Effect of Equivalent Ratio Variation on a Two-Stage Distributed Combustion

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Abstract
Distributed combustion is a recent technology that has a feasibility to apply on the industrial scale. It provides relatively low combustion temperature with rather uniform distribution and very low emission (NOx). The two-stage distributed combustion system is established to reduce the complexity of operating in the industrial application. The operation range of this combustion system is governed by the equivalent ratio of the first and second-stages of combustion. This research is attempted to study the effects of equivalent ratio and to identify the operation range of the combustion system. There are two stages in this experiment: 1. To set up using a diesel combustor (first-stage) 2. To use an LPG distributed combustor (second-stage). Part of the aim of this project is to improve the efficiency of the single-stage diesel combustor. The air to fuel ratio of the first-stage combustor is controlled to provide the 5% excess oxygen for the second-stage combustor. The equivalent ratio of the second-stage is varied as 1, 0.9, 0.8, 0.7 and 0.6. The results have shown that stoichiometric distributed combustion yields the highest outlet temperature and first and second law efficiency. The temperature is decreased when reducing the equivalent ratio. Finally, this research also provides the formula that can be used to determine the appropriate operating range in the term of A/F for the two-stage distributed combustion system conveniently applied to the single-stage available in the industrial.

Keywords: Distributed combustion, Thermal efficiency, Experiment

1. Introduction
The biodiesel is the vegetable oil blended with diesel fuel, which is commonly used in the industry to reduce the fuel price. However, the operation of the biodiesel combustor around stoichiometric range can cause soot, carbon monoxide and nitrogen oxide (NOx). These gases are emissions which
causes pollution issues that directly impact the global environment. Lean combustion is a promising choice to reduce emissions especially soot. However, this combustion process provides lower combustion temperature and thermal efficiency. There are several methods to improve the combustion efficiency [6,11]. To date, there are few studies that have investigated the association between the distributed combustion and the first stage combustor. Currently, there are no data on suitable operation conditions for the distributed combustor [12]. Distributed combustion is the combustion that can reduce pollution especially NOx. The operating temperature range is around 800°C to 1000°C, which has potential in the industrial application [13]. At this temperature, the NOx reaction is inactivated due to the low activation energy [14-15]. The prominent point of this phenomenon is the invisible flame. In conventional combustion, the flame illuminates in yellow light. However, the distributed combustor emits the light in the infrared wavelength due to its low combustion temperature [16]. To achieve this phenomenon, it requires two conditions. The one is to lower the oxygen concentration at the inlet air of the distributed combustion stage from 21% in atmospheric condition to around 5% to 7%. The second condition requires the auto-ignition temperature to ignite and sustain combustion due to the lower inlet oxygen concentration that reduces the activation energy of air [17-18]. The design of combustor to achieve this condition can be developed by several methods such as air recirculation, exhaust gas extractor or preheated air inlet with first-stage combustor [19-22]. In addition, many works operate the disturbed combustion at stoichiometric. There are opportunities to apply distributed combustion as the second stage to take its advantages and also observe operation in lean combustion range. In this research, the biodiesel combustor is operated in lean combustion at the first stage. The air is combusted with biodiesel to provide the high temperature and lower exhaust oxygen concentration before entering the second-stage distributed combustion. LPG fuel is injected into the distributed combustion chamber and ignited by the auto-ignition phenomenon.

\[
C_{3.3}H_{8.6} + 5.74 (O_2 + 19N_2) \leftrightarrow 3.3 CO_2 + 4.3 H_2O + 109.06N_2 \tag{1}
\]

\[
\phi = \frac{(A/F)st}{(A/F)operation} \tag{2}
\]

The stoichiometric reaction of the distributed combustion is shown in equation (1) with LPG fuel that contains 70% propane and 30% butane. The reaction is written in 5% excess oxygen from the first-stage. This oxygen concentration is the widely used value for the distributed combustion. Oxygen dilution increases the nitrogen concentration percentage and is drastically increased. The combustion in higher nitrogen concentration causes reducing combustion temperature because the energy absorbed by nitrogen gas [23]. In an investigation into the operation, the oxygen concentration of the first-stage biodiesel combustor and the equivalent ratio of the second-stage distributed combustor must be controlled mutually to achieve the desired operation. In this research, the effects of the equivalent ratio variation of the second-stage distributed combustion are chosen to study. The equivalent ratio (\(\phi\)) is the ratio of the stoichiometric air-fuel and the operation air-fuel ratio, which is described in equation (2). The equivalent ratio is varied between 0.6 and 1. The first and second law efficiencies are investigated by experiment.

2. Methodology

The model of the second-stage combustor distributed combustor is applied from Hosseini et al. (2014) [24-26] due to its simplicity to apply with industrial application. The dynamic similitude is used for scaling the size of the combustor. Re_{\text{air-\text{fuel}}} ratio is maintained during the resizing from a small lab-scale size to an industrial size, which provides 6-8 kW loads. During the design phase, the simulation
with computational fluid dynamic (CFD) has been applied to observe combustion and flow phenomena inside the combustion chamber [27]. The shape of the combustion chamber is a single-cylinder cover with ceramic insulation. It contains one LPG fuel hole at the center and eight inlet holes connect to the first-stage biodiesel combustor and one exhaust outlet. Three K-type thermocouples are installed to measure the temperature distribution at 60, 120 and 180 cm along the combustion chamber axis. The thermo couple and gas analyzer called ‘Testo-350’ is installed at the exhaust outlet to measure temperature and concentration of oxygen, NOx and CO. In the first-stage combustor, the B7 diesel fuel is injected by the pressurized nitrogen and the atmospheric airflow is injected by the air blower. Air, biodiesel and LPG flow rates are measured by flow meters. The pressure swirl atomizer is used to inject biodiesel fuel into the first-stage combustion chamber. Biodiesel fuel is ignited with an electrical ignition system. The temperature outlet of the first-stage combustor is also measured with the thermocouple. After the design phase, the two-stage combustion system is constructed and tested.

![Diagram of the two-stage combustion system](image)

**Figure 1.** The two-stage combustion system diagram

### 2.1 A two-stage distributed combustion: setting up an experimental system

Before the experiment of the two-stage distributed combustion, the first-stage biodiesel combustor is tested and the temperature outlet is measured. The measurement data is used for calculating the first and the second law efficiencies. In this study, the operation of the biodiesel combustor can provide excess oxygen for the second-stage combustor in a range of 3% - 7%. Below 3% excess oxygen concentration, the soot occurred can cause damage to the measurement system while beyond 7%, the combustion blows off. Therefore, 5% excess oxygen concentration is most appropriate and selected for all experiment. The B7 fuel has a heating value of 42.53 MJ/kg, which corresponding to the flow rate of 19.88 g/s and the airflow rate is 374.33 g/min. This result in a steady outlet temperature is at 610 OC, which is above the auto-ignition temperature of the first-stage LPG fuel.

As shown in Table 1, when combined with the second-stage distributed combustor, the experiment is carried out in five cases of the equivalent ratio varied from 1 to 0.6 with 0.1 decrements. The LPG
supply is decreased and A/F ratio is increased when reducing the equivalent ratio. It contains 70% propane and 30% butane and has a heating value of 49.73 MJ/kg. The condition of the first-stage combustion is maintained constant for all cases. The auto-ignition of the second-stage is occurred by the preheated air from the first-stage diesel combustor. In a preliminary test, it is found that the distributed combustion can be initiated at around 500 °C of the exhaust gases from the first stage for all ranges of the equivalent ratio. One interesting finding is that there are no effects of equivalent ratio.

The temperature distribution along with the combustion chamber axis is measured and used for calculating the first and the second law efficiencies.

### Table 1. Test conditions of the effects of the variation of the equivalent ratio

| Test Conditions | Case I | Case II | Case III | Case IV | Case V |
|-----------------|--------|---------|----------|---------|--------|
| **The first-stage combustion** | | | | | |
| B7 mass flow rate (g/s) | 19.88 | 19.88 | 19.88 | 19.88 | 19.88 |
| Air flow rate (g/min) | | | | | 374.33 |
| Resulted exhaust temperature (°C) | 610 °C | 610 °C | 610 °C | 610 °C | 610 °C |
| **The second-stage combustion** | | | | | |
| Equivalent ratio | 1 | 0.9 | 0.8 | 0.7 | 0.6 |
| Calculated A/F ratio | 63.77 | 70.85 | 79.71 | 91.10 | 106.83 |
| LPG mass flow rate (g/min) | 5.87 | 5.28 | 4.70 | 4.11 | 3.52 |
| LPG temperature (K) | 300 | 300 | 300 | 300 | 300 |
| LPG Composition | C_3H_8 70%, C_4H_{10} 30% |

### Figure 2. Schematic of the single-stage and two-stage combustion systems

2.2 Single-stage and two-stage distributed combustion efficiency analysis

The first law efficiency is calculated and the results of each case are compared. Figure 2 shows the control volumes of single-stage and two-stage combustion system. The first-law efficiency for both systems can be described as:
\[ \eta_{th} = \frac{m_{out} h_{out} - m_{air \text{ inlet}} h_{air \text{ inlet}} - m_{diesel \text{ inlet}} h_{diesel \text{ inlet}}}{(m \cdot LHV_{\text{Diesel}})} \]  
(3)

\[ \eta_{th} = \frac{m_{out} h_{out} - m_{air \text{ inlet}} h_{air \text{ inlet}} - m_{diesel \text{ inlet}} h_{diesel \text{ inlet}} - m_{LPG \text{ inlet}} h_{LPG \text{ inlet}}}{(m \cdot LHV_{\text{Diesel}} + m \cdot LHV_{\text{LPG}})} \]  
(4)

The second law efficiency concept is the comparison between the actual conditions with the best theoretic possible condition. It can be calculated from the proportion of the exergy destroyed and exergy expense substituted from the unity \([28]\) as described in equation \((5)\).

\[ \eta_{ex} = \frac{e_{\text{Outlet}}}{e_{\text{Expense}}} = 1 - \left( \frac{e_{\text{Destroyed}}}{e_{\text{Expense}}} \right) \]  
(5)

While the exergy destroyed can be calculated from entropy generation in equation \((6)\) for the single-stage system and equation \((7)\) for the two-stage system.

\[ e_{\text{Destroyed}} = \frac{(T_{r} S_{\text{gen}})}{a_{0}} \Delta_{\text{UnKnown}} \]  
(6)

\[ e_{\text{Destroyed}} = \frac{(T_{r1} S_{\text{gen1}})}{a_{0}} \Delta_{\text{UnKnown}} + \frac{(T_{r2} S_{\text{gen2}})}{a_{0}} \Delta_{\text{UnKnown}} \]  
(7)

The entropy generation for both systems can be written in equations \((8)\) and \((9)\).

\[ S_{\text{gen}} = S_{\text{prod}} - S_{\text{react}} + \frac{Q_{\text{first-stage}}}{T_{r}} \]  
(8)

\[ S_{\text{gen}} = S_{\text{gen1}} + S_{\text{gen2}} = (S_{\text{prod1}} - S_{\text{react1}}) + \frac{Q_{\text{first-stage}}}{T_{r1}} + (S_{\text{prod2}} - S_{\text{react2}}) + \frac{Q_{\text{second-stage}}}{T_{r2}} \]  
(9)

The entropies of reactant and product are calculated from the summation of the entropy of each species, i, involved in combustion reaction by using the exact entropy analysis equation as shown in equation \((10)\) where \(s_{i}^{0}\) is an absolute entropy \([29]\).

\[ S = \sum N_{i} \bar{s}_{i} = \sum N_{i} [\bar{s}_{i}^{0}(T, P_{m}) - R_{u} \ln(y_{i} P_{m} a_{0})] \]  
(10)

The heat of combustion is written as \(Q_{\text{out}}\) in equation \((11)\) and computed from the enthalpy difference between reactant and product species. The enthalpy of each species is the summation of the enthalpy of formation \((\tilde{h}_{f}^{0})\) and the sensible enthalpy \((\bar{h} - \tilde{h}_{f}^{0})\).

\[ Q_{\text{out}} = \sum N_{r} (\bar{h}^{0} + \bar{h} - \tilde{h}^{0})_{r} - \sum N_{p} (\bar{h}^{0} + \bar{h} - \tilde{h}^{0})_{p} \]  
(11)

Finally, the cost analysis is also performed to observe the advancing of the distributed combustion combination. Equation \((12)\) and \((13)\) is used to calculate the fuel cost for the single-stage and two-stage combustion systems, respectively, in kilowatts hour per baht.

\[ \dot{m}_{g} = \dot{m}_{g} + \dot{m}_{g} + \dot{m}_{g} \]  
(12)

\[ \dot{m}_{g} = \dot{m}_{g} + \dot{m}_{g} + \dot{m}_{g} \]  
(13)

where \(\dot{m}\) is volume flow rate.
3. Results and Discussion

3.1 The effects of the variation of equivalent ratio on the two-stage combustor

In the experiment, the first-stage combustion of biodiesel fuel with the atmospheric air supplied by the air blower is started at first. Then, the second-stage distributed combustor is ignited through the auto-ignition process when the temperature of the exhaust gas from the first-stage combustor reach above the auto-ignition temperature of the LPG fuel. The results of this study show that the distributed combustion can be attained at around 500°C of the first-stage exhaust. Surprisingly, the temperature of the auto-ignition of the butane and propane are 405°C and 450°C, respectively [15]. This can be noticed by the rapid increase of temperature inside the second-stage and provide the non-illuminated flame. Temperature and emissions are observed at the steady condition.

Figure 3 presents the measured temperatures at the exit and inside the second-stage combustion chamber. The results show that the temperature is highest in the reaction zone, which is near the fuel injection nozzle. The uniform-distribution temperature is shown through a slight decrease along the combustion chamber axis. This demonstrates the prominence of distributed combustion. Considering the influence of the equivalent ratio, the temperature is highest at the stoichiometric condition and decreases when lowering the equivalent ratio to 0.6 (the leanest in this experiment). The effects of the equivalent ratio on temperature distribution are also presented.

![Temperature Distribution of Normal Axis](image)

**Figure 3.** Temperature distribution of normal axis of combustion chamber for the equivalent ratio variation

As can be seen from Figure 4, NOx is incredibly low in all cases in the emissions measurement. The values are nearly zero as it was expected. The thermal NOx reaction is suppressed due to the low combustion temperature. For the investigation of CO, it shows the low level for all cases but not nearly zero like NOx. Both NOx and CO are maximized at the equivalent ratio of 1 and decreased when the equivalent ratio is reduced. Reduction of NOx can be explained by the relatively lower temperature of distributed combustion. In the case of CO, the decrease is from the lower feed of fuel as the supplied air is maintained constant.
The first law efficiency for both single-stage and two-stage combustion systems are demonstrated in Figure 6. The results show that the two-stage combustion with 5% oxygen concentration has slightly higher efficiency than the single-stage combustion with excess oxygen concentration, at stoichiometric. The first law efficiency tends to decreased when decreasing the equivalent ratio. It is obviously described by lower outlet enthalpy, as shown in equations (3) and (4). The lowest efficiency is found in the case of the 0.6 equivalent ratio which is about 5% dropped from the stoichiometric condition. It was suggested that there is an opportunity to improve the efficiency of the single-stage
combustion by combining the distributed combustion as the second-stage. In this study, the distributed combustion is only 4.86 kW, which is just 34.73% of the first-stage conventional combustion.

![Thermal efficiency (%) comparison of the single-stage and two-stage combustion](image)

**Figure 6.** Comparison of the first law efficiency between the single-stage and the distributed combustor with the equivalent ratio variation

### 3.3 The analysis of the second law

Regarding the analysis of the second law, it yields the same trend. The highest efficiency is found at the stoichiometric point. Interestingly, the 0.9 equivalent ratio case also has slightly higher second law efficiency than the single-stage combustion. In other cases, the second law efficiency becomes lower when the equivalent ratio is decreased. This is due to the lower heat release rate. The heat is less released because of heat absorption by the more excess oxygen and produced nitrogen in distributed combustion.

As shown in Figure 8, the worthiness in kWh of output per fuel price is found in the case of the distributed combustion with 5% oxygen concentration and equivalent ratio of 1. The cost is worthy than using the single-stage combustion system. Also, the operation (nearly stoichiometric condition i.e. 0.9 equivalent ratios) yields lower cost. Another important finding was that the feasibility of using the two-stage combustion for industrial application i.e. the gas carburizing process [30]. However, this result of the 0.7 and 0.6 equivalent ratios are not appropriate for energy efficiency improvement.

### 3.4 Formulated equations of A/F and equivalent ratio for the two-stage combustion

The distributed combustor is operated by two important control parameters which are the excess oxygen concentration from the first-stage and equivalent ratio of the distributed combustion stage. In the later stage of the study, the simply used equations of equivalent and A/F ratio are developed for the two-stage combustion system with any CxHy fuel. This is because it is expensive in the calculation to have the test conditions of A/F corresponding to require equivalent ratio. The equations are as follows:
\[ \phi = \frac{X + \left(1 - \frac{\phi_{O_2}^{\text{out}}}{\phi_{O_2}^{\text{in}}}ight)}{X + \left[\left(1 - \frac{\phi_{O_2}^{\text{out}}}{\phi_{O_2}^{\text{in}}}ight) \phi_{O_2}^{\text{in}} \right] \phi_{O_2}^{\text{out}}} \]  

\[ \frac{A}{F} = \frac{X + \left[\left(1 - \frac{\phi_{O_2}^{\text{out}}}{\phi_{O_2}^{\text{in}}}ight) \phi_{O_2}^{\text{in}} \right] \phi_{O_2}^{\text{out}}}{\phi_{O_2}^{\text{in}} \phi_{O_2}^{\text{out}}} \]  

X is the amount of carbon atom in fuel chemical formula
Y is the amount of hydrogen atom in fuel chemical formula
\( \phi_{O_2} \) \(^{\text{out}} \) is the mole fraction of the outlet oxygen concentration
\( \phi_{O_2} \) \(^{\text{in}} \) is the mole fraction of the inlet oxygen concentration
\( \phi \) is the equivalent ratio

In equation (14), the \( \phi_{O_2}^{\text{out}} \) is generally not equal to \( \phi_{O_2}^{\text{in}} \) when there is a reaction in the second-stage. In equation (15), \( \phi_{O_2}^{\text{out}} \) cannot be unity. \( \frac{A}{F} \) and equivalent ratio are related and controlled by X, Y, \( \phi_{O_2}^{\text{in}} \), and \( \phi_{O_2}^{\text{out}} \). When the fuel is selected, the controlling parameters are reduced to \( \phi_{O_2}^{\text{in}} \) and \( \phi_{O_2}^{\text{out}} \). These equations are limited to stoichiometric and lean combustion. \( \phi_{O_2}^{\text{in}} \) controls inlet oxygen concentration for the distributed combustion, while \( \phi_{O_2}^{\text{out}} \) controls the desired oxygen concentration at the outlet.

**Figure 7.** Comparison of the efficiency of exergy between the single-stage and the distributed combustor with the equivalent ratio variation.
Figure 8. Cost comparison between the single-stage combustor and the distributed combustor with the equivalent ratio variation.

4. Conclusion
This study set out to combine the LPG distributed combustor to the B7 biodiesel combustor to improve the first law and the second law efficiencies. This study has identified the advantage of using distributed combustion as second-stage. The research has also shown that emissions, NOx and CO are low values especially NOx concentration is nearly zero. The results of this research support the idea that the first law and the second law efficiencies are improved, and it can lead to lower energy cost at the same firing rate. These data suggest that the effects of the equivalent ratio are not recommended for energy cost saving because when the heat release is decreased, it can cause lower efficiency. The present study lays the groundwork for future research into applying the distributed combustion to the conventional single-stage combustion system.

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