SMALL SCALE SURFACE COMBUSTION OF INDUSTRIAL GAS OIL (IGO) SUPPORTED BY COOL FLAMES

Franz v. Issendorff*, André Wittmer°, David Diarra*, Klaus Lucka°, Heinrich Köhne*

*Energie- und Stofftransport/RWTH-Aachen, Kopernikusstr. 16 52056 Aachen
°OWI Oel-Wärme-Institut, Schönauer Friede 108, 52072 Aachen

ABSTRACT

The paper is concerned with the premixed combustion of IGO by a small scale radiant burner provided with a fine wire mesh. The burner has been developed in recent years for household purposes, consequently it has been designed for powers from 8 kW to 25 kW. Its main advantages are not only the reduced burning volume - needed as the heat transfer in the burning chamber is increased by radiation - but also a very steady and therefore quiet combustion. The paper discusses the dependencies of equivalence ratio and firing rate on surface temperatures and the gas temperature at the surface. Results of CFD-simulations of the inner fluid dynamics are presented.

1 INTRODUCTION

The need for small scale burners for household application in the range from 8 kW up to 25 kW has been increasing in recent years. Modern house insulation as well as the demand for space-saving in houses have lead to the replacement of older heating facilities by wall-mounted gas-boilers.

There is still a huge amount of oil-driven heating facilities to be replaced. As a consequence, there is a necessity for improving oil-burners with respect to emissions and minimization of the space required.

Taking these requirements into account, radiant burners can be a useful alternative to common so-called blue flame burners. The heat flux by radiation from the burner surface to the boiler walls reduces the boiler surface needed for convective heat exchange. Additionally radiant burners feature low NO\textsubscript{X} and CO-emissions and quiet operation.

Nevertheless radiant oil-burners are not available on the market. This is due to the problems arising from the complex interactions of the processes of atomization, vaporization and mixing of fuel with combustion air preceding the combustion and the low ignition temperature of over-stoichiometric oil vapour/air mixtures. Depending on the mixture, the theoretical ignition temperature is usually lower than the boiling temperature of the highest hydrocarbons.

The present paper describes a new type of radiant burner [1] for liquid fuel. The cylindrical burner is based on prevaporization and fuel/air-mixing inside a cylindrical wire mesh. The preheated fuel/air mixture flows through the wire mesh and a flame stabilizes close above the surface. The heat needed for vaporization and preheating of the fuel/air mixture is transferred from the surface to the inner parts of the burner and by Cool Flames.

The results of measurements of the NO\textsubscript{X} and CO-emissions and the surface temperatures at various firing rates and equivalence ratios are presented. Furthermore, gas temperature measurements in the flame have been carried out. Results of CFD-simulation of the fluid dynamics inside the cylindrical surface are included.

2 BURNER DESIGN

The burner (Figure 1) consists of a cylindrical surface made of a fine wire mesh and a start-burner below the surface. The start-burner design resembles a blue flame burner. In stationary mode the start-burner and the remaining space below the surface serve for mixing.

The burner is provided with a Simplex pressure atomizer which produces a 80° hollow cone spray. The spray is surrounded by a preheated coflowing air stream. Preheating of the air is necessary to enhance the vaporization.

The preheating of the combustion air takes place in heat exchange tubes through which approximately half of the combustion air is directed. After passing the mixing chamber an air-temperature of approximately 150° C is achieved, varying with power and equivalence ratio.

The fuel/air-mixing takes place in the flame tube, which is provided with a recirculation slot, located at the upstream end.

In order to reduce the burner size the combustion air is swirled. The swirl allows, as is generally known, an intensive mixing of the fuel droplets with the air. Furthermore, it serves the flame stabilization inside the flame tube during the start mode. The swirl is generated by an axial swirler situated in the air nozzle.
Figure 1: Burner design

The surface is made of a fine wire mesh, with the following specifications:

- Porosity: 51%
- Weight: 1.55 kg/m²
- Thickness: 0.4 mm
- Grade of Filtration: 40 μm.

The resulting pressure loss is 50 Pa for an average flow velocity of 0.2 m/s.

3 BURNER OPERATION

In order to avoid the necessity of electrical preheating, the burner is provided with a start-burner. During the start mode a flame is ignited below the surface of the burner and stabilizes inside the flame tube. The flue gases are partly recirculated through the recirculation slot in the flame tube. The surface is heated up to approximately 750 °C by the passing flue gases.

In order to switch from start mode to stationary mode, the fuel supply is interrupted for a short time and the flame inside the burner extinguishes. By opening the fuel feed again, an oil-vapour/air-mixture ignites on the surface. In stationary mode the space below the surface serves as a mixing chamber.

The heat for preheating the combustion air and the vaporization is transferred into the burner by solid body radiation of the wire mesh. Cool flames, which stabilize inside the cylindrical surface, enhance the vaporization. In order to achieve a homogenous mixture below the surface, strong recirculation is utilized.

4 COOL FLAMES

The phenomenon of cool flames describes exothermic reactions of hydrocarbons below the ignition temperature. Cool flames of IGO start at temperatures above 300 °C. Due to reaction inhibition at temperatures of approximately 480°C no further increase in temperature occurs. Consequently, the existence of Cool Flames does not necessarily lead to ignition.

The usage of cool flames facilitates the evaporation and can be used to produce a combustable gas from liquid fuels. This is not only restricted to small scale combustion and could be applied in a wide range of different technical applications.

The appearance of Cool Flames below the surface of the burner is presented in Figure 2. It shows a characteristic temperature of the recirculating gas plotted against the equivalence ratio at different specific firing rates.

It can be seen that for a fixed firing rate the temperature of the recirculating mixture rises with increasing equivalence ratio. This can be explained by the reduced air stream through the burner. A reduction of the air stream influences the recirculation temperature in two ways:

1. less air has to be heated
2. the surface temperature rises as the surface-flame distance is reduced.

Apart from restrictions by the material employed, there are two limiting conditions for operating the burner, i.e. the heat flux for vaporization is insufficient.

Below recirculation temperatures of 200 °C, the vaporization of the fuel is not complete and the burner extinguishes. By increasing the equivalence ratio, the temperature exceeds 300 °C and a sudden temperature rise of 170 K takes place, induced by a Cool Flame. By further increasing the equivalence ratio the temperature rises until a spontaneous ignition occurs below the surface.

Lowering the firing rate increases the recirculation temperature due to the reduced oil/air-mixture to be heated. It can also be seen that spontaneous ignition occurs at lower equivalence ratios, but at similar temperatures for different firing rates.
5 SURFACE TEMPERATURES

For the design of a radiant burner the knowledge of the surface temperature is of importance. The surface temperature influences the durability of the surface and, as could be seen in Figure 2, limits the burner operation due to heat transfer.

Surface temperatures were determined with a two-colour-pyrometer. Different equivalence ratios and firing rates were analysed. The results are represented in Figure 3. The points connected by lines correspond to similar equivalence ratios. As one can see in Figure 3, the surface-temperature rises with the increase of the firing rate up to a maximum. With further increase of the firing rate the surface-temperature slightly falls. The maximum surface temperature of 875 °C was attained at a firing rate of 304 kW/m² and an equivalence ratio of \( \varphi = 0.96 \).

Figure 3 also shows the influence of the equivalence ratio on surface temperature. By reducing the air stream the surface temperature increases due to the higher heat exchange between flame and surface and the reduced energy needed for preheating.

6 GAS TEMPERATURES ABOVE THE SURFACE

In order to investigate the behaviour of the flame front above the surface in greater detail, gas/exhaust gas temperature measurements were executed. Temperatures were measured by use of a thermocouple device, which compensated the influence of radiation. The results in Figure 4 show the gas/exhaust gas temperatures above the surface at a firing rate of 284 kW/m² and an equivalence ratio of 0.91. It is clearly seen that the temperature rises at a distance of approximately 0.3 mm above the surface. It becomes evident that the preheating zone of the flame is situated above the wire mesh, which corresponds to the results of other authors [2] [3]. The temperature rises from 1220 °C to over 1500 °C and achieves a maximum of 1540 °C. Afterwards the exhaust gas temperature constantly drops.

7 EMISSIONS

In Figure 5 the NO\(_x\) - and CO- emissions of the burner are presented at firing rates of 303 kW/m² and 160 kW/m². At a firing rate of 160 kW/m² the emissions remain constant over a wide range of equivalence ratios. Due to the increased surface temperature at a firing rate of 303 kW/m², the NO\(_x\) emissions are higher. The higher CO-emissions at this firing rate can be explained by losses at the flame edges.
The flow inside the burner was simulated with the commercial CFD-Code PHOENICS. The geometry was simplified to an axisymmetric problem to reduce memory usage and CPU time. The preheating tubes were not included. The temperature of the incoming air was set to environment temperature. The grid size was set to 200 cells in axial and 90 cells in radial direction. This leads to a characteristic cell length of about 1 mm. The turbulence was modelled with the well-known k-ε model. Although the swirling flow cannot be modelled accurately, reasonable results can be achieved for the axial and radial velocities. The first calculations were made without radiation, further investigations will be made including radiation effects inside and outside the burner as well as a corrected heat transfer to the incoming air. To simulate the wire mesh, a pressure drop patch was inserted. In addition, the temperature of the mesh was fixed to a value of 875 °C. The mass flow was adapted to a thermal power of about 15 kW at an equivalence ratio of 0.96. This is equivalent to a firing rate of about 250 kW/m². Figure 6 shows the basic flow inside the burner.

The inserted lines of zero velocity (axial: w, radial: v) indicate the locations of the vortices. The gases flowing backwards do not only flow outside the flame tube but also inside of it and the inner recirculation can be stabilized. As the air is accelerated at the end of the injector an effective mixing with the oil spray is achieved.

In Figure 7 the simulated temperature distribution inside the burner is presented. It can be seen, that the recirculating gases passing near the wire mesh are heated up to temperatures above 500 °C before entering the evaporation zone of the oil spray by way of the recirculation holes. The portion of recirculating gases depend on the pressure drop of the wire mesh and the geometry of the flame tube. The recirculating mass of gases is almost 0.3 times the incoming air mass flow. Usual blue flame burners have a recirculation rate in the same magnitude. The recirculation coincides well with the measured temperatures near the recirculation slot of about 480 °C, the inlet temperature below 150 °C and a mixing temperature of more than 400 °C. As shown before, this effect is essential to maintain the Cool Flame inside the burner. The high recirculation rate forces the incoming air to penetrate over a very long distance into the evaporation zone although a realistic strong swirl generation is considered. The oil spray coming into this stream has to be evaporated in a very short time to avoid the collision of droplets on metal parts of the burner. This is important especially near the air injector, where sedimentation of oil droplets occur due to very high temperatures and low gas velocity.
9 CONCLUSION

A small scale radiant burner for the premixed combustion of IGO was presented. Measurements have shown a very quiet combustion and low NOx- and CO-emissions. It was shown that the burner can be operated in a relatively wide range of equivalence ratios and firing rates.

Experimental results of surface temperature measurements demonstrate the influence of equivalence ratio and firing rate on the surface temperature.

Special attention has been drawn to the recirculation of hot gases to support evaporation and the essential Cool Flame for the mixing process. The existence of the Cool Flame was demonstrated by gas temperature measurements below the surface.

The use of CFD made it possible to visualize the flow field and the temperature distribution beyond the surface. The results confirm experimental observations. Due to the presence of large surfaces at high temperatures the influence of radiation should not be neglected. For this reason further investigations will focus on the modelling of radiation to verify its influence.

10 REFERENCES