CHEMICAL ENGINEERING TRANSACTIONS Volume 21, 2010
Editor J. J. Klemeš, H. L. Lam, P. S. Varbanov
 ISBN 978-88-95608-05-1 ISSN 1974-9791
DOI: 10.3303/CET1021188

Local Wall Heat Fluxes in Swirling Non-Premixed Natural Gas Flames in Large Scale Combustor: Data for Validation of Combustion Codes

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The main purpose of this work is the measurement of local heat fluxes (absorbed into water cooled walls) from confined non-premixed swirling gas flame. Presented are two thermal duties 745 kW and 1120 kW fired in a large-scale combustion chamber. Measured data are intended for use as a benchmark. Its purpose is to validate computational fluid dynamics (CFD) codes against reliable and transparent measurements. The uniqueness of reported measurements is in the scale of the experimental facility and its ability to provide accurate data at above thermal duties. The data uncertainty is also discussed. The experimental facility is described including complete geometry of the burner and combustion chamber to enable use of the measured data by other researchers. The entire geometry is made available in the STEP format.

1. Introduction

Study of flame structure is the subject of long-lasting interest of the combustion modelling community. Detailed in-flame measurements of temperature, velocity and species concentrations have served for the validation of all existing combustion models. Unlike the in-flame properties, wall heat fluxes have been used for model validation only rarely. Heat flux measurements reported in the literature are either spot measurements or global heat transfer rates. Spot measurements however mostly provide just the thermal irradiation flux, not the actual radiative or total heat transfer rate, e.g. in the study of industrial furnaces and boilers (Kobayashi et al., 2002; Ströhle, 2004; Hayes et al., 2001). Likewise, global heat transfer rates calculated from the total hot water (steam) production are insufficient for the validation of detailed predictions.
In contrast to that, the interest of engineering community focuses primarily on local heat fluxes and pollutant emissions. Emissions are studied namely to ensure compliance with legislative regulations, e.g. directive (EC 2001), while heat fluxes are required to check proper furnace design and to ensure safe operation. It is thus apparent that the correct prediction of local heat fluxes on heat transfer surfaces is one of the most important aspects of practical combustion simulations that should receive adequate attention.

Swirl-stabilised non-premixed flames are frequently used in industrial burners, but they present one of the most difficult problems to predict computationally. Only with the advances in large eddy simulations (LES), successful predictions of in-flame properties were reported e.g. in (Fureby et al., 2007; Sadiki et al., 2006; James et al., 2007; Dinesh et al., 2010). The LES approach is unfortunately still too computationally expensive for the simulation of large-scale fired heaters due to their huge dimensions (on the order of 10 m) and the need to resolve fine features like gas nozzles with diameters on the order of 1 mm. The only viable alternative for practical predictions in the present as well as for a number of years to come thus consists of models based on first or second-order turbulence closures as eg. in (Khelil et al., 2009).

2. Testing facility

The construction of the semi-industrial experimental facility for burners up to 2 MW was aimed at providing variable length of combustion chamber and accurate heat flux and emission measurements. The main feature distinguishing the test facility at Brno University of Technology from others is the ability to measure local heat transfer rates to the cooled walls, which is enabled by segmental design of the combustion chamber. There are up to seven water cooled segments of the combustion chamber, see Figure 2. All internal segments have the same flame-facing area of 1.57 m^2, whereas the first and seventh section has 1.26 m^2 and 3.14 m^2, respectively. The last three segments are removable which allows adjustment of the combustion chamber length.

The burner was a low-NOx design with staged gas supply (Figure 1) and axial swirl generator, fired by natural gas. Swirl generator with diameter 240 mm consisting of 8 vanes acts as a flame holder. Gas inlet consists of twelve primary nozzles and eight secondary nozzles. Eight of the primary nozzles have diameter 2.6 mm and the other four 3.0 mm. All the primary nozzles are drilled in a nozzle head located on the burner axis.

Secondary gas injection is performed by four additional nozzle heads located in regular intervals around annular air channel which surrounds the primary nozzle head. Each of the four secondary nozzle heads has two nozzles with a diameter of 3.3 mm.

Flame ignition and stabilization is performed by a small (25 kW) premixed natural-draft pilot burner. Its thermal duty was included in the total thermal duty.

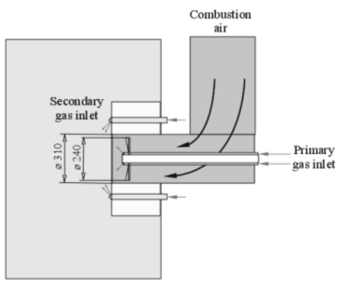

Figure 2: Swirl burner with two gas stages

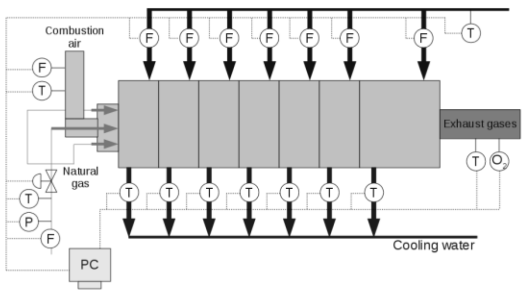

Figure 1: Combustion chamber and main parts of the data acquisition system

The burner performance was set to 745 kW (Case 1) and 1120 kW (Case 2) with excess air ratio 1.1. Measurements previously performed at the same testing facility, although with different objectives than in this work, were described e.g. in (Kermes et al. 2008).

3. Measurements

Each of the two studied regimes (Case 1 and 2) was measured twice to test repeatability. Operating conditions during stabilization time are shown in Table 1. It can be seen that reproducibility is good even though conditions were slightly changed e.g. air temperature 19.3 °C vs. 4.3 °C in Case 1. The deviation in the total extracted heat flux into the water was 0.16 % for the Case 1 and 0.88 % in the Case 2. The maximum load difference was in 3rd section 2.7 % for the Case 2 and 2.4 % in 5th section for the Case 1. The total extracted heat from flame (Q_{total}) was evaluated from natural gas flow rate, temperature and pressure. Equivalent methane mass flow rate (m_{CH4}) is calculated from natural gas flow rate using the ratio of heating values of natural gas and methane. Air flow was calculated based on oxygen concentration in exhaust gases and the fuel flow rate. The data displayed in the tables and graphs are averaged values over 5 minutes (Case 1) and 15 minutes (Case 2). Averaging was applied to remove random fluctuations in the measured values.

Measured heat fluxes (extracted heat rates in each of the seven sections along the combustion chamber) are shown in

Table 2. The displayed data are averages from the two repeated measurements for each of the cases. The averaging period was the same for measured data as for operating conditions. Mean fluctuations during the averaging period are shown as well. All of the fluctuations are under 1 % of measured heat fluxes.

Table 1: Operation conditions of the measurements

		Case 1				*Case 2*			
		Measurement 1	*Uncertainty [%]*	*Measurement 2*	*Uncertainty [%]*	*Measurement 1*	*Uncertainty [%]*	*Measurement 2*	*Uncertainty [%]*
Q_{total}	[kW]	746	1.62	748	2.7	1115	1.6	1124	2.4
m_{CH4}	[kg/s]	0.015	1.62	0.015	2.7	0.022	1.6	0.022	2.4
m_{air}	[kg/s]	0.289	9.8	0.290	10.1	0.435	9.8	0.438	10
T_{fuel}	[°C]	20.1	1.5	12.5	2.6	20.6	1.5	13.1	2.3
T_{air}	[°C]	19.2	1.5	4.3	1.9	20.6	1.4	8.5	1.6
Q_{water}	[kW]	438	4.3	438	5.27	592	3.3	597	4.2
RH	[%]	42		90		41		81	

The fuel distribution among primary and secondary nozzles was determined from numerical simulation of gas distributor based on total fuel flow rate since only the gas main is equipped with flow meter. Fuel flow rate in Case 1 was 3.84 10^{-3} kg/s for primary and 1.1 10^{-2} kg/s for secondary nozzles. In Case 2 the distribution was 5.79 10^{-3} kg/s to the primary and 1.65 10^{-2} to the secondary nozzles.

Using the information from sensor manufacturers, maximum error of heat fluxes was calculated and results are shown as uncertainties in

Table 2. To do this, the theory of error propagation (Braembussche, 2001) was utilized.

Table 2: Measured heat fluxes

	Case 1 Heat flux [kW/m²]	Mean fluctuation [kW/m²]	Uncertainty [%]	Case 2 Heat flux [kW/m²]	Mean fluctuation [kW/m²]	Uncertainty [%]
Section 1	17.25	0.15	8.4%	21.88	0.21	6.4%
Section 2	25.57	0.16	4.8%	34.05	0.27	3.5%
Section 3	40.17	0.14	2.9%	53.28	0.26	2.3%
Section 4	46.41	0.15	2.8%	63.58	0.24	2.0%
Section 5	47.87	0.16	2.6%	65.45	0.27	1.9%
Section 6	42.33	0.17	2.8%	58.9	0.29	2.0%
Section 7	31.4	0.21	2.0%	42.74	0.19	1.6%

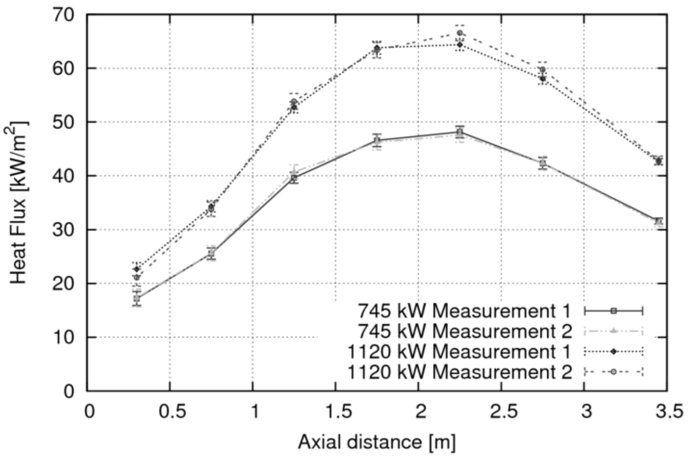

Figure 3: Comparison of measured heat fluxes in seven sections

Geometry of the flow domain in STEP format has been prepared for researchers interested in making their own computational analysis of the present flames. It will be provided upon request by the corresponding author.

4. Verification of air flow rate measurement

An additional procedure for the measurement of air flow rate has been implemented to provide verification of the primary method. The primary indirect measurement based on oxygen concentration in the flue gas has a long response time and rather large uncertainty as documented in Table 1. The calculation of air flow rate is in the primary method based on the measured O_2 content in flue gas and measurement of natural gas flow rate, which itself depends on the readings of three sensors as described above.

The second method of flow rate measurement which provides verification for the first one employs a vane anemometer located directly in combustion air pipe which has inner diameter of 246 mm. Information from the anemometer is also automatically collected by the data acquisition system. The uncertainty provided by manufacturer is ±0.1 m/s and ±1.5 % of measured value. The readings from the vane anemometer are however biased due to natural non-uniformity of flow profile in the pipe and further due to slight non-symmetry of velocity profile at the location of measurement. This is caused by a 90° turn of the pipe, which precedes the probe by approximately 12 diameters. In order to provide a reliable correction function for the vane anemometer, the turbulent flow in the air supply pipe has been modelled using ANSYS FLUENT software system for several flow rates spanning the range corresponding to admissible burner duties. The results displayed in Figure 4 show that the following linear correction function is appropriate:

$$\dot{m}_{air} = 0.972 S v_{anem} \rho - 0.015, \quad (1)$$

where $\dot{m}_{air}$ [kg/s] is total air flow rate through the duct, S [m^2]is cross-sectional area, ρ [kg/m^3] is air density and v_{anem} [m/s] is the velocity measured by the anemometer. In the simulations was applied no-slip condition at the walls and wall roughness height equal to 0.1 mm. Table 3 provides a comparison of the corrected values from the anemometer with data based on the flue gas O_2 measurements.

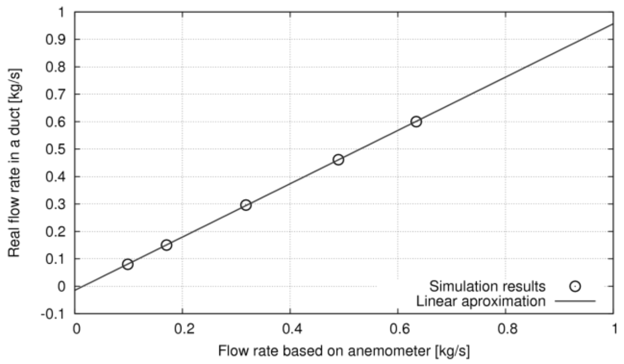

Figure 4: Vane anemometer air flow rate

Table 3: Mass flow rate of air in the supply air duct

Predicted by flue gas analyzer	kg/s	0.283	0.377	0.428	0.510	0.548
Predicted by anemometer	kg/s	0.273	0.393	0.434	0.512	0.585
Relative deviation	%	-3.49	4.31	1.38	0.57	6.74

6. Conclusions

Local wall heat fluxes were investigated in a cylindrical, water-cooled large experimental combustion chamber for non-premixed swirling natural gas flames.

Measured data are provided for two firing rates (745 kW and 1120 kW) together with a complete geometry of the fluid flow domain including air duct, staged-gas burner and combustion chamber. Error analysis is included to complete the data base. This validation benchmark is provided to the research community as a basis for the validation of combustion models and codes using a problem which is both well documented and practically relevant. The article also discusses air flow rate measurement issues when vane anemometry is utilized. Correction function for particular duct size is derived to account for in-duct velocity profile.

Acknowledgement

The authors gratefully acknowledge financial support of the Ministry of Education, Youth and Sports of the Czech Republic within the framework of research plan No. MSM 0021630502 “Waste and Biomass Utilization focused on Environment Protection and Energy Generation” and within the framework of Operational Programme “Research and Development for Innovations” – “NETME Centre – New Technologies for Mechanical Engineering”.

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