Performance Simulation of An Air-Biogas Mixing Device Equipped With Helical Barrier

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Abstract **:** Biogas has been widely used as a petrol fuel alternative in internal combustion engine. As a carburetor, the helical barrier as an air-fuel mixer system has been designed with thermodynamic and fluid dynamics analysis to improve the quality of the mixture according to combustion requirements. The calculation result of the 4-step Otto cycle at a compression ratio of 8.5 obtained a gas flow rate of 0.135 g/s which is used as the basis for calculating the venturi and helical barrier. Then a simulation has been carried out using biogas digester fuel and according to the dimensional data of the engine being referenced. Simulation results that the helical barrier can be used to improve combustion performance and show the quality of air-biogas mixture in the improvement process compared to non-helical barrier. Thermodynamic simulation results, combustion of air-biogas mixture at a distance of 39 mm helical barrier can increase thermal efficiency by 1.29% and also combustion energy by 5%. This shows that the length of the helical barrier is best for use according to the capacity of the engine specifications.

Index Terms **-** Biogas, helical barrier, mixture quality, combustion.

I. INTRODUCTION

Biogas with a methane content higher than 45% is flammable to generates thermal energy (Deublein and Angelika, 2008). The combustion process in an internal combustion engine or thermodynamic and chemical process generates thermal energy to do mechanical work. Combustion in a gasoline engine uses a carburetor as the mixing device of air-fuel before entering the combustion chamber through the intake manifold.

The use of biogas as an internal combustion fuel comes with its own difficulties, which requires modification of some parts of the engine and some engine specifications (Awogbemi, Omojola, Adeyemo and Sunday Babatunde, 2015). Biogas with different properties requires improved mixing and fuel intake systems to reach the gasoline-fueled engine (Klaus von Mitzlaff, 1988). Modifying the biogas intake system is considered the easiest and cheapest way compared to other treatments in the combustion chamber (Rosli Abu Bakar, 2002). An experimental set up with simple modifications on existing electric generator set to supply gas fuel into engine has been studied (K. Mohan Kumar and D. Azad, 2014). However, biogas-fueled engines show a decrease in output power due to low heating values (Rosli Abu Bakar, 2002; Stefan, 2004; Jayesh D. Vaghmashi, D.R. Shah and D.C. Gosai, 2014). It is also caused by low methane content in bio (Md. Ehsan and N. Naznin, 2005) gas (Taiwo. A. and Oje. K, 2008). Modification can increase the friction factor so that it influences flow rate, pressure and speed (Hakan Bayraktar, 2003).

Based on the results of previous studies, further research and development are still needed, especially in mixing modeling, including dynamic modeling scenarios, based on the use of barriers in the mixing device. The flow can be reviewed using mathematical simulation based on thermodynamic and dynamic fluid-based model (John B. Heywood, 1998; Yunus. A. Cengel and Michael. A. Boles, 2002). Calculations with thermodynamic equations of fluids from engine working fluids throughout the cycle can be numerically solved. A fluid-based dynamic model with a multidimensional model can be formulated based on the conservation of mass and energy at various times and the position of the gas in the manifold.

The objective of this research is to study the influence of helical barrier dimension in a biogas-air mixer through a thermodynamic and fluid mechanic simulation.

II. METHODS

The amount of calorific energy of biogas depends on the the calorific value of methane content. The thermodynamic parameters of biogas methane under standard conditions (temperature 273 K, pressure 101,325 kPa) are: specific heat of 2,165 kJ/kg K, molecular weight of 16,043 kg/kmol, density of 0.606 kg / m3 (Klaus von Mitzlaff, 1988). In calculating the biogas property in this study is in accordance with the results of measurements of the methane content in biogas originating from the same digester. Measurement of methane content is carried out using methane sensors. The test result data obtained through the use of the methane sensor is the concentration value of methane gas contained in biogas. The data obtained were then analyzed using ANOVA. The results of the data analysis concluded that there was an effect on reducing the concentration of CO_2 gas. So that in this study, the calculation regarding the biogas property used is in accordance with the measurement results of the methane content in the biogas that comes from the same digester. And in thermodynamic calculations it is also adjusted by referring to the needs in accordance with the Otto motor specification data.

A helical barrier for air-biogas mixing device was designed and used as the base to construct a mathematical model for the simulation. The mixer is intended to be located before the combustion chamber to replace the carburetor. Cross section drawing of the air-biogas mixer shown in Fig. 1.

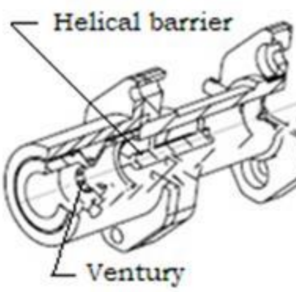

Figure 1. Cross section of ventury and helical barrier showing components

The mathematical model was based on combustion thermodynamics and fluid dynamics. Combustion results in the air-fuel mixture before entering the combustion chamber can be estimated through an Otto cycle thermodynamic analysis (Michael. J. Moran and Howard N. Saphiro, 2006; V. Ganesan, 1987; V.L. Maleev, 1945). One of the determinant of the heat energy produced in the thermodynamic combustion process in the Otto cycle is the quality of the air-fuel mixture (Michael. J. Moran and Howard N, 2006).

Calculation of combustion parameters was based on thermodynamics of a 4-stroke motor cycle (Yunus. A. Cengel and Michael. A. Boles, 2002; Michael. J. Moran and Howard N. Saphiro, 2006) by using equations (1) to (6). Where, m = mass of air (intake volumetric) can be calculated with equation (1), while mass of the fuel can be calculated with equation (2). Where *P* = 1,01325 bar (standard atmospheric pressure); *V1* = cylinder volume, cm³; $\bar{R}$ = 5.183 kJ/kmol.K (individual gas constant); *M* = 16.04 kg/kmol (molecular weight); *T* = 298,15 K (temperature); *m*= mass of mixture (intake volumetric) calculated by:

$$m = \frac{P_1 V_1}{(\bar{R}/M)T_1} \qquad (1)$$

$$m_f = \frac{m}{(AFR+1)} \qquad (2)$$

$$W_{cyc} = m[(u_3 - u_4) - (u_2 - u_1)] \qquad (3)$$

$$mep = \frac{W_{cyc}}{V_1 - V_2} = \frac{W_{cyc}}{V_1\left(1 - \frac{V_2}{V_1}\right)} \qquad (4)$$

Heat added (Q_{23}) and heat rejected (Q_{41}) are used to obtain net work, as in equation (3), and mean effective pressure can be calculated with equation (4), and thermal efficiency was defined as in equation (5).

Determine of thermal efficiency by correlation $\eta_t = 1 - \frac{Q_{41}/m}{Q_{23}/m} = 1 - \frac{u_4 - u_1}{u_3 - u_2}$ (5)

Here, $\frac{Q_{23}}{m}$ = work input, kJ/kg, $\frac{Q_{41}}{m}$ = heat rejected, kJ/kg and u = internal energy.

Ratio of the volume of fresh charge ($V_{ch} = cm^3$) taken during the suction stroke ($V_s = cm^3$) to the full piston displacement called volumetric efficiency (Maleev, 1945) is calculated by using equations (6).

Determine of volumetric efficiency by correlation $\eta_v = {V_{ch}}/{V_s}$ (6)

For stoichiometric condition, air fuel ratio values based on mass and volume are calculated using equations (7) and (8).

$$(aCH4 + bCO2) + c(O2 + 3.76N2) \rightarrow dH2O + eCO2 + fN2 + Energi \qquad (7)$$

$$AFR_{s-f} = \frac{c(O2+3.76N2)}{aCH4+bCO2} \qquad (8)$$

The correlation between mass of air ($m_a = kg$) and mass of fuel/biogas $(m_f = kg)$ is obtained by using equation (9).

$$AFR = \frac{m_a}{m_f} = \frac{m - m_f}{m_f} \qquad (9)$$

Table 1. Parameters of thermodynamic combustion results of air-fuel at the compression ratio cr = 8.5.

Variable	Dimension	Unit
cr (compression ratio)	8.5	-
m-mix (air and fuel)	0.233	g
mf (mass of fuel)	0.037	g
W-cyc (net work per cycle)	0.155	kJ
mep (mean effective pressure)	5.476	bar
η-t (thermal efficiency)	51.1	%
η-v (volumetric efficiency)	74.6	%

$m - \dot{f}r$ (mass flow rate)	0.135	g/s

Thermodynamic combustion process in the Otto cycle was simulated at compression ratio of 7 to 12.5 produces combustion performance as shown in Fig. 2. By simulating the combustion of the Otto air cycle thermodynamically at the compression ratio cr = 8.5, it produces combustion performance as shown in Table 1. Then the mixing device was calculated, namely venturi and helical barrier as the air-fuel mixing system. Mixing was designed to meet the needs of the combustion chamber, namely the mass and flow rate of 0.233 g and 0.135 g/s as the results of the previous calculation in Table 1.

Some dimensions adjustment according to engine technical specifications referred to by input data parameters include: biogas fuel with methane 57.5%, 222 cm^3 combustion chamber capacity and compression ratio = 8.5. The air-biogas flow rate that enters and exits through the intake manifold is obtained by the continuity equation following the ideal gas equation. With a decrease in the cross section of the channel in venturi nozzle hole and helical barrier causing an increase in velocity and flow rate.

Changes in the contour of the helical barrier and the angle of flow also cause a pressure drop along the helical barrier. The pressure drop is affected by friction along the helical barrier, in contrast to the decrease in flow section causing an increase in flow velocity and an increase in combustion chamber temperature.

The results of calculations carried out produce detailed dimensions of venturi and helical barrier as shown in Table 2 with cross section in Fig. 1. Table 2 shows from the results of the design planning obtained detailed dimensions of venturi and helical barrier by adjusting the dimensions of the engine carburetor referred. The diameter and length of the venturi are 18 mm and 50 mm respectively, the diameter of the nozzle hole of 2 mm and the number of nozzle of 8. The helical barrier with length by 39 mm and a shaft diameter of 2 mm with forming an angle of 30 is the most suitable dimension for the machine being referred as shown in detail in Fig. 1.

To determine the effect of using a helical barrier and non-helical barrier to the combustion of the air-biogas mixture, a simulation was performed using the mixing device data in Table 2. The simulation results of the effect of using a helical barrier and non-helical barrier as shown in Fig. 3 and 4.

Table 2. Dimensions of the helical barrier as an air-biogas mixing device used.

Variable	Dimension	Unit
Length of venturi	50	mm
Diameter and nozzle number	2/8	mm
Diameter of cylinder sleeve	18	mm
Length of cylinder sleeve	78	mm
Length of helical barrier	39	mm
Diameter of helical barrier shaft	2	mm

The method of this research was designed and used as the base to construct a mathematical model for the simulation. The simulation performed by calculating the gas flow rate and length of the helical barrier, in order to obtain a helical barrier contour according to the requirements and specifications of the engine referenced. The helical barrier is inside a cylinder sleeve placed before intake manifold. Detailed drawing of the helical barrier was the result of the development of the carburetor on the Otto engine as shown in Fig. 1. The thermodynamic simulation parameters for the helical barrier analyzed include: flow rate, work-cycle, mean effective pressure, combustion energy, thermal efficiency and volumetric efficiency, respectively.

This research can also show the basic calculation of helical barrier and its effect on the quality of an air-biogas mixture based on thermodynamic combustion applied to 222 cm^3 engines, compression ratio = 8.5. And with 57.5% methane content in the biogas used, an effective helical barrier length can be determined to produce the maximum air-biogas mixing quality.

To see a change in the helical barrier, a simulation of the flow rate of air-biogas mixture along the helical barrier is carried out. Based on the dynamic conditions the results of the simulation are analyzed including of the velocity and pressure distribution.

Biogas distribution through several venturi holes spray to the center of the main channel and mixed it with air and through the helical barrier causes irregular flow before entering the combustion chamber. So that it's expected to increase the air-biogas homogeneity in such a way as to increase combustion condition.

III. RESULTS AND DISCUSSION

To see a change in the helical barrier, a simulation of the flow rate of air-biogas mixture along the helical barrier is carried out. Based on the dynamic conditions, the results of the simulation are analyzed including of the velocity and pressure distribution. Biogas distribution through several ventury holes spray to the center of the main channel and mixed it with air and through the helical barrier causes irregular flow before entering the combustion chamber. So that, it's expected to increase the air-biogas homogeneity in such a way as to increase combustion condition. Thermodynamic combustion process in the 4-stroke Otto cycle was simulated at compression ratio of 7 to 12.5 produces combustion performance as shown in Fig. 2. The calculation results of solved by numerical combustion parameters based on thermodynamics with the 4-stroke Otto cycle are shown in Figure 3. Figure 3 shows the correlation of each combustion performance parameter including: mass of mixture (m-m), work cycle (*W-cyc*), mean effective pressure (*mep*), thermal efficiency (η-*t*), volumetric efficiency (η-*v*) and flow rate gas (*fr*) to the variation of compression ratio.

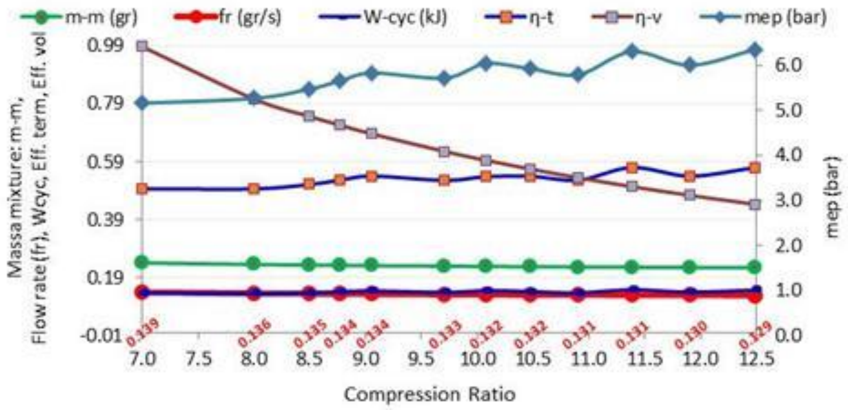

Figure 2: Thermodynamic performance of combustion biogas fuel of Otto cycle for various compression ratios

Figure 3 shows the correlation of each combustion performance parameter including: mass of mixture (*m-m*), work cycle (*W-cyc*), mean effective pressure (*mep*), thermal efficiency (*η-t*), volumetric efficiency (*η-v*) and flow rate gas (*fr*) to the variation of compression ratio 7 to 12.5.

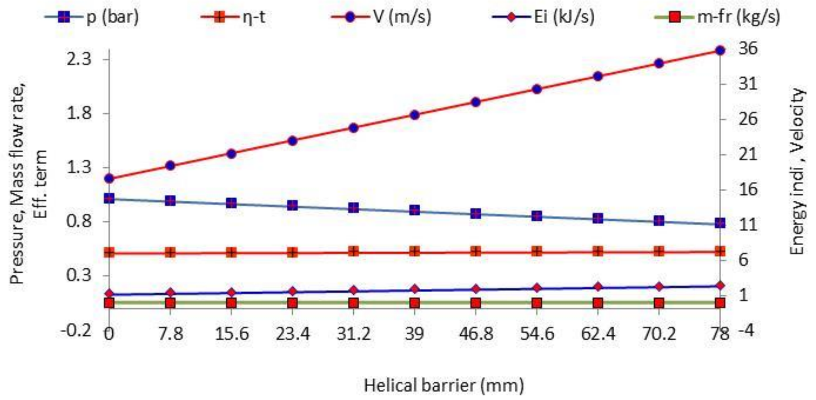

Figure 3 The performance combustion of biogas fuel for 4-stroke Otto motor using a helical barrier.

To determine the effect of using a helical barrier and non-helical barrier to the combustion of the air-biogas mixture, a simulation was performed using the mixing device data in Table 2. The simulation results of the effect of using a helical barrier and non-helix barrier are shown in Figure 2, 3 and 4.

To see the effect of the helical barrier on the combustion performance parameters with the length of the helical barrier variation. Based on the simulation results of a compression ratio of 8.5 have been obtained before (Table 1.), including; mass of mixture (air-fuel) (*m-m*) = 0.233 g, work cycle (*W-cyc*) = 0.155 kJ, mean effective pressure (*mep*) = 5.476 bar, thermal efficiency (*η-t*) = 51.1, volumetric efficiency (*η-v*) = 74.6 and gas flow rate (*fr*) = 0.135 g/s. Simulation result as shown in Figure 3 and Figure 4.

From Figure 3 shows the use of a helical barrier from a distance of 0 to 78 mm when compared to non-helical barrier described as follows. Pressure decreased from 1.01 bar to 0.78 bar, flow rate increased from 0.087 g/s to 0.182 g/s, thermal efficiency increased slightly from 51.1 to 52.5 and volumetric efficiency decreased from 74.5 to 69.5, increase in combustion energy from 1.10 kJ / s to 1.49 kJ / s. An increase in combustion performance shows that the quality of the air-biogas mixture is more homogeneous to produce performance when using a helical barrier.

Figure 4 shows a gradual decrease in pressure as the mixture passes through the helical barrier from 1.01 bar to reach 078 bar, but an increase in velocity from 17.54 m / s to 20.25 m/s enters the combustion chamber and the flow rate from 0.087 g/s to 0.182 g/s respectively from a distance of 0 mm to 78 mm. The improvement in combustion quality indicates the quality of the air homogeneous mixture is more homogeneous to produce performance.

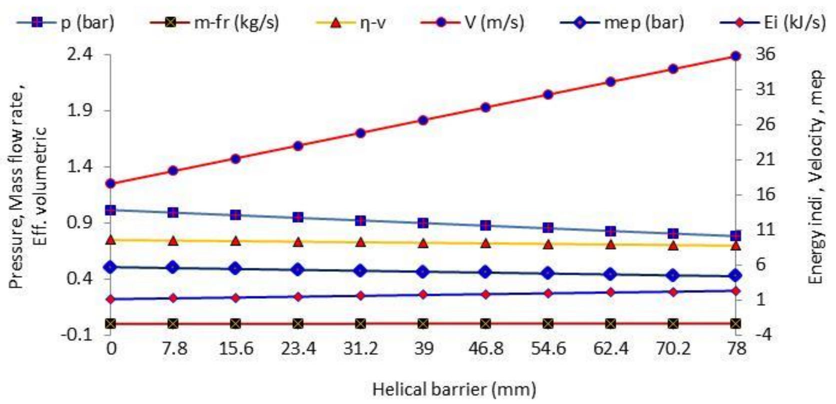

Figure 4. The simulation results of combustion performance using a helical barrier

Furthermore Figure 3 and Figure 4 show the pressure drop and the increase in flow velocity along the helical barrier can increase flow rate of fuel and mixing quality so that combustion performance increases. The figure also shows that the use of helical barrier causes the increase of volumetric efficiency, this illustrates the suitability of increasing flow rate and quality of air-biogas mixing which results in an increase in combustion results that are better to produce an increase in combustion heat as a performance parameter. Figure 4 shows the simulation results of the effect of helical barrier length on the air-biogas mixture including the parameters of velocity, pressure and air-biogas flow period. Some parameters of the air-biogas mixture after passing through the helical barrier undergo a gradual change. Pressure decreases along the helical barrier but the flow velocity increases compared to non-helical barrier.

This result of the simulation using compression ratio cr = 8.5 produces combustion performance as shown in Table 2. Then the dimension of the mixing device were calculated, namely ventury and helical barrier as the air-fuel mixing system. Mixing was designed to meet the needs of the combustion chamber, namely the mass and flow rate of 0.263 g and 0.135 g/s as the results of the previous calculation in Table 1. Some dimensions were adjusted according to engine technical specifications referred to by input data parameters include: biogas fuel with methane 57.5%, 222 cc combustion chamber capacity and compression ratio = 8.5. The air-biogas flow rate that enters and exits through the intake manifold is obtained by the continuity equation following the ideal gas equation. With a decrease in the cross section of the channel in venturi nozzle hole and helical barrier causing an increase in velocity and flow rate. Changes in the contour of the helical barrier and the angle of flow also cause a pressure drop along the helical barrier. The pressure drop is affected by friction along the helical barrier, in contrast to the decrease in flow section causing an increase in flow velocity and an increase in combustion chamber temperature.

The results of calculations carried out to produce detailed dimensions of ventury and helical barrier as shown in Table 2. Table 2 shows from the results of the design planning obtained detailed dimensions of ventury and helical barrier by adjusting the dimensions of the engine carburetor referred. The diameter and length of the venturi are 18 mm and 50 mm respectively, the diameter of the nozzle hole of 2 mm and the number of nozzle of 8. The length of the helical barrier of 39 mm, forming a helical angle of 30º and a shaft diameter of 2 mm, as the most appropriate dimensions for the engine being referenced.

From the results of ploting against the simulation results respectively in Table 1 and Figure 4, from the results of the previous thermodynamic simulation on the compression ratio = 8.5 produces the flow rate of gas entering the combustion chamber by 0.135 g/s (Table 1), the value is plotted at a distance of 39 mm helical barrier (Figure 5). Figure 4 shows a distance of 39 mm using a helical barrier compared with non-helical barrier: - a decrease in pressure from 1.01 bar to 0.78 bar, -an increase in flow velosity from 17.54 m/s to 19.04 m/s and, -an increase in flow rate from 0.087 g/s to 0.135 g/s, increase in combustion energy from 1.10 kJ / s to 0.049 kJ / s. This shows that the length of the helical barrier of 39 mm is the best for the referred engine capacity.

IV. CONCLUSIONS

The helical barrier as an alternative air-biogas mixing device as carburetor in the Otto engine has an impact on improving the quality of the air-biogas mixture when compared to non-helical barrier. The use of helical barrier in producing air-biogas combustion shows that has a positive impact in increasing performance. This research can also show the basic calculation of helical barrier and its effect on the quality of an air-biogas mixture based on thermodynamic combustion applied to 222 cm^3 engines, compression ratio = 8.5, and with 57.5% methane content in the biogas used, an effective helical barrier length can be determined to produce the maximum air-biogas mixing quality.

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