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EMULSIFIED FUEL PERFORMANCE ON COMPRESSION IGNITION ENGINE

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ABSTRACT

The research aims to show the performance of using emulsified fuel in a compression ignition engine. Diesel and four (4) fuel mixtures consisting of different proportions of diesel, bunker, water and catalyst were tested at various storage durations and engine parameters. A batch mixer was designed and fabricated to blend the fuel mixtures. It was observed that the thermal efficiency was lower when firing the engine with fuel mixtures of diesel, bunker, water and catalyst than when firing the engine with straight diesel. The research however, proves that the fuel mixtures can be used as alternative fuels without engine modifications. There is no substantial difference on the engine performance for all the tested fuels at various storage durations.

I. Introduction

The growing interest in emulsified fuels is anchored on the possibility of producing cheaper fuels from residual or crude oils or even from crude oil alternatives by improving its quality, which can be used in existing combustors. Another reason is the passage of stricter environmental standards that have led to various ways to control engine emissions. The development of technologies in emulsification could be an effective way of extracting more energy from oil-based fuel and achieving lower emissions.

Emulsification is usually achieved by mixing liquid fuel with water with the aid of additive called surfactant that keeps very small water droplets in suspension. The surfactant can also acts as a catalyst, when it increases the rate of chemical reactions. Upon combustion, heat causes the water droplets to abruptly turn into steam, resulting in micro-explosions. The micro-explosions promote better mixing between the fuel and the air, which requires lesser combustion time to burn. The addition of water to the fuel mixture may decrease the thermal efficiency due to the energy that goes into the latent heat of vaporization of the water on the emulsified fuel, unless there is a compensating improvement in combustion efficiency. In order to realize the benefits of using emulsified fuels, the size of the water droplets encapsulated with the fuel should be very small and the emulsion must be stable.

An accepted method of reducing nitrogen oxides (NOx) in diesel engine exhausts is the addition of water to the fuel charge either by directly injecting water into the combustion chamber or by emulsifying an amount of water in the fuel. Emulsification has been reported to achieve NOx reductions of 30% to as much as two thirds of the emission, depending on the amount of water added to the fuel charge [4]. Yamashita, et al. came up with the conclusion that the NOx reduction is proportional to the water content of the fuel emulsion while testing a high-speed diesel engine on emulsified fuel [10]. Nakajima, et al. also studied emulsified fuels by running a four-stroke, medium-speed marine diesel engine and found out that increasing the water ratio decreases NOx emissions regardless of engine operating conditions [8].

On a global scale, world economy is expected to grow at an annual average of 2.7 to 3.7% from 1996 to 2010 [2] and may further increase due to the expected improvement in economic activities, population growth, increasing income levels and urbanization, which all indicate increasing energy requirements. In the Philippine energy scene, the volume of energy imports has been continually increasing in the past years. In 1997, total energy imports amounted to 144.6 million barrels of fuel oil equivalent (MMBFOE) with oil accounting for 132.8 MMBFOE, while in 1990, imported energy and oil imports accounted for 79.3 and 76.2 MMBFOE, respectively [3]. This implies an 82.4% increase in energy imports and 74.2% in oil imports within a period of seven (7) years. Increasing concerns over the environment also have prompted the Philippine legislature to enact the Clean Air Act, a law that imposes stricter air quality standards in the whole Philippine archipelago. The use of alternative fuels such as soya oil [7] and emulsified fuels [6] would therefore, lessen the country's dependence on imported energy and at the same time be able to meet the stricter air quality requirements.

This paper reports the results of a study examining the effects of emulsification on the resulting fuel properties and on performance of a compression ignition engine. Four (4) different fuel mixtures consisting of different proportions of diesel fuel, bunker oil, water and catalyst were tested and compared with diesel fuel. Engine performance characteristics such as brake power, brake specific fuel consumption, thermal efficiency and engine exhaust emissions that resulted from the use of the fuel mixtures were determined. Fuel properties such as kinematic viscosity, heating value, pour point, flash point, percentage of sulfur and specific gravity that may affect performance and reliability were also determined. The engine performance tests were again conducted for all the fuels with different storage durations and at various speeds without any engine modifications.

A review of recent energy consumption patterns, oil prices, foreign monetary exchanges and political situation, requires that alternative fuels that can substitute or extend the lifetime of petroleum products be found or developed. An efficient emulsion can optimize the use of heavy petroleum fuels. However, the emulsified fuel must not deteriorate over a long period of time and does not impose major engine modifications. Alternative fuels must be suitable for new, as well as for old fuel burner designs lest it will be difficult to market such a product.

II. Methodology

Four (4) fuel mixtures (A, B, C and D) were prepared using a batch mixer [6]. Fuel A consisted of 60 kg of diesel, 30 kg of bunker, 10 kg of water and 0.3 kg of catalyst. Fuel B had 40 kg of diesel, 45 kg of bunker, 15 kg of water and 0.35 kg of catalyst. Fuel C contained 30 kg of diesel, 53 kg of bunker, 17 kg of water and 0.3 kg of catalyst. Fuel D composed of 20 kg of diesel, 60 kg of bunker, 20 kg of water and 0.3 kg of catalyst. The World Energy Extender Corporation supplied the bunker and the catalyst. Diesel fuel was purchased from a Shell Station. Ordinary tap water was used.

A batch mixer with a capacity of 200 liters was designed and fabricated. It is composed of a mixing drum, steel paddle and a 2 kW electric motor. Each batch contained the exact proportion by weight of diesel, bunker, water and catalyst as formulated above. The fuel mixture was thoroughly mixed for at least 30 minutes. Four (4) batches were mixed per fuel mixture. The first batch was immediately used while the second, third and fourth batches were stored at the laboratory. The second batch was fired after two (2) months. The third batch was considered after seven (7) months while the last batch was experimented after twelve (12) months.

The fuel mixtures were blended and the engine performance tests were performed at the Mechanical Engineering Laboratory of the Department of Mechanical Engineering, University of the Philippines, using a diesel engine. The apparatus consisted of a compression ignition engine with a prony brake, tachometer, multi-point thermometer, timer, Orsat gas analyzer and weighing scale. Lister-Blackstone, Inc. manufactured the type C, LB 142, diesel engine. It is a single cylinder with a bore of 114 mm, a stroke of 108 mm, a compression ratio of 14 is to 1, and water-cooled. The fuel is directly injected and the engine can be operated at different speeds.

The prony brake absorption dynamometer was connected to the engine flywheel. Two (2) metal strips formed into semi-circles were hinged at one end and bolted to each other at the other end. The strips were clipped around the flywheel by means of wooden cleats and tightened or loosened by adjusting the bolts to vary the load exerted on the engine. A brake arm of 1066.8 mm was attached to the strips on one end and to a pedestal mounted on a weighing scale on the other end. A type 75 weighing scale manufactured by Berkel was used to measure the load exerted.

The engine speed was determined by a cirscale tachometer manufactured by Record Company. The amount of fuel consumed was measured through a Toledo computergram weighing scale manufactured by Reliance Company. The weighing scale has a graduation of one (1) gm.

A digital multi-point thermometer manufactured by G. Cussons Company was deployed to measure the entering fuel temperature, exhaust gas temperature, outlet cooling water temperature and ambient temperature in °C. An Orsat Gas Analyzer made by Strohlein was connected to the exhaust pipe to determine the levels of carbon dioxide, oxygen and carbon monoxide in the exhaust gas in percentages.

The engine performance test was based on the Society of Automotive Engineers Diesel Engine Test Code [9]. There were five (5) independent variables: the fuel storage duration, the fuel type, the engine speed, the engine load, and the test run duration. The dependent variables measured were fuel consumed (gm); inlet fuel temperature (°C); outlet cooling water temperature (°C); exhaust gas temperature (°C); and exhaust gas levels of carbon dioxide, oxygen and carbon monoxide (%). The constant parameters were the brake arm length and the compression ratio.

Three (3) engine speeds of 800, 900 and 1,000 revolutions per minute (rpm) were applied for each fuel. The speed was kept constant for the duration of each experimental run by adjusting the fuel intake. The duration of an experimental run was fifteen (15) minutes. Readings were taken every three (3) minutes. There were six (6) up-reading runs and six (6) down-reading runs for each engine speed and fuel type. The load plus the tare weight was varied from 8 to 10.5 kg for the up reading and from 10.5 to 8 kg for the down reading in increments of 0.5 kg. This was done by tightening (for the up-reading) and loosening (for the down-reading) the nut, which holds the brake around the flywheel of the engine. The engine load was computed by subtracting the tare weight from the weighing scale reading. The tare weight was taken by averaging the values taken by manually turning the shaft of the engine clockwise and then counter-clockwise before and after the series of loads were run in the engine. The final readings are taken from the average of the up-reading and the down-reading values.

Frequent checks on the engine speed and engine load were done during the duration of each experimental run to make sure that the values were held constant. The engine was allowed to attain equilibrium conditions before each experimental run.

All of the fuels were made to undergo the same experimental procedures. The same experimental runs were performed for the fuels that were stored for two (2) months, seven (7) months and twelve (12) months. At least six (6) experimental runs were conducted per fuel mixture per time frame. A total of one hundred fifty six (156) runs were conducted. Figure 1 shows the flowchart of the experimental procedure.

The fuel characteristic tests were conducted at the Fuels and Appliance Testing Laboratory of the Department of Energy. The tests were based on the American Society for

Testing Materials (ASTM) [1]. The characteristics tested were heating value (ASTM D240), specific gravity (ASTM D1298), kinematic viscosity (ASTM D 445), pour point (ASTM D97), flash point (ASTM D93), and in percentage by weight of sulfur (ASTM D129). The samples were taken from the middle of each barrel containing the diesel and the various fuel mixtures. The fuel mixture samples were made to pass a 200-mesh or 75-micrometer sieve to determine the size of the water in the fuel mixtures.

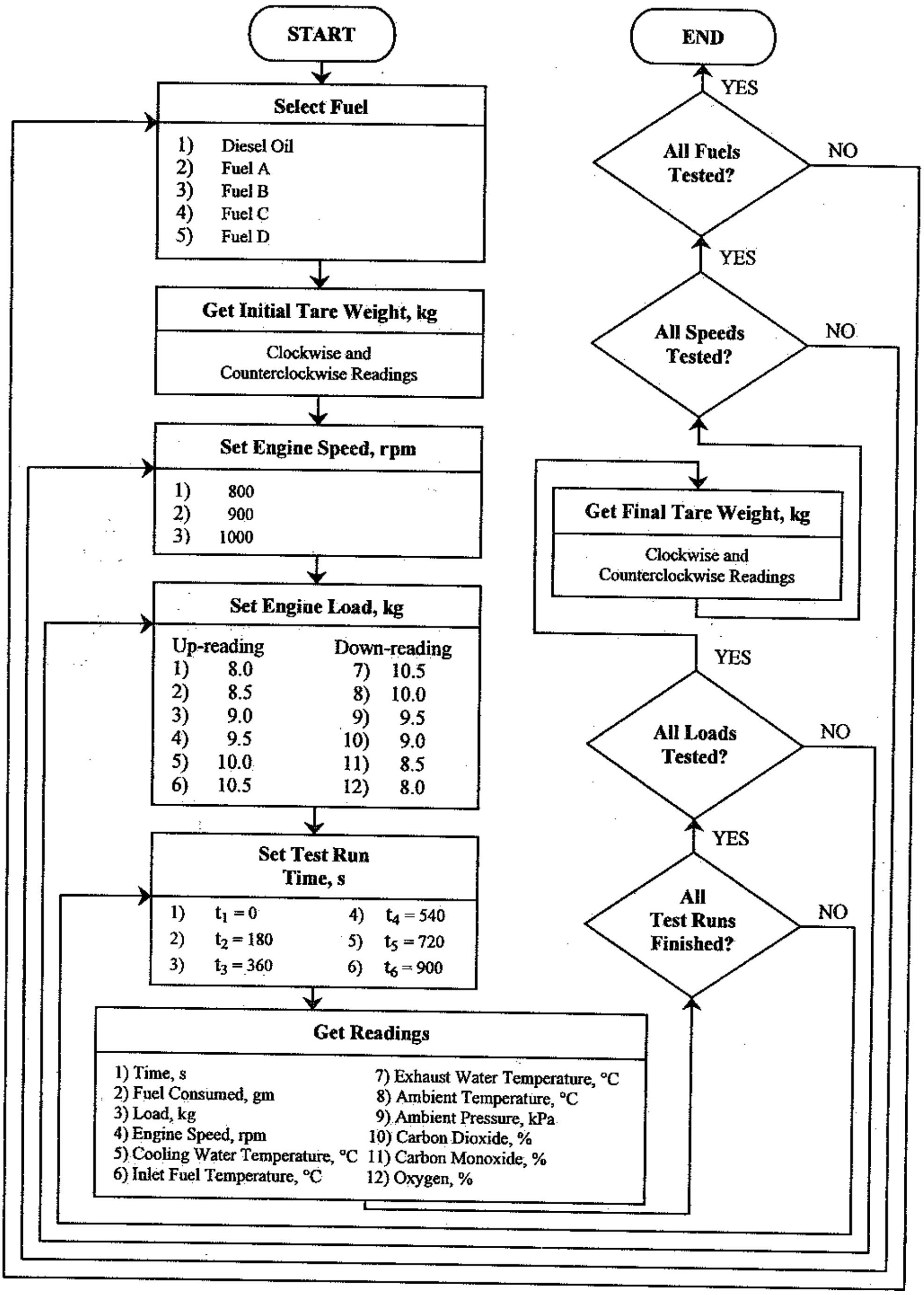

Figure 1. Flow diagram of engine performance test

III. Findings and Analysis

3.1 Fuel Properties

The results of the fuel characteristic tests of diesel and the various fuel mixtures are shown in Table 1. The values obtained are dependent on the sampling method used in getting the samples for testing.

Viscosity, which is a measure of the fuel's resistance to flow, affects the operation of the fuel injection system. The viscosities of the fuel mixtures are substantially higher than that of diesel, which has a value of 4 cSt at 40 °C. The viscosity of fuel A more than doubles that of diesel while those for fuels B, C and D are 14, 18 and 22 times that of diesel, respectively. The increases in viscosity may be accounted for by the addition of bunker oil, water and catalyst. Bunker oil has a viscosity of 612 cSt at 37.8 °C [6].

The heating value is a measure of the energy available from a fuel, which is essential when considering the engine's thermal efficiency. The addition of water and bunker to diesel resulted in heating values lower than that of diesel. The gross heating values of fuels A, B, C and D decreased by 361 kJ/kg, 1,247 kJ/kg, 1,829 kJ/kg and 4,032 kJ/kg, respectively. The heating value decreased by only a small amount with respect to the percentage of water added, e.g., diesel has 44,885 kJ/kg while fuel A has 44,525 kJ/kg or a decrease of about one (1) percent for an addition of about ten (10) percent water in the mixture. The other fuel mixtures used in the engine however did not exhibit the same trend because of the variations on the amount of diesel and bunker in the mixtures.

Pour point is an index of the lowest temperature that the fuel mixture can be used for certain applications. Except for fuel A that has relatively high diesel oil content, pour point increased with the addition of bunker and water. The increases, however, are not that significant to hinder easy engine start-up.

The flash points of the fuel mixtures follow the same trend as the pour points, i.e., it increased with the addition of bunker oil, water and catalyst to the diesel oil. Flash point is a measure of a fuel's flammability. It is the temperature at which the fuel vapors ignite in the presence of an open flame.

Specific gravity is normally an indicator of the fuel's heating value. With ordinary fuel, heating value increases with specific gravity. The values of the heating values in relation with the values of the specific gravity of the fuel mixtures used in this study did not follow this relationship. The addition of bunker, water and catalyst caused increases in the specific gravity of the fuel mixtures but resulted to decreasing heating values.

Measurement of the amount of sulfur in the fuel mixtures enables better determination of the products of combustion. The sulfur content increased because of the addition of bunker to the fuel mixtures.

The fuel mixtures were made to pass through a 75-micrometer sieve to determine the size of water droplets in the fuel mixtures. The size of the water droplets emulsified in the fuel ranged from 1 to 2 mm in diameter. It can be deduced that the mixing procedure used was not efficient, as the water droplets were not fully encapsulated in the fuel. It is necessary that the water droplets should range from about 0.0015 to 0.0075 mm in order for the water droplets to be fully enrobed with the fuel [5]. Another important factor is the stability of the emulsion. Separation should not occur because when the emulsion breakdown at elevated temperatures and the water droplets start to re-combine into bigger droplet sizes, the expected benefits from using the fuel emulsion will not be realized.

The catalyst appears like a greenish gel. The catalyst however was not tested in the laboratory.

Table 1
Fuel Characteristics

Parameters	Diesel	Fuel A	Fuel B	Fuel C	Fuel D
Kinematic Viscosity at 40 °C, cSt	4	11	57	72	89
Heating Value, kJ/kg	44886	44525	43639	43057	40854
Pour Point, °C	-6	-6	3	3	1
Flash Point, °C	88	88	>100	>100	>100
Sulfur, %	0.40	1.27	1.86	2.18	2.17
Specific Gravity	0.853	0.856	0.928	0.915	0.924

3.2 Engine Performance

The brake power in kilowatts (kW) is the rate of doing work and is the product of the torque and the rate of angular rotation. The brake thermal efficiency in percentage (%) is a measure of how much of the energy content of the fuel is converted to usable energy. It is the percentage quotient of the brake power and the product of the fuel flow rate and the heating value of the fuel. Figures 2, 3 and 4 compare the graphs of the brake thermal efficiency against brake power for diesel and the different fuel mixtures at 800 rpm, 900 rpm and 1,000 rpm respectively.

At 800 and 900 rpm, diesel fuel exhibited the highest efficiencies within the brake power range considered while fuel C exhibited the lowest efficiencies. However, at 1,000 rpm, fuels B and C gave higher efficiencies as compared to diesel, as shown in Figure 4. At an engine speed of 800 rpm and engine brake power of 2.5 kW, the engine delivered an efficiency of 16.9% using diesel, which is the highest value among the fuel mixtures. The efficiency of the engine using fuels B, D, A and C are 16.1%, 15.8%, 15.4% and 15.2%, respectively.

Generally, the thermal efficiencies obtained from the fuel mixtures are lower compared to diesel, which indicate that there were no improvements with the use of the fuel mixtures. However, with the addition of water and better blending, heavier fuel can be used with slightly lower efficiency.

The brake specific fuel consumption in kilograms per kilowatt-hour (kg/kWh) is the fuel flow rate necessary to produce a unit brake power. Figure 5 compares the graphs of the brake specific fuel consumption against brake power for diesel and the different fuel mixtures at 900 rpm. The graph indicates that the brake specific fuel consumption against brake power is slightly higher for all the fuel mixtures as compared to diesel. The other speeds follow the same trend.

3.3 Exhaust Emissions

Figures 6, 7 and 8 show the concentrations of CO_2, CO and O_2 respectively, from the engine at 900 rpm using the diesel and the four fuel mixtures. These figures show the typical behavior of the emissions from the engine at the three engine speeds used in the study. CO_2 concentrations increased linearly with the increasing brake power while O_2 concentrations decreased also linearly with the brake power. CO concentrations also increased with increasing brake power.

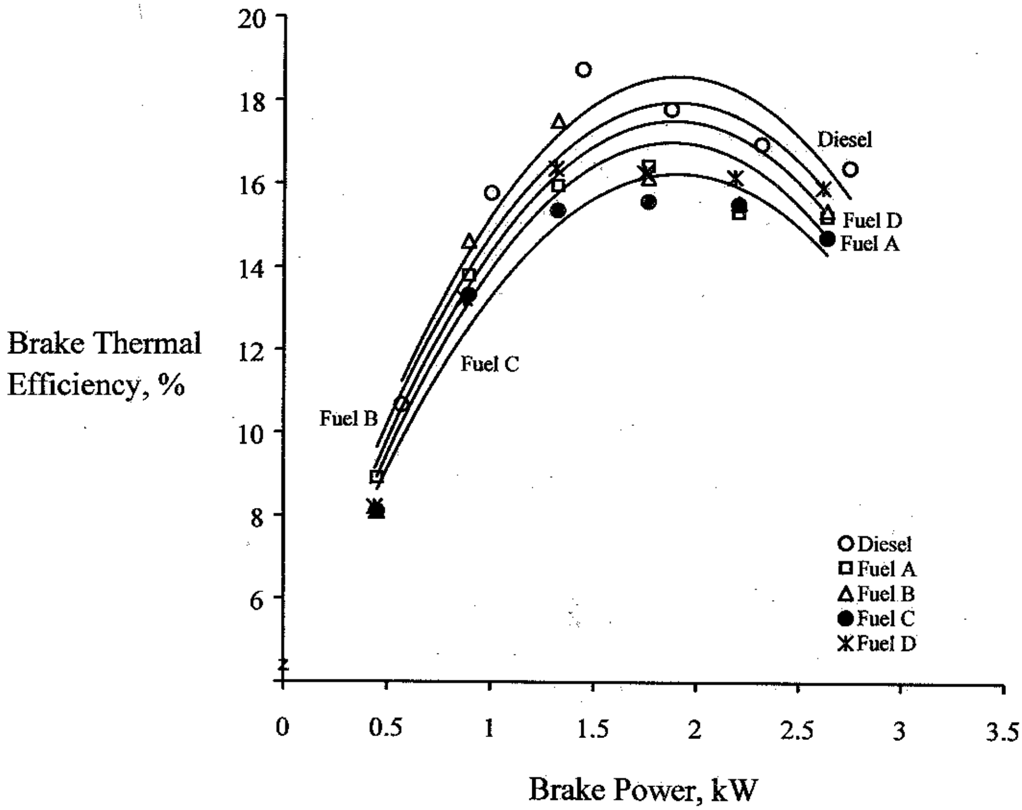

Figure 2. Brake thermal efficiency at engine speed of 800 rpm

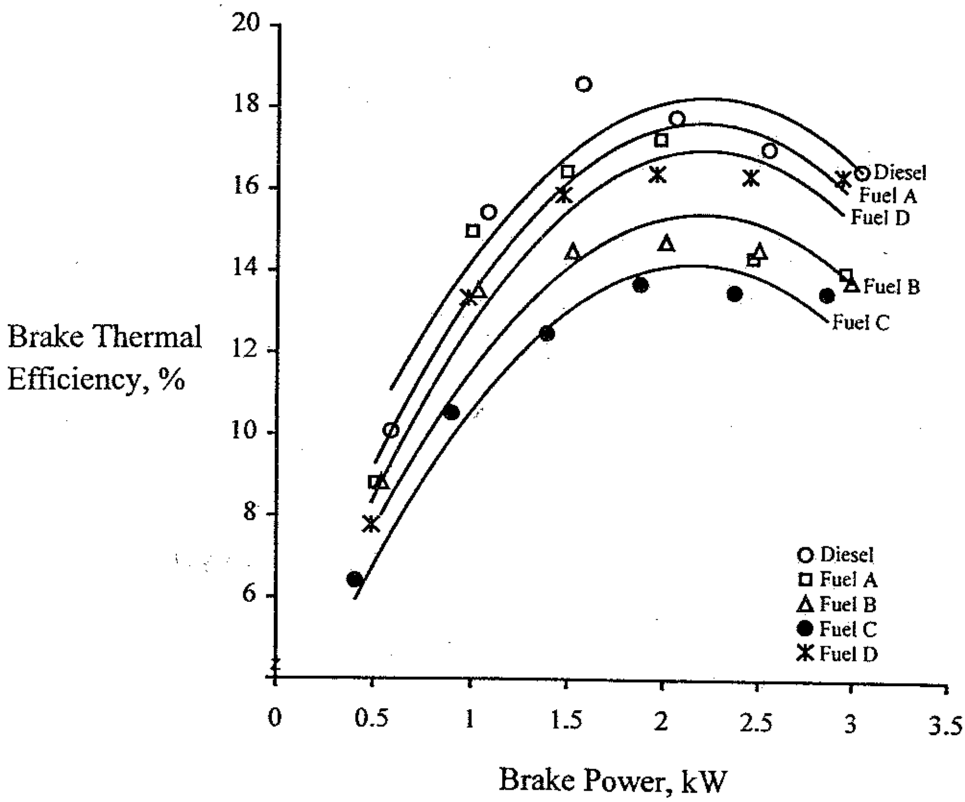

Figure 3. Brake thermal efficiency at engine speed of 900 rpm

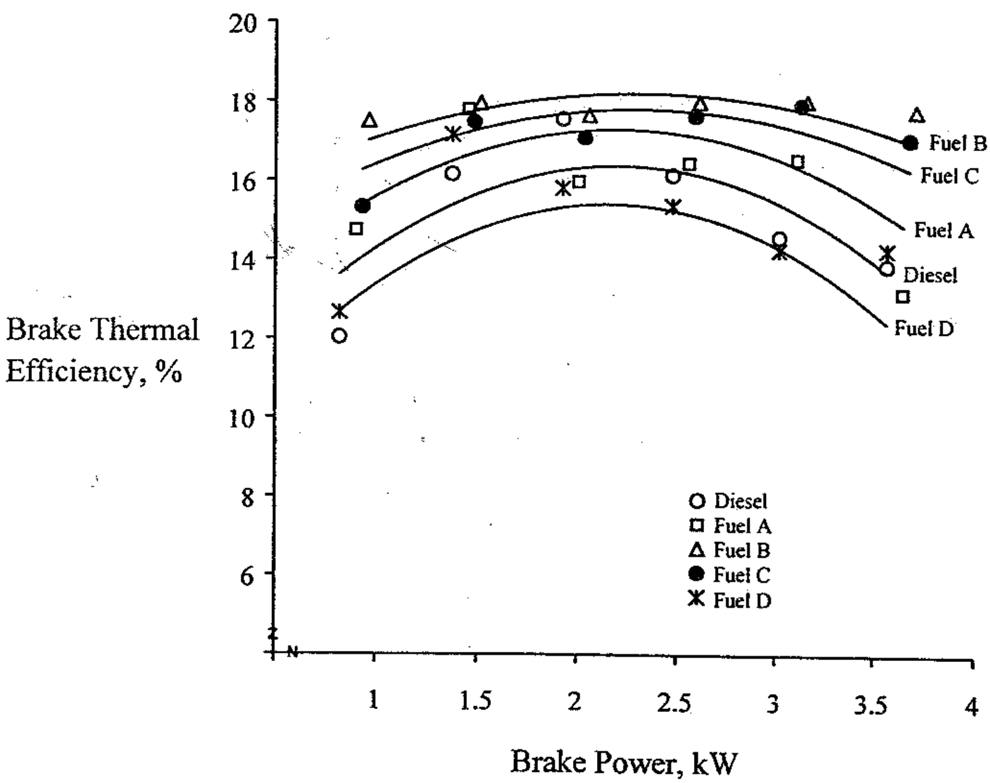

Figure 4. Brake thermal efficiency at engine speed of 1000 rpm

Carbon dioxide concentrations for the different fuels did not vary significantly at the same brake power values, as shown in Figure 6. Differences in carbon monoxide concentrations were also very small, not more than 1% at various brake power values for diesel and the four fuel mixtures as indicated in Figure 7. The difference in carbon monoxide is proportional to the amount of bunker in the fuel mixtures.

Significant differences in exhaust oxygen concentrations (Figure 8) can be observed between the diesel and the fuel mixtures at low brake power values. The differences in concentration, however, diminish as the brake power increases. Oxygen is substantially lower for fuel D as compared to diesel. During the experimental runs, fuel D emitted dense smoke. This indicates high particulate and/or hydrocarbon concentrations in the exhaust.

The graphs of the percentages of carbon dioxide against brake power show that the carbon dioxide concentrations at 800 rpm are higher than the carbon dioxide concentrations at 900 rpm while the carbon dioxide concentrations at 900 rpm are higher than those obtained at 1,000 rpm. The same trend is exhibited for the graphs of the percentages of carbon monoxide against brake power for the different engine speeds. The graphs of the percentages of oxygen against brake power show that the oxygen concentration at 800 rpm are lower than the oxygen concentration at 900 rpm while the oxygen concentration at 900 rpm are lower than those observed at 1,000 rpm.

3.4 Performance at Storage Durations

The storage durations are considered for the five (5) fuels, which were tested at four (4) different dates. In Figure 9, the deviations of the brake thermal efficiency values for diesel and fuels A, B, C and D with respect to storage durations are shown. Values of the brake power obtained from diesel and the four fuel mixtures all demonstrated deviations from their values obtained at zero time. Two months after the first run of tests, the percentage deviations of the brake thermal efficiencies for fuel B, diesel, fuels A, D and C from their initial values are -0.7%, 3.8%, 4.1%, 6.8% and 7.8%, respectively. Seven months after the first run, the percentage deviations of the brake thermal efficiency for diesel, fuels A, D, C and B are -2.7%, -3.0%, -3.3%, 4.1% and 8.7%, respectively. There is no substantial difference on performance twelve months after the first run as compared with the results obtained after seven months.

3.5 Maximum Efficiency and Selected Brake Power

The results of using different fuels on engine performance can be analyzed at selected brake power by determining and comparing the maximum brake thermal efficiencies. Figure 10 shows the maximum brake thermal efficiency (or minimum brake specific fuel consumption) at selected brake power for various engine speeds using the five fuels. A brake power of 2.5 kW was selected at engine speed of 800 rpm and a brake power of 3 kW was selected at engine speeds of 900 rpm and 1,000 rpm.

At brake power of 2.5 kW and engine speed of 800 rpm, the maximum brake thermal efficiency for using diesel is 16.9%; for fuel D, 16.1%; for fuel B, 15.8%; for fuel A, 15.4%; and for fuel C, 15.2%. At a brake power of 3 kW and engine speed of 900 rpm, the maximum brake thermal efficiencies are as follows: 17.4% for diesel; 16.4% for fuel D; 14.7% for fuel A; 14.4% for fuel C; and 14.2% for fuel B. At 3 kW and 1,000 rpm, the maximum brake thermal efficiencies are 22.5%, 17.9%, 16.8%, 16% and 14.8% for fuels B, C, diesel, A and D, respectively.

Another approach to analyze the results of the engine performances is by looking at the brake power at maximum brake thermal efficiency. Figure 11 shows the result of this approach. The figure gives the optimal brake power values that were obtained from the engine at the highest efficiencies for the various fuels at 800, 900 and 1,000 rpm.

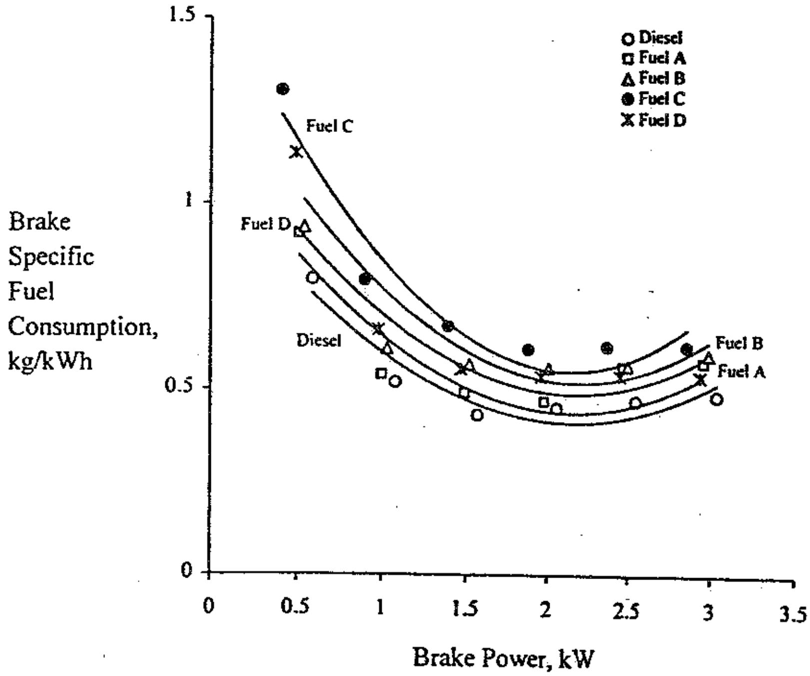

Figure 5. Brake specific fuel consumption at engine at speed of 900 rpm

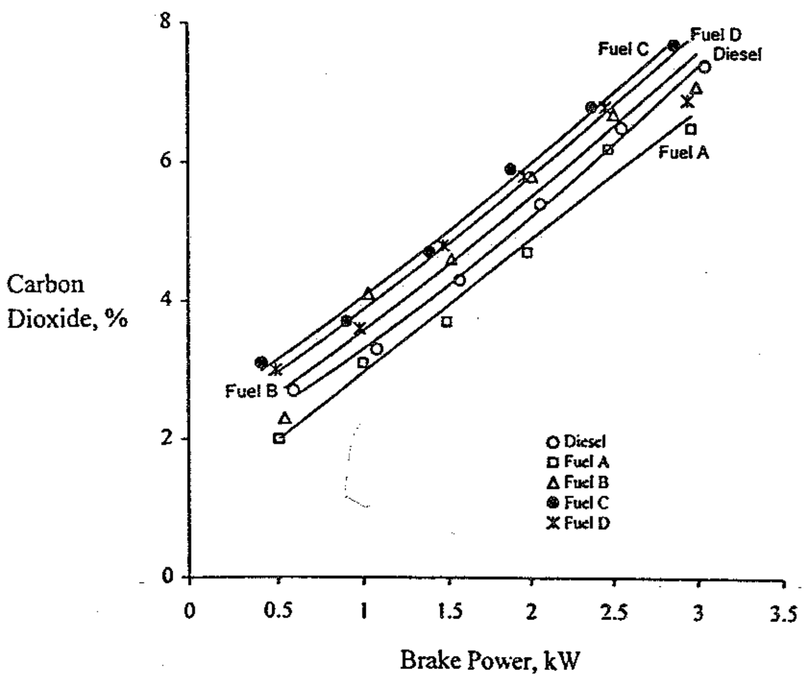

Figure 6. Carbon dioxide emission at engine speed of 900 rpm

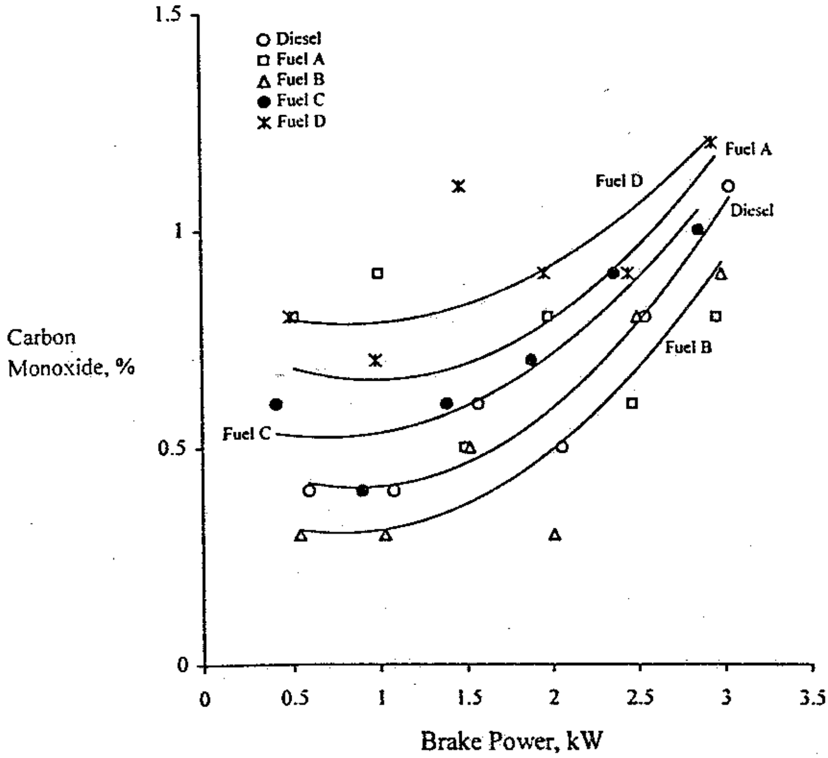

Figure 7. Carbon monoxide emission at engine speed of 900 rpm

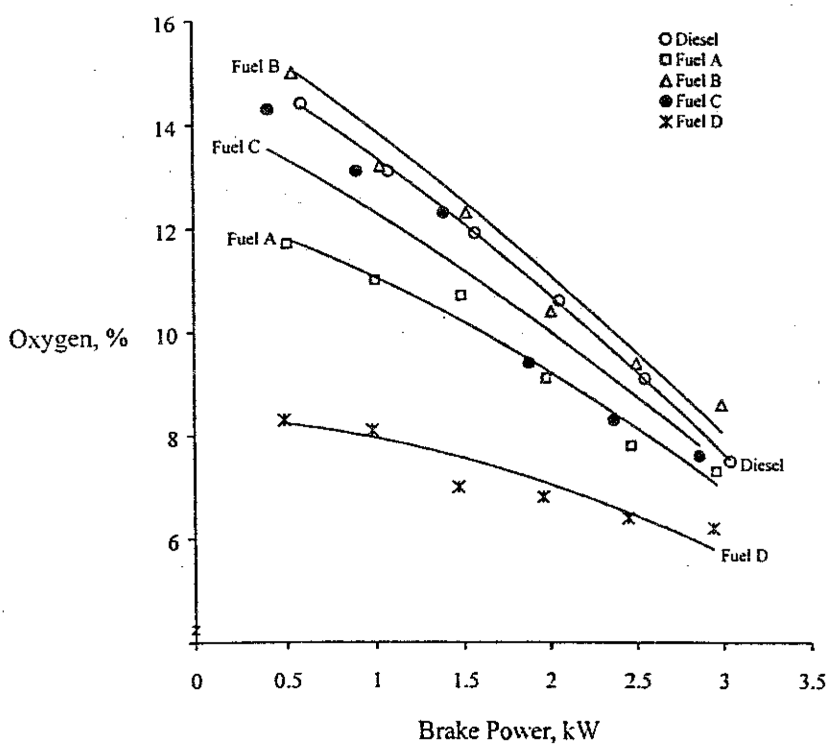

Figure 8. Oxygen emission at engine speed of 900 rpm

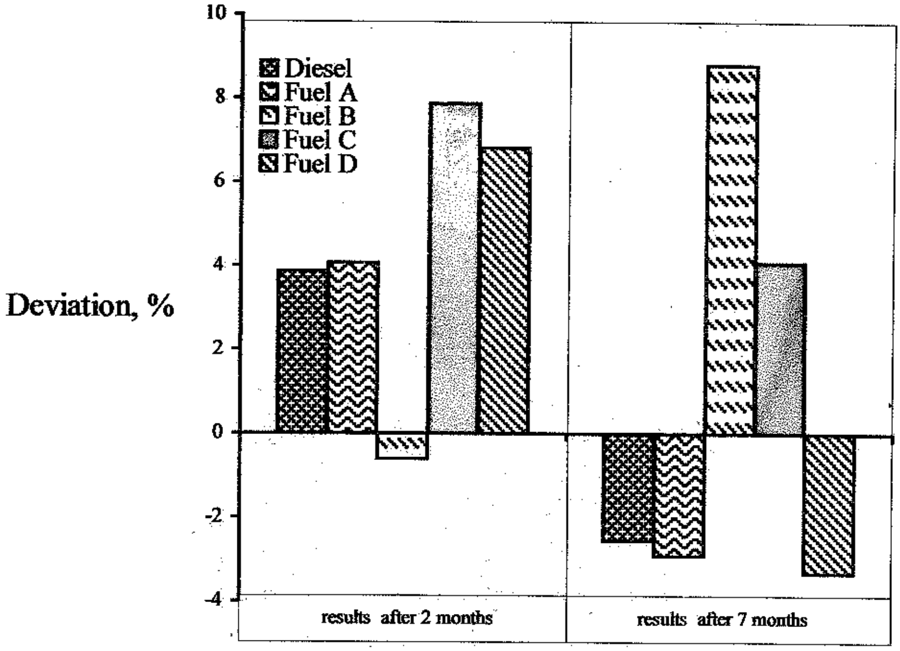

Figure 9. Deviation of brake thermal efficiency with respect to time at 900 rpm

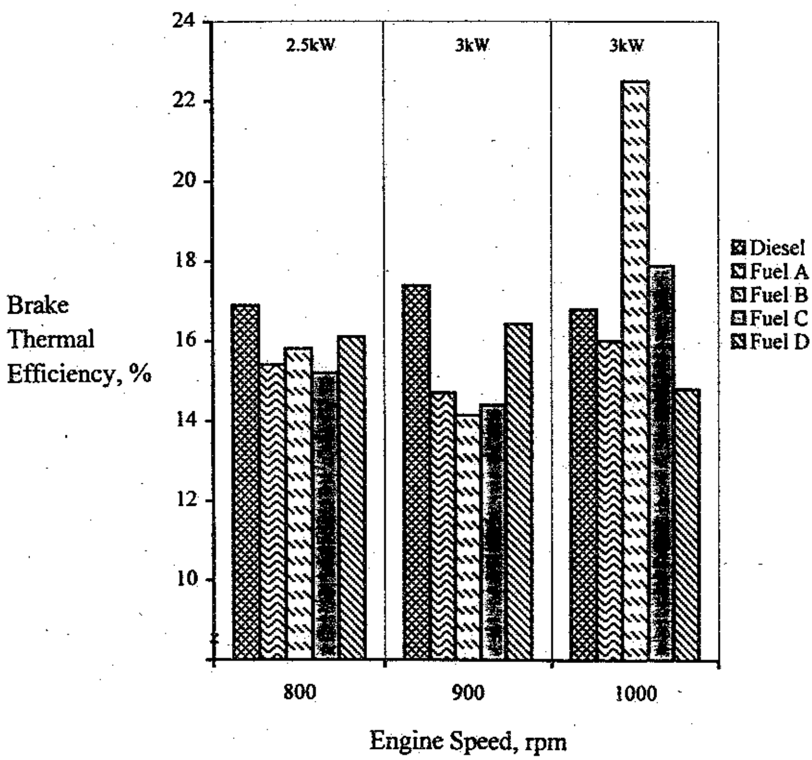

Figure 10. Maximum brake thermal efficiency at selected brake power for various engine speed

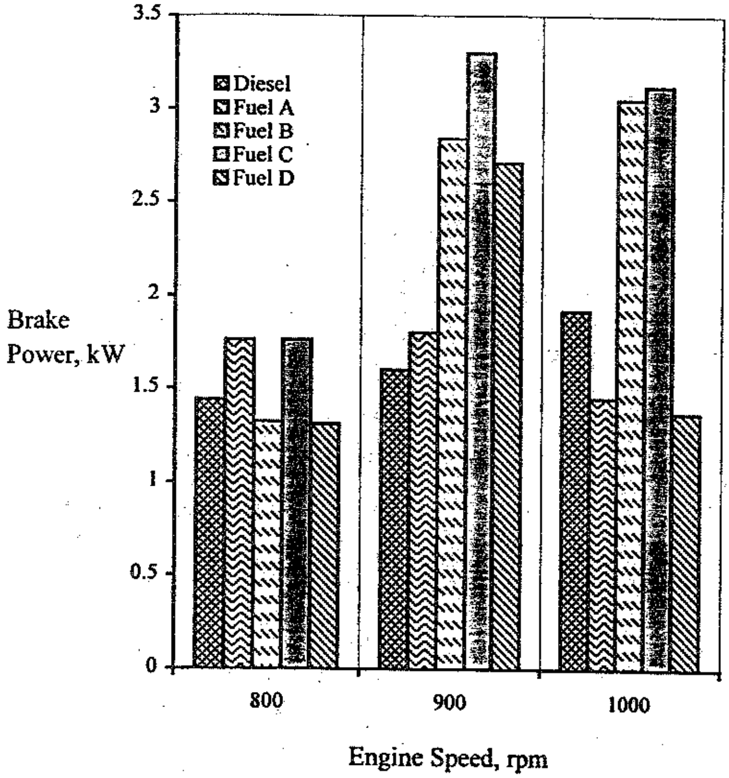

Figure 11. Maximum brake power at optimum brake thermal efficiency

The maximum brake thermal efficiency at 800 rpm for diesel is 18.7% at 1.4 kW; for fuel B, 17.5% at 1.3 kW; for fuel A, 16.4% at 1.8 kW; for fuel D, 16.4% at 1.3 kW; and for fuel C is 15.6% at 1.8 kW. At 1,000 rpm, the maximum brake thermal efficiency for fuel B is 22.5% which was achieved at 3.1 kW; for fuel C, it is 17.9% at 3.1 kW; for fuel A, it is 17.8% at 1.5 kW; for diesel, it is 17.6% at 1.9 kW; and for fuel D, it is 17.2% at 1.4 kW.

The engine can also be operated at minimum emissions. For example, at minimum carbon monoxide of 0% for diesel, the brake power is 0.74 kW; for fuel A, the minimum CO of 0.2% gives brake power of 1.1 kW; the minimum CO of 0% for fuel B resulted to a brake power of 0.86 kW; for fuel C, the minimum CO of 0.2% is attained at a brake power of 1 kW; the minimum CO of 0.2% is also provided by fuel D at 0.96 kW.

IV. Conclusions

The Lister-Blackstone compression ignition diesel engine can be operated using fuel mixtures A, B, C and D without any modification in the engine. There is no substantial difference of the engine performance with respect to the different storage durations. The brake thermal efficiency against brake power is generally slightly lower for the fuel mixtures as compared to diesel. However, it is possible to run diesel engines on emulsified fuels and obtain efficiencies comparable or better than running the engine on diesel in some conditions.

The possibility of obtaining higher efficiencies from emulsified fuels could still be explored by developing a process that could produce smaller water droplet sizes that can be fully enrobed in the fuel. The large water droplet sizes of the emulsified fuels produced in this study may have been a hindrance in obtaining the optimal performance of the fuels.

The proportions and blending of diesel, bunker, water and the catalyst may also be varied to find out which method and proportion would result to the highest efficiency. Concentrations of the components of the flue gas should be measured, particularly NOx. Emulsified fuel results in cooler and/or shorter primary flame zone decreasing the amount of NOx [5].

Some of the properties of emulsified fuels may pose some problems in the operation of engines in actual running conditions. This study shows that the addition of bunker and water to ordinary diesel oil causes significant changes in fuel properties. Further studies may be done to test the effects of emulsified fuels on the different engine components. Other properties of the fuel and the catalyst must also be determined.

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