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Mixer Design for A High Performance Biogas SI Engine Converted From A Diesel Engine

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Abstract

The performance and pollution emission of biogas engines strongly depend on the fuel-air equivalence ratio of the mixture. Thus an appropriate design of the mixer is the main issue in converting an existing diesel engine into a biogas spark ignition engine.

This paper presents some results of simulation and experiment research on a venturi type mixer for a biogas SI engine converted from a ZH1115 diesel engine. The results show that the fuel-air equivalence ratio of the mixture is less dependent on the opening of the butterfly valve which controls the mixture but it sharply depends on CH_4 concentration in the biogas and/or on section of the biogas supplying pipe. At full load, the fuel-air equivalence ratio is slightly changed in relation to the engine speed but at partial load, it strongly depends on the engine speed, particularly at low regime. The dimensionless diameter of the biogas supplying pipe can be expressed by a power relationship with CH_4 concentration in biogas with an exponent of -0.515.

Keywords: *Biogas, Renewable energy, SI engine, Simulation, Mixer*

Nomenclatures

n	Engine speed (rpm)
p	Pressure (Pa)
m	Mass flow rate (kg/s)
V	Flow velocity (m/s)
S	Section (m^2)
x	Percentage of CH4 in biogas (%)
$y=d_{eq}/d_{ad}$	Dimensionless diameter
d_{eq}	Equivalent diameter of biogas supplying pipe (m)
d_{ad}	Diameter of air admission pipe (m)
P_e	Effective power of the engine (HP)
RAFR	Relative Air Fuel Ratio
EAR	Excess Air Ratio
MiCj	Biogas containing 10i% methane and 10j% carbonic in volume
ϕ	Fuel air equivalence ratio
α	Opening angle of the butterfly valve (°)
β	Opening angle of biogas ball valve (°)
ρ	Density (kg/m^3)
Δp	Pressure differential (Pa)

I. Introduction

Biogas is an attractive renewable source of energy for rural areas. It can be produced from organic wastes, such as dung of animals, plant matter and other wastes of agriculture production. Approximately two-thirds of biogas (in volume) is methane and the rest is mostly carbon dioxide. As a fuel, biogas has a low energy density on the volume because of its high CO_2 content. The burning velocity of biogas is low, just at 25 cm/s as against 38 cm/s for LPG, due to the reason that carbon dioxide may change the combustion behavior of the air–fuel mixture. A large quantity of CO_2 present in biogas lowers its calorific value, burning velocity and flammability range compared with those of natural gas. The self-ignition temperature of biogas is high and hence it resists knocking which is desirable for engines with a relatively high compression ratio to maximize thermal efficiency.

Power and thermal efficiency of biogas engines reached their highest values with the *RAFR* between 1.05 and 0.95 [1]. Under these conditions, HC and CO emissions were relatively low but the NO_x values were relatively high. Power and thermal efficiency were reduced for leaner mixtures, particularly, though engine speed was increased, emissions were all reduced [1]. Mixtures richer or leaner than this optimal point will cause incomplete combustion or slow down the burning rate and hence lead to a drop in thermal efficiency. Chulyoung Jeong et al. observed that the maximum values of generating efficiency, cylinder pressure, and NO_x emissions were obtained at an *EAR* of around 1.2 [2], which is slightly higher than the values reported by [1].

A high compression ratio spark ignition engine for biogas can be built by replacing the injectors of a diesel engine by a spark plug and modifying the pistons. It is necessary to maintain a proper ratio between the fuel gas and air in order to attain good combustion [3]. However, since NO_x production in this condition is relatively high, two alternative approaches, slight retardation of spark timing or a little leaner operation which can be applied to control NO_x emission without sacrificing considerable thermal efficiency, were suggested as an optimum and practical operating point for use in an actual biogas site in the future [4].

The supply of the right mixture of air and fuel is therefore of utmost importance for the performance of a biogas spark ignition engine [5]. Further, the engines operating close to the stoichiometric air/fuel ratio display lower levels of emissions of toxic gases. Enhanced methane concentration in biogas (as in methane enriched biogas) significantly improves the engine performance and reduces emissions of hydrocarbons [5].

Thus it is necessary to design appropriate mixers in order to ensure the right mixture with various biogas composition and pressure. In Vietnam, the research team GATEC of the University of Danang [6] has carried out a lot of studies on biogas engine. The results of researches allowed application of biogas on engines in rural areas which is very helpful for climate change mitigation [7]. As the original engines are diverse in structure and dimensions, it is difficult and costly to carry out experiments for determination of basic parameters of appropriate mixers [8-9]. The simulation of mixers will be useful to predict characteristics of the flow under different operation conditions so that we can identify basic dimensions of the mixer corresponding to the size of each engine.

In this research, we study the characteristics of a mixer designed for a biogas spark ignition engine converted from a Jandong ZH1115 diesel engine. The objective of the research is to identify the section of biogas supplying pipe to ensure normal operation of the engine fueled with different components biogas. The experiences taken from this research can be applied on other kinds of diesel engine.

II. Method of Study

1. Mixer Design

The ZH1115 diesel engine with bore of 115mm, stroke of 115mm, and compression ratio of 17 reaches power of 24HP at rated speed of 2,200 rpm. The engine is converted into a biogas spark ignition engine by replacing injection systems with an electronic spark ignition system and a mixer mounted on the intake manifold.

Figure 1 shows the longitudinal section of the mixer system for this ZH1115 engine. A venturi injector of 33mm interior diameter is mounted on a 2.5mm diameter pipe at the narrowest section of biogas supply. The mixer is disposed of for 2 valves: a ball valve for biogas flow control with flow diameter of 18mm and a butterfly valve for mixture flow control. The open angle of the biogas ball valve (compared to the vertical axis) changes from $\beta = 0$ (completely closed) to $\beta = 90$ (fully open). The open angle of the butterfly valve (compared to the center line of the mixer) changes from $\alpha = 0$ (fully open) to $\alpha = 70°$ (fully closed). Relationships between open angle α and the flow section of the intake manifold are shown in Table 1.

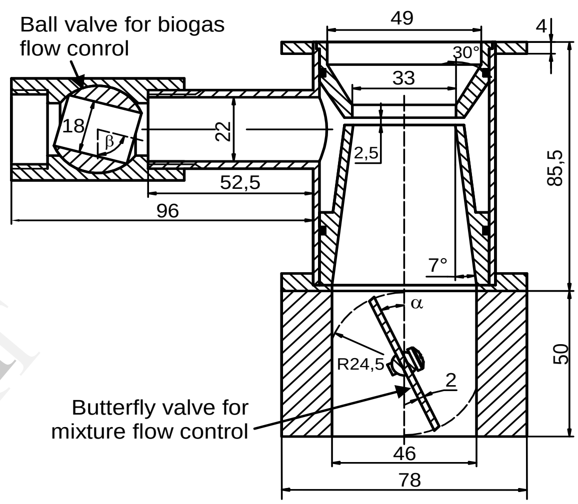


Figure 1: Mixer designed for biogas SI engine converted from ZH1115 biogas engine

Table 1: Relationship between opening of butterfly valve and flow section of intake manifold

α (°)	0	10	20	30	40	50	55	70
Flow section (%)	100%	97%	87%	72%	53%	34%	21%	0%

2. Experimental setup

The experimental setup is introduced in Figure 2. This experimental testing is conducted with the Froude dynamometer on site of biogas production. The ZH1115 engine is converted into the spark ignition engine with the compression ratio of 12. The biogas supplying pipe with variable diameter in accordance with the CH_4 concentrations of biogas. Biogas from the digester is conducted by two different filtration systems. The first system removes only H_2S by bentonite. The second system removes simultaneously H_2S and CO_2 by means of NaOH solution. Biogas mixture coming from these two sources with different concentrations of CH_4 is supplied to the engine. Compositions of biogas are measured by a biogas analyzer GFM 435. Air

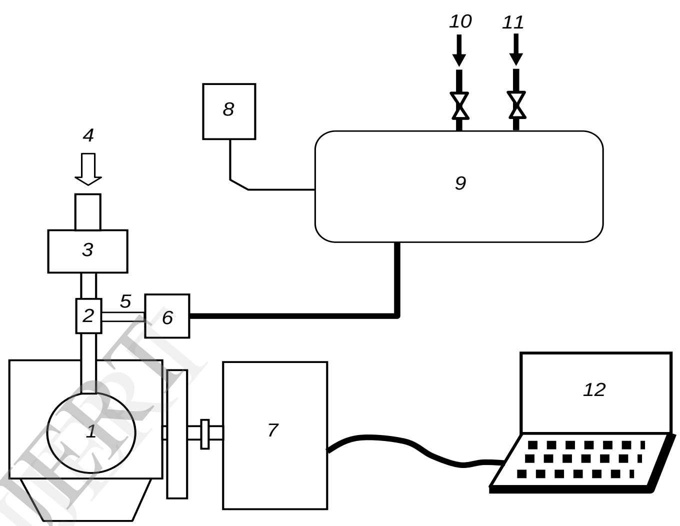


Figure 2: Experimental setup
1. ZH1115 biogas engine; 2. Mixer; 3. Air flowmeter; 4. Air inlet; 5. Biogas supplying pipe; 6. Biogas flowmeter; 7. Froude dynamometer; 8. GFM 435 Biogas analyzer; 9. Biogas mixture bag; 10. Rich biogas source; 11. Poor biogas source; 12. Computer

mass flow is measured by ABB a flow meter. Biogas mass flow is measured by a Sigma flow meter.

Experimental data are transferred to computer via A/D card and Labview software.

3. Simulation

The simulation study has been conducted by using the FLUENT CFD Software in ANSYS 14. The 3D meshing of the mixer (Figure 3) has been established by means of the SOLIDWORKS Software. The boundary conditions include air inlet pressure p_in_air = 0, biogas inlet pressure p_biogas_in = 50Pa. The mixture outlet pressure is calculated based on the crankshaft speed and the structure parameters of the engine.

Density of mixture is supposed to be ρ=1,293kg/m^3. Average mass flow rate of mixture during the intake process is $m = \frac{1.293.V_h.2n}{60}$ (kg/s). The vacuum pressure on the intake manifold is $\Delta p = \frac{1}{2}\rho V^2$. Mass flow rate of mixture is also given as $m = K\rho SV$. In other words, mass flow rate m is proportional to speed V whereas Δp is proportional to V^2, so it is proportional to m^2.

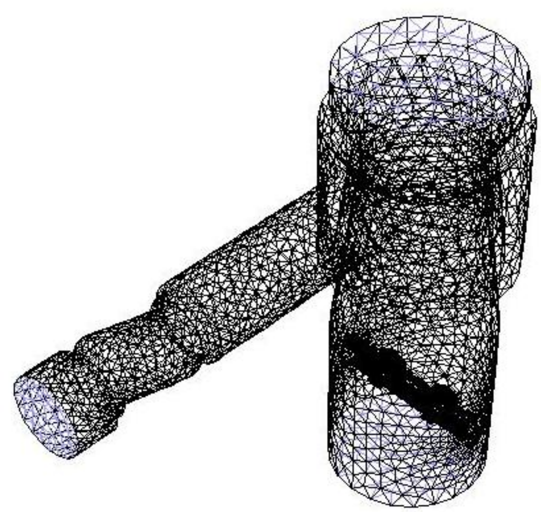


Figure 3: Meshing 3D mixer SOLIDWORKS Software

In order to exclude the coefficient of proportion in the FLUENT calculation, we firstly suppose boundary conditions p_mix_out as p_mix_out_propos. The calculation results will give us the supposed mass flow rate: mmix_out_propos (kg/s). Hence the pressure at mixer outlet with known mass flow rate of the mixture during intake process m_n will be identified by the following expression:

$$p\_mix\_out = p\_mix\_out\_propos\left(\frac{m_n}{m_{mix\_out\_propos}}\right)^2$$

Table 2: Boundary conditions at mixer outlet

n (rpm)	1000	1200	1400	1600	1800	2000	2200
p_mix_out (Pa)	-1608	-2316	-3153	-4118	-5212	-6435	-7786

Pressure at the mixer outlet p_mix_out when the open angle of the butterfly valve is 30°, the open angle of the biogas ball valve is 75°, and the biogas fuel contains 70 CH_4 is illustrated in Table 2.

With mass flows of air and biogas given by simulation of each case, we can then calculate the equivalence ratio of mixture supplied to the engine.

Figures 4a, 4b, 4c and 4d introduce calculation results of velocity field, contour of dynamic pressure, contour of O_2 and CH_4 concentrations on the symmetrical surface of the mixer with open angle of the butterfly valve at 30°. We can predict homogeneity of mixture through the mixer with help based on these results.

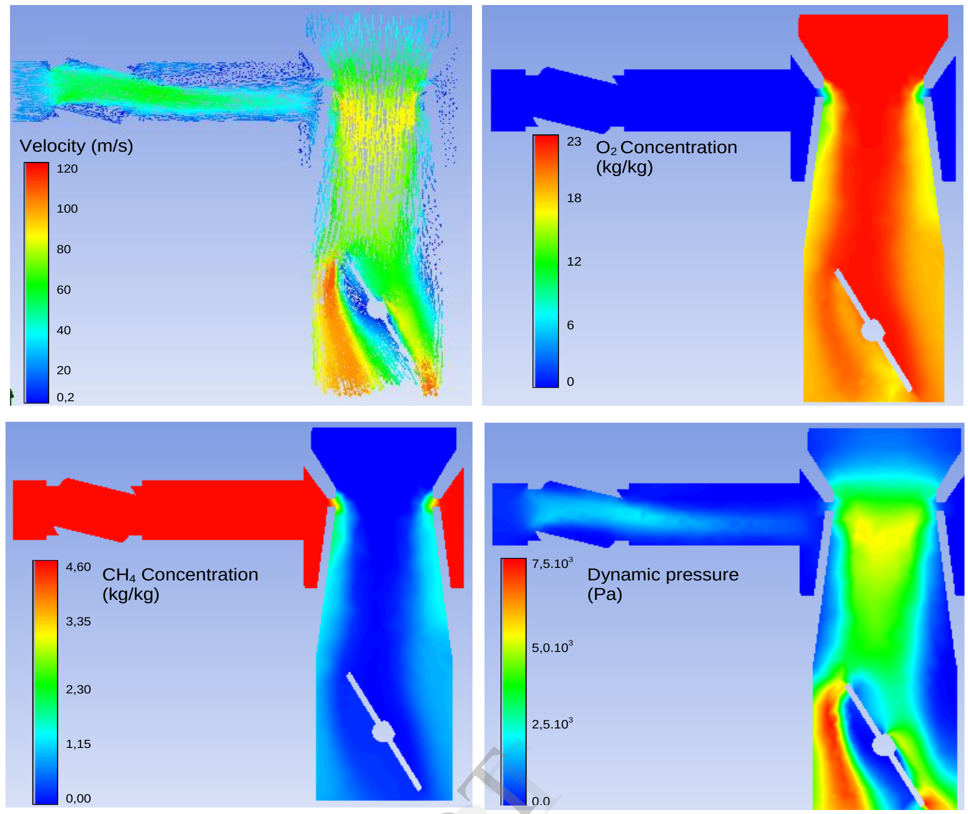


Figure 4: Flow velocity vector (a), contour of dynamic pressure (b), concentrations of CH_4 (c) and O_2 (d) on the symmetrical plane of the mixer.

III. Results and Discussion

1. Simulation Prediction

Figure 5a introduces variations of fuel-air equivalence ratio ϕ of the mixture versus speed of an/the engine fueled with biogas containing 60% of CH_4. The biogas ball valve is fully opened (90°). The butterfly valve is opened at positions of 34%, 72%, and 100%, respectively. The engine speed in each case ranges between 1,000 rpm and 2,200 rpm. The calculation results show that when an engine operates on full load curves (with butterfly valve being 100% open), ϕ of mixture is almost stable (ϕ changes from 1.3 to 1.4). When the engine operates on partial load curves, the curve of the fuel-air equivalence ratio varies in function of engine speed: the steeper the engine speed is, the smaller the butterfly valve open level becomes. The change in concentrations at high-speed positions is less than at low-speed positions. At any opening levels of the butterfly valve, when the engine runs at rated speeds between 1,800 rpm and 2,200 rpm, fuel-air equivalence ratio of the mixture changes narrowly from 1.02 to 1.10.

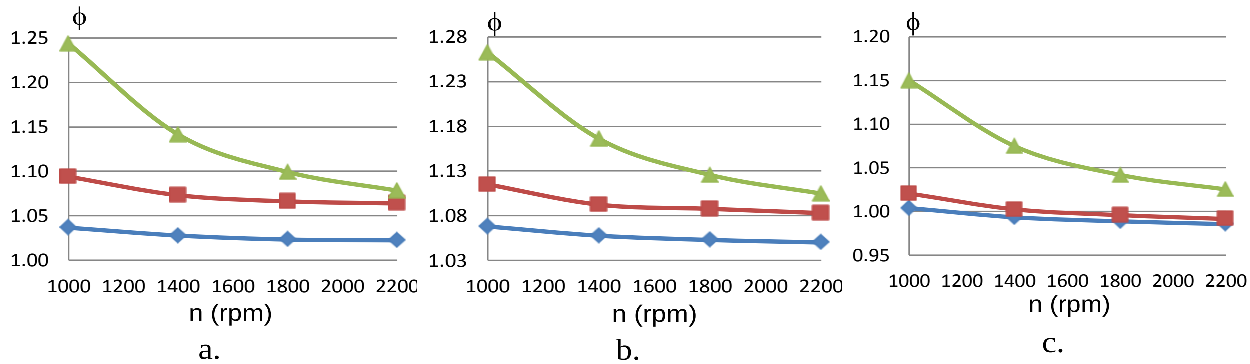


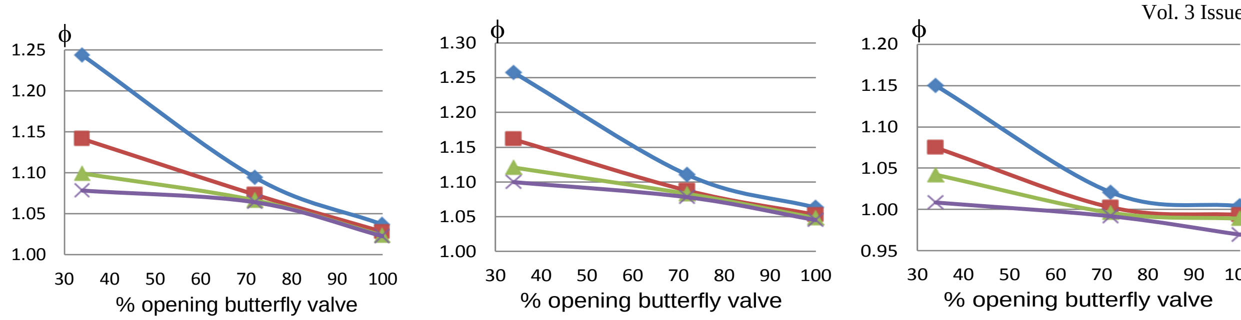
Figure 5: Fuel-air equivalence ratio ϕ vs engine speed
M6C4, ball valve 90°(a); M7C3, ball valve 75°(b); M9C1, ball valve 60°(d);
Butterfly valve opening 34% (▲),72% (■) and 100% (◆)

Figure 5b introduces the same results with biogas fuel containing 70% of CH_4 and an opening level of the biogas ball valve at 75°. Figure 5c introduces the same results with biogas fuel containing 90% of CH_4, with the opening level of the biogas ball valve at the position of 60°. The results manifest the changing principle of ϕ in terms of n similar to the case in which biogas contains 60% of CH_4. With biogas containing 90% of CH_4, if the opening level of the biogas ball valve is at 60°, the mixture is poor. In order to increase fuel-air equivalence ratio ϕ of the mixture in these cases, we can increase the opening levels of the biogas ball valve. On the contrary, in the case that the fuel contains 60% - 70% of CH_4, we can reduce the opening level of the biogas ball valve so as to reduce fuel-air equivalence ratio ϕ of the mixture.

The above results show that, when opening levels of the biogas valve and the butterfly valve are given, the fuel-air equivalence ratio ϕ of the mixture is slightly reduced as decreasing of the engine speed slows. The bigger the opening level of the butterfly valve, the lower the changing degree of ϕ becomes. When CH_4 concentration in biogas is varied, we can adjust the biogas ball valve to achieve the best fuel-air equivalence ratio ϕ. This adjustment can be made once for each kind of fuel. Table 3 summarizes the results of calculations on the opening levels of the biogas ball valve corresponding to the biogas containing different percentages of CH_4 concentration. The results show that with given a CH_4 concentration, we can choose an appropriate opening level of biogas ball valve so that the fuel-air equivalence ratio ϕ is in optimal range according to [1] at any opening level of butterfly valve.

Table 3: Effect of butterfly valve opening from 34% to 100% on fuel-air equivalence ratio at given CH_4 concentration and opening level of biogas ball valve (n=2,200 rpm)

%CH_4 in biogas	60	70	80	90
Opening levels of biogas ball valve (°)	90	75	65	60
ϕ	1.02-1.08	1.03-1.09	0.97-1.01	0.98-1.02



a. b. c.

Figure 6: Fuel-air equivalence ratio ϕ vs opening levels of butterfly valve; M6C4, ball valve 90° (a); M7C3, ball valve 75° (b); M9C1, ball valve 60° (c); n=1,000 rpm (◆), n=1,400 rpm (■); n=1,800 rpm (▲) and n=2,200 rpm (✕)

Figures 6a, 6b and 6c introduce the effect of engine speed on ϕ in relation to the opening level of the butterfly valve with biogas containing 60%, 70% and 90% of CH_4 with opening levels of the biogas ball valve shown in Table 3. The results show that when engine speed and position of the biogas ball valve are fixed, fuel-air equivalence ratio ϕ of mixture is reduced as the opening levels of the butterfly valve are increased. When the butterfly valve is fully open (the engine operates on full load curves), the fuel-air equivalence ratio ϕ of the mixture is almost unaffected by the engine speed. Therefore the mixer must surely supply the best mixed components when the engine operates on full load curves. Under partial load operation, ϕ slightly increases when the opening level of the butterfly valve is reduced. The results show that with 34% opening of the butterfly valve at 1,000 rpm engine speed, ϕ is about 1.25 compared with its value of approximately 1 a fully opened butterfly valve with biogas containing 60-70% of CH_4. Even when the mixture is richer as the engine runs on partial load curves, ϕ is within combustible limit range.

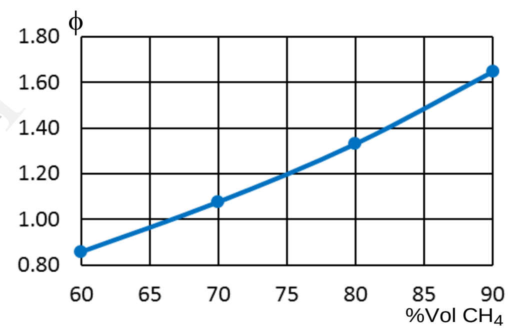


Figure 7: Mixture concentrations proportional to CH_4 concentrations in biogas (ball valve opening 75°; butterfly valve opening 72%; n=2,200rpm)

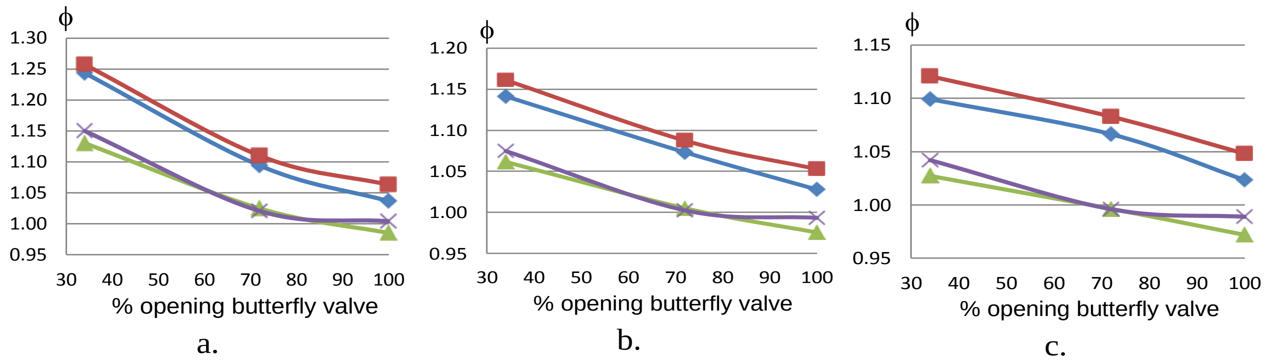


Figure 8: Fuel-air equivalence ratio ϕ vs opening levels of butterfly valve
n = 1,000 rpm (a); n = 1,800 rpm (b); n = 2,200 rpm (c);
M6C4, ball valve 90° (◆); M7C3, ball valve 75° (■);
M8C2, ball valve 65° (▲); M9C1, ball valve 60° (✕).

However, as the biogas ball valve position, butterfly valve position and engine speed are fixed, ϕ changes considerably in accordance with the concentration of CH_4 in biogas fuel. Figure 8 shows that at engine speed of 2,200 rpm, the biogas ball valve is open up to 75° and the butterfly valve is open 72%, ϕ reaches 0.85 and 1.65 with biogas containing 60% and 90% CH_4, respectively. Therefore, to obtain an appropriate fuel-air equivalence ratio ϕ of the mixture as CH_4 concentration in biogas changes, we must change the opening levels of the biogas ball valve.

Figures 8a, 8b and 8c introduce the effect of biogas fuel and opening levels of the biogas ball valve on change of ϕ in function of opening levels of the butterfly valve. With a given engine speed, the tangent of curves are almost similar, independent of CH_4 concentration in biogas fuel. So if we adjust the position of the biogas ball valve so that for a given opening level of the butterfly

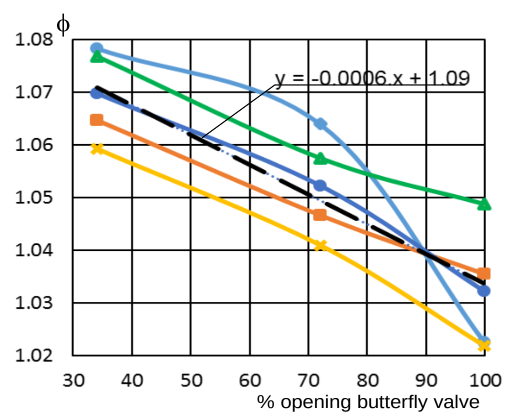


Figure 9: Variation of fuel-air equivalence ratio in accordance with butterfly valve opening levels
M6C4, ball valve 90° (◆);
M7C3, ball valve 74° (■);
M8C2, ball valve 67° (▲);
M9C1, ball valve 61° (✕).

valve we obtain the same ϕ of mixture, then we can represent a linear relationship between ϕ and the opening level of the butterfly valve.

Figure 9 illustrates the variation of ϕ in function of butterfly valve opening levels with biogas containing 60%, 70%, 80%, and 90% of CH_4 and opening levels of the biogas ball valve of 90°, 74°, 67°, and 61°, respectively. At 34% opening level of the butterfly valve, ϕ is in range between 1.06 and 1.08. When the butterfly valve is fully opened ϕ oscillates from 1.02 to 1.05. These results show that the tangent of the curves is -0.0006 (if the opening level of the butterfly valve is in percentage).

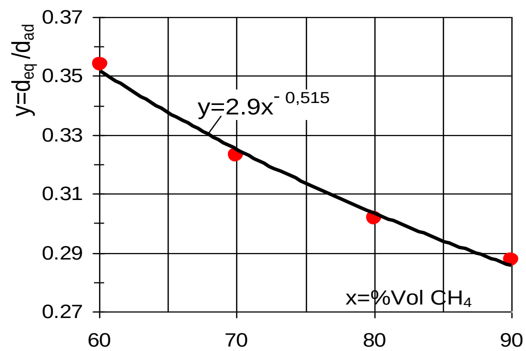


Figure 10: Relationship between dimensionless diameter of biogas supplying pipe and CH_4 concentration in biogas fuel (n=2,200 rpm, ϕ=1±0.02)

As the tangent of the curves is very slight, we can consider ϕ is unchanged in relation to opening levels of the butterfly valve. Contrarily, ϕ of the mixture changes sharply in relation to the opening levels of the biogas ball valve and CH_4 concentrations in biogas fuel. This means that for a given biogas, we need to determine the size of the pipe that supplies biogas to the mixer in relation to the size of the air admission pipe so that ϕ is in optimal range observed by Huanga et al. [1] or by Jeong et al. [2] at any level of butterfly valve opening. In this case, we need not equip the biogas ball valve with the supplying pipe.

Figure 10 introduces the variation of dimensionless diameter $y=d_{eq}/d_{ad}$ of the biogas supplying pipe in accordance with the concentration of CH_4 in the fuel in case of ZH1115 biogas engine. When CH_4 concentration in biogas increases, the amount of biogas supplied to the mixture must decrease to ensure that ϕ of the mixture is unchanged. Because mass flow rate m is proportional to the flow section S, in other words, it is proportional to the square of the dimensionless diameter of the biogas-supplying pipe *y*, or the diameter y is in proportion to $m^{0,5}$. Otherwise, to keep constant ϕ, when the CH_4 concentration in fuel increases, the mass flow rate of fuel decreases. This means that dimensionless diameter of biogas supplying pipe is proportional to $x^{-0,5}$. Figure 10 shows that the exponent of the curve is -0.515. The absolute value of the exponent is slightly higher than 0.5. This is reasonable because when the biogas mass flow rate changes, the air mass flow rate is also changed to ensure the constant value of ϕ.

2. Experiment Valuation

Figures 11a, 11b, and 11c introduce the variation of equivalence ratio versus engine speed at full load regime. Biogas contains 60%, 73%, and 87% of CH_4 concentrations. Biogas supplying pipes are selected with diameters of 18mm, 16mm, and 14.5mm, respectively, corresponding to the relationships shown in Figure 10. During experimentation, the butterfly valve is fully open. The results show that equivalence ratios given by simulation are fitted well to their values given by experiment with different CH_4 concentration in biogas. This confirms that the relationship in Figure 10 is reasonable.

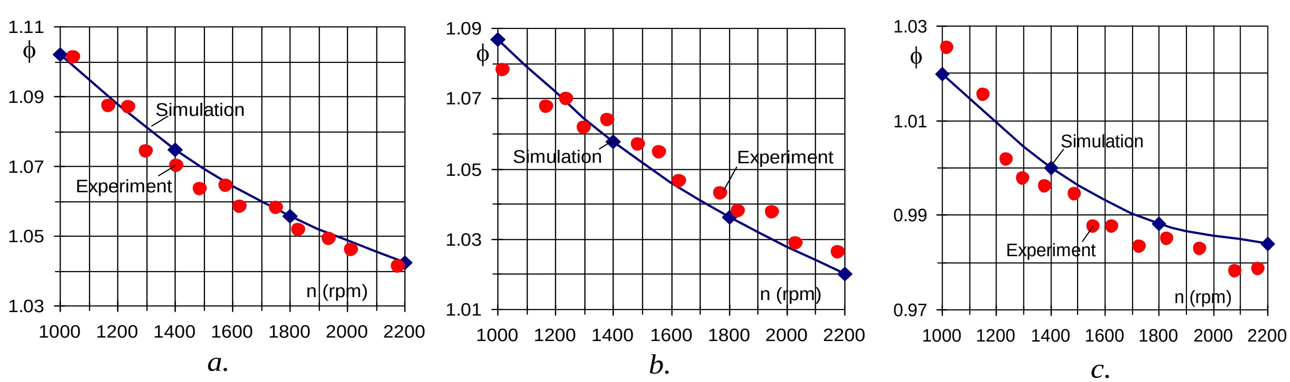


Figure 11: Comparison fuel-air equivalence ratio ϕ given by simulation and by experiment at full load regime with biogas containing 60% CH_4 (a), 73% CH_4 (b) and 87% CH_4 (c)

Figure 12 introduces full load characteristic curves of ZH biogas engine fueled with biogas containing 60%, 73%, and 87% CH_4 concentrations. The results show that at the speed of 2,500rpm, the maximum power of engine run by the biogas containing 87% of CH_4 is 21 HP. Calculated power via proportions of CH_4 passing into a/the cylinder of the engine fueled with biogas containing 73% and 60% of CH_4 will be 20.58HP and 20.03HP, respectively. The experimental data are suitable for the first case, but as for the final case (biogas containing 60% of CH_4), the real power is much lower than that of the calculation. This is because of incomplete combustion due to high concentration of CO_2 in biogas. The suitability of the power proportions when the ZH1115 engine is run by biogas containing different concentrations of CH_4 affirms that the relationships between the biogas supplying pipes with the CH_4 concentrations in biogas shown in Figure 7 are accurate.

The results of this research are very useful to convert an existing engine running on diesel that is largely used in rural areas into biogas engine. Previously for converting a diesel engine into biogas engine we have to conduct a lot of experimental tests to determine appropriate parameters of the mixer. This takes a lot of time and money. Now thanks to this new method of simulation we can orient the technology of conversion. This way can be applied generally to any kind of diesel engine. This is very helpful in reducing the cost of conversion that will encourage numerous farmers to use biogas in their machines. It is an effective contribution to climate change mitigation.

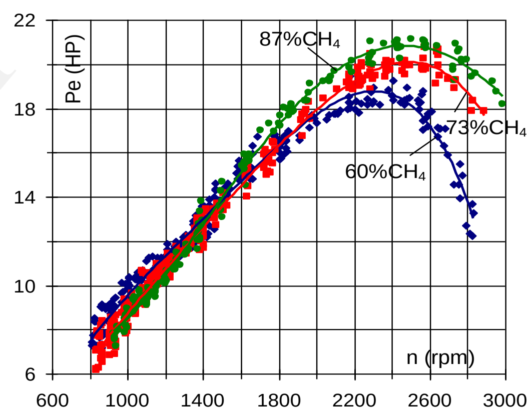


Figure 12: Performance curves of the ZH1115 when operating with biogas containing different CH_4 concentrations

IV. Conclusion

For this paper, we have studied the optimal parameters of a mixer in order to transform a typical diesel engine to a biogas spark ignition engine which can produce high effectiveness. We have drawn the following conclusions from the results.

1. With venturi type mixer designed for biogas SI engine, the equivalence ratio is less dependent on the opening of the butterfly valve which controls the mixture flow but it sharply depends on CH_4 concentration in biogas and/or on sections of the biogas supplying pipe.

2. At full load, the equivalence ratio given by the mixer is slightly changed in relation to engine speed but at partial load, it strongly depends on engine speed, particularly at low regime.

3. The dimensionless diameter of the biogas supplying pipe can be generally expressed by a power relationship with CH_4 concentration in biogas with exponent of -0.515.

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