Overall Assessment of an Innovative Coaxial Air-Staged Burner for Cofiring of Oil and Gas

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Abstract

As a result of the global energy crisis, all countries around the world are resorting to burning using the solid, liquid, and gaseous fuels, whichever is available. However, because of the difficulty of solid fuel combustion, the global trend goes to liquid fuel combustion as it is easier than solid fuel combustion, then goes to the gaseous fuel according to its ease of combustion than the other two types. But, because of the gaseous fuel scarcity and high price, the tendency to co-fire liquid and gaseous fuels is gaining more attention to get Higher performance and solve problems related to liquid fuels. Double swirlers are used to improve the atomization of liquid fuel and improve the mixing of air with fuel to achieve the highest performance of combustion, thus obtaining clean combustion with low emissions. The goal of the current research is to use dual fuels, to increase the efficiency of the combustion and to solve the problem of fuel scarcity and the difficulty of liquid fuel combustion by using gaseous fuels, in addition, to figure out the effect of the recirculation zone and excess air on the combustion characteristics and NOx. The results showed that cofiring dual fuels increases the efficiency of burning liquid fuel as the gaseous fuel form a hot shield around the liquid flame that reduces evaporation time and fires falling droplets from liquid flame and increases soot oxidization, in addition cofiring raises the temperature inside the furnace which enhances convective heat transfer and reduces the visual and thermal flame lengths which decrease the heat flux from the flame. However, the overall cumulated heat transfer is better than the liquid case. Finally, cofiring gives the ability to efficiently burn biofuels in the future to generate thermal power for the industry especially for cement factories that need high power.

Keywords: Co-firing of oil and gas, multiphase combustion, LPG

Nomenclature

Symbol	**Description**	**unit**
CV	Calorific value	[KJ/kg]
D	Diameter of the furnace	[m]
L	Length of the cooling segment	[m]
LGR	Liquid to Gas Ratio of heat share = (Q_l/Q_t) / (Q_g/Q_t)	[-]
$\dot{m}$	Mass flow rate	[kg/s]
Q_t	Total input heat = $\sum \dot{m}_f \times CV_f$	[kW]
Q''	Heat Flux to the furnace wall	$[kW/m^2]$
Q_C	Cumulated heat transfer	[%]
r	Radial distance measured from combustor axis	[m]

R	Inner radius of the combustor	[m]
x	Axial distance along the combustor starting from burner exit	[m]
Subscript		
c	cumulated	
f	fuel	
g	Gas	
in	inlet	
l	liquid	
sn	Segment Number	
t	total	
out	outlet	
w	water	

1 Introduction

Many researchers studied multiphase combustion in internal combustion engines by fumigating air with gaseous fuel such as hydrogen, natural gas, biogas, or alcohol in the intake manifold then injecting a diesel or biodiesel into the combustion chamber. However, multiphase fuel burning in continuous combustion applications is a rarely discussed topic in scientific research, especially non-premixed flames due to the complexity of mapping and controlling combustion systems with two fuel lines. A few scientific research were published in this area, most of them burn syngas premixed with air with diesel or biodiesel being sprayed into the combustion chamber. This opens the way for the simultaneous utilization of massive amounts of bio-oil fuels and syngas to be produced from organic solid waste for use as a renewable power generation source, Agwu, O. and Valera-Medina, A., (2020). This methodology would additionally not only save the reserves of fossil fuel for future generations or use as a raw material in petrochemical industries, but it will also reduce the environmental problems of global warming (zero-carbon solution), Paris Agreement, (2015).

It is worth mentioning that cofiring of oil and gas would promote the vaporization of oil droplets, leading to efficient combustion and higher flame radiation properties. The latter would help in reducing the size of thermal heat transfer surfaces, making the equipment more compact. In other words, oil flames exhibit high heat transfer by radiation while the thermal energy transferred from gaseous flames relies on the convective transfer mode, and so cofiring will be an advantage in transferring the heat generated by both modes. On the contrary, cofiring of biofuels and syngas suffers from much lower heating values as well as poor atomization of biofuels (higher viscosity) and lower percentages of combustible elements in syngas. For these reasons, it appears feasible to adopt the concept of blending with conventional oil and gas to successfully achieve efficient, stable burning together with lower harmful NOx and CO emissions.

Thus, the main goal for burning multiphase fuels is to enhance the combustion efficiency of blended low-grade biofuels and/or syngas. This recently encourages many researchers to develop and test a variety of new burners for cofiring blended biofuels and syngas as explained below. For example, Agwu and Valera-Medina (2020) developed a burner for cofiring of diesel /syngas whereby the syngas is premixed with air and the mixture flows through an annulus swirler and the diesel is centrally injected by a pressure jet atomizer. Tests were carried at a fixed thermal power input of 15 kW and an overall equivalence ratio of 0.7. The test cases cover pure diesel (reference case) and different ratios (energy basis) of syngas/diesel of 10/90, 20/80, and 30/70. The experimental measurements were focused on mapping the flame lean and rich stability limits, the exhaust emissions of CO and NOx. The former was assessed via increasing/decreasing the air flow rate to reach the lean/rich limits of extinction and the stability at a selected operating condition is further assessed by monitoring the C_2^* and CH^* radicals using a CCD imager intense camera. Their results clearly showed that increasing the percent of syngas (at the expense of reducing the oil flow rate) caused (i) an increase in the exhaust emissions of CO coupled with a decrease of NOx and (ii) narrowing the equivalence range for stable operation as due to the poor spray atomization associated with the reduction of the atomization pressure. It is for these reasons that the authors recommended the use of an air blast atomizer to replace the pressure jet atomizer. The assessment of stability via monitoring the C_2^* and CH^* radicals (at a particular operating condition) is achieved by taking the average of their variations over time. The flames giving minimum variations around the mean (i.e., less fluctuations in the reaction rate) are considered more stable. Accordingly, the authors reported the best stability at a syngas/oil ratio of 70/30 among their test cases.

Furthermore, co-firing of palm methyl esters (PME) or diesel fuel and natural gas at a constant thermal input power of 9.3 kW was studied by Chong, C.T., et al., (2020). The authors employed a coaxial burner having a central air-assisted atomizer and an outer vane-type swirler through which a premixed NG/air mixture flows. They tested two base conditions (PME, and diesel fuels), and three cofiring conditions 10/90 and 20/80 and 30/70 natural gas/PME. These five test conditions are repeated at three swirler's angles of 30, 45, and 60 degrees. The experimental measurements were given to mapping for NO and CO emissions from the flame on the range of $\phi = 0.65 \sim 0.9$, flame photographs for different flames at $\phi = 0.65$, and spectroscopic analysis. Their findings are (i) the cofiring flame visual structure of PME and NG is like the PME flame structure. In addition, the diesel flame changes dramatically with increasing the swirl angle from 45 to 60 degrees, the yellow region disappeared with 60° due to the rapid oxidation for soot in the highly swirling air (ii) Dual fuel flame emits lower NO but higher CO at an equivalence ratio of 0.9 with comparing with pure diesel and PME flames. (iii) High CO in the case of dual fuel flame due to low mixing of fuel with air at an equivalence ratio of 0.9, therefore, decreasing the equivalence ratio to 0.65 by introducing more air increases the mixing quality so the CO levels decrease. But high NO levels are due to higher air amounts.

The flame structure and local extinction characteristics of ethanol/methane co-combustion are investigated by Sidey, J. and Mastorakos, E., (2017), the ethanol is sprayed at the center of the burner using a pressure atomizer and the methane is premixed with air. The air/mixture flows through a swirler (downstream the burner rim) and by a conical bluff body at the burner exit. All the tested conditions are at the same amount of air flow rate which achieves velocity at the exit of 18.5 m/s and at the same amount of liquid fuel that achieves $\phi = 0.24$, introducing more gaseous fuel increases the overall equivalence ratio and speed at the burner exit, the tested conditions are (Case1: ethanol only), (Cases2,3,4,5: cofiring), and (Cases 6,7: CH_4 only), all conditions are at $\phi < 1$. The OH^* radicals' concentration is measured using a Mie camera. The results showed that, as the amount of gaseous fuel increases, the air/fuel range of stable flame decreases, and the probability of lifting the flame increases. Sidey, J. and Mastorakos, E., (2018), continued their study on the same burner configuration, experimental program, and measuring technique. However, they used C_7H_{16} instead of methanol. they demonstrated how multiphase combustion alters spray flame behavior and stabilization characteristics as the results showed that adding CH_4 may cause the liquid fuel's attachment point near the bluff-body edge to become unstable, it becomes either a point of local extinction or a region of the system that is too rich to host a flame. Despite this, at the same global equivalence ratio, global extinction characteristics suggest that dual-fuel spray flames are more resistant to blow-off than single-fuel spray flames. Evans, M.J. et al., (2019), used the same burner but with different types of fuel, n-heptane is sprayed at the center and H_2 or CH_4 premixed with air. The temperature is measured at different loading conditions while changing the amount of air through the swirler using a thermal camera. The results showed that, with the addition of H_2, the high-temperature region above the spray apex is reduced, and the zone above the burner base is cooler, in contrast, the addition of NG has little effect on the high-temperature region above the spray apex. When $\phi_{premixed}$ is set to lower flammability limits (LFL), the flame's high-temperature region near the burner base is shifted to the outer flame branch. Exceeding LFL, on the other hand, results in an additional temperature peak, which corresponds to a premixed flame, and the region of highest conditional temperature returns to the burner centerline above the spray apex, but it also increases the probability of lift-off.

Another study is conducted on cofiring fossil fuels (LPG and Diesel) with a dual swirler burner is conducted by Alkadi, et al. (1997), the burner is manufactured in a conical shape to supply air through the tight edge and the swirlers are installed at burner rim (wider edge), the fuel lines are a central line with a pressure atomizer for liquid fuel and four circumferential LPG lines with nozzles at their tips fixed on the outer swirler. The experimental program included five conditions of different fuels (a liquid, a gaseous, and three co-firing with different gas ratios), these conditions are repeated three times for different heat inputs. Their results showed that gaseous fuel flame forms a shield around the liquid fuel flame that enhances droplets vaporization leading to an increase in the combustion efficiency and reduction in the accumulated soot on the combustor's walls and exit. increasing the gas ratio results in the following, (i) Increased convective heat transfer rates, (ii) more uniform heat flux in the downstream direction of the combustor, (iii) raising the thermal efficiency by increasing the cumulative heat transfer to the combustor walls, (iv) the peak value of the heat flux is less sensitive to variations in heat input. Shahien, M.A., (2004), used the same burner to experimentally investigate the influence of the double swirler with different angles on burning characteristics of liquid fuel (light diesel) and the flow field. The study found that (i) the ideal angles for inner swirlers are $30°$ and for outer swirlers are $60°$. (ii) the design of ($30°$in, $60°$out) achieves the best heat transfer to the furnace wall, the best fuel-air pair mixing, the largest central recirculation zone, and the highest axial flame temperature.

The current work aims to assess the new burner design's performance while burning multiphase fuel (oil, gas). The burner's performance is investigated by measuring the temperature distribution, heat flux, and cumulated heat transferred to cooling water for 3 different cases (gas, oil, co-firing). The goal of this burner design is to burn multiphase fuels especially biofuels, However, in this work, the LPG and light diesel fuels (fossil fuels) are used to assess the new burner design and provide a database by which the future work on the biofuels could be compared with.

2 Research Methodology

2.1 Experimental Setup

Figure 1 shows the layout of the experimental setup. It is comprised of a horizontal cylindrical water-cooled combustor having an inner diameter of 0.40 m and a length of 2 m. The cooling jacket is segmented into 9 sections to allow mapping of the longitudinal variations of the heat flux distribution. thirteen measuring taps to allow, flame visual observation, monitoring of the radiant heat flux, and inflame probe measurements. The burner is coaxially mounted at the combustor entry. It has an inner diameter of 0.15 m, giving a sudden expansion ratio of 2.66.

the burner is comprised of a central pressure jet atomizer having a capacity of 2 g/h, delivering a hollow cone spray ($\Theta = 45°$) of light diesel fuel. This atomizer is centrally placed inside an industrial circular stabilizer disc having an outer diameter of 7.6 cm. The inner pipe has a diameter of 8.3 cm, (giving a blockage ratio of 0.72), through which an inner air flow is supplied from a centrifugal fan. Gaseous fuel (LPG) being supplied from pressurized bottles passes to a pressure header, pressure regulator, and metered by a calibrated orifice plate and fed to a coaxially mounted reservoir where it is distributed to twelve circumferentially equally spaced gas nozzles (each having a diameter of 2 mm). An outer circular passage supplies swirling air (swirl angle of 45^o). The inner and outer air flow rate is controlled by gate valves and metered by calibrated orifice plates. The fuel supply flow rate is measured by a calibrated bourdon tube pressure gauge, whereby the pressure reading indicates the fuel flow rate, Figure 2.

Table (1) shows the proximate analysis and calorific value of the diesel fuel as specified by the Egyptian company of petroleum. Table (2) gives the chemical composition and calorific value of the LPG fuel as specified by the Egyptian company of industrial gases.

Table 1 Proximate analysis and calorific value of the diesel fuel*

	Carbon	Hydrogen	Sulfur	Oxygen
Constituents (% wt)	86.5	13.2	0.1	0.2
Calorific Value (MJ/kg)	44			

***** Source: The Egyptian Company of Petroleum

Table 2 Chemical composition and calorific value of the LPG fuel*

	C_3H_8	C_4H_{10}
Constituents (% vol)	30	70
Calorific Value (MJ/kg)	48.5	

* Source: The Egyptian Company of industrial gases

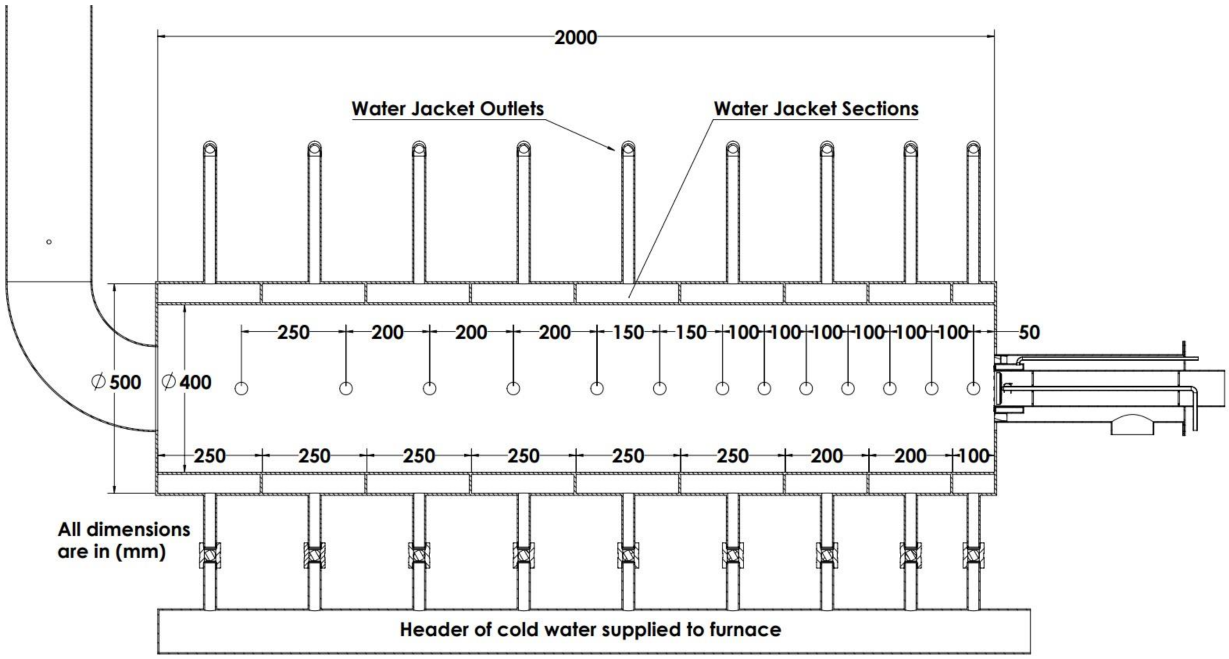


Figure 1 Layout of the Experimental Setup

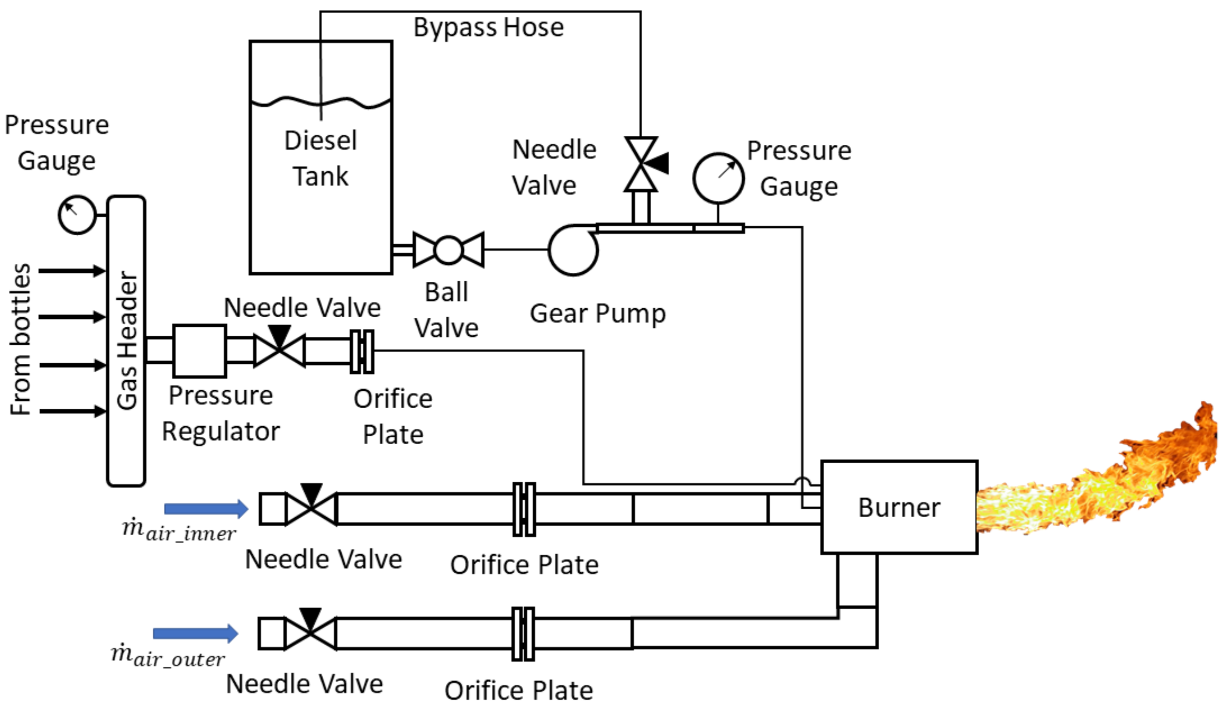
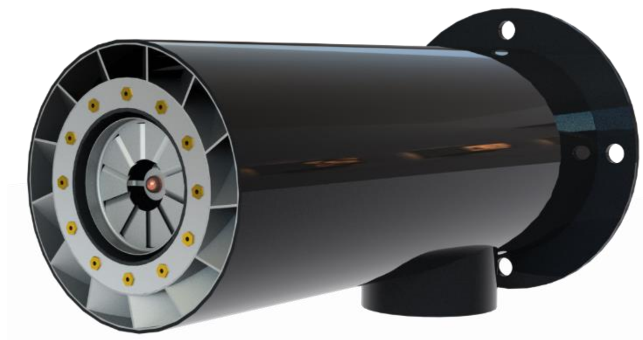
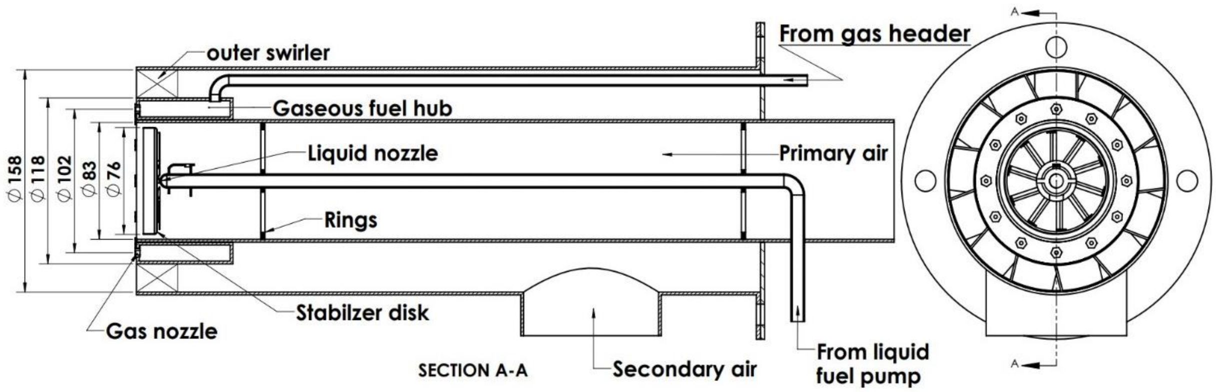


Figure 2 The gaseous and oil fuel feeding lines



Panel (a) Photograph Image of the Burner



Panel (b) Design details of Burner

Figure 3 Photograph and Design details of the co-firing burner

2.2 Measuring techniques

The local mean temperature inside the flame and the hot gases were obtained with a fine wire thermocouple (type S) of 125 μm, connected to an "OMEGA ENGINEERING" digital reader to transmit temperature readings to the software on a computer via a USB cable. The heat flux to the combustor walls and cumulated heat transferred were estimated by the following equations after measuring the rise in cooling water enthalpy at each segment of the cooling jacket shown in Figure 1. The water flow rate and its temperature are measured at each segment by using a calibrated turbine flow meter and thermocouple (type K) that are connected to an Arduino board to present the values on an LCD screen.

$$Q''_{sn} = \frac{\left[\dot{m}_w \times C_{p_w} \times \left(T_{w_{out}} - T_{w_{in}}\right)\right]_{sn}}{\pi \times D \times L_{sn}}$$

$$Q_c = \sum_{sn=1}^{n-1} Q_{c_{sn}} + \frac{\left[\dot{m}_w \times C_{p_w} \times \left(T_{w_{out}} - T_{w_{in}}\right)\right]_{sn}}{Q_t} \times 100$$

2.3 Experimental Program

The experimental procedure aims to investigate and compare both the flame structure and the characteristics of the combustor under three types of firing at a fixed thermal power of 70 kW as shown in Table 3. These types are

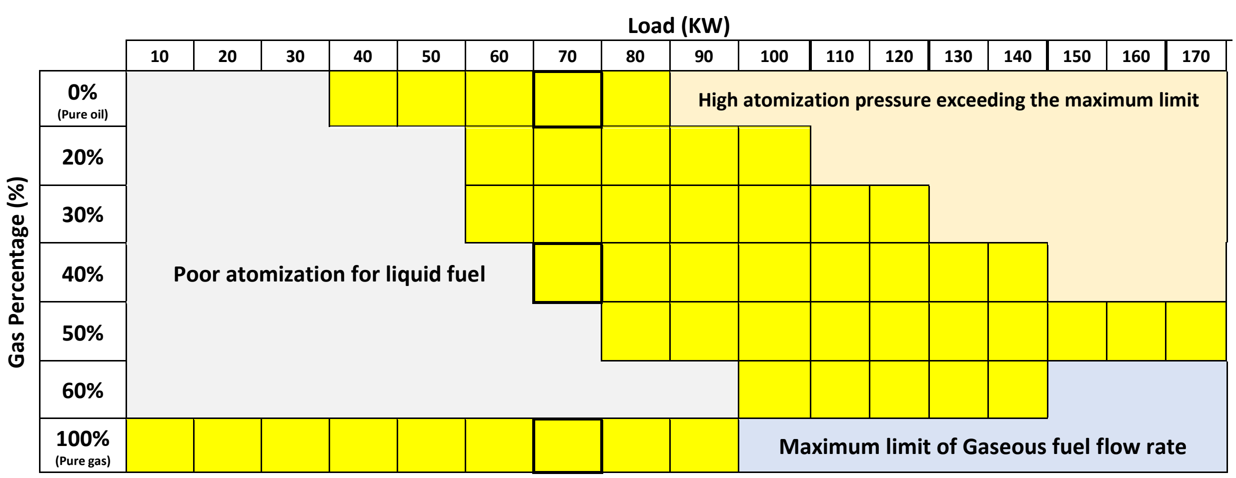
1- Liquid fuel firing
2- Gaseous fuel firing
3- Dual simultaneous fuel firing (Co-firing)

Table 3 Selected Test conditions for heat flux and inflame temperature measurements

Run No.	Fuel Type	LGR [-]	Heat Input [kW]	Equivalence ratio [ϕ]	$\dot{m}_{a,i}$ [kg/h]	$\dot{m}_{a,o}$ [kg/h]	$\dot{m}_{gas}$ [kg/h]	Inner/outer air ratio
1	Diesel	100/0	70	0.5*	50	116	0	30/70
2	LPG	0/100			48	112	5.2	
3	Diesel/ LPG	60/40			49	115	2.1	

* To keep air momentum through the outer swirler in a moderate value that forms a notable swirling effect as the outer swirler's cross-sectional area is large

Table 4 Test cases for mapping of the flames at different loading conditions and Percentage gas ratio at each load



This load mapping suggests using an air-assisted atomizer instead of the pressure jet atomizer to unlink the thermal load from the quality of atomization, which expands the available region of operation and ensures better atomization for cofiring cases.

3 Results and discussion

3.1 Liquid fuel flame structure

The in-flame temperature measurements revealed that the maximum flame temperature in the case of burning oil (Run 1) was 913°C as shown in Figure 4 (right) and Figure 8, which is relatively low, this low temperature because of a sequence of three factors, first, the atomizer spreads fuel droplets in a cone angle of 45°, second, the swirling air converts the spray cone into a wide cloud of droplets, third, the amount of excess air, these three factors widen the flame diameter which reduces the heat release per unit volume and decreases the flame temperatures as well. The maximum temperature is at x = 28 cm (x/D = 0.7) where the maximum diameter of the recirculation zone exists, which indicates good mixing of air and fuel at this distance and shows the thermal flame length as well. Meanwhile, the visual flame length is around 90 cm measured from a scale on Figure 4 (left). This long visual flame length strongly affects the heat flux from the flame especially the radiation, the oil flame shows the highest heat fluxes of 47.6 kW/m^2 at x = 40 cm (x/D = 1) and 43.7 kW/m^2 at x = 65 cm (x/D = 1.62), due to the long and bright visual flame. Despite the high heat flux from the flame, the exhaust gases have relatively lower temperatures than the other two firing cases (Run 2&3) which reduces the amount of heat flux downstream the flame inside the furnace and reduces the cumulated heat transferred from exhaust gases through the combustor to be 41.2% at the last section of the water jacket.

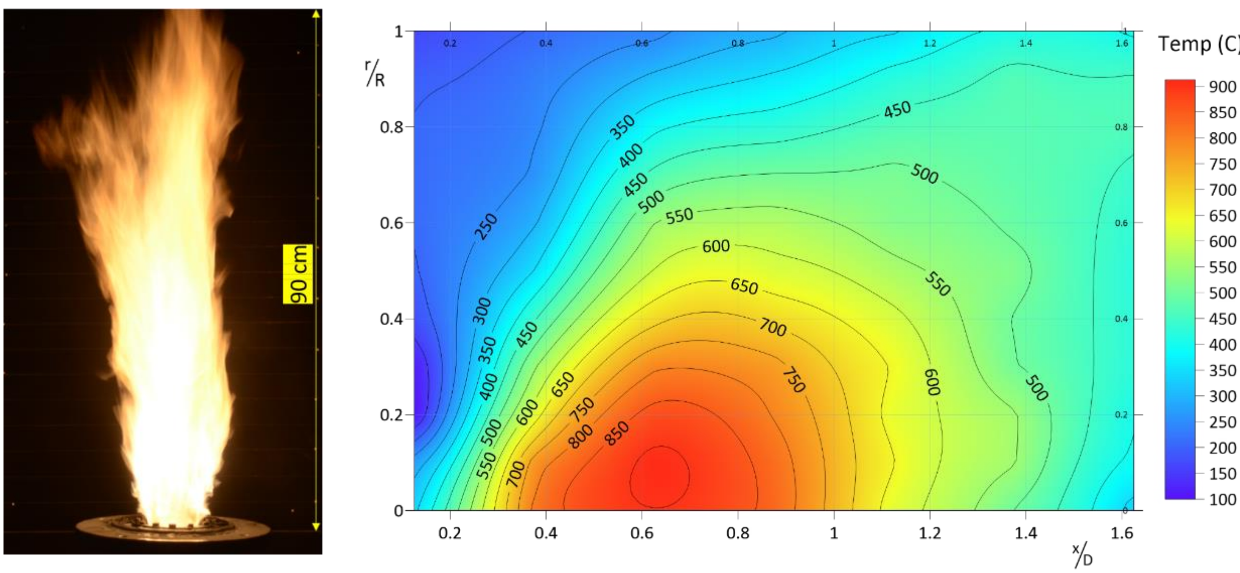


Figure 4 photograph of vertical oil firing (Left), Temperature contours inside the furnace (Right), Run 1

3.2 Gaseous fuel flame structure

The maximum temperature measured in the case of gaseous fuel is 1097 °C at x=15 cm (x/D = 0.38) and maintained approximately at the same value of 1095° C at x = 25 cm (x/D = 0.63), as shown in Figure 5 (right) and Figure 8, which indicates a higher reaction rate for gaseous fuel that enables it to start burning just after entering the combustor and rapid mixing with swirling air. Notably, the temperature contours are longer in this case (Run 2) than the other ones (Run 1&3) because of the high velocity of straight gaseous fuel jets at the gas orifice outlets (v = 21.1 m/s) while the air velocities from both paths are around (v = 7 m/s) that enables the LPG to penetrate deeper into the combustor. however, despite the long penetration of LPG, the visual flame length is around 75 cm which is shorter than oil flame (Run 1) because of the ease of burning gaseous fuel, unlike liquid fuel which takes some time to mix and evaporate with air to be burnt, while the thermal flame length is around x=25 cm corresponding to the maximum temperature. The fast reaction rate of gaseous fuel makes its heat flux higher than the other two firing cases (Run 1&3) at the first two sections of the water jacket up to x = 21 cm (x/D = 0.53), in addition, the heat flux from gaseous fuel downstream the flame is the highest along rest of the combustor due to the high convective properties of the hot exhaust gases. Finally, the cumulated heat transferred in the last section represents the highest value (51%) between the three firing cases because of the high temperature in the exhaust gases.

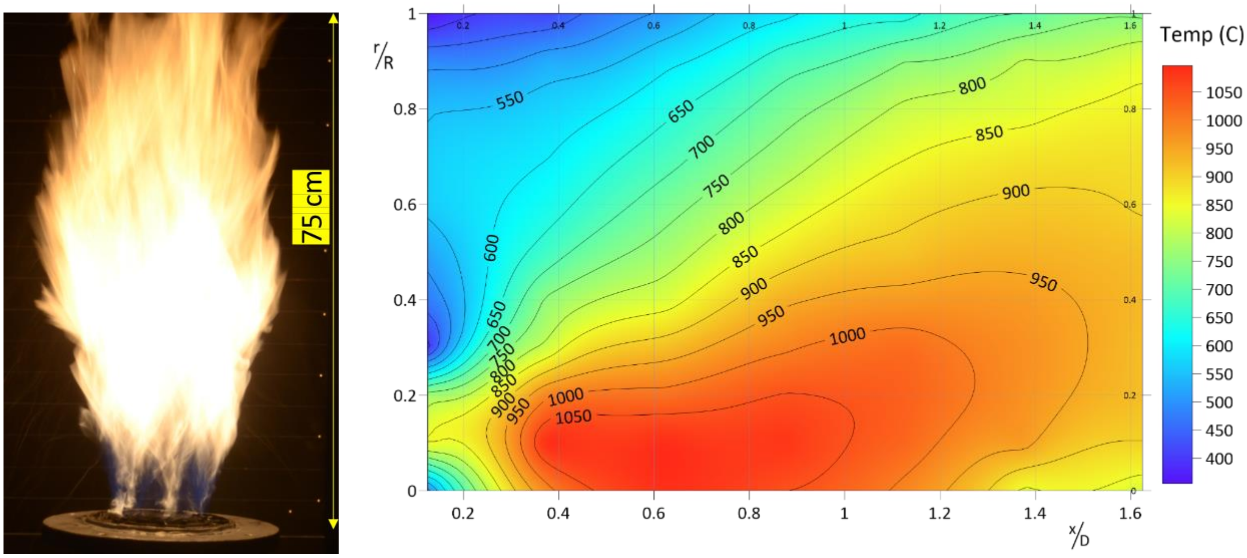


Figure 5 photograph of vertical gas firing (Left), Temperature contours inside the furnace (Right), Run 2

3.3 Dual fuel flame structure

The co-firing case (Run 3) temperatures are in between the other two firing cases (Run 1&2) This is because the co-firing is a mix of the characteristics of each fuel flame as shown in Figure 8. The visual flame length is around 45 cm which is notably shorter than the other cases despite the same thermal power for them all, this is because of the splitting of thermal power to be generated from two fuel streams, each stream has a lower momentum than its pure case (Run 1&2). The thermal flame length is corresponding to the maximum temperature of 1032° C at x = 15 cm (x/D = 0.38) which is shorter as well. Accordingly, the heat flux from the flame of the co-firing case is lower than the other cases (Run 1&2), however, due to the temperature of exhaust gases in the co-firing case being in between the other two cases (Run 1&2), the heat flux downstream the flame is in between them as well, leading to in between cumulated heat transferred to the cooling water (45.7 %) as well.

To achieve lower thermal power from liquid fuel, it’s required to reduce the liquid fuel flow rate which is related to the atomization pressure. Despite this, in the co-firing case, the lower atomization pressure did not cause any falling droplets from the flame because the gaseous fuel that surrounds the liquid acts as a hot shield around the liquid flame which enhances the evaporation process of liquid droplets and fires any falling liquid droplets from the liquid flame. This proves that the co-firing of multiphase fuel with the proposed burner configuration improves the burning of liquid fuel, which makes this burner design suitable for burning biofuels that suffer from a high percentage of impurities.

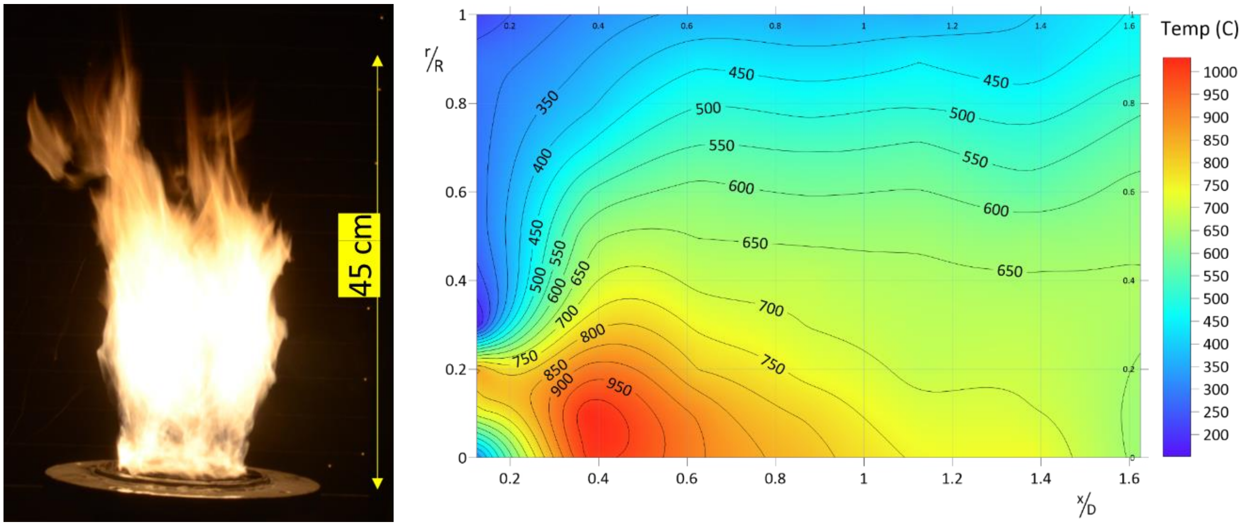


Figure 6 photograph of vertical cofiring of LPG and Light diesel fuels (Left), Temperature contours inside the furnace (Right), Run 3

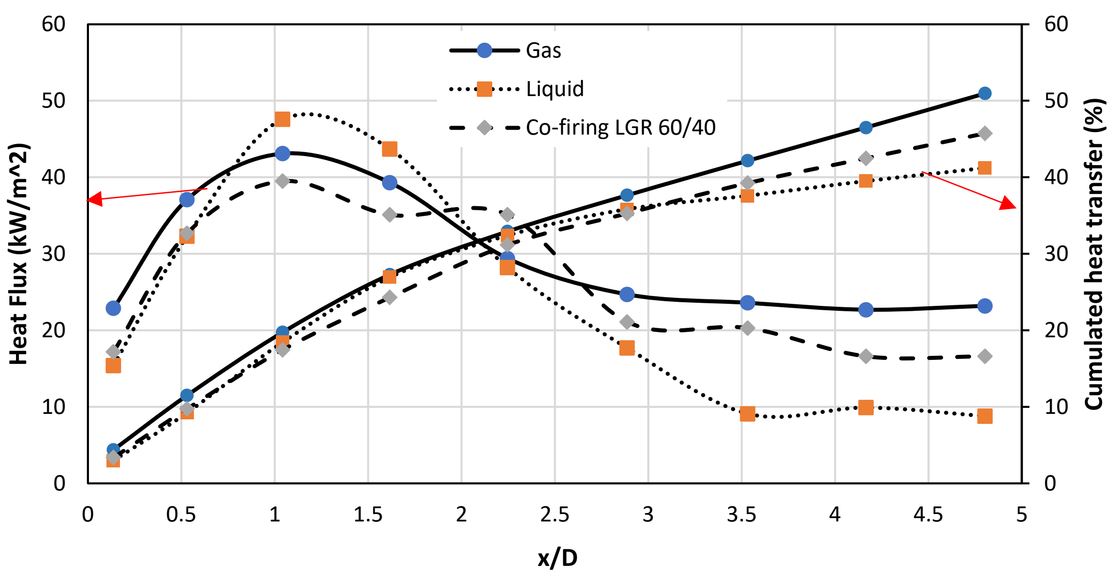


Figure 7 Comparison of heat flux on the furnace wall and cumulated heat transferred to cooling water jacket

The heat liberation near the upstream end has high values due to the existence of the flame which transfers heat by radiation and convection as shown in Figure 7, meanwhile, at the downstream end, the heat flux to the combustor walls has lower values because the heat transfer depends mainly on the convection. Therefore, the higher the exhaust gases temperatures, the higher the heat transfer to the walls, which gives the advantage for burning gaseous fuel as shown in Figure 9 and Figure 8 to have the highest cumulated heat transferred to cooling water.

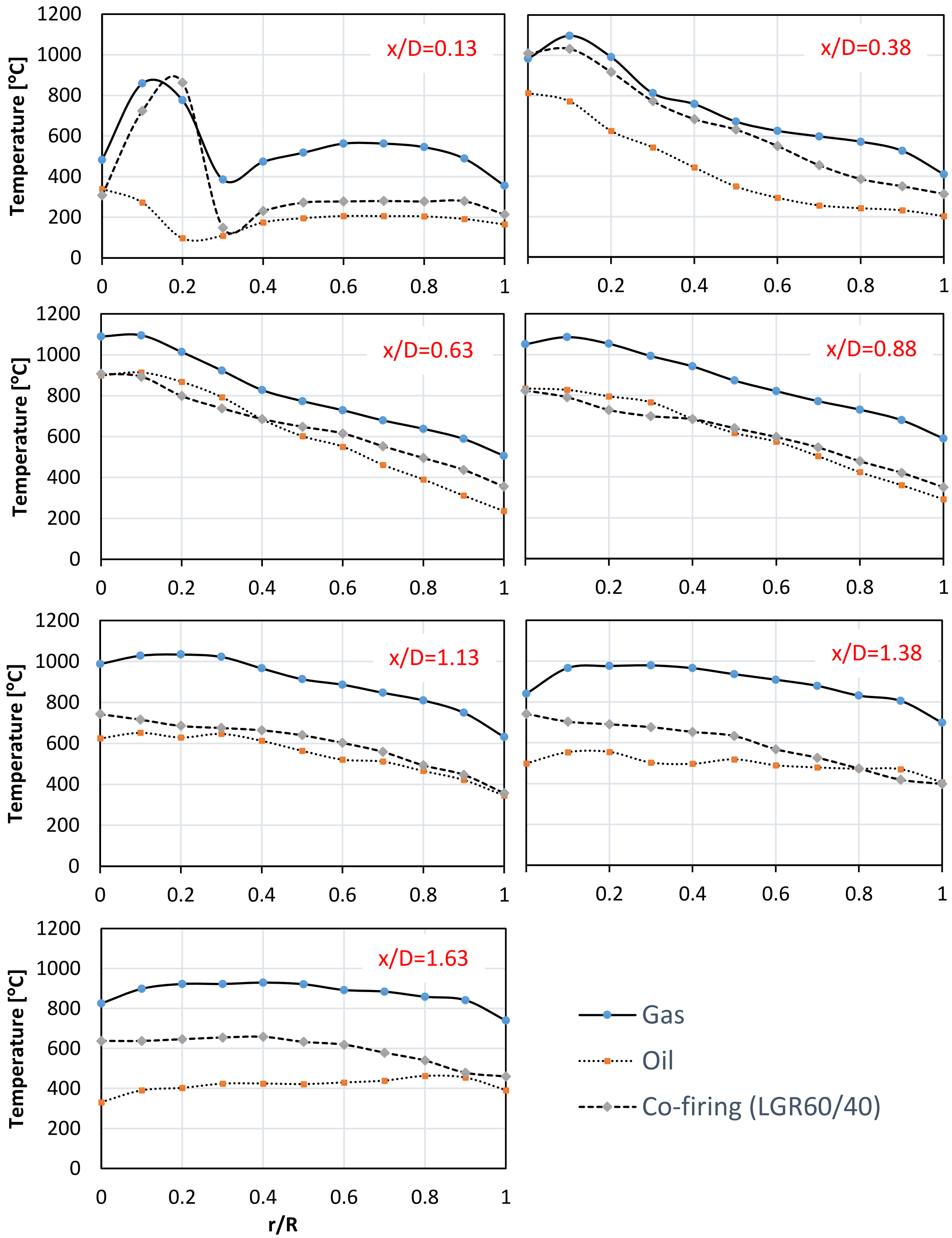


Figure 8 Radial temperature distributions at each measuring hole (x/D)

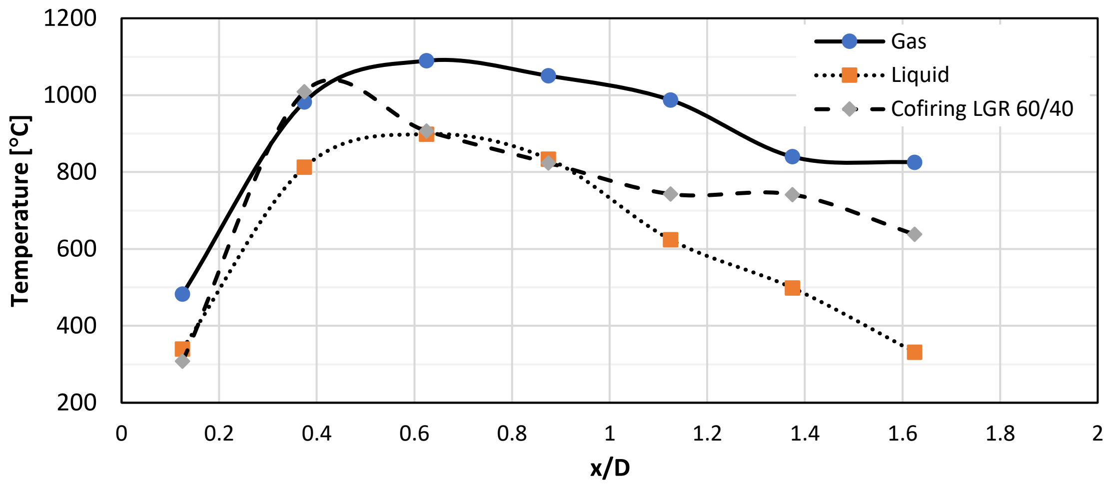


Figure 9 Axial temperature distribution along the centerline of the furnace

4 Conclusion

This work is done to investigate and compare the flame characteristics of burning single-phase fuel (light diesel and LPG) and burning multiphase fuels together (co-firing of liquid and gas fuels) using the proposed burner configuration in this paper. All the tested conditions are conducted at the same thermal power of 70 kW and equivalence ratio of 0.5.

The burning of liquid fuel showed the longest visual and thermal flame length (90 cm and 28 cm respectively) which gives the highest heat flux on the furnace walls from the flame (47.6 kW/m^2) and gives the lowest exhaust gases temperature leading to the lowest heat flux downstream the flame and lowest cumulated heat transfer to the cooling water (41.2 %).

The gaseous fuel has a faster reaction rate than the other types, therefore, the temperatures fastly attain uniform radial distributions which augments heat flux from the exhaust gases downstream the flame, resulting in the highest cumulated heat transferred (51 %). In addition, the visual and thermal flame lengths are shorter than the liquid fuel (75 cm and 25 cm respectively).

By increasing the gas ratio in co-firing, the exhaust gases fastly reach uniform temperature distributions making the heat flux more uniform on the combustor walls, which augment both the heat transfer by convection and the cumulated heat transferred along the combustor, leading to better thermal efficiency. In addition, increasing the gas ratio reduces the evaporation time of liquid fuel and is expected to reduce the soot concentration as well. Furthermore, introducing gas fuel to the combustion alters the heat transfer method to be dependent more on convection which could be considered as an easy way to control heat transfer.

In the co-firing case of liquid and gaseous fuels the temperature distributions are in between the other two cases, therefore, the heat flux from the exhaust gases and the cumulated heat transferred (45.7 kW/m^2) is in between the other two cases as well, furthermore, the visual and thermal flame lengths (45 cm and 15 cm respectively) are the shortest among the other cases because of the low momentum of each fuel stream.

It's recommended for future work to use an air-assisted atomizer instead of a pressure jet atomizer to ensure better atomization quality at low thermal loads especially in the co-firing cases. This agrees with Agwu, O. and Valera-Medina, A., (2020).

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