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A Theoretical Study of the SI Engine Performance Operating with Different Fuels

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Abstract**—**The intension in this work is to investigate the effect of different fuels type on engine performance for different engine speed. Brake Power, Brake Torque, and specific fuel consumption were calculated and presented to show the effect of varying fuel type on them for all cases considered. A special program used to carry out the calculations. A simulation model for one-cylinder spark ignition engine has been built and calculated.

The analysis of the results shows that for methanol the power increases about 30% at 1000 rpm and 16% at 6000 rpm comparing with methane. For the same compared fuels the increment in fuel consumption is about 100% at 1000 rpm and 115% at 6000 rpm. The increment in brake thermal efficiency for gasoline is around 11% comparing with methane at 1000 rpm and 7% for methanol comparing with methane at 4000 rpm.

Keywords**—**Natural gas fuel, spark ignition engines, performance, engine simulation.

I. Introduction

THE control of engine performance and gas emissions has begun to add to the numerous constraints that vehicle manufacturers have to satisfy. The reduction of engine fuel consumption becomes a primary requirement as well as meeting current and future emission legislations. There has been much recent activity on the operation of engines on gaseous fuels both of the spark ignition and compression ignition types. This has been motivated largely by the need to produce engines that can comply with the ever more demanding regulations for cleaner exhaust emissions. Moreover, the operation of engines on the relatively more abundant gaseous fuels such as natural gas, can have distinct potential economic and operational benefits while, if not improving, retaining excellence in reliability and durability [19]. Methanol is an alternative fuel for internal combustion and other engines, either in combination with gasoline or directly ("neat"). It is used in racing cars in many countries. Methanol is less expensive than hydrocarbon to produce sustainably and is a less expensive way to reduce the carbon footprint. However, for optimizing engine performance, fuel availability, toxicity and political advantage, a blend of methane, methanol and petroleum is likely to be preferable to using any of these individual substances alone. Methanol may be made from hydrocarbon or renewable resources, in particular natural gas and biomass respectively. It can also be synthesized from CO_2 and hydrogen. Methanol is far more difficult to ignite than gasoline and burns about 60% slower. A methanol fire releases energy at around 20% of the rate of a gasoline fire, resulting in a much cooler flame. This results in a much less dangerous fire that is easier to contain with proper protocols. Unlike gasoline, water is acceptable and even preferred as a fire suppressant, since this both cools the fire and rapidly dilutes the fuel below the concentration where it will maintain self-flammability. These facts mean that, as a vehicle fuel, methanol has great safety advantages over gasoline.

The performance of spark ignition internal combustion engine fueled with hydrogen or natural gas has been proposed as a method to improve the engine performance and emissions. Several studies have reported on the performance of spark ignition engines with hydrogen - gasoline fuel [1]-[3] and hydrogen - natural gas fuel [4]-[8]. Improving engine performance can be met following different routes each of them could be an effective way with different cost-to-benefit ratio [9]-[12]. Often, it could be observed, it is helpful to adopt numerous solutions contemporaneously. As an example, fast combustion, lean burn, variable valve timing and actuation, manifold bending angle, gasoline direct injection and so long may be reminded [13]-[18].

The objective of this paper is to investigate the performance of SI engines fueled with methane, methanol, and gasoline. Power, torque, thermal efficiency and fuel consumption has been calculated and analyzed for different fuels. The effect of engine speed has also been considered.

II. Development of the Model

For the purpose of analyzing the engine characteristics a single cylinder engine model with port fuel injection has been built and investigated. The combustion process employed a two zone combustion model [20]-[22]. The heat release rate can be defined using empirical heat release functions. The empirical heat release functions are derived from the Woschni heat transfer model. Dissociation effects (CO generation) were modeled through curve fits to the Eltinge diagram, which relates combustion products of CO and O_2 to user specified parameters of air-fuel ratio and mal-distribution [23]. This approach avoids the computationally expensive chemical rate. The valve lift profile is specified by a polynomial consisting of four coefficients and four exponents. The nature of the polynomial is such that the sum of the coefficients is 1.When gas flows through a valve the development of separation and recirculation regions gives rise to a vena-contract where the actual cross-sectional area of the gas stream (effective area) is

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less than the geometric area of the orifice. This phenomenon cannot be simulated directly using a one-dimensional model and has to be characterized using empirical data.

Heat transfer was modeled in all elements. Within cylinders the empirically derived heat transfer correlation proposed by Annand [24] was employed. This model has been derived from a basic Nusselt Number / Reynolds Number correlation for flow in pipes. The model employs coefficients that have been developed to best reproduce the heat transfer results obtained by experiment. The coefficients used by the program may either be the default values or may be tuned by the user to best suit the engine being studied.

The connective heat transfer model proposed by Annand is defined as;

$$\frac{hD_{cyl}}{k} = A\ \mathrm{Re}^{B} \qquad (1)$$

where

h = heat transfer coefficient [W/m2 K]

A = Annand open or closed cycle coefficient = 0.2

B = Annand open or closed cycle coefficient = 0.8

k = thermal conductivity of gas in the cylinder [W/m K]

Dcyl = cylinder bore

Re=Reynolds number based upon mean piston speed and the engine bore. The density is that calculated for the cylinder contents at each crank angle.

Thus the heat transfer per unit area of cylinder wall is defined as:

$$\frac{dQ}{A} = h\ \left(T_{gas} - T_{wall}\right) + C\ \left(T_{gas}^{4} - T_{wall}^{4}\right) \qquad (2)$$

where:

dQ /A = heat transfer per unit area [W/m2]

C = Annand closed cycle coefficient.

The first part of the heat transfer equation is the convective heat transfer and the second part is the radiative heat transfer.

The engine geometry, running conditions and valves data are shown in Table I. All parameters were kept constant through the simulation. The test is carried out for methane, methanol and gasoline fuel. The engine speed is varying between 1000 – 6000 rpm. The brake power, brake torque, brake thermal efficiency, brake mean effective pressure and fuel consumption were calculated and analyzed. The fuel properties are illustrated in the Table II. The spark angle for all tests points are 14o bTDC and kept constant for all engine runs. The ignition delay is assumed to be zero. The turbulent ratio is 1.7. The equivalence ratio ϕ = 1 for all simulations. The ambient air pressure = 1 bar, and temperature = 20^{o}. The model employ H.B. Moss gasoline engine friction model.

TABLE I
ENGINE GEOMETRY

No. of cylinders	1
Bore	104 mm
Stroke	113 mm
Connecting rod length	300 mm
Compression ratio	10.5
Max. intake valve lift	12.1 mm
Max. exhaust valve lift	13.4 mm
IVO angle	25^{o} bTDC
IVC	62^{o} aBDC
EVO	50^{0} bBDC
EVC	20^{0} aTDC
Intake valve throat dia.	46.3 mm
Exhaust valve throat dia.	41.5 mm

TABLE II
FUEL PROPERTIES

parameter	Fuel type		
	methane	methanol	gasoline
Relative density [kg/litr]	0.00074	0.79	0.75
Calorific value [kJ/kg]	46280	20000	43000
Molecular mass [kg/kmol]	17.423	32.04	114.23
H/C ratio	3.87	4	1.8
O/C ratio	0	1	0

III. Results and Discussion

For the engine geometry and running conditions shown above, the simulation has been made for different engine fuels. The fuel types used in the test are methane, methanol and gasoline. The engine speed was varied from 1000 – 6000 rpm. Fig. 1 presents the brake power versus engine speed for different fuel. It shows that methane has the minimum brake power comparing with methanol and gasoline. For low and medium speed gasoline has maximum power than methanol but for engine speed more than 4000 rpm the methanol has the higher brake power. This is due to due to its high octane rating (114) and high heat of vaporization. The brake power is proportional to the engine speed up to 5000 rpm for all fuels type.

Fig. 2 shows the variation of brake torque for different fuel and speed. It shows a decrease of brake torque for all engine speed range. The maximum decreasing percentage in torque is presented for gasoline.

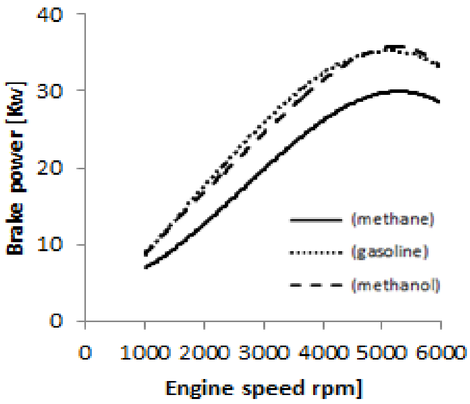

Fig. 1 Brake power for different fuels and engine speed

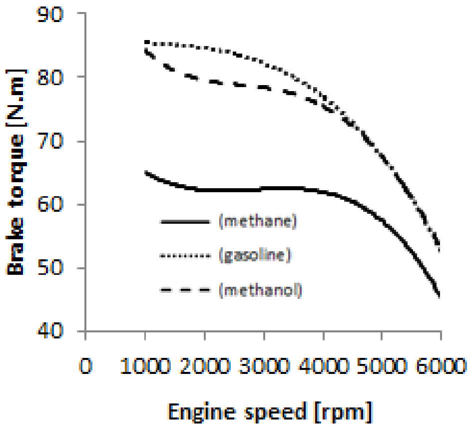

Fig. 2 Brake torque for different fuels and engine speed

Fig. 3 shows the variation of BSFC versus engine speed and fuel, this shows that BSFC is slightly increased for methane and gasoline for all engine speed. But it was sensitive to methanol for engine speeds between (4000-6000 rpm). Moreover, methanol has the highest fuel consumption comparing with gasoline and methane. This due to its low energy content of 19.7 MJ/kg and stoichiometric air-to-fuel ratio of 6.42:1 mean that fuel consumption (on volume or mass bases). Fig. 4 shows the change of brake thermal efficiency versus engine speed for different fuel type. It shows that gasoline has the maximum thermal efficiency for engine speed less than 3500 rpm. For higher engine speed the methanol has the highest thermal efficiency.

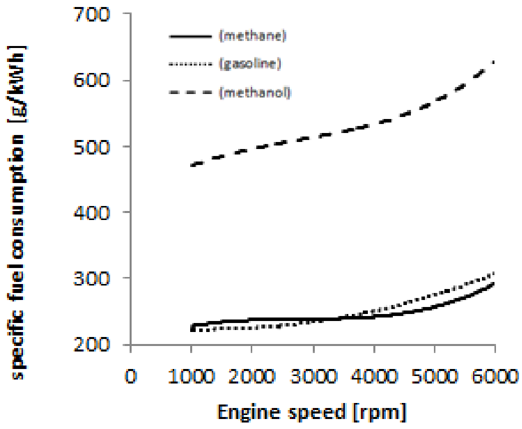

Fig. 3 Fuel consumption for different fuels and engine speed

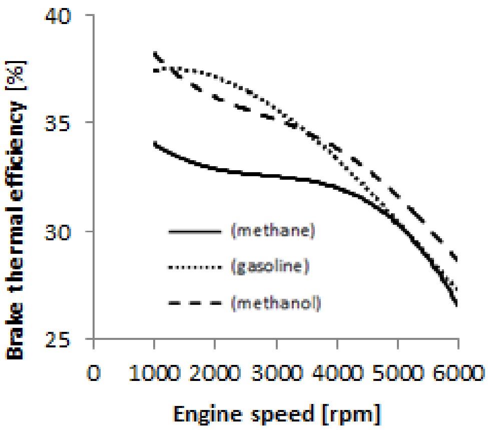

Fig. 4 brake thermal efficiency for different fuels and engine speed

IV. Conclusions

From analysis we can conclude that the use of methanol as a fuel for the SI engine is recommended at high engine speed because it has the highest brake thermal efficiency and output power. The gasoline fuel is more efficient at low and medium engine speed.

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