TECHNICAL NOTE

RESEARCH AND ASSESSMENT OF APPLYING DIMETHYL ETHER "DME" EXTRACTED FROM NATURAL GAS "NG", ON DIESEL ENGINE AS A CLEAN FUEL

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Abstract Due to the shortage of liquid hydrocarbon fuels for compression ignition engines, researchers have constantly been looking for alternative fuels. Recently, dim ethyl ether (DME) with its interesting properties, such as high cetane number and low exhaust emission has drawn a lot of interst as a suitable fuel for diesel engines. The main objective of this study was to assess the potentials and feasibility of using DME in a diesel engine by designing a suitable fuel injection system that would generate a basis for future optimizations and developments. For one model of diesel engines, a new fuel injection pump was designed and manufactured. A series of performance tests based on ISO procedure were performed, and the results showed that DME used in the newly designed fuel injection pump can compete fairly well with the conventional diesel fuel system as far as power and torque output is concerned. Proposals for future development are also presented.

Keywords Dimethylether, Alternative Fuels, Fuel Injection Systems

چکیده عطف به ویژگی های سوخت دی متیل اتر (DME) بعنوان سوخت پاک وجایگزین مناسب با گازوئیل در این تحقیق برپایه استخراج پارامترهای مشخصه موتور دیزلی با پاشش مستقیم (DI-Diesel) معروف به موتور OM314 با سوخت گازوئیل درکلیه شرایط عملکردی از دور و بار برمبنای تحلیل ترمودینامیکی ضریب تراوش حرارتی یا روش Wiebe و مطابقت آن با روش فرگوسن و برآورد انرژی ورودی به موتور با سوخت DME به طراحی ترمودینامیکی و انرژی ورودی و خروجی پرداخته و میزان پاشش سوخت DME را برای عملکرد موتور تحت شرایط مختلف دور و بار بطور دقیق تعیین می‌شود. این تحقیق برای نخستین بار درکشور انجام شده و موتور دیزل با سوخت پاک دی متیل اتر طراحی و نمونه سازی گردیده است. با انجام آزمایش های موتور در شرایط بار کامل و استخراج منحنی های توان، گشتاور و مصرف ویژه سوخت برای موتور دیزلی 314 با سوخت DME و گازوئیل، مقایسه دستاوردهای علمی انجام شده و روش های نظری بکارگرفته شده مورد تایید قرارگرفت.

1. INTRODUCTION

Due to Disel engine's high thermal efficiency and low CO and HC emissions it have always been the most favorite choice among various types of internal combustion engines. Nevertheless, the shortage of liquid hydrocarbon fuels and the high level of particulates and soot in the exhaust gases of the engines at full load conditions have always caused serious concerns worldwide. Therefore many attempts have been made to find an alternative fuel as a substitute for the gas oil in this type of engine.

As experience many alternative fuels have been investigated [1], each with its special advantages and disadvantages: alcohol fuels are difficult to burn smoothly due to their low cetane number, while CNG need to change the cycle to Otto cycle which needs spark ignition with consequent loss of the high thermal efficiency of the diesel cycle.

Recently, attention has been directed towards dimethyl ether [2,3] with its two interesting properties, high cetane number and no soot formation; which will be discussed later. This fuel can be produced from natural gas which is of interest to the countries with their own vast resources of natural gas.

The main disadvantage of DME for application in diesel engines is its low viscosity which causes an unacceptable amount of fuel leakage in the injection pump. Low lubricity of this fuel is another problem [4], so conventional fuel injection systems are not capable of handling this fuel without losing their proper performances.

In this study, with the objective of the application of DME in a diesel engine manufactured in this country, Diesel OM314, first the properties of this fuel is reviewed and its economical production is assessed. Then a fuel injection system suitable for this engine is selected. The size and capacity of the system was determined by thermodynamic cycle calculations of a diesel engine while using the Wiebe function as the heat release model.

Due to the high precision and high technology involved in this system, the provision of a suitable injection pump from industrial companies was not possible due to some “embargos”; therefore it was decided to put utmost effort in the designing and manufacturing of such a system which can be installed on DI diesel engines.

Overcoming the problems and technical complications of this production, which took nearly two years, brought good experiences in the field of alternative fuels for Diesel engines for scientists.

The reason for selecting the engine of Model OM314 was due to the fact that this engine is produced in this country and it is already installed on many transportation fleet minibuses nationwide; therefore, next transformations and developments will be straightforward.

After preliminary tests, the manufactured injection pump was installed on the actual engine, and various tests were performed as will be discussed later.

2. THE PROPERTIES OF DME

In recent decades, the properties of DME have been investigated extensively as it has been used in chemical and pharmaceutical industries for a long time. The most important physical properties of DME relevant to engine applications are given in Table 1, which for reason of comparison, also shows the properties of other fuels mentioned in this paper. It can be seen that DME is a vapor at room temperature and pressure, and in order to be kept liquefied at 25°C, it must be contained under a pressure of at least 5 atmospheres. This is very similar to the vapor pressure of LPG. For this reason, the handling and distribution equipment of this types of fuels are similar.

With respect to its combustion, DME combines a number of advantages which makes it almost the ideal fuel for diesel engines:

- DME has relatively high hydrogen to carbon ratio compared to gas oil. This means that low CO_2 is produced after combustion is completed.
- DME molecules contain oxygen which avoids the formation of soot effectively, low exhaust emissions.
- DME is free of carbon to carbon bonds, the molecular structure of ethers break up readily, which lead to high cetane numbers.

As mentioned before, DME suffers from a low viscosity and low lubricity. It also has a low heating value compared to gas oil. This means that more DME is needed for the same value of energy release. Considering the densities of both fuels, one cubic meter of gas oil is equivalent to 1.85 cubic meter of DME.

TABLE 1. Physical and Chemical Properties of Various Fuels [5,6].

Properties \ Fuel	Natural gas	Methanol	Gas Oil	DME
Chemical Formula	$C_nH_{3.8n}$	CH_3OH	$C_nH_{1.8n}$	CH_3OCH_3
Density kg/lit@ 1 bar and-20°C	-	0.79	0.84	0.66
Boiling Point°C@ 1bar	-161.5	65	180-370	-25
Cetane No.	-	-	40-55	55-60
Viscosity @20°C cP	-	0.6	2-4	0.15
Lower Heating Value MJ/kg	48.5	20	42.5	28.8

2.1. Production Process of DME

DME can be produced from a variety of sources such as natural gas, crude oil, coal, waste, and biomass.

Production of DME from natural gas is based on production of synthesis gases (CO and H_2) and production of methanol, and then dehydration of methanol according to the following steps [7]

$$2CO + 4H_2 \leftrightarrow 2CH_3OH - 182(kJ/kmol)$$

$$2CH_3OH \leftrightarrow CH_3OCH_3 + H_2O - 23(kJ/kmol)$$

$$CO + H_2O \leftrightarrow CO_2 + H_2 - 41(kJ/kmol)$$

Although there has been a lot of technological progress in DME production, the price of gas oil is still lower than that of DME. Low fuel costs are achieved by a high production volume. It has been shown [8] that a production unit with a capacity of 100 tons per day of DME can approach the prices of the two fuels. Bearing in mind that by constantly tightening the emission regulations (and the cost of emission treatments) and the superiority of DME in this regard, the application of DME is economically justified.

3. THERMODYNAMIC-CYCLE CALCULATIONS

In order to estimate the fuel flow rate required for the engine, as well as the capacity and dimensions of the injection system, a thermodynamic cycle calculation was performed. The temperature, pressure, torques and power outputs of the cycle were calculated by using the Wiebe heat release function.

The cumulative heat release fraction x_b (θ) is expressed as [9]

$$x_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^n\right]$$

Where θ = Crank angle, θ_s = start of heat release, θ_d = duration of heat release, n = Wiebe form factor, and a = Wiebe efficiency factor. The parameters a and n are adjustable parameters used to fit experimental data [10]. The rate of heat release as a function of θ is obtained by differentiating the cumulative heat release Wiebe function [9]

$$\frac{dQ}{d\theta} = Q_{in}\frac{dx_b}{d\theta} = na\frac{Q_{in}}{\theta_d}(1 - x_b)\left(\frac{\theta - \theta_s}{\theta_d}\right)^{n-1}$$

Where Q_{in} is the total heat received by the cycle. If the cylinder content is assumed as an ideal gas, applying the First Law of Thermodynamics to the cylinder content results in the following relation between the gas pressure, gas volume, and the crank angle

$$\frac{dP}{d\theta} = -\gamma\frac{P}{V}.\frac{dV}{d\theta} + \frac{\gamma - 1}{V}\frac{dQ}{d\theta} - \frac{\delta Q_w}{d\theta}. \qquad (a)$$

Where γ is the ratio of specific heats, V is the cylinder volume, and Q_w is the heat transfer to the cylinder walls. The value of Q_w depends on the value of heat transfer coefficient hg, which can be calculated from the Woschni's correlation [11]

$$hg = 3.26U^{0.8}p^{0.75}B^{-0.2}T^{-0.55} \quad W/m^2K$$

Where U is the characteristic gas velocity which is equal to 1.68 $\overline{U}$ p,

and

$\overline{U}$ = Mean Piston Speed, m/s
T = Gas Temperature, K
P = Gas Pressure, k P_a
B = Cylinder Bore, m

The cycle pressure trace can be obtained from Equation a, from which the fuel flow rate, engine torque, and power can be calculated.

3.1. Diesel OM 314 Specifications Diesel OM314 is a direct injection diesel engine manufactured in this country under the license of a European company. General specifications of this model are represented in Table 2.

3.2. Engine Test with Gas Oil In order to find the engine performance parameters and optimizing the fuel system with DME under various load conditions, a series of tests were performed according to DIN 70020, for various speeds and load conditions. In these tests the engine was run with gas oil without any modification on engine systems. Therefore the economy range of the engine operation and the amount of required energy can be obtained. Figure 1 shows the variation of the power and torque of the engine. In Figure 1, the correction factor for the ambient conditions of a naturally aspirated engine [12] was applied.

$$f = \left(\frac{99}{p_0}\right)\left(\frac{T_0}{298}\right)^{0.7}$$

where

P_0 = Ambient Pressure, kPa
T_0 = Ambient Temperature, K

In Figures 2-4 the brake specific fuel consumption, BSFC, for three different engine speeds are shown.

The minimum BSFC happens in the region of 50 % to 100 % at full load conditions. Figure 5 presents the amount of fuel consumption for various constant speeds. This figure can be used for optimum design of the fuel control system.

The results of this test (Figure 1) show that the maximum power output is about 54.7 kW at 2600 rpm. This figure is about 13 % less than the figure from the engine manufacturer 63 kW at 2800 rpm, which is the result of low quality fuel and degradation of the quality of the fuel system.

The minimum fuel consumption is 260 g/kWh at 1100 rpm. This figure is again 16 % higher than the manufacturer's data in the engine manual.

4. SELECTION OF FUEL SYSTEM FROM DME

In Diesel OM314, the fuel injectors are installed underneath of inlet manifold, so there is not

TABLE 2. Specifications of OM314 Diesel Engine.

No. of Cylinders	4	Compression Ratio	17:1
Bore	97mm	Max. Power (with gas oil)	85 hp (63 kW)
Stroke	128mm	Max. Torque (with gas oil)	235 Nm
Swept Volume	3780 cm^3	Max. Speed	2800 rpm

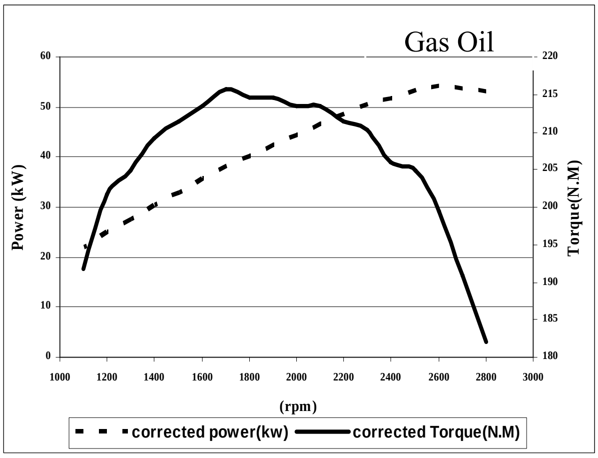

Figure 1. Brake power and torque at full load.

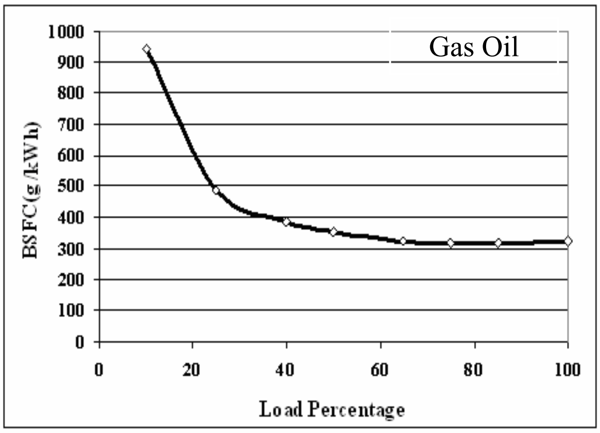

Figure 2. BSFC at constant speed of 2500 rpm.

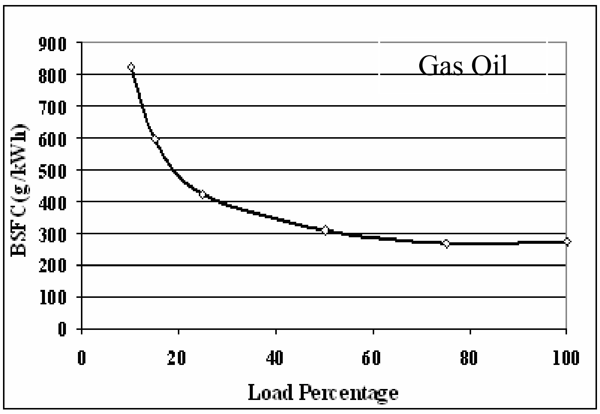

Figure 3. BSFC at constant speed of 2000 rpm.

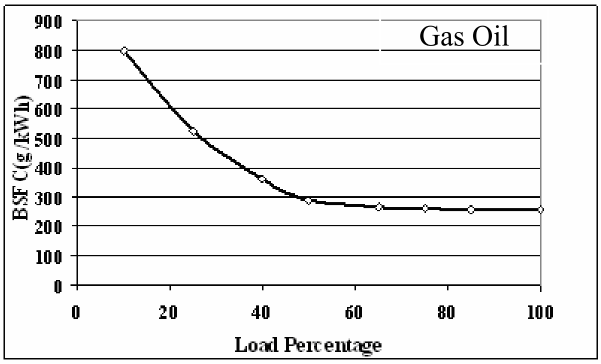

Figure 4. BSFC at constant speed of 1500 rpm.

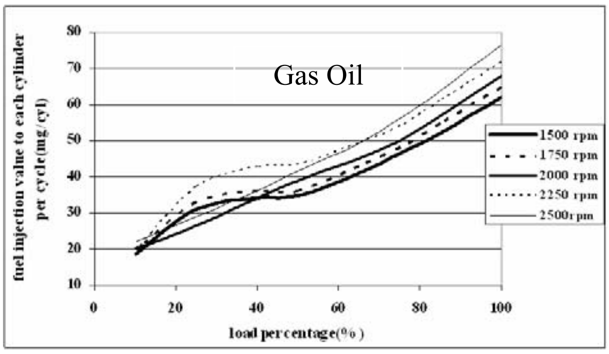

Figure 5. Fuel injection value to each cylinder per stroke (mg/cyl).

enough space for installing common rail systems or unit injectors. In order to avoid any Complication by changing the engine inlet system, the unit pump system was selected. In these units the fuel control system is integrated directly on the pump. Fuel is sent to the pump under the pressure of about 30 bar and the remaining fuel can return to the fuel reservoir under a pressure of about 10 bar. The fuel which is leaked through the moving parts of the pump are collected in the lower parts and returned to the fuel reservoir under a pressure of 1 bar.

Sizing of the fuel system components are based on the results of the previous calculations. The maximum fuel flow rate under the full-load conditions and 2500 rpm was found to be 70.32

mg/cylinder per cycle. To be on the safe side, an injection pump with a flow rate of 100 mg/cylinder per cycle was selected for that purpose. This capacity seems adequate for engine speed of about 2800 rpm. The injection pressure is in the range of 250-300 bar.

4.1. Designing and Manufacturing of a New Injection Pump

As mentioned before, it was decided to design and manufacture the injection pump with the expertise and technical potentials available inside the country.

The salient feature of this pump is its ad hoc level of tolerances involved, as well as the production methods employed for its surface treatment of the moving parts.

The appropriate material, dimensions and the surface quality was selected with high precisions, and the production of the parts was carried out in several stages. Figure 6 up to 9 shows some manufactured parts.

The pump was assembled and the preliminary tests were performed. The pump was then installed on the engine and the first run was arranged for an hour. Because of some unacceptable rate of leakages, which was around eight percent of total fuel injected, the pump was dismantled and all the parts were carefully inspected. It was found that some parts of the barrel and plunger were weared or cracked. [14] So another surface coating and new material were selected. The material finally selected for the plunger was HSS and for the barrel was aluminum-bronze alloy. The coating was Teflon (PTFE-Poly Tetra Florien Ethylene). In addition, a PTFE was inserted on both ends of the plunger. After that, the leakage rate was reduced to about 0.5 percent of the total fuel injected.

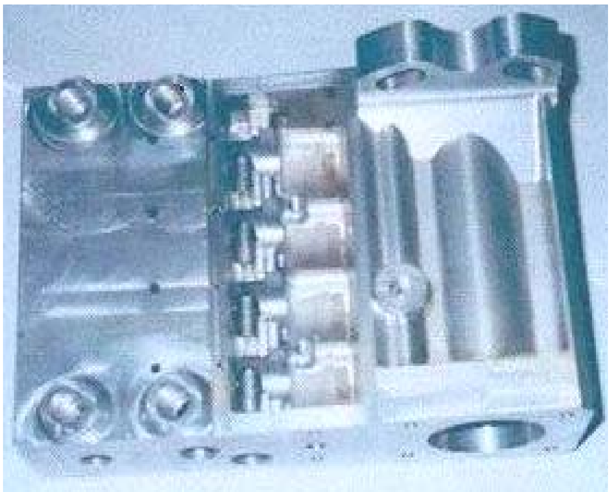

Figure 6. DME fuel pump cylinder block.

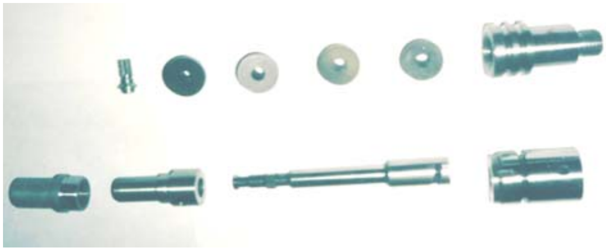

Figure 7. DME pump and plunger component and drawing.

5. ENGINE TEST WITH THE DME

After the preliminary test, the pressure and temperature of the fuel system, as well as the pressure of the injectors were adjusted, and a series of tests were performed according to ISO 3046 I-V.

The pump was run for about 30 hours and the adjustments were made manually by changing the position of the rack on the pump. No automatic control was employed at this stage, although the governor could perform as usual. [13]

In the full-load conditions, the engine speed was varied from 500 rpm to 3100 rpm. The power test results are shown in figure 10 up to 12. The corresponding curves for the tests with gas oil are also included for the purpose of comparison.

6. ANALYSIS OF THE RESULTS

When using gas oil in the engine, the maximum power was 54.70 kW at 2600 rpm, while for the case of DME, one can reach the value 54.85 kW at 3100 rpm. The two figures are almost identical, but the latter is obtained at a higher speed due to the

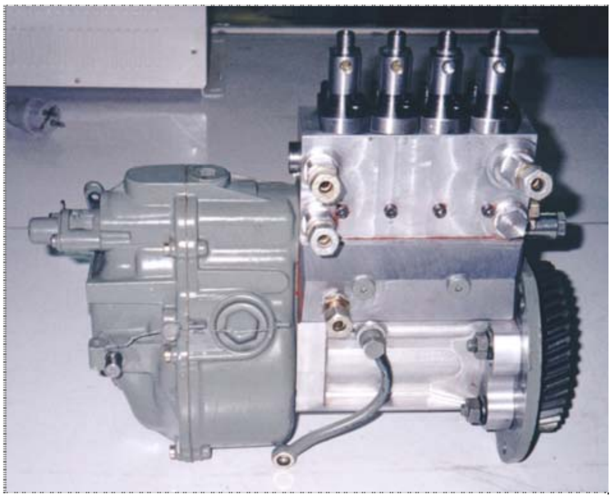

Figure 8. DME fuel pump assembled with engine governor.

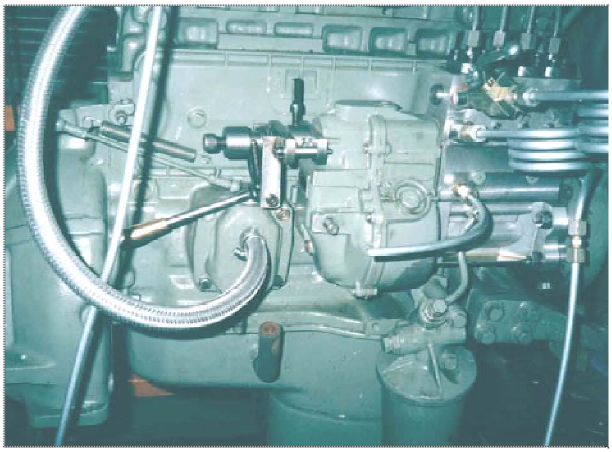

Figure 9. DME fuel injection pump installed on the engine.

fact that the smoke limit has increased without having a loss in thermal efficiency. It should be noted that although the volumetric efficiency decreases at higher speeds [15], the thermal efficiency still sustains its high value.

The maximum engine speed is determined by the allowable dynamic forces in the engine parts, so in the case of DME, the engine acceleration, i.e., the variation of speed in a time interval, has improved for the part-load test. The time needed to increase the speed from 1500 rpm to 1800 rpm was

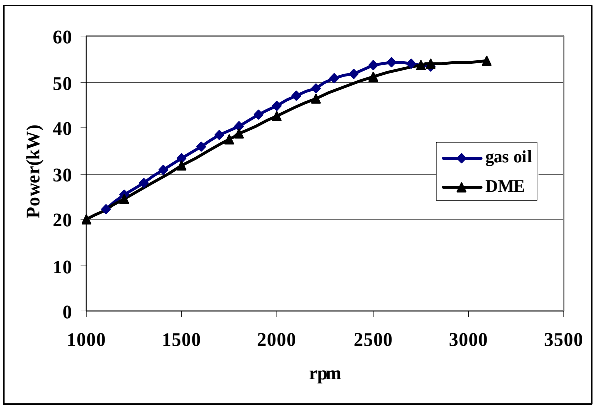

Figure 10. Engine power test result with DME and gas oil at full load.

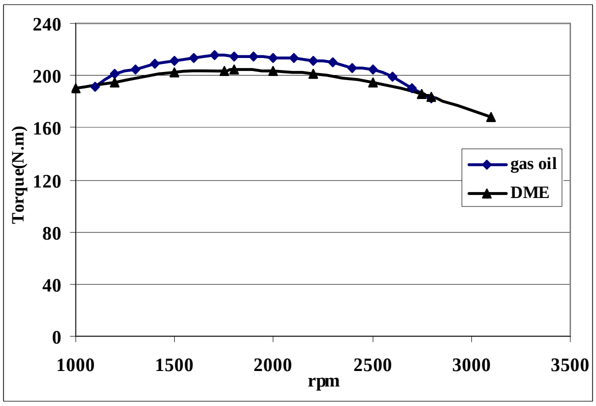

Figure 11. Engine torque test result with DME and gas oil at full load.

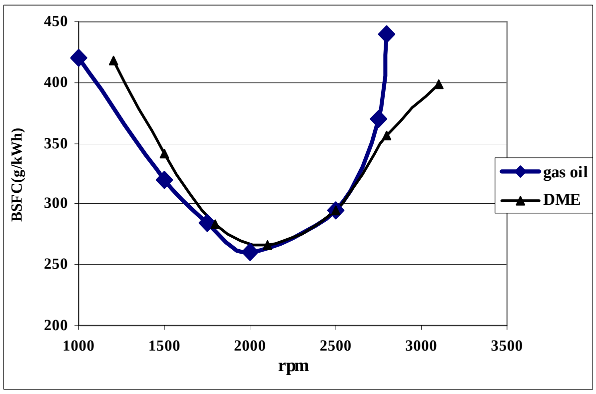

Figure 12. Engine brake specific fuel consumption (BSFC) with DME and gas oil at full load.

20 seconds for gas oil, while this figure is reduced to 8 seconds for DME.

This shows that acceleration characteristics have improved by 250 percent. This seems to be due to a better mixing of the fuel and air, and faster formation and propagation of flame nuclei in the injection and combustion of DME compared to gas oil.

The maximum torque obtained from the engine with gas oil was 215 N.M at 1700 rpm, compared to the value of 205 N. M at 1800 rpm for DME. Here a decrease of 5 percent in maximum torque is observed when using DME. This was somehow expected because of the lower heating value of DME.

The minimum brake specific fuel consumption of the engine with gas oil was about 260 g/kWh at 2000 rpm, while for DME; there is a value of 266 g/kWh at 2100rpm, which seems a reasonable figure at this stage of work. The economy range of the engine with gas oil is between the speed values of 1500 rpm and 2600 rpm, while for DME this range is between 1600 rpm and 2700 rpm, which shows a similar trend for both fuels.

For part-load conditions, DME has a similar or sometimes even better performance than gas oil, which makes it a suitable substitute for automotive applications.

7. CONCLUSIONS

The performance of the manufactured injection pump seems quite satisfactory, and this brings the prospect of extending the application of this type of injection pumps to other models of diesel engines.

The result of this study shows that dimethyl ether can compete fairly well with gas oil, and it can be used as a substitute for gas oil without having a significant loss of torque and power output. The low viscosity of DME was compensated with a better surface treatment and choosing closer tolerances in the injection pump. This resulted in lower fuel leakage and higher resistance to wear.

Since the economical production of DME from natural gas is already feasible by new technologies, it seems reasonable to have a comprehensive plan for substitution of gas oil by DME wherever natural gas is available.

Since for diesel engines, fuel common rails can provide a constant fuel pressure for the injectors, and consequently, a better control can be achieved for fuel flow rate, it is proposed to study the application of common rails for DME systems. Of course modification of inlet manifold is necessary for adaptation of common rails on the conventional systems.

8. ACKNOWLEDGEMENT

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