

Study on air-staging combustion of natural gas-hydrogen blend in high temperature industrial heating process

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ABSTRACT

The decarbonization of energy-intensive industrial sectors requires the implementation of low-carbon fuels and advanced combustion technologies. This study investigated the co-combustion of hydrogen and natural gas in an industrial diffusion burner, with particular emphasis on pollutant emissions and heat transfer. Experimental investigations and theoretical analyses were conducted for the fuel blends containing up to 60 vol.% hydrogen over a broad range of operating conditions. A modified burner incorporating staged combustion air, with primary and secondary air supplied separately, was tested in a semi-industrial combustion chamber under high-temperature conditions. The results showed that hydrogen addition increased the total heat flux by up to 3.5%, primarily because of the higher combustion and flue-gas temperatures and changes in the radiative properties of the combustion products, particularly CO₂ and H₂O. However, the higher oxidation rate of hydrogen reduced the flame length, partially limiting radiative heat transfer directly from the flame. Experimental evaluation of the proposed combustion strategy confirmed stable burner operation with very low carbon monoxide emissions, typically not exceeding a few ppm, even at high secondary air ratios. Although hydrogen enrichment initially increased NO emissions, the application of air staging significantly suppressed NO formation, achieving reductions of more than 55% at a secondary air ratio of 0.5. The emission-reduction strategy was more effective at lower excess-air ratios. Overall, the results demonstrated that hydrogen–natural gas co-combustion in industrial furnaces is feasible and can maintain NO emissions below 50 ppm under the investigated operating conditions without requiring secondary flue-gas treatment technologies. The proposed approach, therefore, represents a promising pathway for the decarbonization of high-temperature industrial processes.

Keywords: natural gas hydrogen blend, air-staging combustion, heat transfer, toxic compounds emission, nitric oxides.

INTRODUCTION

Decarbonizing industries that require high levels of process heat is crucial for achieving global climate targets because of their substantial greenhouse gas (GHG) emissions and continued reliance on fossil fuels. Aluminum production is one such process, with an estimated carbon footprint of approximately 11.7 Mg CO₂ per Mg of product. Other energy-intensive sectors include metal heat treatment and steel production, with CO₂ emissions of approximately 2.1 and 1.8 Mg CO₂ per Mg of product, respectively. These processes are often conducted at temperatures ranging from 1080 to 1780 K.

Industry is exploring numerous decarbonization strategies, including the electrification of heating systems, the integration of renewable energy sources, the use of alternative fuels such as hydrogen and ammonia, the optimization of production and auxiliary processes, and the implementation of carbon capture, utilization, and storage (CCUS) technologies. However, CCUS has the major drawback of high energy consumption, typically amounting to approximately 3–4 GJ per Mg of captured CO₂, which can significantly reduce the overall efficiency of the system [1].

Electrification enables precise and efficient heat generation through technologies such as resistance, induction, and electric-arc heating.

Nevertheless, its widespread implementation is constrained primarily by the capacity and availability of the electricity grid infrastructure [2]. Fossil-fuel consumption may also be reduced by integrating solar energy with thermal energy storage (TES) systems, which can provide heat for industrial processes [3]. However, the temperature limitations of many solar-thermal systems and the intermittent nature of solar irradiance remain important challenges to the wider application of this approach.

Another strategy for decarbonizing high-temperature industrial processes is the combustion of alternative fuels. Low-emission synthetic and renewable fuels can be produced through electrolysis using the electricity generated from renewable sources, allowing them to serve as chemical energy carriers and storage media [4]. Among these fuels, hydrogen deserves particular attention. Global industrial hydrogen consumption is projected to increase approximately eightfold by 2050, reaching around 700 million Mg annually [5].

However, hydrogen differs substantially from hydrocarbon fuels in terms of its physicochemical and combustion properties, which affect the operation of thermal equipment with respect to combustion behavior, pollutant emissions, heat transfer, and operational safety. The parameters that differ most significantly from those of natural gas (NG) include laminar flame speed, adiabatic flame temperature, flammability limits, and lower heating value.

Dong et al. [6] demonstrated that, under operating conditions representative of low-power boilers ($T_{\text{air}} = 293 \text{ K}$, $P = 1 \text{ bar}$, and $\phi = 0.8$), the ratio of the laminar flame speed of hydrogen to that of natural gas (SL_{H_2} to SL_{NG}) is approximately 6.7. The substantially higher laminar flame speed makes hydrogen combustion more susceptible to flashback in premixed burners, which may pose a significant safety risk.

The higher adiabatic flame temperature of hydrogen, expressed by a ratio of ($TA_{\text{H}_2}/TA_{\text{NG}} = 1.1$), may contribute to increased nitrogen oxide emissions. At the same time, it may intensify radiative heat transfer within the combustion chamber, potentially improving process efficiency. The volumetric lower heating value of hydrogen is approximately one-third that of natural gas. Consequently, hydrogen substitution may reduce the thermal input of existing equipment unless the fuel supply system and burner are appropriately modified [7].

Another important property of hydrogen is its wide flammability range in air, extending from approximately 4 to 75 vol.% under normal conditions, compared with approximately 4.3 to 15 vol.% for natural gas. Hydrogen is also characterized by a very low minimum ignition energy of approximately 0.02 mJ, compared with about 0.28 mJ for natural gas [8, 9].

The distinct combustion properties of hydrogen have prompted numerous research institutions to investigate the optimization of hydrogen combustion and co-combustion technologies for both stationary energy systems and transport applications.

El-Ghafour, El-Dein, and Aref conducted experimental investigations of a free turbulent diffusion jet flame surrounded by a co-flowing air stream. The experiments were performed at a constant jet Reynolds number of 4000 and hydrogen volume fractions ranging from 0 to 50 %. The results showed that increasing the hydrogen content improved flame stability, shortened the flame, and increased the combustion temperature. An increase in NO emissions was observed, whereas CO emissions initially increased and then decreased at higher hydrogen fractions [10]. Lindmüller et al. investigated hydrogen enrichment in an unmodified natural-gas swirl burner. Their results showed that hydrogen addition increased the NO_x emission index by up to a factor of three while the thermal input was maintained constant [11]. Daurer et al. performed experimental and numerical analyses of an industrial low-swirl burner operating with hydrogen fractions of 0, 20, 50, and 100 vol.% over a thermal input range of 75–120 kW. Increasing the hydrogen content enhanced heat transfer to the cooling lances by up to 20 %, resulting in an efficiency improvement of 5–10 percentage points. For a hydrogen fraction of 50 vol.%, NO_x emissions decreased by 5–10 % at all operating points. During pure hydrogen combustion at furnace temperatures below 1473 K, the reduction reached approximately 20 %. The authors attributed these trends primarily to changes in the flow regime and flame structure [12]. Gee et al. investigated a scaled industrial low-swirl burner and found that complete substitution of natural gas with hydrogen reduced the radiant heat fraction generated by the flame by 33%. The authors attributed this effect to the elimination of soot radiation. Hydrogen addition also increased NO_x emissions by 25% and 47% at hydrogen fractions of 10% and 20%, respectively. Pure hydrogen combustion resulted in a 4.8-fold

increase in NO_x emissions [13]. As the studies discussed above demonstrate, hydrogen addition may lead to different trends in nitrogen oxide emissions during combustion. These inconsistencies indicate that further research is required to better understand how hydrogen affects emission levels under different operating conditions. In the present study, an experimental investigation was conducted using a diffusion gas burner modified to enable air-staged combustion of fuels containing high hydrogen fractions under high-temperature conditions. The experimental results were complemented by a heat-transfer analysis, making the findings presented in this paper a useful knowledge base for the researchers and engineers interested in the application of H₂/NG blends in atmospheric-pressure combustion systems.

METHODS

Heat transfer analysis

The total heat flux transferred to the combustion chamber consists of flame radiation, exhaust gas radiation, and convection, and can be described by Equation 1.

$$THF = Q_{R,fl} + Q_{R,fg} + Q_c \text{ [W]} \quad (1)$$

where: *THF* – total heat flux, *Q_{R,fl}* – flame radiative heat transfer, *Q_{R,fg}* – flue gas radiative heat transfer, *Q_c* – convection heat transfer.

A change in fuel type affects the parameters governing all of these heat-transfer mechanisms. Therefore, the present study analyzed the influence of the physicochemical properties of the investigated fuels on heat transfer. The calculation procedure used to determine the heat-transfer rate to the combustion chamber walls is described by Equations 2–5. The radiative heat-transfer rate from the flame to the chamber walls was calculated using Equation 4, following the methodology reported in [14].

$$Q_{R,fl} = 5.67 F_{fl} \frac{1}{\frac{1}{\epsilon_{fl}} + \frac{F_{fl}}{F} \left(\frac{1}{\epsilon_w} - 1 \right)} \left[\left(\frac{T_{fl}}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] \text{ [W]} \quad (2)$$

The flame emission coefficient (ϵ_{fl}) for the analyzed fuels was determined according to formula (5) using the model described in [15, 16], where the weighting factors were calculated based on model proposed by Elhme et al. in [17].

$$\epsilon_{fl} = \sum_{i=1}^n a_i [1 - \exp(-k_i XPL)] \quad (3)$$

where: *F_{fl}* – flame area; ϵ_{fl} – flame emissivity coefficient; ϵ_w – wall emissivity coefficient; *a_i* – weighting factor; *k_i* – absorption coefficient; *L* – length; *p* – pressure in combustion chamber; *X* – sum of the molar fraction of absorbing species; *T_{fl}* – adiabatic flame temperature, *T_w* – combustion chamber wall temperature.

The heat-transfer rate between the flue gas at temperature *T_{fg}* and the furnace wall at temperature *T_w*, accounting for the wall emissivity, was determined using the methodology described in [18, 19]. The effective emissivity of the flue gas was calculated as a function of the partial pressures of CO₂ and H₂O and the mean flue gas temperature in the combustion chamber, following the procedure reported in [20].

$$Q_{R,fg} = 5.67 F (\epsilon_{fg} + 1) \cdot$$

$$\left[\epsilon_{fg(T_{fg})} \left(\frac{T_{fg}}{100} \right)^4 - \epsilon_{fg(T_w)} \left(\frac{T_w}{100} \right)^4 \right] \text{ [W]} \quad (4)$$

where: *F* – heat exchange area; ϵ_{fg} – flue gas emissivity coefficient.

For the convective heat flux, the Mendelev correlation (5) was applied. This equation is valid for Reynolds numbers in the range of 2300 to 10000 [21].

$$Q_c = 0.00263 F \frac{h \cdot Pr^{0.35}}{L_c} \cdot Re^{0.8} \cdot (T_{fg} - T_w) \text{ [W]} \quad (5)$$

where: *h* – heat exchange coefficient; *Re* – Reynolds number; *Pr* – Prandtl number; *L_c* – characteristic length.

Since the combustion chamber walls are made of insulating brick, a constant wall emissivity coefficient of $\epsilon_w = 0.7$, corresponding to a temperature of 1073 K, was assumed in the calculations. In addition, an average wall temperature of *T_w* = 1088 K was adopted, corresponding to the wall average temperature during the experimental tests.

Experimental investigations

Experimental studies were conducted in a combustion chamber equipped with a modified

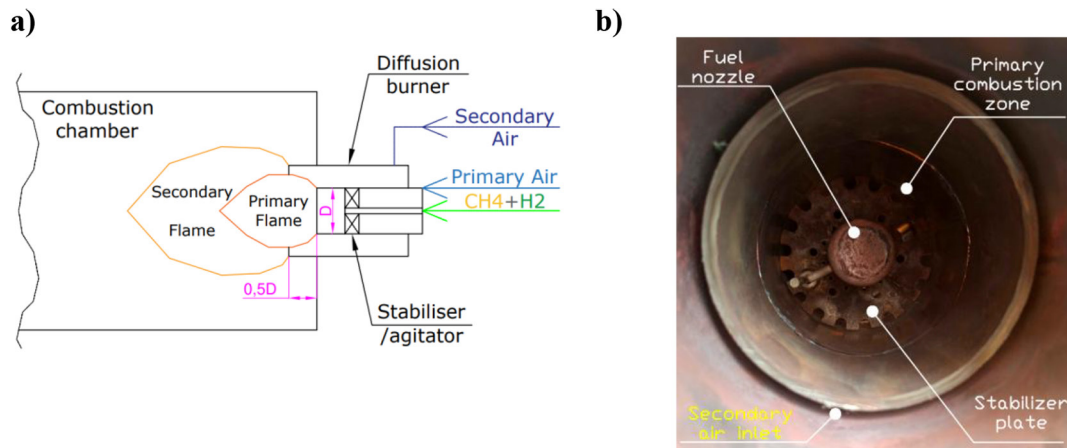


Figure 1. Scheme of gas burner: a) diagram of air and fuel distribution, b) photo of nozzles

industrial natural gas burner to assess the effect of air staging on the emission characteristics of natural gas–hydrogen blend combustion. Air was supplied to the primary combustion zone through a stabilizer plate and to the secondary zone located at a distance corresponding to half of the burner diameter. The natural gas–hydrogen blend was injected into the primary zone through a fuel nozzle positioned along the burner axis. A schematic diagram and a photograph of the burner are presented in Figure 1.

The hydrogen and natural gas flow rates supplied to the fuel line were controlled using Bronkhorst mass flow controllers to ensure the required fuel composition. Primary and secondary air were supplied by two separate blowers equipped with frequency converters and butterfly valves for flow-rate regulation. A total heat-flux

probe was installed 450 mm downstream of the burner to measure changes in heat transfer. During the experiments, the combustion chamber wall temperature was monitored using a K-type thermocouple located at the mid-length of the chamber to ensure stable, steady-state operating conditions. The wall temperature was regulated using an air-cooled heat exchanger installed in the floor of the combustion chamber and was maintained at 1088 ± 11 K. The flue gas composition, including the concentrations of NO, O₂, CO, and CO₂, was measured using an NGA 2000 gas analyzer manufactured by Rosemount Inc. The sampling point was located in the exhaust stack downstream of the heat exchanger. The measured emission concentrations were normalized to a reference oxygen concentration of 0% O₂ according to Equations 6 and 7 [14]. A schematic diagram

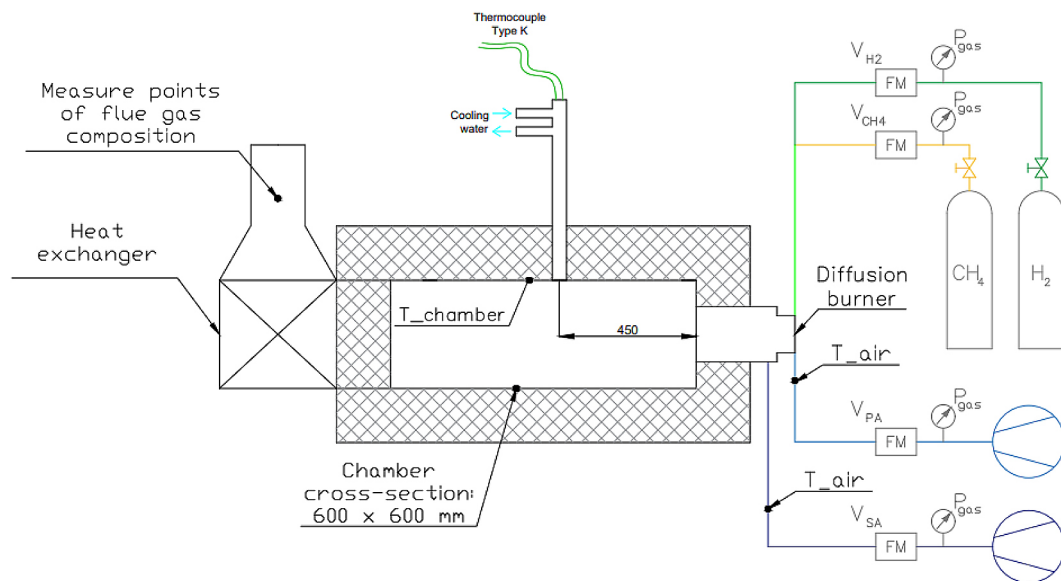


Figure 2. Schematic diagram of the combustion test rig

Table 1. List of measurement devices

Sensor/devices	Measurement	Measurements range	Accuracy
Gas analyzer Rosemont-Emerson NGA300	Flue gas	O ₂ : 0–21% CO: 0–100 ppm NO: 0–100 ppm CO ₂ : 0–25%	±1% FS
Mass flow controller Bronkhorst EL-FLOW	NG flow rate	0–150 NI/min	±0.5 % FS
Mass flow controller Bronkhorst EL-FLOW	H ₂ flow rate	0–200 NI/min	±0.5 % FS
Manometer MRU DM9200	Fuel pressure	350 mbar	±1 % FS
Thermocouple Type K	Temperature	-20 to 1273 K	±1.5 K
Heat flux probe	Total heat flux	200–600 kW/m ²	±1.5 % MV

Table 2. Selected properties of tested fuel blends

Fuel name	NG	H ₂	H ₂ heat power contribution	T _A ¹	S _L ¹	LHV
	% vol.	% vol.	%	K	cm/s	MJ/Nm ³
NG100	100	0	0	2212	32.9	36.1
NG_H2_45	54	46	20	2241	48.6	24.4
NG_H2_60	41	59	30	2255	60.3	21.1

Note: ¹Calculated for NG/H₂ blend in T = 273.15 K, P = 1 bar, λ = 1.

of the experimental setup is presented in Figure 2, while the measurement equipment used in the study is listed in Table 1.

$$NO_{0\%} = NO \frac{21 - O_{2ref}}{21 - O_{2mes}} [ppm] \quad (6)$$

where: *NO* – measured *NO* emission, *O_{2ref}* – reference oxygen concentration, *O_{2mes}* – measured oxygen concentration.

$$CO_{0\%} = CO \frac{21 - O_{2ref}}{21 - O_{2mes}} [ppm] \quad (7)$$

where: *CO* – measured *CO* emission, *O_{2ref}* – reference oxygen concentration, *O_{2mes}* – measured oxygen concentration.

The experimental investigations were conducted at a constant thermal input of 70 kW and a constant air temperature of 295 K. Pure natural gas and fuel blends containing 46% and 59% hydrogen by volume were investigated. These hydrogen fractions corresponded to 20% and 30% of the total thermal input, respectively. The basic properties of the investigated fuel blends are summarized in Table 2.

During the experiments, the effect of the distribution of air between the primary and secondary

Table 3. Experimental test parameters

Test number	Fuel name	λ	SAR
-	-	-	-
1	NG100	1.05	0
2	NG100	1.05	0.2
3	NG100	1.05	0.3
4	NG100	1.05	0.4
5	NG100	1.05	0.5
6	NG_H2_60	1.05	0
7	NG_H2_60	1.05	0.2
8	NG_H2_60	1.05	0.3
9	NG_H2_60	1.05	0.4
10	NG_H2_60	1.05	0.5
11	NG_H2_45	1.05	0
12	NG_H2_45	1.05	0.2
13	NG_H2_45	1.05	0.3
14	NG_H2_45	1.05	0.4
15	NG_H2_45	1.05	0.5
16	NG_H2_60	1.16	0
17	NG_H2_60	1.16	0.3
18	NG_H2_60	1.16	0.5
19	NG100	1.16	0
20	NG100	1.16	0.3
21	NG100	1.16	0.4
22	NG100	1.16	0.5

combustion zones on flue-gas emissions was investigated. In the following sections, this parameter is referred to as the secondary air ratio (SAR), defined as the ratio of the secondary air mass flow rate to the total air mass flow rate. The experiments were conducted at SAR values ranging from 0 to 0.5 and at two oxygen concentrations in the dry flue gas, namely 1% and 3%, corresponding to excess-air ratios (λ) of 1.05 and 1.17, respectively. A summary of the experimental cases and their operating parameters is presented in Table 3.

Each experimental run consisted of two stages: stabilization of the operating conditions and acquisition of the measurement data. During the experiments, the following parameters were recorded: combustion chamber temperature, flue gas temperature, combustion air temperature, natural gas and hydrogen flow rates, heat flux, and flue gas composition. For each operating condition, data were collected for 120 s at a sampling frequency of 1 Hz. The recorded values were subsequently averaged and used in further analysis.

RESULTS AND DISCUSSION

Effect of fuel composition on heat transfer

Experimental and numerical investigations were conducted to determine the effect of fuel composition on heat-transfer phenomena in a semi-industrial furnace for all investigated fuels. The calculations were performed at a constant excess-air ratio of $\lambda = 1.05$ and a constant combustion chamber wall temperature ($T_{\text{wall}} = 1100$ K) corresponding to the experimental conditions. The flame length used in the heat-transfer calculations was determined by analyzing images of

the gaseous flames, with flame luminosity adopted as the primary criterion. The flame region was defined using a threshold corresponding to 80% of the maximum image brightness [22]. An example of the flame-length determination procedure is presented in Figure 3.

Experimental and numerical studies of the effect of changes in the chemical composition of fuel on heat transfer phenomena in a semi-industrial furnace were carried out for all types of studied fuels. The calculation has been prepared for a constant value of air excess ratio equal $\lambda = 1.05$ and a constant temperature of combustion chamber walls related to experimental result ($T_{\text{wall}} = 1100$ K). The flame length used for heat transfer calculations was determined based on the analysis of gas flame images, using flame luminosity as the primary criterion. It was assumed that the flame region corresponds to 80% of the maximum brightness [22]. An example of the flame length determination procedure is presented in Figure 3.

In addition to the fuel composition and oxidizer flow rate, flame length is influenced by the diameter and geometry of the burner nozzle. Therefore, the measured flame lengths were normalized with respect to the value obtained for the reference fuel, NG100. The results indicate that hydrogen addition reduced the flame length by 5.3% and 11% for NG_H2_45 and NG_H2_60, respectively. The shorter flame affects radiative heat transfer from the flame to the combustion chamber walls. However, it should also be noted that increasing the hydrogen content increases the combustion temperature. The results of the heat-transfer analysis were normalized with respect to the values obtained for natural gas combustion and are presented in Table 4.

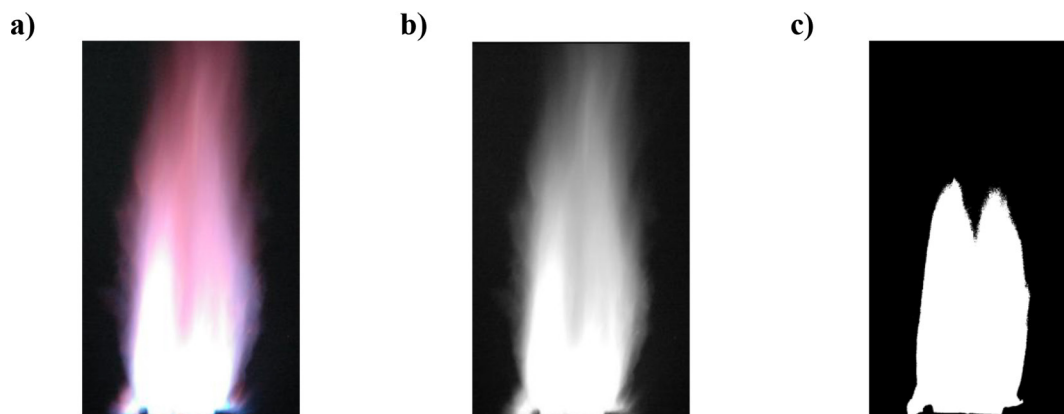


Figure 3. An example of the flame length determination procedure (NG_H2_45; $\lambda = 1.05$): a) raw photo, b) grey scale, c) flame contour for 80 % of maximum luminosity

Table 4. Heat transfer normalized to NG100 fuel

Parameter	Symbol	NG_H2_45		NG_H2_60	
Convective heat flux	Q_c	1.020	↑	1.027	↑
Radiative heat flux from flue gases	$Q_{R,fg}$	1.015	↑	1.046	↑
Flue gas emissivity coefficient in T_{fg}	ϵ_{fg}	0.938	↓	0.940	↓
Radiative heat flux from flame	$Q_{R,fl}$	0.987	↓	0.978	↓
Flame emissivity coefficient	ϵ_{fl}	1.039	↑	1.099	↑
Total heat flux ($THF = Q_c + Q_{R,fg} + Q_{R,fl}$)	THF	1.021	↑	1.050	↑
Experimental total heat flux	THF_{exp}	1.023	↑	1.035	↑

Experimental measurements of the total heat flux (THF), performed using a water-cooled probe, yielded results consistent with the analytical calculations of heat transfer from the flame and flue gases in the semi-adiabatic combustion chamber. Compared with the reference fuel, NG100, the THF increased by 2.3 % and 3.5 % for NG_H2_45 and NG_H2_60, respectively. The theoretical analysis indicates that this increase was primarily attributable to enhanced convective and radiative heat transfer from the flue gases. For the hydrogen-enriched fuels, the reduction in effective flue-gas emissivity was compensated for by an increase in the flue-gas temperature. At the same time, radiative heat transfer directly from the flame decreased by 1.3% and 2.2% for NG_H2_45 and NG_H2_60, respectively, despite the increase in flame emissivity. This effect resulted from a reduction in the flame surface area associated with the shorter flame length caused by the higher hydrogen combustion rate.

Analysis of toxic compounds emission

Pollutant emissions are among the key challenges associated with the combustion of fossil fuels. Therefore, as part of the present study on the co-combustion of hydrogen and natural gas in a semi-industrial furnace, nitrogen oxide and carbon monoxide emissions were measured.

The available literature does not provide a clear consensus on whether increasing the hydrogen content in the fuel leads to higher or lower NO emissions, as the observed trends strongly depend on the combustion system configuration and operating conditions. Preliminary investigations conducted by the authors showed that the combustion of natural gas–hydrogen blends in an industrial diffusion burner increased NO emissions from 56 ppm for NG100 to 73 ppm and 76 ppm for NG_H2_45 and NG_H2_60, respectively. These

values correspond to increases of 30% and 38% relative to the reference fuel.

To mitigate the increase in NO emissions, the burner was modified by staging the combustion air. Primary air was supplied in the vicinity of the fuel outlet, whereas secondary air was introduced at a distance corresponding to 0.5 burner diameters downstream of the burner outlet, as illustrated in Figure 1. The emission results obtained for the investigated fuels at secondary air ratios ranging from 0.2 to 0.5 are presented in Figures 4 and 5. Stable burner operation was maintained at a SAR of 0.5, as confirmed by the low carbon monoxide emissions recorded during the experiments and presented in Figure 6.

As shown in the results, dividing the combustion air into primary and secondary streams reduced NO emissions for all investigated fuels when the SAR exceeded 0.3. At SAR = 0.2, a slight increase in NO emissions was observed. This may be attributed to the combination of sufficiently high temperatures and the availability of atomic oxygen in the vicinity of the fuel nozzle, which promoted NO formation through the thermal mechanism. Increasing SAR reduced the amount of oxidizer supplied to the primary combustion zone, thereby creating locally fuel-rich conditions. Consequently, the combustion temperature decreased by more than 200, 350, and 500 K at SAR values of 0.3, 0.4, and 0.5, respectively. These conditions suppressed NO formation, but simultaneously increased the concentrations of carbon monoxide and CH radicals, which subsequently had to be oxidized by the secondary air. The measured NO emissions for the hydrogen-containing fuels remained higher than those for natural gas under the corresponding operating conditions. Nevertheless, at SAR values of 0.4 and 0.5, the NO mole fractions were lower than those measured during the pure natural gas combustion in the unmodified burner.

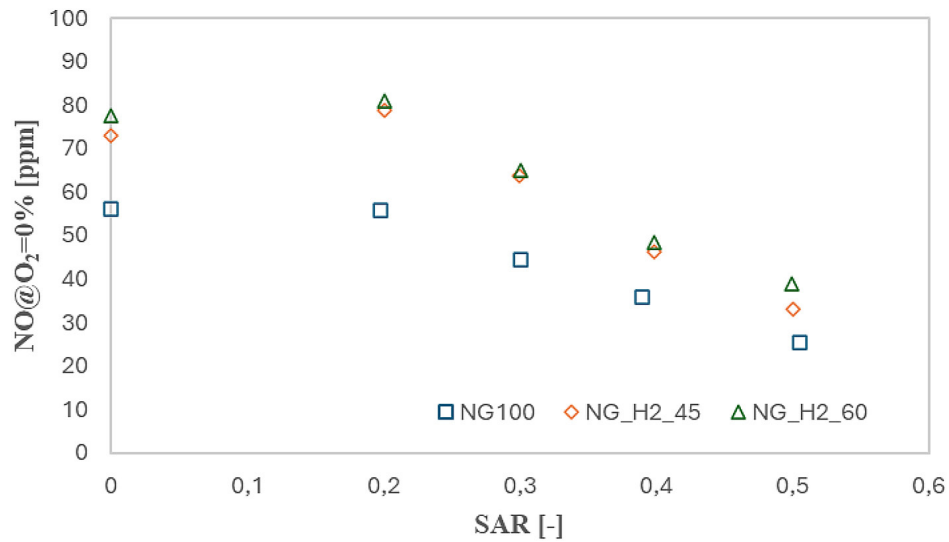


Figure 4. Molar fraction of nitric monoxide in the flue gas versus SAR for $\lambda = 1.05$

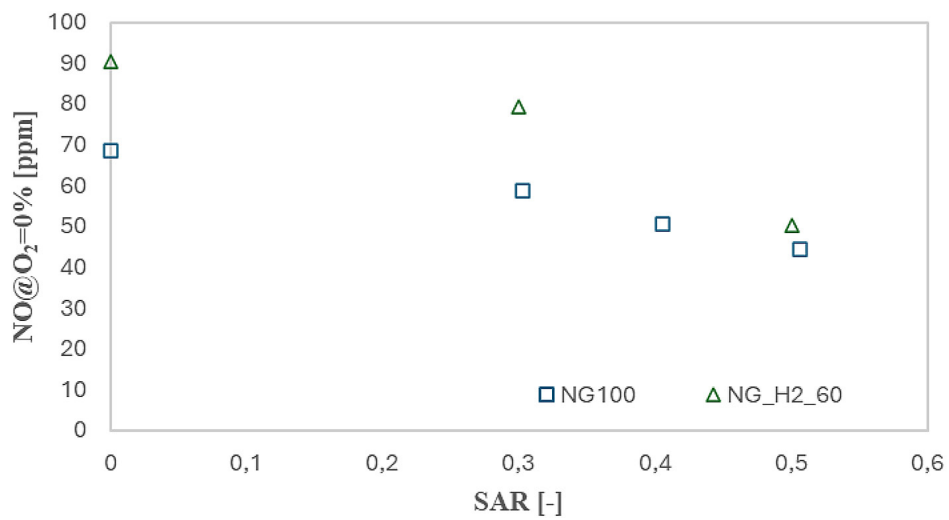


Figure 5. Molar fraction of nitric monoxide in the flue gas versus SAR for $\lambda = 1.17$

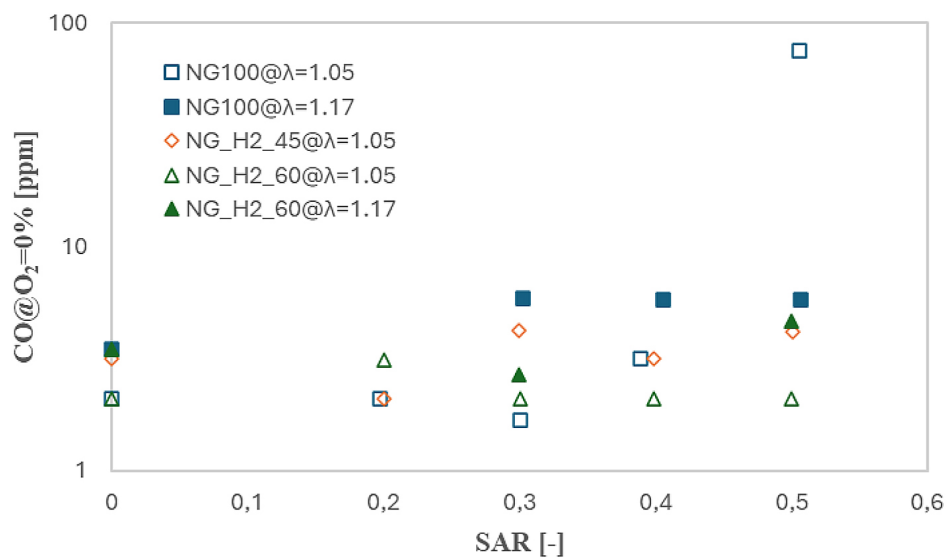


Figure 6. Molar fraction of carbon monoxide in the flue gas versus secondary air ratio

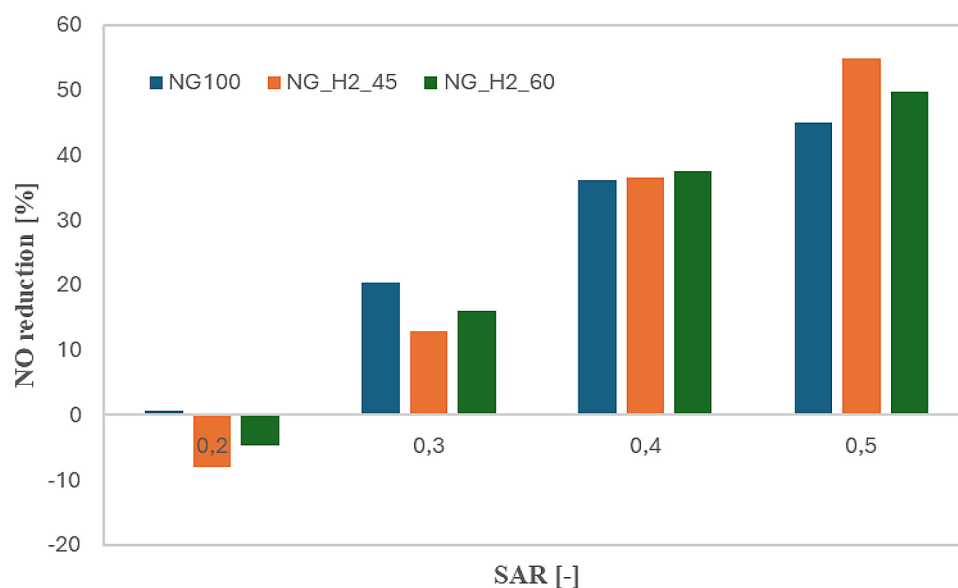


Figure 7. NO reduction (relative to SAR = 0) for the investigated fuels at $\lambda = 1.05$

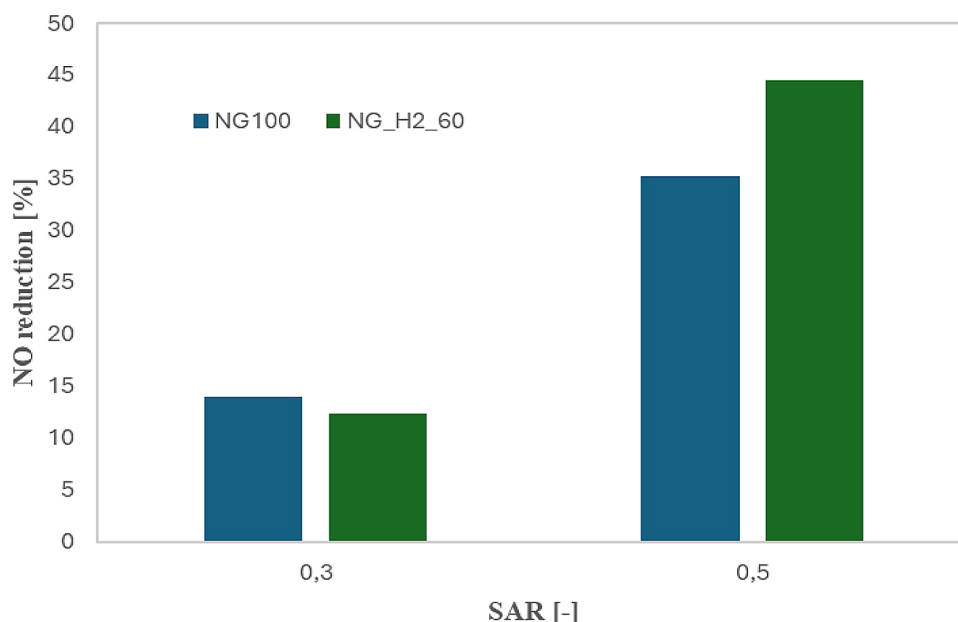


Figure 8. NO reduction (relative to SAR = 0) for the investigated fuels at $\lambda = 1.17$

As shown in Figure 6, carbon monoxide emissions did not exceed 10 ppm for most measurement points. The only exception was observed for NG100 at SAR = 0.5, for which the CO concentration increased to 76 ppm. This increase was attributed to the formation of a highly fuel-rich mixture in the primary combustion zone, combined with a relatively low flue-gas temperature in the secondary zone.

The NO emission reduction factors presented in Figures 7 and 8 indicate that the applied mitigation strategy was more effective at lower global excess-air ratios. For example, for NG_H2_60 at

SAR = 0.5, the reduction in NO emissions relative to the corresponding baseline case for the same fuel at SAR = 0 was 50% at $\lambda = 1.05$ and 44% at $\lambda = 1.17$. The greater oxygen availability in the primary combustion zone at $\lambda = 1.17$ increased the flame temperature and the concentration of O radicals, thereby intensifying the NO formation through the Zeldovich mechanism.

The results of the emission analysis indicate that the combustion of the NG/H₂ blends with high hydrogen content in high-temperature combustion systems used in industrial furnaces is feasible while maintaining low pollutant emissions.

CONCLUSIONS

The combustion of pure hydrogen or its co-combustion with hydrocarbon fuels reduces specific carbon dioxide emissions. However, hydrogen addition may also affect key combustion-process parameters, including heat transfer and pollutant formation. The theoretical analysis, supported by experimental heat-flux measurements during the combustion of hydrogen-enriched gaseous fuels, demonstrated that hydrogen addition had a beneficial effect on heat transfer. Increasing the hydrogen content in the fuel increased the total heat flux by 2.3% and 3.5% for fuels containing 45 vol.% and 60 vol.% H₂, respectively. The theoretical analysis indicated that this enhancement was primarily attributable to the higher combustion and flue-gas temperatures and to changes in the radiative properties of the flame and combustion products, particularly those associated with CO₂ and H₂O. A factor limiting radiative heat transfer from the flame was the reduction in flame length resulting from the higher oxidation rate of hydrogen.

The results obtained for the proposed modification of the industrial diffusion burner, based on combustion-air staging in the fuel-injection region through the division of air into primary and secondary streams, demonstrated its applicability to both natural gas and fuels containing substantial hydrogen fractions. The experiments showed that the modified burner maintained stable combustion conditions and generally produced very low carbon monoxide emissions. Even at SAR values as high as 0.5, CO emissions did not exceed a few ppm for the hydrogen-containing fuels. For SAR values above 0.3, the measured NO emissions were lower than the reference values obtained for pure natural gas combustion without air staging. The degree of NO reduction increased with the fraction of air supplied to the secondary combustion zone, reaching a maximum of more than 55 % for the fuel containing 45 vol.% H₂. The investigated emission-reduction strategy was also found to be more effective at lower excess-air ratios, making it particularly suitable for high-temperature industrial combustion processes. The experimental results and the analyses performed indicate that, within the investigated range of operating parameters, the co-combustion of hydrogen and natural gas using the proposed high-temperature combustion technology is feasible while maintaining low pollutant emission levels, without the need for secondary emission-control measures.

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