Swirler geometry effects \((d_h/d_o)\) ratio on synthetic gas flames: Part 1: Combustion and emission characteristics

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Swirling flows increase combustion performance via favouring flame stability, pollutant emissions, and combustion intensity. The strength of a swirling flow is characterized by a parameter known as swirl number, which is highly related to the \(d_h/d_o\) ratio. In this study, effects of the swirler \(d_h/d_o\) ratio on combustion and emission characteristics of the synthetic gas flames of premixed 20\%CNG/30\%H\(_2\)/30\%CO/20\%CO\(_2\) mixture were experimentally investigated in a laboratory-scale swirl stabilized combustor. For this purpose, twelve different swirl generators were designed and manufactured. \(d_h/d_o\) ratios of these swirlers were set as 0.30 and 0.50, and the geometric swirl number was varied between the values of 0.4 and 1.4 (at 0.2 intervals). All experiments were conducted at a fuel-lean equivalence ratio \((\phi = 0.6)\), room temperature, and local atmospheric conditions of the city of Kayseri, Turkey. A data logger was utilized to plot axial and radial temperatures and NO\(_x\), CO, and CO\(_2\) profiles, which were exploited to assess combustion and emission performance. Results showed that the \(d_h/d_o\) ratio had a non-monotonic effect on the behaviour of combustion and emission of the tested synthetic gas mixture. Depending on the swirl number, increments and decrements were observed in temperature and emission values.

**Keywords:** synthetic gas, \(d_h/d_o\) ratio, swirler, combustion, emission

INTRODUCTION

In a flow, swirl can be generated in two ways, actively (1) or passively (2). The active technique requires an external power source to mechanically introduce a tangential component to the flow; in the passive technique, the flow structure is altered by the implementation of a geometric design to the burner nozzle or surface. Compared to the former one, the latter technique favours by means of energy demand and structural complexity [1]. The swirl number (the ratio of axial flux of tangential momentum to axial flux of axial momentum) is the parameter
that characterizes the strength of the swirl and can be presented as

\[
\text{Swirl number } (S) = \frac{G_w}{RG_{\text{sw}}} = \frac{\int_0^R w u r^2 \, dr}{\int_0^R u^2 r \, dr},
\]

(1)

where \( R \) – outer radius of the annulus; \( w \) – tangential velocity component; \( u \) – axial velocity component; \( r \) – radial position \([2]\). The strength of swirl also can be approximated by using the formula below:

\[
S = \frac{2}{3} \left[ \frac{1 - \left( \frac{d_h}{d_o} \right)^2}{1 - \left( \frac{d_o}{d_o} \right)^2} \right] \tan(\theta)
\]

(2)

where \( \theta \), \( d_h \), and \( d_o \) are swirler vane angle, hub, and outer diameters of the swirl generator, respectively \([3]\). Many researchers used this formulation and conducted both experimental and numerical studies on swirling flows.

Ilbas et al. conducted numerical studies on hydrogen containing fuel blends to investigate effects of the swirl number on combustion and emission behaviour of such mixtures. They varied the swirl number between the values of 0.2–0.8 (swirler vane angle was also varied accordingly) and kept \( d_h/d_o \) ratio constant. Results showed that an increasing swirl number in tested range causes emissions of \( \text{NO}_x \) and temperature values in radial direction to increase \([4]\). The recirculation zone formed in a swirling flow (after a critical condition-swirl number) enhances residence time, improves the fuel/air mixing condition, and hence reduces pollutant emissions and increases flame stability. Therefore, swirl generator design is of great importance by means of flame stability, combustion efficiency, pollutant emissions, and pressure losses. Considering this, Khandelwal et al. investigated effects of design parameters of a swirl generator (such as swirler vane angle and the number of vanes) and the mass flow rate through swirler on the non-reactive flow field. For this purpose, they designed five different swirl generators with different swirl numbers (0.625–1.34) and a constant \( d_h/d_o \) ratio (0.5). It was concluded that increasing vane angle led to reverse flow velocity and turbulence in axial direction, and pressure drop to increase. They commented this situation as a better mixing condition at high vane angles with the payoff of the pressure drop. Moreover, they reported that turbulence energy rose and flow velocity reduced with vane number increments \([5]\). Ishaka et al. studied effects of inlet velocity on the structure of the swirling flow in a combustor by keeping all design parameters of the swirl generator constant. They found that inlet velocity slightly altered the flow field and as inlet velocity increased, reverse flow velocity rose but the area of core flow did not significantly change \([6]\). Yilmaz et al. built an experimental test rig to analyse the effects of the swirl number on temperature and pollutant distributions throughout the combustor, and stability limits of (blowout and flashback equivalence ratios) synthetic gases in respective burner. They also examined swirl number effects on flame structure by utilizing instant flame images. It was shown that the swirl number had a non-monotonous effect on stability limits and these limits were differently affected by the swirl number; the place where the concentration of reactive intermediates was high moved towards combustor walls and axial temperature values decreased as swirl number increased; CO emissions were highly susceptible to the swirl number \([3]\). Readers may refer to the literature to find more studies related to both reactive and non-reactive swirling flows \([7–13]\).

In this study, effects of swirler hub diameter to outer diameter on combustion and emission behaviour of synthetic gas flames of premixed 20%CNG/30%H\_3/30%CO/20%CO\_2 mixture were experimentally investigated in a laboratory-scale combustor. To this end, 12 different swirl generators with different swirl numbers (0.4–1.4, at 0.2 intervals) and \( d_h/d_o \) ratios (0.3 and 0.5) were designed and manufactured so that effects of the \( d_h/d_o \) ratio could be evaluated at different swirl numbers. During experiments, the equivalence ratio (0.6) and thermal power of the combustor (3 kW) were maintained constant. For temperature and emission measurements, K- and B-type thermocouples and a portable flue gas analyser were used, respectively.

**EXPERIMENTAL SETUP**

The layout of the overall combustion system is presented in Fig. 1. Each synthetic gas constituent
is provided from a gas cylinder and their amounts are adjusted by a digital mass flow controller, which is governed by a vacuum system controller. Pressure regulators (on the gas cylinder and flow line) are used to drop pressure to the desired value (burner requirement, 20 mbar). For safety purposes, a solenoid valve is assembled for each flow line. This valve cuts off the gas flow in the case of flame absence, which may be caused by blowout, flashback, lift off, etc. Fuel gases are then mixed in a collector and directed to a static pre-mixer, where combustion air and fuel gases are completely mixed before entering the combustor.

The combustor is 1755 mm long, its walls are 5 mm thick; it is made of stainless steel. It has many slots for thermocouple and other measurement equipment installations. It also contains an external air fan for cooling the combustor material. The burner is also made of stainless steel and can operate thermal powers of up to 10 kW. It also incorporates a pressure sensor. In addition, there is a pilot ignition system to ignite fuel air mixture.

All experiments have been conducted at local atmospheric conditions. Mixing of fuel and air takes place at room temperature. As stated previously, equivalence ratio and thermal power
of the combustor are 0.6 and 3 kW, respectively. Mass flow rate of air and each synthetic gas constituent were specified based on these values.

RESULTS AND DISCUSSIONS

Temperature profiles throughout the combustor are one of the decisive parameters that characterize effectiveness of a combustion process. In Fig. 2, temperature profiles at different swirl numbers and $d_h/d_o$ ratios are illustrated. Temperature values are differently affected by the variation of $d_h/d_o$ ratio at different swirl numbers. In other words, effect of $d_h/d_o$ ratio on temperature distribution is not monotonic. However, all temperature curves show a good consistency by means of trend.

![Temperature profiles at different swirl numbers and $d_h/d_o$ ratios](image)

**Fig. 2.** Temperature profiles at different swirl numbers and $d_h/d_o$ ratios
At 0.4 swirl number, temperature values at 0.5 $d_{hub}/d_{oute}$ ratio are higher than those at 0.3 $d_{hub}/d_{oute}$ ratio throughout the combustion chamber. Nevertheless, this difference diminishes towards the outlet sections of the combustion chamber and becomes nearly insensitive to the axial location after the axial distance of 300 mm. At 0.6 swirl number, temperature curves in the flame zone are almost coincident for both tested $d_{hub}/d_{oute}$ ratios. Starting from the axial position of 200 mm, lower temperature values form at 0.3 $d_{hub}/d_{oute}$ ratio. The opposite behaviour is the case for 0.8, 1.2, and 1.4 swirl numbers. While temperature values are lower in and near the flame zone at 0.5 $d_{hub}/d_{oute}$ ratio (this difference is much more distinct at 1.2 swirl number), temperature profiles are nearly inline in the post flame zone (slightly lower at 1.4 swirl number). At 1.0 swirl number, higher temperature values form at 0.3 $d_{hub}/d_{oute}$ ratio throughout the combustion chamber unlike other swirl numbers tested. In conclusion, it can be said that the effect of the $d_{hub}/d_{oute}$ ratio is mainly controlled by the swirl number. Depending on the swirl number, this ratio slightly or dramatically varies temperature distribution in and near the flame region. However, post flame region temperature values are nearly irreversible to the $d_{hub}/d_{oute}$ ratio.

All emission measurements were performed at a fixed position at combustor outlet by waiting at least 5 min to reach the thermal equilibrium. In Fig. 3, measured CO values at different swirl numbers and $d_{hub}/d_{oute}$ ratios are illustrated.

Likewise, $d_{hub}/d_{oute}$ ratio effects on temperature profiles, $d_{hub}/d_{oute}$ ratio non-monotonically affects emissions of CO. At 0.4 swirl number, the difference between CO emissions at different $d_{hub}/d_{oute}$ ratios is very high (the highest at 1.4 swirl number). Consistent with temperature profiles, lower CO values form at 0.5 $d_{hub}/d_{oute}$ ratio since CO oxidation kinetics favour at higher temperatures. At 0.6 swirl number, emissions of CO do not change with $d_{hub}/d_{oute}$ ratio. At 0.8 swirl number, emissions of CO are higher at 0.3 $d_{hub}/d_{oute}$ ratio, although temperature values are higher at 0.3 $d_{hub}/d_{oute}$ ratio than at 0.5 $d_{hub}/d_{oute}$ ratio. This situation indicates that CO emissions depend not only on temperature distribution but also on flow field variations. CO emission values at 1.0, 1.2, and 1.4 swirl numbers also confirm this phenomenon. Overall, it can be concluded that emissions of CO are negatively affected by the decrement in the $d_{hub}/d_{oute}$ ratio (except for 1.0 swirl number).

Fuel components that contribute to the emissions of CO$_2$ are CO, carbon containing CNG constituents, and CO$_2$ itself. CO$_2$ emissions at 0.5 $d_{hub}/d_{oute}$ ratio do not significantly change with the swirl number (Fig. 4). When the $d_{hub}/d_{oute}$ ratio is 0.3, emissions of CO$_2$ vary identically at lower swirl numbers (0.4–0.8) but the variation becomes more unambiguous after the swirl number of 0.8. Consistently with the measured CO values, CO$_2$ concentrations in total exhaust gases are lower at 0.3 $d_{hub}/d_{oute}$ ratio.

**Fig. 3.** Variation of emissions of CO (ppm) with the swirl number and the $d_{hub}/d_{oute}$ ratio

**Fig. 4.** CO$_2$ concentrations (% – as a percentage of total exhaust gases) at different swirl numbers and $d_{hub}/d_{oute}$ ratios
Because all experiments were conducted under fuel lean equivalence ratio and there is no fuel originated nitrogen, measured NO\textsubscript{x} values are low (Fig. 5). Similar to CO\textsubscript{2} emission values, NO\textsubscript{x} values barely change with the swirl number at 0.5 \(d_h/d_o\) ratio. At 0.3 \(d_h/d_o\) ratio, emissions of NO\textsubscript{x} largely increase mainly because of the flow field alterations (temperature increments also increase NO\textsubscript{x}).

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{fig5.png}
\caption{Measured NO\textsubscript{x} values at different swirl numbers and \(d_h/d_o\) ratios}
\end{figure}

CONCLUSIONS

In this study, effects of the \(d_h/d_o\) ratio on combustion and emission behaviour of premixed 20%CNG 30%H\textsubscript{2} 30%CO 20%CO\textsubscript{2} mixture were experimentally investigated in a laboratory-scale combustor. Combustion behaviour was analysed by examining axial temperature profiles, whereas emission behaviour was evaluated by inspecting measured CO, NO\textsubscript{x}, and CO\textsubscript{2} values at the combustor outlet. Within the scope of this study, swirl generators with different swirl numbers (0.4–1.4, at 0.2 intervals) and \(d_h/d_o\) ratios were produced and tested under the same physical and boundary conditions. Main findings of this study are summarized below:

- \(d_h/d_o\) ratio does not substantially change trend of temperature profiles and temperature values in post flame region but it differently affects temperature values in and near flame region depending on the swirl number. Besides, some similar behaviours were also observed at 0.8, 1.2, and 1.4 swirl numbers.

- emissions of CO are non-monotonically affected by \(d_h/d_o\) ratio.

- at 0.5 \(d_h/d_o\) ratio, emissions of CO\textsubscript{2} and NO\textsubscript{x} are barely affected by swirl number variations. At 0.3 \(d_h/d_o\) ratio, both pollutants become more prone to the swirl number. In particular, emissions of NO\textsubscript{x} change dramatically with the swirl number.

ACKNOWLEDGEMENTS

We would like to thank the Scientific and Technological Research Council of Turkey (TÜBİTAK-MAG-215M821) for its financial support.

Received 4 June 2021
Accepted 10 August 2021

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