A Novel Burner for Investigating End-Gas Effects of Transient Flame-Wall Interaction


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Abstract
This paper presents the progress of the characterisation of a novel burner to investigate exhaust carbon monoxide emissions from premixed flames undergoing quenching and transient flame-wall interaction. The axisymmetric, atmospheric, fully premixed, forced laminar burner is designed to investigate the quenching behaviour as the flame oscillates. A speaker in the plenum induces these oscillations, and a cooled central tube quenches the inner portion of the flame. An emissions bench is used to accurately measure the end-gas CO and other species of interest. This enables systematic investigation of the contribution of CO emissions produced as a result of transient flame-wall interaction, relative to the overall end-gas emissions. The axisymmetric nature of the design provides scope for future detailed chemical kinetic modelling to be conducted, as well as providing validation data for this modelling. Furthermore, this burner is optically accessible, allowing in-situ studies to be undertaken using line-of-sight techniques.

Introduction
Emissions are a limiting consideration for industrial gas turbines. The two main emissions of interest are nitrous oxides (NOx) and carbon monoxide (CO), associated with high and low operating temperatures respectively. While NOx emissions have been well studied, CO emissions have not [9]. In general, CO is correlated with increased cooling air [8], however the controlling mechanisms are not well understood. It has also been shown in recent literature that the behaviour of CO emissions as a result of flame-wall interaction (FWI) differs between steady state and transient cases [10]. The impact of FWI on the CO formed with respect to the overall CO emissions produced by the flame, has not been systematically studied.

The concept of FWI is relevant to most combustion technologies. For many years, it has been linked to the production of pollutants, such as unburnt hydrocarbons (UHC) and CO. While FWI effects on UHC have proved to be minor [15], its effects on CO emissions have been harder to ascertain. Despite this uncertainty, it is still widely believed to be a major factor in the production of CO emissions [8, 9].

Early work on this topic has relied on numerical simulations [12, 3, 4, 13, 6] due to the challenges of experimentally measuring quantities close to a wall. The focus of these works have been on flame dynamics, with only a few [13, 6] using a detailed chemical kinetic model. The exclusion of detailed chemistry from these modelling studies make it difficult to draw inferences regarding the CO emissions produced by a flame undergoing FWI.

Two studies [5, 14] used laser based techniques to show that there is a build up of CO near the cold wall for a steady flame. While one [5] showed a qualitative increase in CO, the other [14] showed a quantitative increase, though this increase was minimal. Using a transient case, a study showed that the CO build up at the wall is different when compared to the steady case [10]. Their quantitative laser based in-situ CO measurement showed that a steady flame undergoing FWI only leaves a minor amount of CO at the wall. However, the transient case showed a significantly larger CO concentration upon contact with the wall. This indicates that there might be different chemical pathways that govern CO oxidation in a transient case [10].

The study of these potentially different chemical pathways is important in the context of gas turbine design. However, the more pressing study for gas turbine design is to determine the impact of FWI on CO emissions in engine-out exhaust. This engine-out exhaust is ultimately what is measured to determine whether the engine complies with the regulatory standards.

To that end, a burner has been designed to study the effect of the reported CO emissions due to transient FWI, on the overall exhaust. The burner has a premixed flame with a water-cooled quenching wall through the centre to induce the FWI. An automotive exhaust analyser was used to measured the CO emissions. In addition, a speaker in the plenum can induce transient FWI. This paper presents the characterisation of this burner in the unforced case, as well as preliminary CO measurements, both with and without induced FWI.

Experimental Method

Burner Configuration

The burner is a modified version of that used by Ref. [7], originally designed to study the response of acoustically forced laminar flames. Their design included the flame holder and the plenum, which has a speaker attached to the bottom. The flow rates of the reactants are controlled by two MKS 100B mass flow controllers. The fuel and air are combined and pass through honeycomb and a series of steel meshes to remove flow unsteadiness. The gas then passes through a ducted flame holder with a 25mm diameter circular port. The working section is confined in a quartz tube to allow optical access and shield from external air.
The WCT configuration. In this configuration, a 6
Cooled Tube (NCT) configuration and the With Cooled Tube
The design of the burner allows for two configurations: the No
Cooled Tube (NCT) configuration and the With Cooled Tube (WCT) configuration. Figures 1a and 1b show the burner in the
WCT configuration. In this configuration, a 6.35mm water-cooled tube (the cooling tube) passes through the centre of the flame holder and quenches the central section of the flame. This method of inducing FWI reduces the challenges associated with line-of-sight techniques for conducting in-situ, non-intrusive, FWI measurements. The temperature of the inlet and outlet water that cools the cooling tube, is measured by two thermo-couples shown in Figure 1b. The NCT configuration does not include the cooling tube.

Liquid petroleum gas (LPG) has been used as the fuel. Ignition of the flame is achieved with a small pilot flame, shown in Figure 1b, controlled by the ball valve highlighted in the box. This pilot flame does not enter the main gas path, and produces a small non-premixed flame outside the lip where the flame stabilises. The pilot flame is extinguished once the main flame is lit and stabilised.

The main exhaust path extends vertically up, then turns 90° towards the main exhaust outlet. A sampling tube is positioned at the base of the main exhaust outlet which is connected to the emissions analyser. The sample gas path, measured from the nozzle, travels approximately 450mm before entering the analyser inlet. As it enters the inlet, a thermocouple measures the gas temperature and a water trap removes condensed water. This temperature must be below 100°C so that the sample gas is dry, as required by the analyser. The whole assembly of fittings is supported on three adjustable rails, which allow for thermal creep.

The "M"-shaped flame is characterised by the outside edges of the flame stabilising on the outer edge of the circular port, while the centre of the flame stabilises on the cooling tube. This flame shape generally appears at \( \phi < 0.90 \) and \( \phi > 1.10 \). As the flow rate increases, the sides of this "M"-shaped flame stretch under the increasing flow velocity, eventually detaching and becoming unstable.

The "V"-shaped flame is characterised by a detached outside edge and an attached central edge. The corners of the flame curl towards the port, and the flame is only stabilised on the cooling tube. This shape generally appears at \( 0.90 < \phi < 1.10 \), except at low flow rates. The position of this flame along the tube is largely determined by the flow rate. As seen in Figure 3, at low flow rates the flame moves along the cooling tube upstream and causes flashback. At higher flow rates, the bulk flow velocity pushes the flame downstream. Figure 3 also shows that at high flow rates and near the lean blow-off limit, the "V"-shaped flame is also exhibited, albeit highly unstable.

At moderate to low flow rates (\( \leq 18slm \)), near the lean blow-off limit, the flame first exhibits a conical shape, similar to Figure 2a, before transitioning into a lifted "M"-shaped flame. At this point, the equivalence ratio needs to be increased up to 0.85 before the "M"-shaped flame again appears. The flame transitions through a "V"-shaped flame before finally stabilising into this "M"-shaped flame. The large range of different flame shapes warrants further study, but is outside the scope of this paper.

**CO Emissions Measurement**

The exhaust sample is analysed by an automotive exhaust analyser (Autodiagnostics ADS9000) using a Non-Dispersive Infrared (NDIR) technique. The CO span gas concentration of the analyser is 10% CO by volume and its resolution is 0.01%, giving an uncertainty of 0.005% or 50ppm. This analyser was verified against a Horiba 200 Series Emissions Bench, showing excellent CO measurement agreement [2]. In addition, a calibration was also conducted with a high (2% CO-N\textsubscript{2}) and low range (0.1% CO-N\textsubscript{2}) calibration gas. During this calibration, an STEC SGD-78 standard gas divider was used to meter a percentage of the calibration gas into the sample line, with

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\phi = 0.85. \quad \phi = 0.85. \quad \phi = 0.95. \]

Figure 2 shows three different steady and stable flame shapes achieved for each of the two configurations. Figure 2a shows the flame shape of the burner in the NCT configuration. This is similar to the conical flame studied by Ref. [7], which is stabilised on the lip of the outer edge of the circular port. In the WCT configuration, the two flames that are stable across the largest variation of flow rates and equivalence ratios (\( \phi \)) are the "M"-shaped flame (Fig. 2b) and the "V"-shaped flame (Fig. 2c). The exact appearance of both these flame shapes are strongly dependent on equivalence ratio and mass flow rate.

![Image 1](image1.png)

**Flame Characterisation**

![Image 2](image2.png)

![Image 3](image3.png)

Figure 2: Images of premixed LPG flame luminescence for characteristic stable flame shapes taken with a 5ms exposure. Mass flow rate is given in standard litres per minute (slm). The burner rim is shown for spatial context.
Preliminary investigation of the effects of burner heat loss were burner heat losses that decrease the flame temperature. This discrepancy could be the result of many factors, including reported in the literature for steady state FWI. The NCT case. This observation is consistent with the behaviour can be seen that they overlap closely with the measurements of 5 also shows measurements for the flame in the WCT case. It \[ \phi \approx 1.00 \] emission region occurs at \[ T_{\phi} \] equivalence ratio region. At a lean equivalence ratio, \[ CO \] emissions are minimal. The transition to the high \[ CO \] emission region occurs at \[ \phi \approx 1.00 \] and does so steeply. Figure 5 also shows measurements for the flame in the WCT case. It can be seen that they overlap closely with the measurements of the NCT case. This observation is consistent with the behaviour reported in the literature for steady state FWI.

Results

Experimental Measurements

Figure 5 shows a plot of the dry \[ CO \] concentrations with respect to the equivalence ratios for two experimental measurements and two simulation results. The measurements of the NCT case in Figure 5 show the highest \[ CO \] emissions produced in the rich equivalence ratio region. At a lean equivalence ratio, \[ CO \] emissions are minimal. The transition to the high \[ CO \] emission region occurs at \( \phi \approx 1.00 \) and does so steeply. Figure 5 also shows measurements for the flame in the WCT case. It can be seen that they overlap closely with the measurements of the NCT case. This observation is consistent with the behaviour reported in the literature for steady state FWI.

Chemical Kinetic Simulations

To better understand these results, a 1-D chemical kinetic simulation of an adiabatic, freely propagating, premixed, propane-air flame was conducted using CHEMKIN [1] with the NUI-G NG mechanism [11]. Comparing these results to the experimental measurements in Figure 5, it can be seen that the predicted \[ CO \] concentrations are significantly higher than the measurements. This discrepancy could be the result of many factors, including burner heat losses that decrease the flame temperature.

Preliminary investigation of the effects of burner heat loss were attempted by scaling the temperature profiles of the adiabatic freely propagating flame using an empirically derived relation (Eq. 1). The adiabatic flame temperature profile at a given equivalence ratio \( (T_\phi) \) is scaled by a factor \( (a) \) while keeping the unburnt gas temperature \( (T_{unburnt}) \) the same. This results in a scaled temperature \( (T_{\phi,scaled}) \) with a lower flame temperature and the same flame thickness.

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T_{\phi,scaled} = a(T_\phi - T_{unburnt}) + T_{unburnt} \tag{1}
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These temperature profiles provide the temperature across the domain of a burner stabilised premixed flame simulation. The simulation then used this temperature profile to solve for the species distribution across the domain, using the same conditions otherwise as the adiabatic freely propagating flame simulation. The resulting end-point \[ CO \] is corrected to a dry basis to compare with the experimental data, as the measurements are dry concentrations. The results of the 30% scaling case is also shown in Figure 5, which shows improved agreement with the experimental measurements.

Discussion

While Figure 5 clearly shows that the adiabatic freely propagating flame simulation is far from the NCT measurements, the non-adiabatic simulation is closer. At a heat loss that is represented by the 30% scaling, it approaches the magnitude of the experimental data. This particular chemical kinetic model contains a comprehensive propane combustion mechanism, but little work has been done to test the mechanism’s ability to model \[ CO \] oxidation at low temperatures. Given the gas temperature and the sample line length, the low temperature \[ CO \] oxida tion behaviour could contribute significantly to the post-flame exhaust \[ CO \] emissions.

In addition to the chemical kinetics, the simulation setup could also contribute to the discrepancy. The simulation results presented in Figure 5 are of freely propagating and burner stabilised flames only, and hence do not account for the sample line length. The cooling effect imposed by the sample line likely leads to different chemical equilibria compared to those of the simple simulation presented in Figure 5. The magnitude of this effect, and those of the chemical kinetic model inaccuracy, warrant further investigation.

Conclusion

A novel burner has been developed to study the effect on combustion emissions from premixed flames undergoing flame-wall interaction. The burner is a confined, atmospheric, forced laminar burner with an optional central cooling tube. It is connected to an exhaust manifold that permits undiluted sampling of exhaust emissions using an automotive emissions analyser. Its axisymmetric configuration is suitable for numerical modelling and its convex quenching region is amenable to optical diagnostics. Preliminary \[ CO \] emissions measurements have been conducted and compared to those predicted for adiabatic and non-adiabatic premixed flames.

The results show a discrepancy between the flame measurements, and those predicted for an adiabatic freely propagating flame. Simulating heat loss, with a burner-stabilised premixed...
flame with scaled temperature profiles, decreases the discrepancy but do not completely match the measured data. Measurements of the unforced flame with the central cooling tube show minimal difference to the results found for the unforced flame without the central cooling tube. Given the expectation of increased exhaust CO with flame quenching, this result suggests that complex competing mechanisms will need to be explored further. These mechanisms include the relationship between steady and unsteady quenching as suggested by Ref. [10], the role of post-flame exhaust temperatures in the global consumption of CO, and the accuracy of the CO sub-mechanisms in current chemical kinetic models.

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References