THE 29TH TLM - IEA IMPLEMENTING AGREEMENT ON ENERGY CONSERVATION AND EMISSIONS REDUCTION IN COMBUSTION

Gembloux (Belgium), 02-06 September 2007

CO emissions of fuel oil boilers in transient and cyclic regimes: performance of burners without and with "anti-drip" systems

Sub - Task 2.1H

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CONTENTS

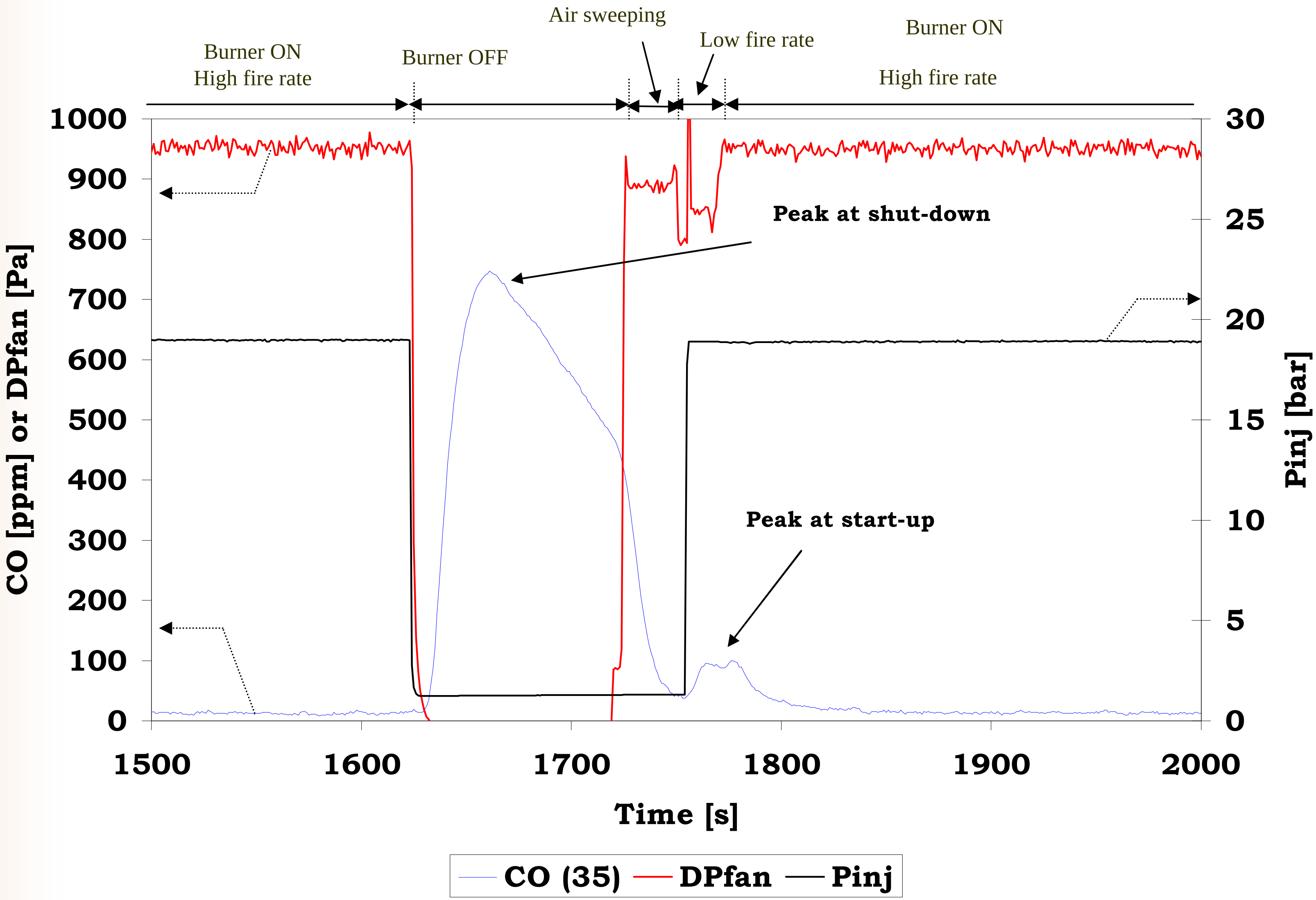
INTRODUCTION

Objective: investigation of CO emissions of fuel oil boilers in transient and cyclic regimes.

European standards for steady-state CO emissions of boilers (output $\leq$ 400kW) : 110mg/kWh.
Transient and cyclic CO emissions not regulated.

Literature review → Other studies mention CO peaks at start-up and shutdown of fuel oil burners.
At start-up: Poor fuel atomisation
At shutdown: Oil after-drip → Incomplete combustion

CO emission for one cycle

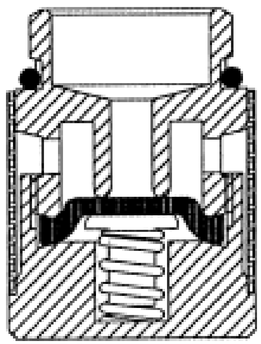


Ref: Cuevas and Ngendakumana, 28th TLM, Heidelberg 2006, « CO emissions of fuel oil boilers used in domestic heating »

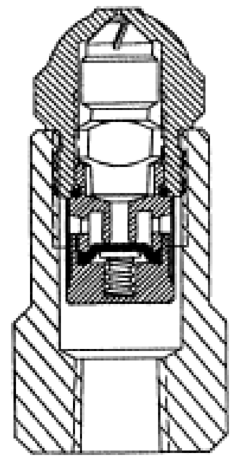
Fuel after-drip at shutdown

To avoid oil after-drip
→ Install an anti-drip system on the nozzle

Anti-Drip System



EcoValve™



Nozzle Assembly
with
EcoValve™

TEST BENCH

Boiler : Viessmann Paromat-Triplex

type: KN-032

Power : 320-370 kW

Burner : ABIG

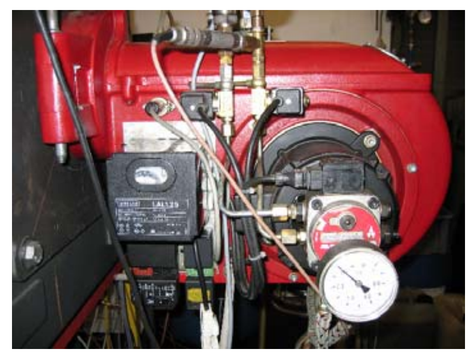
type AW-1ZV

Power: 118-652 kW

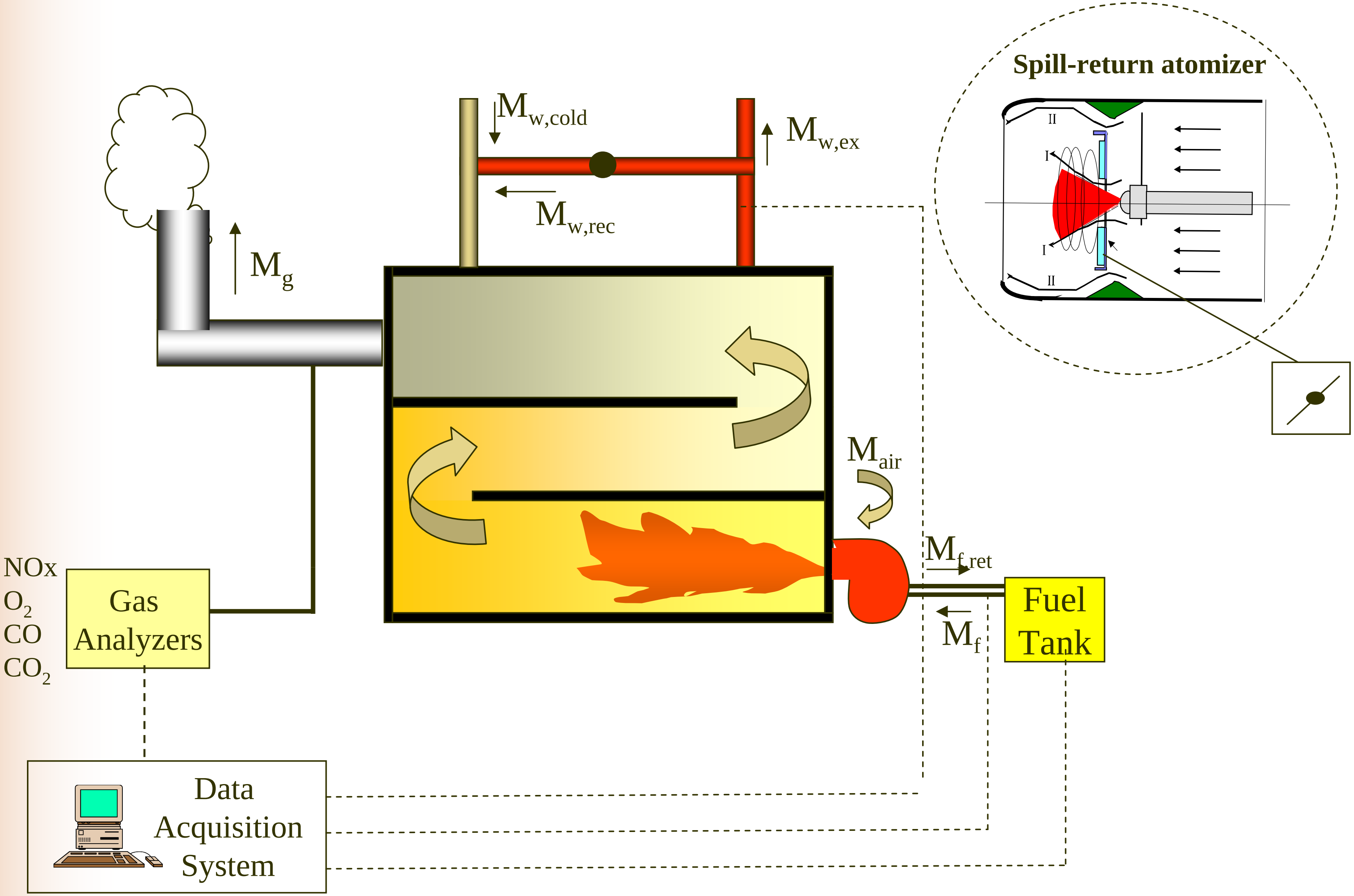
Nozzle : spill return and 45° type

Nominal capacity rate: 35kg/h

Nominal injection pressure: **20 bar**



TEST BENCH



Spill-return atomizer
II
I
I
II
$M_{w,cold}$
$M_{w,ex}$
$M_{w,rec}$
M_g
M_{air}
$M_{f,ret}$
M_f
Fuel Tank
NOx
O_2
CO
CO_2
Gas Analyzers
Data Acquisition System

TESTS PERFORMED

→ Tests performed with and without an anti-drip system:

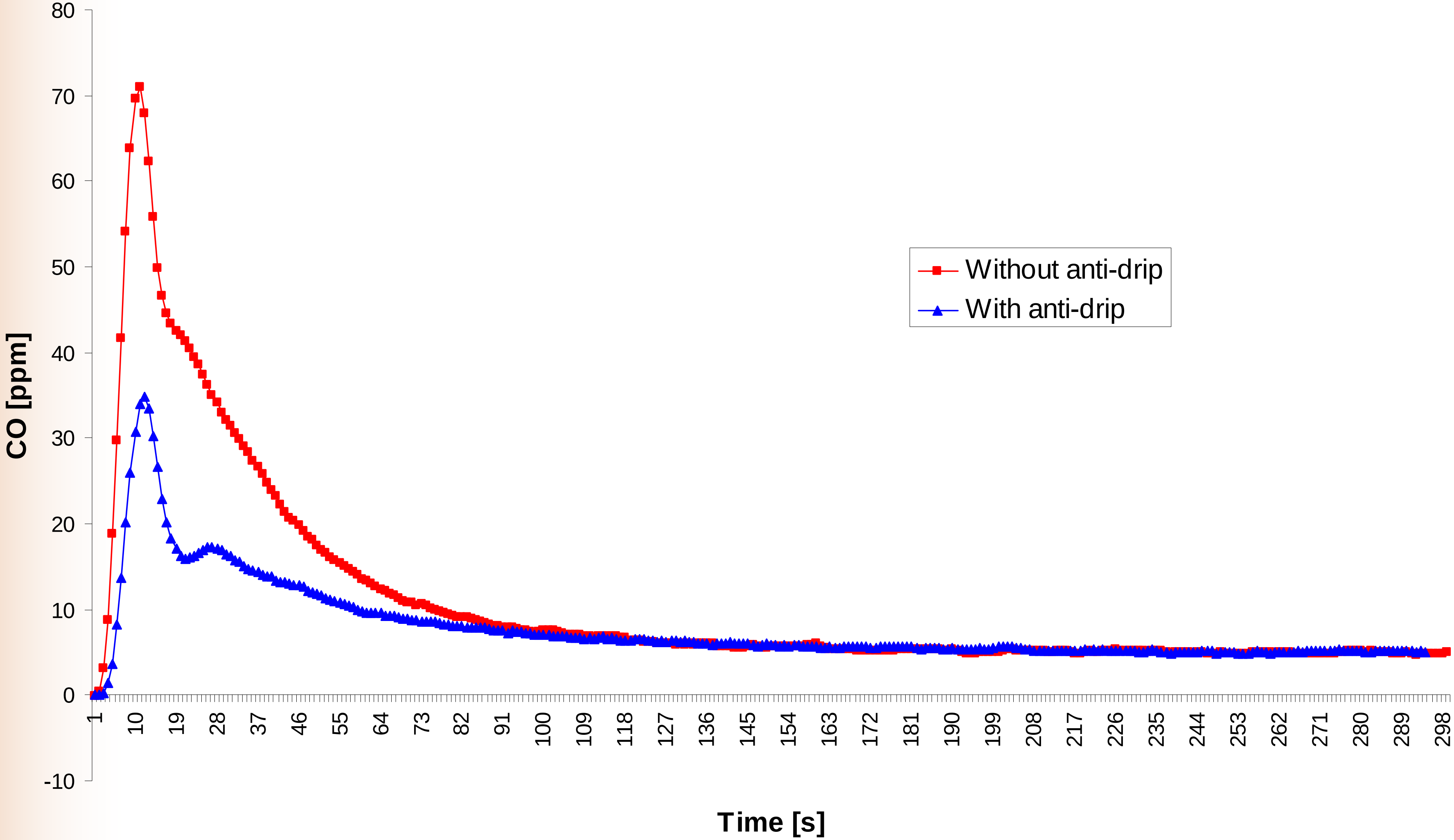
- Cold start tests
- Cyclic regime tests

→ Tests conditions:

	Patm [mbar]	tamb [°C]	Rhamb [%]	Pret [bar]	Pinj [bar]	M fuel [kg/s]	M w [kg/s]	O2 [%]
Without anti-drip system	985	27,7	53,0	15,2	19,0	0.01052	1,39	3,0
With anti-drip system	963	21,8	46.8	15,3	19,8	0.01046	1,38	3,0

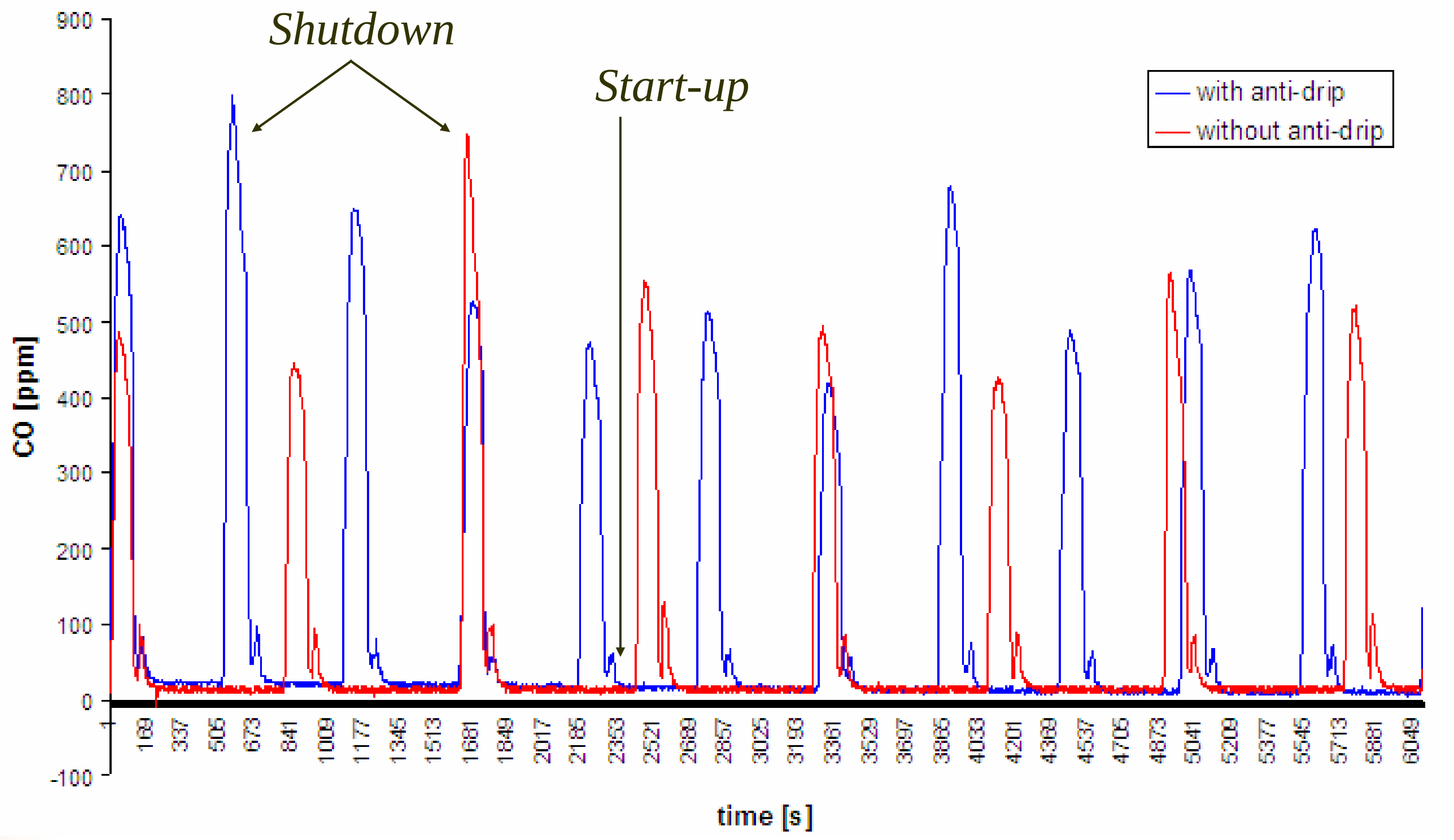
RESULTS

Cold Start



RESULTS

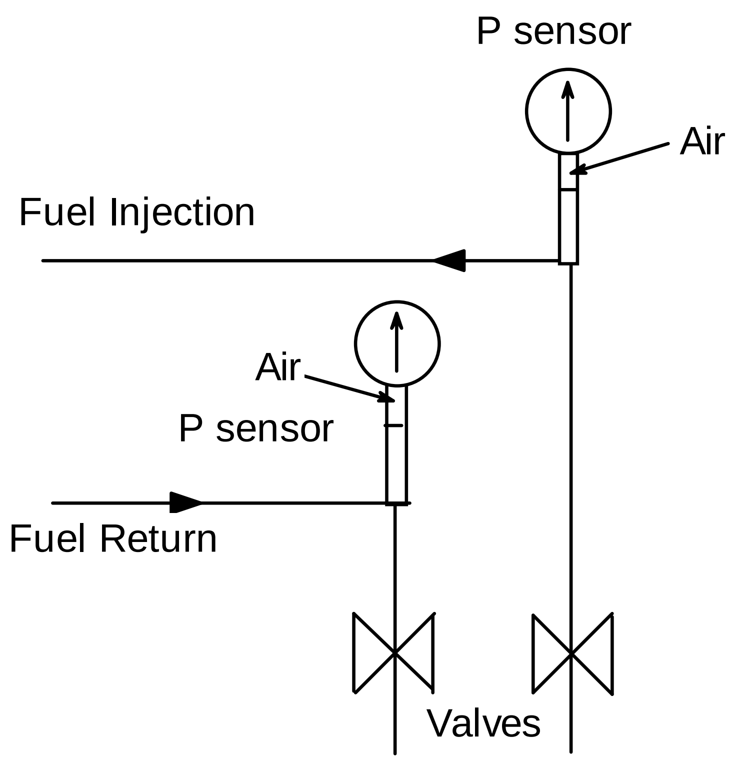
Cyclic Regime



No difference in cyclic regime ?

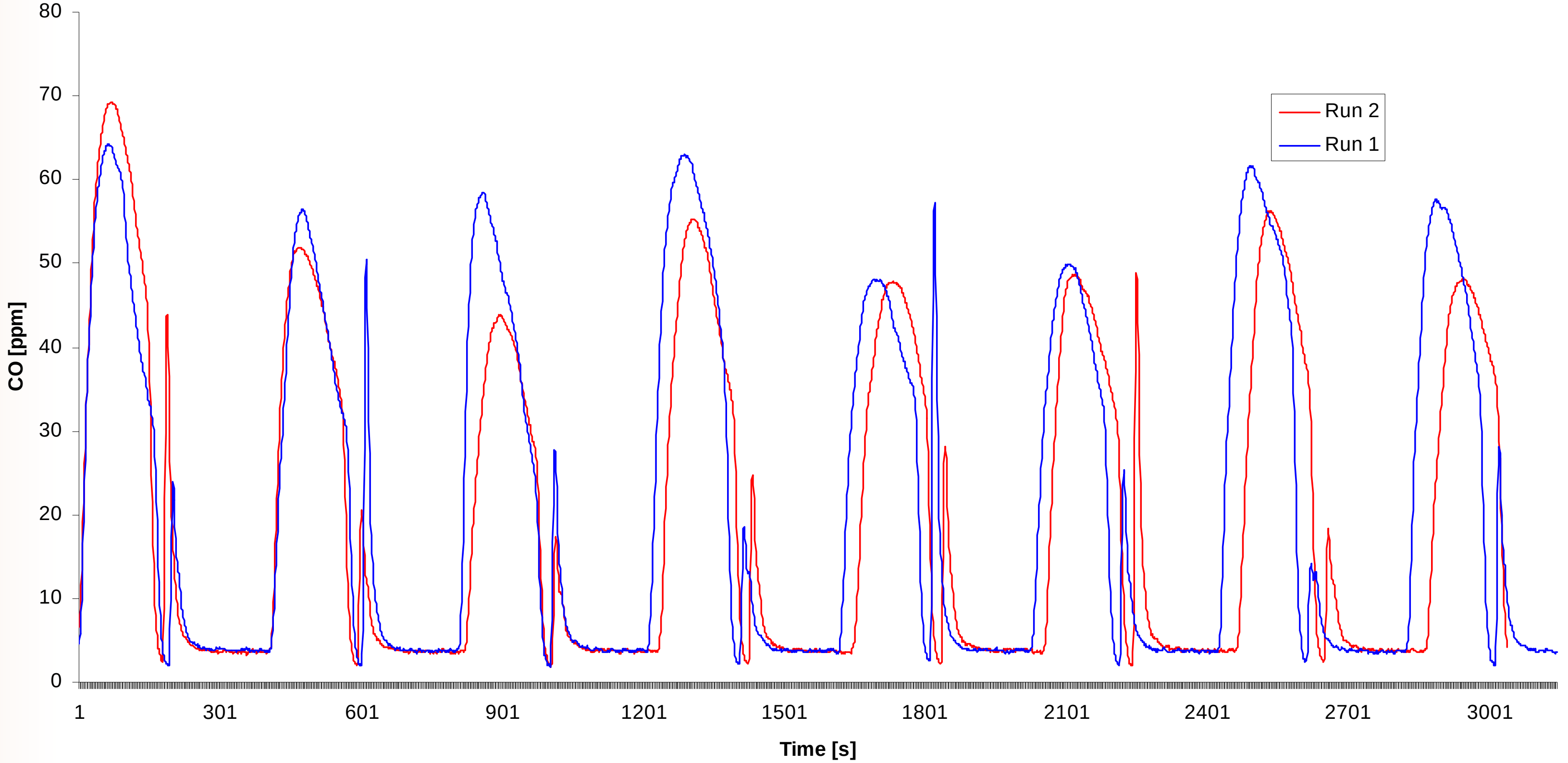
→Experimental artefact due to air volumes trapped in the injection system

→Complementary tests to confirm this hypothesis



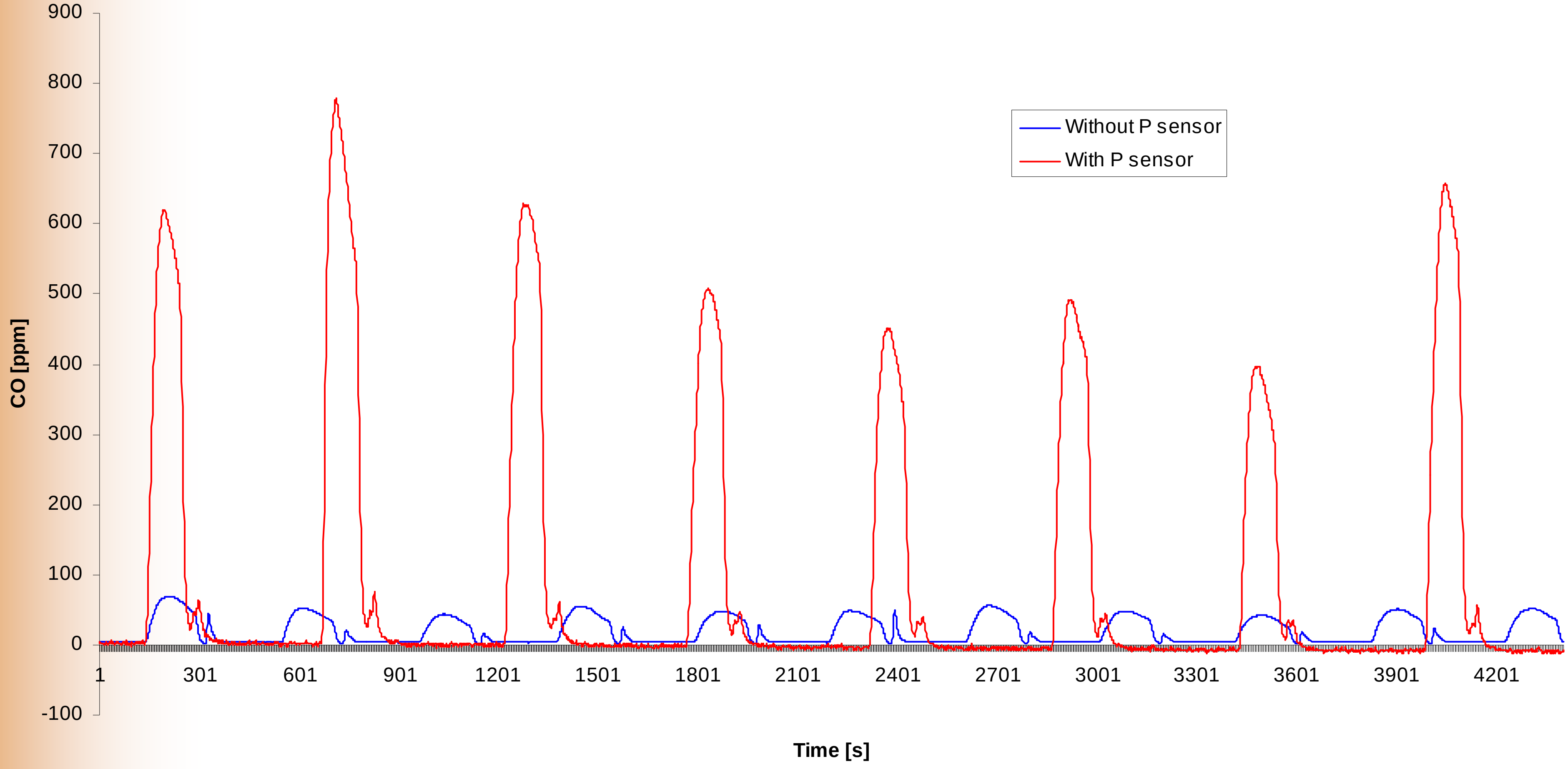
RESULTS

Without P sensors



RESULTS

Comparison of cyclic CO emissions with and without P sensors



CONCLUSION

- 50% of reduction at cold start with the anti-drip system.
- Presence of air in the fuel line led to important CO peaks at shutdown when cycling.
- The anti-drip system did not completely remove the CO peaks.
- Transient CO emissions remain below steady-state emissions European standards with the anti-drip system and if the air problem in the fuel line is avoided.

Acknowledgements

Financial support from **Walloon Ministry for Technology and Energy** (WMTE) Belgium, in the frame of IEA combustion activities, and **VIESSMANN** and **ABIG** manufacturers who have provided (free of charge) the experimental boiler and the burner respectively used in this investigation.

CO emissions of fuel oil boilers in transient and cyclic regimes: performance of burners without and with "anti-drip" systems

(Sub-Task 2.1 H)

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Abstract

CO emission peaks from fuel oil boilers in transient and cyclic regimes have been measured in previous studies. In order to evaluate these emissions, tests have been performed without and with an anti-drip system installed on the burner nozzle. Flue gas composition was measured using classical analysers (CO_2, O_2, CO and NO_x) and CO emissions results were analysed and discussed.

It was found that the anti-drip system reduced cold start CO emissions and did not affect steady-state CO emissions. At first, no reduction was noticed when the burner was cycling. Further investigation indicated that there was an experimental artefact caused by an air volume trapped in the pressure measurement system. When the air was purged and with the anti-drip system, CO peaks when the burner was cycling were strongly reduced and CO emissions remained under European standards limits.

Keywords: Fuel oil boilers, CO emissions, transient regime, cyclic regime, oil after-drip

1. Introduction

Carbon monoxide (CO) is a toxic gas and is considered as a pollutant. It has an indirect radiative effect by increasing concentration of methane and tropospheric ozone [1]. Carbon monoxide is mainly a result of incomplete combustion of hydrocarbon fuels. The most well known cause of this is a lack of air in the combustion zone. There may be other causes, such as inadequate mixing of fuel and air or quenching of the products before the combustion completion [2]. CO may be also produced by dissociation of CO_2 at high temperature.

According to European standards, CO emissions from gas and oil boilers, whose useful output power is lower or equal to 400kW, are limited in steady-state regime to 110mg/kWh [3]. CO emissions in transient regimes are not regulated by European standards.

CO emissions of fuel oil burner-boiler systems in transient and cyclic regimes have been previously measured by Cuevas and Ngendakumana [4]. Their study shows important CO emission peaks at start-ups and shutdowns of the burner. One of the main conclusions of the study is that these peaks are due to an oil after-drip of the burner nozzle.

The aim of this paper is to compare CO emissions in transient and cyclic regimes with and without an “anti-drip” system. First, a literature review has been carried out in order to determine if CO peaks have already been measured by other researchers. Then, tests have been performed with an anti-drip system installed on the burner nozzle.

2. Literature review

Only few studies mention CO measurements in transient and cyclic regimes as CO transient emissions are not regulated on central heating boilers. In general, the main objectives of these researches were not CO measurements so that, when CO peaks were observed at boilers start-ups and shutdowns, they were not explained.

While comparing performances of a boiler fuelled by different mixtures of biodiesels, Krishna [5] observed important CO emissions at the start of the boiler (approx. 250 ppm for a normal fuel and 400 ppm for a B30). At steady-state operation, CO concentration remained at a constant level of 50 ppm. Krishna did not explain the cause of CO start-up emissions.

Lee et al. [6] have compared emissions of different blends of soy methyl ester/diesel in a 30kW boiler. CO peaks were observed at the start of the boiler. No peak was mentioned at shutdown. To evaluate the influence of fuel temperature on emissions, they have measured emissions at different fuel temperatures. For a fuel temperature of 5°C, CO concentration at the start was around 160 ppm while it was only 100 ppm at 15°C. In steady-state operation, CO concentration was around 30 ppm for all tests.

Litzke [7] compared three types of burners: a conventional flame-retention head burner, a high static-pressure head burner and an air-atomising burner. Tests were performed using different oils at both steady-state and transient behaviours under typical cyclic operating conditions. During steady-state operation, when the oil burner stays on continuously, there were negligible flue gas emissions, such as carbon monoxide (CO) and particulates. Higher concentrations were observed only during start-up and shutdown in cyclic operation. CO peaks were more important at start-up than at shutdown for the flame-retention burner and for the high static-pressure head burner. This was the opposite for the air-atomising burner. The most significant effect is observed for highly viscous fuels. Litzke concludes that CO level depends on the burner design and that CO emissions result of incomplete combustion, which could be due to an insufficient air/fuel mixture or a low temperature in the combustion zone.

During tests performed to evaluate fuel sulphur influence on emissions in a domestic boiler, Lee et al. [8] have observed CO peaks at start-ups and shutdowns. It was shown that these were more important at shutdown than at start-up. Moreover, the intensity of the peaks were variable. Lee et al. did not give any explanation to these peaks.

These observations are identical to Cuevas and Ngendakumana’s [4], who have studied CO emissions of a fuel oil boiler not only at start-ups but also during the cyclic regime. They also showed that CO emissions at start-ups increase when the nozzle capacity increases. During the cycling regime, peaks at shutdowns were more important (vary between 350 ppm to 780 ppm) than the ones at start-ups (from 80 ppm to 270 ppm). They conclude that these peaks are due to incomplete combustion of fuel because of a fuel drip from the burner nozzle when it is turned off.

Nozzle manufacturers have already detected these problems when the burner turns off, so that they have developed “anti-drip” systems to avoid fuel drip. Delavan [9] proposes a system called Protek, which can be directly installed on the nozzle instead of the filter. This system provides a sharp cut off of flow without creating a pressure drop. It uses a valve which is calibrated to open and close at certain pressure levels. Hago [10] offers a similar system called Ecovalve. These systems are suitable for simplex nozzles. Fluidics [11] and Riello [12] propose systems which are suitable for spill-return nozzles. It must be pointed out that anti-drip systems are usually not installed on burners by default as transient emissions are not regulated.

3. Experimental

Tests have been performed on the boiler available at the laboratory. The experimental set-up is the same than the one described in Cuevas and Ngendakumana’s study [4] : 370 kW Viessmann boiler equipped with an Abig burner. A schematic representation of the set-up is shown on Figure 1.

The burner was fitted with a spill return nozzle that allowed fuel injection pressure and mass flow rate adjustment. The burner nozzle was characterized by a 45° spray angle and a nominal capacity of 35 kg/h at an injection pressure of 20 bar.

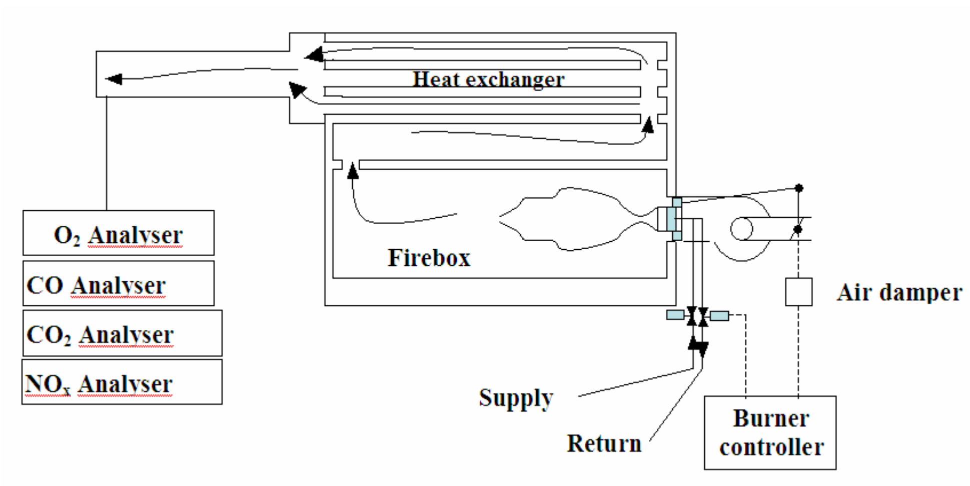

Figure 1: Experimental set-up [4]

An anti-drip system made by Riello was installed on the nozzle. The fixing system had to be modified to fit the nozzle. Figure 2 shows the nozzle coupled to the anti-drip system.

Emissions were measured at the burner start from cold conditions and during the burner cycling. During cyclic regime, the burner worked at part load by fixing hot water temperature to 75°C. The analyser for CO measurement is a CO/NO Hartmann & Braun analyser (Uras 10E). Acquisition time was shortened as much as possible and a measure was taken every second.

Similar test conditions were approached for all tests with and without the anti-drip system. The fuel mass flow rate was adjusted around 0.0105 kg/s (37.8 kg/h) and for each test, excess air was kept constant (3% of oxygen at the stack).

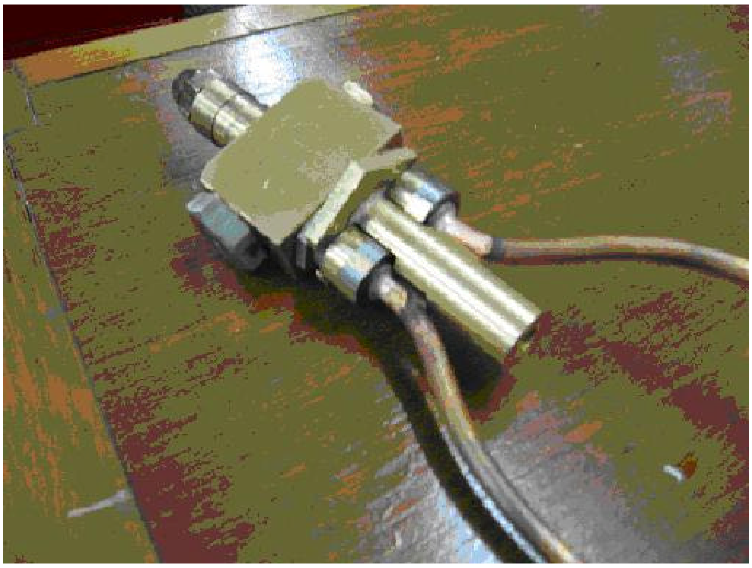

Figure 2: Nozzle coupled with the anti-drip system

4. Results and discussion

4.1. Cold start

Table 1 gives the test conditions for cold start measurements.

Table 1: Test conditions

	P_{atm} [mbar]	t_{amb} [°C]	RH_{amb} [%]	P_{ret} [bar]	P_{inj} [bar]	$\dot{M}_{fuel}$ [kg/s]	$\dot{M}_{water}$ [kg/s]	O_2 [%]
Without anti-drip system	985	27,7	53,0	15,2	19,0	0.01052	1,39	3,0
With anti-drip system	963	21,8	46.8	15,3	19,8	0.01046	1,38	3,0

CO emissions at cold start are plotted on Figure 3. The CO peak at the burner start-up is reduced from 71 ppm without the anti-drip system to 35 ppm with the system installed. This reduction could be explained by a better vaporization of the fuel at the start with the anti-drip system. This system tends to eliminate the incomplete fuel atomisation at the start when the pump is not at full pressure as the system valve opens at approximately 10 bar.

European standards impose CO emissions lower than 110 mg/kWh, which corresponds to 86 ppm for the test boiler (400 kW at 3% O_2). Thus, CO cold start emissions are lower than the limit imposed by standards. During steady-state regime, CO emissions do not exceed 6 ppm. The anti-drip system does not influence steady-state emissions.

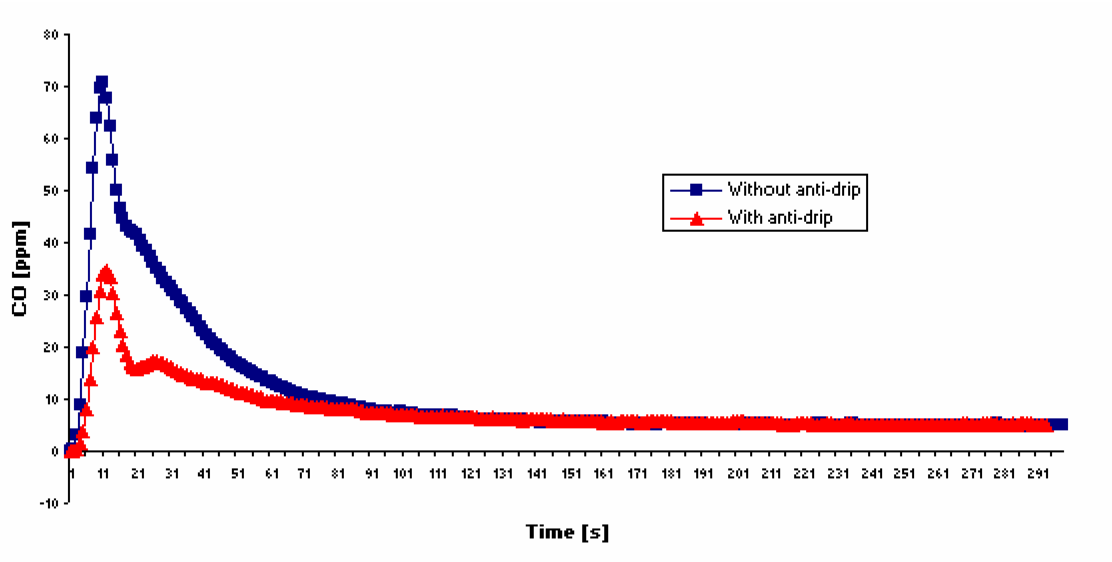

Figure 3: CO emissions at cold start

4.2. Cyclic regime tests

The tests conditions were the same than the ones in Table 1. Figure 4 shows the comparison between CO emissions before and after the anti-drip system installation. It can be seen that there was no significative difference between the two configurations during cyclic regime. The maximum value for the shutdown CO peak observed on the figure is 800 ppm with the anti-drip system whereas it is 780 ppm without the anti-drip system.

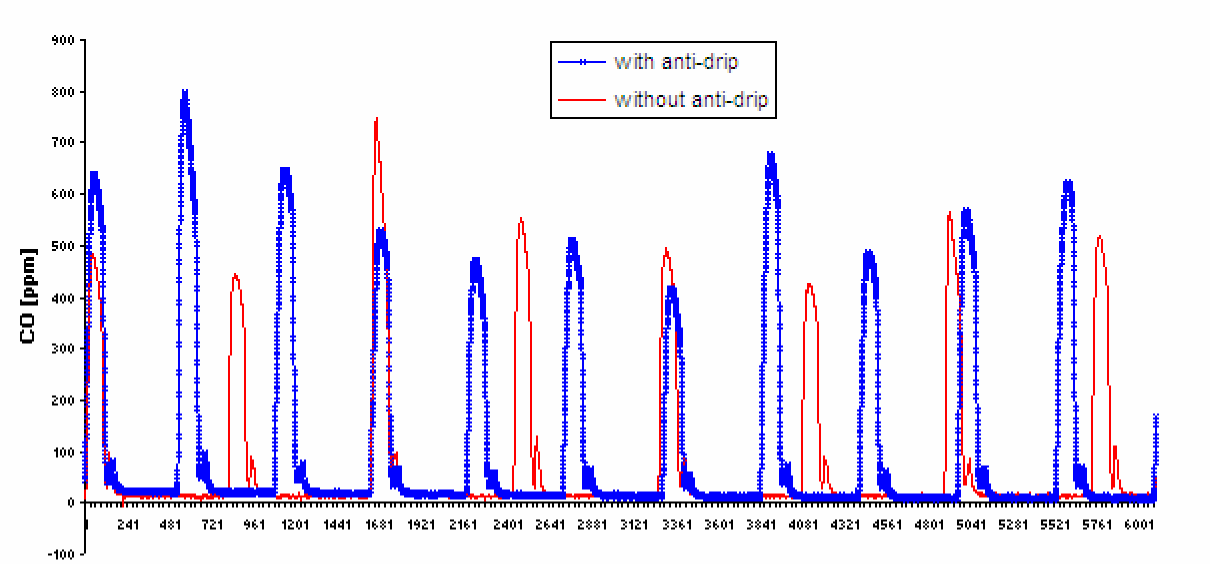

Figure 4 : CO emissions with and without anti-drip system during cyclic regime

The main function of the anti-drip system is to reduce the fuel drip at the burner closure. As CO peaks are still important, the experimental set-up has been checked to determine the possible cause of these peaks.

An experimental artefact has been detected. Pressure sensors had been installed on the fuel line to measure injection and return pressures (Figure 5). Residual amounts of air get trapped in the pressure sensors before the solenoid valves. These air volumes are compressed when the pressure is established by the pump and distended when the burner shuts down. The distending air expels the remaining fuel into the combustion chamber. To solve this problem, trapped air should be removed from the fuel line. In that configuration, it was not possible to purge the air from pressure sensors. So, they have been removed for further tests.

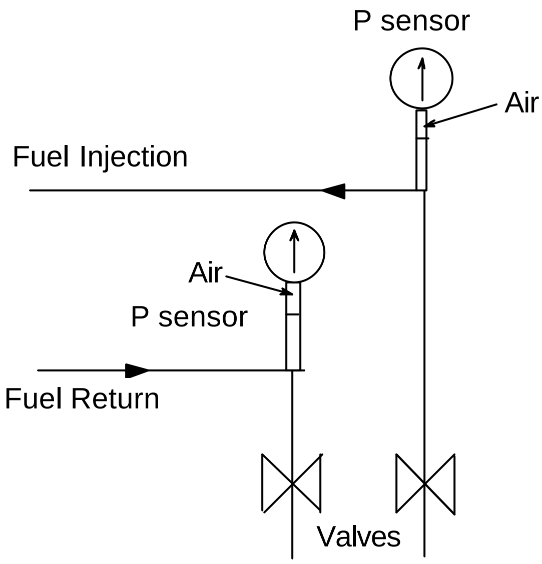

Figure 5: Fuel injection and return pressure measurements

Two batches of complementary tests without pressure sensors have been performed to confirm this hypothesis. The adjustment of the burner stayed identical to previous tests. The results are shown on Figure 6. The level of CO peaks at shutdown are reduced to a maximum of 70 ppm for both runs, which is lower than standards limits. This significant decrease can be seen if CO emissions with and without the pressure sensors are compared (Figure 7).

As it was not possible to take back the anti-drip system to get the initial fuel injection system back, CO cyclic emissions without the anti-drip and without pressure sensors could not be compared to the emissions with the anti-drip and without pressure sensors. Thus, the effect of the only anti-drip system could not be quantified for CO emissions at the burner shutdown.

CO peaks during cyclic regimes are not completely removed with the anti-drip system. When the burner is turned on, the valve inside the nozzle opens before the nominal injection pressure (approximately 10 bar). Thus, fuel atomisation is not optimal and incomplete combustion still occurs. When the burner is turned off, the valve of the anti-drip system does not close directly and some fuel still drips into the combustion chamber and incomplete combustion occurs. To reduce the CO peaks more, another anti-drip system characterized by operating pressures closer to the nozzle nominal injection pressure should be installed .

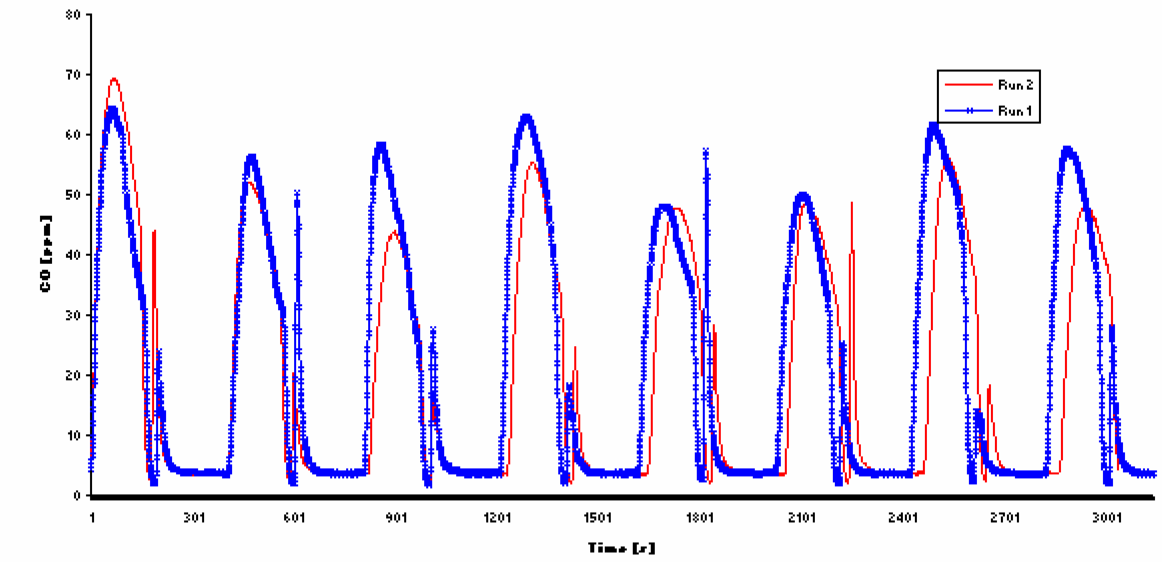

Figure 6: Cyclic CO emissions without the pressure sensors with the anti-drip system

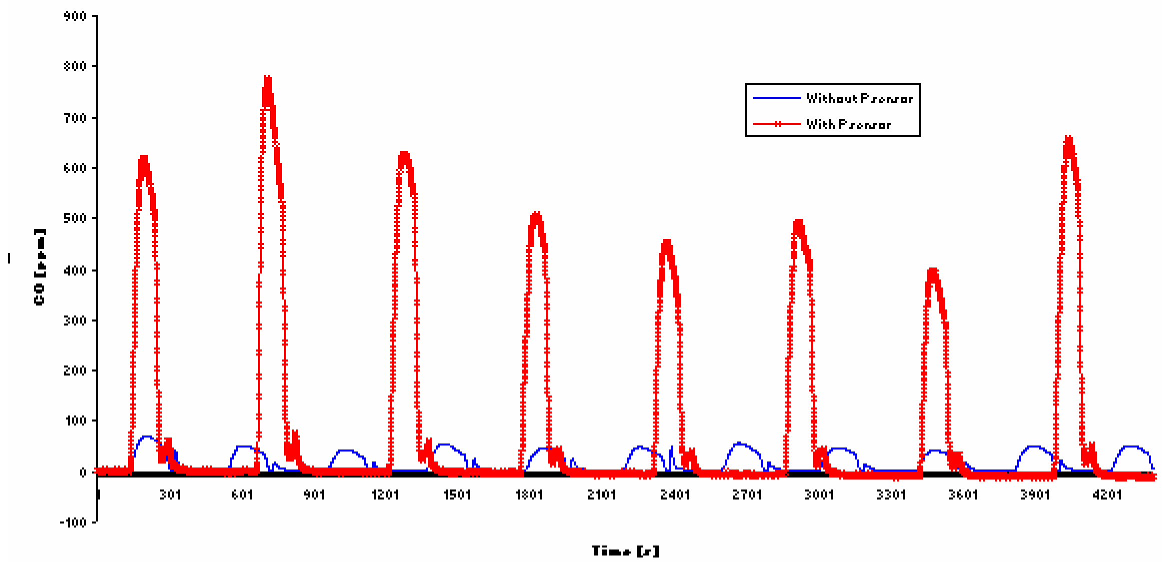

Figure 7: Comparison of CO emissions with and without the pressure sensors installed (with the anti-drip on the burner nozzle)

5. Conclusion

The aim of this study was to measure CO emissions at cold start and during cyclic regimes for a fuel oil burner equipped with an anti-drip system.

Cold start CO emissions were compared in the different configurations (with and without the anti-drip system). A 50% reduction was shown at cold start. It could be explained by a better fuel vaporization at the start because the optimal injection pressure is more rapidly reached.

Complementary tests showed an experimental artefact coming from air trapped in the injection pressure measurement system, which led to important CO emissions at shutdown. In fact, when the air was purged from the fuel line, CO emissions were lowered from 700 ppm to 70 ppm.

The only anti-drip effect could not be quantified because it was not possible to get exactly the initial system back: the nozzle without the anti-drip system and without pressure sensors. However, it can be concluded that CO emissions with the anti-drip system remained lower than European limits (86 ppm for the test boiler) when the burner was cycling.

In short, the anti-drip system is efficient to reduce CO emissions at the start of the burner. To avoid CO emissions when cycling, it is necessary to purge the fuel line from residual air and install an anti-drip system working in the right operating pressure range. It is also better not to undersize heating systems so that the burner can work continuously.

Nomenclature

$\dot{M}$: mass flow rate [kg/s]
P : pressure [bar]
t : temperature [°C]
RH : relative humidity [%]

Subscripts
atm : atmospheric
amb : ambient
ret : return
inj : injection

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Acknowledgment

The authors acknowledge the Walloon Ministry for Technology and Energy (WMTE)-Belgium for the financial support of this work. They are also grateful to VIESSMANN and ABIG manufacturers for their cooperation in providing free of charge respectively the experimental boiler and the burner used in this investigation.

POLYTECH.MONS

Thermal Engineering & Combustion Laboratory (FPMs)
Thermodynamics Laboratory (ULg)

Feasibility study of the diluted combustion in a semi-industrial boiler at low temperatures

Work in progress

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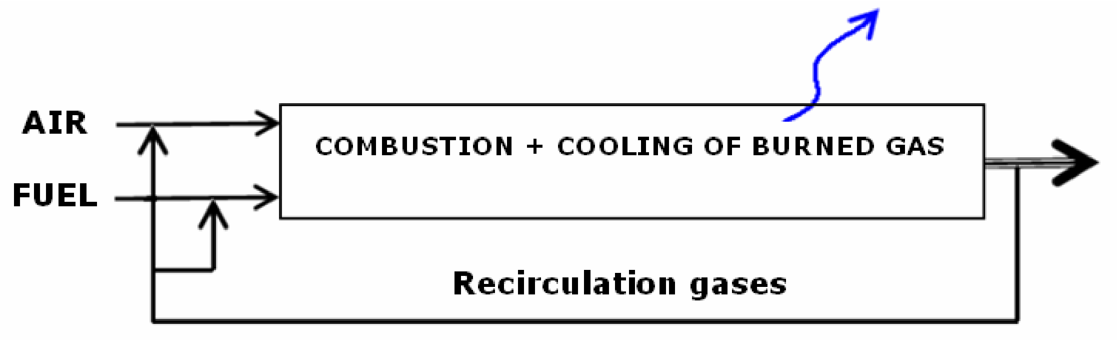
2. Experimental setup

3. Numerical study

4. Conclusions

1. Introduction



$$K_V = \frac{\dot{M}_{recirculation\ \ gas}}{\dot{M}_{fuel} + \dot{M}_{Air}}$$

• Diluted combustion has been acknowledge as one of the most effective combustion technologies to meet both the targets of high process efficiency and low pollutant emissions.

• 2 fundamental requirements :

- The process temperature must be above the mixture autoignition temperature (for methane-air : 500°C theoretically, 800°C practically)

- The recirculation ratio (**K_v**) (i.e., the ratio between the recirculation gases mass and the incoming mixture mass) must be higher than a threshold

1. Introduction

• High level of dilution with flue gases ➔ slower reaction in a larger volume ➔ temperature field more homogeneous and no peak values ➔ reduction of NO_X

• Principally used in the high temperature process furnaces but interest at low temperatures (even if less affected by NOx emission)

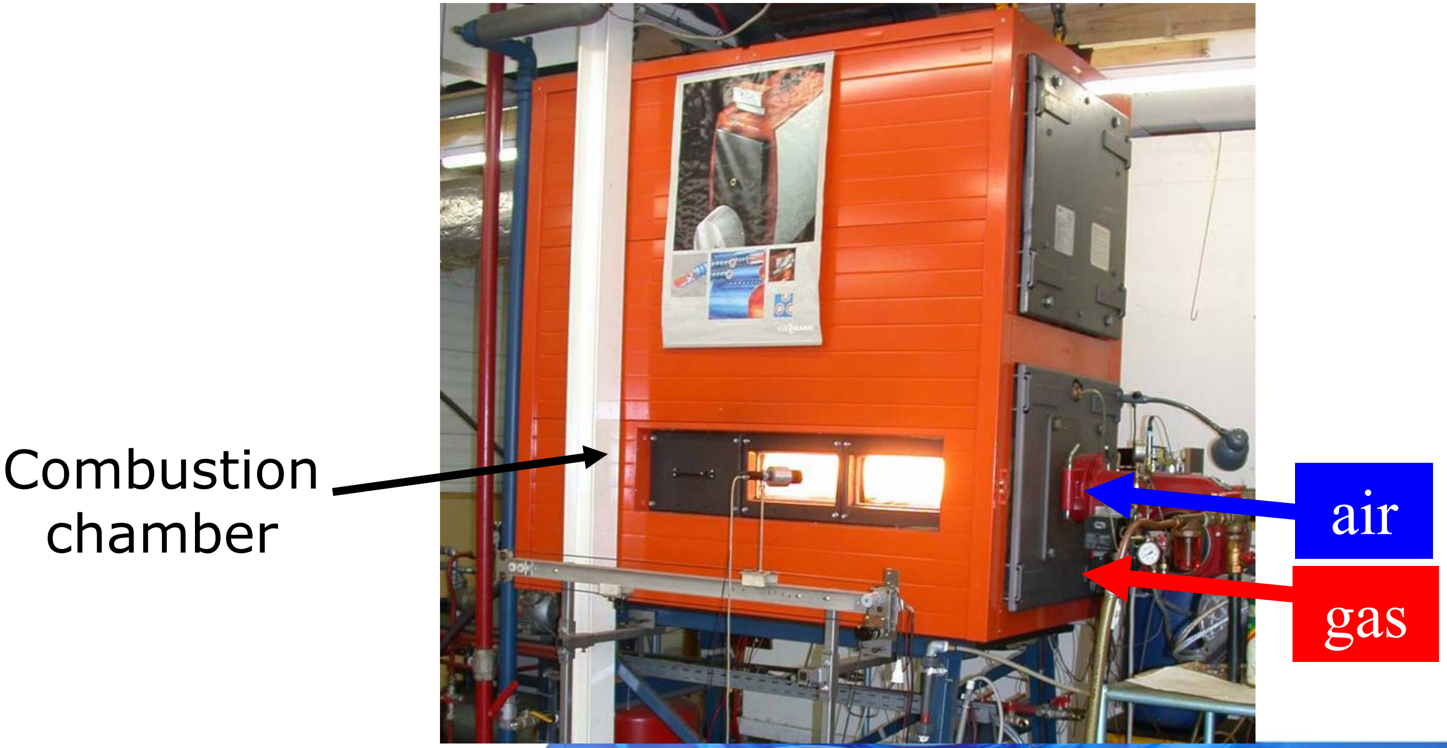
• Combustion chamber furnaces ≠ combustion chamber boiler in:

- Geometrical and thermal confinement
- Water-cooled ➔ high wall heat losses

➔ Feasibility study of the diluted combustion in a semi-industrial boiler at low temperatures (without preheating the combustion air)

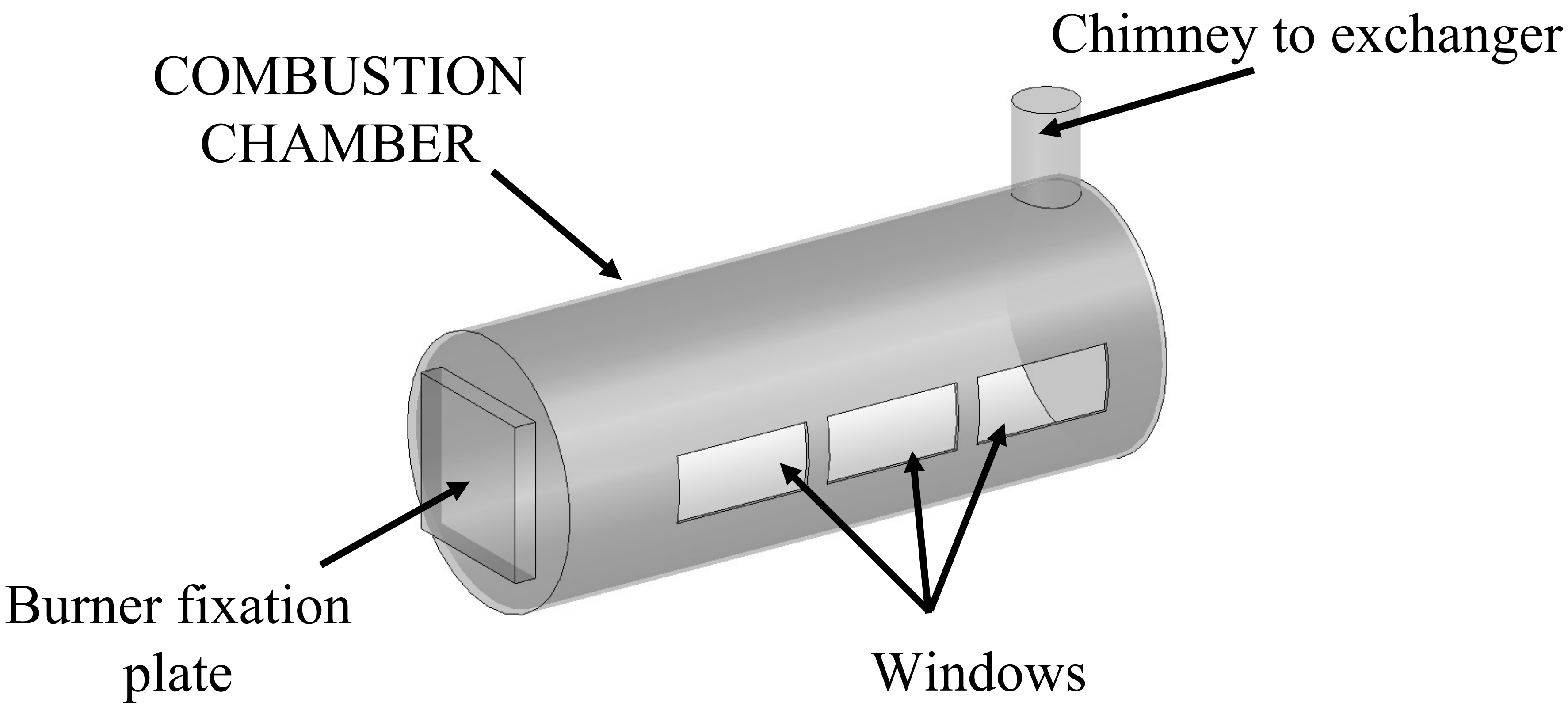
2. Experimental setup

- Viessmann Paromat-Triplex boiler (Thermodynamics Laboratory ULg)
- To get diluted combustion : air and gas injection separeted
  - Central injection for the air
  - Gas injection (under the air injection)



2. Experimental setup

- Boiler = combustion chamber (water-cooled) + exchanger



2. Experimental setup

• Use of a gas burner jet in place of the fuel-oil burner : Thermjet 100 Eclipse burner (293 kW)

• Classical working mode to heat up the combustion chamber : Starting in flamme mode (classical combustion) till the max capacity of the burner with air excess of 15%

• Then, the mass flow of air injected in the burner is increased and the gas corresponding to this increasing is injected by direct injection

• Finally, the gas injected by the burner will pass to the gas injector.

• 370 kW in the combustion chamber by direct injection

3. Numerical study

Recall

• Last year a preliminary CFD study ➔ selection of a jet burner and a secondary gas injector able to generate good condition for the diluted combustion : sufficient dilution and temperature of the air and gas jets before they meet in the combustion chamber of the boiler

Numerical study

• investigation on the boundary conditions and the parameters of the combustion model

• optimization of the position, the incline and the diameter of the gas injector

3. Numerical study

Half of the combustion chamber of the boiler and the burner modelled (for the burner only the quarl) using FLUENT 6.2 ®, by means of 450,000 hexahedral cells grid.

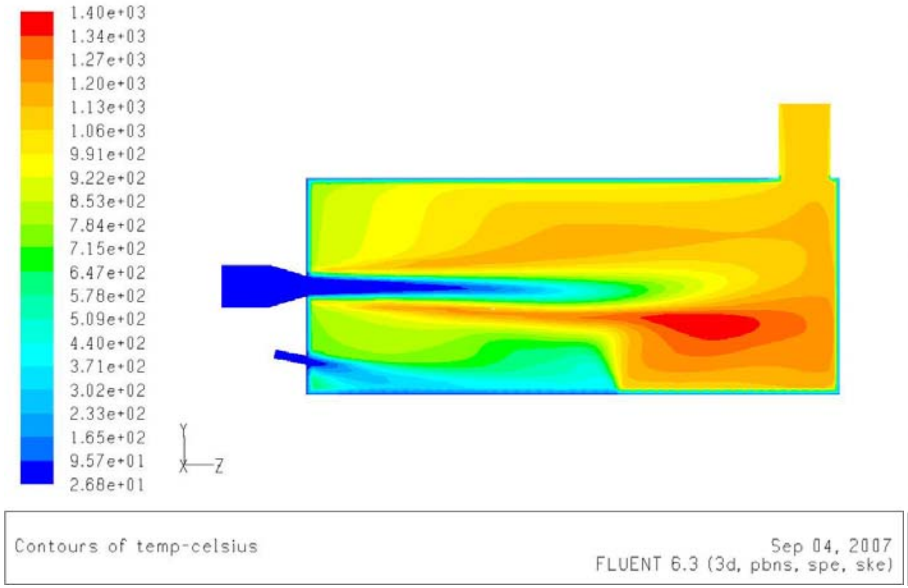
Standard models implemented in FLUENT are used:

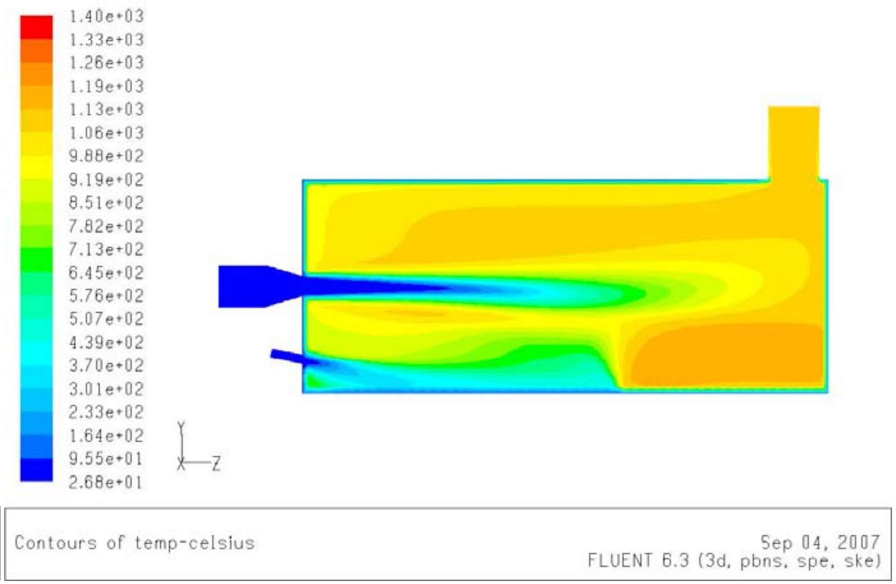
- Turbulence modelled using standard k-ε model, with standard wall functions.
- Radiative heat transfer modelled with the discrete ordinates approach where absorption coefficient is computed with the weighted sum of gray gases assumption.
- A transport equation is solved for each species involved in the combustion reaction mechanism. Fluent proposes several models which differ by the way they compute the average reaction rate. We use the "Eddy-Dissipation Model" which assumes that the reaction rates are fully controlled by turbulent mixing parameters (A, B).

3. Numerical study

Model EDM

Standard mixing parameters proposed in Fluent modified ➔ the modification of the "Eddy-Dissipation Model" mixing parameters better reproduce the furnace temperature profiles measured in pilot-scale furnaces working in diluted combustion.





Standard parameters A=4, B=0.5

Modified parameters A=0.6 , B=10 e+20

3. Numerical study

Boundary conditions

combustion chamber water-cooled ➔ 2 possibilities to reproduce the wall flux loss of the chamber:

- to impose a uniform temperature on the wall (Dirichlet)

- to impose a heat transfer coefficient by fixing the temperature of the cooling water (Fourrier).

Solution

Fourrier

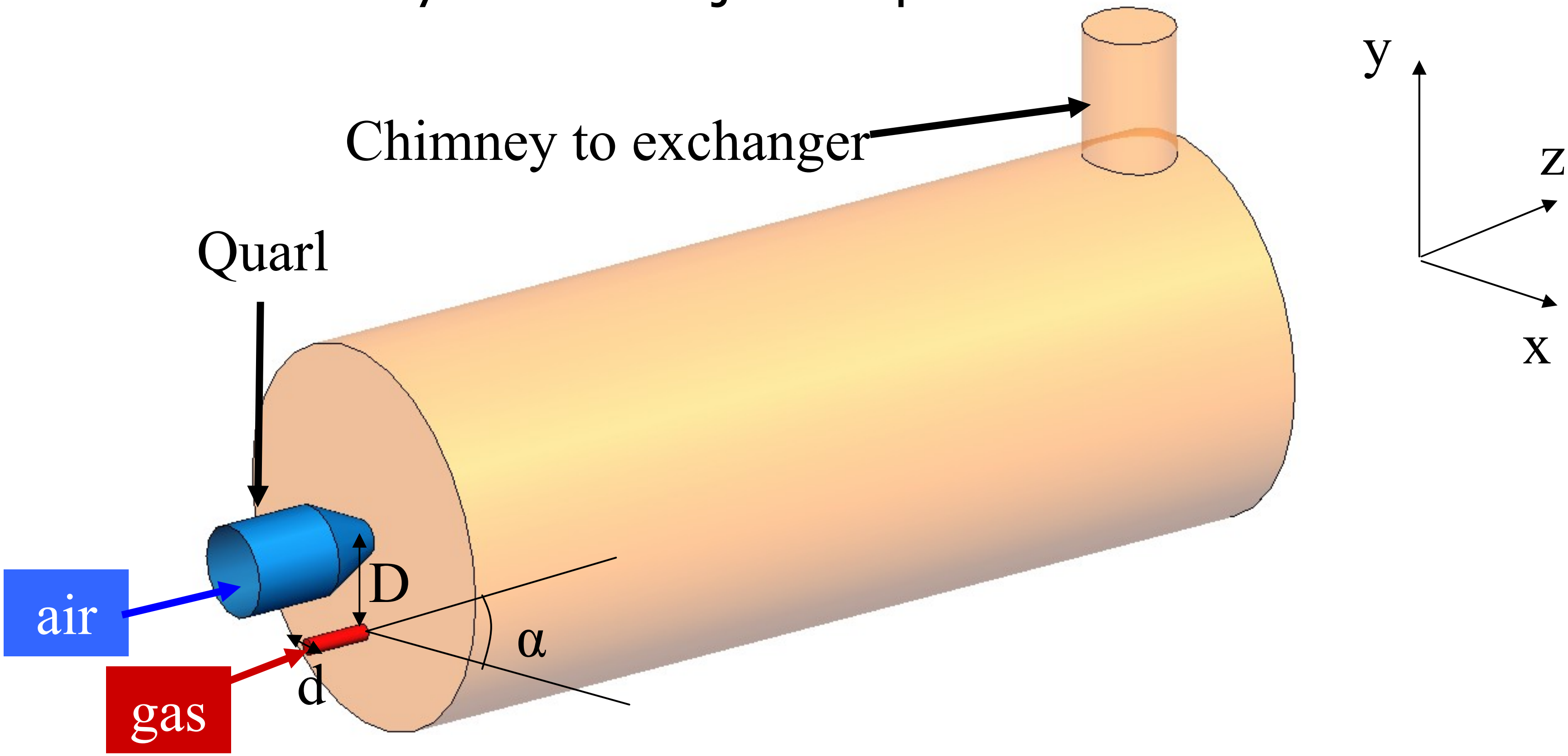
K = 4000 (W/K m²)

Tf = 350 K

		Gas injector diameter (24 mm)	
		T max (°C)	Max heat release (kW/m³)
Wall T imposed			
350 K	76,85 °C	1179,40	47404,46
400 K	126,85 °C	1187,17	47632,42
450 K	176,85 °C	1195,40	47835,84
Wall convection coefficient (W/K m²)			
1000		1188,76	47542,36
4000		1181,75	47439,74

3. Numerical study

- Parametric study on the injector position and incline

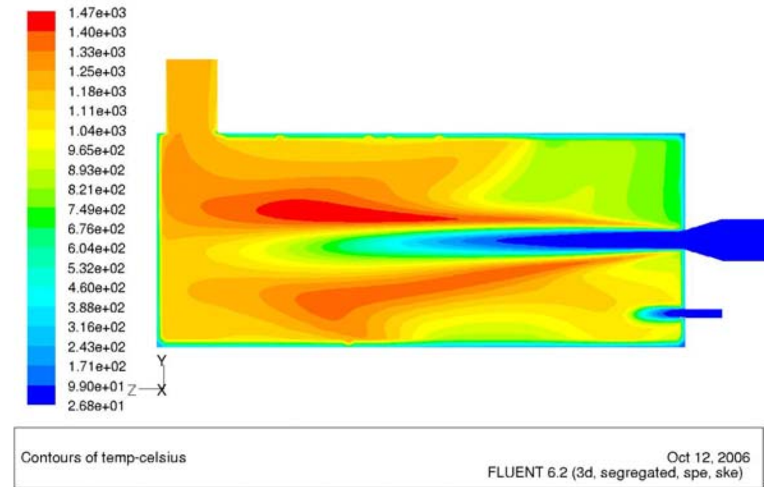


3 parameters for the direct injection :
- length D (max~150mm)
- angle α
- diameter d

3. Étude numérique

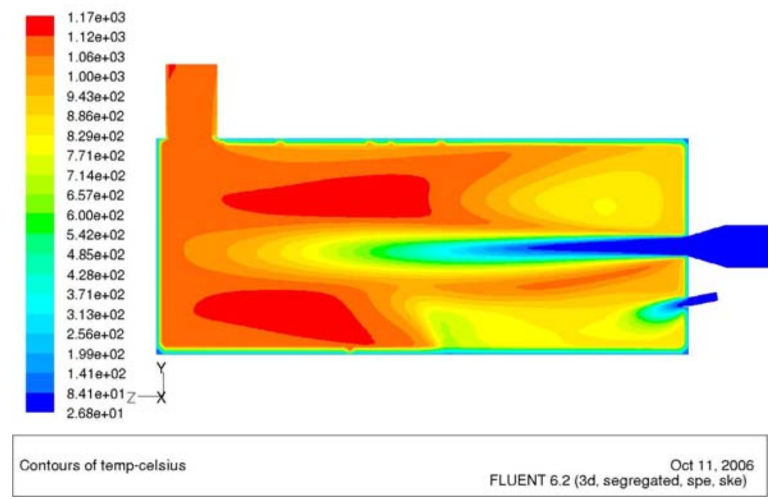
α = 0°

Temperature field not uniform enough (for diluted combustion) and the peak of temperature too high



α = 11°

Temperature field more uniform and the peak of temperature lower than for α = 0°



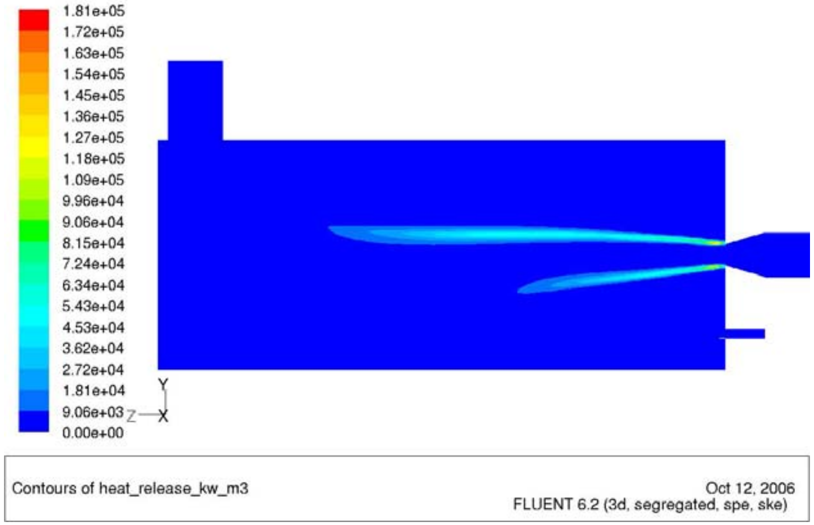
3. Étude numérique

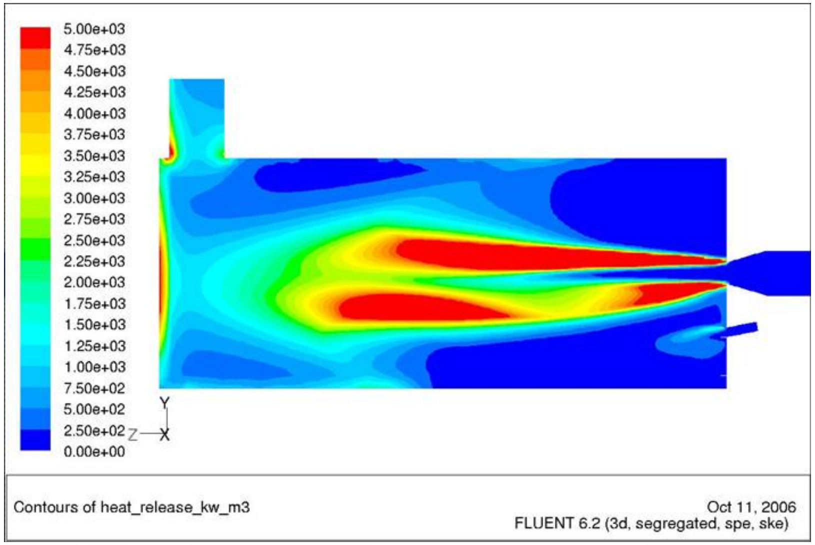
$\alpha = 0°$

Heat release field not spread enough and the peak value is too high for diluted combustion (risk of flame anchor)

$\alpha = 11°$

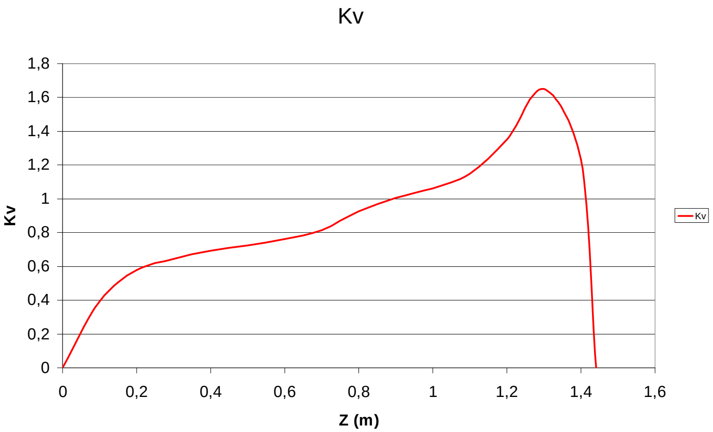
Heat release more spread and peak value lower than for $\alpha = 0°$

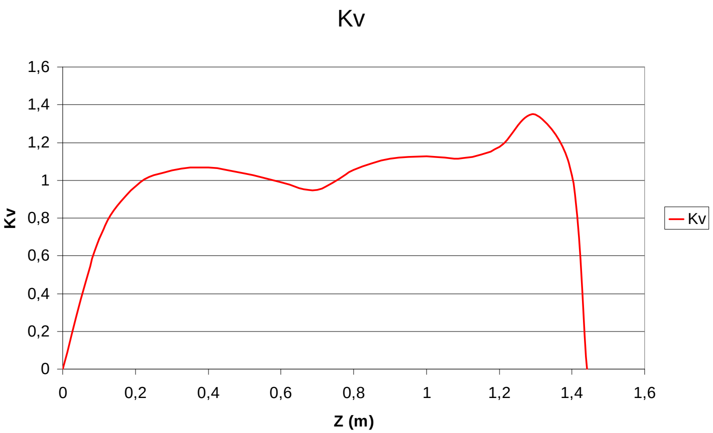




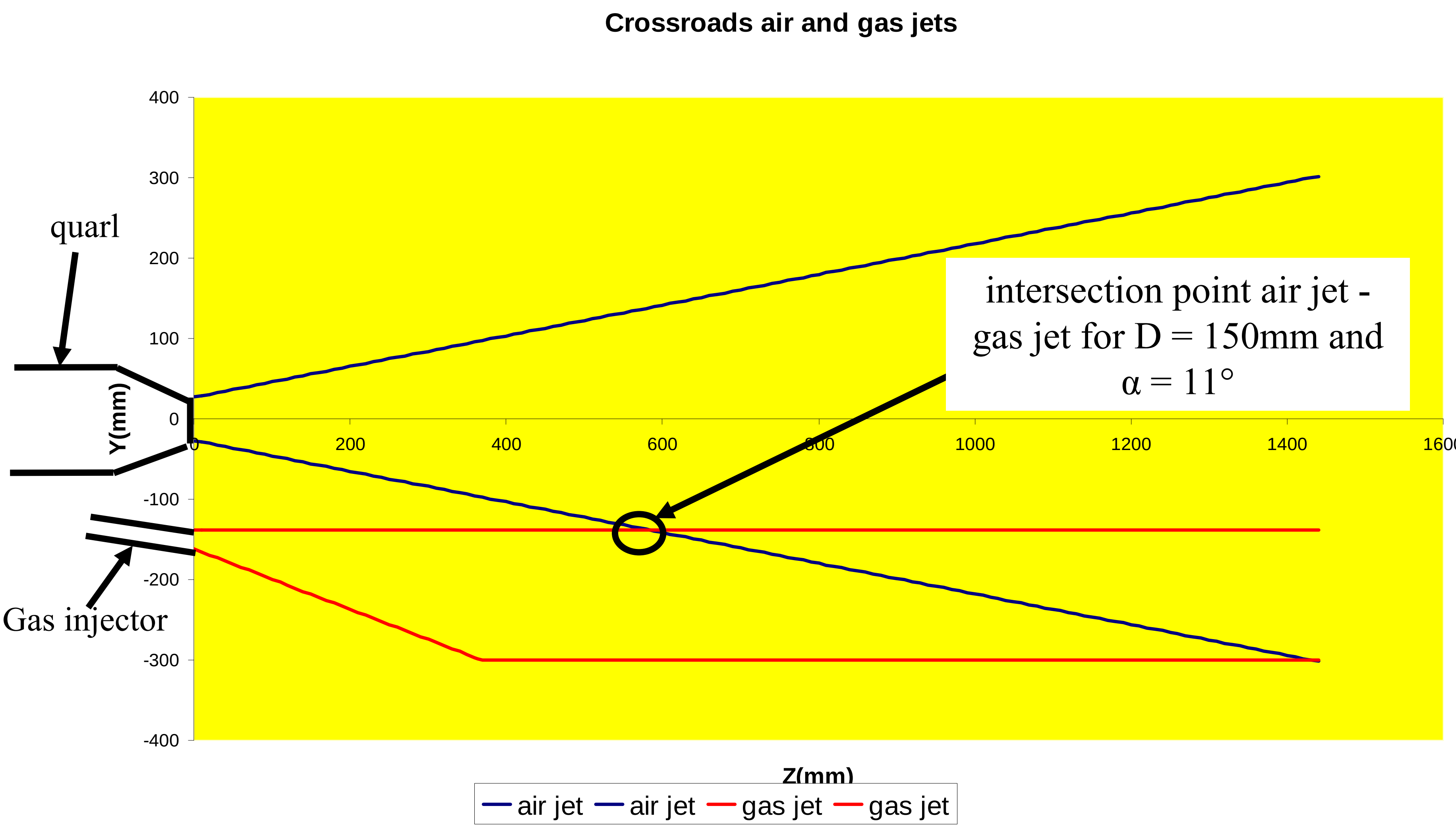
3. Étude numérique

- α = 0°
  - At the intersection between the air jet and the gas jet (Z=0,6 m), the value of Kv=0,7
- α = 11°
  - The value of Kv is higher than for α = 0° at the intersection of the two jets : Kv=1 at Z=0,6





3. Numerical study

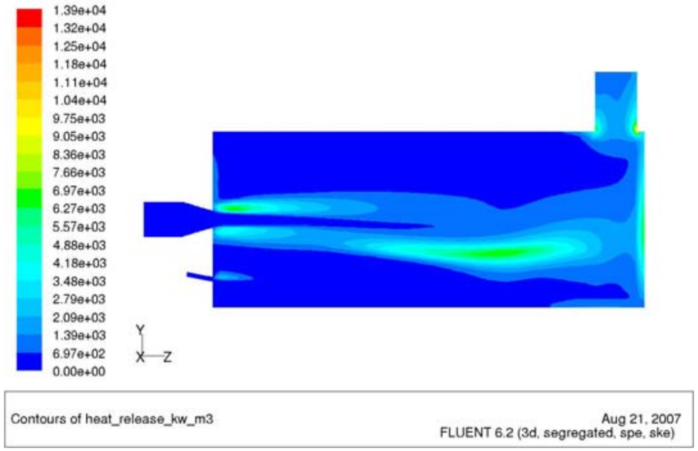


3. Étude numérique

- Parametric study on the gas injector diameter d

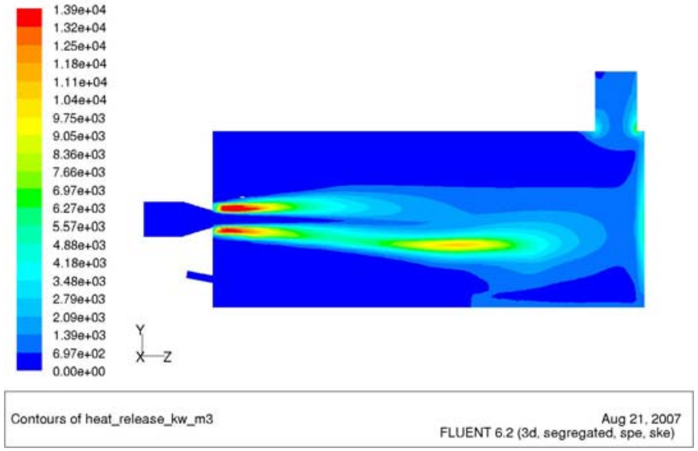
d = 16 mm

Heat release less intense and level of unburned gas quite high

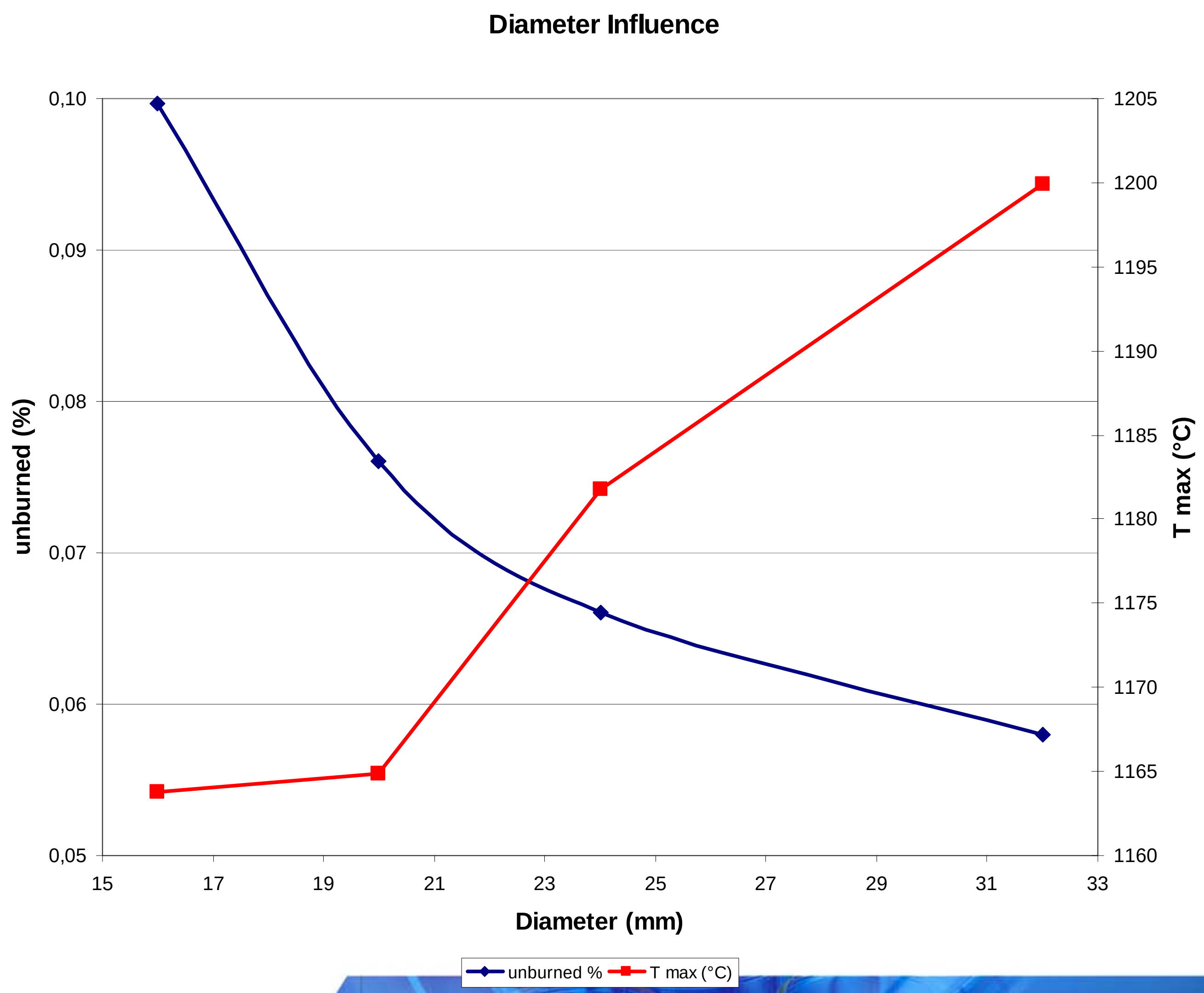


d = 24 mm

Peak of heat release 2 X higher and level of unburned gas lower than for d = 16 mm



3. Étude numérique



4. Conclusions

Done

- Preliminary CFD study has been performed
- Parametrical study : D = 150 mm, α = 11° and d = 24 mm
- Order the selected material
- Burner installation

Following of the work

- Experimental study in classical combustion mode
- Numerical simulation with adjustment of the parameters
- Injector installation
- Feasibility study in diluted combustion mode

I wish to thank the Walloon Government for continuous financial support of research in the framework of the IEA

RÉGION WALLONNE

THANKS FOR YOUR ATTENTION

Feasibility study of the diluted combustion in a semi-industrial boiler at low temperatures

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(Sub - Tasks 2.1 H & 2.1 I)

Abstract

The aim of this work is to assess the technical feasibility and highlight the specific problem of application of diluted (or mild) combustion in a medium scale boiler by direct gas injection. The main difficulty is due to the high geometrical confinement and heat losses of a typical boiler combustion chamber, which prevents from getting the minimum level of reactants dilution and temperature needed to reach mild combustion regime. Last year, a preliminary CFD study was performed to select a jet burner and secondary gas injector able to generate sufficient dilution and temperature of the air and gas jets before they meet in the combustion chamber of the boiler [1]. In this work, the optimization of the position and the diameter of the gas injector has been performed. Finally, the boundary conditions and the parameters of the combustion model have been investigated.

Introduction

Diluted combustion technology, also called "flameless oxidation", allow high process thermal efficiency with low pollutant emissions. This technology is today mainly used in high temperature process furnaces, to lower mainly the NO_x emissions. The principle consists in providing a high level of dilution of the reactants with flue gases before combustion reaction occurs, to get a slower reaction in a much larger volume than in classical combustion [3]. The resulting lower local heat release leads to a more homogeneous temperature field in the furnace, without peak values responsible of high thermal NO_x formation [2,3,4,5]. However, the increase in furnace temperature uniformity can also have an interest for lower temperature processes, less affected by NO_x emission level restrictions. In this work, the application of the principles of diluted combustion is considered in the combustion chamber of a semi-industrial boiler, which differs from a furnace mainly in geometrical and thermal confinement.

Safe diluted combustion conditions request that the reactant mixture temperature remains above its self-ignition level everywhere in the furnace. In the case of a boiler, the combustion chamber is generally water-cooled, and the very high wall heat losses are therefore not compatible with the self-ignition temperature requirement.That is why a preliminary study is necessary to assess the technical feasibility of the diluted combustion in a semi industrial boiler at low temperature (without preheating of the combustion air).

In this work, diluted combustion is performed by using direct injection of the reactants into the combustion chamber of a boiler. A jet burner is first used for preheating the combustion chamber with classical combustion. When the mixture self-ignition temperature is reached, combustion air and gas are then injected separately into the combustion chamber.

Combustion in diluted mode in the boiler is first studied by CFD modelling, which will be validated by an experimental study on a semi-industrial boiler test bench.

Experimental setup

The test bench consists in a boiler from Viessmann (Figure 1) whose nominal output power is about 370 kW, and which is located at the Thermodynamics Laboratory of University of Liège (ULg, Belgium). This boiler was initially working with a fuel-oil burner. Optical access is available through quartz windows placed in one side wall (see Figure 1).

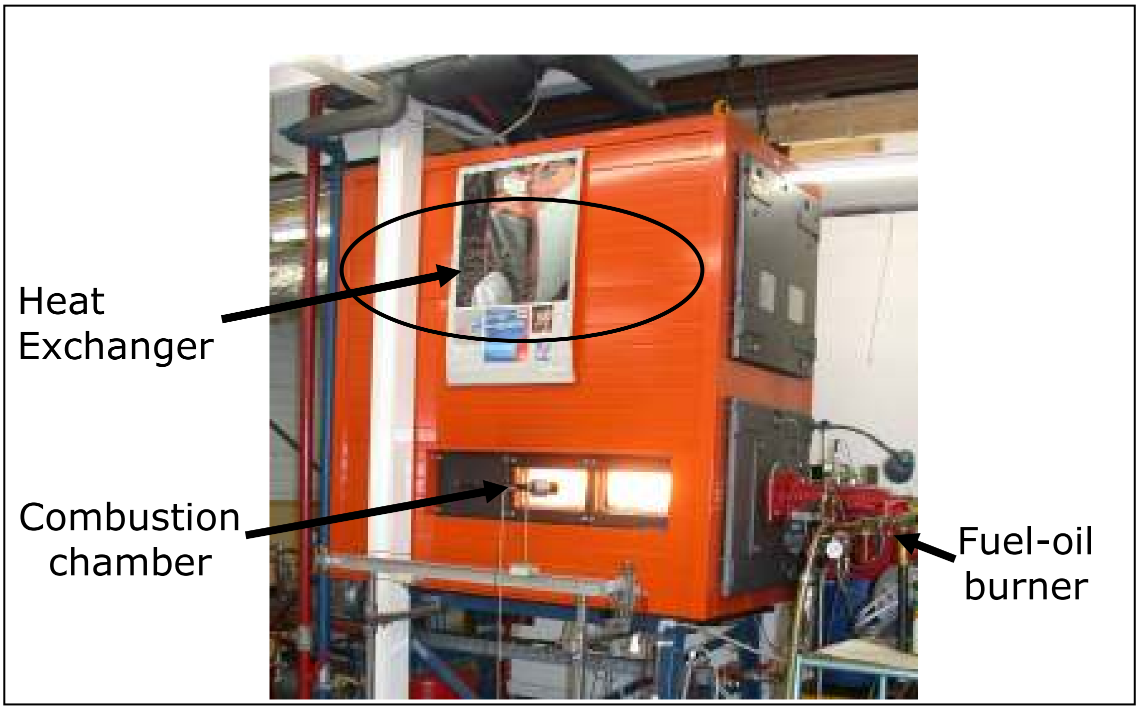

Figure 1

In our study, the fuel-oil burner is replaced by direct injection of air and natural gas (Figure 2). Air is injected horizontally (in the boiler axis) and the position of the gas injection is under the air injection [1].

Figure 2

As it can be seen on Figure 1, the boiler can be subdivided in 3 parts: the combustion chamber, the heat exchanger and the chimney between them. In our simulation, only the combustion chamber and the chimney are modelled (Figure 3) using Fluent 6.2.

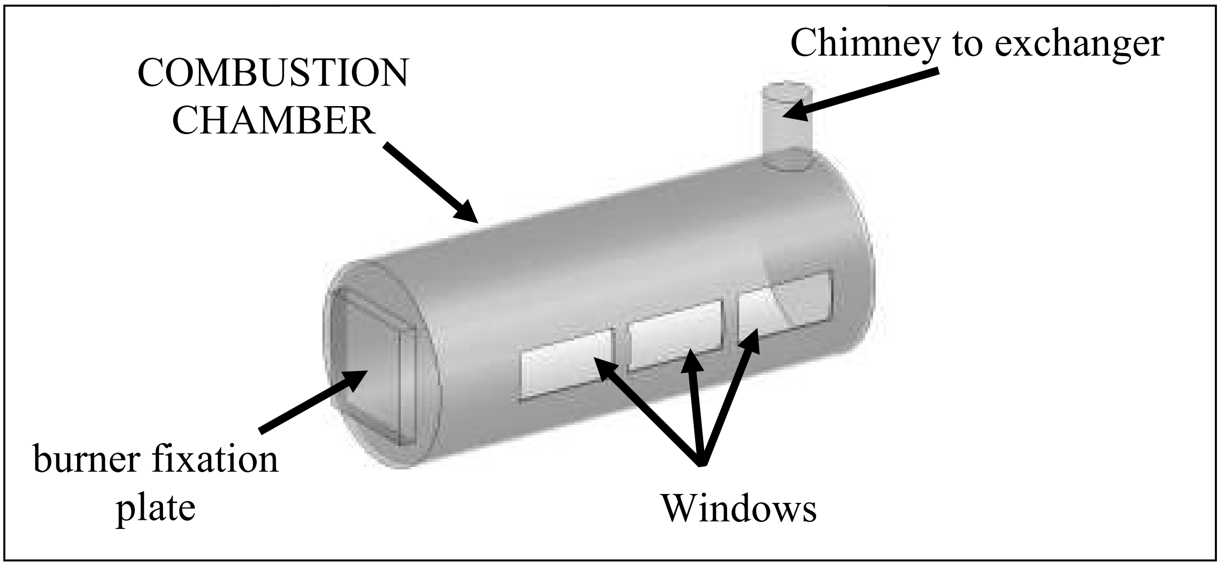

Figure 3

The chosen solution is the use of a natural gas jet burner (Eclipse Thermjet 100 [1]) in place of the fuel-oil burner to heat up the combustion chamber. To start the system, the jet burner will be used in a classical flame combustion mode, in order to heat up the combustion chamber above the mixture self-ignition temperature level. When the temperature is high enough, we pass gradually from a classical combustion operation to a diluted combustion mode, by using the jet burner as the air injector and adding a second separated injector for the gas. The gas injector has to be under the burner and the diameter has to be optimized to get the better condition for diluted combustion.

Numerical study

The Thermjet 100 (293 kW) Eclipse burner has been modelled with the combustion chamber of the boiler using FLUENT 6.2 ®, by means of 450,000 hexahedral cells grid.

Standard models implemented in FLUENT are used:

- Turbulence is modelled using standard k-ε model, with standard wall functions.
- Radiative heat transfer is modelled with the discrete ordinates approach where absorption coefficient is computed with the weighted sum of gray gases assumption.
- A transport equations is solved for each species involved in the combustion reaction mechanism. Fluent proposes several models which differ by the way they compute the average reaction rate. We use the “Eddy-Dissipation Model” which assumes that the reaction rates are fully controlled by turbulent mixing parameters.

Standard mixing parameters proposed in Fluent have been modified according to [2,6]. The work of Lupant [2] and Malfa [6] has shown that the modification of the “Eddy-Dissipation Model” mixing parameters better reproduce the furnace temperature profiles measured in pilot-scale furnaces working in diluted combustion.

Boundary conditions

For two different gas injection diameter (d = 16 mm and d = 24 mm), the boundary condition at the combustion chamber wall has been varied. In fact, the combustion chamber is water-cooled and this cooling has to be reproduced. We have two possibilities to reproduce the wall flux loss of the chamber: to impose a uniform temperature on the wall (Dirichlet) or to impose a heat transfer coefficient by fixing the temperature of the cooling water (Fourrier) (Table 1).
After a numerical study for these two conditions and two different diameter, we opt for a Fourrier boundary condition for the wall with a heat transfer coefficient K of 4000 (W/K m²) and a water temperature of 350 K.

Nominal Power		**370 kW**	
		Gas Injector Diameter (24 mm)	
Wall T imposed		**T max (°C)**	**Max heat release (kW/m³)**
(K)	**(°C)**		
350	76,85	1179,40	47404,46
400	126,85	1187,17	47632,42
450	176,85	1195,40	47835,84
Wall Convection Coefficient (W/K m²)			
1000		1188,76	47542,36
4000		1181,75	47439,74
		Gas Injector Diameter (16 mm)	
Wall T imposed		**T max (°C)**	**Max heat release (kW/m³)**
(K)	**(°C)**		
350	76,85	1161,40	13848,96
400	126,85	1169,06	14224,36
450	176,85	1177,19	14592,39
Wall Convection Coefficient (W/K m²)			
1000		1171,06	14164,68
4000		1163,76	13930,51

Table 1

Parametric study

Last year, the first part of the numerical study has been realized. This part has consisted of the study of the classical combustion and of the determination of the gas injector position under the quarl. In this work, a parametric study consisting of the optimisation of the gas injector position and diameter has been performed.

In fact, the position of the gas injection is defined by two parameters: the distance (D) between the injector and the quarl and the angle (α) of the injector axis in the plane YZ (Figure 4).

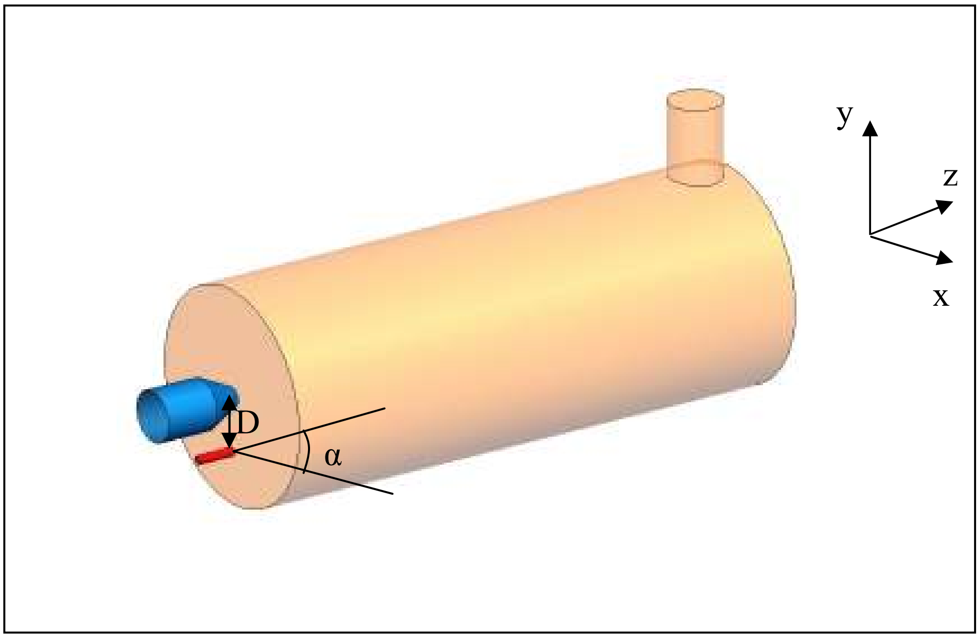

Figure 4

By varying these 2 parameters, we can obtain an intersection point between the 2 jets as far as possible in the combustion chamber (Figure 5). The dilution of the reactants can be calculated at intersection, following Eq.1. The maximum computed values of K_v for each set (D,α) is plotted on Figure 6.
Practically, the distance D has to be limited at 150 mm. The angle α has to be limited to keep a temperature of the recirculation flue gas high enough to reach the auto-ignition threshold. The choice of an angle α of 11° allows a quite good dilution of gas and air before they meet and a mixture temperature above the auto-ignition threshold.

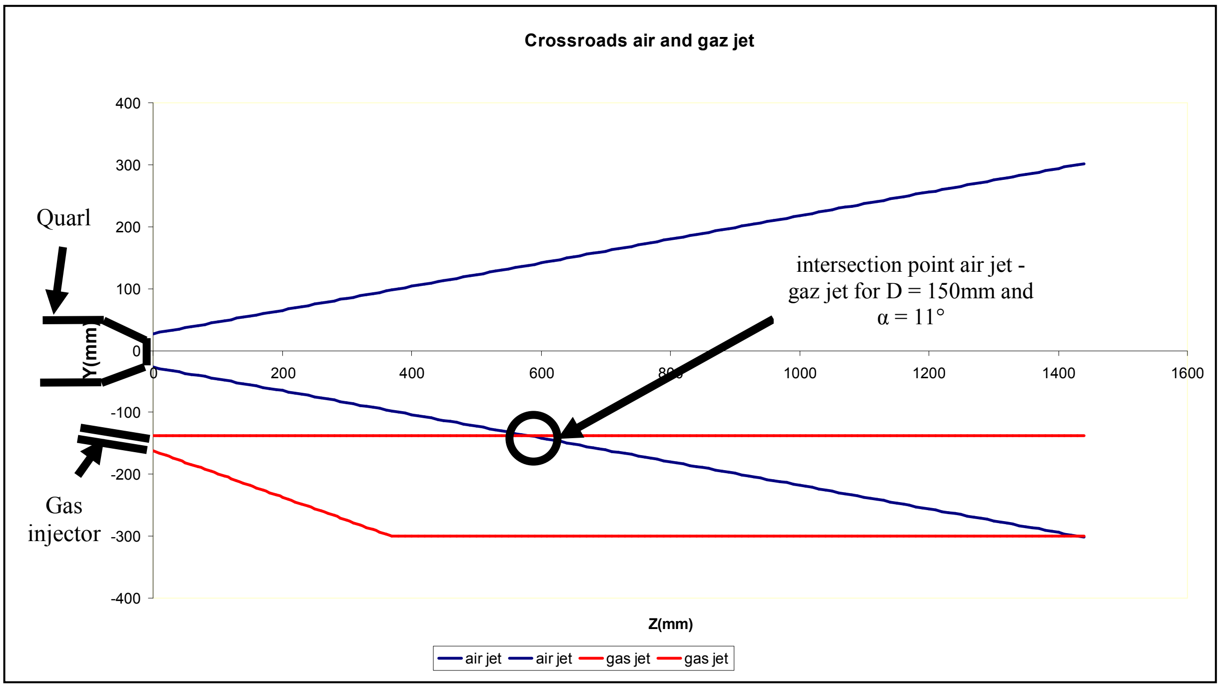

Figure 5

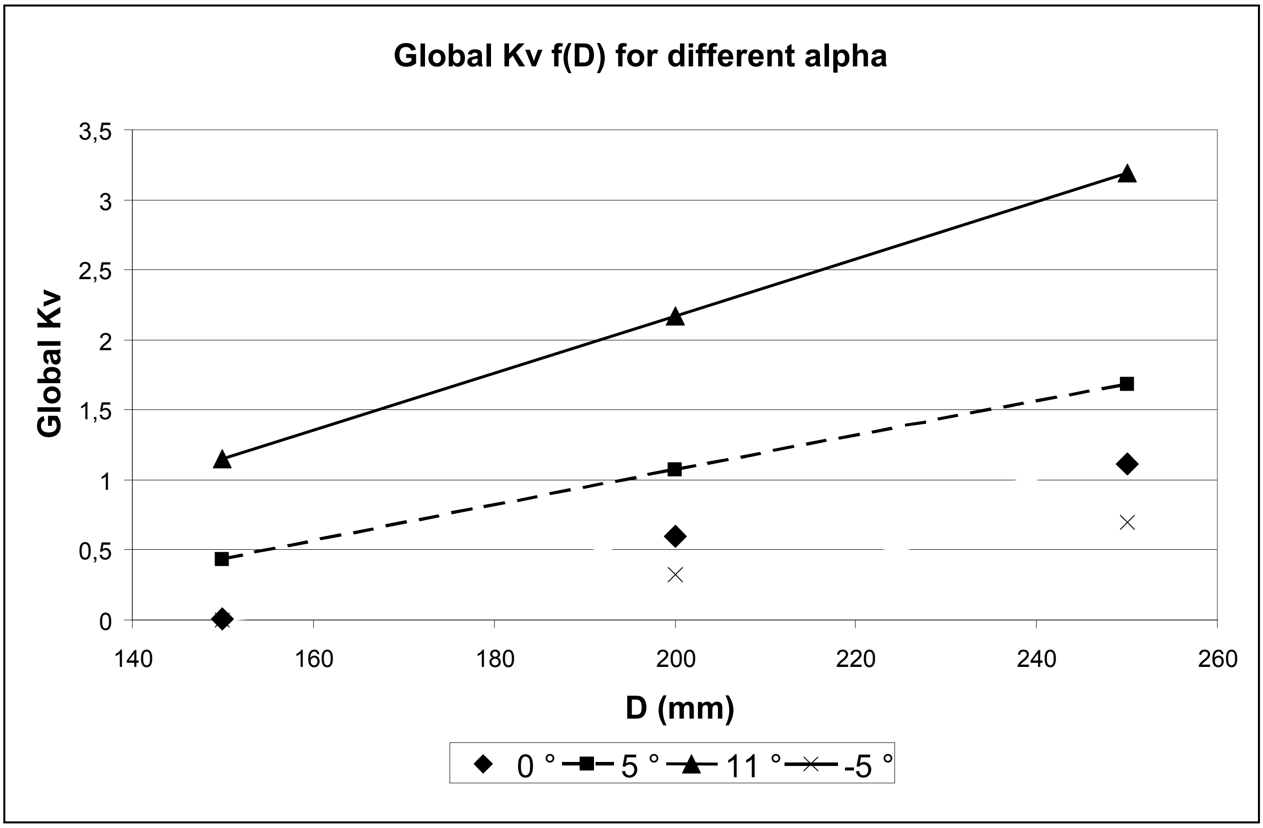

Figure 6

The influence of the diameter can be characterized by the unburned content at the exit of the combustion chamber and the maximal temperature reached in the chamber (Figure 7). A diameter reduction leads to an increase of the unburned level in the chimney and a decrease of the maximal temperature; this is a consequence of the increasing of the reactants dilution by the acceleration of the gas jet. The gas injection diameter has to be optimized to insure a sufficient dilution of the reactants for maintaining the maximal temperature below a critical level for the NOx formation, while conserving the unburned rate under an acceptable level.

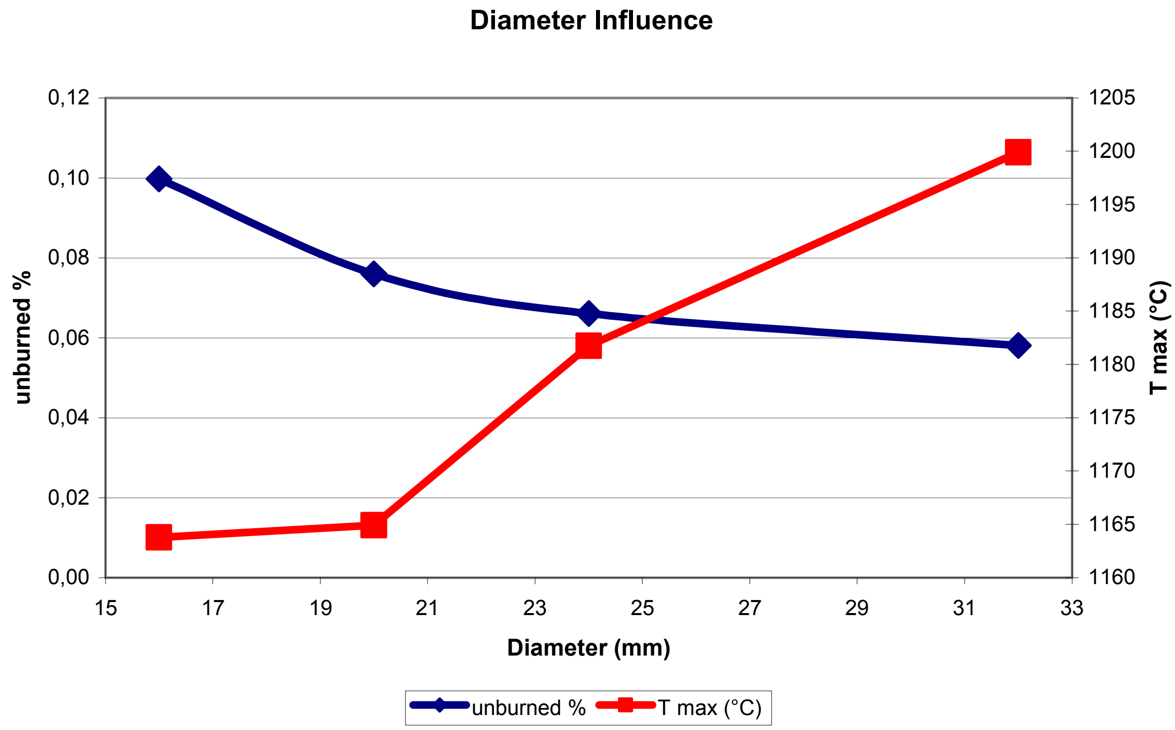

Figure 7

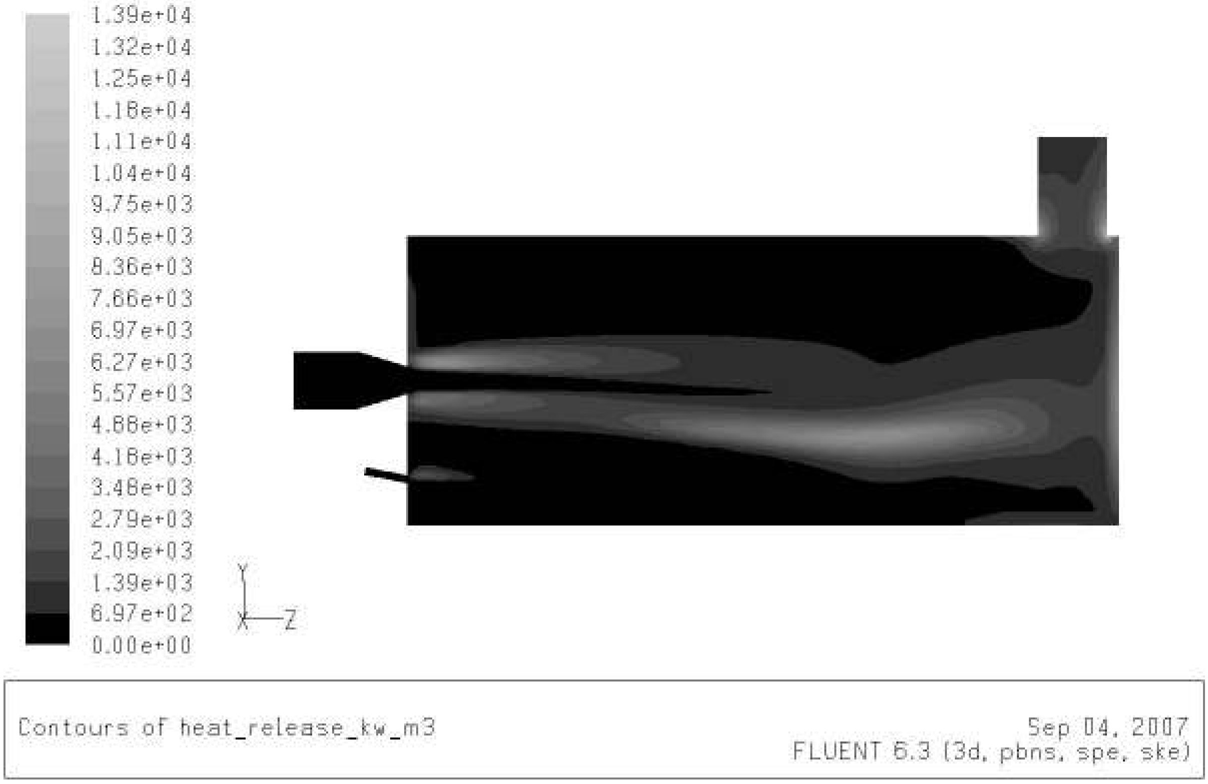

Figure 8: heat release field in the plane YZ for d = 16 mm

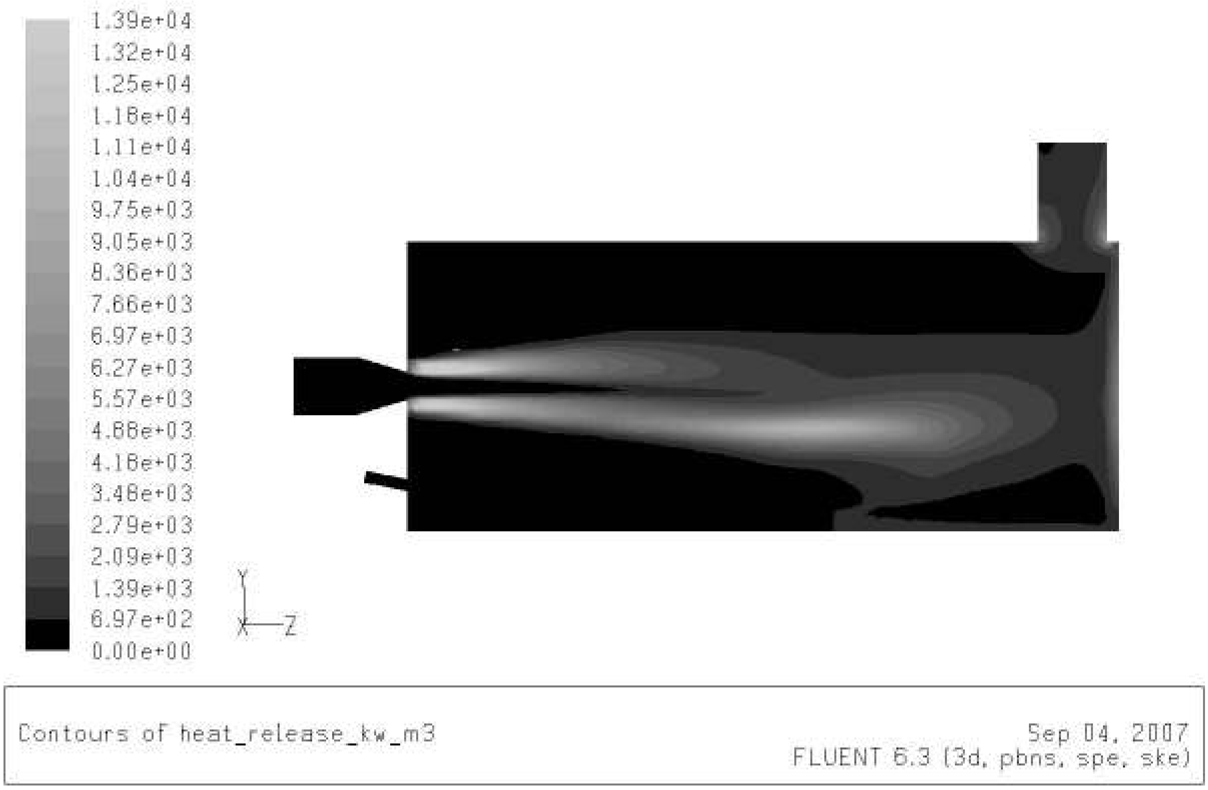

Figure 9: heat release field in the plane YZ for d = 24 mm

By comparing the Figure 8 and 9, we can observe that the heat release is broader for a diameter of 24 mm than for a diameter of 16 mm although the peak value is higher for d = 24 mm. With a diameter of 24 mm, the gas jet velocity allows a good dilution of the reactants in the chamber and a residence time long enough for limiting the unburned rate. While with a diameter of 16 mm, the time of residence in the chamber is reduced that's why the unburned rate is higher than for d = 24 mm. Finally, we opt for a diameter of 24 mm which corresponds to the researched optimum between the dilution and the unburned rate.

Conclusions and perspectives

To assess the feasibility of diluted combustion by direct gas injection in a medium scale boiler at low temperature, a preliminary CFD study has been performed, to select a standard jet burner from the current market, able to generate sufficient internal recirculation for reactants dilution in the boiler. The dependence of the present numerical results to the parameters of the turbulence and combustion models has been examined. In the meantime, a first parametric study has been performed to determine the optimum location, injection angle and diameter of a secondary gas injector to maximize the dilution of the air and gas jets before they meet. The optimum is obtained with a distance D of 150 mm, an angle α of 11° and an injector diameter of 24 mm.
In a second step, an experimental characterization of the selected material will be performed on the semi-industrial boiler available at the Thermodynamics Laboratory of Liège University, to validate numerical results.

Acknowledgements

The authors wish to thank the Walloon Government for continuous financial support of research in the framework of the IEA « Implementing Agreement for a Program of Applied Research, Development and Demonstration in Energy Conservation and Emissions Reduction in Combustion » since 1992.

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