APPLICATION OF BIOGAS-GASOLINE HYBRID FUEL MIXTURE ON THACO TOWNER 750 LIGHT TRUCK ENGINE

ỨNG DỤNG NHIÊN LIỆU HYBRID BIOGAS-XĂNG TRÊN ĐỘNG CƠ XE TẢI NHẸ THACO TOWNER 750

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Abstract - The combination of various fuels in an internal combustion engine is a major area of research in the field, aimed at conserving fossil fuels and reducing environmental pollution. This paper examines the performance and pollution emissions of a DA465QE Towner engine using a biogas/gasoline hybrid fuel in a light truck engine. Results show that when the engine runs on methane, the cycle indicator work is reduced by up to 13% compared to gasoline. When running on biogas M6C4, the cycle indicator work decreases the most by 27% compared to gasoline, and when running at high speeds, the engine power increases. The optimal gasoline content blended with M7C3 biogas is 30%, with the cycle indicator work and NO_x emission increasing quickly and CO and HC emissions decreasing sharply with less than 30% gasoline content.

Key words - Renewable fuel; Biogas engine; Hybrid fuel; equivalence ratio; Pollution emissions

Tóm tắt – Trong lĩnh vực nghiên cứu động cơ đốt trong, việc kết hợp đa dạng các loại nhiên liệu trở thành một hướng tiên tiến, nhằm mục đích giảm sử dụng nhiên liệu hóa thạch và hạn chế phát thải gây ô nhiễm môi trường. Bài báo trình bày kết quả nghiên cứu đánh giá tính năng kỹ thuật và mức độ phát thải ô nhiễm động cơ TOWNER DA465QE sử dụng nhiên liệu hybrid biogas-xăng. Kết quả chỉ ra rằng trong cơ chế phối hợp nhiên liệu hybrid biogas-xăng, khi chạy bằng mêtan thì Wi giảm nhiều nhất 13% so với khi động cơ chạy bằng xăng. Khi chạy bằng biogas M6C4 thì Wi giảm nhiều nhất 27% so với xăng, và khi hoạt động ở tốc độ cao, công suất động cơ tăng lên. Hàm lượng xăng tối ưu trung bình pha vào biogas M7C3 là 30%. Wi và nồng độ NOx tăng rất nhanh trong khi phát thải CO, HC giảm mạnh theo hàm lượng xăng pha vào biogas khi hàm lượng xăng nhỏ hơn 30%.

Từ khóa – Nhiên liệu tái tạo; Động cơ biogas; Nhiên liệu hybrid; Hệ số tương đương; Phát thải ô.

1. Introduction

The use of a hybrid fuel mixture, consisting of multiple fuel types, in an internal-combustion engine is a major area of research for scientists. The goal is to improve combustion efficiency and reduce pollutant emissions. Biogas, a renewable fuel with a high-octane content but low burning rate, is a promising option. Combining biogas with fuels of high calorific value and/or high combustion rate can increase its efficiency in internal-combustion engines.

In tropical countries, biogas is an abundant renewable fuel source that does not contribute to an increase in atmospheric CO_2 levels. In fact, its use also limits CH_4 emissions, a greenhouse gas with a potency 23 times higher than CO_2. Biogas contains inert CO_2 gas, which leads to low calorific value, slow combustion, and reduced engine power. However, its high-octane value provides good anti-knock properties [1, 2].

The use of a biogas-gasoline hybrid fuel mixture overcomes the limitations of biogas and maximizes its benefits, contributing to the development of renewable fuel applications in internal-combustion engines, including vehicle engines. In this study, conventional biogas is used as a hybrid fuel in vehicles, with biogas serving as the main fuel for low to medium engine regimes. Gasoline is added to enhance the energy of the combustion mixture when high torque is needed. When operating at high loads, gasoline is used to support the engine, as biogas alone cannot meet the power requirements at this stage. Most of the vehicle's operating time is in low to medium regime, with gasoline being used only when operating at high loads.

2. Materials and Methods

In this study, we used a DA465QE engine mounted on a 750kg Thaco Towner light truck. This engine was manufactured and assembled by the Chu Lai-Truong Hai Automobile Factory using modern lines. The engine has a 65.5mm cylinder diameter, 72mm piston stroke, and a power output of 35 kW at 5000 rpm and 72 Nm of torque at 3500 rpm when running on gasoline. The engine was modified to run on a biogas-gasoline hybrid fuel by adding a venturi throat and biogas injector to the intake. The computational model includes a cylinder with a piston, combustion chamber, intake manifold, and TDC combustion chamber (Figure 1). The simulation was conducted using ANSYS FLUENT software and the input parameters were engine speed, fuel composition, pressure, supply method, and throttle position (Table 1). The setting of the model and boundary conditions are described in [3]. In this work, the equation system was closed by the Re-Normalized Group RNG k-ε model. The thermodynamic parameters of the mixture are calculated using the Partially Premixed model. Whenever we change the fuel, we recalculate the thermodynamic parameter probability density function (PDF) table. Simulation of the supply process, combustion and pollutant emissions of the DA465QE engine has been presented in our previous works [4, 5]. Experimental research is shown in the work [6]. The following part will assess the performance and pollutant emissions by engine using biogas/gasoline hybrid fuel.

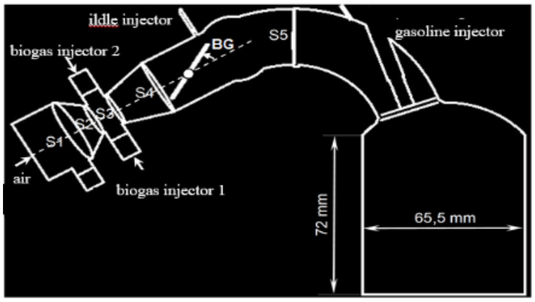


Figure 1. *Survey sections on the intake engine*

Table 1. *Calculated values of simulated boundary conditions*

Parameter	Unit	Value
Engine Speed (n)	rpm	2000; 3000; 3500; 4000; 5000; 6000; 7000.
Biogas component	%	M6C4; M7C3; M8C2; M100.
Gasoline content blend biogas	%	G0; G5; G10; G15; G20; G30; G100.
Throttle opening (α)	%	0; 30; 45; 100.

3. Results and Discussion

3.1. Assessment of engine performance

Figure 2 introduces the influence of the gasoline component blended into the Biogas M7C3 on the cycle indicator work of the engine. At the same operating conditions, the higher the gasoline content blended into the biogas, the higher the maximum pressure, and the larger the area of the cycle indicator work. (At the same operating conditions, the higher the gasoline content blended into the biogas, the higher the maximum pressure, and the larger the area of the cycle indicator work.") (ratio of the cycle indicator work and engine power). The above results show that when the engine runs on M7C3 biogas at a speed of 3000 rpm, the cycle indicator work has 244J/ct, but when 30% gasoline is blended into Biogas, the cycle indicator work increases to 294J/ct, about 20% increase.

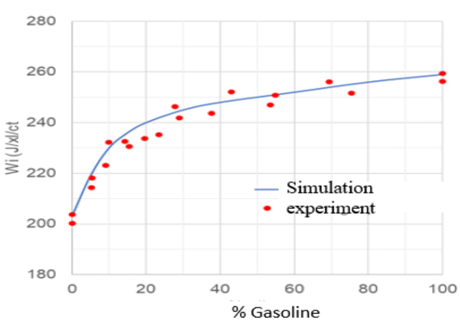
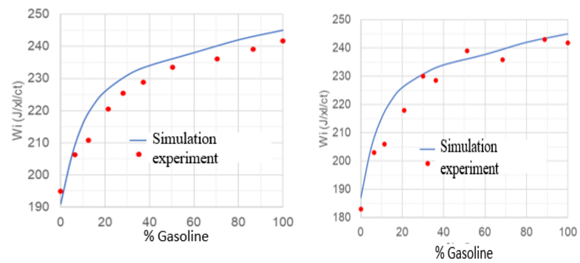


Figure 2. *Comparison of cycle indicator work variation to gasoline content by simulation and experiment (M7C3 biogas, n=3000 rpm)*

This result shows that low gasoline content, the effect of gasoline content on cycle indicator work is higher than with high gasoline content.

Figure 3 shows that the results given by simulation and experiment have a deviation of no more than 5%. The experimental measurements are lower than the simulation results because, in fact, the combustion is not as perfect as in the simulation calculation Table 2 and Figure 3a show that n=5000 rpm, the comparison of Wi to gasoline added to biogas follows the same rule as n=3000 rpm. However, the cycle indicator work value corresponding to a given gasoline content is smaller. Because the engine's intake coefficient decreases with increasing speed. Figure 3 shows that the experimentally given Wi value is about 5% smaller than the simulated value at most gasoline content values.



a *b*

Figure 3. *Comparison of cycle indicator work to gasoline content blended to M7C3 (a) and M6C4 (b) by simulation and experiment (n = 5000 rpm)*

Table 2. *Comparison of cycle indicator work by simulation and experiment (M7C3, n=5000 rpm)*

Simulation result		Experimental results	
G	W_i (J/xl/ct)	G	Wi (J/xl/ct)
0	191	0	194.93
5	206	6.33	206.3
10	216	12.48	210.76
15	222	21.45	220.54
20	226	28.13	225.39
30	231	37.27	228.81
40	234	50.29	233.5
60	238	70.33	236.12
80	242	86.68	239.07
100	245	100	241.77

Table 3. *Comparison of cycle indicator work by simulation and experiment (M6C4, n=5000 rpm)*

Simulation (M6C4, 5000 rpm)		Experiment	
G	Wi (J/xl/ct)	G	Wi (J/xl/ct)
0	186.98	0	183
5	204.63	6.5	203
10	215.16	11.42	206
15	221.71	20.87	218
20	226	30.12	230
30	230.71	36.24	228.52
40	234	51.29	239
60	237.7	68.42	235.82
80	242	88.67	243
100	245	100	241.77

Explained by 3 reasons. One is because the actual intake coefficient is smaller than the theoretical coefficient. The second is because the actual early ignition angle (designed for gasoline) is unsuitable for the gasoline-biogas mixture. Third, the actual combustion process is deteriorated in the high-velocity region to incomplete combustion.

The case of M6C4 (Table 3 and Figure 3b), it shows that the Wi results given by the simulation are consistent with the experimental results. Compared with the case of M7C3 (Figure 3a), when the gasoline content is less than 30% Wi in the case of M6C4 is smaller. But when the gasoline content is greater than 30%, the Wi given by the two cases is equivalent.

Figure 4 compares the torque and power Characteristics given by the simulation and experiment when the engine runs on gasoline, with M7C3 biogas and M7C3 biogas blened with 30% gasoline. Because the pressure in the combustion chamber is not measured, the torque and power of the engine are calculated from the cycle indicator work by the mechanical efficiency $P_e = i.\eta_m.W_i.n/120$.

Figure 4 shows that the maximum torque at the engine speed is about 3500. This is exactly with the data published by the manufacturer. Experimental results at most measuring points are about 10% smaller on average than simulation results. Because the actual combustion is not as ideal as assumed in the simulation. The actual early ignition angle is installed in the ECU by the manufacturer for gasoline fuel, not completely suitable for biogas fuel and biogas blend gasoline. On the other hand, the pressure on the intake manifold is also not properly included in the simulated conditions, to an error between simulation and experiment.

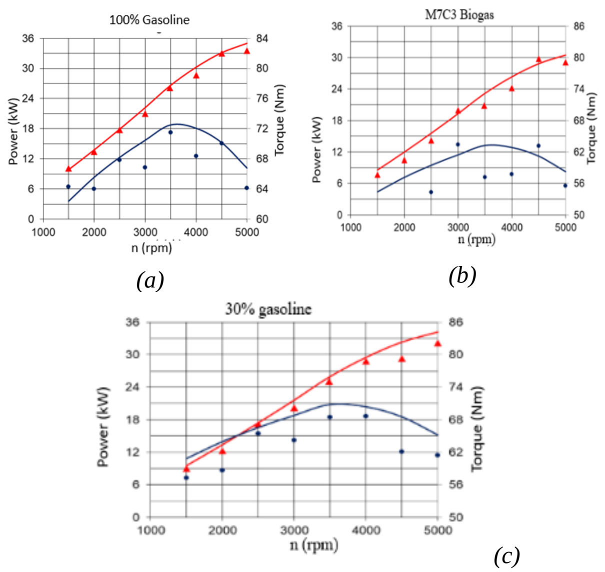


Figure 4. *Compare the torque and power characteristics given by the simulation and experiment when the engine runs on gasoline (a), M7C3 biogas (b) and M7C3-30G (▲: Experimental power, •: Experiment torque)*

3.2. Assessment of pollutant emission

Figure 5a, b, c compares the effects of gasoline content blended with M7C3 biogas to CO, HC and NO_x emissions given by simulation and experiment when the engine runs at 3000 rpm. When increasing the gasoline content in the fuel mixture, the amount of CO_2 inert gas is reduced, the burning speed is improved to help the combustion process take more completely. Figure 5a shows that the CO concentration given by the experiment is 5-20% higher than that given by the simulation. This can be explained because the actual combustion is not as ideal as assumed in the simulation. The composition of the mixture in actual is also not completely homogeneous. Therefore, incomplete combustion can take locally, to higher CO emissions in actual than calculated values.

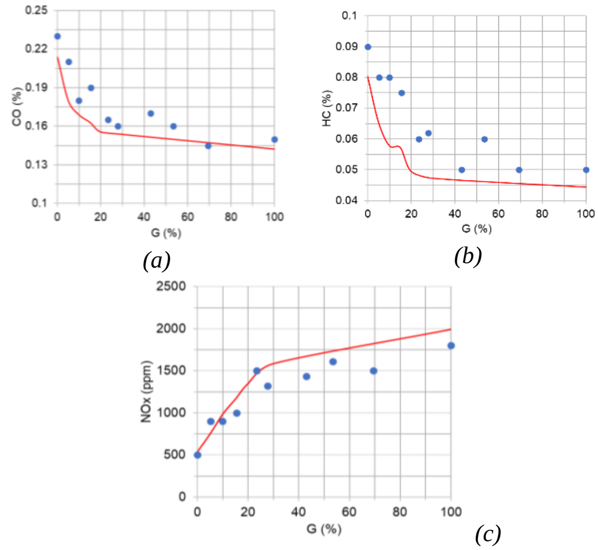


Figure 5. *Comparison of the effect of gasoline content blended with M7C3 biogas to CO (a), HC (b) and NO_x (c) emissions is given by simulation and experiment (n=3000 rpm, —:simulation, •: experiment)*

Similarly, CO, HC emissions given by the experiment is also 10-20% higher than the simulated calculated value due to the fact that the combustion is not completely localized in reality (Figure 5b).

Unlike CO and HC, the NO_x concentration given by the experiment is about 10% lower than the simulated value. NO_x formation is highly dependent on combustion temperature.

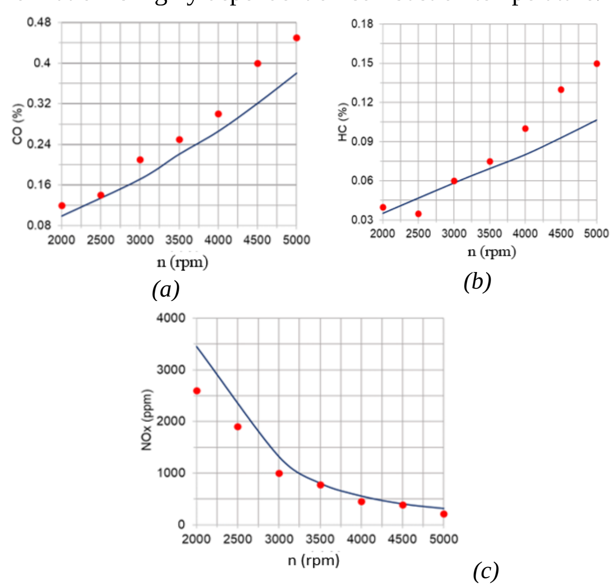


Figure 6. *Compare the effect of speed to CO (a), HC (b) and NO_x (c) emissions given by simulation and experiment (M7C3-30G, ϕ=1, φ_s=25°CA, n=3000 rpm, —:Simulation, •: experiment)*

Figure 6a introduces the variation of CO concentration to engine speed when running on M7C3-30G fuel given by simulation and experiment. We see the same fuel supply conditions, CO concentration increases with increasing engine speed. Because as the speed increases, the burning time decreases leading to incomplete combustion increasing CO emissions. The results above 10a show that the CO concentration in the experiment is about 10% higher than that given by the simulation in the low speed region and increases to about 20% in the high speed region. This is because in our simulation calculations we assume complete combustion in an ideal homogeneous mixture. in fact, there are always areas of unusual mixtures to incomplete combustion locally. The higher the speed, the lower the actual combustion quality. That is why the difference between experiment and simulation increases with speed.

For HC emissions, the law of evolution to engine speed is similar to that of CO. Figure 6b shows that the value given by the simulation is consistent with the experimental value in the low speed region but in the high speed region, the experimental value is higher than the simulated value. At the speed of 5000 rpm, the HC concentration value given by the experiment is 1.5 times higher than the simulated value. This can be explained similarly to the case of CO.

Figure 6c shows the variation of NO_x to engine speed given by simulation and experiment when running on biogas. We see that both simulation and experiment show that NO_x concentration decreases with increasing engine speed.

This was explained in the simulation by two reasons. First, because the maximum temperature decreases with increasing engine speed, NO_x formation decreases. Secondly, because the time when the combustion mixture exists at a high temperature decreases with increasing engine speed, it reduces the NO_x concentration during combustion. The results of Figure 10c show that the simulation results are consistent with the experimental results in the high-speed region because NO_x formation is slightly reduced. In the low speed region, the NO_x concentration given by the experiment is about 15% smaller than the value given by the simulation on average. This is because the effect of temperature on NO_x formation is more significant when the time the mixture is present at a high temperature is prolonged.

4. Conclusion

The results of this study allow us to draw the following conclusions:

- Full-methane hybrid engine (M100) has a maximum Wi reduction of 13% compared to gasoline engine. When running on M6C4 biogas, the cycle indicator work decreased the most by 27% compared to gasoline, 17% compared to methane and 10.5% compared to biogas M8C2.

- Full-Methane hybrid engine has the smallest HC and CO emissions. When gasoline hybrid engines has the highest NO_x emissions, while M6C4 biogas engines have the highest CO emissions but the lowest NO_x emissions.

- The combination of using gasoline/biogas, i.e. the combination of gasoline/methane/carbon dioxide can help the engine achieve a harmony between performance and pollutant emissions.

- The optimum gasoline content for mixing M7C3 biogas is 30%. Cycle indicator work and NO_x concentration increased very quickly while CO, HC emissions decreased sharply with the content of gasoline mixed into biogas when the gasoline content was less than 30%. After this value, the improvement in engine performance is insignificant.

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NOMENCLATURE

CA Degree crankshaft angle;

MxCy Biogas constituted by 10x% CH_4 and 10y% CO_2 by volume;

ϕ Equivalence ratio;

φ Crankshaft angle;

n Engine speed (rpm);

π Pressure (bar);

α Rotation angle of butterfly valve in range of 0° and 60°;

TDC Top dead center;

Gx x percent gasoline fuel blend with biogas.

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