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THE EFFECT OF USING ARGON IN THE INTAKE CHARGE ON THE PERFORMANCE OF A DIESEL ENGINE UNDER SYNTHETIC ATMOSPHERE CONDITIONS

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Abstract

During the recent decades there have been substantial researches aimed to produce non-nuclear power systems for use in restricted fresh air supplies which has been identified by both the naval and the civilian communities. One particular class of power plant which has attracted a great deal of attention is the Air Independent Propulsion (AIP) systems.

The simplest of AIP systems is the Closed Cycle Diesel Engine (CCDE). A CCDE works like a regular Diesel engine, except that it uses synthetic atmosphere for the combustion of diesel fuel.

The present work is established with the overall objectives to determine the performance of diesel engine operating on non-air mixtures compared with the normal air, and the effect of Argon on the combustion of Diesel engine.

The test-rig with a single cylinder diesel engine was erected to evaluate the performance of the closed cycle diesel engine with Nitrogen or Argon as an inert gas balancing with CO_2 with different percentages and constant 21% O_2 by volume at different loads and constant speed. A computer simulation technique using single-zone model was used to predict the steady state performance for both closed cycle and normally aspirated diesel engine. The validation model has been used as a predictive tool for the study of Nitro-diesel closed cycles.

The deteriorating performance of the diesel engine when increasing the percentage of CO2 in the synthetic atmosphere is determined. Also the beneficial effects of using argon instead of nitrogen are analyzed.

KEY WORDS
Closed Cycle Diesel Engine, Non Air Mixtures, Air Independent Propulsion systems,

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NOMENCLATURE

Symbols

C_p	Specific heat at constant pressure (KJ/kg.K)
C_v	Specific heat at constant volume (KJ/kg.K)
CO_2	Carbon dioxide
h	Enthalpy (N.m/kg)
m	The mass contained in the cylinder (kg)
$m^{\cdot}_{int}$	The instantaneous mass flow rate through the inlet valve (kg/s)
$m^{\cdot}_{f}$	The rate of imaginary fuel addition (kg/s)
$m^{\cdot}_{ex}$	The instantaneous mass flow rate through the exhaust valve (kg/s)
$m_{f.burnt}$	Mass of fuel burnt (kg)
$m_{f.injected}$	Mass of fuel injected (kg)
N_2	Nitrogen
O_2	Oxygen
PC	Personal Computer
Q	Air Flow Rate (m^3 /s)
Q_i	Rate of heat transfer
X_f	The ratio between fuel burnt and fuel injected
x_M	Molecular weight of the synthetic air to the normal air
x_R	Gas constant of the synthetic air to the normal air
x_u	Internal energy of the synthetic air to the normal air

Abbreviation

AIP	Air Independent Propulsion
BMEP	Brake Mean Effective Pressure [bar]
BSFC	Brake Specific Fuel Consumption [gm/kW.hr]
CCDE	Closed Cycle Diesel Engine
WMS	Water Management System

1. INTRODUCTION

Over the past twenty years the need for non-nuclear power systems for use in the restricted fresh air conditions has been identified by both naval and civilian communities. One particular class of power plant which has attracted a great deal of attention is the Air Independent Propulsion (AIP) systems [1]. These are energy conversion systems capable of producing the necessary power output using synthetic instead of fresh air.

One of the greatest problems that limit the operation of a submarine is the dependence on air for its propulsion. For this reason there is a need for power systems that can fill the gap between the low performance battery systems and high performance nuclear reactors. The innovations recently introduced have opened the way to a new technology of non-nuclear submarines, the third generation, with a propulsion system independent of the air (AIP system).

The simplest form of AIP systems is the Closed Cycle Diesel Engine (CCDE) which works like a regular diesel engine but with synthetic air. The system employs a commercial diesel engine with adaptations to meet the special needs resulting from the closed cycle operation mode [2]. The exhaust gas leaving the diesel engine consists mainly of carbon dioxide, argon or nitrogen, water vapor and a small amount of unburned oxygen. After being cooled down (spray cooling), the exhaust gas is fed through the absorber which dissolves the carbon dioxide and condenses the water vapor. The absorber takes advantage of the high capacity of water to dissolve carbon dioxide and its lesser capacity to incorporate oxygen and argon or nitrogen, as shown in Fig. (1)

The residual gas mixture is enriched with oxygen and brought back to the engine's intake for a new combustion cycle. In order to maintain the adiabatic exponent of the intake gas similar to ambient air, a small amount of monatomic gas, Argon, is added.

The engine performance under closed cycle conditions is not totally documented or understood [3]. In the present work, which is mainly experimental, the Diesel engine performance is evaluated when running with synthetic atmosphere. The "quasi-steady filling and emptying" concept was employed to simulate the normal and synthetic operation of the diesel engine under steady state conditions.

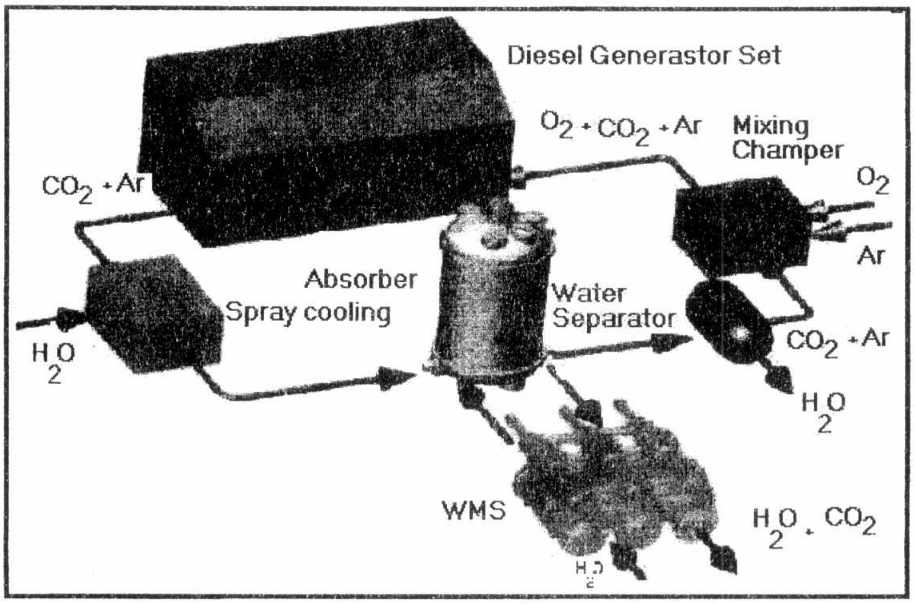

Fig. (1) Operation of Closed Cycle Diesel Engine

2. EXPERIMENTAL WORK

A single cylinder direct injection diesel engine (Deutz-F1L-511) with maximum power 7.7 kW at 1500 rpm and compression ratio 17 was used. The test rig includes all necessary instrumentation for measuring engine power, fuel consumption, air consumption, engine speed and exhaust temperature fig. (2).

The mixed gases (O_2, CO_2, and N_2 or Ar) are stored with different volume concentrations. The mixture is fed to the engine through an upstream surge tank of 0.2 m^3 volume. Fig.(3) gives a general scheme of the complete test rig showing numbered locations.

The engine speed was fixed at 1000 rpm and the engine load was changed from 0 to 4.7kW. The data acquisition system which is used in the present work consists of three main elements: BNC connector panel, PC based data acquisition card and a PC computer with data acquisition software. The data acquisition card is used to collect the measured data from the measuring instruments through the BNC connector panel directly or after amplification and converts the analog inputs to digital data which is then recorded by the computer under software control [4].

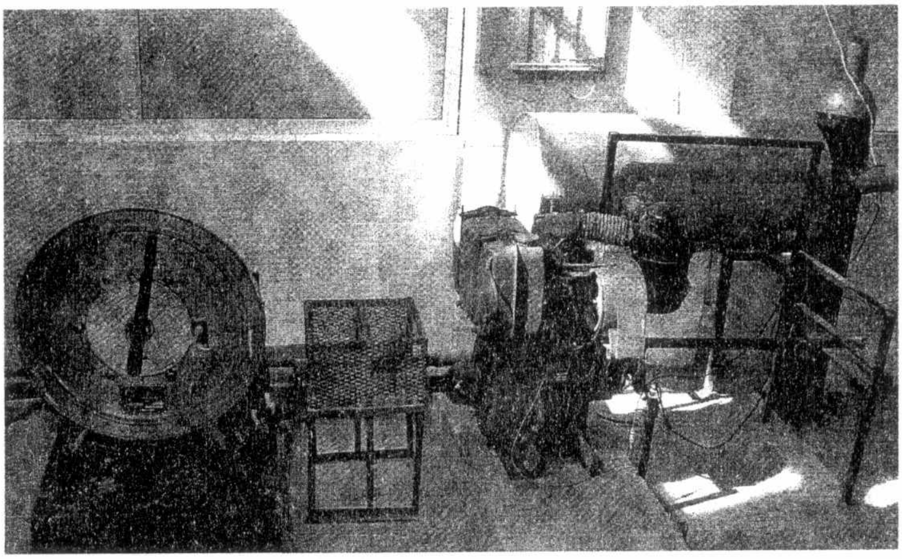

Fig.(2) The test rig of the experimental work

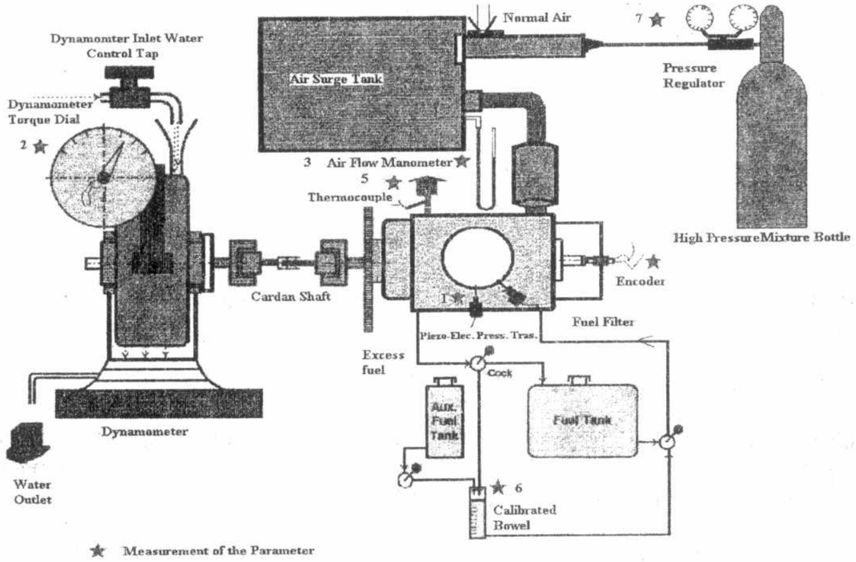

Fig. (3) Location of the measured parameters

Location	Parameters	Instrument used
1	Cylinder Pressure	Piezo-electrical transducer
2	Dynamometer Reading	hydraulic dynamometer
3	Air Flow Rate	Manometer
4	Crank Angle & Engine Speed	Encoder
5	Exhaust Temperature	Thermocouple
6	Fuel Flow Rate	flask of 10 cc volume
7	Cylinder Mixture Pressure & Outlet Pressure	Pressure regulator

3. SIMULATION TECHNIQUE

The concept of "Quasi-steady filling and emptying model" [5,6,7] was used in this work to simulate the closed cycle diesel engine under steady operating condition.

The engine is treated as a series of thermodynamic control volumes with the following components, as shown in fig.(3):

1. Engine cylinders, each bounded by the gas-exposed metal surfaces and the inlet and exhaust valves.

2. Exhaust manifolds, each bounded by the gas-exposed surfaces and exhaust valves.

3. Induction manifold bounded by the gas-exposed surfaces and inlet valves.

The laws of conservation of mass and energy were applied to each control volume under the following assumptions:

a. Thermodynamic equilibrium and ideal gas behavior exist at all times.

b. All control volumes contain homogeneous mixtures of air and combustion products. Therefore, phenomena such as temperature, entropy and fuel air ratio as well as pressure waves, non-equilibrium composition, etc. are ignored.

c. Mixing of the flow streams from different thermodynamic states and/or different chemical species, is treated as being instantaneous and perfect.

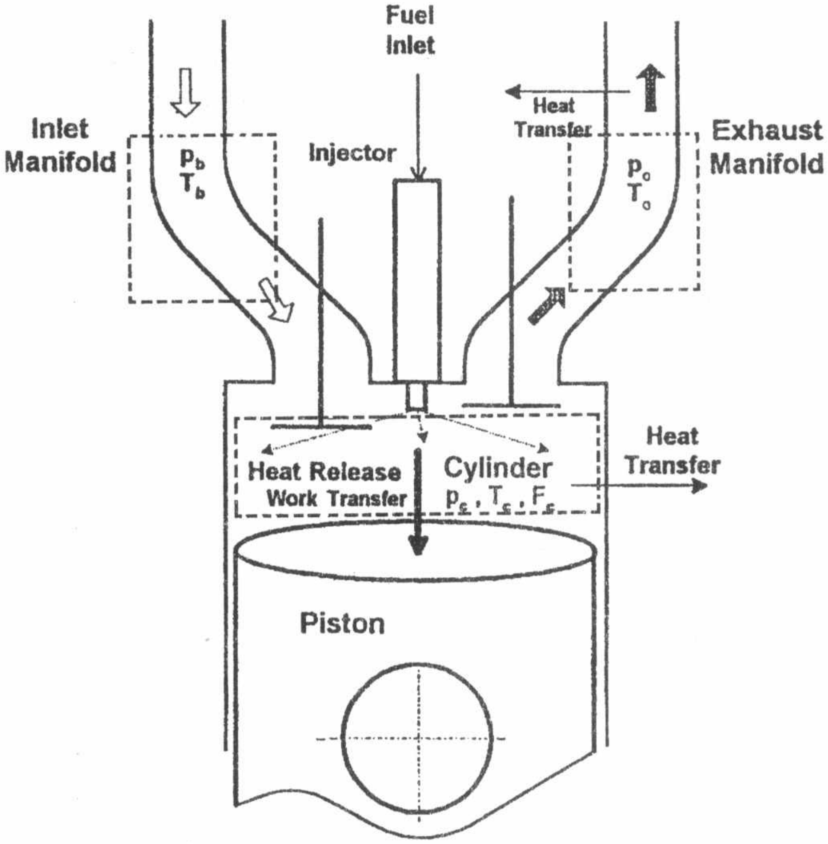

Fig. (4) Thermodynamic model of the engine

a - Continuity Equation

$$\frac{dm}{dt} = \dot{m}_{\text{int}} + \dot{m}_f - \dot{m}_{ex} \tag{3.1}$$

m	The mass contained in the cylinder (kg)
$\dot{m}_{\text{int}}$	The instantaneous mass flow rate through the intake valve (kg/s)
$\dot{m}_f$	The rate of the imaginary fuel addition (kg/s)

$m^{\bullet}_{ex}$	The instantaneous mass flow rate through the exhaust valve (kg/s)

The rate of change of the mass inside each control volume is equal to the sum of the rates at which mass flows in or out. Fuel is mathematically treated as being injected and burnt instantaneously

b. Energy Equation

$$\frac{d}{dt}(mu) = pV^{\bullet} + \sum_i Q_i^{\bullet} + \sum_j h_j m_j^{\bullet} \tag{3.2}$$

m	Mass inside the control volume (kg)
u	Specific internal energy of the medium (N.m/kg)
p	Pressure inside the volume (N/m^2)
$V^{\bullet}$	Rate of volume change (m^3/s)
$Q_i^{\bullet}$	Rate of heat transfer through surface i (N.m/s)
h_j	Specific enthalpy of the medium entering or leaving the control volume through the flow area j (N.m/kg)
$m_j^{\bullet}$	Rate of mass flow through the flow area j (kg/s)

The program was basically developed to simulate normally aspirated diesel engines. The working gas was assumed to be a homogeneous mixture of air and fuel. Thus, it was necessary to make some modifications.

The mixture properties were calculated at standard temperature and pressure from the properties of the constituting gases in the following procedure:

The calculated properties at different volume percentages of the constituting gases were then normalized with respect to normal air:

$$x_M = \frac{M}{M_{air}} \; ; \quad x_R = \frac{R}{R_{air}} \; ; \quad x_u = \frac{C_v}{C_{v.air}} \tag{3.3}$$

Figure (4) show the near linear dependence of the normalized factors (namely x_M, x_R, x_u) on the percentages of the constituting gases. Least square curve fitting technique was then used to correlate the normalized factors to the percentages of N_2, Ar and CO_2, which gave the following results:

$N_2\% + CO_2\% + Ar\% = 79\ \%$

$$x_M = 1.0 + 0.005548\ CO_2\% + 0.041413\ Ar\ \% \qquad (3.4)$$

$$x_R = 1.0 - 0.0019\ CO_2\% - 0.00512\ Ar\ \% \qquad (3.5)$$

$$x_u = 1.0 - 0.00119\ CO_2\% - 0.00594\ Ar\ \% \qquad (3.6)$$

Experimental data showed that the fuel consumption increases as the percentage of CO_2 in the mixture increases at the expense of Nitrogen or Argon. The ratio of the amount of fuel really burnt to that injected was estimated from experimental data to be as follows:

$$x_f = \frac{m_{f.burnt}}{m_{f.injected}} = (1 - 0.4\ \frac{CO_2\%}{100}) \qquad (3.7)$$

4. Experimental Results

The data obtained are those related to:

- Brake specific fuel consumption (BSFC), as a performance parameter.
- Cylinder pressure-crank angle history and the ensuing ignition delay period, burning duration, and maximum cylinder pressure, as combustion parameters.

Experimental work was carried out in two sets of experiments having the same test conditions but differs only in the composition of the aspirated air. The constituents of synthetic air in the first set included carbon dioxide, Nitrogen, and oxygen; while in the second set carbon dioxide, argon, and oxygen. In both sets the oxygen volumetric concentration was kept constant at 21%, while carbon dioxide volumetric concentration was altered to be from 0 to 30 %.

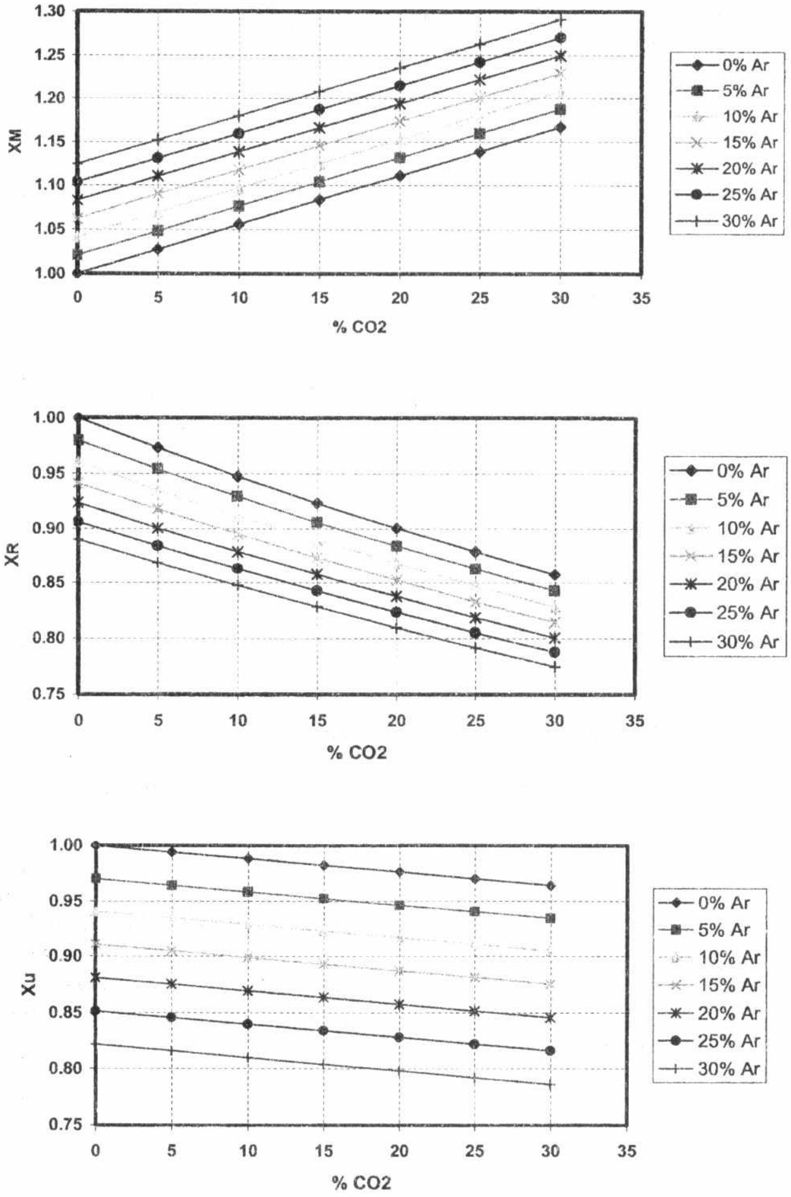

Fig.(5) Gas properties ratios for mixture of O_2, N_2, Ar and CO_2 (fixed O_2 21% by volume)

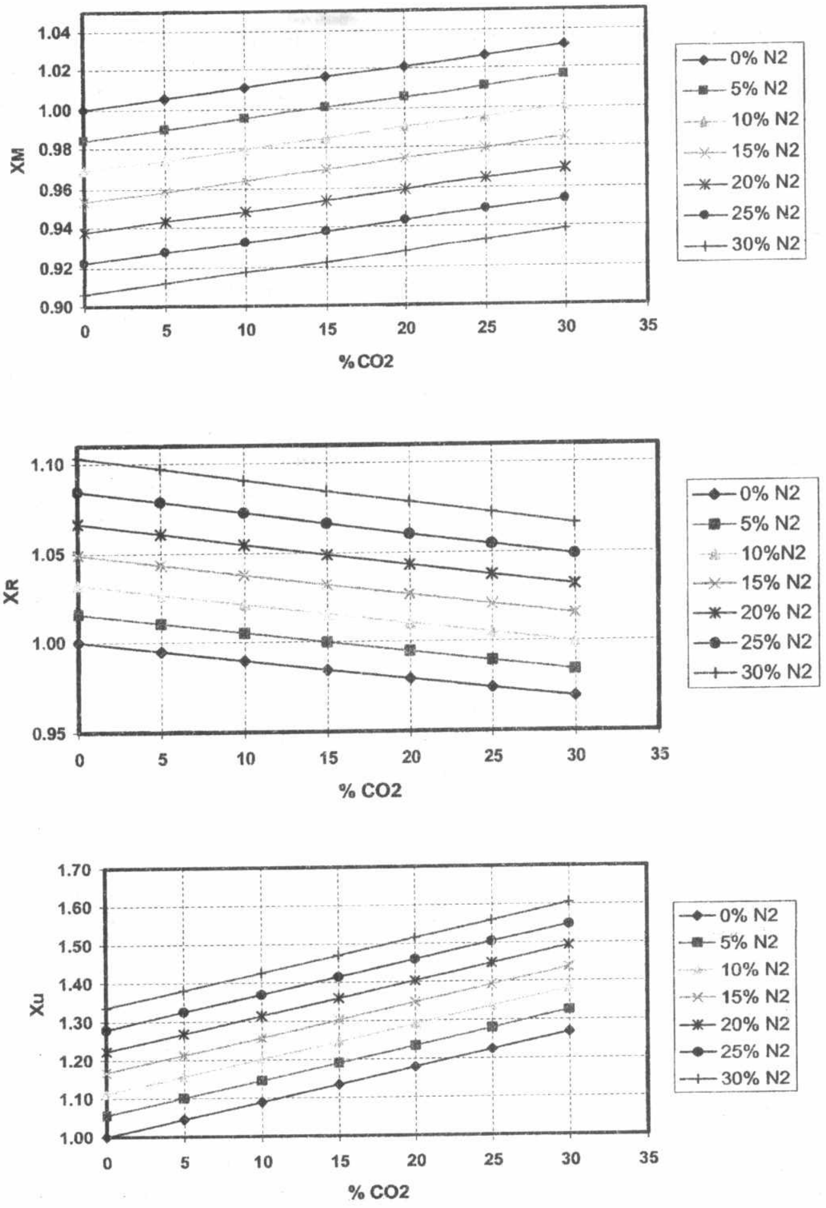

Fig.(6) Gas properties ratios for mixture of O_2, N_2, Ar and CO_2 (fixed O_2 21% by volume)

During the tests, the engine speed was fixed at 1000 rpm while the load was changed from 0 to 6.8 bar (BMEP). The measured values include engine speed, load, time of consumption of 5 cubic cm of fuel, air-orifice manometer reading, and exhaust temperature. These measured values were used to calculate engine power, torque, air flow rates, specific fuel consumption and air/fuel equivalence ratio.

4.1 Brake Specific Fuel Consumption (BSFC)

Figure (7) illustrates the variation of BSFC with CO_2 partially replacing nitrogen while fixing oxygen content at 21%. As shown from this figure, the higher CO_2 content, the more fuel is consumed (50 % increase as CO_2 rise from 5 to 30 % by volume with respect to normal air). This trend is more pronounced at lower engine loads due to the decline in mechanical efficiency. The increase in CO_2 concentration raises the mass of synthetic air, but simultaneously lowers its content volume specific heat with less extent. This combined effect leads to a colder combustion temperature and hence lower peak cylinder pressure, Fig.(7). Furthermore, the fuel is struggling to react with the oxygen and this leads to an increase in the BSFC. This causes more drop in the cylinder pressure, as shown in fig.(10) and hence output power. In order to maintain constant power, more fuel has to be injected to compensate the unburned fuel portion. The exhaust temperature is therefore seems to be unchanged, Fig. (15).

The effect of CO2 on BSFC in the second set, where argon replaces nitrogen and varying CO2 content at 21% oxygen, offers the same trend as the first set but with lesser extent (27 % increase as CO2 rise from 5 to 30 % by volume with respect to normal air). This is because although both mass and specific heat of synthetic air increase, Fig. (8), less reduction in the combustion temperature and pressure is observed, Fig.(16).

4.2 Cylinder pressure-crank angle history

The experimentally recorded data of cylinder pressure versus crank angle were analyzed to provide the peak cycle pressure for each test conditions. The recorded data were also fed to a computer program developed by Krieger and Borman. The program outputs were manipulated to calculate the ignition delay period and the burning duration.

4.2.1 Ignition delay period

The ignition delay, as shown in Fig.(11), considerably increases (by about 70%) as CO_2% increases from 5 to 30% concentration; the increased value is nearly the same for all loads. This longer delay may be attributed to the properties of CO_2 and also the drop in combustion peak pressure is less than that obtained from the motoring situation and results in a late burn conditions. CO_2 absorbs more energy than air in order to reach the same final temperature and hence the non-air mixture does not attain the auto ignition temperature at the same position of the piston. Subsequently, the onset of combustion is delayed which leads to incomplete burning of the fuel.

However, replacing nitrogen by argon (in the second set of experiments) offers somewhat different trend in the ignition delay versus CO2, Fig.(12). As shown from the figure, the delay period does not change when raising CO2 up to 25% concentration. This may be attributed to the faster chemical reactions caused by argon which may balance the adverse effect of CO2. Above 25% concentration of CO2, longer ignition delay is observed as the effect of CO2 in lowering the chemical reaction rates becomes more dominant.

4.2.2 Burning duration

Figure (13) shows the effect of CO_2 presence on the burning duration. The duration becomes shorter by about 14% as CO_2 raises from 0 to 20%. This may be due to the increased portion of unburned fuel during the diffusion mode of the combustion due to the presence of CO_2.

The decrease of burning duration in the second set of experiments becomes less about 20 % for the same range of CO2 variation, Fig.(14). This confirms the faster chemical reactions in the presence of argon that leads to improvement in the diffusion combustion mode. This justifies the improvement in the specific fuel consumption when replacing nitrogen by argon, as stated before.

5. Simulation results

An extensive series of simulation program runs have been carried out at different operating conditions. The test conditions were introduced into the simulation program as input data. The experimental results were then compared with those obtained from the simulation program.

The measured data along with those predicted from the simulation model for the specific fuel consumption and peak cylinder pressure are plotted in figures (17), (18), (19) and (20).

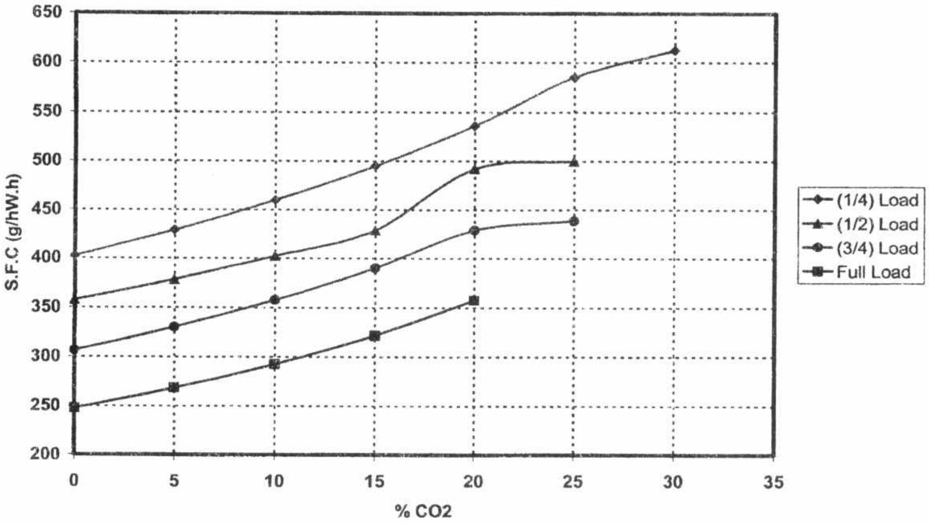

Fig. (7) B.S.F.C. 21% O_2+79%(N_2 + CO_2)

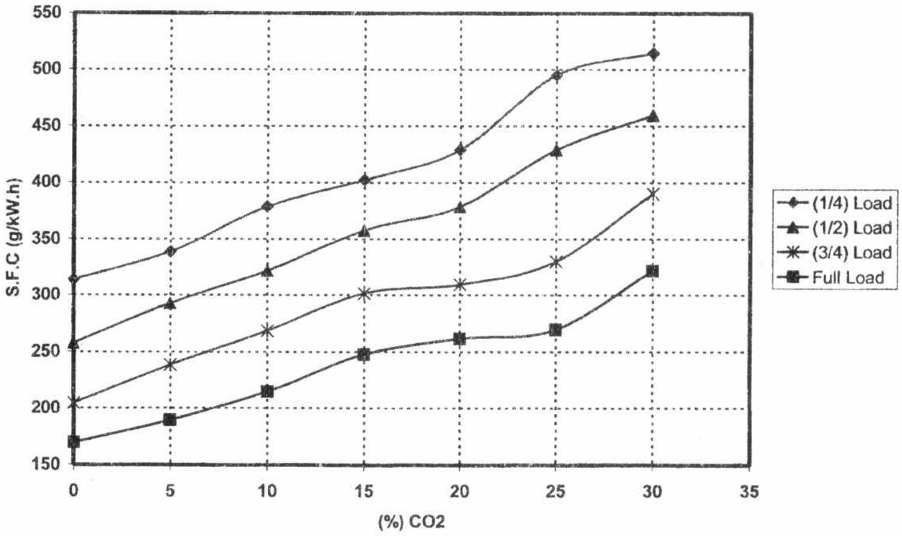

Fig. (8) B.S.F.C. - 21% O2+79%(Ar + CO2)

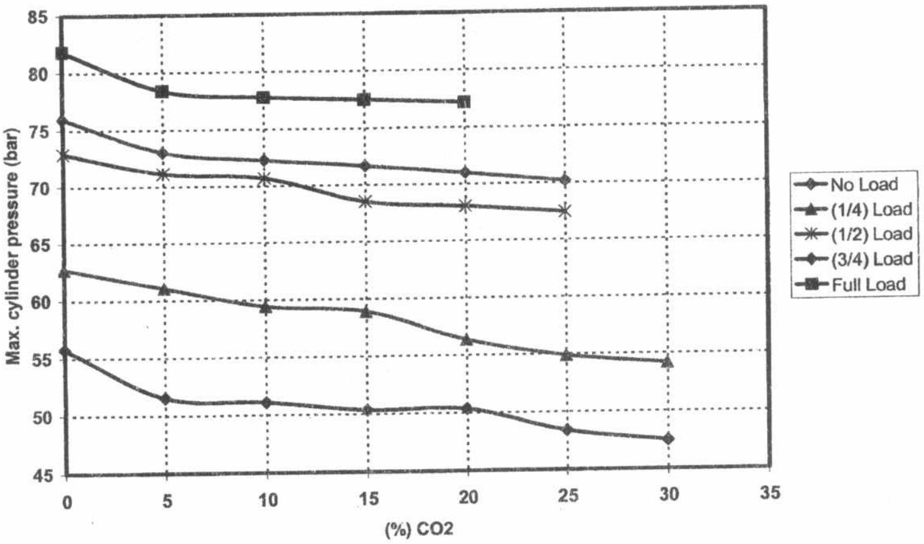

Fig. (9) Maximum pressure in cylinder - 21% O_2+79%(N_2 + CO_2)

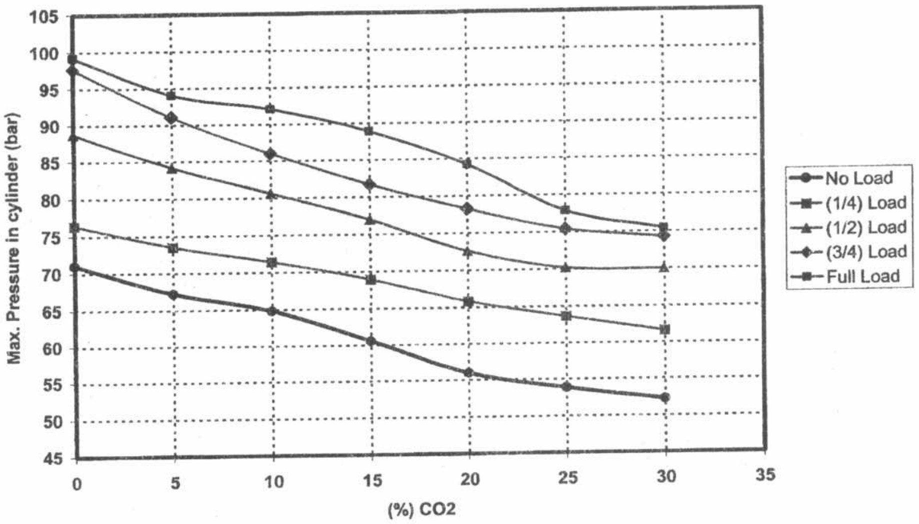

Fig. (10) Maximum pressure in cylinder - 21% O2+79%(Ar + CO2)

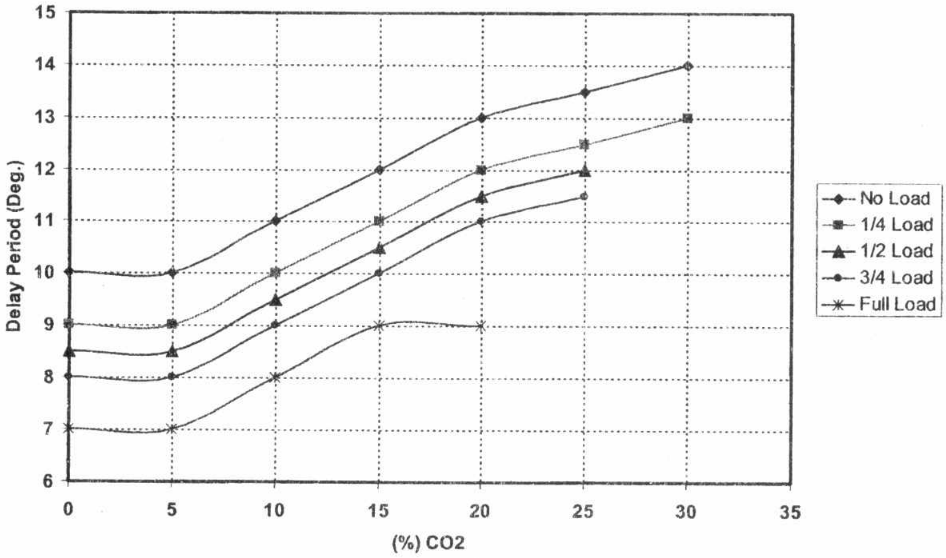

Fig. (11) Delay period - 21% O_2+79%(N_2 + CO_2)

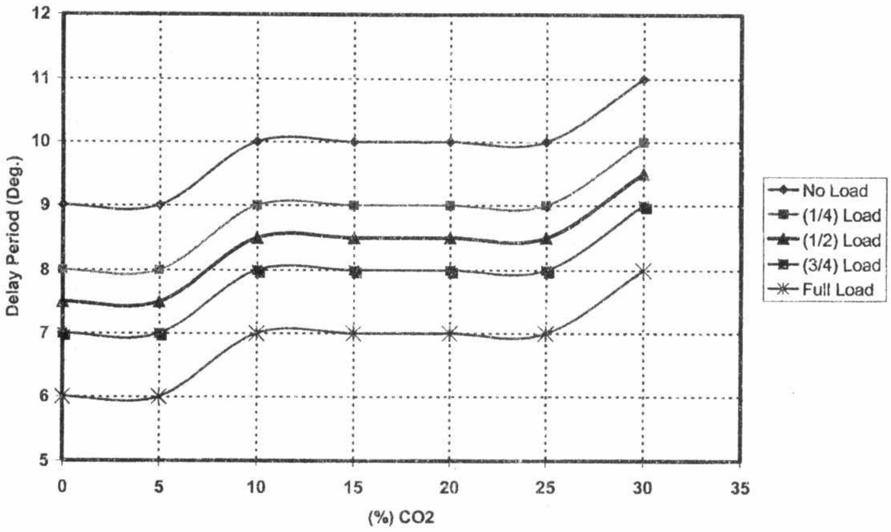

Fig. (12) Delay period - 21% O2+79%(Ar + CO2)

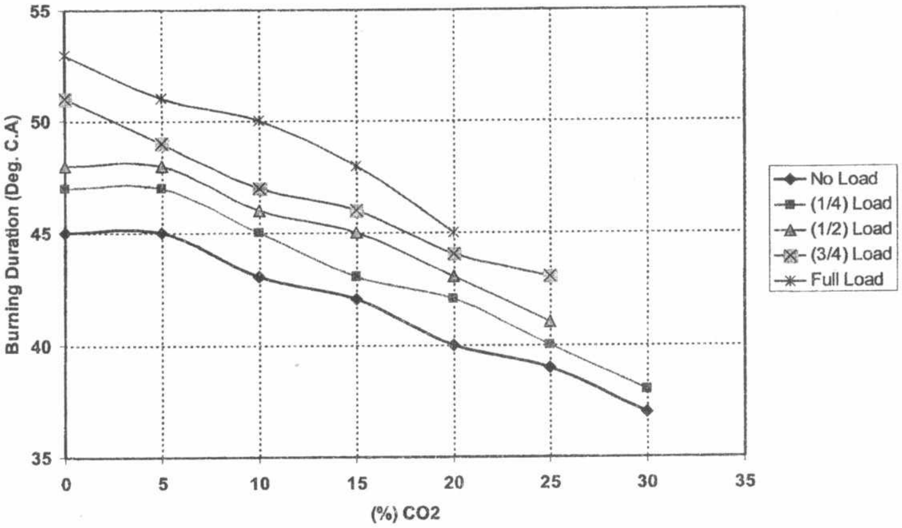

Fig. (13) Burning duration - 21% O_2+79%(N_2 + CO_2)

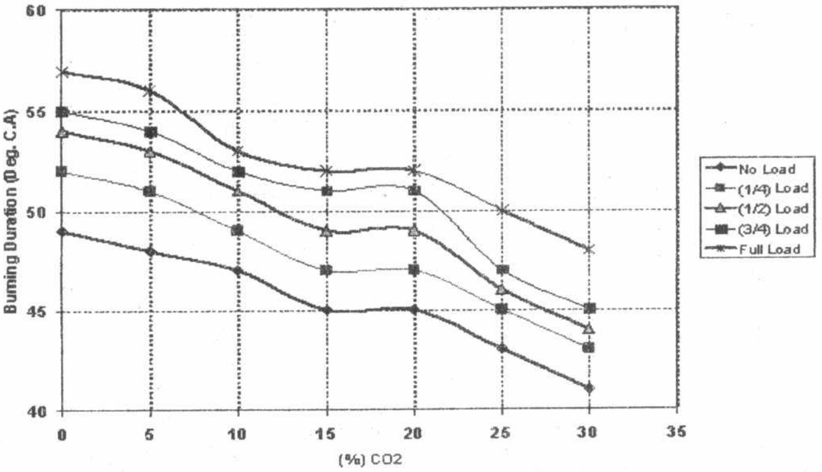

Fig. (14) Burning duration - 21% O2+79%(Ar + CO2)

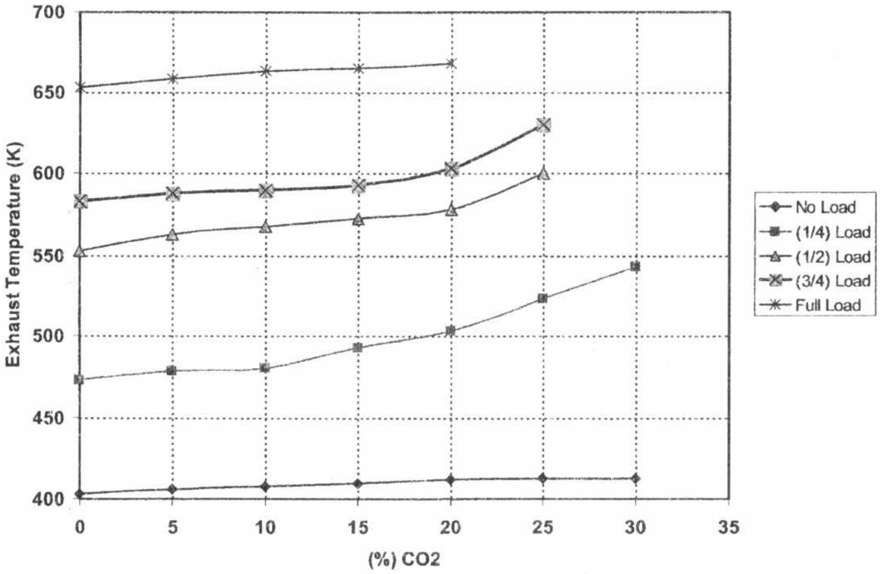

Fig. (15) Exhaust temperature - 21% O_2+79%(N_2 + CO_2)

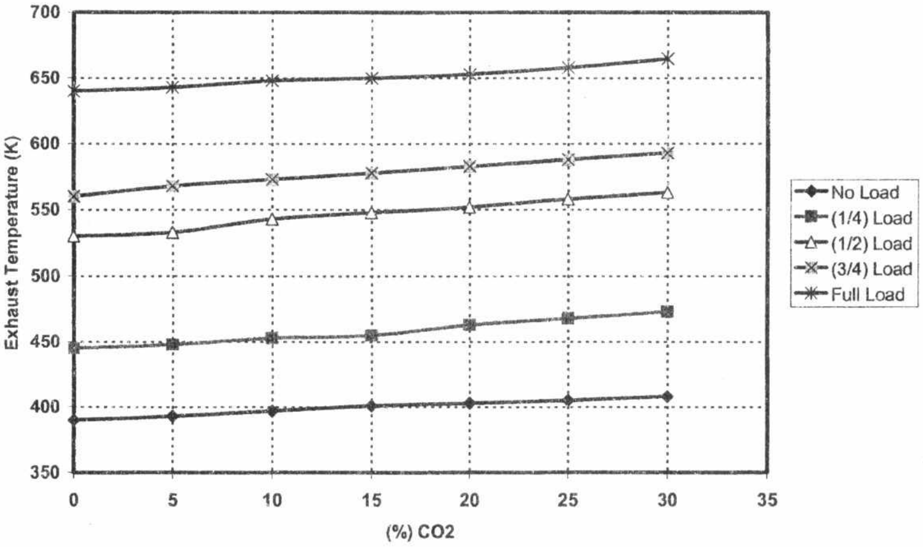

Fig. (16) Exhaust temperature - 21% O2+79%(Ar + CO2)

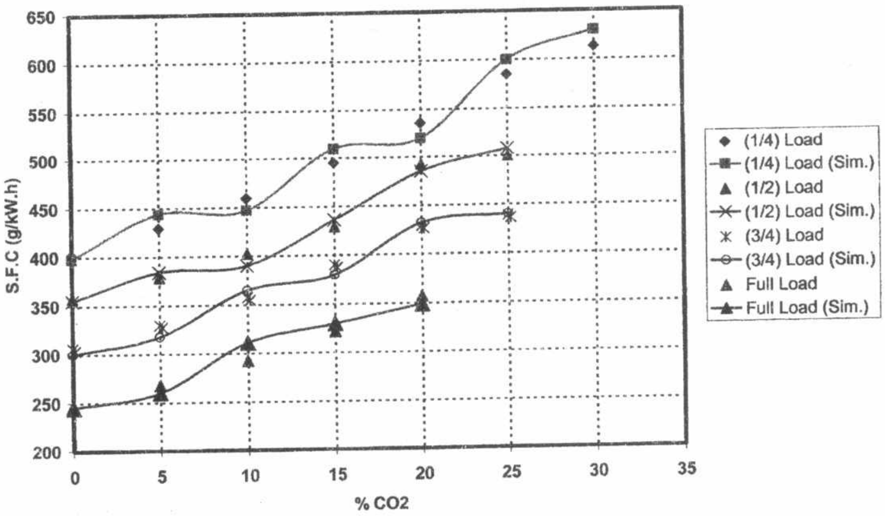

Fig. (17) B.S.F.C. - 21% O_2+79% (N_2 + CO_2)

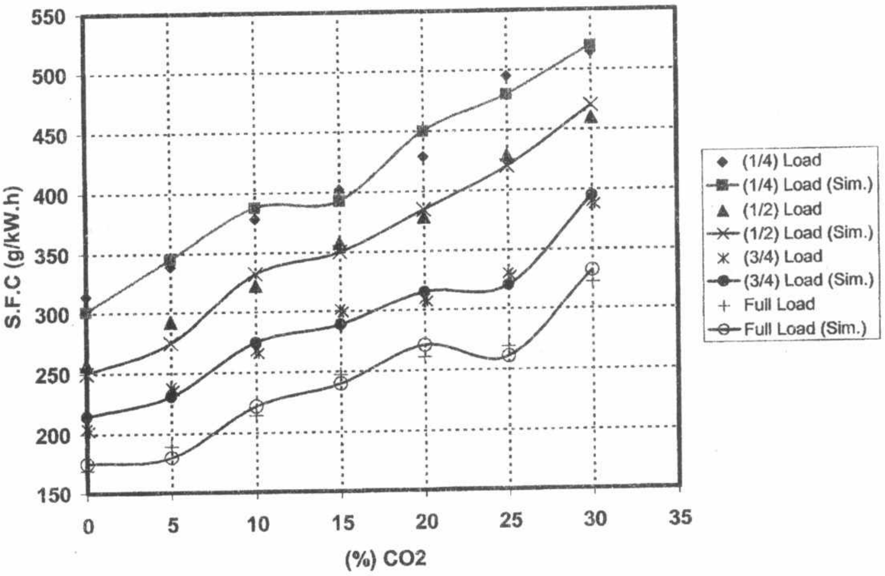

Fig.(18) B.S.F.C. - 21% O2+79% (Ar + CO2)

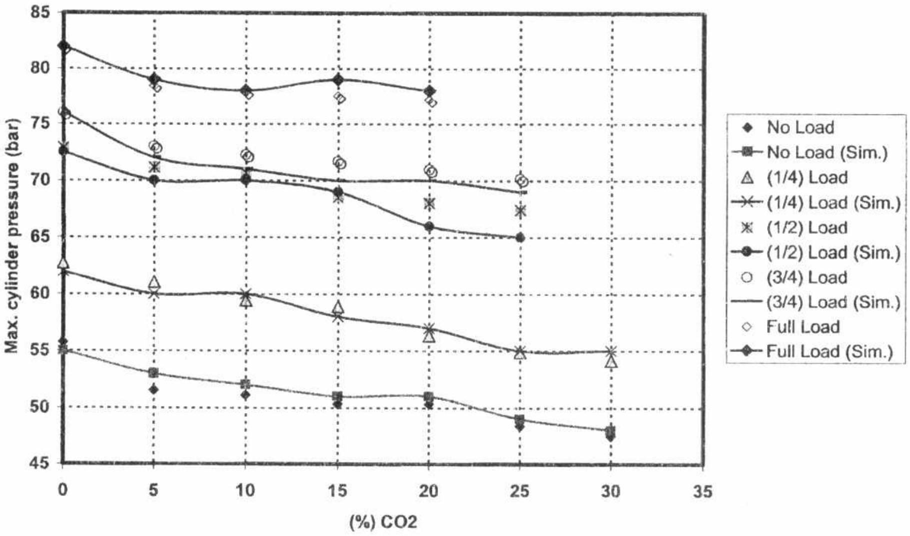

Fig.

(19) Maximum pressure in cylinder - 21% O_2+79% (N_2 + CO_2)

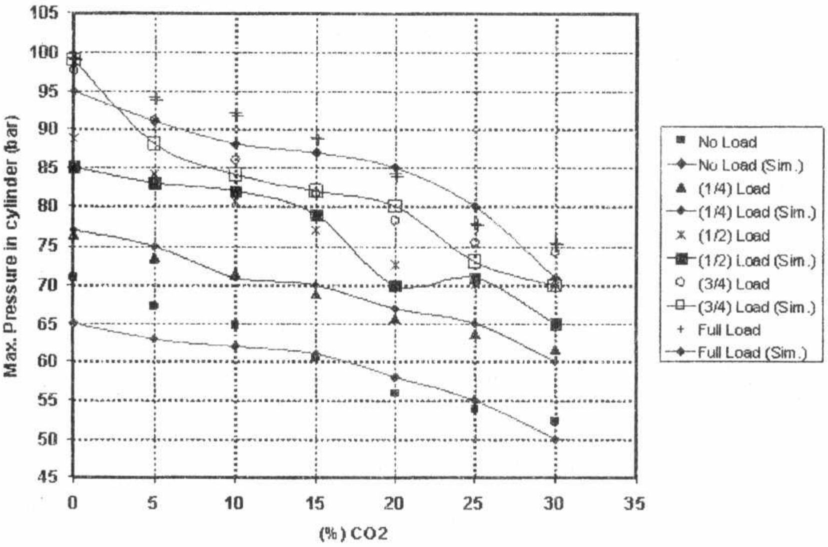

Fig.(20) Maximum pressure in cylinder - 21% O2+79% (Ar + CO2)

5. CONCLUSION

1- BSFC increases when CO2 replaces nitrogen while fixing oxygen content at 21% (50 % increasing as CO2 rise from 5 to 30 % by volume with respect to normal air). When argon replaces nitrogen and varying CO2 content at 21% oxygen, offers the same trend but with lesser extent (27 % increase as CO2 rise from 5 to 30 % by volume with respect to normal air).

2- Cylinder pressure decreases when CO2 replaces nitrogen while fixing oxygen content at 21%. When argon replaces nitrogen and varying CO2 content at 21% oxygen, offers the same trend but with lesser extent.

3- The ignition delay increases as CO2 goes up from 5 to 20% concentration when CO2 replaces nitrogen while fixing oxygen content at 21%. When argon replaces nitrogen and varying CO2 content at 21% oxygen, offers the same trend but with lesser extent.

4- The burning duration becomes shorter as CO2 raises when CO2 replaces air nitrogen while fixing oxygen content at 21%. When argon replaces nitrogen and varying CO2 content at 21% oxygen, offers the same trend but with lesser extent.

5- Argon improves the performance of closed cycle diesel engine.

6- The problem of CO2 contamination in the intake charge of the engine is mainly a thermodynamics problem.

7- The simulation program has proved that it is capable to be used as a predictor tool to obtain the performance of closed cycle diesel engine.

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PRODUCTION TECHNOLOGY

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