the increase of engine speed. This is due to shorter air-fuel mixing time available at extremely higher speed. Finally it was also observed that when the inlet air temperature was increased the efficiency also increased at a certain speed.

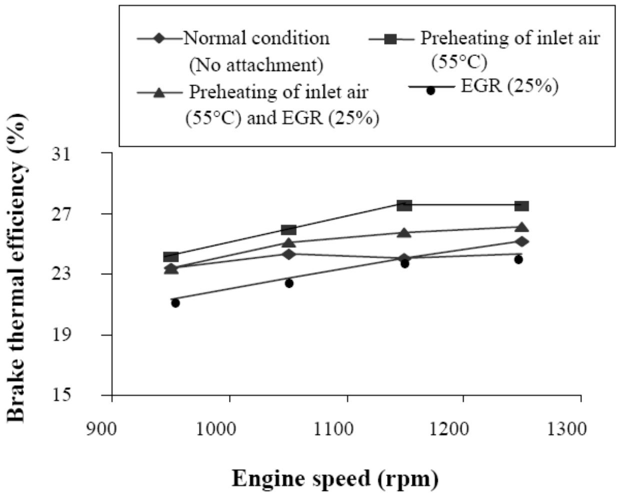

Fig. 5. Effect of inlet air preheating and EGR system on brake thermal efficiency at different engine speed (Load 10kg).

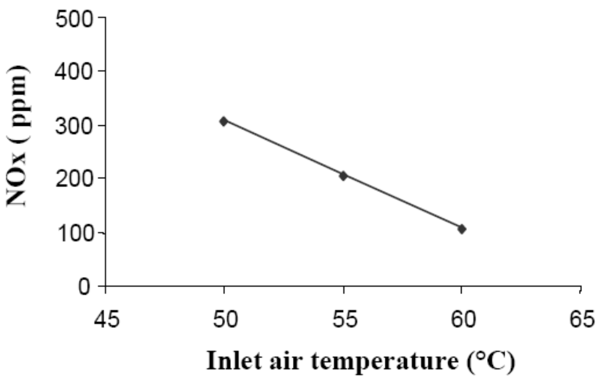

Fig. 6. Effect of inlet air temperature on NOx with neat diesel at constant engine speed (1150 rpm) and constant (Load 10kg).

Figure 6 displays the effect of inlet air temperature on NO_x emission at a constant load of 10kg and speed of 1150 rpm. The inlet air temperature was varied from 50ºC to 60ºC by maintaining proper flow of flue gases through heat exchanger. In this figure it was seen that with increasing inlet air temperature, NO_x emission gradually decreases. This may be due to higher inlet air temperature shortening the ignition delay period and duration of premixed combustion. Ishida *et al.* [25] showed that with the combination of EGR and air preheating with natural gas as fuel, NOx and smoke emission reduced markedly.

Figure 7 displays the effect of inlet air temperature on CO emission when the engine was run at a constant load of 10kg and speed of 1150 rpm. The inlet air temperature was varied from 50ºC to 60ºC as earlier. From this figure it was seen that CO emission gradually decreased with increasing inlet air temperature. It was also observed that at the maximum inlet air temperature, CO emission is minimum. This may be because higher inlet air temperature provided better air fuel mixing and higher rate of oxidation of CO.

Figure 8 displays the effect of inlet air temperature on brake thermal efficiency at a constant load of 10kg and speed of 1150 rpm. The inlet air temperature varied from 50ºC to 60ºC as earlier. It was clear from this that with increasing inlet air temperature brake thermal efficiency also increased. The reason for this may is that preheated inlet air provides better mixing of air and fuel in the combustion area giving lower CO production as well as the improving of combustion efficiency.

Figure 9 demonstrates the effect of adiabatic flame temperature on NOx emission with conventional diesel fuel. The figure is plotted based on the computed data. It can be seen from the figure that with the increase in equivalence ratio adiabatic flame temperature increases. The adiabatic flame temperature reaches for a maximum value at an equivalence ratio of 0.9 and then decreases. NOx emission shows the same trend as the adiabatic flame temperature. It is interesting to note that with the increase of adiabatic flame temperature NOx emission increases. However, NOx emission decreases with the decreases of adiabatic flame temperature. Though the computation data does not support experimental data quantitatively, it supports qualitatively. The experimental result of reduced NOx emission with EGR may be explained as the lower combustion flame temperature as seen from Figure 9.

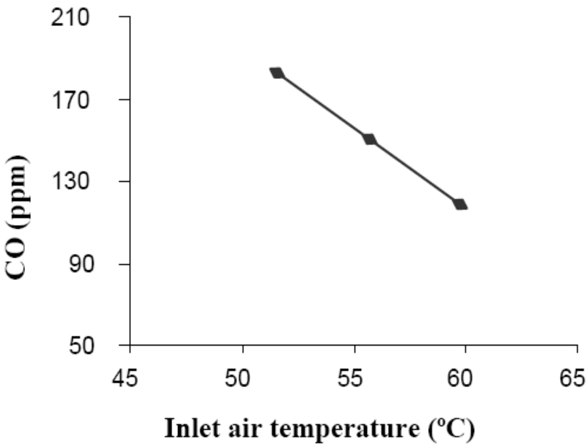

Fig. 7. Effect of inlet air temperature on CO emission with neat diesel at constant engine speed (1150 rpm) and constant load (10kg)

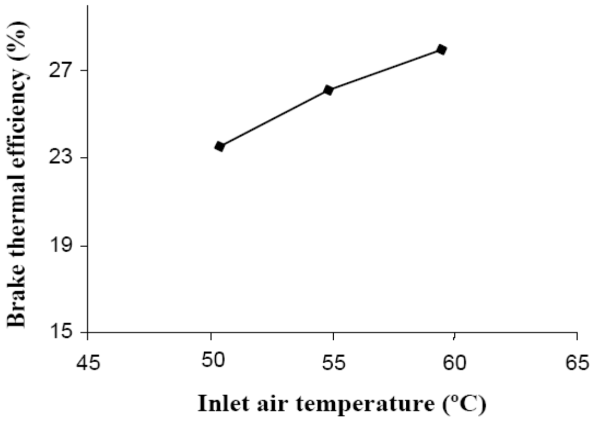

Fig. 8. Effect of inlet air temperature on break thermal efficiency with neat diesel at constant engine speed (1150 rpm) and constant load (10kg)

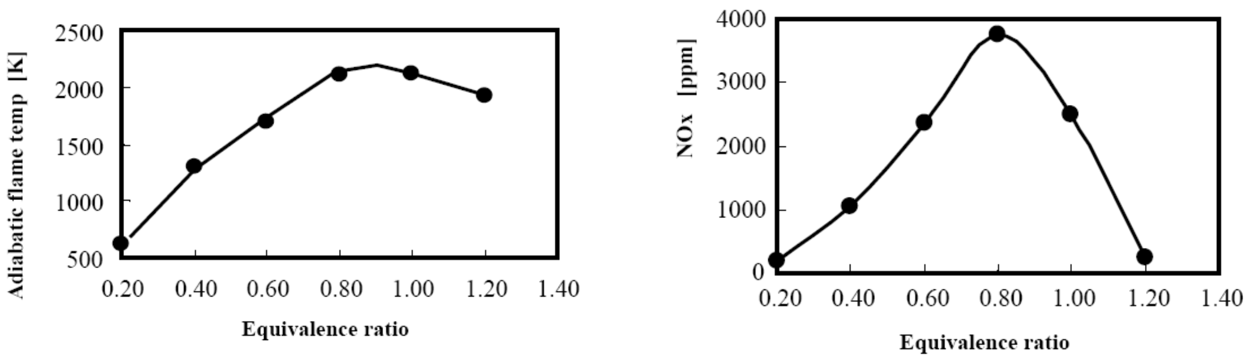

Fig. 9. (a) Adiabatic flame temperature; and (b) NOx emission with conventional diesel fuel

5. CONCLUSIONS

From this investigation, it is observed that there is positive effect of inlet air preheating and EGR with neat diesel fuel in diesel engine on engine performance and exhaust emissions, such as NO_X, and CO. Results of this study can be summarized as follows:

- By newly designed inlet air preheating system, it is possible to extract maximum 35% energy of exhaust gas energy during experiment.
- It is found that maximum amount of NO_X emission is reduced during the combination of EGR and inlet air temperature at 60°C by preheating system.
- It is also found that comparatively more CO emission can be controlled when only inlet air preheating system attached with the engine than others condition of attachment.
- It is notably observed that NO_X and CO formation in the combustion chamber decrease significantly with the increase of inlet air temperature.

- The experimental result of NOx emission is verified with that of simulated result.

NOMENCLATURE

BP	Brake power
BSFC	Brake specific fuel consumption
CV	Calorific value
C	Carbon
CO_2	Carbon dioxide
DI	Direct injection
KJ	Kilojoules
KW	Kilowatt
m_f	Mass of fuel
N	Engine speed
s	Second
NO_x	Oxides of nitrogen
η_{th}	Brake thermal efficiency

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