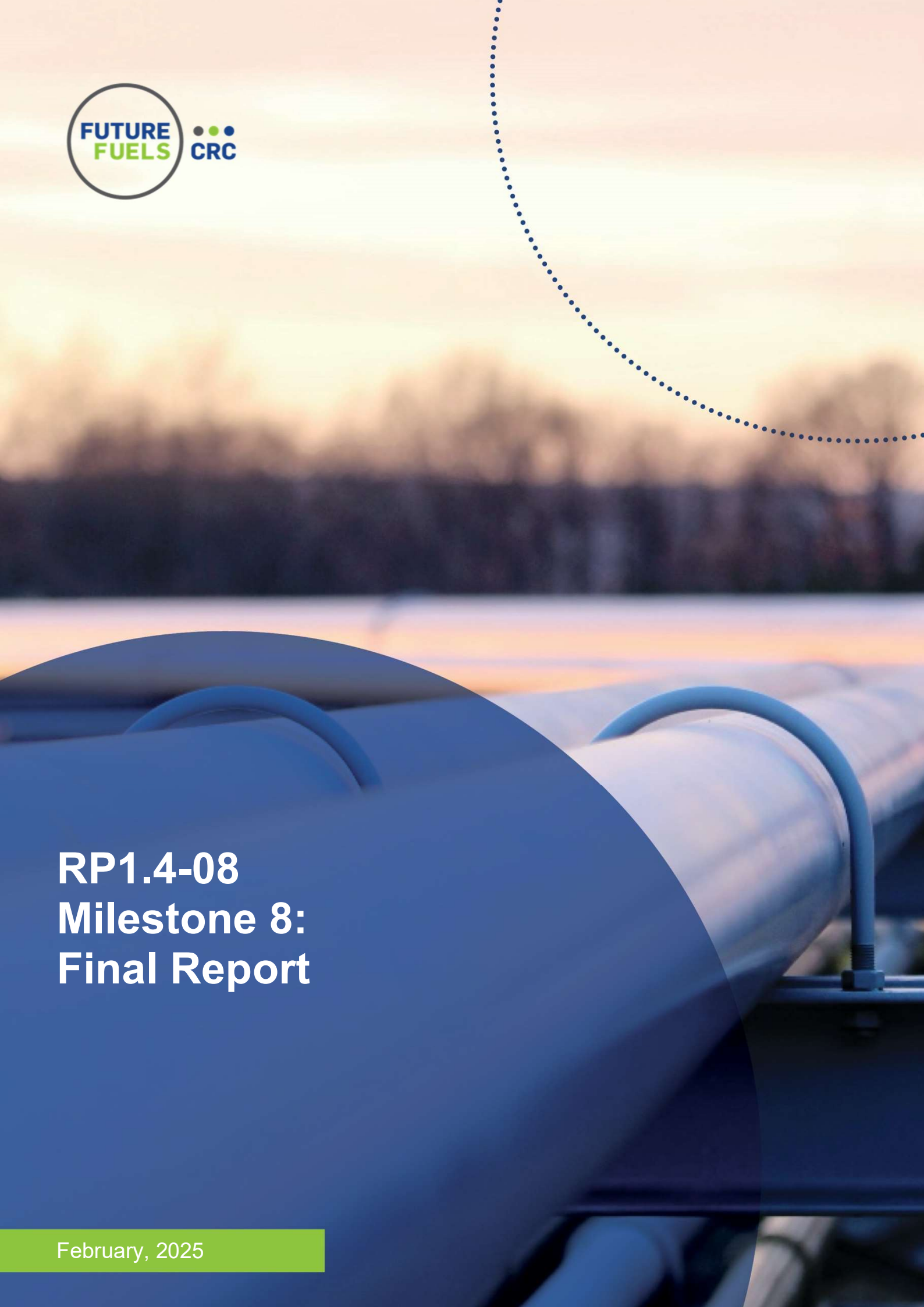

RP1.4-08 Milestone 8: Final Report

February, 2025

Project number: RP1.4-08

Detailed assessment of Type B appliances with blends of hydrogen and natural gas and 100% hydrogen

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Business Cooperative Research Centres Program

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Summary

1.1 Background and aims

This document is a compilation of reports from project RP1.4-08 covering experimental work performed at the University of Adelaide, funded by the Future Fuels Cooperative Research Centre.

The first aim of the project was to establish whether a wide range of commercial and industrial natural gas appliances are suitable for use with blends of up to 20 volume% hydrogen, which may be used in some gas networks in the future. University combustion research typically uses generic burner designs so that scientific testing can be undertaken, but here testing the performance of real-world burners was targeted.

The burners used in the project were designed for natural gas, rather than hydrogen or blends, and so it was expected that if blends with higher fractions of hydrogen in the fuel were used issues may arise. The second aim was to explore the limits of hydrogen blending and operation with 100% hydrogen without changing the natural gas burners. In other words, deliberate stress testing of burners beyond their design capability. (A similar project conducted by the same team has investigated domestic and commercial catering appliances).

1.2 Burners

The following new burners/appliances were purchased specifically for the project and tested:

- A partially-premixed, naturally-aspirated burner with a Flame Retention, FR8 nozzle
- A partially-premixed, naturally-aspirated burner with an AN-style nozzle rated at 35 kW (120 MJ/h)
- A forced draft premixed burner. Riello package burner (Gulliver BS2F) with integral fan, rated at 35–92 kW with a 106 mm diameter burner tube
- An Eclipse air heat burner, AH0050, rated at 25 – 530 MJ/h (147kW)
- A forced-draft North American Tempest 4442 high-velocity nozzle-mix burner rated at ~20 kW (72 MJ/h), using an air pressure of 7 kPa
- A Thermjet TJ0015, Medium velocity alloy tube natural gas burner
- Pilot burners, flame ionisation rod detectors and UV flame detectors integral to some of these burners

In addition to the purchased burners, a second-hand Airatherm, SuperRay, WEA-24 ported ceramic premixed radiant surface burner (7kW) was tested and an older Riello two-stage package burner was used in a demonstration in an existing industrial furnace.

Further plans to test some additional burners did not come to fruition, but the burners tested cover a wide range of types and results are relevant for other similar burners.

1.3 Key findings

As hydrogen is blended into natural gas the Wobbe index reduces slightly which results in slightly less heat throughput at the same pressure. This was illustrated in tests with the FR8 nozzle and in a demonstration in an industrial furnace. However, it is common practice to control fuel rate to a set temperature, which will automatically increase fuel flow as required.

The lower Wobbe index could also cause a restriction through the burner valve train, but only if it is already near maximum capacity. However, the difference in fuel input rate at the same pressure between natural gas and a 30 vol% hydrogen blend is of the order of -7%, which can be accounted for by a small increase in inlet pressure.

Open firing tests showed that flame appearance, flame length, radiation and flue gas emissions were very similar to natural gas operation for the likely blend ratios to be used in networks (up to 20 vol% hydrogen). Hence a wide range of Type B appliances will be suitable for these blends.

Open firing showed that the maximum operable H_2% without changes is approximately 40 vol% for partially premixed appliances, but higher for appliances with nozzle mix burners. Burner parts are subjected to higher temperatures above 50 vol% and can exceed 600ºC for 100% hydrogen in open firing, showing that some burners will need to be changed for 100% hydrogen firing. (Since the tests were done, communication with Riello has confirmed they have a new design for 100% hydrogen which does not result in high metal temperatures). Ionisation rod flame detector performance depended on the burner and appliance. Ionisation rod flame detectors

are not suitable for 100% hydrogen flames but performed with up to 99 vol% hydrogen for one burner, whilst others performed up to approximately 80 vol%. They can be replaced with UV flame detectors in these cases.

In confined firing steady-state laboratory experiments heat transfer and NOx and CO emissions did not vary greatly with fuel composition for four enclosures representing furnaces (AN burner in a cylindrical duct, the Tempest burner in two different sized rectangular box shaped enclosures and a package burner in a cylindrical enclosure). Heat transfer rates, which would translate to product quality in an industrial furnace, had a stronger dependence on air rate and fuel rate than on fuel composition. This implies that either automatic temperature control or slight adjustments of fuel and air rates will be suitable for possible network blends and blends with higher vol% H2 blends and even 100% hydrogen, unless the process relies on flame radiation.

Calculations confirmed that air rates have a large effect on both radiation and convection heat transfer and that combustion gases from 100% hydrogen flames produce slightly lower radiation than natural gas and blends, Hence, if using 100% hydrogen slightly higher fuel input rates may be required in some applications where gas radiation is important, but it would only amount to a few percent of the total. A small proportion of biomethane in hydrogen would improve radiation.

Carbon monoxide and nitrogen oxide emissions from 10-20 vol% hydrogen blend flames were similar to those from natural gas flames. Emissions of nitrogen oxides from 100% hydrogen operation were significantly higher than natural gas and blends for some burners. However, burner design and operation will receive attention to address this issue, noting that in this project the burners were designed for natural gas not hydrogen.

A two-day industrial furnace demonstration with a 30 vol% hydrogen blend illustrated that for a short trial, it was possible to use existing natural gas certified valve train components, controllers and materials after performing only a “soapy water” hydrogen pressure test on all the fittings. No leaks were observed with personal gas alarms during the two days, so if there was any increase in leakage rate with the blend it was minor. It has been noted that others have shown higher leakage rates with hydrogen and blends than natural gas, so leakage from existing natural valve trains when operated on blends warrants further investigation. Hydrogen rated components are available, so suppliers can be consulted if a permanent installation is planned, but valve train pressure testing using the blend and comparison to natural gas results would establish whether any changes are required.

In this project the Type B approval process that currently applies for natural gas appliances as outlined in Australian Standards was followed for enclosed firing of the Tempest burner and the Riello package burner. This process showed that appliances could be operated using the start gas requirements in AS 3814:2018, but the lower stoichiometric requirement of hydrogen means that very high flue oxygen concentrations are required to avoid an explosive mixture if the flame is lost.

As the fuel’s hydrogen fraction increases the flue gas water vapour concentration increases, and carbon dioxide concentration decreases. This leads to an increase in flue gas dew point. For natural gas, the dew point is approximately 59^{o}C, for 20vol% hydrogen blend it only increases very marginally to 60^{o}C and for 100% hydrogen it is 73^{o}C. Process operators need to keep the flue gas temperature well above the dew point to avoid condensation and subsequent corrosion. Experiments did not investigate this since the data is clear. Using up to 20 vol% hydrogen will make no noticeable difference, but for 100% hydrogen the flue gas temperature will have to kept well above 73^{o}C, or condensation will occur, and there will be greater volume of condensate than would occur with natural gas.

The overall summary is that there is a high level of confidence that Type B appliances may be operated with no changes, or minor changes, when up to 20 vol% hydrogen is blended into natural gas, which is the upper limit proposed for gas network blends. Any small changes in fuel and air flow-rates or pressures will typically be performed by existing automatic temperature control. For other blends and 100% hydrogen, changes required are summarised in the table below.

vol% H2 in blend	Operation observations	Adjustment or change required
0-20%	Normal operation of all burners	Automatic temperature control on fuel rate. Air rate may need slight adjustment
20% - 40%	Beware flashback in premixed burners	Limit primary air on premixed burners
40% - 50 %	Beware flashback in premixed burners	Replace premixed burners with nozzle mix
50 % – 100 %	High burner metal temperatures	Modified or new hydrogen burners
> 70%	Ionisation rod flame detector perform	Replace with UV flame detector
>80%	Potential for high NOx emissions	Replace with low NOx hydrogen burners
90 -100%	Slight reduction in heat transfer	Design hydrogen burners for required duty. Potentially blend with 10% biomethane.

RP1.4-08
Milestone 3: Interim report and preliminary results

August 1st, 2022

Project number: RP1.4-08

Detailed assessment of Type B appliances with blends of hydrogen and natural gas and 100% hydrogen

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Summary of Report

This short report contains details of some of the background work that has led to the project and some preliminary experimental results. The project plan has also been updated and is included in this report.

The motivation of the project is to provide confidence that Type B appliances in hydrogen injection trials will be compatible with 10 and 20 mol% hydrogen blends and determine if a higher intermediate %hydrogen blend is practical as part of a transition to 100% hydrogen. It will also provide insight into changes required to use 100% hydrogen in Type B appliances. The knowledge gaps are:

- Are there any issues with 10 - 20 mol% hydrogen blends in a wide range of burners and appliances?
- Is there a maximum grid injection hydrogen% with no changes or minor changes to equipment?
- Can burners and appliances run on 100% hydrogen?
- Can testing and demonstration be done economically?

The impacts of this project will be demonstrated by closing these knowledge gaps. The specific aims and expected outcomes of the project are related to the knowledge gaps.

Aims:

- Ensure planned 10 mol% hydrogen injection pilot trials can proceed and determine if any changes are required.
    - Tests will include an assessment at approximately 20 mol% hydrogen for a range of types of burners and appliances and their systems, to provide a confidence margin. Assessments will include performance, emissions, safety, control (flow control, flame detection, BMS), product quality.
- Determine if there are relatively simple modifications that can extend the blending range since some end-users may have access to hydrogen and could blend at higher proportions than supplied via pipeline.
- Investigate economic retrofit options for some appliances for 100% hydrogen duty.
- Provide feasibility studies (cost estimates etc.) for large scale end-user trials
- Information sharing of results

Expected Outcomes:

- Confidence with changes to 10 mol% hydrogen / natural gas blend
- Definition of an upper limit in blend ratio with minor changes such as changes to supply pressure or orifice size.
- Description of blending options for progression to 100% hydrogen (e.g. 10 or 20% and then directly to 100% or the circumstances under which an intermediate step may be useful)
- A legacy portable blending station for future on-site testing
- Information for regulators, installers, servicers and end-users re updates, modifications or operating procedures
- A model to predict burner compatibility and required changes
- Reports and journal papers as outlined in the project agreement

The original project plan, as described in the project proposal is summarised in Appendix 8.3. The original plan was reconsidered because of logistical issues and because it became clear that smaller burners and appliances than originally planned could be tested at University as the first step in the plan. The logistical issues related to the original plan to test large appliances offsite where overnight hydrogen storage was not possible. This would result in time wasting in packing up, setting up and travel twice per day.

The use of small burners at the University of Adelaide also meant that the rate of hydrogen use would be less and it became possible to also consider 20 mol% blends as a new upper limit for the pilot regions instead of 10 mol% hydrogen as in the original plan. By doing this some tests will be conducted at 31% to provide a large safety margin for 20 mol% injection trials. Use of the small burners at the University of Adelaide also allowed

testing for maximum hydrogen percentage and 100% hydrogen at same time as tests with 10 -20%, i.e., while the burner is set up. These changes have been approved in a project variation.

The plan involves testing a range of small burners and appliances in open firing tests in 2022, followed by testing the same burners in confined firing in early 2023. Testing of some larger burners and appliances will occur in the second half of 2023. An onsite demonstration in an industrial furnace will be conducted as soon as practical in the program, at this stage it is planned for 2023. It will involve some in baseline tests with natural gas before moving on to hydrogen blends later. A model for combustion performance will be developed in parallel with the experimental parts of the program. Planning and costing of larger on-site tests and provision of information and results will be performed in 2024.

The current report also includes some preliminary results from open firing of an inspirated “FR” nozzle. These results are preliminary and presented to illustrate some of the techniques involved. Results illustrate the similarity between the appearance of, and NOx and CO emissions, from natural gas flames and those using 10, 20 or 31 mol% hydrogen blends. The results also illustrate the very different appearance and higher NOx emissions of the 100% hydrogen flame. More detailed tests will be done in the second half of 2022 and results presented in December 2022.

2 INTRODUCTION

2.1 Project Motivation

2.1.1 Provide confidence that Type B appliances in hydrogen injection trials will be compatible

Operators of hydrogen grid injection trials in Australia require confidence that 10 mol% hydrogen blends with natural gas will be effective and safe. There are several pilot projects in Australia, which include commercial and industrial equipment. Hydrogen grid injection trials are starting with 2 – 10 mol% hydrogen blends but could extend to 20 mol%. Examples are:

- AGIG hydrogen injection trials at Tonsley Park, Gladstone (Queensland) and Albury Wodonga.
- Jemena JGN network including 7 industrial users (brick kilns), 39 commercial users (Bakeries, Distilleries, Kitchens etc) and residential users. Larger customers have already shown support for further research (Panek et al. 2020).
- AusNet has a blending project underway in Ballarat.
- ATCO's Clean Energy Innovation Hub and Park in WA includes some engines, but future projects are likely to also include other commercial appliances.

Hydrogen has wide flammability limits, lower stoichiometric air requirement, and higher flame speed, as comparted to natural gas. These properties have been well documented, including in the final report of RP 1.4-02 (Panek et al. 2020). For addition of 2 - 20 mol% hydrogen to natural gas, the effects of the small increase in flame speed and minor changes in the stoichiometry are most likely to be insignificant for many industrial atmospheric burners because they generally have a wide operating range. Emissions and heat transfer may be affected to a small degree. For appliances with partially premixed or premixed burners there will be slightly more chance of light-back with these small quantities of hydrogen, so it will be prudent to check a representative sample, especially at low rate, where light-back is most likely. Some appliances might require a slight increase in gas rate or pressure, but only if already operating at full rate, because adding 10 or 20 mol% hydrogen to natural gas slightly reduces the heating value. In many cases the heating value reduction will not be noticeable. Demonstration that the effects are minor is required for confidence in operation. Confidence also needs to be gained in flame detection and control systems with these levels of hydrogen addition.

Pressurised combustion in turbines, microturbines, engines and fuel cells have been previously reviewed (Panek et al. 2020). These pressurised combustors all have a limited capability, without modifications, to operate with addition of hydrogen to natural gas or with variable fuel composition. Panek et al. (2020) detailed a range of turbine capabilities with different capabilities for handling hydrogen blends. Some have very little capability without modifications. Likewise, natural gas reciprocating engines may be limited to 5 mol% hydrogen, although modern air fuel ratio control systems may extend this. On the other hand, with modifications any hydrogen blend can be accommodated. For example, the Clark energy website describes different conversions of natural gas engines to 25 mol% hydrogen or to 100% hydrogen[1]. Capstone microturbines can operate well beyond 10% hydrogen[2]. The recurring theme is that the manufacturers of pressurised combustors have been developing retrofits and new designs for any hydrogen blend ratio. The consequence for hydrogen injection trials is that pressurised combustors will need to be identified and operational capability or requirements for changes will need to be discussed with suppliers and upgrades to equipment carried out where necessary.

The current project focuses on atmospheric combustion appliances rather than pressurised combustion. In contrast to pressurised combustion, many of the burner types used in atmospheric combustion are expected to be suitable for up to 20 mol% hydrogen blends without modification, or with only very minor changes. Some types are also likely to be suitable for 100% hydrogen with modifications, while other types may not be suitable for 100% hydrogen at all.

There may be some regions or industries where enough hydrogen may become available to move rapidly to 100% hydrogen, or perhaps an intermediate concentration between 20% and 100%. For 100% hydrogen, burner designs and operation may need to be changed completely. Blending to between 20 and 100% hydrogen is less

1 https://www.clarke-energy.com/gas-engines/
2 Risk Validation Report for HyP Murray Valley Market/Hydrogen Appliance Study Project, GPA, 2022.

likely but may be applicable for operations with access to 100% hydrogen, thus allowing on-site blending at higher levels of hydrogen addition. Hence, there is merit in determining a maximum hydrogen blend % with minor or no changes for some burner designs and in gaining confidence in operation with these blends and with 100% hydrogen. Small grid trials with >20mol% or 100% hydrogen may occur in the future and thus results will be useful for those trials.

2.1.2 Conduct R&D in Australia to address local issues and contribute to the global effort

There have been grid injection trials overseas, some testing of appliances with blends of hydrogen and natural gas, and some development of 100% hydrogen appliances. However, many of the programs are incomplete and there are some differences in the types of appliances used, so Australia needs to investigate issues of importance locally and also contribute to the global effort so as to be ready for the transition to hydrogen.

Many research programs and grid trials were reviewed in the reports of projects RP1.4-02 and RP1.4-06, but it is worth highlighting some of these programs and the key results, which informs the scope of the current work.

In Australia, ITP Thermal Pty Ltd (2019) examined the options for replacement of fossil fuels with renewable resources in the generation of process heat for the Australian industry. The potential for green hydrogen was seen to be most likely for ammonia production in the Direct Reduced Iron (DRI) process, but costs of renewable hydrogen may be a barrier in alumina refining, cement manufacturing and other industries.

The European Gas Research Group (GERG) prepared for hydrogen grid injection by reviewing likely impacts on pipelines, pressurised combustion, atmospheric combustion, vehicles and CNG storage. For atmospheric combustion they raised potential issues with Wobbe Index, flame length and NOx emissions, which are now well understood. The latter two will be studied experimentally in the current project.

Hodges et al. (2105) assessed the suitability for hydrogen/natural gas blends up to 20 vol%. They suggested that forced draft burners were likely to be more suitable and that natural draft burners may be more susceptible to lightback or overheating of the burner due to the flame burning too close to the tip. Both types of burners will be studied the in the current project.

A European consortium (THyGA - Testing Hydrogen Admixtures for Gas Appliances), established in 2020, is assessing both domestic and commercial appliances with blends of hydrogen and natural gas with low (<10 vol%), medium (10-30 vol%) and high (>60 vol%) levels of hydrogen in the blends - although more recent information suggests that their main focus will be at <30%. The types of industrial and commercial appliances included are standard boilers and condensing boilers, water heaters, space heaters, combined heat and power systems (with internal combustion engines or microturbines, fuel cells, Stirling engines), commercial dryers, air heaters and infrared radiant heaters (Schaffert, et al., 2021).

THyGA developed a testing protocol through an iterative process with input from many experts. It was noted that the percentage hydrogen addition will have a different effect depending on the Wobbe number of the original natural gas it is blended with, so there was discussion about using both “EU low” and “EU high” natural gases (which have quite different Wobbe number) for the blending tests. Some of the tests are deemed mandatory (i.e. agreed as part of THyGA funding). These are:

- Safety, emissions, and efficiency tests
- Operation and safety for different adjustments (light-back and CO emissions)

Other additional tests that will be done are:

- Extreme conditions – e.g., cold start, hot start, low air temperature
- Flame detector ionisation signal measurement
- Light back analysis
- Delayed ignition test
- Soundness of the gas circuit (no leaks)
- Qualitative response to quick variation in flow and shut-off
- Metal temperatures to check for overheating

Hy4Heat is a project commissioned by the UK Department for Business, Energy and Industrial Strategy to assess the issues of conversion of industrial gas appliances in the UK to 100% hydrogen. It deals with commercial and industrial appliances in two separate work packages. The demarcation between commercial (WP5) and industrial (WP6) was made at a heat input rating of 1MW and, in addition, commercial heating was restricted to appliances supplied from the network at less than 7 bar. A review under the Hy4Heat WP5 (Ryan & Iliffe, 2020) lists only the following types of appliances as "commercial" in the UK:

- boilers
- radiant tube space heaters
- luminous ceramic radiant heaters
- direct- and indirect-fired warm air heaters and water heaters (storage, circulating and instantaneous)
- commercial catering appliances and tumble dryers in commercial laundries
- gas engine driven heat pumps.

The Hy4Heat WP5 review found that there are a wider variety of burner types in commercial appliances than in domestic appliances, and normally performance needs to be more tightly controlled. Boilers use either pre-mix burners in condensing boilers, or package burners[3] in both condensing and non-condensing boilers. It was considered that it will not be economical to convert non-Hydrogen-ready boilers to hydrogen and they will need to be replaced (Ryan & Iliffe, 2020). However, it was considered that all other appliances are likely to be able to be converted to 100% hydrogen appliances cost-effectively. This is a conclusion that is significant for the current FFCRC project with respect to the investigation of retrofitting appliances. It is also worth noting that boilers are a much larger part of the commercial appliance landscape in the UK than they are in Australia.

Hy4Heat WP6 deals with industrial applications and includes many types of dryers, ovens and kilns. A report from Element Energy, Advisian (Worley) and Cardiff University outlined the estimated costs of conversion to 100% hydrogen (Durusut, 2019). For a range of industries, the average estimates were approximately £150,000 for 1MW or £600,00 for 10 MW. However, since Hy4Heat is only concerned with 100% hydrogen, it did not consider the potentially lower costs of retrofitting for blends of hydrogen and natural gas. The types of industrial natural gas users (in order of most frequent to least frequent) in the UK were identified as:

- Boiler (steam)
- Oven
- Boiler (hot water – they call these "boilers" even though they are in-fact water heaters)
- Direct dryer
- Kiln - ceramics (e.g. bricks, tiles and tableware)
- CHP
- Furnace - metal melting, metal rolling including reheating, and metal treatment
- Furnace - glass
- Kiln - lime
- Indirect dryer
- Kiln - other

Potential issues for 100% hydrogen were identified as:

- Low radiant heat transfer may be a limiting factor in glass furnaces, ceramics kilns and lime kilns
- Low convective heat transfer due to lower air requirement. The use of flue gas recirculation or higher excess air could be considered, but will also lower temperatures
- Water in the flue gases might damage some products
- NOx emissions may be higher and this needs to be investigated
- Some users require onsite demonstration trials (particularly if direct fired), but others are happy with guarantees from OEM testing (particularly indirect fired)

Many grid injection trials have commenced in Europe, but with little involvement of industrial of commercial customers. Stage Two of the HyDeploy project includes one school and some small businesses at up to 20 mol%

3 Package burners have their own fan and control system and typically a nozzle mix burner, but sometimes use a partial premixed burner.

hydrogen[4]. The H21 Leeds City Gate included an assessment which concluded that converting some industrial burners to 100% hydrogen may require changing to non-premixed mode and may increase NOx emissions or require additional abatement measures [44]. This is an issue that will be addressed in the current work.

In previous FFCRC projects, common types of Type B burners and furnaces were reviewed (see Appendix 8.1), and a compilation of Type B appliances found in hydrogen injection sites and throughout Australia was made (see Appendix 8.2). These Appendices are included here to illustrate the variety of appliances and burners in operation in Australia, which shows that judicious choices of the most representative burner types to test will be necessary.

In addition to the reviews and grid injection trials, many burner manufacturers have already developed burners suitable for up to 30% hydrogen in natural gas, and other burners are under development for higher concentrations of hydrogen. Some burner manufacturers are developing products for 100% hydrogen. Examples include Honeywell, who have a range of hydrogen-ready and hydrogen-capable (i.e., small changes required) burners and control systems covering industries as diverse as incineration, paint bake ovens, process heaters, dryers, bakery ovens, coffee roasters, glass melting, metal melting and treatment, dryers in the pulp and paper, construction, and textile industries[5].

The burner types that are more readily adaptable to 100% hydrogen duty are diffusion, duct, and nozzle mix burners. This includes oxy-fuel burners for glass kilns and melting applications, radiant tube space heaters, but not premixed burners. Fives Combustion, have demonstrated “Pillard LONOxFLAM® G2” burners for 100% hydrogen duty in a chemical plant[6]. This is an example of being able to adapt a nozzle mix burner to hydrogen. In general, adaption to 100% hydrogen should be possible for all nozzle mix burners, but in many cases may also require changes in nozzle geometry. Performance and emissions would then have to be optimised for each process and may not be suitable for all processes. Domestic and small commercial 100% hydrogen boiler prototypes have already been developed by Bosch and Baxi, amongst others.

In summary, there is a substantial interest in identifying potential issues when adapting existing burners to hydrogen blends and 100% hydrogen. Burner and appliance suppliers are very active in this area. However, much of the work is ongoing and incomplete and the types of appliances being investigated overseas does not match the range in use in Australia, although there are some that are the same. The FFCRC can contribute to global effort to demonstrate compatibility of appliances with initial blending trials at up to 20 mol% hydrogen and research and demonstration of changes required for >20mol% hydrogen and 100% hydrogen.

2.2 Knowledge gaps:

2.2.1 Are there any issues with 10 - 20 mol% hydrogen blends in a wide range of burners and appliances?

For blending of 10 mol% hydrogen into natural gas, existing appliances are likely to operate well without changes. However, since there are so many different types of appliances it is impossible to test them all. It is, however, possible to test common burner types and control system types. At such low levels of hydrogen addition, flame stability and appearance, heat transfer and emissions should all be similar to natural gas, but this needs to be confirmed. Tests at 21.7 mol%, or similar, provide further confidence in operation at 10 mol%. Tests at 31 mol% provides confidence for operation at 20 mol%.

As illustrated by the review of studies and grid injection trials, other projects have cross-over but are testing some appliances that are not as relevant in Australia. Burner suppliers are doing their own development, but demonstration is important too, and this project will include some trials and demonstrations in real processes and preparation for larger demonstrations.

[4] https://hydeploy.co.uk/about/

[5] https://thermalsolutions.honeywell.com/content/dam/hpsbt/en/documents/downloads/HONEYWELL-THERMAL-SOLUTIONS-Harnessing-the-promise-of-hydrogen-to-reduce-global-emissions.pdf

[6] https://combustion.fivesgroup.com/ko/news/combustion-news/expertise-in-hydrogen-combustion.html

2.2.2 Is there a maximum grid injection hydrogen% with no changes or minor changes to equipment?

It may be possible to suggest a maximum above the 20mol% blend that can be used for all types of burners or a limited region without changes, and potentially a higher maximum with minor modifications. The maximum will be set by the burner type which has the lowest tolerance for hydrogen blending, probably partially pre-mixed or premixed burners. The question is how much hydrogen can be blended to natural gas before some types of burners become susceptible to unacceptable performance such as lightback at low rates or reduced heating capacity or failure of control systems or flame detection. Minor changes to overcome a particular limiting issue would include options like increasing injector sizes, or increasing pressure, or perhaps tuning controllers or changing to a UV flame detector. The knowledge gained will be useful in understanding the transition options.

2.2.3 Can burners and appliances run on 100% hydrogen?

For partially premixed and premixed combustion, for which conversion to 100% hydrogen duty is difficult, burner suppliers may look to different types of burners for 100% hydrogen. However, burner suppliers are confident in being able to offer diffusion, duct, and nozzle mix burners that can operate with 100% hydrogen. In each case operation will need to be tailored to the needs of the process for heat transfer, mixing, and emissions. Air flows will need to be adjusted from natural gas versions of the burners and this will have a large effect on operation, so process performance will depend on how the burners are operated and where air is added or flue gas recirculated. Hence the knowledge gap is not whether these types of burners can be adapted for 100% hydrogen, but what steps need to be taken to use them effectively. This can be assessed by modifying burners and testing them under different operating conditions and measuring heat transfer and emissions. Tests in a limited number of furnaces and appliances can then also be done.

2.2.4 Can testing and demonstration be done economically?

A further gap regards whether Type B or industrial equipment can be tested economically, while producing useful results. The estimated costs for retrofits to furnaces, kilns, dryers, and ovens identified in a Hy4heat WP6 report illustrates that it would be impossible to fund large demonstration projects for 100% hydrogen conversions within the current Future Fuels CRC budget. However, the testing program outlined here provides a useful way of testing a wide range of burners and appliances.

2.3 Scope and reasons for changes from the original plan

The original project plan, as described in the project proposal, is summarised in Appendix 8.3. The plan has been altered and the current plan is detailed in the sections below. The intended impacts, aims and outcomes have not changed, except for addition of initial testing at 20 mol% hydrogen and additional effort to demonstrate 100% hydrogen burners.

The original plan was reconsidered because of logistical issues and because it became clear that smaller burners and appliances than originally planned could be tested at University as the first step in the plan. The logistical issues related to the original plan to test larger appliances offsite where overnight hydrogen storage was not possible. This would result in time wasting in packing up, setting up and travel twice per day.

The use of small burners at the University of Adelaide also meant that the rate of hydrogen use would be less and it became possible to also consider 20 mol% blends as a new upper limit for the pilot regions instead of 10 mol% hydrogen as in the original plan. By doing this some tests will be conducted at 31% to provide a large safety margin for 20 mol% injection trials. Use of the small burners at the University of Adelaide also allowed testing for maximum hydrogen percentage and 100% hydrogen at the same time as tests with 10 -20%, i.e., while a burner is in position. These changes have been approved in a recent project variation. The main reason for the change to operate at smaller scale in the initial experiments is to be more productive in terms of the number of burners that can be tested and the number of tests that can be performed.

Commencement of experiments has been slightly delayed as a result of changes to the plan, the time required for ordering and delivery of the small burners, and for the project team to prioritise two other shorter projects relating to Type A appliances (i.e. RP 1.4-07 and RP 1.4-01C) which were added after the current project was originally scheduled. The current report includes some preliminary results, the main purpose of which is to demonstrate that experimental work had just commenced.

The scope is limited to atmospheric combustion appliances, not pressurised combustion as has been noted already.

2.4 Impacts, Aims and Outcomes

The impacts of this project are to:

- Gain confidence for operation of Type B appliances with blends of 10 vol% hydrogen in natural gas. (Extend this to 20 vol%).
- Recommend an upper blend limit – with minor changes and without changes – this may be useful for some sectors or regions or types of equipment.
- Options for progression to 100% hydrogen - this could identify types of appliances or sectors or regions or pilot trial areas that may be suitable for earlier deployment of 100% hydrogen than others.
- Preparation for on-site tests

Aims:

- Ensure planned 10 mol% hydrogen injection pilot trials can proceed and determine if any changes are required.
    - Tests will include an assessment at approximately 20 mol% hydrogen for a range of types of burners and appliances and their systems, to provide a confidence margin. Assessments will include performance, emissions, safety, control (flow control, flame detection, BMS), product quality.
- Determine if there are relatively simple modifications that can extend the blending range since some end-users may have access to hydrogen and could blend at higher proportions than supplied via pipeline.
- Investigate economic retrofit options for some appliances for 100% hydrogen duty.
- Provide feasibility studies (cost estimates etc.) for large scale end-user trials
- Information sharing of results

Expected Outcomes:

- Confidence with changes to 10 mol% hydrogen / natural gas blend
- Definition of an upper limit in blend ratio with minor changes such as changes to supply pressure or orifice size.
- Description of blending options for progression to 100% hydrogen (e.g. 10 or 20% and then directly to 100% or the circumstances under which an intermediate step may be useful)
- A legacy portable blending station for future on-site testing
- Information for regulators, installers, servicers and end-users re updates, modifications or operating procedures
- A model to predict burner compatibility and required changes
- Reports and journal papers as outlined in the project agreement

3 PROJECT PLAN

3.1 Open firing of a Flame Retention (FR) nozzle

Preliminary tests were conducted at University of Adelaide, Thebarton campus, and are reported in section 3.

Flame Retention (FR) nozzles are a simple entrainment type of burner which is ideal for initial tests – to test ignition, stability, flame length and flame shape (video) and flue emissions (CO and NOx vs CO_2 and oxygen) at various H_2 percentages from 0 – 100%). The FR nozzles used will have a rating of approximately 30 kW at lower end of its normal pressure operating range. The nozzle is a short pattern inspirator, FR 8, with a nominal gas pressure range of 0.75 kPa to 6 kPa.

Open firing tests with NG, 10mol% H_2, 21.7 mol% H_2, and 31 mol% H_2 blends are for confidence with up to 20% in practice with the 31% blend included to give additional confidence. Tests will be performed at the nominal minimum gas pressure to establish techniques and assess effects of hydrogen blending. For each different blend the pressure and flow-rates will be slightly adjusted to maintain constant minimum net heat input because users may operate with similar heating rates to current rates if using a blend or 100% hydrogen – instead of maintaining constant pressure.

Experimental Plan:

- Test flame stability, flame length and flame width (long exposure photo or video) and flue emissions (CO and NOx vs CO_2 and oxygen) at various H_2% from 0 – 100% (Note - more detailed tests with measurement of heat flux are not included at this stage).
    - The photos will be descriptive because as % hydrogen in the blend increases the flames are less visible than natural gas due to reduction in soot radiation. The purpose of the photographs are to image flame length and width, and provide a qualitative visual comparison.
    - Test flame length and emissions for 0%, 10%, 21.7%, 31% H_2 blends
    - Test 50%, 70%, 80%, 90% and 100% hydrogen blends without adjusting air if they don't light-back
    - If light-back occurs, then tests can be re-run with reduced primary air to prevent light-back

3.2 Open firing tests with other small burners and appliances

These tests are planned for late 2022, Reported by Dec 20, 2022. For these trials burners with capacities below 500MJ/h (140 kW) are to be used. The appliances will be:

- Partial premix inspiratory - pizza oven burner (radiant) with AN style nozzle 35 kW (120 MJ/h) AVANZINI
- Forced draft high velocity nozzle mix burner 65 MJ/h (~20 kW) max, air pressure of 7 kPa – North American Tempest 4442 – narrow flame
- Forced draft premixed – Riello small package burner BS2F with integral fan. Rated at 35 – 92 kW with a 106 mm diameter burner tube
- Eclipse air heat burner – rectangular mounting – produces hot air (not flame) 25 – 500 MJ/h = 7-139 KW
- Raypak B0200 water heater, 56 kW (200 MJ/h)
- A small premixed radiant burner
- A pilot burner will also be trialled if available, e.g., Kromschroder (10.8 MJ/h)
- Flame rods and UV detectors integral to some of these burners will also be tested

The burners to be tested are either new equipment or used equipment available on loan from collaborators, GasTrain, Furnace Engineering, or Ignite. These burners include different types of partially and fully premixed and nozzle-mix burners that are the main types of burners used in Type B appliances such as those found in hydrogen injection trials. The tests will investigate ignition (normal, delayed, and at maximum turn-down), light-back (at normal rates and at maximum turndown), leakage, and flame stability. Each of these tests will be performed with natural gas, 10 mol% hydrogen in natural gas, 21.7 mol % hydrogen, and up to a maximum % hydrogen (without changes to any equipment). The "maximum" will be determined by a safe limit prior to light-back or instability by monitoring flames as % hydrogen is increased. The testing procedures will be similar to those for the FR nozzles, and again cover tests with NG baseline, 10%, 21.7%, 31% H2, maximum H2% with and without changes and operation at 100% hydrogen with necessary adjustments, where it is possible to

achieve 100% firing. Tests will be performed at min and maximum pressures in the normal operational range for each burner. Prior to initial tests, further discussion with combustion experts will be undertaken to refine the program.

For water heaters and other appliances and processes to be tested throughout the program, in which the control system works to a setpoint process temperature, the performance will be checked by recording fuel pressure, flow and the process variable temperature achieved.

The open firing assessments will include:

- Leakage rates of valve trains and burners used in the experiments will be tested using the procedure in AS 5601.1:2013 (Appendix E), which calls for ensuring that a higher pressure than the normal operating pressure can be held for 5 minutes. However, in this case the time will be extended to one hour, and any leakage will be determined by pressure reduction as a function of time. Tests will be done for a natural gas baseline and blends of 10 mol% and 21.7 mol% hydrogen in natural gas. A hydrogen leak detector will be used to find any leaks and remedy them, allowing for a pressure re-test. Tests could also potentially be performed with 100% hydrogen. These leak tests will only provide an early indicative assessment of whether using hydrogen blends in existing valve trains is likely to cause significant leakage problems. It may not be widely applicable because each valve train contains a different range of materials and components but may assist gauge the likely significance of leakage. It will provide experience for testing larger and more complicated valve trains at industrial and commercial sites.
- Ignition and stability at different conditions including cold conditions, high or low air-fuel ratio
- Flame length and width
- Emissions – NO, NO_2 and CO, CO_2
- Light-back at normal conditions and turndown
- Determine maximum H_2% possible in a blend with no changes to equipment – for example where there is a gas-air ratio controller it will be operated without a change.
- Assess maximum hydrogen % with changes (such as air reduction) at constant heat input
- Radiant heat flux from limited flames for most of the burners
- Instrumentation and valve operability
- Flame detection operation for different types, and ionisation signal strength for those with ionisation rods, UV detectors integral to some of these burners will also be tested.
- Effect of rapid variability in %hydrogen on stability
- Assess simple retrofits that may be applicable for operation of equipment with 100% hydrogen (for example air reduction, gas injector diameter, gas pressure), and radiant heat transfer and NOx emissions. Where retrofits are successful establish operation ranges (hydrogen pressure range and hydrogen / air ratio range)

The December 2022 report will also include results of discussions, reviews of other work and supplier documentation on the effects of hydrogen addition on flue gas condensation, temperature and amount.

3.3 Confined firing tests and initial model development

These tests are planned for 2023, with an interim report by June 30, 2023, and further report by Dec 20, 2023 (with other results). The reports will also include information of development of a model. The June 2023 report will include preliminary work on development of a predictive model, which will at this stage only be to model the results observed.

To test the effect of confinement and to assess heat transfer and emissions, the small burners used in previous open firing experiments (except for the premixed radiant burner) will be fired into a thin wall 2m long, 200mm diameter stainless steel duct. The thin wall is important so that temperatures reach a steady state rapidly to reduce experimental run-time and minimise bottled hydrogen use, which is important because the capacity of hydrogen bottles is much less than other fuels. These tests will require a flame detector and burner management system. Temperatures will be assessed using an IR imaging camera and multiple thermocouples mounted on the duct wall. A heat flux sensor can also be used for measuring radiation from the duct to the room and could also be used to measure flame radiation at various locations if suitable openings are made in the duct. Flue temperature and emissions will also be measured. These tests will all be done at firing rates to allow the tube to reach high temperatures so that radiation from the tube to the room is significant.

For the package burner a test may also be conducted in a closed loop water boiler, which will provide information on operation in a typical appliance, with the required heat load. In this case the output measurements are water temperature and emissions, so there are less details than from the 200 mm duct, but it provides a good example of a practical application.

A literature review of heat transfer measurement methods and calculations of convective and radiant heat transfer will be conducted prior to the confined flow tests to assist in preparation of the measurement methods and decide if openings will be made in the duct to use a heat flux sensor for direct flame radiation, or if this will not be necessary.

The experimental plan for the first half of 2023 is:

- Initially perform tests at minimum rates while the stainless steel remains in shiny state. Approximately 10 kW heat input will ensure that the stainless steel will not discolour. Repeat the turndown tests done in the open, but this time firing into the tube at approximately 10 kW for most of the burners with NG, 10%, 21.7 and 31% hydrogen.
- Then oxidise the tube to transform it into a high emissivity tube. To do this it will to be operated with steel at approximately 1000^{o}C for some time by running much higher natural gas rate. After oxidation the tube will lose more heat to the surroundings by radiation and can then be operated at fuel inputs of 30 – 50 kW. Experiments can then be continued experiments with rates around 30 – 50 kW with high wall temperatures. At these rates many of the small burners can be operated at full rate.
- Alternatively, if this technique is not useful, the 200mm tube with multiple wall temperature measurements can be used in a water bath for higher heat transfer rates. A decision on which method to be used will be made at the time.
- Test each of the small burners with NG, 10 and 21.7%, 31% and maximum % hydrogen blends initially with no change to inlet pressure. This will mean a slight reduction in heat rate when hydrogen is blended into natural gas. Measure wall temperature profiles with thermocouples and IR camera, and flue gas temperature and emissions.
- Then repeat with NG, 10 and 21.7%, 31% and maximum % hydrogen blends allowing increases in pressure to maintain constant heat rate. Compare results to the constant pressure case to assess the relative importance of heat rate and hydrogen blend composition on heat transfer and emissions, and the potential implications for processes. Decide if can recommend that don't need to make changes to allow for 10 and 21.7% - or small changes.
- Perform confined tests on any successful simple retrofits for 100% hydrogen from the open firing tests at both ends of operation ranges (hydrogen pressure range and hydrogen/air ratio range) and measure heat flux and emissions.

The experimental plan for the second half of 2023 is:

- Establish maximum hydrogen % with small changes for burners that would limit grid hydrogen injection %, such as partial premixed burners.
- For burner types that allow it, re-test a 100% hydrogen flame, with change / retrofit to adjust heat transfer to be like a natural gas flame.
- For 100% hydrogen flames that were shown to have high NOx emissions, test a low NOx retrofit – either a new low NOx burner or alter air-fuel mixing or test flue gas recirculation (which can be simulated by using hot flue gas from a small natural gas burner).
- Confined firing may also allow for testing of control systems and purging. AS 4625:2008 (Electronic flame safeguards and flame detectors) describes flame safeguard systems including the provision for flame detection, flame proving and purging. An assessment of the need for any changes required to purge times for different fuel blends will be made. An assessment of common flame detection and flame safeguard or burner management system (BMS) controllers will also be made.
- Newer appliances use microprocessors BMS and process controllers (such as LMV Siemens). An assessment of how they deal with, or can be programmed to deal with, variable hydrogen % in the fuel will be undertaken. Information on gas composition may be required, or potentially a measurement of density could potentially be used to infer it. Some older appliances still use mechanical linkages for air / fuel mixing by controlling air and fuel valves in predetermined ratios.

These are less accurate than microprocessor controllers, and provide poor control at turndown, resulting in high excess air and hence poor efficiency. They will be phased out. However, there are enough left in operation to warrant assessment and testing with hydrogen / natural gas blends. The performance of ionisation rod, UV and IR flame detectors with the range of hydrogen blends will be tested.

3.4 Tests with burners rated 500 MJ/h and modelling

This work will take place in the second half of 2023 and reported in Dec 2023 along with the report on operation of smaller burners at maximum rate with retrofits and 100% hydrogen with retrofits. Development of a combustion model based on experimental results and using correlations for prediction of performance, failure modes and operational limits will be conducted and reported.

Preparation for a combustion trial at one full-scale test site (A burnoff furnace at Furnace Engineering in Melbourne) will be undertaken and reported. This furnace operates c, so a trial can be organised with 10% and 21.7% hydrogen blends and results compared to a natural gas baseline. Other site-tests could be added – for example, detailed measurements at sites within a 10% hydrogen blending project. Plans for such tests will be reported on in Dec 2023, or earlier, if the opportunity arises earlier.

Burners and appliances that differ from those tested earlier in the project will be selected for testing at larger scale, ~ 500 MJ/h, which although smaller than some real-world applications, is expected to be representative. Discussion with the project team will finalise which burners and appliances and gas blends are to be tested. More detailed information on the types of burners and appliances in use in Australia may become available at this stage of the project and further options may be added. It is likely that tests with up to 30 mol% hydrogen will be performed. The following burners / appliances are likely to be selected:

- A condensing boiler
- A forced draft nozzle mix burner – either an example available to Gas Train or a medium velocity Maxon Kinemax 160 kW) (from Techrite) if considered significantly different from others tested
- Linear air heat style (from All Controls)- Eclipse AH0050, 25 – 500 MJ/h (already tested at low rate in first set of trials)
- Maxon, NP1-LE duct burner (from Techrite) > 500 MJ/h even for a 12” length
- Raypak B0538 on/off water heater, (500 MJ/h) and/or Raypak Modulating Vertical Burner Water Heater, MV910500H (500 MJ/h)
- A modulating or two stage package burner (for example, one is available at Gas Train, but it is large, so a smaller one may be found or purchased)

3.5 Site tests, information, modelling, and full-scale costings

A report will be provided by June 2024 covering the following:

- A site test will be conducted at Furnace Engineering in Melbourne (or potentially elsewhere) – testing a 60kW burner in a burn-off furnace. The purposes of this trial are to demonstrate whether any adjustments are necessary in this example of a practical furnace with 10 and 20 mol% blends of hydrogen and natural gas, and to determine if a 100% hydrogen burner design can produce the required performance. The tests will require an initial visit to survey the site and decide what will be measured and what additional instrumentation and equipment is required to provide useful data from the trials e.g., input flowrates, pressures, measured temperatures, emissions and defining a performance measurement. After preparation, a trial will be conducted at baseline conditions with natural gas using the same instrumentation as will be used for subsequent runs with hydrogen blends. A range of different operating conditions can be run during this trial using natural gas only. Data from the natural gas trial will be analysed and a 100% hydrogen burner will be purchased (or designed and built), a subsequent trial running the furnace with 10 vol% and 20 vol% hydrogen blends using the natural gas burner, and with 100% hydrogen using the new burner. Finally further data analysis and reporting of the results will be done.

- Completion of development of the combustion and heat transfer model based on experimental results and using correlations for prediction of performance, failure modes and operational limits (latter two terms from End-user Roadmap).
- Results of model predictions and cost estimates for demonstration tests at small- and large-scale sites with a 10 mol% hydrogen blend.
- A design and costing tool for modifications to key burner types for different blend ratios and for 100% hydrogen (for those where it is practical and potentially economic).
- Cost estimates of conversion/retrofit of appliances for 10% vs maximum % hydrogen with minor changes and for retrofits to 100% hydrogen vs new 100% hydrogen appliances (currently being developed), and in some cases, comparison to electrical appliances.
- Details on any changes required to equipment or operating procedures.
- Recommendations for updates in appliance standards will be made for 100% hydrogen appliances. Discussions with relevant Standards committees will be undertaken on recommendations.
- Training on any updates.

A report will be produced by Dec 20, 2024, including:

- Implications for Standards and a description of a legacy portable blending station for future on-site testing
- Preparation of a proposal or other funding arrangements for on-site demonstrations to be performed outside the current project, by end of 2024 (if demonstrations are warranted).
- Results of application of the predictive model to burners and appliances not tested.
- Comparison of results with those of THyGA and other research

4. PRELIMINARY OPEN FIRING TESTS

The current plan starts with open firing tests. Initial results from one burner have been obtained and are provided here to illustrate some of the techniques that will be used in reporting on the open firing tests in the next project report. The results are very preliminary to demonstrate that the program is underway. The next report will have much greater detail on the performance of burners.

The flame tested demonstrated minor differences when 10-31% hydrogen was blended into natural gas, but for 100% hydrogen the preliminary results confirm the need for burner modification.

4.1 Burners and equipment

The initial part of the project plan is to test the following small burners in open firing tests (to be followed in 2023 by confined firing tests):

- Partial premix inspiratory - pizza oven burner (radiant) with AN style nozzle 35 kW (120 MJ/h) AVANZINI
- Forced draft high velocity nozzle mix burner 65 MJ/h (~20 kW) max, air pressure of 7 kPa – North American Tempest 4442 – produces a narrow flame
- Forced draft premixed – Riello small package burner BS2F with integral fan. Rated at 35 – 92 kW with a 106 mm diameter burner tube
- Eclipse air heat burner – rectangular mounting – produces hot air (not flame) 25 – 500 MJ/h = 7-139 KW
- Raypak B0200 water heater, 56 kW (200 MJ/h)
- A small premixed radiant burner
- A pilot burner will also be trialled if available, e.g., Kromschroder (10.8 MJ/h)
- Flame rods and UV detectors integral to some of these burners will also be tested

A flame retention nozzle burner has been used in preliminary trials (see next section) but will be replaced by the pizza oven burner in further tests since it is a new unit with original injector size and has an ignition and control system.

The small Riello package burner to be tested is a forced draft premixed burner. The mixing between the multiple gas injector tubes and air supplied by a fan occurs within the combustion head, but the bulk of the combustion occurs downstream of the burner. Manufacturers such as Riello provide a variety of burner heads so that these types of burners can be adapted for fuels such as towns gas, which has approximately 50% hydrogen (depending on location). However, for a burner intended for natural gas, such as the one to be tested. The addition of more than 20% hydrogen may result in flame instability or overheating and loss of throughput. There may be some noteworthy results from this burner. In contrast, some package burners are nozzle mix burners which are likely to be suitable for quite high hydrogen percentages because there is no premixing.

4.2 Preliminary tests – Flame Retention nozzle

4.2.1 Experimental equipment and settings

These preliminary tests were conducted at University of Adelaide, Thebarton campus, in June 2022. Flame Retention (FR) nozzles are a simple entrainment type of burner which is ideal for initial tests – to test ignition, stability, flame length and flame shape (video) and flue emissions (CO and NOx vs CO_2 and oxygen) at various H_2 percentages from 0 – 100%). The nozzle used is a short pattern inspirator, FR 8, shown in Figure 1, with a nominal gas pressure range of 0.75 kPa to 6 kPa. It requires no air supply because the gas pressure entrains primary air into the nozzle and then some additional air in also entrained after the mixture leaves the nozzle.

The FR8 nozzle used has been used previously elsewhere. It is nominally rated at 20 kW when operated at the lower end of its normal pressure range. However, tests with the 5mm injector supplied with this nozzle produced 121 MJ/h (34 kW) at 0.675 kPa, indicating that the injector had been drilled out from its original size.

The open firing tests used a constant heat input rate of 121 MJ/h (34 kW), based on HHV, for all blends of natural gas and hydrogen. The flowrates of bottled hydrogen and line natural gas were controlled accurately by mass

flow controllers to maintain the constant total energy input independent of blend ratio. This method of comparison was chosen because users are likely to operate with similar heating rates to current rates whether using natural gas, a blend or 100% hydrogen (depending on what other adjustments may be required to allow for different heat transfer characteristics of the flames). In contrast, maintaining constant pressure would cause a reduction in energy input as hydrogen is blended into natural gas.

Figure 1 The FR 8 nozzle with air inlet closed

Vertical upwards open firing was conducted with natural gas, 10mol% H2, 21.7 mol% H2, and 31 mol% H2 blends for relevance for initial blending projects with up to 20% hydrogen. The 31mol% test provides a buffer margin like “Nb” gas when testing Type A appliances. Tests were also conducted with blends containing 50%, 70%, 80%, 90% hydrogen and with 100% hydrogen. All these preliminary tests were used to establish techniques for further work in the project.

Flue gas emissions were measured with a Teledyne T200 analyser for NOx and O2, and a T300 for CO and CO2 measurements. The NOx analyser measures NO via the chemiluminescence method and calculates the NO2 concentration by switching modes every other measurement period, such that the NO2 is converted to NO using a catalyst, enabling the total NOx concentration to be measured and the NO2 fraction to be determined via subtraction. This allows both NO and NO2 to be measured down to the ppb level with an accuracy of ±0.5% of the reading above 50 ppb. The CO measurements also have an accuracy of ±0.5% of the reading above 5 ppm, while the CO2 sensor has an accuracy of ± (0.02% + 2% of reading). The O2 sensor has an accuracy of ±0.1%. The Teledyne gas analysers were factory-calibrated, and an in-house calibration/linearity check was also performed using existing span gases and dilution with UHP nitrogen via the mass flow controllers to ensure the readings in the measurement range were within the stated margin of error. The gas samples for the two analysers were taken continuously from a single sampling point using a stainless-steel probe fitted with a thermocouple at the tip for monitoring temperature during the measurements.

Flame photographs were taken with a Canon EOS 1500D DSLR camera with a long shutter speed of 3.2 seconds to show a qualitative impression of the average flame envelope. The other camera settings were kept constant and used to provide enough exposure of the flames with 90 vol% and 100% hydrogen, although this results in saturation in the images with 0 – 31 vol% hydrogen. The images were taken with an aperture of F29 and digital “film speed” of ISO 400.

4.2.2 Flame images and flame stability tests

The flames were stable with the air inlet fully open for all the blends. Blow-off did not occur in any of the cases. Light-back only occurred when turning the flames off, when the fuel rate was reduced to a very low rate and occurred with natural gas just as readily as it did with hydrogen blends. Because the flames were stable, the tests were all performed with the air inlet fully open.

Figure 2 shows long exposure photographs of the 121 MJ/h vertical open flames from the FR nozzle with the 5 mm injector. Not all the blends tested are shown. The images show the relative similarity of the flames with 0-31% hydrogen. The 100% hydrogen flame appears quite visible but lacks the flame radiation from hydrocarbon fuel flames. (Future work focussing on the range from 0 -31% hydrogen with other burners will adjust exposure settings to avoid saturation).

Heat flux measurements were not taken in these preliminary experiments but will be measured in future work with other burners to quantify any changes in thermal radiation in the range 0 – 31% hydrogen and to the quantify the reduction from 100% hydrogen flames. Further experiments planned in a confined ducts (for 2023) will assess both convection and radiation heat transfer.

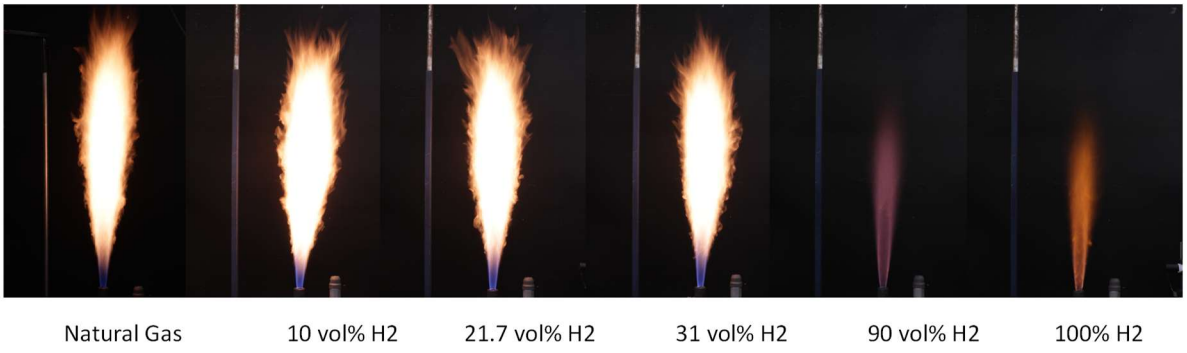

Figure 2 Long exposure photographs of flames from FR nozzle with 5mm injector at 121 MJh

4.2.3 Effect of maintaining constant heat input on gas pressure

The Wobbe index of the blend reduces as hydrogen is added, as shown in Figure 3, and this required increased pressure to maintain constant energy input. The measured pressures are shown in Figure 4. The pressure measurements indicate that it is likely that adjustments to pressure or fuel flowrate will be required when blending with 10 or 20 mol% natural gas. In many appliances and processes the control of gas flowrate is automated based on a temperature setpoint, but for those where this is not the case, some other adjustment may be necessary.

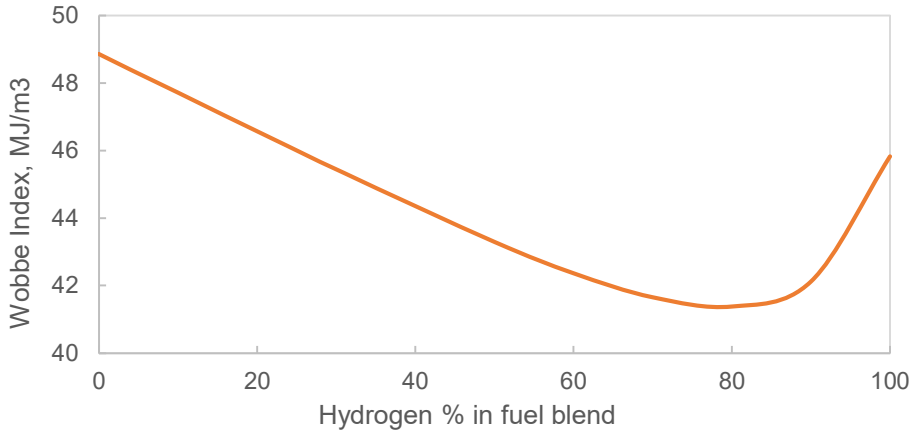

Figure 3 The effect of blending hydrogen into natural gas on Wobbe index

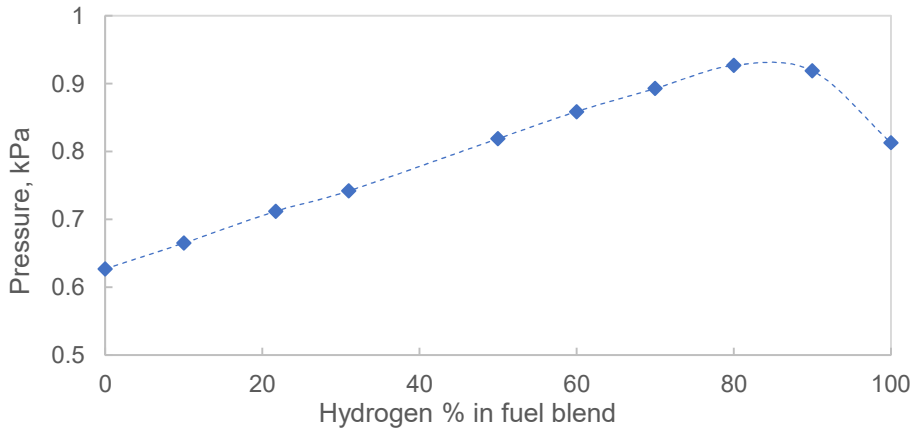

Figure 4 Measured gas pressure for the 5mm injector to maintain 121 MJ/h energy input

4.2.4 Effects of hydrogen blending on the emissions from an FR nozzle

Figure 5 shows CO emissions from the open flames as a function of vol% hydrogen in the blend. The raw samples were measured well above the flame and oxygen levels were approximately 18% to 19%, so the raw emissions have been corrected to 0% O2 for presentation. All the CO emissions were low, and unsurprisingly CO emissions reduce as hydrogen added due to reduced carbon in the fuel. The ratio of CO/CO2 produced is relatively constant, with a slight increase for 90% hydrogen blend. The overall mass rate of CO reduces steadily as the percentage of hydrogen in the blend increases.

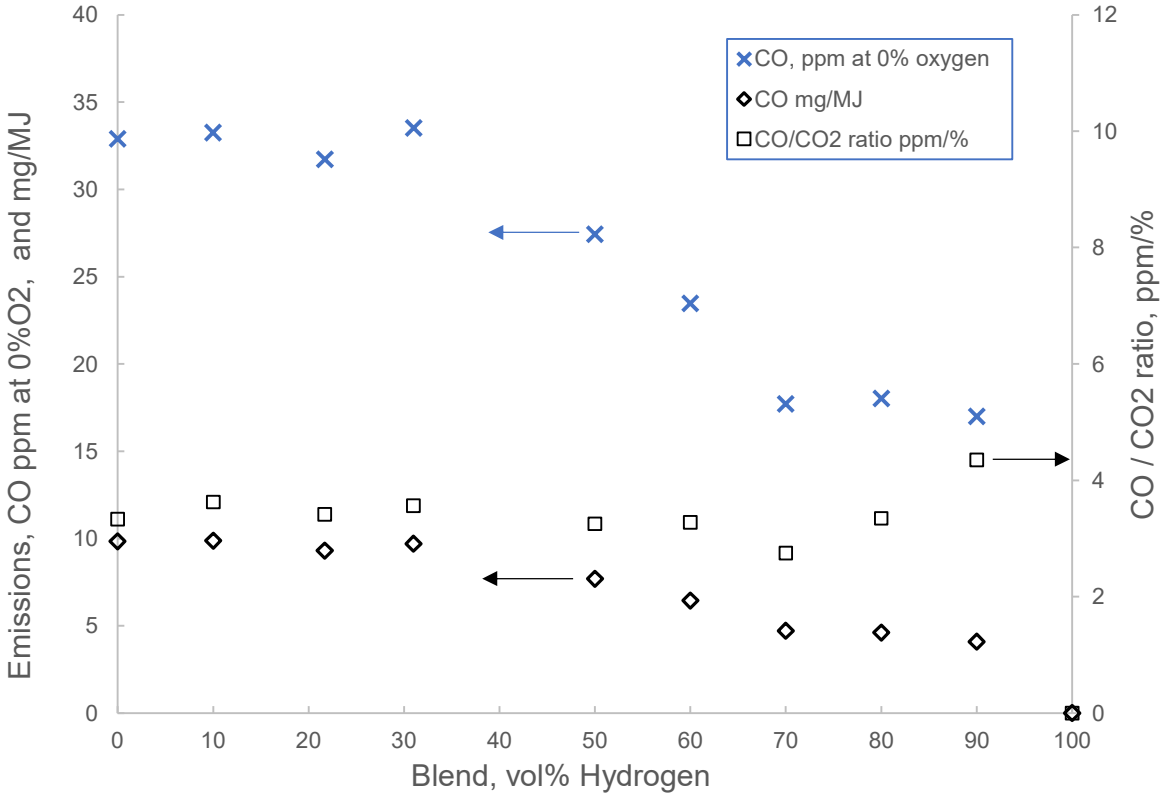

Figure 5 CO emissions from open FR8 nozzle flames

NOx emissions from commercial and industrial flames are often of greater concern than CO emissions. Figure 6 shows the NO, NO2 and NOx emissions from the flames corrected to 0% O2. The correction to 0% O2 doesn't consider the effect of changing stoichiometric air requirement and entrainment rates as the percentage of hydrogen in the fuel changes. Consequently, it is more informative to calculate NOx emissions relative to the heat input of the fuel.

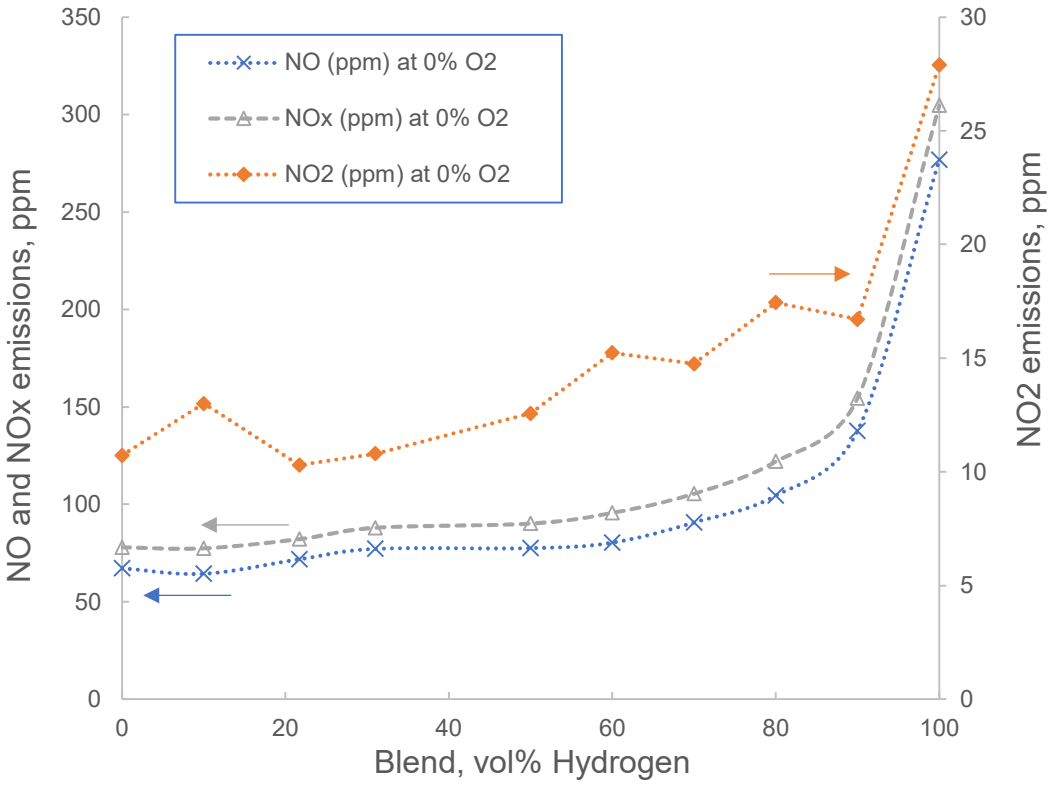

Figure 6 NO, NO2 and NOx emissions corrected to 0% O2

Figure 7 shows the emissions in mg/MJ as a function of the % of hydrogen in the fuel. Note that the units mg/MJ are equivalent to ng/J, used in Type A emissions testing. Flue gas NOx emissions limits are often stated in units of mg per Nm3 of flue gas, but this does not account for the potential reduction in flue gas volume when hydrogen is burnt. Hence the units mg/MJ have been used in the current work. NO2 emissions are much lower than NO emissions and Figure 8 shows that there is a slight reduction in the ratio of NO2 to total NOx for the 100% hydrogen flame. The NOx emissions form the natural gas flame, and the 10% and 21.7% hydrogen flames are virtually identical, emphasising the overall similarity in flame characteristics. This is a promising early result for pilot trial regions.

The NOx emissions from the 100% hydrogen flame are significantly higher than the natural gas flame. This is not unexpected since it is known that hydrogen can produce higher flame temperatures than natural gas which can lead to higher NOx emissions. However, NOx emissions also depend on the air fuel mixture ratio and the mixing processes, and the heat transfer mechanisms involved. Hence there is a dependence on burner and appliance design and operation. These open flames self-cool by radiant heat transfer to the room, with a greater cooling effect for the more luminous natural gas dominated flames, it is expected than the flames with 0 – 31% hydrogen would have lower NOx emissions than the 100% hydrogen flame.

The FR nozzle tested is not a low NOx design, but there are known techniques for reducing NOx emissions, and burner suppliers have a range of low NOx designs for natural gas burners. It can be expected that low NOx designs will be developed for hydrogen burners also and these will also be investigated in the current project. As such the results with 100% hydrogen seen here are not a cause for concern. The current results serve as an illustration of the techniques that can be employed with other burners in the current project. Adding heat flux and flame temperature measurements will increase the understanding of the NOx production from these flames.

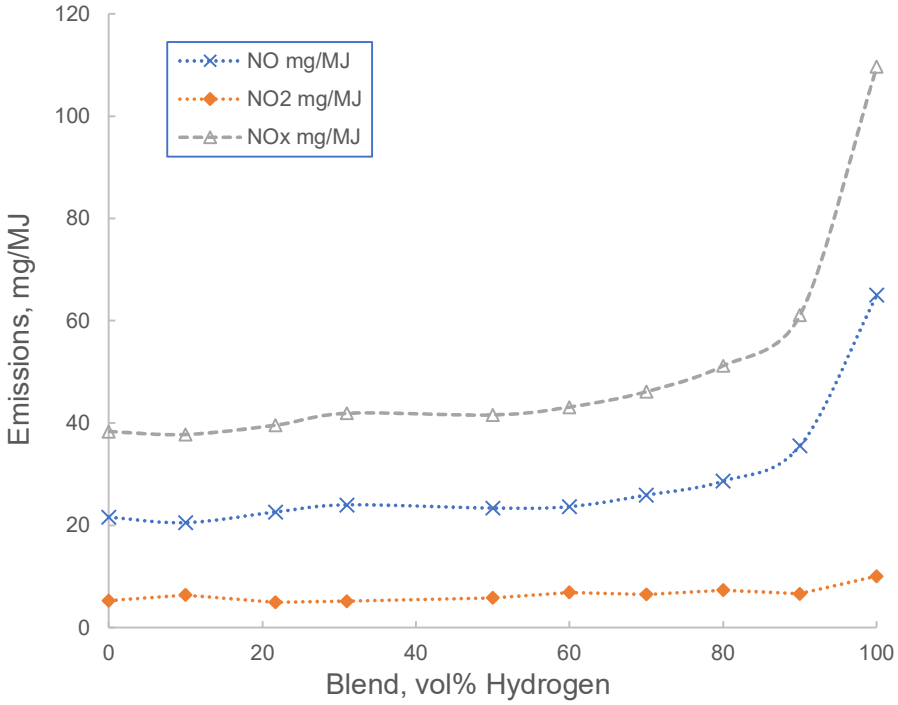

Figure 7 NO, NO2 and NOx emissions in mg/MJ

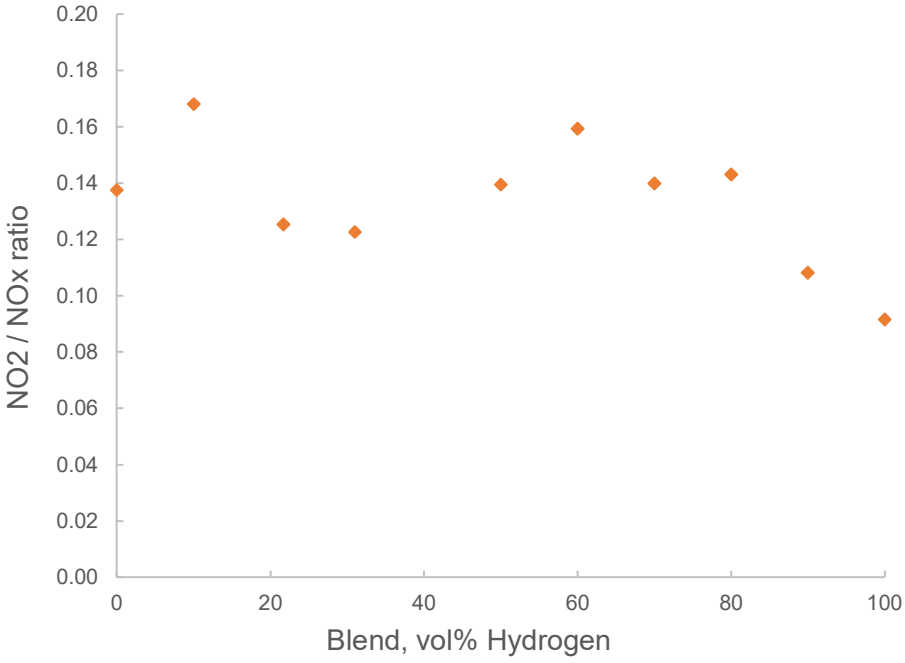

Figure 8 NO2/NOx ratio for FR nozzle flames

5 CONCLUSIONS

The report outlines the motivation for the project and provides details on, and reasons for, the changes to the project plan. The changes have been made to the project plan extend the scope to providing confidence with 20 mol% blends in addition to 10 mol% blends and to provide more research impact on use of 100% hydrogen in Type B appliances.

The report provides very preliminary results from open firing of an FR nozzle as an illustration that the detailed test plan in underway. Preparation for more detailed open firing tests of multiple burners have been made.

6 IMPLICATIONS AND RECOMMENDATIONS FOR INDUSTRY

The implications of the results to date are that they are early results that illustrate that the experimental work is underway. The experiments to date have clarified which naturally aspirated burner to use in further work. It also highlights that this type of burner is likely to be one that can operate well with up to 20 mol% hydrogen, and that NOx emissions will need to be addressed if it is used for 100% hydrogen. The implications of the planning work described is that coming reports will have many more useful results.

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8 APPENDIX

8.1 Type B appliances – variety of burner sub-types and furnaces

The focus of this project is non-pressurised combustion. Examples of burners given in AS 3814:2014 include atmospheric burners in which all the air is provided by the inspirating effect of the gas and/or natural draft, induced draft burners and forced draft burners. A broader categorisation includes non-premixed (diffusion), premixed, partially pre-mixed, and nozzle mix burners (Panek et al. 2020).

Specific burner types include the following (Hazlehurst, 2009):

- Diffusion:
    - These have separate air and fuel streams through the burner. Only a small amount of primary air is introduced through the burner and is not mixed with the fuel. The rest of the air is supplied through a larger inlet which delays mixing and produces large and radiant flame in kilns and furnaces.
    - They are used in cement, glass, alumina calcining, smelting, rotary kilns.
    - Most of the air is entrained into the flame in the combustion chamber and is not introduced through the burner, e.g. – rotary kilns, glass tanks – gas at up to 10 kPa (Hazlehurst, 2009) into high T air, neat gas burners (which are probably rare).
- Atmospheric natural draft partial premix:
    - Use gas at relatively low pressure, e.g., 1 - 2 kPa which entrains 50 -55% of stoichiometric air, with additional air entrained after leaving the burner.
    - Examples include most industrial burners – drilled bar, single port (Bunsen), ribbon ported, ring, assembles of drilled.
    - Burner port loading is about 10 MW/m2.
    - Retention burners using some form of stabilisation, e.g., bluff body, often with a small fraction of the gas to provide a stable attached flame, ribbon burners also have small auxiliary flames from small openings which stabilise larger flames from larger openings.
    - Pilot burners – can be natural draft atmospheric or forced draft.
    - High gas pressure, natural draft. A gas pressure of approximately 2 bar entrains enough air to be fully premixed.
- Fully premixed:
    - Air blast burners – air at approximately 7 kPa is supplied from a fan and entrains the gas producing intense small flames (i.e., full stoichiometric mixing). Air blast burners are typically used with quarls or refractory tunnels. Radiation from the hot refractory stabilises the flame. If used without quarls, normally a Retention or non-blow-off tip is used. These have a small axillary flow which acts to stabilise the flame. They are metal – so used without being fully enclosed to avoid high temperatures. For atmospheric pressure furnaces, control is simple, gas entrainment rate is linear with air rate, and can be adjusted manually. Non-return valves are required on both air and gas supplies. These burners are used for heat treating and annealing and re-heating furnaces to produce high temperatures with compact flames.
    - Porous radiant burners – used for space heating, grills, high temperature drying - supplied by either an atmospheric injector or and air-blast burner. Types are: porous refractory, perforated refractory tiles, wire gauzes (lower temperature grills and space heaters), catalytic combustion (for LPG, whereas Natural gas has methane slippage), high temperature porous medium burners.
- Nozzle mix (including package burners):
    - Air and gas are supplied at pressure separately to the burner exit, and normally swirled, sometimes into a quarl, to create rapid mixing between. The quarl aids stability by providing recirculation and radiation to incoming fuel. This can result in a small blue flame, but only if there is rapid mixing. Different air registers may have different air velocities and varying the air ratio between registers can alter flame length and intensity. Often flue gas recirculation or mixing related NOx control

measures are also used. They are highly adjustable and have high intensity. Specific types include:
    - High velocity burners – full combustion in a metal can and quarl, then high velocity gases out to cause turbulence in the furnace. Can operate on high excess air.
    - Flat flame burners to avoid impingement
- Radiant tube burners for indirect heating – many types use nozzle mix or air-blast burners or flameless oxidation
- Self-recuperative burners
- Regenerative burners
- Package burners – have own fan, control system and valves – typically they use nozzle mix burners but can also be partial premixed.
- Partial premix with ratio control of fuel air mixing to provide a non-flammable fuel-rich mixture – have been used in past after converting from manufactured gas to a gas with higher WI (i.e., natural gas), so that the new gas can be diluted down to the same WI as the original – but not relevant for the current work.

A wide variety of burner types are used in furnaces and ovens as illustrated by the list above. Many different types of furnaces and ovens are used, with a small selection listed in Table 1 (Green, 2008).

Table 1 Some types of furnaces uses and their characteristics

Glass	Many variations on regenerative and recuperative
Batch	Heat treatment, drying, calcining ceramics
Direct fired	Side-fired, over-fire, underfired, limited to about 1000^{o}C
Indirect fired	Radiant tube or muffle
Protective Atmosphere	Heat treatment of metals in indirect furnaces. Gases used are hydrogen, dissociated ammonia, reducing gases - combinations of H_2, N_2, CO_2, CO, CH_4

8.2 Type B appliances in Australia

Discussions with network operators and local Regulators and assessors identified many appliance types in pilot trial areas in Australia. A consolidated summary of the types of appliances present is given in Table 2.

Table 2 Type B appliances in hydrogen injection pilot trial regions in Australia

High efficiency Boiler	Paint spray booths	Arc furnace
Condensing hot water heater	Powder coat drying oven	Burn-off furnace
condensing boiler	Paint drying oven	Laminator for plastics
Hot water heater	Indirect air heater for metal drying	Annealing furnace
Pizza ovens	Scrap metal heater	Exothermic atmosphere generator
Coffee Roaster	Brick Dryer	Endothermic gas generator
Rotary rack baking oven	Grain Dryer	Cremator
Continuous conveyor baking, drying, grilling	Clothes dryer	Incinerator
Various commercial catering	Process Ovens	
Tank heater	Curing oven	Air conditioning - engine driven
Immersion tube heater	Powder cure oven	Generator
Hot oil heater	Drying oven	Turbines
Oil heater with package burner	Concrete curing system	
Direct fired space heaters		
Indirect fired space heater		

8.2.1 Appliances in other regions in Australia

The list of appliances in pilot trial areas was expanded through prior knowledge and discussions with industry advisers, certifiers, regulators, and suppliers, and referencing those listed in the relevant Australian Standards.

shows this expanded list with appliances categorised by broad combustion type (premixed, partially pre-mixed etc), and with more details about burner types and with some suppliers listed. The expanded list illustrates that the appliances in pilot trial areas are representative of those throughout the broader economy. The types of burners used in most of these appliances are represented in the list to be tested in the screening tests (see Section 2.2 -1b on page 6 of the Proposal for future work).

The aim of categorising appliances and burners is to find common or representative types, so that the testing and assessment can be as representative as possible.

potential to reduce the required testing program to a manageable level. It will also be important to assess common burner control systems and flame detection systems.

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The aim of categorising appliances and burners is to find common or representative types, so that the testing and assessment can be as representative as possible.

potential to reduce the required testing program to a manageable level. It will also be important to assess common burner control systems and flame detection systems.

Table 3 Appliances in Australia, categorised

Appliance /application	Combustion type	Example suppliers
Pressurised combustion		
Co-gen, tri-gen	Engine	
Co-gen, tri-gen	Turbine	
Co-gen, tri-gen	Microturbine	Capstone
Air-conditioner	Engine	Yanmar
Generator	Engine	Kohler at Gas Train (from Clark Energy), CD Power Generac in WA
LNG production (huge user of NG)	Compressors, refrigeration	
Diffusion / non-premixed		
traditional regenerative glass furnaces, alumina calcining, smelting, rotary kilns (cement, lime, mineral processing, plasterboard)	Diffusion • some have primary air as nozzle mix – e.g. rotary kilns • Glass often just 2 co-annular gas streams	
Heat recovery Steam Generators (e.g. in Co-gen plants), driers	Duct burners (non-premixed) – linear and grids - fuel exits on lee side with flame holder	
Direct Driers	Duct burners	
Partial premixed		
Pilot burner for large ovens and kilns	Atmospheric partial premix with ID fan and pressure switch	
Tumble dryer etc	Normal package burner. Some are partial premix with ID fan	
Coffee roaster	Some older smaller ones partial pre-mixed – latest are package burners	
Porous radiant burners – stadium space heaters and low temperature drying of paints, textiles, paper, heat treatment of steels, plastic forming, conveyors	Atmospheric partial premix – work like a grill in a Type A oven (or some BBQs have one under a hood)	
ribbon burners –wire, glass blowing (Ribbon burners are high energy and don't flashback and don't lift)	Some are atmospheric partial pre-mix venturi into a mixing chamber and then multiple exits (others are forced draft)	
water bath heaters in the gas distribution system, range from 3-10 GJ/h (Potential for full-scale test)	inspirated premixed burners with tall stacks to provide natural draft (others are forced draft nozzle mix)	
metal treating – galvanising lines (e.g. burners used for indirect heating of galvanising bath)	some atmospheric partial premixed burners with a flame retention nozzle on the end - and some forced draft nozzle-mix	
Pizza ovens (Flame luminosity may be important – so this may change. Quick response of gas compared to electricity, with automatic control is a plus. Gas fired units may be cheaper to buy and connect.)	A variety of curved line burners or multiple simple partially premixed burners firing upwards.	Euroquip
Direct fired gas heaters	Multiple partial Premix venturi tubes	AIRA Dravo by Seeley
Process Ovens	Some are partial premixed	
Plasterboard manufacture	Some are FD partial premix?	
Paint spray booths	Most commonly use Riello package burner (FD) firing into a heat exchanger which heats air than circulates over the parts to be baked	Aussie Spray booths, Lowbake, Seetal,

	Or IR gas burner – forced air partial premix burner heating steel that radiates. Some use atmospheric direct air burners – i.e. inspirating type with no fan. Some use catalytic IR burners mostly firing LPG	Complete booth services, Truflow
Hot oil heater	Atmospheric partial premix	Thermal and Pressure Engineering, Consolidated Fire and Steam
Brick kiln burners		
Continuous conveyor baking, drying, grilling	Pre-mixed Gas mesh - like a domestic griller	(Wesmartin) Furnace Engineering
Fully Premixed		
Strip line burners for metal heat treating, laminating plastics, plastics forming, low temperature drying of paints, textiles, paper, biscuits	Fully premixed ribbon burners and other porous burners, Possibly some air-blast	
Burners for singeing textiles	Fully or partial premixed	Furnace Engineering
Strip line burners for metal treating	Radiant tube burner – some use “MILD” or “Flox”	
Furnaces	Premixed Radiant wall burners tolerant to H2 up to 70% (RP1.4-02).	
Process heaters	Radiant wall burners, custom made, fired heaters such as SMR, ethylene crackers, process heaters in refineries, e.g. in Kwinana	
Endothermic gas generator (used for metals treatment – e.g. carburizing or to prevent oxidation, carburizing (case hardening).	Uses a retort with catalyst and requires a carbon-based gas.	
Forced Draft Nozzle Mix (including package burners) – some also ID fan		
Utility scale boilers – only 2 left in Australia	Nozzle mix or tangential	
Industrial steam raising boilers (some industrial ones also used to generate electricity) + Commercial sector boilers used for steam – circular burners with range of swirl air, FGR or internal FGR for NOx control	Older are Nozzle mix Newer are package burners Small versions are FD only, larger may be both FD and ID	Trevor, Maxitherm, Enertech, Tomlinson, Cleaver Brooks
Water bath heaters in the gas distribution system - nozzle mix forced draft burners are used in some water bath heaters	Nozzle mix	
Fired heaters – have a radiant and convective section. NOx control techniques similar to boiler burners	Nozzle mix	
Metal treating – galvanising lines have some premixed burners and some nozzle-mix	F.D. Nozzle mix	
Strip line burners for metal treating	Nozzle mix	Major Engineering
Controlled atmosphere furnaces	Nozzle mix	Major Engineering
Self-regenerative burners, which have a large mass of refractory material - used in pairs in a cycle (flue gas heats out of service burner refractory). Applications include continuous and batch metals heating and forging operations and ceramics and pottery kilns	Nozzle mix	Orora (Gawler)
Self-recuperative burners (flue gas heats incoming air)	Nozzle mix	Maybe not in Aus?
Side-fired glass kilns with recuperators, brick tunnel kilns, ceramics kilns and multiple hearth magnesia calciners	Nozzle mix	Traditional glass kilns
High efficiency Boiler	Package burner	
Condensing boiler - can put modular multiple	Multiple premix burners controlled by air gas	Baxi,

units to partially replace old boilers. (Condensing boilers will be even more efficient than non-condensing boilers with blends of hydrogen since more of the flue condenses. Flue design will need to be assessed for 10% and 21.7% hydrogen	ratio valves. Much more efficient than non-condensing.	Applied Heating, Moorea
Hot water heater & condensing hot water heater	Package burner	Chappee, Raypak, William Boby, Hunt, Hoval, Rheem, Aira
Cremator	In a small furnace (incinerator) + package afterburner for stack	
Incinerator		Megtec
Immersion tube burner	High velocity burner (Tolley Ch 4, Fig 4-25) – exhaust gas at high vel. for convective heating	Garnic
Indirect fired drier – i.e. combustion products heat coils containing fluid to be heated	Package burner (can also be atmospheric or nozzle mix)	
Coffee Roasters	2 package burners, one for roasting, then an afterburner in flue for odour and burn light husks	Probat
Radiant space heaters linear or U-tube	small package burners – flue gas down tube which acts as radiant heater	
Tumble dryer	Package burner (or atmospheric with ID fan)	Jensen, Passat, Lavatech
Ironing machines	package burner with Thermal oil heater	
Furnaces – reheating, annealing, melting, forging, reverberatory, arc, burn-off	Package burners	Furnace Engineering
Furnaces – reheating, annealing, melting, forging, reverberatory, arc, burn-off	Nozzle mix	
Ovens and furnaces, e.g. curing ovens, paint drying, powder coat drying	package burners	Riello, Furnace Engineering, Garnic
Timber drying kiln – may be more likely to use waste wood		
Plasterboard manufacture	Package burner	
HVAC indirect air heater	Some are package burners, others are atmospheric	
Process ovens	Some use package burners	Furnace Engineering
Baking ovens, e.g. rotary rack	Package burners, mainly 2 stage package burners – i.e. low fire and high fire option, that don't modulate between Flat bread tunnel ovens -FD package burners	
Grain dryers	Probably package burners	Agridry GrainTech Engineering
Powder cure oven, drying oven	Burner systems not clear	Garnic
Concrete curing system	Probably package burners providing hot combustion gases	Kraft
Methane as a feedstock		
Ammonia production	Steam methane reforming	
Sodium cyanide (3 plants fed via transmission lines)	Reactor (methane+NH_3+ O_2)	
Petroleum refining – hydrogen for hydrocracking	SMR	
Those specifically in AS 3814 or AS 4563		
Process after-burners		
Steam and water boilers		
Duct fired Air heaters and curtain		
Smoke ovens	Atmospheric, some package burners	
Direct fired rotary kilns and ovens		
Special Atmosphere Generators		
Cremators and incinerators		
Steam and Hot water boilers		
Stationary Gas engines and turbines		
Water heaters		
Multi-fuel firing systems		

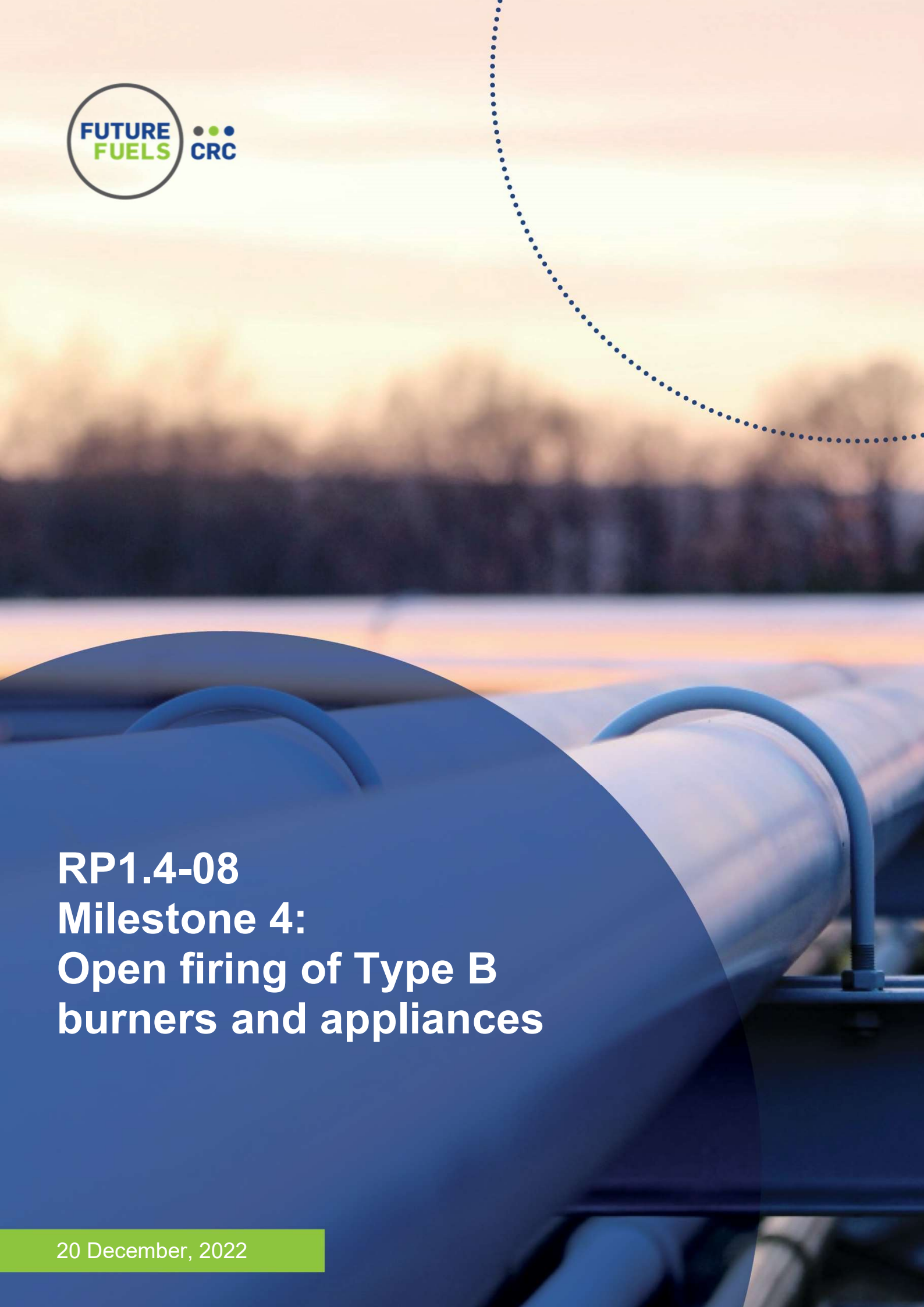
8.3 Original Project Plan (from the Project Proposal)

1. Compatibility of equipment listed in Table 2 (in the Appendix) with various gas blends, and model development (report and paper).
   a) Screening tests at GasTrain using the following list of burner types with 10 mol% and 21.7 mol% hydrogen blends with natural gas, and maximum hydrogen% without changes, and (for some of the burners) up to 3% oxygen and up to 10% nitrogen. The list of burners covers as close as practical to the entire range used in Type B appliances. (Package burners alone are used in a large proportion of Type B appliances.) A condensing boiler will also be included, but the others in the list to follow are all burners, rather than complete appliances:
      - Two different styles of package burners (these are the most common type of burner used in Type B equipment).
      - 1 nozzle mix burner.
      - Fully premixed non-radiant burners (for furnaces and Neon sign manufacture and pilot burners) – these have larger burner port areas than other types.
      - Air blast burner – uses air at about 7 kPa, compared to most others which are atmospheric.
      - Atmospheric partial premixed venturi burner (many uses, e.g., ducted space heaters, pizza ovens, oil heaters, water heaters).
      - Atmospheric partial premixed burner with ID fan (e.g., used in tumble dryers and pilot burners).
      - Atmospheric natural draft partial premix porous ceramic radiant burner (e.g., used in space heating, paint drying, metal heat treatment).
      - Atmospheric forced draft partial premix porous ceramic radiant.
      - Ribbon burners (partial premix and forced draft versions).
      - Atmospheric partial premix with flame retention nozzle on end (used in galvanising).
      - forced air partial premix infra-red burners (catalytic and non-catalytic) used in paint spray booths.
      - Forced draft nozzle mix (e.g., some galvanising line burners).
      - Forced draft “high velocity” nozzle mix burner as used in immersion tube burners and some water bath heaters.
      - Water bath heaters – scaled down version of the inspirated premixed type with tall stack for natural draft.
      - Condensing boiler.
   b) Perform detailed tests for those burners / appliances in which there were no issues on the screening tests at (i) University of Adelaide with smaller burners and appliances, (ii) Furnace Engineering in Melbourne in a real furnace to assess effects on product quality and emissions, (iii) Ignite Gas or one of the other locations with appliances used for water heating, (iv) GasTrain and (v) Jemena, Western Sydney, where some burner types can be tested, but also a Fuel Cell and a Microturbine can be demonstrated with maximum allowable hydrogen % blends. At each location, use hydrogen mole percentages of 0, 10, 21.7, and a maximum (without changes to any equipment). The detailed tests are for leakage, emissions, heat transfer duty and efficiency, metal temperatures, damage, control, operability, flame detection, flue gas condensation, variability in %hydrogen of the blend.
   c) Development of a model for combustion and heat transfer – at this stage only replicating the results observed.
   d) At the end of this stage of the project, Dec 2022, have a review, with potential to alter direction of the project.

2. Burners / appliances and predictive models with changes for maximum hydrogen blend % and 100% hydrogen:
   a) Obtain and report on documentation and other evidence from burner and appliance suppliers already marketed for 100% hydrogen and other levels of hydrogen above 25%. Collaborate with a burner supplier such as Techrite (who handle the Honeywell range) to provide the most compelling demonstrations of their burners and other components for 100% hydrogen and other % hydrogen (e.g., documentation, video, site visit demonstration or demonstration testing by UoA).
   b) For some of the burners, design changes required to hardware and control systems to increase max% hydrogen, based on previous trials and theory.

c) Complete models and use to predict performance of burners not tested and with the design changes to control and hardware.
d) Detailed tests of performance and failure modes for max% hydrogen blends.
e) Adjust models based on experimental results.
f) Review literature and for appropriate cases, design retrofits using 100% hydrogen and make models for 100% hydrogen.
g) Detailed tests on 100% hydrogen retrofits and adjust model.
h) This stage of the project will occur in 2023.

3. End-user site test preparation, generation of model, costing and training outputs
   a) Predict performance and results at water bath heater site and two other sites.
   b) Perform cost estimates for demonstration tests at these sites with a 10 mol% hydrogen blend. Discuss the viability and importance of doing these tests and if they will be done write a proposal or arrange funding outside the current project for results by Dec 2024.
   c) Extend the predictive model for performance, failure modes and operational limits to burners and appliances not tested, leading to development of a design and costing tool for modifications to key burner types for different blend ratios and for 100% hydrogen (for those where it is practical and potentially economic).
   d) Perform the cost estimates, and work required to detail any conversion/retrofit to equipment for 10% (if any), max% or 100% hydrogen, operating procedures or standards, and provide training as outlined in Output 3.
   e) For a small network, feeding a range of industrial users, perform a cost estimate for a 10% hydrogen blending trial based on usage data.
   f) This stage of the project will rely on some of the earlier results, and will be the final part of the project, but parts of it will progress throughout the entire project timeframe.



RP1.4-08
Milestone 4:
Open firing of Type B burners and appliances

20 December, 2022

Project number: RP1.4-08

Detailed assessment of Type B appliances with blends of hydrogen and natural gas and 100% hydrogen

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Summary of Report

This report investigates the effects of the addition of hydrogen (H_2) to natural gas on the combustion performance of a range of Type B gas appliances, commonly used in industrial and commercial settings in Australia. Five different burners were studied under open firing conditions, with a focus on identifying the maximum allowable hydrogen content in the fuel with no major changes being made to the appliances. The burners were initially operated without any modification following a standard installation for, and operation with, natural gas, and the hydrogen concentration was incrementally increased until a practical upper limit was observed. Adjustments to the appliance were then performed, including changes in flame detection and/or air and fuel flowrates, as well as the investigation of simple retrofits in some instances. In addition to qualitative observations—including flame stability (i.e. light-back), appearance, noise and burner temperature—quantitative comparisons of emissions (CO and NOx) and radiative heat flux were performed for the various fuel blends. A brief description of the five appliances tested and a summary of the notable findings for each of them is provided below.

A naturally aspirated “atmospheric nozzle” (AN) style burner was tested. It has an adjustable screw to control the level of primary aeration. Initially the burner was operated on natural gas at full rate (35 kW) with the air inlet fully open, and the hydrogen content was increased until light-back of the flame into the air-fuel mixing tube occurred at 65% H_2. This test was repeated at the turn-down setting of 20 kW (on natural gas) and light-back was observed at 55% H_2. The primary air inlet was then closed, and the test was repeated at high rate. In this case the limiting factor was the flame rod failing to detect the flame, which occurred at 80% H_2 causing the control system to cut off the gas flow. A simple retrofit was performed by replacing the flame rod with a UV sensor, which allowed the burner to be operated on 100% H_2 with the air inlet closed. The photographs of the flames show that the 100% H_2 case with no primary aeration produces a flame of similar length and width to the natural gas flame with full aeration, although there is a significant increase in NOx emissions for 100% H_2 and a slight reduction in thermal radiation. Tests were performed with a sealable enclosure to measure and control the amount of air entrainment, and it was found that introducing 10% of the stoichiometric air requirement as primary air leads to a reduction in NOx emissions of approximately 20% for 100% H_2.

A forced air package burner with a specified operating range (on natural gas) of 35–92 kW was also tested. The fan operates at a single speed, with an adjustable damper to modulate the air flow. The burner was tested at the upper and lower heat input limits, and two air settings were considered for each firing rate. It was found that for all settings, the hydrogen concentration could be increased up to approximately 98% before the flame rod cut off the fuel supply. Again, the flame rod was replaced with a UV sensor which enabled operation at 100% H_2; however, additional heating of the burner head components when high levels of hydrogen are used may impose a practical limit. The manufacturer also offers an alternative combustion head assembly for operation with towns gas (which can contain up to 60% H_2), so this configuration was also explored as a potential retrofit option for 100% H_2. The results with the towns gas head suggest that it is suitable for 100% H_2, while maintaining similar NOx emissions to the original burner head.

Another appliance tested was an air-heat burner with a maximum fuel input of 147 kW, featuring an integral fan to supply the air similar to the package burner. The burner is designed to provide a small portion of premixed air when operating at low rate to improve flame stability, while the fuel and air are completely separated prior to reaching the exit ports at high firing rate. During testing it was observed that the V-shaped air diffuser section of the burner becomes visibly hotter at high levels of hydrogen addition (i.e. > 50%) when running at high firing rate, while light-back was observed at low firing rate at 80% H_2. Excessive flame noise also becomes increasingly apparent as the hydrogen concentration is increased beyond 40%.

A high-velocity nozzle-mix burner, commonly used in industrial furnaces and kilns and with a maximum rate of 20 kW, was found to operate well up to 100% H_2. Fuel and air are supplied separately and undergo rapid mixing and combustion inside a chamber within a refractory brick, with a narrow flame issuing from a small hole in the brick. The burner was operated at three turn-down settings, with the gas flowrate controlled via a ratio regulator which ideally maintains a constant air-fuel ratio as the air flowrate is adjusted. The flowrates of air and the H_2/NG blend were metered while increasing the hydrogen concentration to 100%, and the operation of the ratio regulator to control the fuel flowrate was assessed. It was found that there is an increase in the amount of excess air (as a percentage of stoichiometric air) as hydrogen is added up to approximately 80%, while the excess air percentage decreases from 80–100% H_2 due to the increase in fuel flowrate (note however that the 100% H_2 case still has more excess air in comparison with natural gas).

The final appliance tested is a radiant surface burner rated at 7 kW, which is a Type A appliance but features a similar design to those used in certain Type B installations. There are three separate burners which each have a gas injector that entrains air into a chamber behind a ceramic radiant tile. The igniter is on one side of the three burners and a flame detection thermocouple is located on the other side. The burners cross light across the face of the three ceramic elements. The results of the testing show good performance up to 31% H_2, with radiant heat flux measurements indicating that the efficiency increases with hydrogen addition. Light-back during ignition was found to occur approximately one in five times at 40% H_2, which poses a potential upper limit for this specific appliance.

The results of this investigation provide confidence for the compatibility of existing atmospheric-pressure Type B appliances with blending ratios that are likely to be used in Australian distribution networks in the near-future; that is, up to 20% (by volume) H_2. Specifically, no issues were observed for any of the burners operating on blends of up to 31 vol% H_2 in natural gas. The maximum hydrogen content with no changes was found to vary significantly between appliances, and for some burners the possibility of minor adjustments to facilitate operation at 100% H_2 was demonstrated. These results will be accompanied in the future by confined-firing tests which will provide further insights regarding the impacts of hydrogen addition on emissions and heat transfer characteristics under more practical conditions.

1 INTRODUCTION

1.1 Background and motivation

The substitution of natural gas with hydrogen presents a significant opportunity for the reduction of carbon emissions. The potential to use existing infrastructure—such as distribution networks and end-use combustion appliances—is a particularly attractive feature of hydrogen as an energy carrier. While the compatibility of hydrogen (both blended with natural gas and 100% H_2) with distribution infrastructure has been explored in detail [1,2], the behaviour of the combustion appliances which use this gas has received relatively little attention from a research perspective. One of the challenges in ensuring confidence with the transition to a hydrogen gas network is the wide variety of natural gas combustion appliances currently in use, with different burner designs potentially responding differently to hydrogen admixture. In Australia, combustion appliances are classified as either "Type A" or "Type B". In this scheme, Type A refers to mass-produced (typically domestic or light-commercial) appliances for which a certification scheme exists, while Type B consists of any other appliance with a gas consumption of greater than 10 MJ/h, for which a certification scheme does not apply, typically used in large commercial or industrial applications.

In the context of Type B appliances, a useful distinction to make is the division between combustion devices which operate at elevated pressures, and those which operate at near-atmospheric pressures. Elevated-pressure combustors typically include devices whose function is to generate motion (e.g., for electricity generation) such as gas turbines and reciprocating engines, whilst atmospheric-pressure burners are more common in both direct- and indirect-fired heating applications, such as furnaces, boilers and kilns. The Australian manufacturing sector—which comprises the majority of atmospheric-pressure Type B appliances—consumes approximately 294 petajoules annually (excluding as a chemical feedstock and for electricity generation); a total consumption 1.7 times greater than that used by residential (i.e., Type A) gas appliances [3].

Significant progress has recently been made in developing the capability of land-based gas turbines (used for electricity generation) to operate with increasing quantities of hydrogen in the fuel blend, with certain designs able to use both natural gas and 100% H_2 while maintaining high efficiencies and low emissions [4,5]. Similarly, the use of hydrogen-natural gas blends in automotive engines has received significant research attention, and has been demonstrated in practice [6]. While these developments clearly show that it is possible to design combustion systems which accommodate both hydrogen and natural gas (in varying proportions), it is also essential to investigate the behaviour of existing natural gas appliances as hydrogen is added to the fuel, without any changes to the burner. It is also of relevance to understand what minor adjustments and/or modifications can be made to existing designs to optimise performance at varying levels of hydrogen admixture. This is particularly important for the atmospheric-type combustion systems which are found in a more widespread range of applications (and are the focus of this report), since the complete replacement of existing installed appliances with updated designs may not be economically viable.

There are several important differences between hydrogen and natural gas in terms of their combustion properties, which have been well-documented in the past. These differences include the fact that hydrogen has a wider flammability range in air in comparison with natural gas, lower volumetric energy density, reduced stoichiometric air requirement, higher adiabatic flame temperature and a higher burning velocity/flame speed [7]. For hydrogen-natural gas blends with less than ~20 vol% H_2, previous experimental work suggests that the slight changes in flame speed and stoichiometry are unlikely to cause complications for the majority of industrial applications, at least in terms of the operation of the burner itself. Due to the increase in flame speed with hydrogen addition, an important factor to consider is the increased potential for light-back (i.e. propagation of the flame upstream into the mixing region) for premixed and partially premixed systems. For Type B appliances in which hydrogen addition causes light-back, it may be useful to explore burner adjustments that allow higher levels of hydrogen addition without light-back to maintain the required level of both safety (e.g. flame stability) and performance (e.g. emissions, heat transfer). The reliability of flame detection and burner control systems must also be assessed to provide confidence for the use of hydrogen in industrial gas burners.

Previous work conducted as part of the Future Fuels CRC (RP1.4-01 and RP1.4-05) found that blending hydrogen into natural gas in concentrations of up to 10% (by volume) results in very little impact on the operation of Type A appliances, while additional testing with a limit gas (for the 10% blend) of 21.7 vol% H_2 suggested that higher blend ratios are also feasible with minimal disruption. Recent injection trials at the sub-grid scale—such as the HyP SA project [8]—have confirmed this feasibility in practice. As these trials continue to be expanded upon

(in terms of both the area covered and the hydrogen blend concentrations) the impact of hydrogen admixture on industrial customers must also be considered, particularly concerning the safety and performance of Type B appliances.

For Type B appliances it is of interest to determine if there is a practical maximum level of hydrogen admixture to natural gas that is different from that observed for Type A appliances. For Type A appliances it is likely only practical to blend hydrogen in distribution networks up to a maximum percentage that can be tolerated by all Type A appliances without changes, followed by a single changeover to new designs in conjunction with a change to 100% H_2. However, for Type B appliances, smaller local industrial or commercial networks could potentially be operated at higher blend percentages. It may also be practical to make minor adjustments to individual Type B appliances in such a network, or to appliances at a particular site if hydrogen were generated locally. Additionally, many industrial gas users have set ambitious decarbonisation targets and are therefore interested in assessing the feasibility of higher levels of hydrogen addition, including the viability of 100% H_2 to replace natural gas.

1.2 Project scope and aims

This project is focussed on the investigation of the effect of hydrogen addition on the safety, performance and emissions characteristics of Type B combustion appliances operating at atmospheric (or near-atmospheric) pressures. Specifically, the current report aims to address the following key questions, based on the testing of a series of burners operated under “open-firing” conditions:

1. Does blending 10–20 vol% hydrogen to natural gas present any safety or performance concerns to the operation of atmospheric-pressure Type B appliances, assuming no changes are made?
2. Without making any changes, what is the maximum allowable hydrogen concentration in the fuel which can be achieved while maintaining safe operation?
3. If applicable, what minor adjustments (e.g. tuning of air-fuel ratio controller) can be performed to facilitate higher hydrogen content, and what is the allowable concentration in this case?
4. What types of burners can be operated on 100% H_2 with either no change, minor adjustments, or with a simple retrofit?

Note that for the above questions, it is not expected that there will be a straightforward, non-ambiguous answer for each of the burner types tested. As a result, the discussion accompanying the experimental results is focussed on identifying a range of possible answers that are applicable for various situations. Ultimately, this report aims to provide guidance for network operators, technical regulators, burner suppliers and end-users to enable the development of the optimal strategy to reduce carbon emissions without major disruption to industrial processes.

2 EXPERIMENTAL EQUIPMENT AND METHODS

2.1 Burners and appliances

The burners tested are new products, purchased from an appropriate supplier, with the exception of a radiant heater, which is on loan from a collaborator. The burners include different types of partially and fully premixed and nozzle-mix designs, representing the main types of burners used in Type B appliances commonly found in hydrogen injection trial areas. The following burners/appliances were tested:

1. A partially-premixed, naturally-aspirated burner with an AN-style nozzle rated at 35 kW (120 MJ/h)
2. A forced draft premixed burner. Riello package burner (Gulliver BS2F) with integral fan, rated at 35–92 kW with a 106 mm diameter burner tube
3. An Eclipse air heat burner, rated at 147 kW (530 MJ/h)
4. A forced-draft North American Tempest 4442 high-velocity nozzle-mix burner rated at ~20 kW (72 MJ/h), using an air pressure of 7 kPa
5. An Airatherm, SuperRay, WEA-24 ported ceramic premixed radiant surface burner. The appliance is rated at 24 MJ/h (7 kW).
6. Pilot burners, flame ionisation rod detectors and UV flame detectors integral to some of these burners

For these appliances, a range of operating conditions were assessed where relevant, and appliance-specific modifications were made in some instances to improve or enable operability with 100% H_2, as detailed in Section 2.3.

The burners selected cover a range of designs and uses. Naturally aspirated burners are used in cooking applications (such as pizza ovens) and in launders in metallurgical industries. Package burners are the most widely used burner in Type B installations, used in boilers, ovens, furnaces, dryers, process heaters, air heaters and others. The Eclipse air heat burners are a linear burner design that produces hot gas with a short flame. They can be ordered with a range of lengths so are useful for heating air flowing in rectangular ducts or in rectangular ovens or processes. The Tempest forced-draft high-velocity burner is designed for operation in high temperature furnaces, with multiple burners typically used in a single furnace. In such an environment, the high velocity induces recirculation of furnace gases which facilitates a reduction in NOx emissions and improved flame stability. The radiant surface burner is a Type A appliance, but is similar in design to many used as Type B appliances in ovens and drying applications.

2.2 General testing methodology

Tests were performed across the normal operational range for each burner, to assess the safety and performance of the burners up to a maximum hydrogen content. Initially, H_2 concentrations of 10%, 21.7% and 31% (by volume) in natural gas were studied. Note that the blend of 31% was chosen because it provides a margin above other potential blend concentrations of 25% or 30%. The $\%H_2$ was then incrementally increased to identify a limit (if applicable), and normal adjustments and minor modifications/retrofits were made to facilitate operation up to 100% H_2 where possible. The general approach that was implemented for the various burners is outlined in the following steps, with appliance-specific details provided in Section 2.3 (note that not all of the below steps were performed for all of the burners, due to differences in their operation, e.g. not being able to adjust regulator pressures):

7. Burner is installed according to the manufacturer/supplier instructions, with appropriate adjustments made for operation at maximum rate on natural gas (e.g. adjusting air settings and regulator pressures). Emissions recorded on natural gas to ensure the burner is operating as expected and to obtain baseline.
8. Ignition, flame stability (e.g. light-back) and emissions are examined for 10%, 21.7% and 31% H_2 in NG blends with no changes made (including maintaining regulator pressure such that heat input is reduced in line with the Wobbe Index). Pre-blended bottles with 10, 21.7 or 31% by volume hydrogen in methane were used for short duration tests, such as ignition tests. For longer duration tests, mass flow controllers were used to accurately blend the specified ratio of hydrogen (from a Grade 5.0 cylinder) with natural gas from the network at the specified ratio.
9. To determine the maximum hydrogen blend concentration without changes, the burner is operated and the $\%H_2$ is increased incrementally above 31% whist sampling flue gases, until one of the following occurs:

a. Main flame or pilot propagates upstream of the burner exit/combustion head (i.e. light-back) or otherwise becomes unstable

b. Flame detector (i.e. flame rod or UV detector) fails to detect the flame

c. Flame-induced noise levels increase significantly beyond what was experienced during normal operation

d. Significant over-heating of the burner is observed/measured in areas not designed for high temperatures.

10. Ignition safety is then assessed at this maximum %H_2.
11. Steps 3 and 4 are repeated at the turn-down setting if applicable, with appropriate adjustments made for NG as in Step 1.
12. Steps 3 and 4 are repeated while maintaining constant heat input equal to the maximum firing rate, by setting H_2 and NG flowrates with flow controllers (see below for details).
13. Adjustments (e.g. aeration settings and UV detector installation) are made where possible, and Steps 3 and 4 are repeated to determine a "maximum hydrogen % with minor modifications".
14. Additional modifications are then made where appropriate to accommodate 100% H_2.

The simplest modification which can be made to the burners is adjusting the flowrate of fuel being combusted, by changing the inlet pressure to the burner. End-users will likely look to maintain their output characteristics if the fuel is varied, such as heating an equivalent volume of water to the same temperature in the same amount of time, or maintaining the throughput of raw materials into a furnace. It is therefore of interest to keep the heat input to the burner (per unit time) constant., which—if thermal efficiency remains constant—would result in constant useful heat output. This also enables a more direct comparison of heat transfer and process performance metrics.

The addition of H_2 to the fuel stream up to approximately 80% leads to an initial reduction in the supplied heat input for a constant inlet pressure, after which there is an increase in heat input with further addition of H_2. This is a consequence of the variation in the Wobbe Index with H_2 addition to natural gas, as shown in Figure 9. Therefore, it is necessary to change the inlet pressure to maintain constant heat input, or alternatively, the size of the orifice in burners featuring an injector can be increased to provide additional heat input without changing the pressure. Since it is generally simpler to increase a regulator set pressure rather than changing injectors for each fuel blend, the constant heat input tests were performed without changing injectors. The pressure on the regulator was set higher than required, and the flowrates of H_2 and natural gas were independently controlled by mass flow controllers to achieve the required heat input and blend ratio, with the flowrates of H_2 and natural gas calculated in advance based on the higher heating value of the fuel mixture.

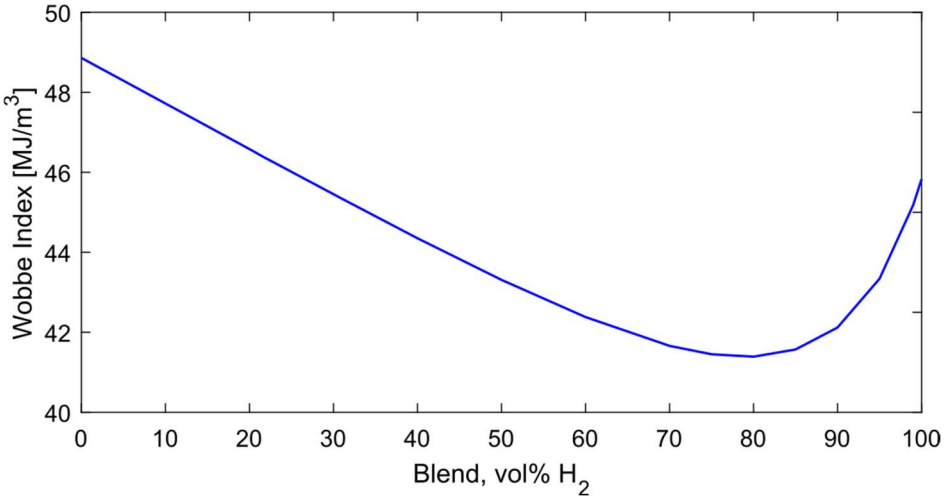

Figure 9: Wobbe Index as a function of volumetric H_2 content in natural gas. Note that the limits for the Wobbe Index of natural gas as per AS 4564:2020 are 46–52 MJ/m³.

The following subsections outline in greater detail the specific experimental techniques used to study the burners/flames. It should be noted that some of these tests were only performed on a limited set of flames for the various burners in order to achieve a compromise between capturing a range of burners/fuel blends, and obtaining fundamental insights that are applicable to other burners.

2.3 Description of appliances and adjustments/modifications

2.3.1 Atmospheric nozzle (AN) inspirator burner

The AN burner (shown in Figure 10) is unique among the burners tested in that it relies upon a passive air entrainment mechanism rather than a forced air mechanism. It features an adjustment screw to control the level of primary aeration, such that very little air is entrained with this screw completely tightened. Based on information from the burner provider, this type of burner is typically operated at the maximum aeration setting when using natural gas; that is, with the screw loosened off. Therefore, the tests were initially performed with maximum aeration when adding hydrogen to the fuel. The control system also offers a HIGH and LOW setting for the burner, with the HIGH setting corresponding to 35 kW and LOW corresponding to 20 kW on natural gas (the LOW setting is adjustable via a separate pressure regulator but was maintained at the original "as delivered" setting for this testing).

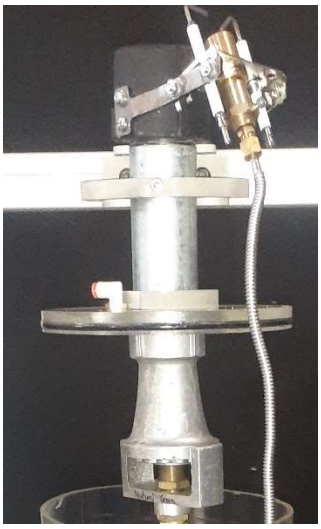

Figure 10: AN burner with pilot assembled.

The burner also features a pilot flame, which has a separate air entrainment mechanism. The burner control system initially ignites this pilot flame, and the flow of gas to the main burner is activated if the presence of the pilot flame is successfully detected by a flame rod detector. The main flame is initially ignited at LOW rate, and can only then be increased to the HIGH setting.

Steps 1–8 (in Section 2.2) were carried out for this burner, with the following adjustments/modifications made to increase the allowable hydrogen content:

15. **Minimal aeration**: Air entrainment screw fully tightened to limit entrainment, and entrainment gap for pilot burner is blocked.[7]
16. **Flame detection:** UV flame sensor installed instead of flame rod (in conjunction with limiting aeration).

In addition to the standard emissions and flame stability tests, a more detailed series of tests was performed for this burner to provide fundamental insights in terms of the combustion behaviour as hydrogen is added. In order to quantify the level of primary aeration and the effect that this has on the flames, an apparatus was developed to measure the rate of air entrainment. This consists of a sealable, cylindrical enclosure which mounts to the burner

[7] It should be noted that preventing entrainment in the pilot burner required the installation of a physical "collar" around the entrainment section and is therefore a modification rather than an adjustment.

and surrounds the injector nozzle and aeration opening, as pictured in Figure 11. The enclosure contains two inlets at the base which enables air to be introduced to the cavity, with the flowrate of this air controlled and metered via a mass flow controller (MFC). The inlets are fitted with a diffuser, such that the flow disturbance within the enclosure is minimised. There is also a pressure tapping at the top of the enclosure to enable the pressure to be observed with a digital manometer. The general operating principle is that the burner is turned on and the fuel flowrate set while the enclosure is sealed off (i.e. air flowrate set to zero via the MFC), such that a negative pressure is produced within the enclosure due to the entrainment effect. Air is then introduced incrementally via the MFC, until the enclosure reaches atmospheric pressure as read via the manometer, allowing sufficient time to observe any unsteadiness. At this point, the air being introduced into the enclosure (and thus, the amount of air entrained by the burner) is equal to the entrainment rate of the burner when operated without the enclosure. This enables the change in air entrainment with varying hydrogen content to be characterised, as well as the effect of changing the degree to which the adjustment screw is opened/closed.

Figure 11: Enclosure for measurement/control of aeration for the AN burner.

Using this enclosure, a range of different conditions were studied, largely focussed on optimising the performance at 100% H_2. Flame photography, heat flux measurements and emissions sampling were performed for the various cases with constant heat input, with comparisons made against the equivalent natural gas flames. The enclosure was also used to completely seal the aeration inlet in some instances, because tightening the “aeration screw” as far as possible does not completely prevent air from being entrained, since it does not form a perfect seal against the nozzle. Additionally, the amount of air entrained at this “minimum” setting appears to be sensitive to the exact position of the screw and is not a repeatable condition. Therefore, rather than performing tests at this minimum aeration setting, entrainment of air was prevented altogether via the enclosure when performing the emissions and thermal heat flux measurements.

2.3.2 Riello package burner

The package burner that was tested is a Riello BS2F model (shown in Figure 12), with an operating range from 35–92kW. It features a fan which operates at a single speed, with an adjustable damper to modify the flowrate of air. It also features a combustion head with a unique geometry to provide fuel staging for low NOx emissions, whereby a portion of the fuel exits at the base of the head in a low-pressure region, accompanied by six fuel tubes angled away from the central axis and spaced concentrically downstream of the initial combustion region. The air enters through a perforated plate at the base of the burner, and a portion also flows around the outside of the combustion head. The outer housing is seen to taper inwards at the exit of the burner, presumably to promote greater mixing between fuel and oxidant. The relative position between the combustion head and this outer

housing must be adjusted to a specific setting depending on the heat input, as outlined in the manufacturer's installation instructions.

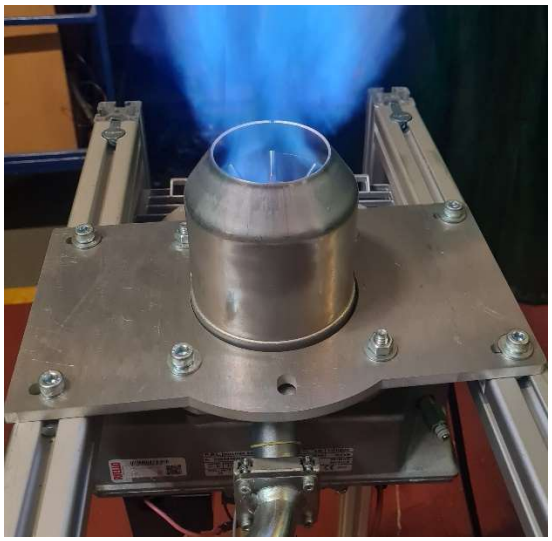

Figure 12: Riello BS2F package burner.

The package burner has a wide range of operability, and the fact that the flowrates of fuel and air are decoupled from one another (i.e. there is no ratio controller) means that the number of different combinations that could be studied is essentially unbounded. It was therefore decided that the burner would be operated at its minimum and maximum rated heat input, and two air damper settings—corresponding to how the burner would typically be operated on natural gas—were studied as hydrogen was added incrementally. Similar to the AN burner, the package burner was provided with a flame rod for flame detection, and was also tested with a UV detector. In addition to measuring emissions for the various cases and capturing photographs of the flames, the effect of hydrogen addition on the temperature of the burner head was also investigated via infra-red imaging.

This burner also features an option for replacing the combustion head and outer housing with a "towns gas" head. Since towns gas is known to contain up to 60% H_2, it is feasible that this modification would be advantageous for operation on 100% H_2. Therefore, the burner was tested with this alternative configuration after the initial testing, with a focus on higher levels of H_2 up to 100%. The towns gas burner head differs from the natural gas burner head in having a large central axial hole which allows the exiting gas to produce a longer flame with less fuel/air mixing in the burner housing. This is a more appropriate design for blends with greater hydrogen content because they are likely to have less stability problems, and the design may reduce overheating and NOx concerns.

2.3.3 Eclipse air-heat burner

The Eclipse air-heat burner (AH0050)—pictured in Figure 13—is a line-type burner typically used for producing large volumes of hot air in applications such as ovens, dryers and fume incinerators. The burner has an adjustable firing rate ranging from 10–147 kW on natural gas (controlled via a pressure regulator), with an integral blower to supply air to the burner, which is controlled via an adjustable damper. The appliance also features a method of premixing a small portion of the supplied air into the fuel stream when the burner is fired at low rates. This is achieved via a check valve which opens when the gas pressure is lower than the air pressure at the inlet to the burner, providing a more stable flame when operating at low rate due to this partial premixing. At high firing rate, the gas pressure is higher than the air pressure, and the air and fuel streams are separated prior to mixing in the burner exit section.

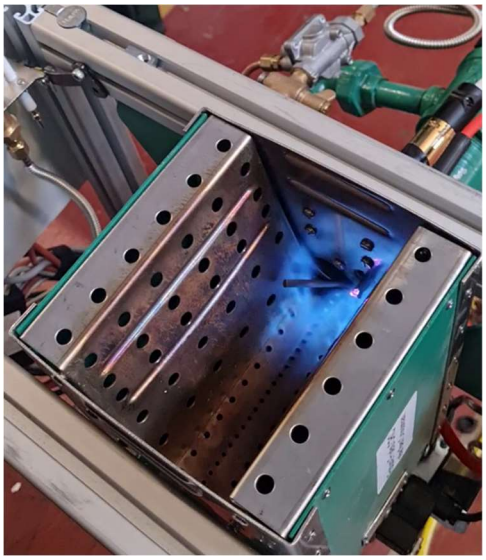

Figure 13: Eclipse air-heat burner during ignition on natural gas.

2.3.4 Tempest high-velocity nozzle-mix burner

Nozzle-mix burners are those in which fuel and air enter the combustion zone separately prior to mixing and ignition, commonly used in furnaces/kilns. The burner tested is a North American Tempest burner (model 4442, maximum heat input of ~70 MJ/h), shown in Figure 14. The burner produces a narrow, high-velocity flame which burns partially within a mullite combustion chamber enclosed in a refractory block before exiting through a small hole into the furnace/kiln (or, in this case, into the open atmosphere). The high-velocity jet flame leads to high levels of entrainment of the surrounding gas, which promotes exhaust gas recirculation when fired into an enclosed environment, in-turn leading to improved combustion stability and reduced NOx emissions following the warm-up period. Since the current testing is limited to open-firing, the main focus here is on the ignition behaviour and flame stability as hydrogen is added, rather than emissions, which will be investigated further in the following phase of this project. This burner was provided with a UV sensor for flame detection.

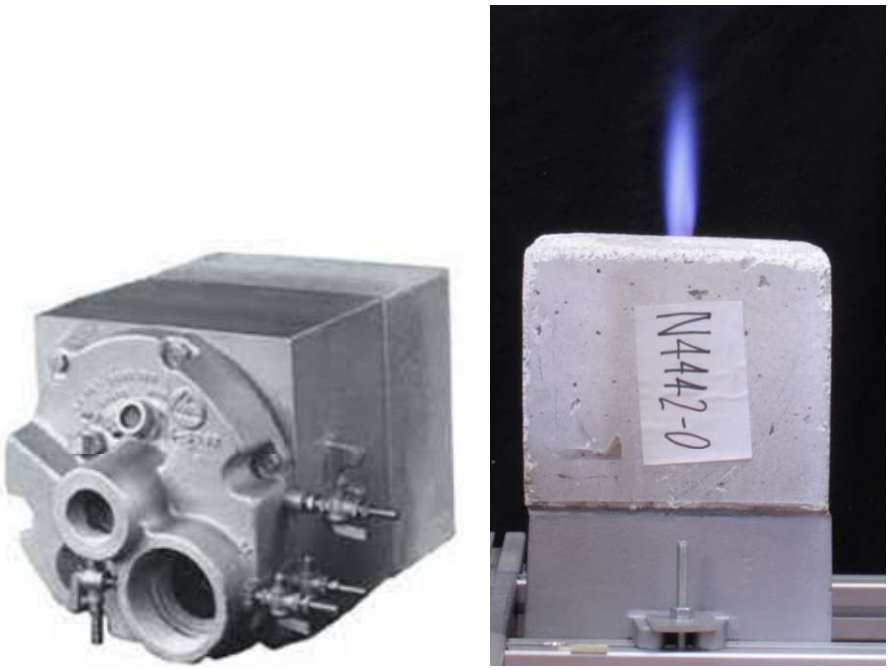

Figure 14: Left: Tempest burner assembly (from techritecontrols.com.au), and right: burner in operation on natural gas.

The valve-train used for the Tempest burner also features an air-fuel ratio regulator, which enables a constant excess air setting to be used across the turn-down range following installation. The appliance was initially configured to provide an excess air ratio of 30% on natural gas, with the air and fuel flowrates both monitored via mass flow controllers. After confirming that the ratio controller operated as intended on natural gas across the operating range, the effect of hydrogen addition on this behaviour was then investigated without making any changes to the valve-train configuration.

2.3.5 Radiant surface burner

An Airatherm, SuperRay, WEA-24 ported ceramic premixed radiant surface burner was tested. The appliance is rated at 24 MJ/h (7 kW). This is a Type A appliance but is similar in design to many used as Type B appliances in ovens and drying applications. There are three separate burners each with an injector that entrains air into a chamber behind a ceramic burner. The igniter is on one side of the three burners and a flame detection thermocouple is located on the other side. The burners cross light across the face of the three ceramic elements. The appliance features a non-adjustable built-in regulator such that it was not possible to perform the tests with constant heat input.

2.4 Measurement techniques

The effect of hydrogen addition on combustion behaviour was studied via flame photography, radiative heat flux measurements and emissions sampling. For all burners, a series of photographs were captured with a DSLR camera to investigate the overall appearance of the flames in terms of colour, luminosity and flame shape. Heat flux measurements were performed for both the AN burner and the radiant heater, using a Medtherm 64-Series heat flux transducer. In the case of the AN burner, the sensor was fitted with a view restrictor, such that it only measured the radiation from a localised portion of the flame. The sensor was then traversed along the length of the flame, enabling an axial profile of the thermal radiation from the flame to be generated.

For all burners tested, the composition of the flue gas stream was measured via multiple gas analysers. NO_x (NO and NO_2) measurements were of particular interest, since hydrogen addition may promote NO_x formation due to the possibility of higher flame temperatures. The concentrations of CO, CO_2, O_2, and unburnt fuel (CH_4 and H_2) were also measured using gas analysers.

Two different Teledyne “T-Series” gas analysers were used; a T200 analyser was used to measure concentrations of NO_x and O_2, while a T300 was used for CO and CO_2 measurements. The NO_x analyser measures NO via the chemiluminescence method and calculates the NO_2 concentration by switching modes every other measurement period, such that the NO_2 is converted to NO using a catalyst, enabling the total NO_x concentration to be measured and the NO_2 fraction to be determined via subtraction. This allows both NO and NO_2 to be measured down to the ppb level with an accuracy of ±0.5% of the reading above 50 ppb. The CO measurements also have an accuracy of ±0.5% of the reading above 5 ppm, while the CO_2 sensor has an accuracy of ± (0.02% + 2% of reading). The O_2 sensor has an accuracy of ±0.1%. The Teledyne gas analysers were factory-calibrated, and an in-house calibration/linearity check was also performed using span gases and dilution with Grade 5.0 N_2 via the mass flow controllers to ensure the readings in the measurement range were within the stated margin of error. The gas samples for the two analysers were taken continuously from a single sampling point using a stainless-steel probe fitted with a thermocouple at the tip for monitoring temperature during the measurements.

In order to compare the emissions results for different fuels and with different levels of dilution, it is necessary to convert the raw readings of pollutant concentration (e.g., ppm CO or NO_x) to an “emission rate”. This rate has units of mass per unit energy (e.g., mg/MJ), and essentially represents the total mass of the pollutant emitted, normalised against the total heat input of the fuel. For example, using this measure, a 10 kW flame that produces 10 mg of CO every hour would produce an equivalent result to a 1 kW flame that produces 1 mg of CO in the same time period. To calculate this emission rate, the pollutant concentrations measured from the diluted sample (i.e. from the flue gases mixing with the surrounding air) must be multiplied by a “dilution factor”, to yield the concentration that would be present if the sample was not diluted and the combustion process occurred with stoichiometric air. In practice, this can be performed using either the measured CO_2 or O_2 concentration in the diluted sample, since the theoretical, undiluted concentrations are known in advance for a given fuel mixture. It should be noted, however, that the CO_2 method (which is used for calculating the NO_2 emission rate for certain Type A appliances in the AS/NZS 5263 standard series), is not applicable to 100% H_2 since there is no carbon in

the fuel. Therefore, in the presentation of the emissions results in this report, the rates shown are calculated based on the O_2 concentration in the diluted sample. The procedure for calculating the emission rate from the diluted flue gas sampling for a variable H_2:NG blend ratio based on these two methods is described in Appendix A.

3 RESULTS

3.1 Maximum H_2 content with no modification and normal adjustments

3.1.1 Atmospheric nozzle (AN) inspirator burner

To assess the effect of hydrogen addition on the operability of the AN burner, a range of scenarios corresponding to the adjustments outlined below were tested:

17. **No change**: Burner operated with maximum aeration at normal inlet pressure at high and low rate; that is, at 35 kW and 20 kW on natural gas, respectively.
18. **Minimal aeration**: Air entrainment screw fully tightened to limit entrainment, and entrainment gap for pilot burner is blocked.
19. **Constant heat input, max aeration**: Inlet pressure increased and flowrates controlled to give desired blend ratio with constant heat input (35 kW).
20. **Constant heat input, min aeration:** As above, but with aeration limited via tightening screw.
21. **Flame detection:** Steps 1–4 repeated with UV flame sensor instead of flame rod.

For each of the above adjustments, the burner was initially ignited on natural gas, and the H_2 concentration was incrementally increased until a limiting condition was observed. The results of these tests, in terms of the maximum $\%H_2$ for each case and the limiting factor(s) are shown in Table 4.

Table 4: Maximum H_2 level under various "normal adjustment" scenarios, and reason(s) for this maximum.

Scenario	Max H_2 Concentration	Reason
No change (high rate)	65%	Light-back (pilot + main flame)
No change (low rate)	55%	Light-back (main flame)
Minimal aeration (flame rod)	80%	Flame detection
Minimal aeration (UV sensor)	100%	Nil
35 kW, max aeration	70%	Light-back
35 kW, min aeration (flame rod)	80%	Flame detection
35 kW, min aeration (UV sensor)	100%	Nil

In addition to incrementally increasing the $\%H_2$ with the burner already lit, the AN burner was also tested in terms of the ignition process via the burner management system. It was ultimately found that ignition did not present a lower limit on $\%H_2$ below the limit imposed by light-back at low rate—that is, 55% H_2 for the case of maximum aeration. With the aeration screw completely tightened, ignition was safely achieved at 100% H_2 when using the UV sensor. Further details regarding the appearance of the flames, thermal radiation and emissions for the AN burner are provided in Section 3.2.1.

3.1.2 Riello package burner

The Riello package burner (Gulliver BS2F) is a single stage, forced air burner with a heat input range of 35–92 kW. It operates at a single fan speed, with an adjustable damper to control the amount of air that is delivered. It also features an adjustable combustion head position, which is set to a specific location for a given heat input in order to optimise the combustion process (e.g. to achieve low NO_x emissions). Due to the wide range of adjustability of this burner, and the fact that the air and fuel settings are essentially decoupled, it is necessary to sample the flue gases while adjusting the air damper such that an operating setting with both relatively low CO and NO_x emissions is achieved.

It was found that the NO_x and CO emissions when running on natural gas were particularly sensitive to the air damper setting, with a high air flowrate leading to very high CO emissions in some cases (presumably due to flame quenching/extinction from the excess air). Furthermore, the addition of hydrogen was seen to both improve and worsen this behaviour depending on the specific combination of flowrate and air damper setting. It was not possible to measure the flowrate of air with this burner under the open firing conditions tested; this aspect of the burner will instead be quantified when performing the confined firing tests in the next phase of this project. To gain an overall understanding of the effect of hydrogen addition across the range of operation, the burner was

operated at its maximum and minimum firing rate (i.e., 35 and 92 kW on natural gas), and two air damper settings were considered for each firing rate; one of which produced relatively high CO levels under natural gas, and the other producing low CO but higher NO_x emissions (emissions results are presented in Section 3.2). The maximum allowable hydrogen content with no further changes for each of these cases was then investigated.

Unlike the AN burner in which the flame was observed to light-back, the package burner did not present a hard limit on hydrogen content. In terms of flame stability, there does not appear to be any limit on hydrogen content, and the burner was able to be operated at 100% H_2 with the only required change being the replacement of the flame rod with the UV sensor, for all four combinations of gas pressure and air damper setting. Interestingly, the flame rod was able to detect the flame at H_2 concentrations up to 99% for all cases investigated, but was found to shut-down for all cases for 100% H_2. This indicates that the presence of small quantities of hydrocarbon impurities has a profound impact on the flame ionisation signal. It is also interesting to note that the flame rod failed at a much lower concentration of 80% H_2 for the AN burner, which suggests that the flame rod was either positioned in a more intense flame region for the package burner, or the circuitry was configured to allow a wider range of signal intensities.

While there is no obvious cut-off point in regard to a maximum $\%H_2$, the major concern that was observed for this burner as the H_2 concentration was increased was a visible increase in the temperature of the combustion head, with the metal of the burner producing a more intense glow as hydrogen was added. This is expected, due to the higher flame temperature of hydrogen coupled with the higher reactivity, which leads to more combustion occurring within the burner head rather than outside of it. To study this change in temperature, a thermal imaging camera was used to image the temperature inside the combustion head immediately after turning off the flame (noting that this technique cannot be used while the flame is present), for the two air settings used at high firing rate. Note that this is by no means intended to be used as a quantitative measure to set a limit on $\%H_2$. Rather, it is intended to provide an indication of the effect of hydrogen addition on the tendency of the burner to overheat; information which is useful for end-users in terms of monitoring the condition of their burners and developing maintenance schedules when operating with hydrogen blends. Figure 15 displays the maximum measured temperature over a 50 second duration after turning off the burner for the various fuel blends up to 100% H_2 for the “low air” cases, while Figure 16 compares the thermal images for the natural gas and 100% H_2 cases at a time of 5 seconds after turning off the flame for the two air settings.

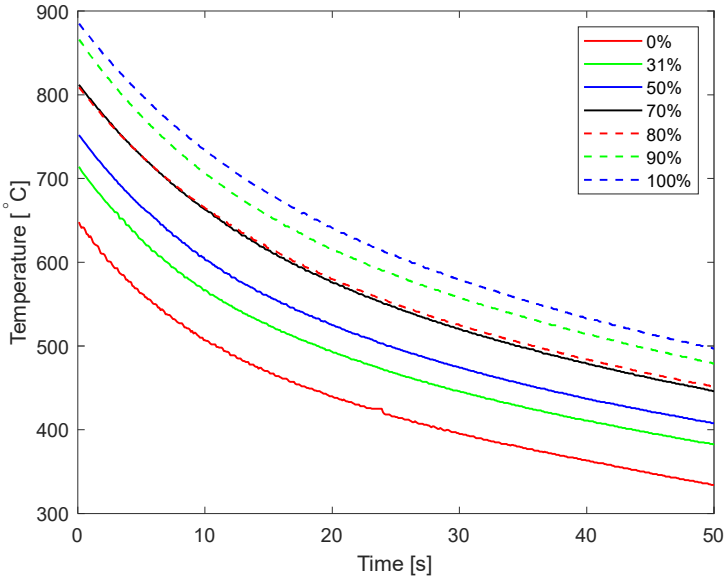

Figure 15: Decay of maximum burner temperature after flame is turned off, for 0–100% H_2.

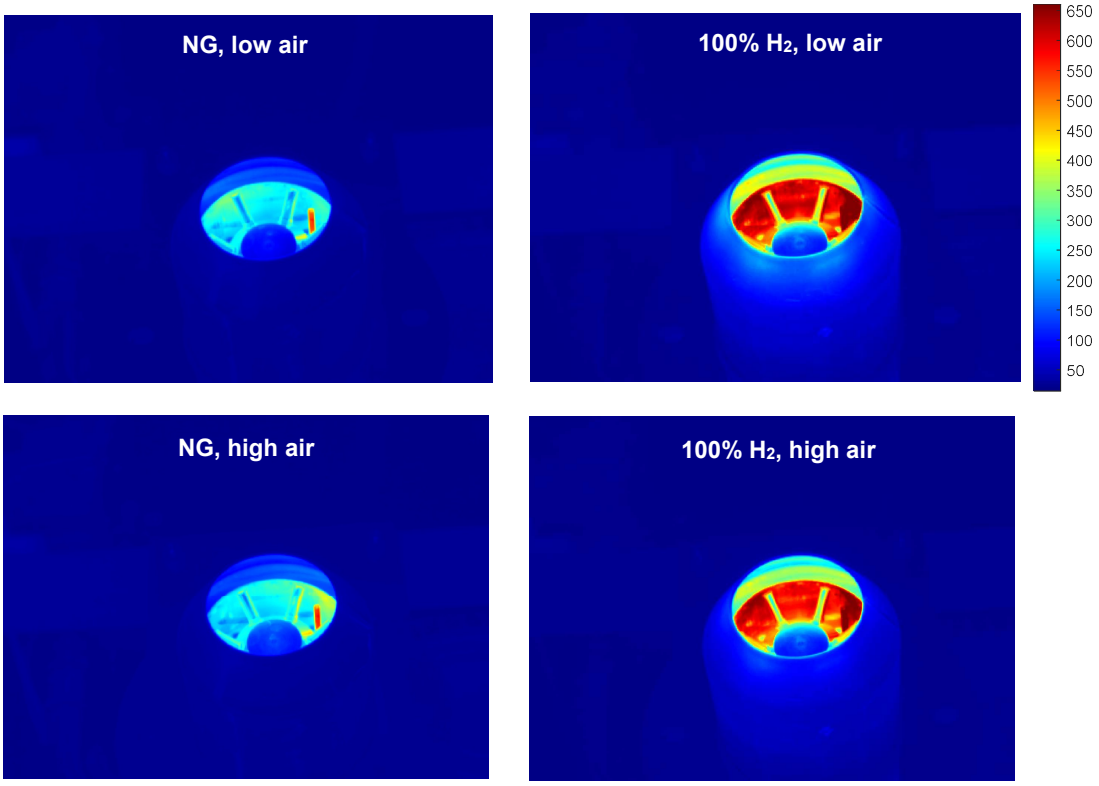

Figure 16: Thermal imaging results for natural gas and 100% H_2, at a time of 5 s after flame is turned off. Note that the same colour scale limits are used in all of the images shown.

These results highlight the significant impact of hydrogen on the heating of the burner head. It should be noted that no physical damage to the burner was observed when operating on 100% H_2, although the duration of the tests was limited to approximately 5 minutes. It is difficult to comment on the importance of this additional heating due to hydrogen addition, without knowing the exact details and ratings of the materials the burner head comprises. The specific application in which the burner is used will also determine whether or not the excess heating is a limiting factor; for example, if it is used in a furnace/kiln then it is possible that parts of the burner could become even hotter than they otherwise would. A potential option for addressing the overheating concern is by using an alternative “towns gas” combustion head which is offered by the manufacturer, which leads to more combustion occurring outside of the burner. A comparison of the operation with the towns gas head and original configuration is included in Section 3.2.2.

3.1.3 Eclipse air heat burner

To determine the maximum allowable hydrogen percentage in a blend with no changes or minor changes, light-back is one issue to investigate. At low rate (10 kW) blending ratios were trialled up to 80 vol% hydrogen with good operation, but light-back occurred with the 90% hydrogen blend and with 100% hydrogen. However other issues arise with blends with lower percentages of hydrogen. Noise is one such issue. Low-rate tests (10 kW) were performed with the check valve operating as designed. Gas pressure to the burner was adjusted to maintain constant heat input as the blend ratio varied. The check valve was in operation at these low rates, providing some air for premixing into the fuel before exiting the small fuel holes in the base of the burner. For the 10, 20, 31 and 40 vol% hydrogen blends, the noise differences from natural gas case were not significant. At low rate the 50 vol% hydrogen blend produced significant combustion noise, which increased further at higher vol% blends. This combustion noise may be the limiting factor which determines a maximum practical limit in the hydrogen ratio in the blend, occurring at lower ratios than light-back. To investigate further, the check valve was removed, and pipe sealed so there was no premixing. Repeating the tests at 10 kW for blends stepping up from natural gas to 100% hydrogen in 10 vol% steps produced insignificant noise for all blends. This illustrates that removing the check valve and preventing premixing is a potential simple solution for the noise issue, but it would also make the flame less stable if the burner was used on natural gas.

Burner overheating and high temperature damage is another issue that will limit the amount of hydrogen that can be blended into natural gas. Operation with a 40 vol% hydrogen blend at 10 kW indicated that the burner was visibly hotter than with natural gas, a trend that intensified as more hydrogen was added to the blend. Brief tests with 100% hydrogen at higher rates rapidly produced glowing red metal with air set at either 2.5 mbar or 1.0 mbar. In progressing towards full rate with 100% hydrogen, the rate was turned up as far as 85kW with air at 1.0mbar, but the V-shaped burner air diffuser rapidly became red hot, and the test was aborted to prevent burner damage. Overheating was even more rapid with the air set at 2.5 mbar. An attempt was made to measure the temperature of the metal in region where it appeared hottest, but the recordings were not accurate due to the cooling effect of air in addition to the flame. There is no simple retrofit that can address the overheating issue. It requires a burner design change to reduce mixing of air and fuel within the burner, which although achievable is a step beyond the scope of this report.

To summarise, with no changes except an increase in gas pressure, the burner is suitable with up to 31 vol% hydrogen blended into natural gas and probably with a 40 vol% hydrogen blend. With higher hydrogen percentages, overheating or noise will limit operation.

3.1.4 Tempest high-velocity nozzle-mix burner

After installing the Tempest burner and adjusting the regulators for reliable ignition and air-fuel ratio control on natural gas, hydrogen was added to the fuel stream with no other changes to the burner and control system. Interestingly, it was found that the ignition process became quieter in comparison to natural gas as hydrogen was added up to 100%. This is in contrast with ignition results for the other burners, in which there is generally increased noise when igniting with high levels of hydrogen due to the higher flame speed. The behaviour observed for the nozzle-mix burner appears to be related to the igniter being located away from the fuel injection point, relying on the air stream mixing with the fuel and transporting it towards the igniter. Since hydrogen has much wider flammability limits in comparison to natural gas, this ignition process occurs faster with higher levels of hydrogen in the fuel, preventing the build-up of unburnt gas prior to ignition.

The flowrates of air and fuel were also controlled/monitored as hydrogen was added, to investigate the change in air-fuel ratio and the behaviour of the ratio regulator. For this burner, the turn-down setting is controlled by the air flow; that is, the air flowrate determines the fuel flowrate. In practice this would be achieved using an air blower with a variable output, while in the experiments the air was supplied from a compressed air line and controlled using a mass flow controller. Three different air flowrates were studied, corresponding to three different nominal heat inputs on natural gas, specifically 21 MJ/h, 36 MJ/h and 72 MJ/h. The air-fuel ratio regulator was set to provide approximately 30% excess air on natural gas. The combination of fuel and air settings for natural gas are summarised in Table 5 below.

Table 5: Operational settings on natural gas, and calculated excess air.

Heat input (MJ/h)	Air flow (SLPM)	NG flow (SLPM)	% Excess air (vol)
21	125	10	31.3
36	207	17	29.4
73	415	34	28.2

The three air flowrate setpoints shown in Table 5 were maintained as hydrogen was added to the fuel, while the Alicat flow controllers were used to control the relative flow between hydrogen and natural gas to give the desired concentration of H_2. This allows the operation of the burner with variable levels of hydrogen addition to be explored; results are shown in Figure 17, in terms of the change in; (a) excess air percentage (which is determined by the air-fuel ratio and fuel stoichiometry), (b) total fuel flowrate, and (c) heat input.

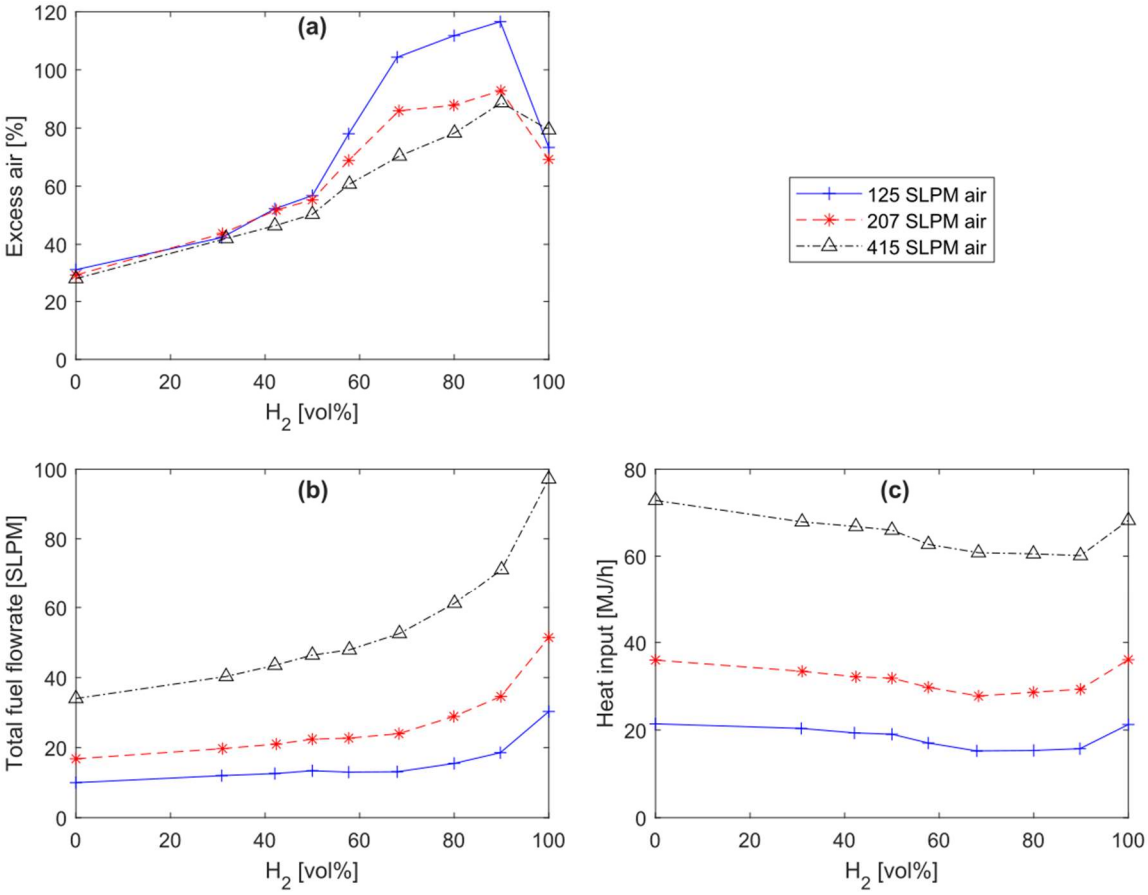

Figure 17: Effect of hydrogen addition on excess air percentage, total fuel flowrate and heat input, for the three air flowrates (which correspond to the three turn-down settings).

The various plots displayed in Figure 17 offer an interesting insight into the operation of burners with an air-fuel ratio controller up to 100% H_2. Since H_2 requires four times less air in comparison with CH_4 (by which natural gas can be approximated) for a stoichiometric mixture, an increase in hydrogen content leads to an increase in the amount of excess air for a fixed fuel and air flowrate. However, the reduced density of H_2 leads to increased flow for a fixed inlet pressure to the burner, leading to a competing effect between the change in total flowrate of fuel and the change in stoichiometric air-fuel ratio. Interestingly, this leads to a general increase in excess air up to 90% H_2, followed by a reduction for all three turn-down settings for 100% H_2 due to the more significant increase in flow from 90%–100% H_2. In terms of heat input, a similar trend is observed, whereby there is a general reduction with increasing hydrogen addition up to approximately 70% H_2, and an increase for higher H_2 concentrations.

In addition to the change in excess air due to stoichiometry and total gas flow, there also appears to be a change in the ability of the ratio controller to maintain a fixed air-fuel ratio for the different fuel blends. While the fuel flowrate is shown to increase as the air flowrate is increased for all blend ratios, the behaviour appears to be non-linear for high concentrations of H_2 in the fuel mixture; that is, the proportion by which the fuel flowrate changes is not equal to the proportion by which the air flowrate changes. This is most clearly seen in Figure 17(a) for the cases with %H_2 ranging from 60–90%, where there are large discrepancies in the excess air values for the same H_2 concentration. This indicates that re-tuning of the ratio controller is necessary in order to achieve the correct behaviour across the turn-down range, although it should be noted that this would likely be required anyway due to the change in stoichiometry. It is also interesting to note that the reduction in excess air with a further increase to 100% H_2 is accompanied by a more stable behaviour of the ratio controller.

3.1.5 Radiant heater

Light-back on ignition results suggest that the maximum practical hydrogen blend percentage is 31 vol%. The burners did not light-back on ignition with natural gas nor with any blend up to 31 vol% hydrogen. With 40 vol% hydrogen light-back on ignition did occur in 2 out of 10 attempts on the centre burner. After this result, tests were

repeated in more detail with the 31 vol% blend with fifteen tests conducted with the burner in both hot and cold condition. There was no light-back in any of these further tests using the 31 vol% blend. Tests were then repeated with 40 vol% blend and light-back occurred two out of five times when hot and one out of five times when cold.

Further tests were conducted for light-back during normal operation after ignition. In these tests the burner was lit on natural gas and the blend percentage of hydrogen was gradually increased whilst maintaining constant heat input. In these tests light-back did not occur up to 70 vol% hydrogen but occurred at 80 vol% hydrogen. However, the previous light-back on ignition tests will set a practical limit of less than 40 vol%. The appliance also has a built-in gas pressure regulator which is not adjustable, so it was not feasible to repeat the testing at constant heat input. Due to the nature of the burner, it was not possible to make any other simple adjustments/modifications to increase the allowable hydrogen content.

3.1.6 Summary of operation with no changes and normal adjustments

The five appliances have each been tested to identify the maximum allowable H_2 concentration under a range of operating conditions and with minor adjustments where feasible. The burners were initially operated without any alteration (including keeping the regulator set to the nominal pressure based on the rated natural gas heat input), and those burners which came with a burner management system (both for ignition and flame monitoring) were operated with this system in place. For burner systems which are designed to be adjusted as part of "normal operation" (such as changing aeration, for example), appropriate adjustments were made with the intent of accommodating higher levels of hydrogen in the fuel. Many of the burners featured a flame ionisation/rectification rod as part of the burner management system to detect the presence of a flame, and these flame rod detection systems tend to be problematic at high levels of H_2. Therefore, in situations where the flame rod was found to be the limiting factor regarding maximum H_2 content, it was replaced by a UV sensor to detect the presence of the flame, which allowed the other aspects of the ignition/control system to be operated normally.

The operation of the various burners, in terms of their maximum allowable H_2 concentration prior to an unsafe condition or forced shutdown due to flame detection, is summarised below. This is shown for both the "no change" scenario (Table 6), and allowing for "normal adjustments" (Table 7), such as replacing the flame rod with the UV sensor or maintaining constant heat input. The following section will investigate the combustion performance and appearance of the flames up to this maximum hydrogen concentration.

Table 6: Maximum %H_2 for the various burners with no modification and the reason for this maximum.

Appliance/burner	**Max H_2 [%]**	**Reason/observation**
AN burner (open)	55	Light-back at low rate
AN burner (closed)	80	Flame detection (flame rod)
Riello package burner	99	Flame detection (flame rod) – possibly lower (overheating)
Eclipse air-heat burner	40–50	Overheating of burner (+ noise)
Tempest nozzle-mix burner	100	No issues in flame detection or stability
Radiant burner	40	High probability of light-back on ignition

Table 7: Maximum %H_2 for the various burners with normal adjustments and the reason for this maximum.

Appliance/burner	**Max H_2 [%]**	**Change made**	**Reason/observation**
AN burner (open)	70	Constant heat input of 35 kW	Light-back
AN burner (closed)	100	Flame rod replaced by UV sensor	No limit observed
Riello package burner	100	Flame rod replaced by UV sensor	Overheating may be a concern
Eclipse air-heat burner	40–50	Check valve sealed off	Overheating of burner
Tempest nozzle-mix burner	100	No change necessary	No limit observed
Radiant burner	40	No adjustment feasible	High probability of light-back on ignition

3.2 Flame behaviour and emissions

3.2.1 Atmospheric nozzle (AN) inspirator burner

As discussed in Section 3.1.1, the AN burner was operated at both the maximum aeration setting and with the aeration inlet closed off. With maximum aeration, the burner could be operated at up to 70% H_2 (at constant heat input) before light-back into the mixing tube occurred, while 100% H_2 was possible with the aeration limited and the UV sensor installed. To provide further insights into the combustion behaviour with hydrogen addition under both of these scenarios, additional tests were performed focussing on flame appearance, thermal radiation and emissions (specifically CO and NOx), using a constant heat input of 35 kW. For the "minimal aeration" scenario, the aeration inlet was completely sealed off for these tests, to ensure consistency between different cases. Figure 18 and Figure 19 display a series of photographs at a range of H_2 concentrations with the aeration inlet open and closed (respectively) for the AN burner.

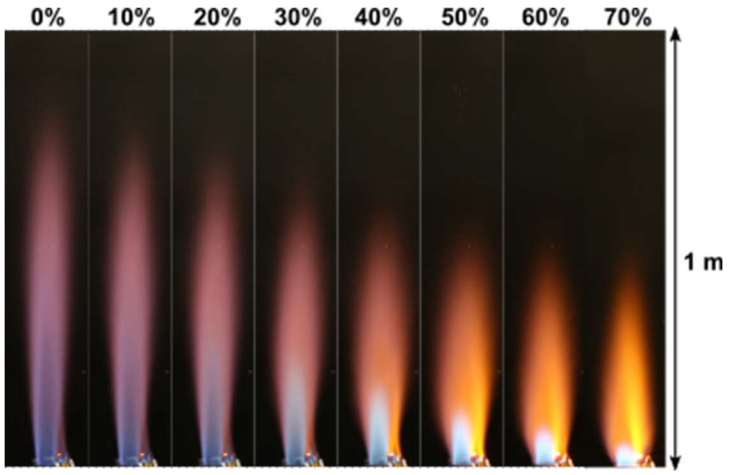

Figure 18: Photographs of AN burner flames with hydrogen addition at constant heat input. Images correspond to a constant exposure time of 3 s and ISO setting of 400.

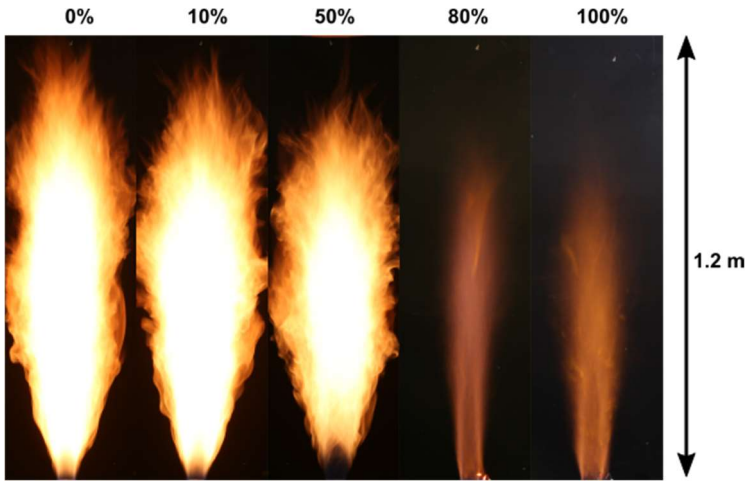

Figure 19: Photographs of AN burner flames with constant heat input and no primary aeration, ranging from 0–100% H_2. Cases with 0%, 10% and 50% H_2 correspond to an ISO setting of 100, while 90% and 100% cases were captured with an ISO setting of 800.

The photographs of the aerated flames shown in Figure 18 display a consistent trend in flame appearance as hydrogen is added to the fuel up to 70%. An approximately linear reduction in the visible flame length is apparent up to 50%, as well as a widening of the flame near the base which becomes particularly noticeable for H_2 concentration > 30%. These trends are expected, due to the higher flame speed, reduction in air required for stoichiometric combustion, and the higher diffusivity of hydrogen in comparison to methane. Additionally, the increase in injection pressure (and therefore the velocity of fuel exiting the injector) leads to increased air

entrainment in the burner venturi as hydrogen is added, which further changes the equivalence ratio. It is also interesting to note the change in colour of the flames, from a relatively pale flame which transitions from blue to red along the length of the flame in the case of natural gas, to a flame with an intense light-blue colour in the "inner cone" region at the flame base with a yellow-orange tail for the 70% H_2 case. The formation of the inner cone structure with the addition of H_2 is particularly interesting, as it suggests that there are distinct premixed and non-premixed (diffusion) regions of the flame. The orange colour that can be seen for the >30% H_2 flames is most likely due to the excitation of sodium ions in the air, which has been observed in a previous Future Fuels CRC project (RP1.10-04).

The photographs of the flames with the primary aeration inlet closed off (Figure 19) show a significantly different appearance to those in Figure 18. The influence of primary aeration on the sooting behaviour of the flames is notable in comparing the cases up to 50% H_2, with a highly luminous region for the non-premixed flames apparent due to this soot. It is interesting to note that the addition of 50 vol% H_2 to natural gas does not significantly change the overall flame appearance, although there is a reduction in flame length and a delayed onset of the sooting region. It should be mentioned that this type of burner is generally not operated with the aeration inlet closed off; as such the natural gas flame shown does not represent the typical flame appearance, except in certain cases where a highly luminous and radiant flame is desirable, such as in some pizza ovens. At 80% H_2 this sooting behaviour is seen to disappear, noting that the orange colour in the 80% and 100% H_2 flames is due to sodium excitation. Comparing the 80% and 100% H_2 flames in Figure 19 to the aerated natural gas flame in Figure 18 (which represents the baseline condition), it is interesting to observe the similarities in the flames in terms of their width and height. This is shown more clearly in Figure 20 which compares the 100% H_2 non-premixed flame to the partially premixed natural gas flame.

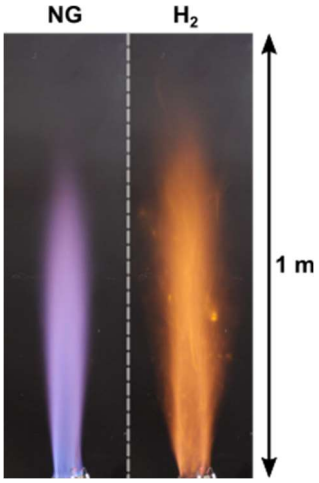

Figure 20: Comparison of natural gas (with aeration) and 100% H_2 (no aeration) flames for the AN burner.

In addition to their visible appearance, the flames were also studied in terms of the heat radiated from the flames via measurements of radiative heat flux. The heat flux sensor was traversed along the length of the flame to obtain an axial profile of thermal radiation for the various cases, with the measurement locations ranging from 100–1300 mm downstream of the burner exit. Figure 21 displays these results for the natural gas, 10% H_2 in NG, and 50% H_2 in NG cases with the aeration inlet open, while Figure 22 shows the results up to 100% H_2 with no aeration.

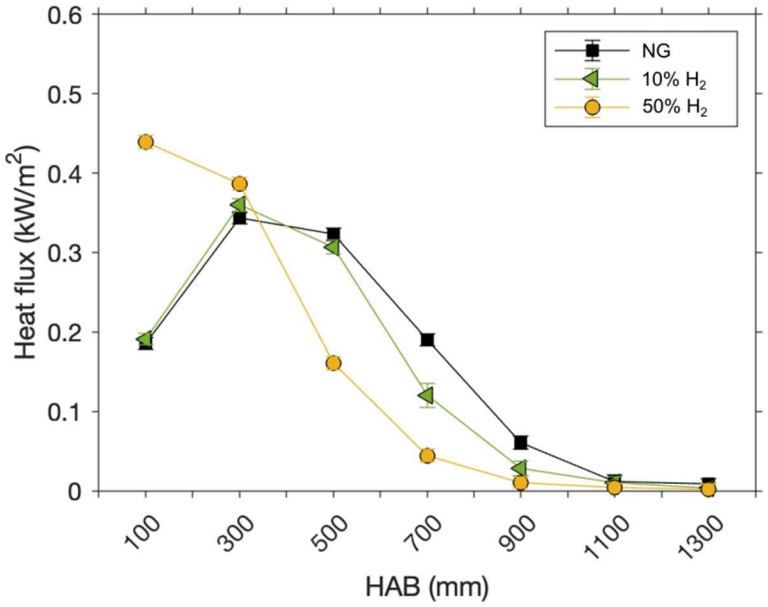

Figure 21: Axial profiles of radiative heat flux as a function of the height above the burner exit (HAB) for flames with 0, 10 and 50% H_2 in the fuel stream, for the AN burner with maximum aeration setting and constant heat input.

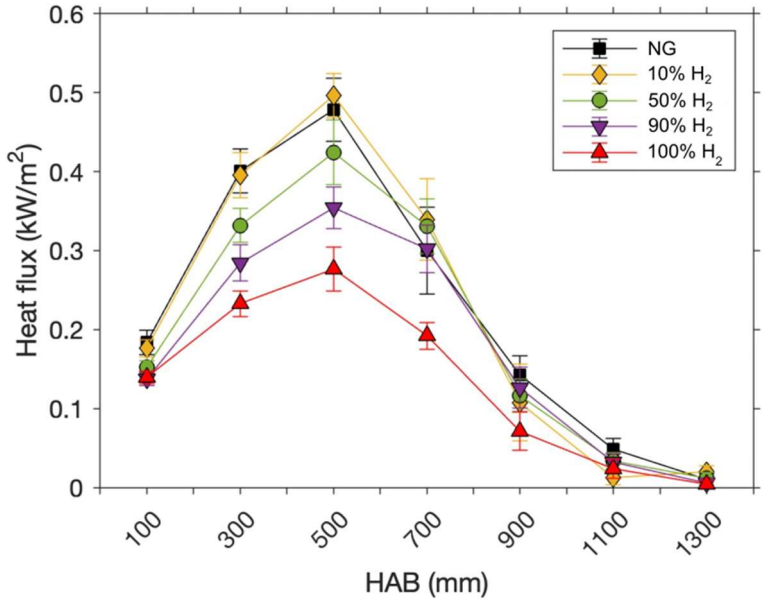

Figure 22: Axial profiles of radiative heat flux for flames with 0, 10, 50, 90 and 100% H2 in the fuel stream and no primary aeration, for the AN burner with constant heat input.

The axial heat flux profiles in Figure 21 show an interesting change in behaviour as hydrogen is added up to 50%. The 10% H_2 in NG and the purely NG cases display a similar profile, with the 10% H_2 case having a slightly higher peak at 300 mm, followed by a steeper reduction at locations above this height. The 50% H_2 case is seen to produce much higher levels of thermal radiation near the burner exit, although this radiation then decreases rapidly for heights above 300 mm from the burner exit, below the values measured for the 10% H_2 and NG flames. These results are consistent with what is expected based on the photographs shown in Figure 18, where the reaction zone appears to be much more intense near the burner exit for the 50% H_2 flame, with a significantly reduced flame length. The cases without aeration (shown in Figure 22) can be seen to all show a similar axial profile, with a consistent reduction in peak thermal radiation as the hydrogen concentration is increased. It is again interesting to note the similarities between the 100% H_2 flame without aeration and the natural gas flame with aeration, which also have similar axial profiles, although the natural gas flame is seen to produce slightly higher thermal radiation in regions near the jet exit.

The exhaust gas composition was also measured for these flames, to study the change in NOx emissions with hydrogen addition. To compare the results with different levels of dilution due to the entrained air, the NOx concentrations measured from the samples were normalised to their "non-diluted" values, based on the O_2 concentration measured in the diluted sample and the fact that the stoichiometric combustion products nominally have 0% O_2. These results are shown in Figure 23, for the cases with aeration (up to 50% H_2) and without aeration (up to 100% H_2). Also shown in Figure 23 are the emissions results with primary air entrainment corresponding to 10% of stoichiometric air for each fuel blend, which was performed using the sealable enclosure.

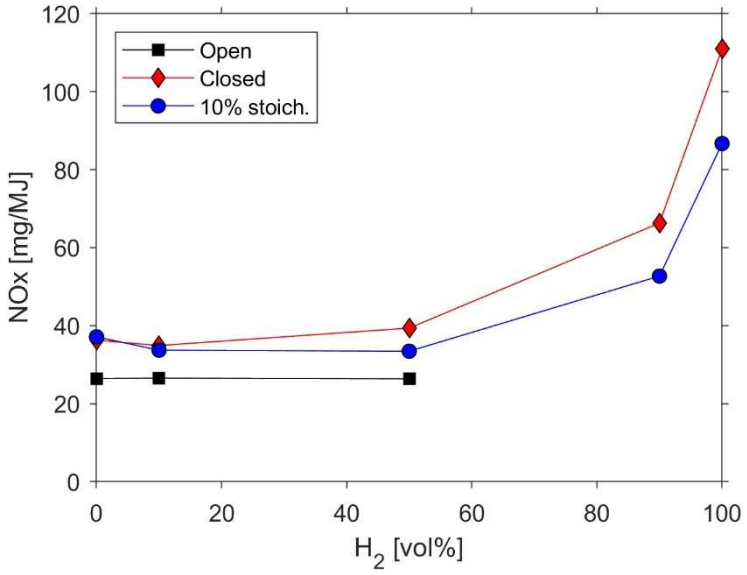

Figure 23: NOx emissions (corrected for dilution and normalised against heat input) for the AN burner, for aerated, non-aerated and 10% stoichiometric air cases from 0–100% H_2.

A significant increase in NOx emissions is observed for the series of flames with no primary aeration, particularly as the hydrogen content is increased beyond 50%. This indicates that—although limiting aeration is effective in providing a flame similar in length, shape and thermal radiation to the baseline natural gas case—additional steps must be taken to maintain acceptable NOx concentrations. The addition of 10% stoichiometric air is seen to reduce the NOx emissions, although a noticeable increase is still apparent for 100% H_2.

3.2.2 Riello package burner

As mentioned in Section 3.1.2, the package burner was able to operate up to 100% H_2 without any modification to the burner. There is, however, a noticeable change in appearance as hydrogen is added, in addition to the increase in the temperature of the burner as discussed previously. Photographs of a series of flames corresponding two different air flowrates at the minimum and maximum firing rate are shown in Figure 24, again showing the characteristic orange colour of the hydrogen flames.

As mentioned in Section 2.3.2, the package burner was also tested with the "towns gas" head modification, to explore the viability of this option for 100% H_2. A comparison of the appearance of the towns gas head flames (with maximum air damper setting) against the original configuration is shown in Figure 25, again for natural gas and 100% H_2 (note that towns gas was not actually used with the towns gas head). A noticeable change is observed with the towns gas head, where much more of the combustion appears to occur outside of the burner head, with the soot formation for the natural gas flame suggesting much less mixing between fuel and air. During the experiments, it was also observed qualitatively that the burner did not appear to overheat to the same extent with the towns gas head in place, suggesting that this could be a viable option for applications where the flame length and shape is not important.

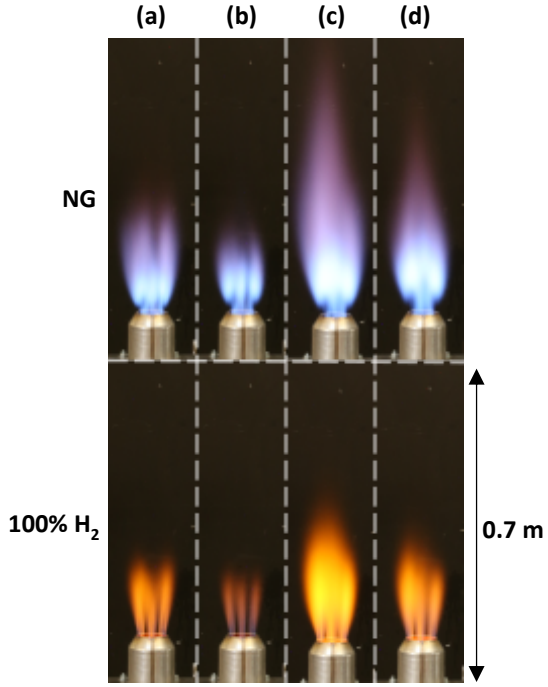

Figure 24: Photographs of package burner flames with natural gas and 100% H_2, for high and low air settings at minimum and maximum rate. From left to right, columns correspond to: (a) low rate, low air, (b) low rate, high air, (c) high rate, low air, (d) high rate, high air.

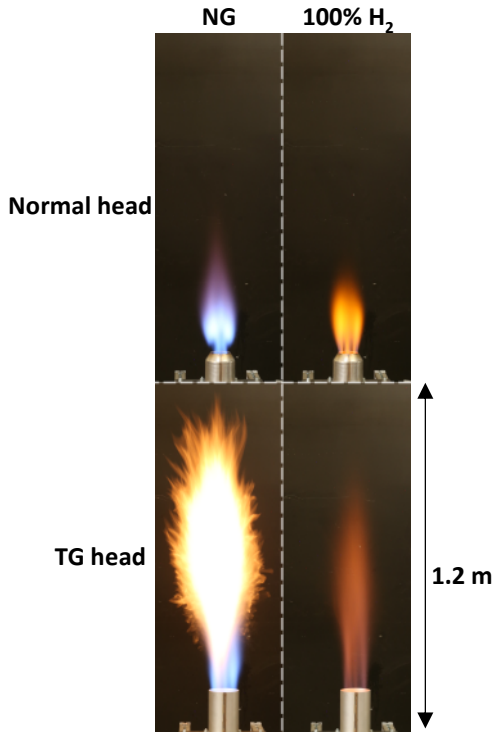

Figure 25: Comparison of flames using both the original combustion head and the towns gas head, for natural gas and 100% H_2.

Emissions were also measured for both combustion head configurations to explore the change in CO and NOx formation as hydrogen is added under the different scenarios. These results are shown in Figure 26, for the two air settings at high firing rate for the original head, and at maximum aeration for the towns gas head, for H_2 concentrations ranging from 0–100% (only 40–100% shown for TG head). The towns gas head and “high air”

flames are seen to produce relatively high CO emissions which trend to zero as the hydrogen content is increased (which is a reflection of the carbon content in the fuel and not the combustion process), while the low air setting has generally low CO emissions across the range of H_2 concentrations, which is at the expense of higher NOx emissions. The towns gas head produces slightly lower NOx emissions compared to the low air case, with the high air case with the original burner head slightly lower again. For all cases, a noticeable increase in NOx formation can be seen for 100% H_2, although this increase is relatively minor compared to the results of the AN burner.

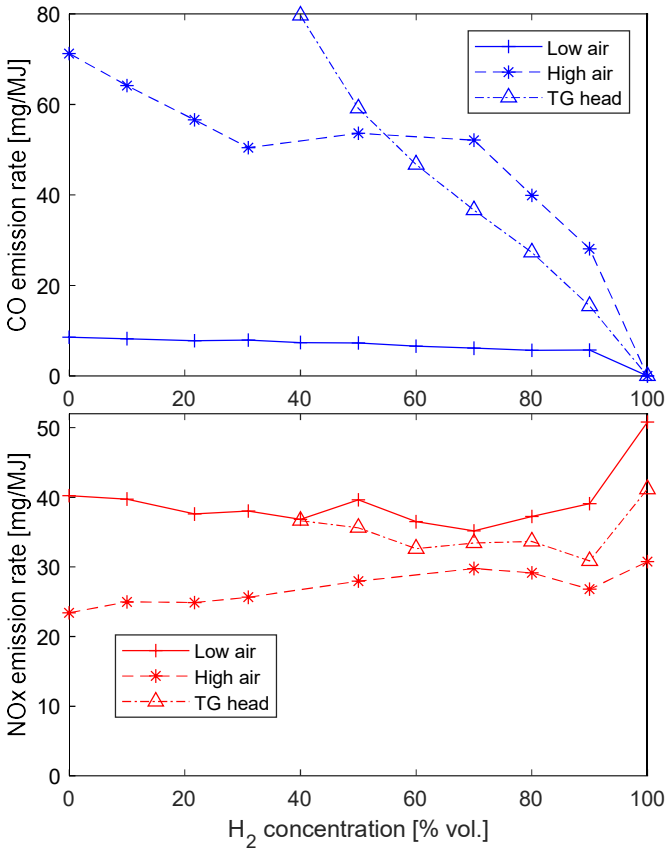

Figure 26: CO (top) and NOx (bottom) emissions for the package burner with three operation conditions: Low and high air flowrate with the original burner head, and high air with the towns gas (TG) head.

3.2.3 Eclipse burner

Emissions were also measured for the Eclipse burner up to a blend of 31% H_2 in natural gas. Similar to the package burner, emissions were measured with the burner fired both at the maximum and minimum rate (147 kW and 10 kW, respectively), with air supplied at a pressure of both 1.0 and 2.5 mbar for the two firing rates. These results are shown graphically in Figure 27, for the four different operating conditions.

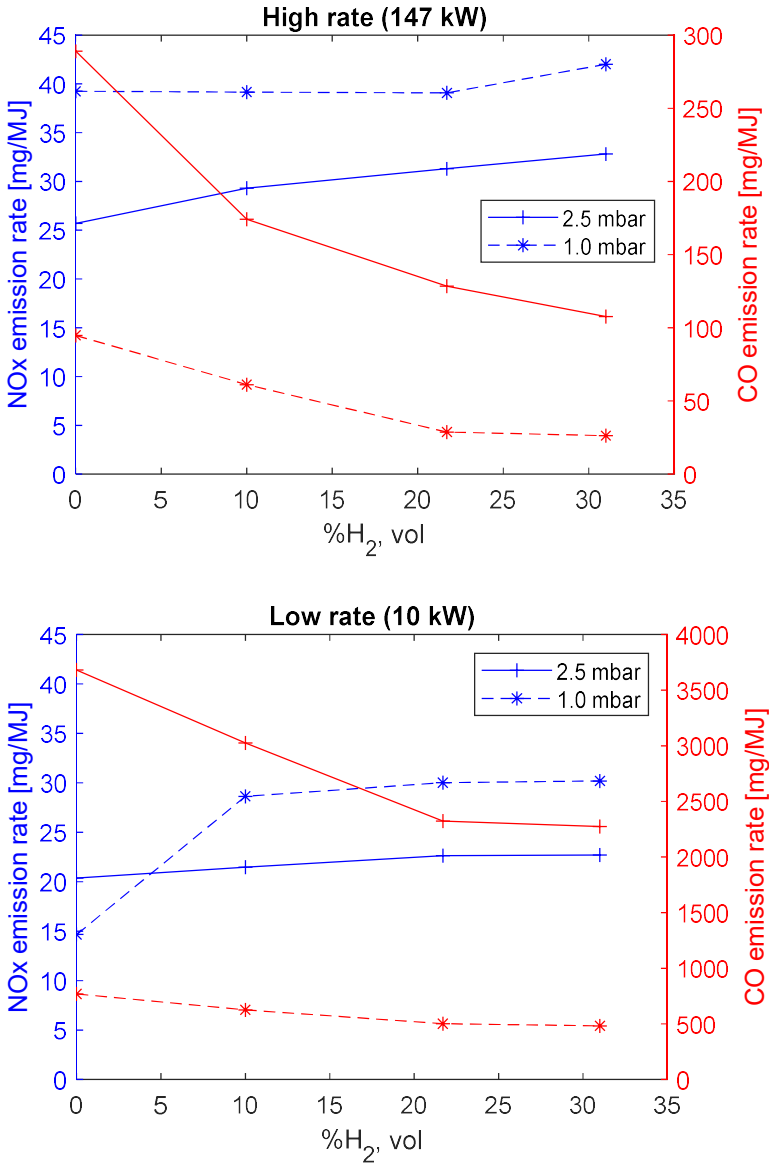

Figure 27: NOx emissions (left axes, in blue) and CO emissions (right axes, in red) for the Eclipse burner at high and low rate, with air supplied at both 1.0 and 2.5 mbar for each firing rate.

These results show that there is a general reduction in CO emissions as hydrogen is added, accompanied by an increase in NOx formation for all cases. The most notable increase in terms of NOx emissions is when blending 10% H_2 into the fuel for the 10 kW case with reduced air supply, in which there is an approximate doubling of the NOx emission rate from 15 to 29 mg/MJ, with further increase in H_2 concentration having relatively little effect. The results for the different air settings also indicate that higher inlet pressure (and hence flowrate) on the air line can reduce NOx emissions, albeit at the expense of higher CO emissions. Ultimately, the emissions data indicate that the operation of this appliance is satisfactory up to 31% H_2, although it may be necessary to adjust the air setting to prevent an increase in NOx emissions for certain applications.

NOx emissions in practical ovens, kilns, boilers etc. will vary from the results recorded in the open firing tests reported here due to the different temperature environments and amounts of excess air used, and in some cases, the use of further NOx reduction techniques. The open firing tests only give an indication that addition of up to 31 vol% hydrogen is likely to only cause small increases in NOx, and that for some burners 100% hydrogen may cause significantly higher NOx, and so will need to be studied further in confined experiments.

3.2.4 Tempest high-velocity nozzle-mix burner

As mentioned previously, the focus for the nozzle-mix burner was on the effect of hydrogen addition on ignition and air-fuel ratio control rather than emissions, the results of which were presented in Section 3.1.4. To highlight the change in flame appearance as hydrogen was added to the fuel, photographs of the flames up to 100% H_2 are displayed in Figure 28.

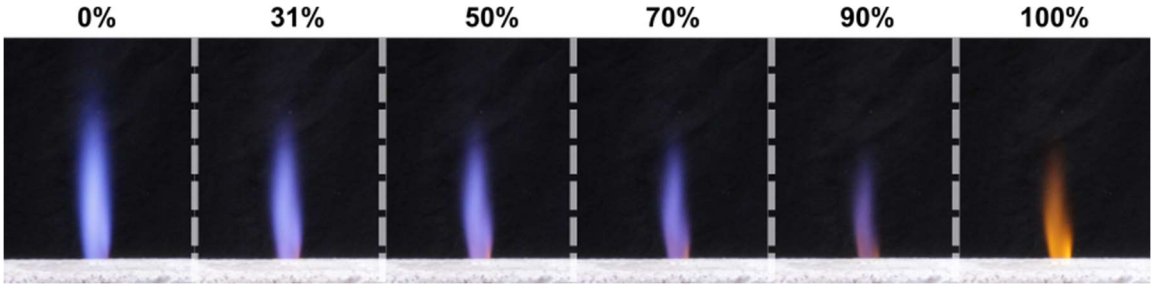

Figure 28: Photographs of Tempest burner flames with air set to 415 SLPM.

3.2.5 Radiant heater

For the radiant heater, emissions and radiative heat flux measurements were performed for natural gas and blends of 10, 21.7 and 31% H_2 in natural gas. These results are shown in Table 8.

Table 8: Emissions and radiative heat flux results for H_2 blends in natural gas up to 31% for the radiant heater.

H_2 content (%vol in NG)	NOx [mg/MJ]	CO [mg/MJ]	Radiative heat flux [kW/m^2]
0%	8.3	324.3	1.26
10%	7.7	273.0	1.27
21.7%	6.7	231.8	1.28
31%	5.9	205.6	1.27

The results shown in Table 8 indicate that the radiant heater performs well with blends of hydrogen up to 31%. In fact, both the CO and NOx emissions (normalised against heat input) can be seen to decrease with hydrogen addition, while the thermal radiation (as measured by the heat flux sensor) changes very little for the different blends. The consistent radiative heat flux is particularly interesting considering the reduction in heat input with increasing hydrogen addition (due to the reduction in Wobbe index of approximately 7% from 0–31% H_2, as shown in Figure 9), which indicates that the appliance has an increased efficiency with increasing hydrogen content up to 31%.

4 CONCLUSIONS

The compatibility of five gas burners with variable blends of hydrogen in natural gas has been investigated. The burners were selected to encompass a range of different appliance types that are commonly used in industrial and commercial applications in Australia. In addition to providing confidence for the introduction of hydrogen to the gas distribution network in concentrations of up to 20 vol%, this report also examines the potential for the various appliances to accommodate higher blend concentrations, as well as 100% H_2.

The naturally aspirated AN burner was found to operate safely on up to 55% H_2 with no changes made, with higher concentrations of hydrogen susceptible to light-back at low rate. With the primary air inlet closed and the ionisation rod substituted with a UV sensor for flame detection, the AN burner could be operated at up to 100% H_2, noting that the flame rod becomes unreliable at approximately 80% H_2. Radiative heat flux measurements show a reduction in thermal radiation with hydrogen addition for the cases with no primary aeration, although the 100% H_2 flame produces a similar result to the fully aerated natural gas baseline case. Emissions were measured while the entrainment of air was controlled/measured via a sealable enclosure, and it was found that entrainment of 10% of the stoichiometric air produces lower NOx emissions in comparison to the cases with no aeration, although there is a significant increase in comparison to the natural gas cases.

The package burner that was tested was able to be operated up to 100% H_2 at both high and low rate and with the air damper set to provide different air flowrates. The only change that was required to achieve this was the replacement of the flame rod with a UV sensor, noting that the flame rod was able to detect the flame at approximately 99% H_2 but not at 100% H_2 across the operating range. This suggests that the flame ionisation signal is highly sensitive to low levels of natural gas in the fuel. While the burner could be operated at 100% H_2, excessive heating of the burner head may present a lower limit, although this is difficult to quantify. An alternative towns gas combustion head was fitted to the burner and was found to accommodate 100% H_2 with no indication of overheating and only a slight increase in NOx emissions relative to natural gas.

An air-heat burner was also found to be susceptible to overheating with hydrogen addition, accompanied by a significant increase in flame noise for H_2 concentrations above 40%. Additionally, when the check valve which provides partial premixing at low rate was not sealed, light-back was observed at low firing rate at 80% H_2. The emissions results show that an increase in air pressure can effectively reduce NOx, albeit at the expense of CO emissions. Since CO emissions tend to decrease with hydrogen addition and NOx emissions can be higher, this may be an effective measure to achieve a compromise between NOx and CO.

Following a standard installation for natural gas, a high-velocity nozzle-mix burner could be safely operated up to 100% H_2 with no subsequent changes. The air-fuel ratio controller was found to be effective in modulating the gas flowrate, although the lower stoichiometric air requirement of hydrogen relative to natural gas leads to an increase in the excess air percentage up to 80% H_2, followed by a reduction up to 100% H_2 due to the non-linear increase in gas flowrate.

Finally, a radiant surface burner showed good performance up to 31% H_2, with radiant heat flux measurements indicating that the efficiency increases with hydrogen addition, as well as a reduction in both CO and NOx emissions. While incrementing the hydrogen content following ignition allows operation up to 80% H_2 (at which point light-back occurs), light-back during ignition was found to occur approximately one in five times at 40% H_2, presenting a practical upper limit for this appliance.

5 IMPLICATIONS AND RECOMMENDATIONS FOR INDUSTRY

The results and conclusions in this report illustrate that in Type B appliances there will be no difficulty using the 10 -20 vol% hydrogen blend ratios likely to be available in distributed gas.

In some local areas it may be possible to deliver distributed gas with higher hydrogen percentages than used in general. If this is the case it appears that Type B appliances indicative of those models tested here will be capable of handling 30 vol% blends without changes, and up to 70 vol% with minor changes for some burners, but complete replacement for others. Likewise for conversion to 100% hydrogen some burners need to be completely replaced, whilst others need only minor changes, although practical NOx emissions and heat transfer assessments need to be performed, and changes made to mixing and other NOx controls may be required.

The current report details open firing tests, which are useful for determining any issues with ignition and light-back and providing indicative emission trends. Confined testing in the next part of this project will provide insight on the practical blend ratios possible in geometries representing ovens, furnaces, boilers and water tube heaters.

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APPENDIX 1

Formula/method for determining "undiluted" pollutant concentrations from diluted gas sampling, using either CO2 or O2 to determine "dilution factor" (for either a dry or wet flue gas sample)

Background:

Assume we have a mixture of CH_4 and H_2, and let x be the molar/volumetric fraction of CH_4, and $(1-x)$ the fraction of H_2.

We can then consider the methane/hydrogen mixture as having the chemical formula "C_xH_y", where $y = 4x + 2(1-x) = 2x + 2$.

The general formula for the stoichiometric combustion of the fuel with a mixture of 21% O_2 and 79% N_2 is as follows:

$$C_xH_y + \left(x + \frac{y}{4}\right)(O_2 + 3.76N_2) \rightarrow xCO_2 + \frac{y}{2}H_2O + \left(x + \frac{y}{4}\right)3.76N_2$$

Substituting in $y = 2x + 2$:

$$C_xH_{2x+2} + (1.5x + 0.5)(O_2 + 3.76N_2) \rightarrow xCO_2 + (x+1)H_2O + (1.5x + 0.5)3.76N_2$$

Analysis:

Let us initially consider the scenario where we measure a diluted (with air) sample of flue gases without condensing the H_2O in the exhaust gases (i.e. "wet" measurement)

The molar concentration of CO_2 in the undiluted flue gases is calculated from the molar coefficients of each of the combustion products as follows (from the balanced eq. above):

$$[CO_2]_{w,stoich} = \frac{x}{x + (x+1) + 3.76(1.5x + 0.5)} = \frac{x}{7.64x + 2.88}$$

While the concentration based on a dry sample (in which H_2O is condensed out) is given by:

$$[CO_2]_{d,stoich} = \frac{x}{x + 3.76(1.5x + 0.5)} = \frac{x}{6.64x + 1.88}$$

Additionally, since the combustion is stoichiometric, the concentration of O_2 in the undiluted flue is obviously zero based on both the dry and wet flue gas.

Now, suppose we measure a certain O_2 concentration in the diluted flue sample:

$$[O_2]_{w,meas} = z$$

Where ***z*** is the measured molar fraction.

Since all of this O_2 comes from the surrounding air, this means that the total concentration of "diluting air" is equal to $4.76z$ (since the ratio of O_2 to N_2 is 1:3.76).

We can then use this to determine a "dilution factor" (D) based on the O_2 measurement. First of all, D is defined as the ratio of the undiluted molar species concentrations to the diluted concentrations, or equivalently, the factor by which the "measured" readings are multiplied in order to give the stoichiometric or undiluted concentration. Since the mole fraction of air is equal to $4.76z$, the remainder comes from the flue gases:

$$[flue] = 1 - 4.76z$$

By definition, the mole fraction of the undiluted flue is equal to unity, such that we have the following:

$$D = \frac{1}{1 - 4.76z}$$

Noting that this is equivalent to $\frac{21\%}{21\% - [O2]_{meas}}$

We can also get the dilution factor by measuring the CO_2 concentration in the diluted sample. Firstly, based on the wet sample:

$$D_{wet} = \frac{[CO_2]_{w,stoich}}{[CO_2]_{meas}}$$
$$= \frac{x}{7.64x + 2.88} \times \frac{1}{[CO_2]_{meas}}$$

Or, if the sample is dry:

$$D_{dry} = \frac{[CO_2]_{d,stoich}}{[CO_2]_{meas}}$$
$$= \frac{x}{6.64x + 1.88} \times \frac{1}{[CO_2]_{meas}}$$

Now, if we measure a pollutant (such as NOx) in the diluted flue sample, this can be converted to the "undiluted" concentration using the dilution factor. First of all, consider the case where we use the dilution factor based on the O_2 reading (i.e. correcting to 0% O_2), assuming the pollutant is NO_2:

$$[NO_{2,undil}] = [NO_{2,meas}] \times D$$
$$= \frac{[NO_{2,meas}]}{1 - 4.76z}$$

Recalling that z is the measured O_2 concentration.

If we assume the concentration of NO_2 in the undiluted exhaust gases to be small (e.g. <1000 ppm), then the number of moles of NO_2 per mole of fuel is given by the following:

$$n_{NO2} = [NO_{2,undil}] \times \sum_i n_i$$

Where $\sum_i n_i$ is the sum of the molar coefficients of each of the products (including H_2O for wet sampling but excluding it for dry sampling) from the balanced equation. This is essentially the number of moles of combustion products per mole of fuel consumed. So, for the wet sample initially:

$$n_{NO2,w} = \frac{[NO_{2,meas}]}{1 - 4.76z} \times (7.64x + 2.88)$$

Or, for a dry sample:

$$n_{NO2,d} = \frac{[NO_{2,meas}]}{1 - 4.76z} \times (6.64x + 1.88)$$

Now consider if we use the CO_2 reading to do the same correction (firstly based on a wet sample):

$$[NO_{2,undil}] = [NO_{2,meas}] \times D_{wet}$$

$$= \frac{x \times [NO_{2,meas}]}{[CO_2]_{meas} \times (7.64x + 2.88)}$$

And repeating the previous step of converting this to the number of moles per mole of fuel:

$$n_{NO2,w} = [NO_{2,undil}] \times (7.64x + 2.88)$$

$$= \frac{x \times [NO_{2,meas}] \times (7.64x + 2.88)}{[CO_2]_{meas} \times (7.64x + 2.88)}$$

$$= \frac{x \times [NO_{2,meas}]}{[CO_2]_{meas}}$$

So, the term representing the number of moles of products per mole of fuel is seen to cancel out. This also applies if we assume that we have a dry sample and use the corresponding dilution factor:

$$[NO_{2,undil}] = [NO_{2,meas}] \times D_{wet}$$

$$= \frac{x \times [NO_{2,meas}]}{[CO_2]_{meas} \times (6.64x + 1.88)}$$

$$n_{NO2,d} = [NO_{2,undil}] \times (6.64x + 1.88)$$

$$= \frac{x \times [NO_{2,meas}] \times (6.64x + 1.88)}{[CO_2]_{meas} \times (6.64x + 1.88)}$$

$$= \frac{x \times [NO_{2,meas}]}{[CO_2]_{meas}}$$

This shows that if the “dilution correction” is carried out based on the measured CO_2 concentration, then it does not matter if the flue gas sample has the H_2O condensed out or not prior to analysis. This can be understood by considering the fact that the measured CO_2 and measured NO_2 values are offset by the same factor if H_2O is either included or not included.

Note that the previous statement is also true when measuring O_2, but the difference is that the equation for the dilution factor based on O_2 concentration does not factor in whether or not the sample is wet or dry (since the concentration of O_2 in the undiluted flue gas is zero in both instances).

While the equation for n gives the number of moles of pollutant per mole of fuel, the value that we are interested in is the number of moles (or alternatively the mass) of pollutant resulting from a given heat input from the fuel (i.e. the emission rate, E), which depends on the fuel composition. Therefore we have to divide the equation by the HHV of the fuel (on a per mole basis):

$$E = \frac{n}{HHV_{mol}}$$

$$= \frac{n}{HHV_{CH4,mol} \times x + HHV_{H2,mol}(1 - x)}$$

The mass-based emission rate (in units of mass/energy—e.g. mg/MJ) can then be calculated from the above by multiplying by the molar mass of the pollutant of interest, which is approximately 46 g/mol for NO_2.

Summary

- This means that it is very important to know if you are measuring a dry or wet sample for cases with high %H_2, since using the O_2 concentration is either the only option, or the CO_2 method will have high error (e.g. for 90% H_2 where CO_2 concentration is very low)
- For 100% H_2, $x = 0$, so the difference based on using the wet vs dry equation is (1.88/2.88), i.e. using the dry method when it is actually a wet sample will lead to a NOx value that is approx. 65% of what it should be
- The n_{NO2} expressions can be converted to the [mass/energy] value by dividing by the heating value of the fuel (in J/mol), and then multiplying it by the molar mass—i.e. 46 g/mol for NO_2 (and multiplying by the required power of 10 for the appropriate units).

RP1.4-08
Milestone 5: Preliminary confined firing of Type B burners and appliances

14 July, 2023

Project number: RP1.4-08

Detailed assessment of Type B appliances with blends of hydrogen and natural gas and 100% hydrogen

Authors:

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Australian Government
Department of Industry, Science, Energy and Resources

Business Cooperative Research Centres Program

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IMPORTANT DISCLAIMER

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PROJECT INFORMATION	
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Summary of Report

Preparations have been made for confined firing of four different Type B burners in 3 different enclosures that represent different types of ovens, furnaces, or duct heaters. Confined experiments will provide more meaningful emissions and heat transfer results than previous open firing experiments. The enclosures have all been made and one has undergone preliminary testing with one of the burners to develop a methodology for quantifying heat transfer.

The preliminary tests were performed in a vertical duct using a naturally aspirated burner fired at 32 kW. Measurement of fuel flow rate, flue gas temperature and dilution and wall temperatures allowed calculation of heat transfer through walls. The method is still undergoing verification, and in fact further preliminary results than are reported here have already been obtained but are still being processed. The methods will be refined and used to provide properly detailed results for all burners and enclosures in the next Milestone report.

1 INTRODUCTION

1.1 Background and motivation

With continued interest in the use of hydrogen in natural gas distribution networks (both blended and as a complete replacement of natural gas), it is important to understand the effect of hydrogen addition on the behaviour of "Type B" combustion appliances. Type B appliances are typically used in industrial and some commercial applications, and are distinct from "Type A" appliances which are more commonly found in domestic and light-commercial settings. Due to the differences in physical and chemical properties of hydrogen and natural gas, a number of changes in combustion behaviour are expected with hydrogen addition, such as emissions, flame shape/appearance, heat transfer and flame stability/light-back. Some of these changes were explored in the previous milestone report of this project[8], which presents the results of the open firing of a series of Type B burners. While open firing enables the general operation of a burner to be assessed (i.e. impact of hydrogen on flame stability and ignition process), the assessment of the burners under such conditions does not provide representative results for other key parameters, notably heat transfer and emissions.

Measuring the effect of hydrogen addition on emissions and heat transfer from industrial burners during real-world operation is complicated by access limitations and process fluctuations. Instead, measuring emissions and heat transfer in representative laboratory experiments provides good quality data that can be used to validate models of real processes. In this project, laboratory combustion experiments will be performed in which Type B burners (that are typically used in heaters, ovens, furnaces, and kilns) are fired into confinements/enclosures. Since the experiments will use industrial burners they involve representative fuel/air mixing, temperatures, heat transfer and general combustion conditions. The current milestone report deals with establishing the methodology for measuring heat transfer and emissions using one burner mounted in a duct.

1.2 Scope and aims

This project is focussed on the investigation of the effect of hydrogen addition on the safety, performance and emissions characteristics of Type B combustion appliances operating at atmospheric (or near-atmospheric) pressures. Research questions include:

- Does blending 10–20 vol% hydrogen to natural gas present any safety or performance concerns to the operation of atmospheric-pressure Type B appliances, assuming no changes are made?
- Without making any changes, what is the maximum allowable hydrogen concentration in the fuel which can be achieved while maintaining safe operation?
- If applicable, what minor adjustments (e.g. tuning of air-fuel ratio controller) can be performed to facilitate higher hydrogen content, and what is the allowable concentration in this case?
- What types of burners can be operated on 100% H_2 with either no change, minor adjustments, or with a simple retrofit?

The current report aims to provide a background on the methods that will be used to assess confined flames and preliminary results.

[8] RP1.4-08 Milestone 4 Report: Open firing of Type B burners and appliances

2 EXSPERIMENTAL EQUIPMENT AND METHODS

2.1 Description of burners

The burners were previously tested in open firing, as reported in the Milestone 4 report. They are:

- A partially-premixed, naturally-aspirated burner with an AN-style nozzle rated at 35 kW (120 MJ/h)
- A forced draft premixed package burner with integral fan, rated at 35–92 kW with a 106 mm diameter burner tube
- An air heat burner, rated at 147 kW (530 MJ/h)
- A forced-draft high-velocity nozzle-mix burner rated at ~20 kW (72 MJ/h), using an air pressure of 7 kPa

In the current milestone report, preliminary results are presented for the AN-burner only. The confined test results of the remaining burners will be presented in the following milestone report.

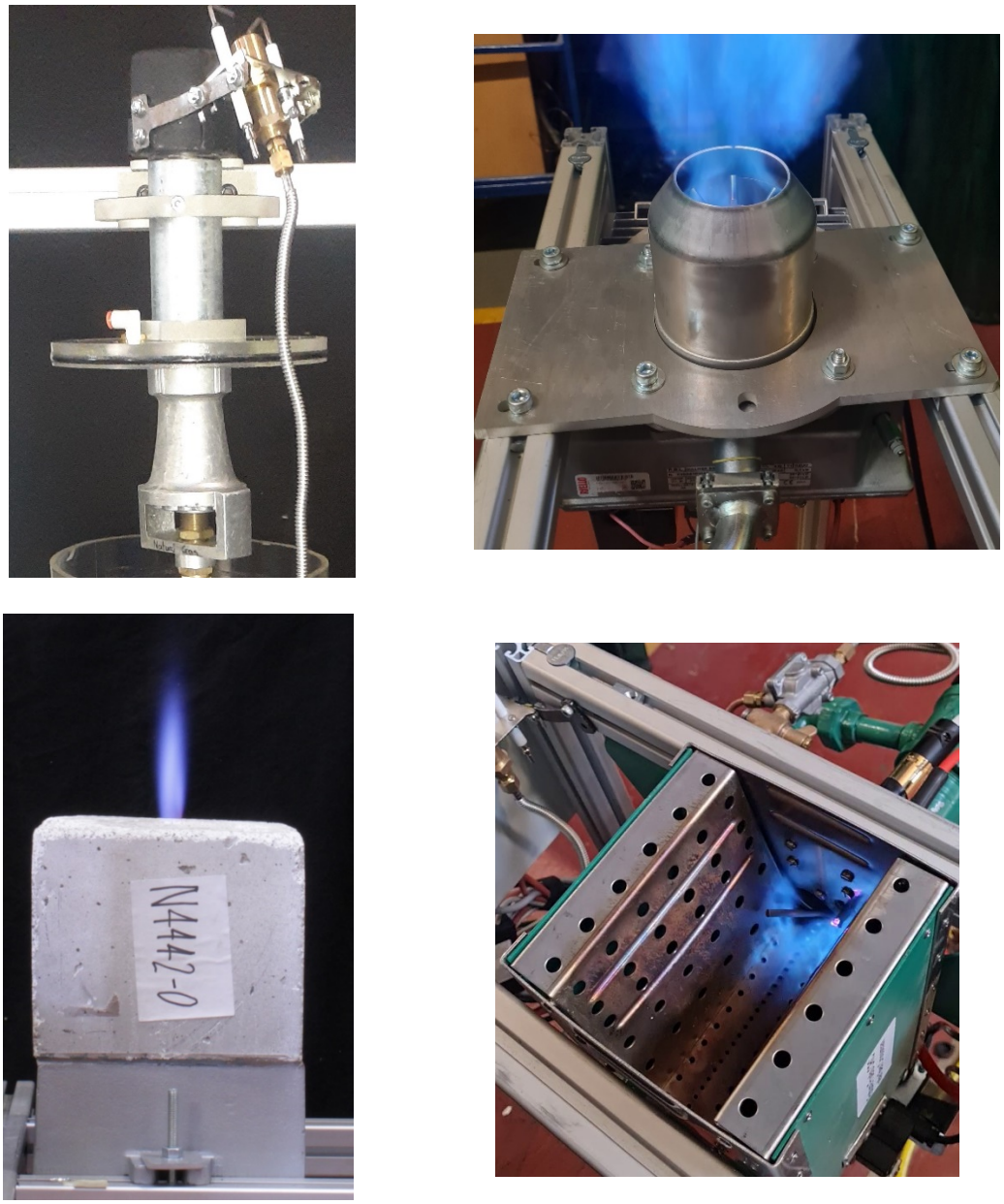

Figure 29: Photographs of the four burners to be involved in confined-firing study (three shown open-firing natural gas). Clockwise from top-left: AN burner, package burner, air-heat burner, and high-velocity nozzle-mix burner.

2.2 AN burner confined firing: preliminary test method

Due to the differences in burner and flame geometries and fuel-air mixing processes, multiple confinements will be used in this phase of the study. In order to verify and refine the experimental technique, preliminary measurements of heat transfer and emissions have been conducted with the AN burner operating on natural gas in a confined configuration. The burner was fired vertically into a 200-mm diameter, 1-m long duct, painted black to prevent any changes or non-uniformity in emissivity as the duct was heated up and discoloured. The burner sits slightly below the inlet to the duct, which ensures that entrainment of secondary air is sufficient to achieve complete combustion. A photograph displaying this configuration is shown in Figure 30.

The external wall temperatures along the length of the duct were monitored and recorded via eight K-Type washer thermocouples, clamped to the outside surface via ring clamps. The thermocouples were evenly spaced in the axial direction, and were also positioned in different circumferential locations to assess the uniformity of the wall temperature. An adjustable butterfly damper was mounted to the outlet of the duct, to control the amount of secondary air entrained and hence excess air level. The temperature and composition of the flue gases (specifically; NO, NO_2, CO, CO_2 and O_2) immediately upstream of the butterfly valve were sampled using a Testo 350 gas analyser. During the preliminary experiments, the firing rate of the burner and the damper setting were adjusted to achieve an O_2 concentration in the flue gas of approximately 3 vol%.

Figure 30: AN burner fired into cylindrical duct.

2.3 Other confinements and test methods

The high velocity burner will be fired into a furnace type of confinement (Figure 31. In practical furnaces, refractory and insulation limits heat loss and preheated air is often used so high temperatures are achieved. In this project, the furnace type experiments will instead use a thin-walled rectangular stainless steel box shaped enclosure with wall temperatures limited to 800°C to 1000°C due to heat loss from uninsulated walls, which provides a representation of heat transfer to the load in a real furnace.

The package burner and air heat burner will be fired into a cylindrical confinement with a length to diameter ratio of 1.5 to represent a common shape in some boilers. Boiler type experiments will reproduce the heat transfer limitation on the flame / gas side (as opposed to the steam side) by using an uninsulated enclosure with thin metal walls to allow heat loss to the surroundings. Wall temperatures will be in the range 100°C to 500°C.

The independent variables are fuel composition, fuel rate and excess air. Independent control of wall temperatures can also be used to generate different experimental conditions. Measured variables include total heat transfer (convective and radiant), heat transfer profiles, flue emissions and flue gas temperatures, providing data to perform mass and energy balances.

Previous work has indicated that blending up to 30 vol% hydrogen into natural gas does not create major changes in heat transfer and NOx. This assumption will be assessed and efforts to improve NOx or heat transfer results, such as adjustment of fuel rate or excess air, will be made.

For 100% hydrogen flames it is expected that radiation heat transfer will be reduced compared to natural gas. For these geometries, adjustments will be made to fuel rate, excess air, and addition of small quantities of methane will be trialled. All these adjustments will be made to explore whether heat transfer and NOx emissions can be like those of baseline natural gas flames. However, it is expected that further changes to burner design, secondary air mixing and recirculation of flue gas will be required.

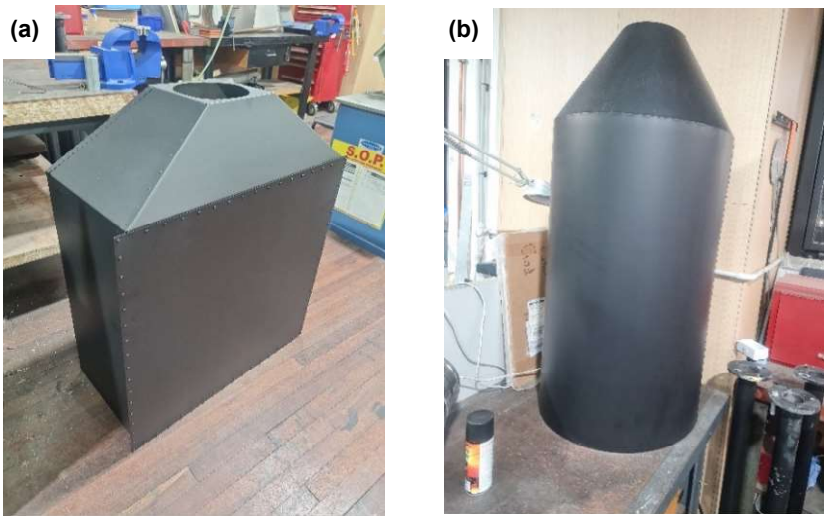

Figure 31: Enclosures to be used for future confined firing tests – enclosure (A) to be used for high-velocity nozzle-mix burner, enclosure (b) to be used for package burner and air-heat burner.

2.4 Heat transfer methodology

The method is described for the Atmospheric Nozzle (AN) naturally aspirated burner but will also apply to the other burners and confinements. The AN burner was fired vertically upwards into a 200mm thin-wall (0.6mm thick) stainless steel duct located in the open air in the laboratory. This arrangement models an indirect heater, which is one that uses a metal heat transfer surface (in this case the duct) between the flame and the fluid to be heated (in this case the surrounding air and room). Heat loss from the duct walls is by radiation and natural convection to the surroundings. Radiation loss is maximised by painting the outside of the duct with high temperature rated matte black paint. The aim of the experiments is to explore the effect of hydrogen blending on heat transfer from flame and combustion gases to the inside of the duct.

The laboratory experiment differs from many commercial applications of indirect fired duct heaters in which a liquid (rather than the surrounding air) is heated. For example, low-pressure water tube heaters use a fire tube inside a water bath. The heated water then transfers heat to another fluid circulating inside another coil. Another variation is constructing a firetube from the inside of a helical tube in which a process fluid to be heated flows. The key similarity in these two variants is that there is a relatively small diameter, long confinement for the flame. Process fluid temperatures can vary from under 100°C for water heaters, to up to 500°C for oil heaters. Firing is normally horizontal but vertical options are available.

The damper located will allow heat transfer and emissions to be compared at equal excess air even though the required stoichiometric air changes as hydrogen is blended into natural gas. NOx and CO emissions and heat flux all show a dependence on excess air, so it is useful to compare results at equal excess air.

A schematic of the various heat transfer processes for the burner fired into the duct is displayed in Figure 32, with energy balance derivations and definition of the key terms included below:

q_{in} is the heat of combustion of the fuel
q_{flue} is the sensible heat leaving in the flue gas
q_1 is heat transfer from flame to wall (radiation and convection)
q_2 is sensible energy gain of the wall material = $m_{ss}C_p\Delta T/\Delta t$
$q_3 = q_{cond}$ is heat conduction through the wall = $-kA(T_{wo} - T_{wi})/\Delta x$
q_4 is heat lost from outside wall by natural convection and radiation.

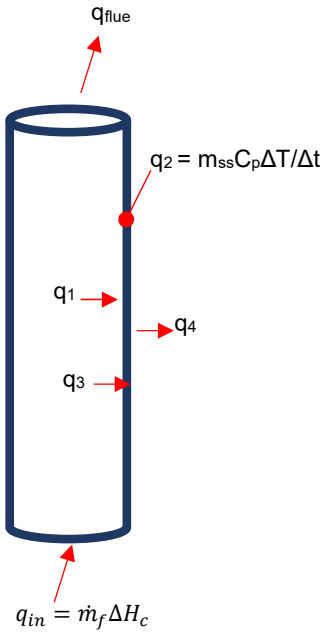

Figure 32: Energy balance on a duct with an internal flame losing heat to surroundings

Where $\dot{m}_f$ is the mass flow of fuel
m_{ss} is the mass of stainless steel.
$\Delta T/\Delta t$ is the change in wall temperature with time,
ΔHc is the heat of combustion of the fuel,
T_{wo} is the duct outside wall temperature,
T_{wi} is the duct inside wall temperature,
Δx is the duct wall thickness.

The overall energy balance is:
Overall balance: $q_{in} = q_2 + q_4 + q_{flue}$
At steady state the wall temperatures are constant so, $q_2 = 0$ and, $q_1 = q3 = q_4$.

The aim of the experiment is to understand the changes in total heat transfer and the heat transfer profile from flame and hot gases to the wall under different conditions, , i.e. q_1, which at steady state is equal to q4. So, the aim can be addressed by estimating q_4 from wall temperature measurements and correlations for natural convection heat loss and calculations of radiant heat loss, using an assumed wall emissivity, from the outside of the duct.

However, a simpler way to calculate the steady state heat transfer ($q_1 = q_2 = q_4$), is to observe the rate of cooling of the wall immediately after shutting off the flame. This method relies on the insight that that the heat loss from the outside of the duct at steady state is a function of wall temperature, and that immediately after turning off the flame the initial temperature of the duct is unchanged so heat is still being lost by natural convection and radiation from the outside of the duct to the room at the same rate as when the flame was on, i.e. q_4 is initially unchanged. If the temperature of the duct then drops linearly with time for a short period, the heat loss will reduce, but the temperature reduction can be used to calculate the initial sensible heat change in the duct, q_2, which is equal to q_4 when the flame was turned off and before the flame was turned off.

The method is to first record emissions during operation at steady state when temperatures are at their highest, then turn off the fuel and close the flue damper so there is no induced air low through the duct and observe the reduction in wall temperatures with time. In the period after shutting off the flame, the wall temperature drops, so that q_2 becomes negative and can be easily calculated knowing the wall mass and specific heat.

The initial rate of q_2 at shutdown is equal to what q4 was during steady state. In the immediate period after the flame is shut off, q_{in}, q_{flue}, q_1, q_3 are all zero and so $-q_2 = q_4$. A plot of wall temperature vs time in the first 10 - 30 seconds after shutdown can be used to calculate and average temperature and hence and overall heat loss, q_4
Or, a heat flux profile can be calculated from individual temperatures and incremental area segments ($\pi D \Delta l$), where D is the duct diameter and Δl is the incremental duct length.

A wide range of heat fluxes can be explored in the experiments by heating the tube to different temperatures before performing the experiment.

In the next report, tests using burners in the other confinements will be reported. Some of these represent kilns, ovens and furnaces with refractory walls operating at much higher temperatures than indirect fired heaters. However, the same method can be used since the important heat flux is from the flame and furnace gases to a process material or objects inside the kiln. Laboratory wall temperatures will not be as high as a refractory lined furnace, but high enough for NOx and heat flux results to show relevant trends.

3 PRELIMINARY RESULTS/ANALYSIS – AN BURNER IN A DUCT

3.1 Wall temperature measurements

To verify the method outlined in Section 2.4, preliminary experiments were performed with the AN-burner operated on natural gas and fired into a one-metre cylindrical duct (painted matte black for uniform emissivity). A restriction was placed at the outlet of the duct to achieve an O_2 concentration in the range of 2–5 vol% in the (wet) undiluted exhaust gases. The wall temperatures were monitored via eight thermocouples uniformly spaced along the length of the duct, and the burner was operated until steady state was reached while the flue gases were sampled. For the case analysed in this section, the burner was operated at a gas rate of 50 SLPM

(equivalent to 32 kW based on the higher heating value), resulting in a concentration of 2.5 vol% O_2 in the exhaust gases.

Significant variation in the wall temperature along the length of the duct was observed, as shown in Figure 33, with the thermocouples located in the upper half of the duct (T5, T6, T7 & T8) reaching similar steady state temperatures in the range of 450–500 °C. Figure 33 also shows the linear reduction in the wall temperature immediately following shut-down of the burner; this can be used to calculate the steady-state heat transfer, which is the focus of the analysis in this section.

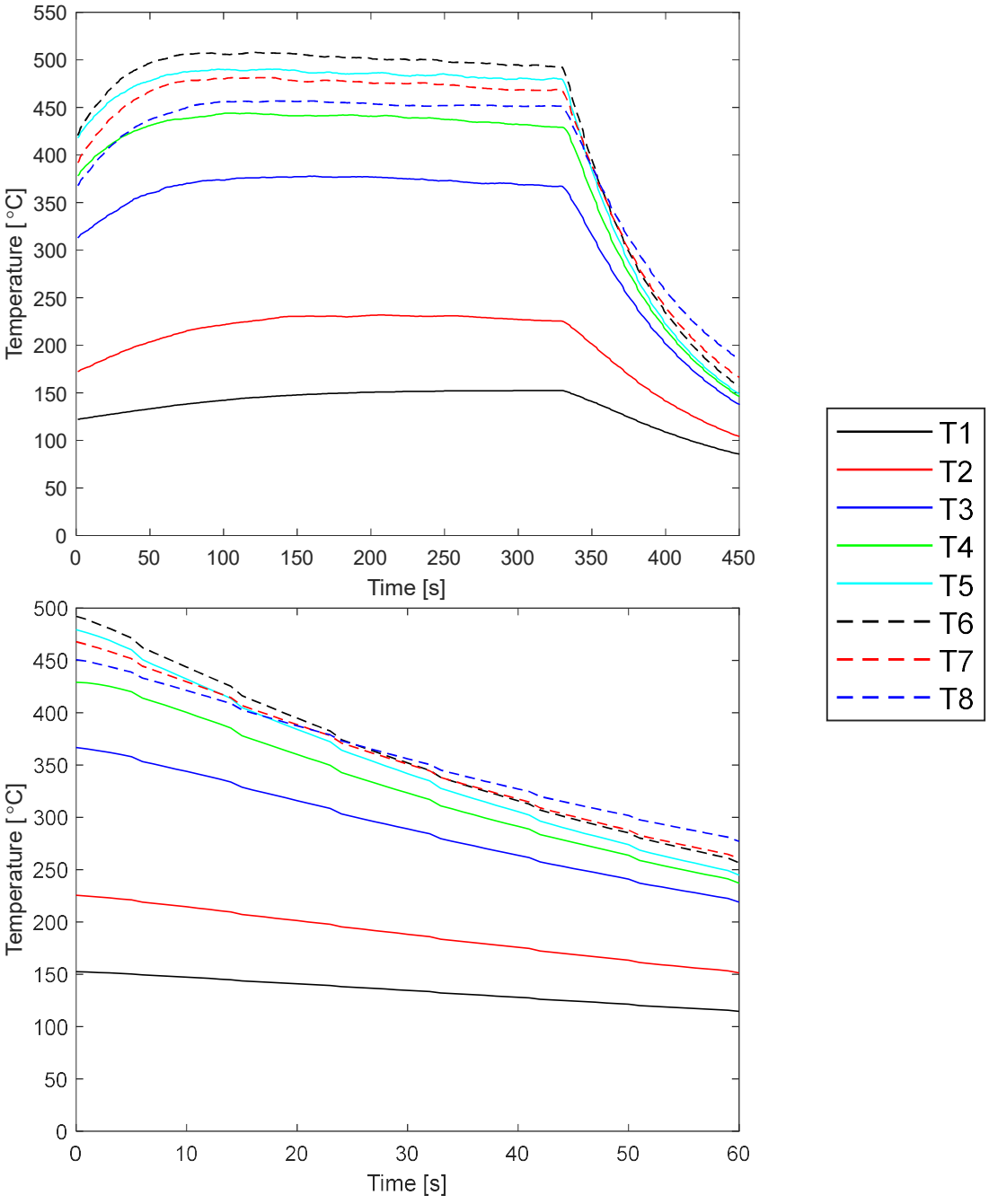

Figure 33: Temperature of thermocouples – top graph showing wall temperature reaching steady state and decay after flame turned off, and bottom graph showing linear temperature reduction in 60 s after shut-down.

3.2 Heat transfer analysis

As described in Section 2.4, the steady state rate of heat transfer from the flame to the walls (i.e. the useful heat transfer rate) can be determined from the rate of temperature decay during the period immediately following shut-down of the burner. Specifically, the linear decay in temperature (see Figure 33) allows the calculation of q_2, defined as the sensible energy gain (or loss, if the value is negative) of the wall material:

$$q_2 = \frac{m_{ss} C_p \Delta T}{\Delta t}$$

To calculate q_2 for the entire duct from the discrete thermocouple measurement locations, multiple methods of analysis can be used to yield the same result. Given that q_2 is a linear function of ΔT and the thermocouples are evenly spaced—and assuming that the density of the duct material and the specific heat capacity are constant along the length of the duct—the simplest method for calculating q_2 in this case is to simply take the mean of the eight temperature measurements. For the case where the thermocouple locations are not evenly distributed, this method can be generalised by fitting a curve to the temperature decay (ΔT) measurements as a function of the axial distance, and then taking the mean of this function rather than the arithmetic mean.

For the case shown in Figure 33, the mean reduction in temperature based on the eight measurement locations—calculated from the initial (i.e. steady-state) temperatures and the temperatures 60 s after turning off the flame—is 162.9 °C. For illustration purposes, the axial plot of ΔT for the eight locations is shown in Figure 34.

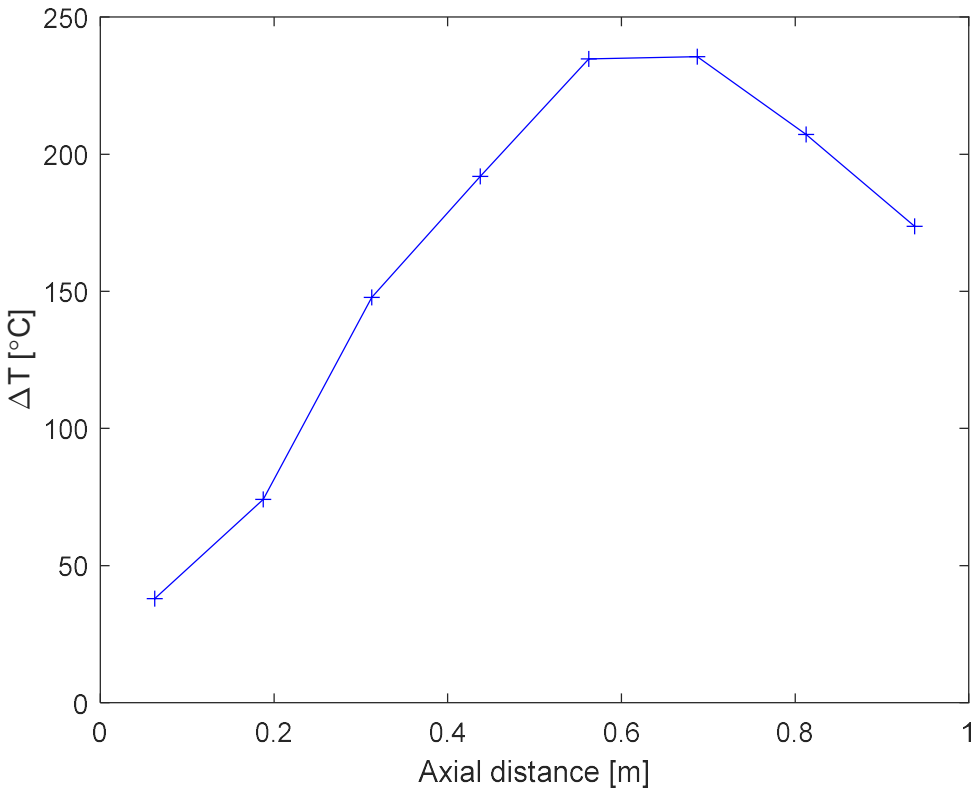

Figure 34: Axial profile of temperature decay 60 s after turning off the flame.

Using the average value for ΔT, along with the mass of the 1-m duct plus the thermocouple clamps which is 3 kg, and estimating the specific heat capacity of stainless steel as 0.5 kJ/kg.K, we get:

$$q_2 = \frac{3 \times 0.5 \times (-162.9)}{60}$$
$$= -4.1 \text{ kW}$$

Note that the value used for ΔT is negative as it is a reduction in temperature, which results in a negative value for q_2, since it is defined as the enthalpy gain (such that a negative value represents a heat loss). From Section 2.4, the rate of heat loss from the walls immediately after shut-down is equal to the heat transfer rate from the walls to the surroundings at steady state, which is equal to the heat transfer rate from the flame to the inside of the duct during steady state—that is:

$$q_1 = q_4 = -q_2$$
$$\therefore q_1 = 4.1 \text{ kw}$$

As mentioned, this result corresponds to a 32-kW heat input flame, so the heat transfer from the flame to the walls of 4.1 kW gives a thermal efficiency (based on the higher heating value of the fuel) of approximately 13%. This is a relatively low efficiency, which is consistent with the fact that the flame was of low radiance and there was minimal direct contact between the flame and the walls of the duct, such that most of the heat from the flame remains in the flue gases. In saying this, further measurement and testing is required in order to fully validate these calculations as a reliable method for comparing heat transfer when changing the fuel blend, which will be expanded upon in future reporting.

The flame shutdown method is still in the preliminary stages of verification and will be further refined using shorter time periods. It will also be compared with calculations of heat transfer rates from an energy balance and heat transfer correlations based on measurements taken at steady state when the flame is on. Ultimately the best method will be selected for the experiments.

4 CONCLUSIONS

Preliminary experiments have been conducted to assess heat transfer from a naturally aspirated burner to an enclosing duct. Analysis of preliminary results shows that more development and refinement of the method is required, and this is ongoing. There are many ways to assess heat transfer and once the most accurate method has been chosen, it will be used to assess four burners in appropriately shaped enclosures to represent ducts, furnaces, and ovens. The effect of hydrogen blending into natural gas on emissions and heat transfer will be assessed.

5 FUTURE WORK

The next phase of the work is to assess the effects of blending hydrogen into natural gas on flame heat transfer (by a combination of radiation and convection) to confining duct walls and NOx emissions. The effects of adding 10, 20 and 30 vol% hydrogen to natural gas in confined configurations will be explored once the heat transfer methodology has been finalised.

If the hydrogen blends produce poorer heat transfer or higher NOx emissions than natural gas, simple methods to reduce the differences such as varying primary and secondary air rates can be explored. Changing to 100% hydrogen involves a larger change in flame conditions, but likewise methods to reduce differences with current natural gas flames will be explored. The results will contribute insights into how hydrogen can be used in ducts, water tube heaters, furnaces and boilers and what changes are required.

Following the confined firing tests with the four burners and three enclosures outlined in the current report, the next phase of the work as outlined in the Milestone 2 Report, Section 2.4, is to test additional burners. Following the most recent discussions with the industry advisers, this set of burners has been modified to the following which all have differences from the burners already available:

- The smallest model Maxon Ovenpak burner which is a common burner for furnaces.
- The smallest model Eclipse Thermjet.
- A small condensing water heater or boiler.
- Further tests on the efficiency of a legacy Raypak Type A water heater with 10-20% hydrogen blends, after checking it is the same design as the common Type B units.

After open-firing tests of these burners, a decision will be made on whether confined firing will be useful also. There are also other components to the project, related to modelling, full-scale testing and at Furnace Engineering and preparation for other full-scale tests, as outlined in Section 2.5 of the project plan in the Milestone 2 Report.

6 IMPLICATIONS AND RECOMMENDATIONS FOR INDUSTRY

This report has focussed on development of a method which once developed further in the next stage of the project will yield useful practical results relating to the applicability of hydrogen in existing Type B furnaces, duct heaters and ovens.

RP1.4-08
Milestone 6: Confined firing of Type B burners and appliances

12 December, 2024

Project number: RP1.4-08

Detailed assessment of Type B appliances with blends of hydrogen and natural gas and 100% hydrogen

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Summary of Report

Confined firing tests have been performed for three different Type B burners in three different enclosures to investigate the effects of the addition of hydrogen (H_2) to natural gas on combustion performance. These enclosures were made of thin stainless-steel sheet which allowed the calculation of heat transfer through the walls using the measured wall temperatures. In addition, comparisons of NOx emissions were performed for various fuel blends. To improve the performance of the burners running with H_2 and natural gas blends, modification has been proposed and tested under confined firing conditions. A summary of the key findings for each burner is shown below.

A naturally aspirated “atmospheric nozzle” (AN) burner was fired upwards into a 200mm diameter vertical duct. With a constant heat input (35 kW), the peak heat flux to the wall shifted closer to the burner head with the addition of H_2 when the burner was operated in a partial-premixed mode. On the other hand, while the burner was operated in a non-premixed mode, H_2 addition has insignificant influence on the heat flux profile. Furthermore, it was found that the heat flux profile of the natural gas reference flame can potentially be matched by adjusting the primary air and excess air setting. In the present study, the amount of entrained primary air was adjusted by changing the fuel injector orifice size, while a flue damper was installed to adjust the excess air ratio. For example, the heat flux profile when using a H_2 fraction of 90 vol% and 10 vol% natural gas, and a fuel injector size of 8mm, matches the heat flux profile of the natural gas refence case with a standard fuel injector size of 4mm. The NOx emissions of the natural gas reference case were approximately 40 mg/MJ and no significant impact was found for the addition of H_2 up to 50 vol%. When the H_2 fraction was increased to 90 vol%, the NOx emissions approximately doubled compared to the reference case. For an H_2 fraction of 100 vol%, the NOx emissions were approximately four times higher than the reference case. No clear impact of the primary air settings was found on the NOx emissions. However, an increase in excess air reduced the NOx emissions slightly for high H_2 fractions ($\geqslant$90 vol%).

A forced air package burner with a specified operating range (on natural gas) of 35–92 kW was tested via firing horizontally into a 600mm diameter cylindrical enclosure. The fan operates at a single speed, with an adjustable inlet damper to control the air flow rate. The burner was tested at the 92 and 60 kW. Air damper settings were adjusted for each firing rate to achieve excess air ratios from 0 to 150%. It was found that for all studied operating conditions, H_2 addition had an insignificant impact on the heat transfer performance in the furnace. An increase in H_2 fraction up to 50 vol% increased the NOx emissions up to 25 mg/MJ from 20mg/MJ for natural gas. A further increase in H_2 fraction up to 100 vol% was found to reduce the NOx emissions. For the heat input of 92 kW, the NOx emissions from 100% H_2 flames were lower than corresponding pure natural gas flames with the same air damper position. A “town gas” head designed for fuels containing relatively higher H_2 fraction was tested. For the flames using the “town gas” head, the effect of H_2 addition on the wall heat flux profile is small. Compared with the corresponding flames using the original burner head, the heat flux along the entire furnace length is consistently lower for the cases using the “town gas” burner head. For the “town gas” head, as the H_2 fraction is increased from 0 to 75 vol%, the NOx emissions increase slightly (<30%). As the H_2 fraction was increased further from 75 vol% to 100 vol%, significant increases in NOx emissions were observed. For example, at one damper position NO emissions from 60 kW flames increased from 36.4 to 57.4 mg/MJ.

A high-velocity nozzle-mix burner was tested by firing horizontally into either of two box-shaped enclosures at a rate of ~15 kW. The burner design includes a mullite refractory chamber insulated by refractory brick in which fuel and air are rapidly mixed and partially combusted before exiting through a small hole as a narrow flame into the box-shaped thin metal enclosure. The air flowrate was controlled by a mass flow controller to achieve various excess air ratios from 10 to 50%. It was found that the influence of H_2 addition on the wall heat flux profile is small (<15% variation for all the studied flames). Fuel composition had an insignificant effect on NOx emissions for H_2 fractions lower than 60 vol%, with the results ranging from 36 to 41 mg/MJ. A slight increase in NOx emissions was observed as the H_2 fraction was increased from 60 to 100 vol%. For example, for an excess air ratio of 30%, the NOx emissions index increased from 40.3 to 47.6 mg/MJ.

The results of this investigation show that H_2 addition has varying effects on the wall heat flux depending on the type of burner, enclosure and operational conditions. For the studied conditions, the addition of H_2 shifted the heat flux peak location towards the burner head while the AN burner was operated in partial premixed mode. This impact can be mitigated by modifying the primary air ratio. The influence of H_2 addition on the wall heat flux profiles was less significant for both the high velocity burner and the package burner. The only significant influence of hydrogen addition on NOx emissions was for the AN burner and H_2 fractions >90 vol% for the

studied conditions. An additional noteworthy point is that for all the burners and enclosures when operated a H_2 fraction of up to 30 vol%, the confined firing produced similar results to natural gas firing (both heat flux and NOx emissions).

1 INTRODUCTION

1.1 Background and motivation

This Milestone 6 report deals almost exclusively with details of confined firing tests on three different styles of Type B burners. This covers most of section 2.3 of the project plan that was detailed in the Milestone 2 report. The confined firing tests provide new information on the effects of hydrogen blending and of using 100% hydrogen on heat transfer and NOx emissions.

Model development and full-scale tests of 30vol% hydrogen in a pyrolysis burn-off furnace and open firing of a new high velocity “Thermjet” burner will be reported in Milestone 7 report rather than the current report. The project plan also included open firing of some other burners. However, these tests were not included in consultation with industry advisers for various reasons. For example, the air-heat linear burner had already been tested and results reported in the Milestone 5 report. A two-stage package burner was not included in the open-firing tests because one was tested in a full-scale test at an industrial trial to be reported in Milestone 7 report. It was also found that the “Testing Hydrogen admixture for Gas Applications” consortium (THyGA), had already sufficiently investigated the impact of blending hydrogen on the performance of condensing boilers [1]. During their tests, no malfunction associated with the variation of condensates was found. Therefore, no adjustment is required for the existing appliances. However, the variation of condensate flowrate differed slightly between appliances (for a H_2 fraction of 40 vol%). Therefore, when designing a new appliance for high H_2 fractions, condenser design should be investigated and modified in order to achieve maximum performance. No increase was found in acidity of the condensate with an increase in hydrogen fraction. Similar findings were reported in other literature [2, 3].

1.2 Scope and aims

This project investigates the effect of hydrogen addition on the safety, performance and emissions of Type B combustion appliances operating at atmospheric (or near-atmospheric) pressures. The current report aims to address the following questions, by using experimental testing of burners under “confined” conditions:

- How does blending hydrogen with natural gas or biomethane affect heat transfer performance and emissions from enclosed flames for a range of industrial burners?
- Are there any simple and cost-effective modifications that can be made to the Type B appliances to mitigate any negative impacts of hydrogen addition on heat transfer and emissions characteristics?
- Without significantly impacting heat transfer and emissions performance, what types of burners can be run with either no change, minor adjustments, or with a simple retrofit?

2 EXPERIMENTAL EQUIPMENT AND METHODS

2.1 Burners and appliances

The burners tested are new and include different types of partially and fully premixed and nozzle-mix designs, representing the main types of burners used in Type B appliances commonly found in hydrogen injection trial areas. The following burners/appliances were tested:

A partially-premixed, naturally-aspirated burner with an AN-style nozzle rated at 35 kW (120 MJ/h) was used to fire into a thin-wall enclosure.

- A forced draft nozzle-mix burner package burner with integral fan, rated at 35–92 kW with a 106 mm diameter burner tube was used to fire into a cylindrical enclosure.
- A forced-draft high-velocity nozzle-mix burner rated at ~20 kW (72 MJ/h), using an air pressure of 7 kPa, was fired into two rectangular box enclosures of different sizes.

2.2 General testing methodology

Confined firing tests investigated the heat transfer performance and NOx emissions of the three commercial burners. The burners were fired into thin-wall enclosures (furnaces) which act as heat sinks. The temperature distributions on the wall were measured to calculate the heat flux. In addition, flue gas NOx emissions were measured for all the flames. The influence of H_2 addition on the heat transfer and NOx emissions were investigated for various operating conditions including a few simple modifications. The variables included excess air ratio, heat input (package burner only) and the primary air to fuel ratio (only for AN burner).

The confined firing apparatus for each burner was manufactured and installed according to the supplier's instruction. Type B submissions were prepared for both package burner and high velocity burner. The details of these submissions are presented in Appendix 1.

2.3 Description of appliances and adjustments/modifications

2.3.1 Atmospheric nozzle (AN) inspirator burner in duct

The AN burner was fired upwards into a thin-wall (0.6 mm thick) vertical, 200mm diameter, 2m long stainless steel cylindrical enclosure, Figure 35, which acts as a heat sink for the combustion process. The flow rates of the fuels were precisely controlled by mass flow controllers thus conserving the heat input (35kW, based on lower heating value, LHV) at various H_2 fractions. The fuel stream from the fuel injector (4 mm standard orifice diameter) entrains and mixes with primary air before exiting the burner, thus producing partially-premixed flames. Non-premixed (NP) flames can be obtained by closing the primary air valve. Secondary air enters the enclosure from the open bottom of the 200 mm duct by natural draft due to entrainment and buoyancy effects. The fuel is combusted within the enclosure, and the flue gas exits through an adjustable damper installed at the top of the enclosure. The damper facilitates adjustment of the secondary air flow rate, and hence % excess air and the concentration of O_2 in the flue gas. A sample collection probe with a thermocouple installed was used to measure the flue gas temperature and to sample flue gas for the Testo 350 gas analyzer which measures O_2 and NOx concentration on a dry basis.

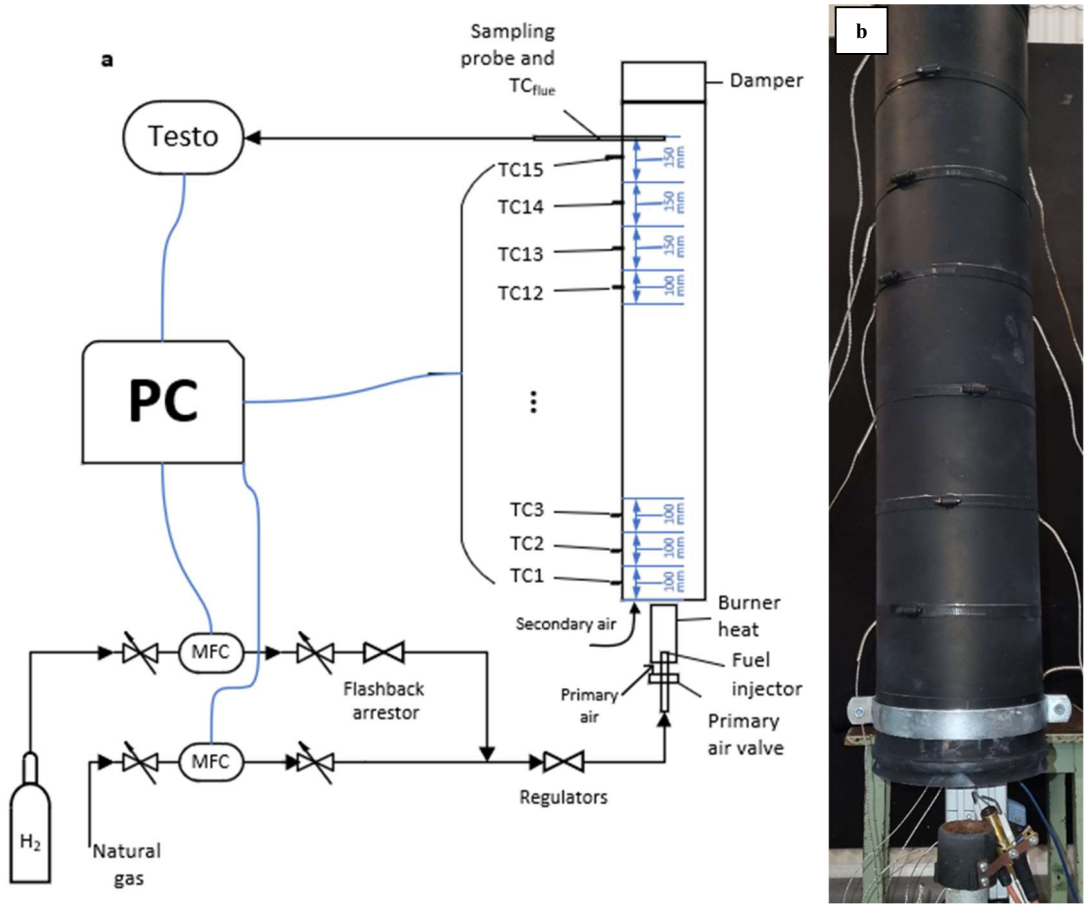

Figure 35 Schematic diagram (a) and photo (b) of the confined firing apparatus with an AN-burner.

2.3.2 Package burner in cylindrical enclosure

A package burner was fired horizontally into a 600mm diameter, 1m long cylindrical enclosure (Figure 36). The heat input was maintained at either 60 or 92 kW (LHV) for various H_2 fractions by precisely controlling the flow rates of the fuels using mass flow controllers. Air was supplied by a fan operated at single speed while an adjustable inlet damper was used to control the air flow rate. The sampled flue gas was rapidly diluted with ambient air in a purpose built a probe. The dilution factor was controlled by adjusting a valve on the ambient air inlet. In addition, a hot air gun was used to ensure that the sample temperature was higher than 100 °C to avoid water condensation. An infrared camera was used to measure the temperature distribution on the furnace wall.

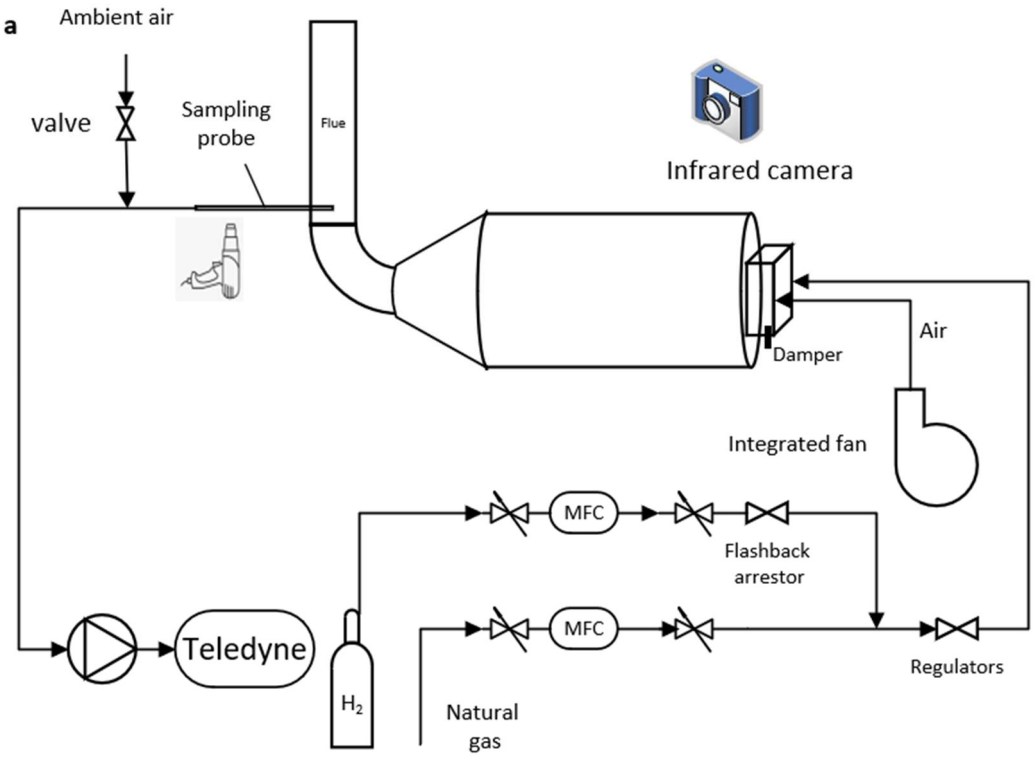

Figure 36 Schematic diagram (a) and photo (b) of the confined firing apparatus with a package burner.

2.3.3 High-velocity nozzle-mix burner in rectangular enclosures

A high velocity burner was fired horizontally into either of two box enclosures (Figure 37). The box photographed is the larger of the two, being 800mm high, 675 mm in length to an internal baffle and 400 mm deep. The other box (not shown) was designed to have a much smaller volume to reduce heat losses and increase temperatures. It was 450mm high, 500 mm in length to an internal baffle, and 300 mm deep. The heat input was maintained at 15kW (LHV) for various H_2 fractions by precisely controlling the flow rates of the fuels using mass flow controllers. The required flow rate of air was calculated according to the H_2 fraction and excess air ratio and was controlled by a mass flow controller. The fuels were partially burned within a mullite combustion chamber insulated by a refractory block which forms part of the burner, and the resulting flame exits through a small

opening with high velocity. A baffle board was located in the furnace close to the outlet of the box enclosure to enhance the flue gas recirculation in the box enclosure. Two gaps between the baffle board and the box enclosure were located at the top and bottom of the baffle board, respectively, to allow the flue gas to flow through. The method of flue gas sample measurement was the same as that used for the package burner apparatus. An infrared camera was also used to measure the temperature distribution on the furnace wall.

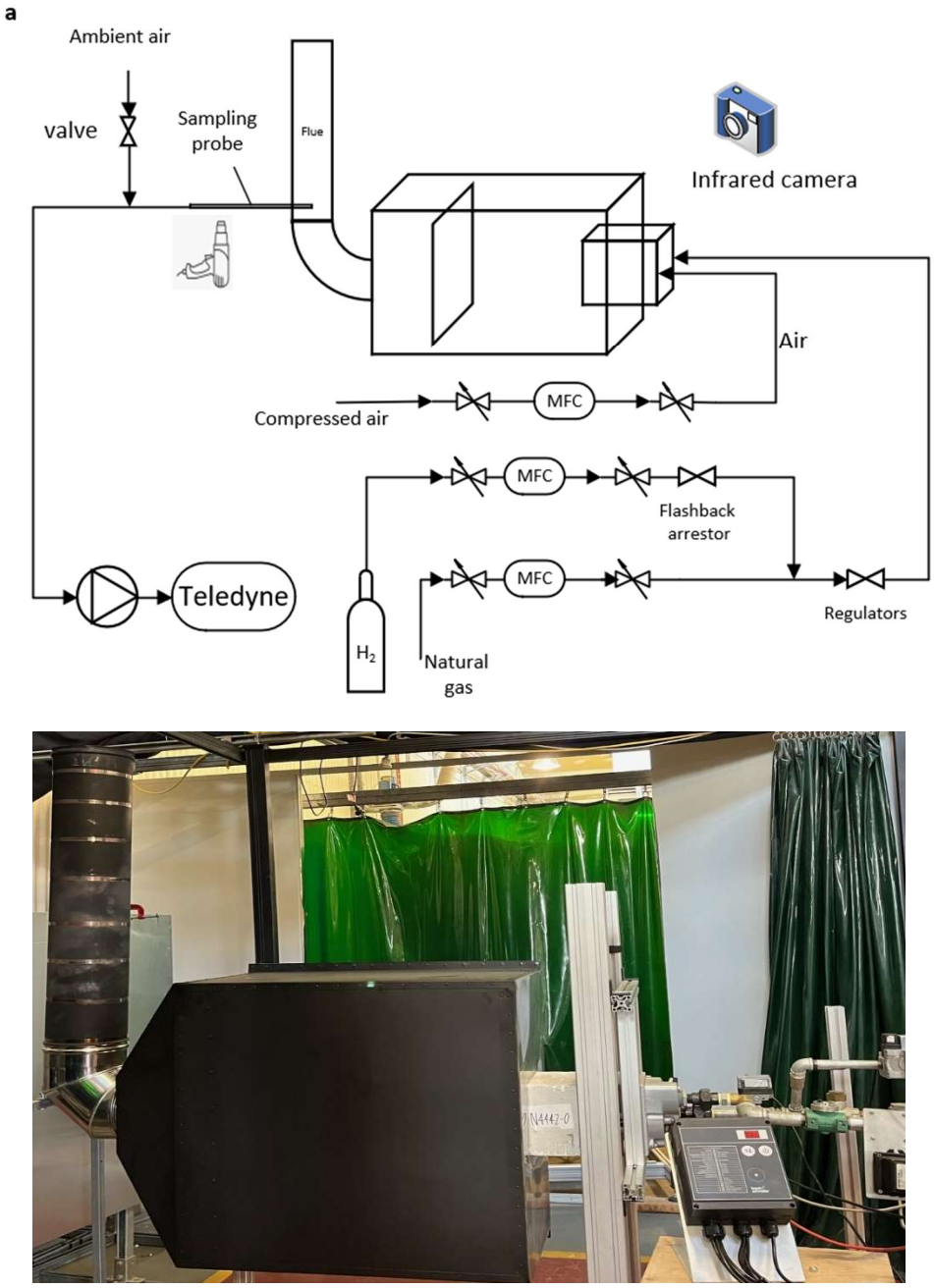

Figure 37 Schematic diagram (a) and photo (b) of the confined firing apparatus with a high-velocity nozzle-mix burner.

2.4 Analysis methods

2.4.1 Wall heat flux calculation

The heat flux transferred from the combustion process to the wall can be calculated from an energy balance with the heat lost to the room. For the heat flux analysis, the furnace walls were divided into sections and an energy

balance was then performed on each section. For the AN burner, one thermocouple was mounted onto the centre of the outside wall of each of 15 sections (as shown in Figure 38). An infrared camera was used to measure the wall temperature distribution for the package burner and high velocity burner enclosures. For the high velocity burner flames, a two-dimensional temperature matrix on one of the box walls was used for the energy balance (as illustrated in Figure 38). For the package burner, since the wall of the enclosure is not flat (cylindrical shape), only an axial temperature distribution along the furnace was exported from the thermal image to investigate the wall heat flux (as illustrated in Figure 38). Since the aim of this study is to investigate the impact of H_2 addition on the wall heat flux, the attachments of thermocouples, the positions of the enclosures and the settings and positions of the infrared camera were all kept constant for all the flames.

According to energy conservation, the heat transferred to each inside wall section is equal to the sum of heat loss from the outside wall to the surrounding environment and the sensible heat stored in the wall material (Equation **Error! Reference source not found.**).

$$m_{ij}C_v\frac{dT_{i,j}}{dt} + q_{wall,loss,[i,j]} \times A_{ij} = q_{flame/gas\ to\ wall,[\,i,j]} \quad (1)$$

Where:
$T_{i,j}$ is the temperature of wall section [i,j], (K),
$A_{i,j}$ is the area of wall section [i,j] (m^2),
$m_{i,j}$ is the mass of wall section [i,j] (kg),
C_v is the thermal capacity of the wall material (kJ/kg/K).

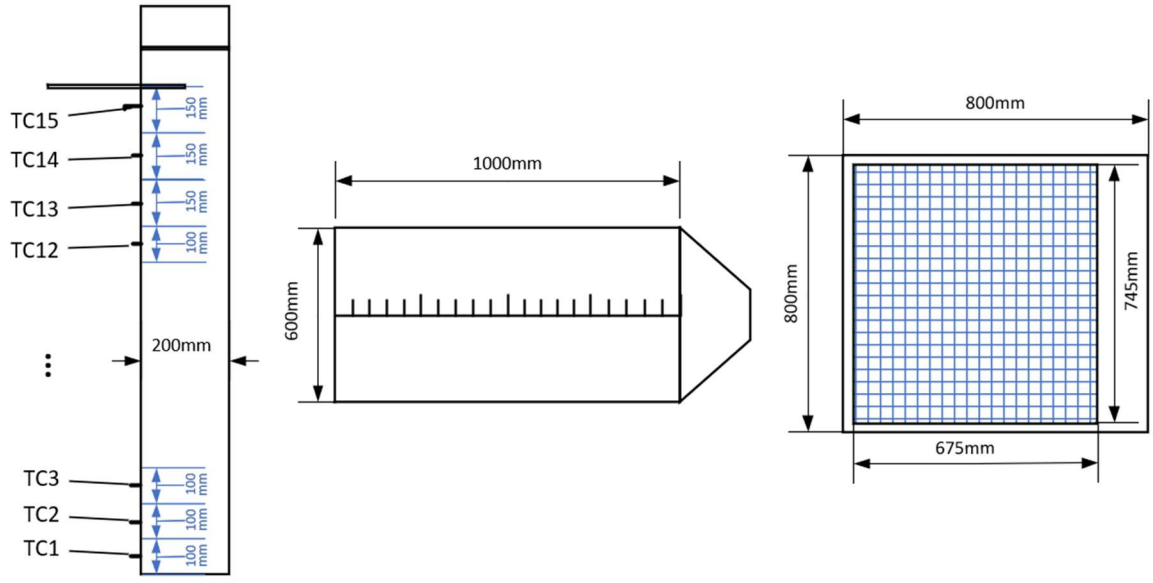

Figure 38 Diagram of sections of wall temperature measurement for AN burner (left), package burner (centre) and high velocity burner in the large box (right).

The first term on the left of Equation (1)**Error! Reference source not found.** is the sensible heat stored in the wall material, which is zero when the wall reaches a steady state (thermal equilibrium). Note that Equation **Error! Reference source not found.** could also include thermal conduction between wall sections, but it has been ignored since the quantity is negligible because the wall section temperature differences are small, and conduction through the wall occurs at a much higher rates since the wall is thin (~0.6 mm). Therefore, at steady state, the heat transferred from the combustion process to each wall section equals to the heat loss from the wall section to the surrounding environment by convection and radiation which can calculated according to Equation (2) using the measured temperature of the wall section [4-6].

$$q_{wall,loss,[i,j]} = h_{i,j}\left(T_{i,j} - T_0\right) + \varepsilon\sigma\left(T_{i,j}^4 - T_0^4\right) \quad (2)$$

Where:
$h_{i,j}$ is the natural convective heat transfer coefficient for wall section [i,j] calculated using the vertical mean temperature (kW/m^2/K),
ε is the emissivity of the wall surface, which is assumed to be 0.95 for the black matte painted surfaces used.

The overall energy balance is described in Equation (3), namely, that the heat input (heating value of the fuel) is equal to the energy output. The energy output (Equation (4)**Error! Reference source not found.**) is the sum of heat transferred from flame/gas to the furnace wall, heat loss in flue gas, and the other heat losses (i.e., radiation losses from the open top and bottom of the duct).

$$Q_{in} = Q_{out} \tag{3}$$

$$Q_{out} = \sum_{i=1}^{15} q_{flame\ /gas\ to\ wall,\ i} \times A_i + Q_{flue} + Q_{o,loss} \tag{4}$$

$$Q_{flue} = \sum_{s=1} m_s\, h_s \tag{5}$$

Where:
Q_{in} is the heat input rate to the combustion process (kW, based on LHV of the fuel),
$q_{flame/gas\ to\ wall,i}$ is the heat flux from flame/gas to the furnace wall section *i*, (kW/m^2),
Q_{flue} is the sensible heat in the flue gas (kW),
$Q_{o,loss}$ is the other heat losses from the system not measured in the present study (kW),
m_s is the mass flow rate of species *s* in the flue gas (kg/s, calculated using O_2 concentration in the flue gas and the fuel flow rate and composition),
h_s is the enthalpy of species *s* in the flue gas (kJ/kg, calculated using the temperature of the flue gas).

2.4.2 Emissions analysis

2.4.2.1 Testo 350 data analysis

The NOx concentration in the flue gas was measured using a Testo 350 gas analyzer. The measurement accuracy for NOx (NO and NO_2) is ±5 ppm for readings from 0 to 99.9 ppm and ±5% of reading for readings from 100 to 1999.9 ppm. The water vapor was assumed to be saturated for all the gas samples sent into the Testo 350. Raw NOx data is converted to an emission index (EI, mg/MJ) defined by solving the equations below:

$$EI = \frac{m_e}{HHV} \tag{6}$$

$$m_e = \frac{F_0}{1 - z_0} \times NO_x \times 10^6 \times M \tag{7}$$

$$F_0 = F + F_1 \times ER - W - F_1 \times (1 + ER) \times RH_0 \times z_0 \tag{8}$$

$$ER = \frac{x \times \frac{F_0}{1 - z_0}}{F_1 \times x_0} \tag{9}$$

Where:
m_e is the generated NO_x per mole fuel (mg/mol),
HHV is the higher heat value of the fuel (MJ/mol),
F_0 is the dry flue gas factor which is the volumetric ratio of dry flue gas to the fuel gas,
NO_x is the sum of readings of NO and NO_2 on the gas analyser for sample gas (ppm),
M is the molecular weight of NO_x, equal to 46 mg/mol, (standard practice is to use molecular weight of NO_2)
F is the flue factor which is the volumetric ratio of wet flue gas to the fuel gas at stoichiometric condition,
F_1 is the air factor which is the volumetric ratio of stoichiometric air demand to the fuel gas,
ER is the excess air ratio,
W is the volumetric ratio of produced water to the fuel gas,
RH_0 is the relative humidity of ambient air,
z_0 is the water vapour saturated pressure at ambient temperature,

x is the reading of oxygen concentration on gas analyser for sample gas,
x_0 is the reading of oxygen concentration on gas analyser for ambient air,

2.4.2.2 Teledyne data analysis

As shown in Figure 36 and Figure 37, a special probe was developed to dilute the flue gas sample before sending into Teledyne gas analyser. In order to avoid any condensation in the gas line, the flue gas should be sufficiently diluted with a dilution factor greater than a minimum value (DF_{min}). Therefore, during sample gas measurement, the dilution valve was adjusted to make sure the reading of oxygen concentration (x) was higher than the minimum oxygen concentration (x_{min}) calculated by the equations below:

$$DF_{min} = \frac{x_0}{x_0 - x_{min}} \tag{10}$$

$$DF_{min} = \frac{\frac{W}{z_0} - (F - F_1) \times RH_0}{(1 - RH_0) \times F} \tag{11}$$

The emission index of NOx (EI) was also calculated using Equation (6). Different from Testo gas analyser, the parameter n is calculated using the equations below:

$$EI = \frac{m_e}{HHV} \tag{12}$$

$$m_e = ((F \times DF) \times NO_x - ((DF - 1) \times F + F_1)) \times 10^6 \times M \tag{13}$$

$$DF = \frac{x_0}{x_0 - x} \tag{14}$$

DF is the dilution factor, the volumetric ratio of wet diluted gas to the wet flue gas produced at stoichiometric condition,
$NO_{x,0}$ is the ambient NOx concentration, ppm.

2.4.3 Operating conditions

2.4.3.1 AN burner

The modifications proposed by Gee at al. [7] to adjust the primary air entrainment (use fuel injectors with different orifice diameters of 8 and 12 mm) were also investigated in the present study to understand their impacts on heat transfer and NOx emissions. Table 9 shows the H_2-blending limits and the corresponding ranges of equivalence ratio for various primary air entrainment settings operated up to its H_2-blending limit at 35 kW in open firing.

Table 9 H_2-blending limits and ranges of equivalence ratio obtained by Gee at al.[7].

	Partial-premixed			NP
Fuel injector orifice diameter (mm)	4	8	12	
H_2 blending limit (vol%)	50	90	100	100
Range of equivalence ratio	1.27-1.54	2.88-3.75	5.76-7.87	0

Investigations were conducted up to the previously identified H_2-blending limit for each primary air entrainment setting. For each primary air entrainment setting, the flue damper was adjusted to produce variations in the O_2 concentration in the flue gas (and therefore excess air) so that effects on heat transfer and NOx emissions could be measured.

The controlled variables that are considered in the present study are (1) the primary air entrainment settings (primary air valve opening and fuel injector orifice size), (2) the H_2 fraction in the fuel, and (3) the excess O_2 concentration in the flue gas (controlled by adjusting the damper opening). Therefore, to simplify the presentation of results in the following section, a case naming convention of the form '**A**x-**H**y-**O**z' and '**NP**-**H**y-**O**z' was defined to indicate the combination of the three variables for partial-premixed and non-premixed cases, respectively. In

these codes, *x* is the orifice size of the fuel injector (mm), *y* is the H_2 fraction (vol%), and *z* is the O_2 concentration in the flue gas (vol%). For example, **A**04-**H**030-**O**5 means that the fuel injector diameter is 4 mm (standard size), the H_2 fraction is 30 vol%, and the O_2 concentration in the flue gas is 5 vol%. Likewise, **NP**-**H**100-**O**6.2 means that the burner is operated in non-premixed mode, the H_2 fraction is 100 vol%, and the O_2 concentration in the flue gas is 6.2 vol%.

2.4.3.2 Package burner

Investigations were conducted at two heat inputs (60 and 92 kW) with various H_2 fractions (0, 25 vol%, 50 vol%, 75 vol% and 100 vol%) and various inlet air damper positions (positions 1 to 6). The air flow rate and hence excess air was increased by increasing the damper opening indicated by larger air damper positions. In addition, to reduce burner head overheating and noise, a towns gas (TG) head (designed for fuels containing relatively high H_2 fraction (> 40 vol%) was also tested and compared with the original standard burner head. The TG head has a different geometry which increases the axial momentum of the fuel to reduce mixing between fuel and oxidant in the near burner zone.

2.4.3.3 High velocity burner

Investigations were conducted for the high velocity burner with various H_2 fractions (0, 30 vol%, 60 vol% and 100 vol%) and various excess air ratios (ER=10%, 30% and 50%). The fuel input was 15kW for all of the flames with various H_2 fractions.

3 RESULTS

3.1 Heat transfer characteristics with no modification and normal adjustments

3.1.1 Atmospheric nozzle (AN) inspirator burner

3.1.1.1 Validation of energy conservation

In the present study, the 'wall heat flux' calculated from the sum of convection and radiation loss to the room, by using Equation (2), was used to evaluate the heat transfer from the combustion process. To validate the calculation, the energy balance of the combustion process was investigated. Figure 39 presents the calculated heat input and heat losses (wall heat loss and sensible heat lost in the flue) for the duct furnace with an AN-burner and a constant heat input of 35 kW (LHV), as a function of fuel H_2 fraction. The unmodified burner with a standard 4 mm fuel injector was studied for H_2 fractions up to 50 vol% which is the upper limit for stable flames [7]. To study the energy balance for scenarios with higher H_2 fraction (90 and 100 vol% were selected), the burner was operated at NP condition (primary valve closed).

Figure 39 shows that for all H_2 fractions from 0 to 100 vol%, the fuel heat input of 35kW is slightly higher than calculated losses. The maximum difference is about 16% and likely results from other energy losses not included in the calculation (i.e., radiation losses from the open ends of the duct), uncertainties in assumptions (i.e. emissivity of the wall surface and natural convective coefficient), and measurement errors. The difference between the heat input and the sum of flue loss and total wall loss is relatively consistent which indicates that the applied measurements and calculations can provide reasonable accuracy to investigate the impact of H_2 addition into natural gas on the heat transfer characteristics.

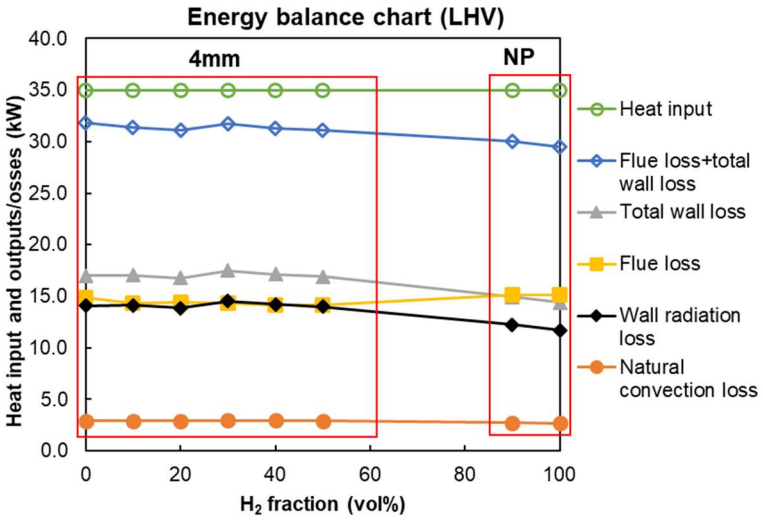

Figure 39 Components of the energy balance for confined flames with varying H_2 fractions and a constant heat input of 35 kW.

3.1.1.2 The influence of H_2 addition on wall heat flux profile

The axial wall heat flux profile is an important characteristic that may be impacted by the addition of H_2. For the AN burner with a constant heat input of 35 kW and a 4mm fuel injector, the influence of H_2 addition on the actual and normalized (against the natural gas case) wall heat fluxes at heights above burner (HABs) from 50 to 1570 mm are presented in Figure 40a and Figure 40b, respectively. An increase in the H_2 fraction from 0 to 50 vol% in 10vol% steps resulted in the peak heat flux location moving sequentially closer to the burner, from 650 to 350 mm above the burner, without significantly changing the value of peak heat flux. This reduction in peak heat flux location was the result of shorter flames with increasing H_2 fraction as observed clearly in open-firing results [7]. The magnitude of the peak heat flux decreased by about 2.8% over the same range of fuel compositions.

Plotting the heat fluxes normalised against the natural gas flames in Figure 40b illustrates the magnitude of the change in position of the heat flux curve. It shows that an increase in the H_2 fraction resulted in an increase in wall heat flux for HAB less than 500 mm. At HAB of 150 mm, the wall heat flux for a 50 vol% H_2 natural gas blend

flame was more than three times as high as that for a natural gas flame at the same location. With the replacement of natural gas by H_2, a reduction in the soot volume fraction leads to a reduction in measured total radiative heat flux of these AN burner flames in open firing, but radiative losses were in the region of only 10 to 20% of the heat input [7]. In addition, it was previously found that the open firing flame length for the AN burner with a 4mm fuel injector is shorter than 700 mm, and reduces further as the hydrogen content is increased from 0 - 50 vol% [7]. This reduction in flame length indicates that more heat is released closer to the burner as hydrogen is added. Adding hydrogen may also increase flame temperature [8, 9], which can lead to increases in both radiative and convective heat fluxes. The earlier combustion and potentially higher temperatures probably cause higher convective heat transfer from the flame.

On the other hand, the enthalpy/temperature of the flue gas above the flame (HAB>700 mm) could be lower for the scenario with a higher H_2 fraction due to greater heat losses and at lower positions.. Therefore, for an HAB greater than 700 mm, the wall heat flux is slightly decreased with an increase in H_2 fraction, as shown in Figure 40b.

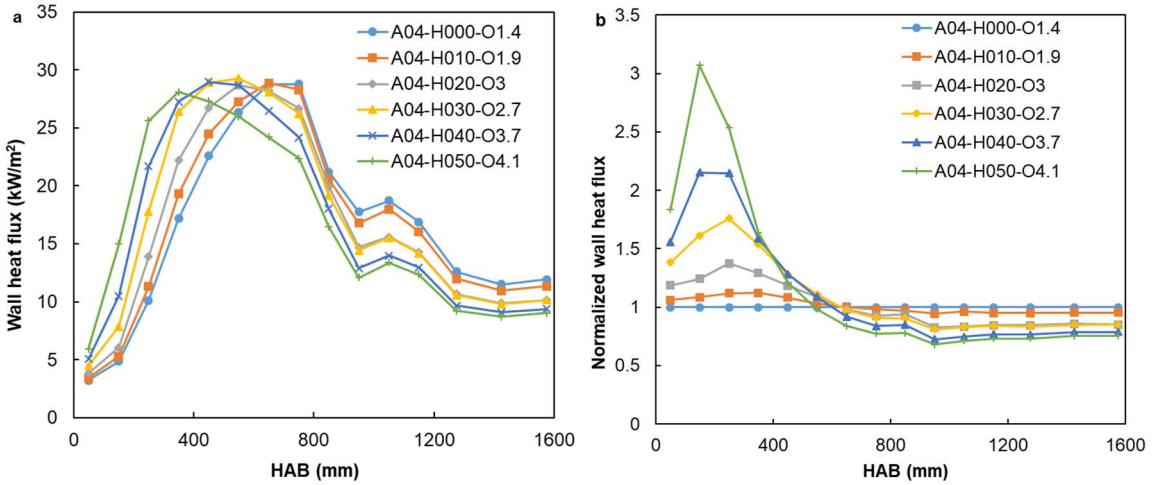

Figure 40 The (a) actual and (b) normalized (against the natural gas case) axial wall heat flux profiles for various H_2 fractions. Constant heat input (35 kW) and a 4 mm fuel injector.

3.1.1.3 The influence of H_2 addition on the heat flux profile for non-premixed cases

Wall heat flux profiles from four non-premixed flames are presented in Figure 41 alongside two partially premixed flames using standard 4 mm fuel injector for comparison (dashed lines). The peak heat flux from the NP flames was lower and shifted away from the burner head relative to the partially premixed flames.

A direct comparison can be made for an H_2 fraction of 50 vol%, the location and value of the peak heat flux for the NP flame were 750 mm above the burner and 23.6 kW/m^2, respectively, compared to 350 mm above the burner and 28.1 kW/m^2, for the partially premixed case using 4 mm fuel injector. These observations are due to longer NP flames compared with partial premixed ones [7]. On the other hand, H_2 blending fraction has an insignificant effect on the heat flux profile for NP cases. The radiative heat flux of the open fired versions of these flames were reduced by the H_2 blending [7]. However, here in the confined firing cases, an increase in convective heating may have rebalanced that effect to result in an insignificant total effect of H_2 addition on the heat flux profile and peak heat flux.

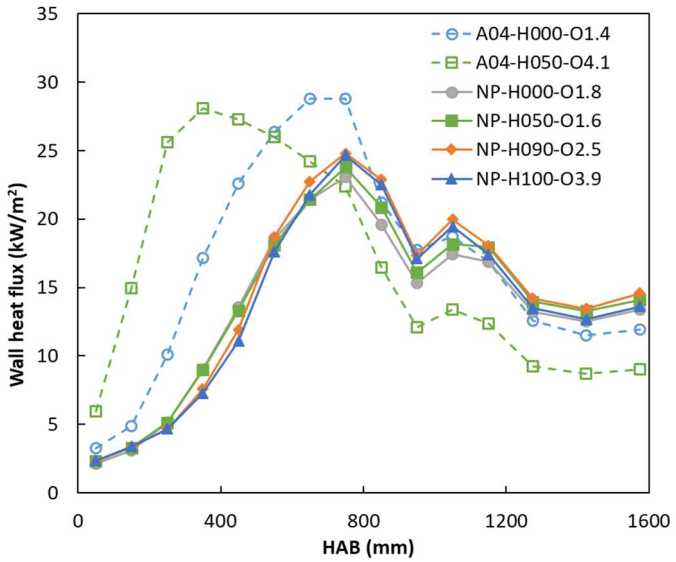

Figure 41 Wall heat flux profiles for non-premixed (NP) and corresponding partial premixed cases for various H_2 fractions.

3.1.1.4 The influence of excess air on the wall heat flux profiles of partially premixed flames

Excess air is an important operating parameter that can impact the heat transfer characteristics of the combustion process. Figure 42 presents the influence of excess air on the wall heat flux profiles for AN burner flames with various H_2 fractions. Two natural gas flames from the unmodified AN burner operated with two damper positions to produce 1.4 and 5 vol% O_2 concentrations in the flue gas are shown with the blue symbols in Figure 42a. For both damper openings, flames with various H_2 fractions were investigated. The yellow symbols in Figure 42a show that for 30vol% hydrogen the same damper settings produced 2.7%O_2 and 6.1% O_2 in the flue gas (higher than natural gas because of lower stoichiometric requirement) and earlier heat release than for natural gas. The green symbols show that these trends of higher flue O_2% and earlier heat release continued for 50 vol% hydrogen. This reinforces the strong effect of fuel composition on heat flux observed earlier. However, these results also enable an investigation of the significance of excess air on heat flux.

Figure 42a shows that for each fuel composition pair, the higher excess case produces insignificant changes in the location of the peak heat flux. Locations were unchanged for the H_2 fractions of 0 and 30 vol% but shifted 100 mm away from the burner head for an H_2 fraction of 50 vol%. On the other hand, Figure 42a shows that for each fuel composition pair, the higher excess case produces a significantly lower peak heat flux and generally lower heat flux than the low excess air case. This is illustrated again in Figure 42b which shows the heat fluxes for the higher excess cases normalized against the lower excess air flames. It illustrates that the high excess air cases have a lower heat flux over the whole range of locations for all fuel compositions.

The reduction in heat flux with excess air is expected due to a reduction in temperature, which evidently has a greater effect than the increase in velocity might have on convective heat transfer. In a practical process, if the desire is to achieve similar wall heat flux to natural gas cases when using a hydrogen blend, these results show that increasing excess air does not correct the earlier heat release caused by adding hydrogen. Instead, a different strategy would be necessary.

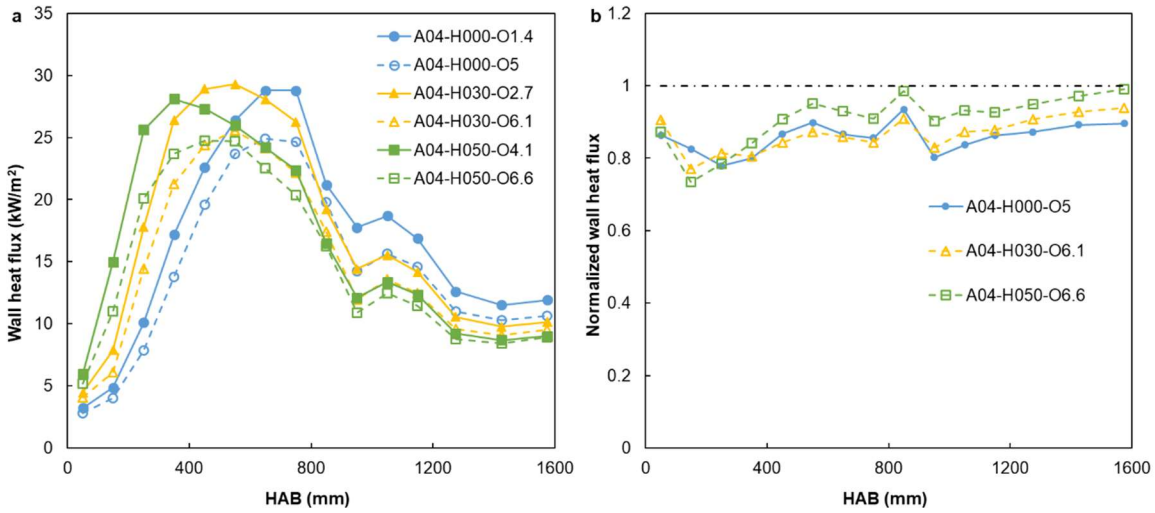

Figure 42 The influence of excess air on the (a) actual and (b) normalized wall heat flux profiles for an AN-burner with various H_2 fractions.

3.1.2 Package burner

3.1.2.1 Averaged wall heat flux

The air flow rate used in the package burner flames was not measured directly but can be calculated from flue oxygen concentration and fuel rate. To run the experiments the air rate was varied by adjusting the opening of the air inlet damper. Higher damper position numbers correspond to larger damper openings and higher excess air, although the actual values of excess air were only calculated after the experiment was performed. Figure 43 shows the increases in excess air at equal damper positions as more hydrogen is blended into natural gas. For example, at 60 kW the burner could be operated at damper position 1 on natural gas with 11% excess air, but this increased to 34% for 100% hydrogen at the same heat input, due to the difference in stoichiometric requirement, although the damper position and air flowrate did not change. Likewise, at damper position 4 the excess air increased from 84% for natural gas to 122% for hydrogen. A similar pattern was observed for 92kW flames, although due to the higher fuel rate, the damper positions were in a different range and even at position 6 the maximum excess air was 30%.

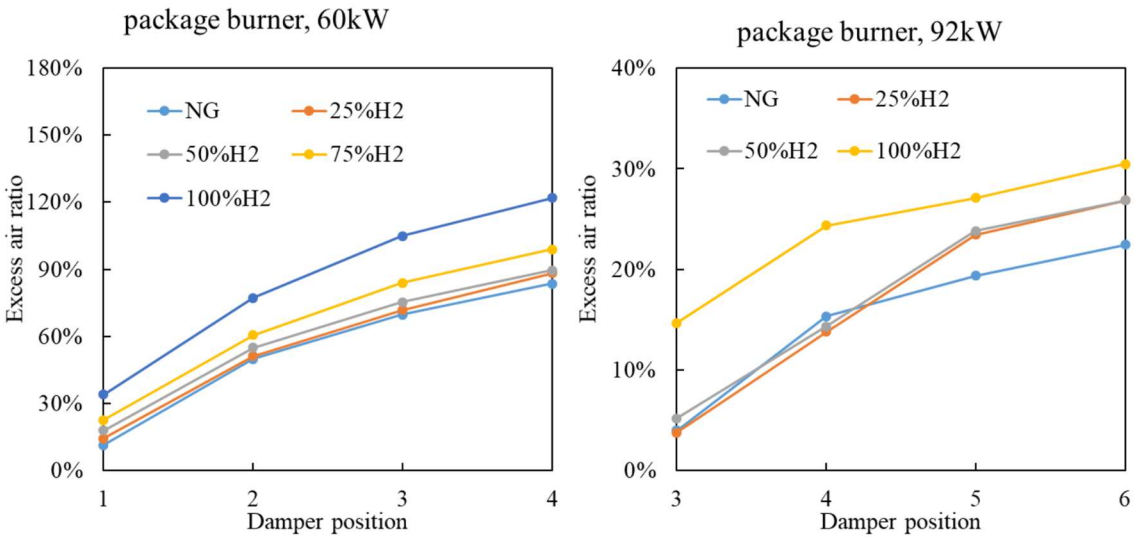

Figure 43 The effect of package burner air inlet damper position on excess air for various fuel compositions

Figure 44 shows the effects of air inlet damper position and H_2 fraction on the averaged wall heat flux for the package burner operated at 60 and 92 kW. There is a general trend of larger damper openings (higher excess air) resulting in slightly lower averaged heat fluxes on the furnace wall. These reductions are probably caused by reduced flame temperatures due to dilution by increased excess air. For 60kW flames there also appears to be a reduction in heat flux for high H_2 fractions compared to natural gas at large damper positions, probably due to the higher excess air, as shown in Figure 43. Further analysis of the effect of excess air will be presented in the next report, where the heat flux data will be replotted as a function of excess air. Once the effect of stoichiometry is removed it is likely that fuel composition has less effect on the averaged wall heat flux.

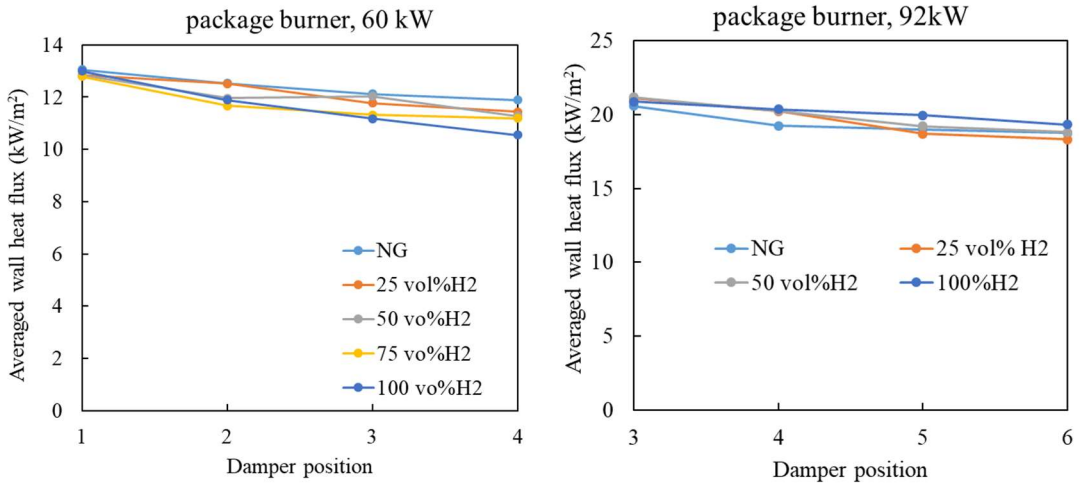

Figure 44 The influences of air damper position and H_2 fraction on the averaged heat flux on the furnace wall for the package burner.

3.1.2.2 Wall heat flux profile

Figure 45 shows the effects of damper position and H_2 fraction on the wall heat flux profile along the furnace length for a package burner operated at 60 and 92 kW. For the 60 kW flames the peak heat flux and the overall heat flux were reduced by increasing the damper opening from position 1 to position 4, probably due to decreased flame temperature from the increase in excess air. The effect of fuel composition is minimal at damper position 1 (with excess air varying from 11 to 34% depending on composition), but an increase in hydrogen content in the fuel slightly reduces in heat transfer at damper position 4 (84 to 122% excess air). However, the high excess air ratios at position 4 are of less relevance than those at position 1. Hence for most practical situations (damper position 1) there is little effect on heat flux over the whole range of compositions up to 100% hydrogen. At 92 kW, damper position 6 is the most industrially relevant (excess air of 20 – 30 %), and there are only minor differences in heat flux over the fuel composition range.

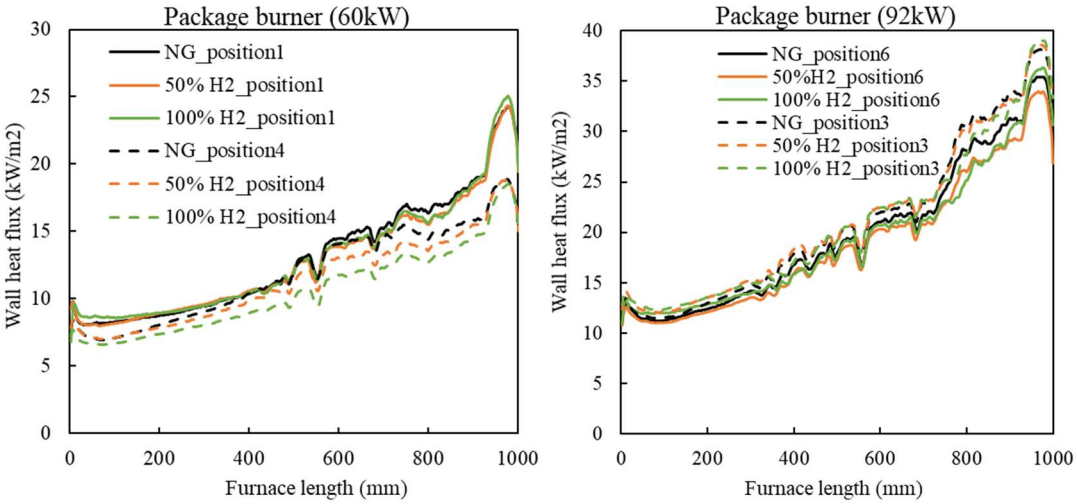

Figure 45 The influence of H_2 addition on heat flux profile for various damper positions

3.1.2.3 Burner head temperature

In the open firing tests conducted previously, a potential issue of burner head overheating was identified at high heat input (92 kW). Therefore, for the confined firing tests, a thermocouple was installed inside the burner head with the thermocouple junction in contact with the metal of the burner head. Figure 46 shows a significant increase in the temperature of the burner head as H_2 fraction of the fuel is increased, especially as H_2 fraction increased beyond 50 vol%. For 100 vol% H_2, the temperature of the burner head was approximately 700 °C, and would likely be higher in furnaces with refractory lining, showing that this burner is not suitable for 100% hydrogen. However, the manufacturer is aware of the issue and has the "town gas" head for use with fuels with approximately 50 vol% hydrogen content.

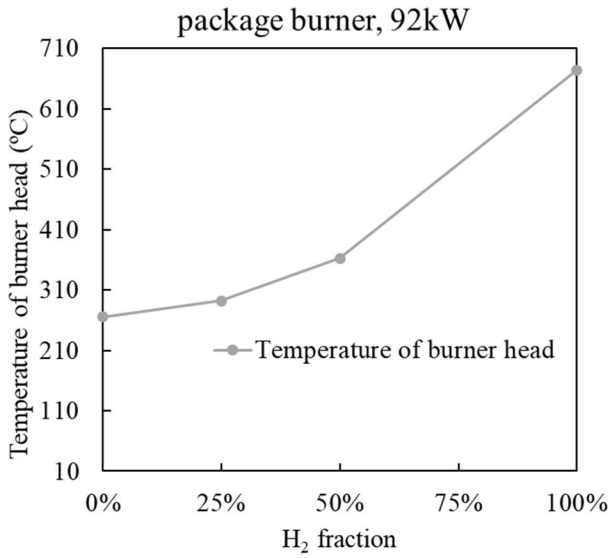

Figure 46 The influence of H_2 addition on the temperature of burner head

3.1.3 High-velocity nozzle-mix burner

3.1.3.1 Averaged wall heat flux

For the high velocity burner operated in the larger box shaped enclosure, Figure 47 shows that the influences of both the excess air ratio and fuel H_2 fraction on the averaged wall heat flux are insignificant. As the excess air increases, the flame temperature decreases which has the potential to decrease the wall heat flux. However, the increase in flow rate and velocity could increase convection heat transfer in the furnace and cancel this effect to a certain extent. Similarly, for a constant excess air ratio, the flame temperature increases as hydrogen is added, which competes with a reduction in flue gas flow rate and velocity. These competing changes may explain the overall insignificant effect on heat flux as hydrogen is added.

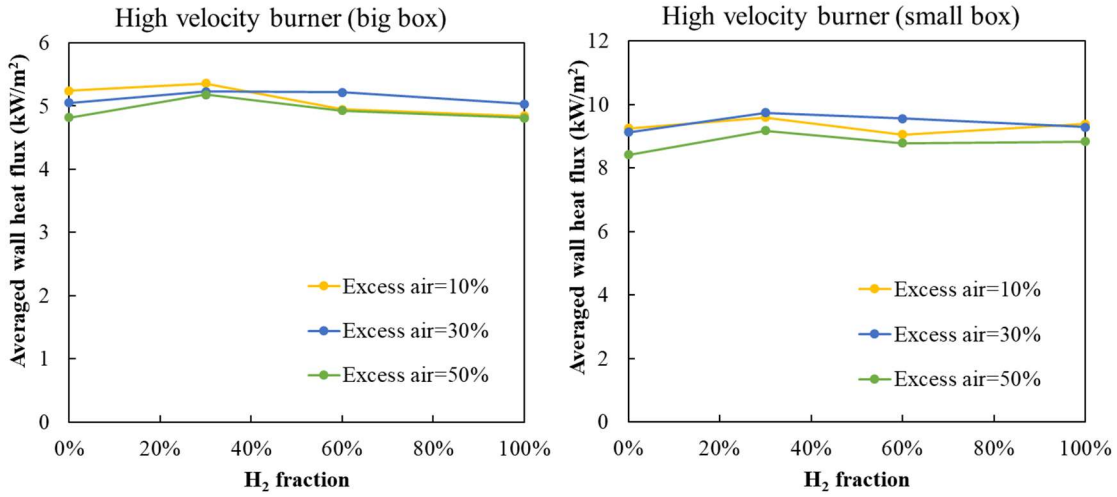

Figure 47 The influences of excess air and H_2 fraction on the averaged heat flux on the furnace wall for the high-velocity nozzle-mix burner with a big box enclosure (left) and a small box enclosure (right).

3.1.3.2 Wall heat flux profile

Figure 48 presents the influence of excess air and fuel composition on the wall heat flux profile for the high velocity burner in the both furnaces. Solid lines are for excess air of 10%, and dashed lines are for excess air of 50%. For natural gas, an increase in excess air reduced the wall heat flux, which could be a result of lower temperatures caused by the increase in excess air. There is a similar but less significant reduction in heat flux with increased excess air for the 30vol% hydrogen blend and 100% hydrogen. The smaller reduction may bea result of competing effects of decreased temperature and increased velocity with increased excess air. It can also be observed that natural gas and 30 vol% hydrogen blend operated at 10% excess air have slightly higher peak heat flux than the 100% hydrogen flames in the large box, but they are all similar in the small box.

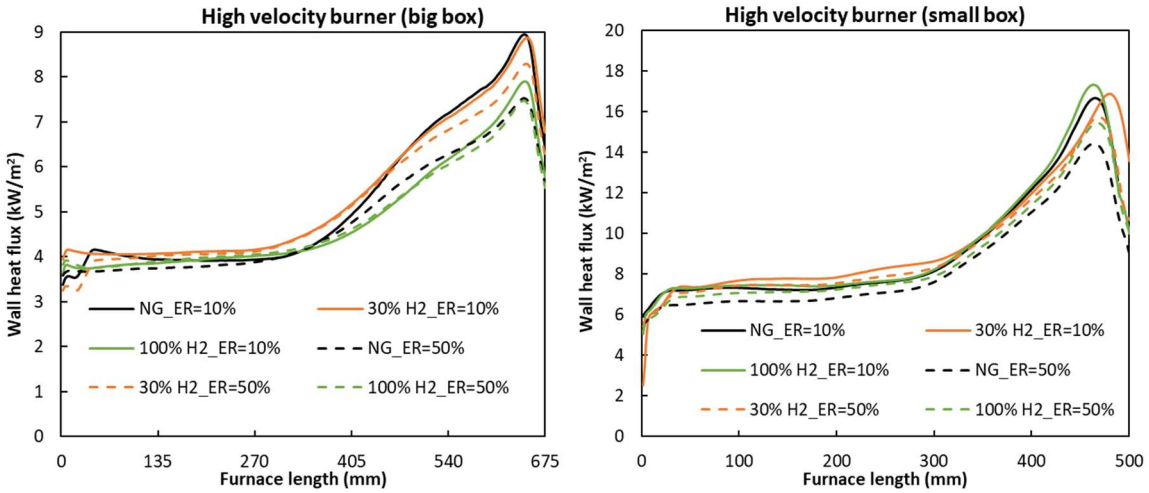

Figure 48 The influences of excess air and H_2 fraction on the wall heat flux profiles for a high velocity burner with a large box enclosure (left) and a small box enclosure (right).

3.2 NOx emissions with no modification and normal adjustments

3.2.1 Atmospheric nozzle (AN) inspirator burner

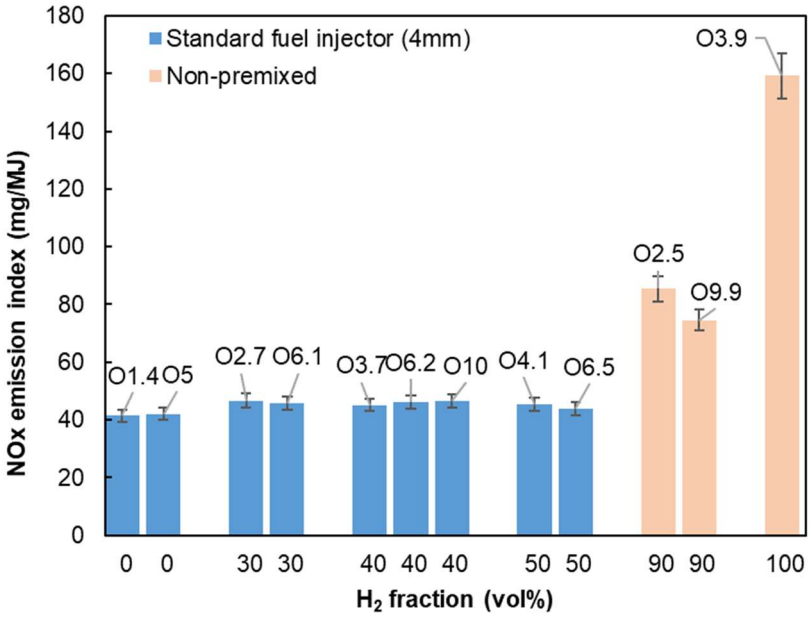

Figure 49 The influence of H_2 vol% on NOx emissions for an AN burner operated with various flue O_2 concentrations (%, as prefixed with 'O'). Fuel injector size is 4 mm..

Figure 49 shows the influence of H_2 addition to natural gas on NOx emissions for the AN burner operated with various primary air settings (closed primary valve or fuel injectors with various orifice sizes) and secondary air damper openings to produce varying flue oxygen concentrations (e.g. O1.4 = flue oxygen of 1.4%). In the fuel composition range from 0 to 50 vol% hydrogen addition of H_2 and variation in excess air both have negligible influence on NOx emissions (~40 mg/MJ). These NOx emissions obtained from the confined firing tests were consistently higher than those achieved from the previous open firing tests [7], which was expected due to higher flame temperatures in the confined condition. To avoid flash back at higher H_2 fractions, the primary air valve was closed to produce non-premixed flames. For a H_2 fraction of 90 vol%, the NOx emission index is about two times the reference natural gas case. For the 100% hydrogen flame, the NOx emission index almost doubled again.

3.2.2 Package burner

The effect of excess air and fuel composition on NOx emissions from the package burner were investigated at 60 and 92 kW. Figure 50 shows that there was a trend of lower NOx emissions with increasing excess air when the burner was fired at 60 kW probably due to decreasing flame temperature. For 92 kW the maximum excess air that could be achieved was less than 40%, so this effect was only observed as slight Nox reduction at excess air levels greater than 30%. Figure 50 also shows that for both fuel inputs, NOx emissions increased as the H_2 fraction increased up to 50 vol% and then decreased from 50 to 100 vol% H_2. For a heat input of 92 kW, NOx emissions from 100% H_2 flames are lower than those of the natural gas flames.

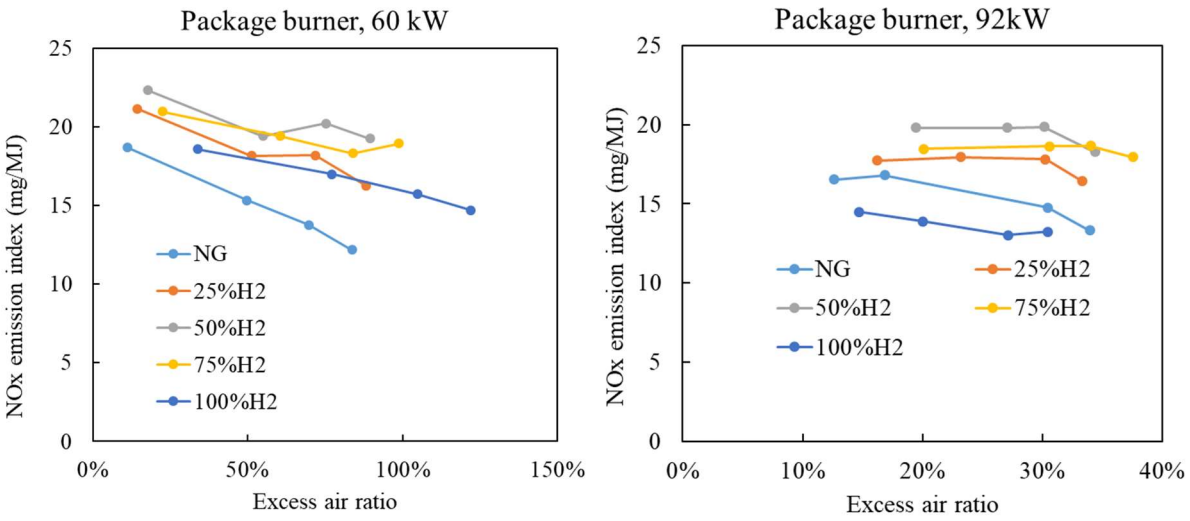

Figure 50 The influence of H_2 fraction and excess air ratio on NOx emissions for a package burner operated at 60 kW(left) and 92 kW(right).

3.2.3 High-velocity nozzle-mix burner

Figure 51 shows the effect of excess air ratio and H_2 fraction on NOx emissions for a high velocity burner in the two box enclosures. The influence of excess air is not very clear, with relatively small differences between cases, but some points can be noted. For both boxes fired with natural gas and 30vol% hydrogen, the NOx emissions were lower for 10% excess air than 30 and 50% excess air. For 100% hydrogen, NOx emissions are generally slightly higher than for natural gas and 30 vol% hydrogen blend. In most cases, 30% excess air produces slightly higher NOx than the other excess air levels.

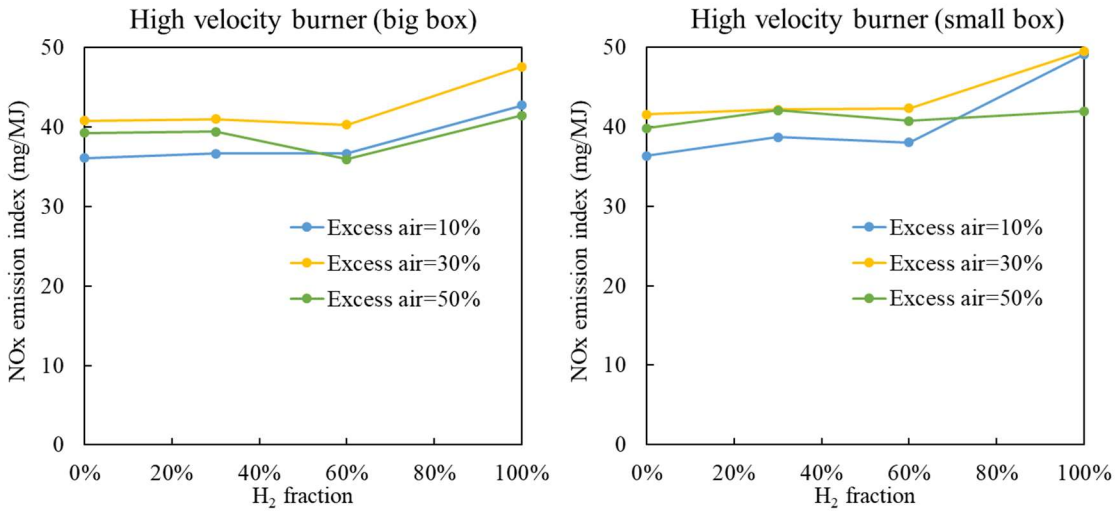

Figure 51 The influences of H_2 fraction and excess air on NOx emissions for the high velocity burner with a large box enclosure (left) and a small box enclosure (right).

3.2.4 Summary of operation with no changes and normal adjustments

3.2.4.1 Heat transfer characteristics

For the AN burner operated in partially premixed mode, the addition of H_2 into natural gas shifted the location of peak wall heat flux towards the burner head due to the shorter length and higher temperature of the flame. On the other hand, the effect of H_2 addition on the heat flux profile is insignificant for the AN burner operated under

non-premixed mode. An increase in excess air leads to a decrease in peak heat flux but has an insignificant effect on the axial location of the peak and the shape the wall heat flux profile. On the other hand, for both package burner and high velocity burner, the effects of H_2 addition on the heat transfer characteristics are minor.

3.2.4.2 NOx emissions

The NOx emissions of the AN and high velocity burners operated with natural gas were similar (~40 mg/MJ). Insignificant changes in NOx emissions were observed for both burners as H_2 was added up to 60 vol%. However, for 100% hydrogen a significant increase in NOx emissions (to 160 mg/MJ) was observed for the AN burner, and a 15% increase was observed for the high velocity burner for most levels of excess air in both box enclosures

The NOx emissions from the package burner operated on natural gas were lower (<20 mg/MJ) than the AN and high velocity burners. An increase in NOx emissions of up to 5 mg/MJ was found as H_2 was added up to 50 vol%. However, NOx emissions dropped slightly for 60 kW, 100% H_2 flames and to levels below that of natural gas for 92 kW, 100% H_2 flames.

3.3 Heat transfer characteristics with burner modifications

3.3.1 Atmospheric nozzle (AN) inspirator burner

As shown previously (Figure 41), for non-premixed AN burner flames adjusting H_2 fraction from 0-100 vol% had little effect wall heat flux profile. However, all the NP flames had much lower and later peaks in heat flux than the reference natural gas case (A04-H000-O1.4). So, using the non-premixed approach to modify the AN burner is not effective for matching the heat flux profile of natural gas flames. An alternative is to use partially premixed flames from larger injectors to reduce the quantity of primary air entrained (as shown in Table 1). Two fuel injectors with orifice sizes of 8 and 12 mm were tested with flames using an H_2 fraction of 90 vol%. The heat flux profiles of the reference natural gas case and the flames with 8 and 12 mm fuel injectors are shown in Figure 52. For an H_2 fraction of 90 vol%, the heat flux profile of the flame from an 8 mm fuel injector matched the heat flux profile of the reference case. Open firing tests have shown that it has a similar flame length to reference case [7] On the other hand, heat flux matching was not achieved for the 12 mm fuel injector with any H_2 fractions across the entire range from 0 to 100 vol%. These tests were all done with the primary air valve fully open, so it is feasible that further investigation could find that selecting the appropriate fuel injector diameter (smaller than 12mm) and primary air valve opening could be used to adjust the heat flux profile for any fuel blend.

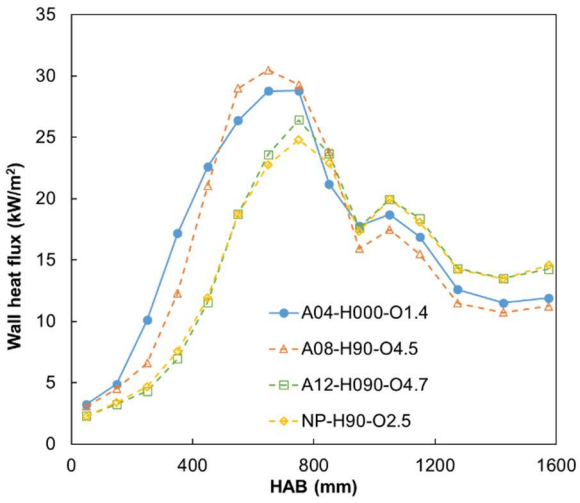

Figure 52 Wall heat flux profiles of the reference case (natural gas) and 90 vol% H_2 flames with modifications.

3.3.2 Package burner

A TG head was installed to replace the original standard burner head, and a significant reduction of the noise has been observed, although not quantified. In addition, previous opening fire tests suggested that overheating of the burner head was less significant for the TG head. In this report, confined firing tests have been conducted to investigate the impact of this burner head replacement on the furnace wall heat flux.

3.3.2.1 Averaged wall heat flux

Figure 53 shows the effect of damper position and H_2 fraction on the averaged wall heat flux for a package burner operated at 60 and 92 kW with a TG head. The figure needs to be read along with Figure 43 which shows the variation in excess air with damper position. The average heat flux results show a similar reducing trend with increasing excess air as for the normal head (Figure 44), but for the TG head the magnitude of the heat flux is lower in all cases. In addition, variation in fuel H_2 fraction has a less significant effect on the averaged wall heat flux for the TG head, with 100% hydrogen being similar to NG.

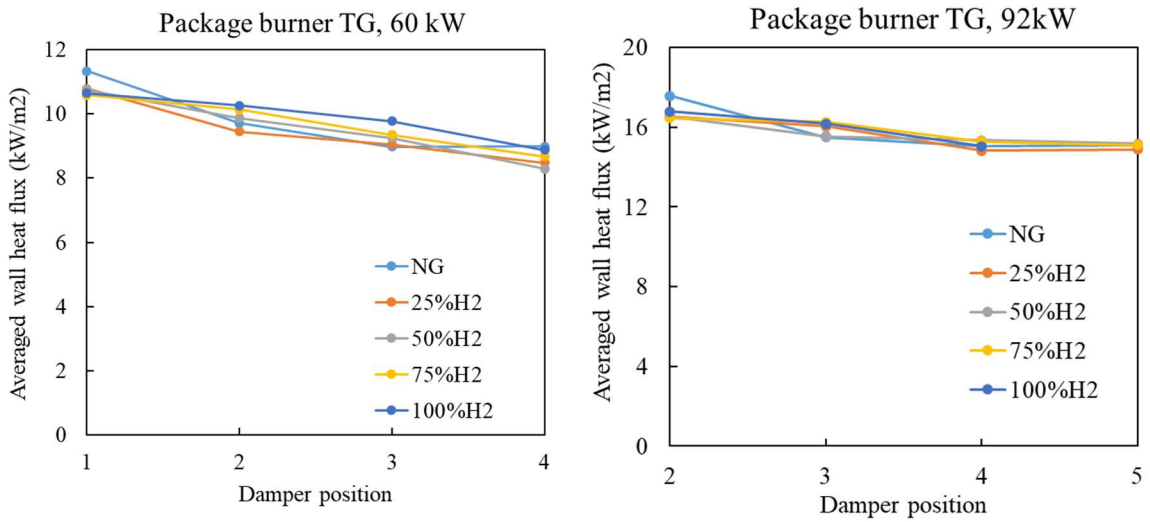

Figure 53 The influence of damper position and H_2 fraction on the averaged wall heat flux for a package burner operated at 60 and 92 kW with a TG head.

3.3.2.2 Wall heat flux profile

Figure 54 shows the effect of H_2 fraction on the wall heat flux profile for a package burner operated at 60 and 92 kW with a TG head. H_2 fraction has an insignificant effect on the wall heat flux profile for the TG as was the case for the original burner head. On the other hand, the wall heat fluxes along the furnace length are consistently lower than the NG case with the original head.

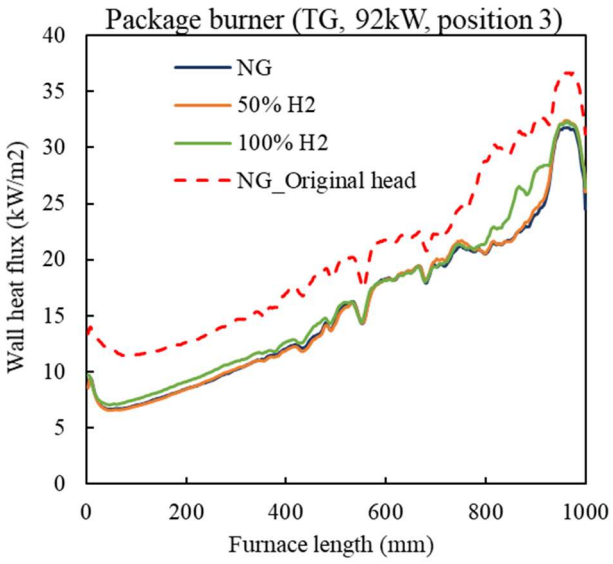

Figure 54 The influence of H_2 fraction on the wall heat flux profile for a package burner operated at 60 and 92 kW with a TG head.

3.4 NOx emissions with burner modifications

3.4.1 Atmospheric nozzle (AN) inspirator burner

Figure 55 presents NOx emissions from the AN burner operated 90 vol% or 100% hydrogen with various fuel injector sizes and excess air ratios (flue oxygen concentrations). No clear impact of the modification of fuel injector size (which affects primary air ratio) and flue oxygen concentration on the NOx emissions was found for the studied flames. Therefore, for the AN burner at high H_2 fractions (>90 vol%), the H_2 fraction dominates the NOx emissions.

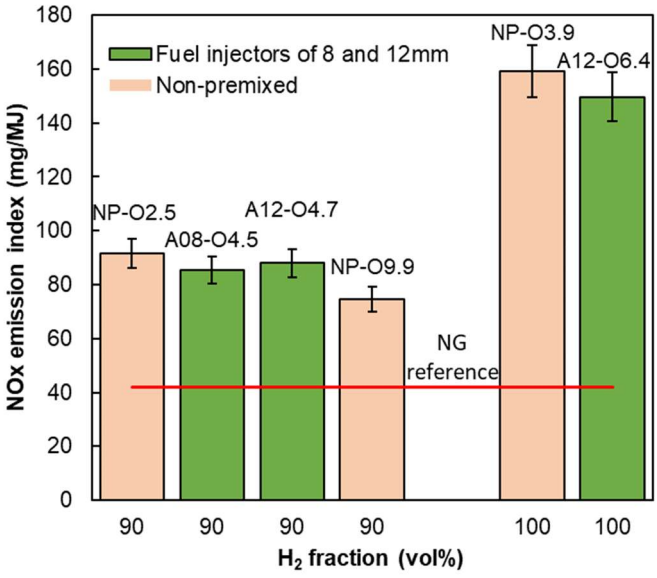

Figure 55 The influence of H_2 fraction on NOx emissions for an AN burner operated with various fuel injector orifice sizes and flue O_2 concentrations

3.4.2 Package burner with TG head

Figure 56 shows the effect of excess air ratio and H_2 fraction on the NOx emissions from a package burner fitted with a TG head operated at 60 and 92 kW. In general the NOx emissions obtained for the TG head are much

higher than those for the original head and increase with increasing excess air. As H_2 is added up to 75 vol%, slight increases in NOx emissions (<25%) were found relative to natural gas. Further significant increases in NOx emissions were observed for 100% hydrogen.

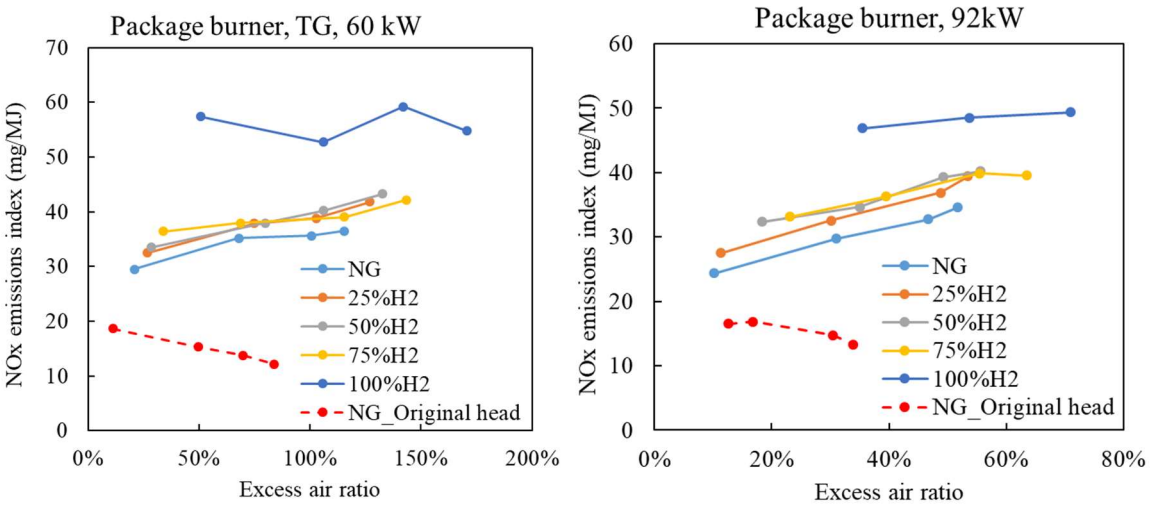

Figure 56 The influence of excess air ratio and H_2 fraction on the NOx emissions for a package burner operated at 60 kW (left) and 92 kW (right) both with a TG head.

3.4.3 Summary of operation with burner changes

For the AN burner operated in partially premixed mode, the heat flux peak and heat flux profile of a 90 vol% hydrogen flames can be adjusted to match that of the baseline natural gas flame by adjusting the primary air ratio and excess air. However, flashback is far more difficult to avoid for 90vol% and 100% hydrogen flames than for natural gas. NOx emissions for 90 vol% and 100% hydrogen were mainly dependent on the fuel composition with little effect from burner operation changes such as partial premixing and excess air.

For the package burner, with the same damper position, the TG head produces lower wall heat fluxes and higher NOx emissions than the original head. For 100% H_2 flames, the NOx emissions from the TG head were about four times higher than the original head.

4 CONCLUSIONS

A simple and robust method was applied to investigate the influence of the addition of H_2 to natural gas on the heat flux profile and NOx emissions from three commercial burners firing into thin-wall enclosures. The effects of low-cost burner modifications on mitigating the challenges of H_2 blending on the heat flux profile and NOx emissions were investigated.

It was found that, for the unmodified AN burner operated under conditions which ensured flashback did not occur (up to 50 vol% H_2 fraction), the blending of H_2 shifted the peak heat flux location towards the burner head without significantly changing the peak heat flux. Shifting of the peak heat flux location can be explained by the shorter flame with the blending of H_2. On the other hand, for the non-premixed cases (achieved by closing the primary air valve of the burner) H_2 addition had an insignificant effect on peak heat flux and the heat flux profile. An increase in secondary air controlled by increasing the opening of a flue damper caused a decrease in heat flux peak but insignificant change in the heat flux profile. Furthermore, the modification in fuel injector size, and hence primary air rate, was able to modify the influence of H_2 addition on heat flux profile. As an example, for an H_2 fuel fraction of 90 vol%, operation with ~4.5 vol% O_2 in the dried flue gas when using the 8 mm fuel injector provided a good match in both the peak heat flux and heat flux profile to the reference natural gas flame. The NOx emission index of the reference natural gas flame was approximately 40 mg/MJ and no significant change was found for the blending of H_2 up to 50 vol%. However, for H_2 fractions of 90 vol% and 100%, the NOx emission indexes were approximately doubled and quadrupled, respectively, compared to the reference case.

A forced air package burner was tested at 92 and 60 kW and excess air ratios from 0 to 150%. The heat transfer in the furnace was not significantly influenced by the addition of H_2. NOx emissions, slightly increased from approximately 20 to 25 mg/MJ as the H_2 fraction was increased to 50 vol%. A decrease in NOx emissions was found as the H_2 fraction was further increased from 50 to 100 vol%. For a heat input of 92 kW, the NOx emissions with 100% H_2 were lower than the corresponding natural gas flames with the same inlet air damper position. A “town gas” head was applied to the package burner and lower noise was observed. For the “town gas” head, the wall heat flux profile was consistently lower than that from the normal head, but the influence of H_2 fraction was small within the TG head cases. The NOx emissions were found to be much higher for the TG head than those using the original head especially for the 100% H_2. For example, for the package burner with the TG head operated at 60 kW using 100% H_2, and the inlet air damper set to position 1, the NOx emission index was 57.4 mg/MJ but only 18.6 mg/MJ using the original head.

For the high-velocity nozzle-mix burner, it was found that H_2 addition had only small effects on the wall heat flux profile. NOx emissions, for H_2 fractions lower than 60 vol% were within the range from 36 to 41 mg/MJ. NOx emissions were slightly higher for higher H_2 fractions. For example, for 60 vol% H_2 and 100% H_2 with 30% excess air, the NOx emissions indexes were 40.3 and 47.6 mg/MJ respectively.

5 IMPLICATIONS AND RECOMMENDATIONS FOR INDUSTRY

The research outcomes from this package have been instrumental in enabling low-level blends of hydrogen into the existing natural gas networks. The prevalence and diversity of these appliances amongst customers means that actual test data is required to support decisions by the networks on how much hydrogen can be blended, and also used in discussions between networks and customers on how their appliances will behave once the blending occurs. It has also been critical in guiding industry discussions on how applicable Australian standards should be updated to account for hydrogen.

The influence of H_2 addition on heat transfer and NOx emissions were varied significantly between burners/enclosures. However, in general, for the studied burners operated with a H_2 fraction of up to 30 vol%, confined firing produced similar performance results to natural gas firing. For the three tested burners, previous open firing tests showed no issues (with flashback etc) until higher than 50 vol% H_2 was used. Hence, if gas network operators progress to a 20 vol% hydrogen blend with natural gas or biomethane then there won't be operational issues. Slight adjustments may be required to fuel and air rates, but where there is automatic control based on temperature, the fuel rate will be adjusted automatically.

For the high velocity burner, the manufacturer already has a 100% hydrogen version, with improved high temperature materials, but other than that this work confirms that the burner operates well with hydrogen.

For the package burner and AN burner, different burner designs are required for 90 vol% or 100% hydrogen, with the modifications attempted here only partially successful in matching both heat transfer and NOx emissions of the base case natural gas flames. However, the objective was not to provide functional burner designs but to illustrate the types of changes to burner design and operation that will be necessary to convert to 90 vol% or 100% hydrogen.

The focus in this report is on experimental results, whereas the next report in this project (Milestone 7) will go into more detail on the implications for industry.

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APPENDIX 1 TYPE B SUBMISSIONS

Type B Submission

Stainless steel combustion chamber with single-stage package burner (PC001)

a. Name and address of appliance manufacturer.
The University of Adelaide

The University of Adelaide, North Terrace campus, Adelaide, South Australia, 5005, Australia

b. Name and address of the authorised installer or conversion contractor.
Ignite services,
Address: 8 Cardiff Court, Cavan SA 5094

c. Name and address of commissioning person.
Terence Ricketts
Address: ignite services, 8 Cardiff Court, Cavan SA 5094

d. Name and address of organization(s) responsible for risk assessment.
The University of Adelaide
The University of Adelaide, Adelaide, South Australia, 5005, Australia

e. Name and address of organization where appliance is or is to be installed.
The University of Adelaide
Building 11, 35-37 Stirling Street, THEBARTON SA 5031

f. Customer contact and telephone number.
Neil Smith
The University of Adelaide
0487645550

g. Appliance type, operation and process description.
Confined firing test for H_2 and natural gas blends

- Appliance type: A stainless steel cylindrical experimental combustion chamber with a single-stage package burner
- Operation & process: Burner fired into a stainless-steel combustion chamber with blends of hydrogen/natural gas in the University's Thebarton combustion laboratory, under constant supervision from University researchers and technical staff (note that burner has previously been proven to operate safely up to 100% H_2 based on open-firing tests).

h. Number of burners and type.
One (1) package burner with Siemens LME 39.100 C2 controller and UV flame detection.

i. Nominal gas consumption for total appliance and for each main burner.

330 MJ/h (maximum rated heat input on natural gas)

j. Gas consumption at ignition for each burner.
60 SLPM natural gas (133 MJ/h)

k. Air flow rate at ignition for each burner.
At the maximum firing rate, air flow is approx. 100 m^3/h or 1700 SLPM.

l. Volume of each combustion chamber.
Only one combustion chamber with the volume of 0.29m^3

m. Total volume swept by the combustion products from the burner(s) to each flue connection.
0.327 m^3

n. Air flow rate during purge periods.
>1100 SLPM (ref. purge time calculations)

o. Details and method of operation of any combustion air or flue dampers.
The air is supplied by a blower that is housed within the package burner assembly. There is an adjustable damper for the air supply, which will be set to ensure sufficient air supply (as per pre-purge calculations). The air pressure switch is adjusted to ensure sufficient air supply.

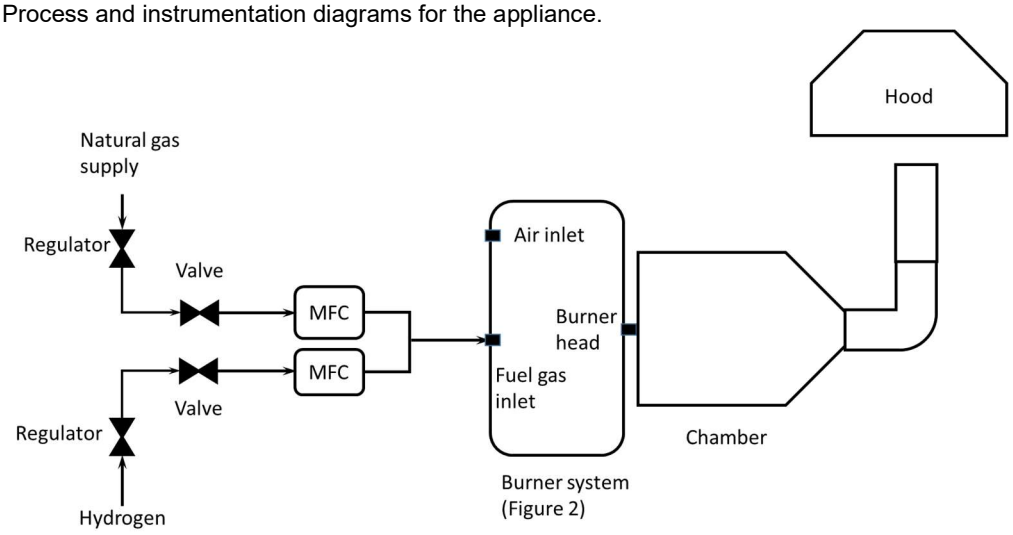
p. Details of any explosion reliefs including location, cross-sectional area and weight together with calculations (refer AS 1375 Appendix E).
The strength of the enclosure is estimated at 7 kPa, based on Clause E4.2 of AS 1375 App. E note 1(c):

"Ducts of normal light sheet metal construction". As per the same clause, "it is assumed that an internal pressure below 7kpa is generally unlikely to endanger personnel, although it may damage the appliance."

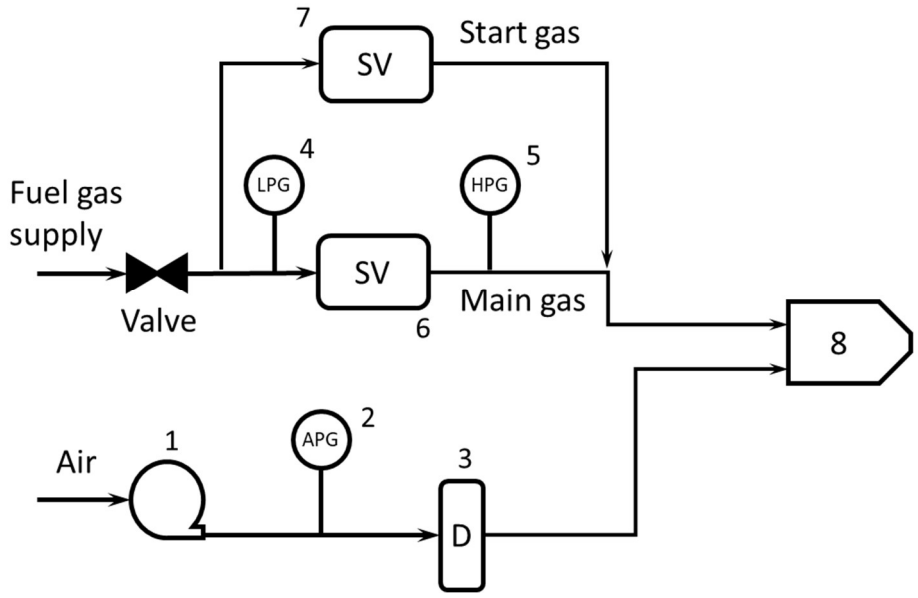
q. Appliance marking plate details (refer Clause 4.1 for requirements).
All appliances shall have clear, permanent markings that are readily accessible and easy to read when the appliance is installed.

The OEM's name: The University of Adelaide
The model identification: Combustion chamber, PC001 (means confined package no. 001).
The nominal gas consumption: 300 MJ/h
The gas type: H_2 and natural gas blends
The maximum and minimum gas supply pressure (kPa): 7 kPa (fixed supply pressure)
Purge times (minors): 90s
The gas pressure at burner head, for the nominal gas consumption (kPa): TBC Check
The combustion chamber volume (m^3): 0.29
The maximum appliance process design temperature (°C) and pressure (kPa):
T_{max} = 1100 °C (stainless steel)
P_{max} = 7 kPa
The total volume swept by the combustion products in passing from the burner to the flue connection (m^3): 0.327
The Serial number: PC001 (means confined package no. 001)
The date of manufacture: September 2023
The purge volume for combustible atmospheres (m^3) (when applicable): 1.64m^3
The dilution air volume m^3/min at 20 °C (where applicable): N/A
The type and maximum quantity of the design solvent(s) (L/h): N/A

r. Process and instrumentation diagrams for the appliance.



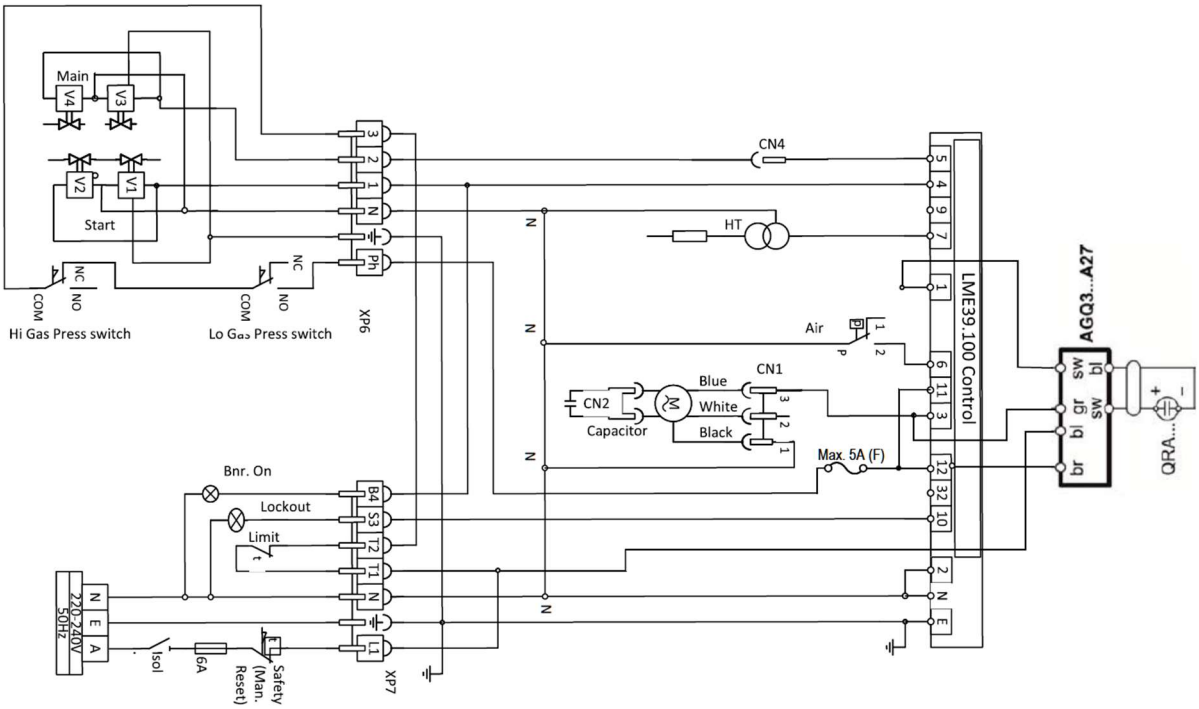
s. Schematic drawing (Figure 2) of the combustion air control system and fuel valve train specifying all valve train components (i.e. their brand, model number, size and rated working pressure) and the proposed setting of all adjustable devices



BOM for BS2F burner system						
	Part no.	Description	Size	Qty	Rated pressure [kPa g.]	Proposed setting
1		Centrifugal fan with forward curve blades		1		
2	LGW 10 A1	Air pressure switch		1	50 (withstand) 1 (adj. range)	
3		Adjustable damper		1		
4	DG 50N-3	Low gas pressure switch,	1/4"	1	60 (withstand) 5 (adj. range)	
5	DG 50H-3	High gas pressure switch,	1/4"	1	60 (withstand) 5 (adj. range)	
6	CG10R70-D1W6A	Double block valve for start gas	1/2"	1	7	
7	CG10R70-D1W6A	Double block valve for main gas	1/2"	1	7	
8	BS2F	Burner, 127-330 MJ/hr		1		

t. Schematic electrical wiring diagram showing the safety and control circuits including details of the brand, model number and method of operation of each major component and the proposed setting of

any adjustable device.



u. Purge time calculations.
The minimum purge air flow rate is equal to the air supplied by the blower with the damper set the "most-closed" position, which is 1100 SLPM.
Purge time=5*V/V_a, V is the purge volume (327 L) while V_a is the air volumetric flow rate (1100 SLPM).
Therefore, Purge time=5*327/1100=1.49mins=89.4 s - therefore the purge time on the programmable controller is set to 90 seconds.

v. Calculations of start gas rate conditions (refer Clause 3.2.3).
In the appliance, the incoming fuel is mixed continuously with air within the appliance and with incoming air. Therefore, to calculate the start gas rate conditions, the case B method introduced in Appendix D of AS 1375 is used/referred.

$$T = \frac{V_a G}{F} \ln\left(\frac{G}{G - \left(L/K\right)}\right)$$

Where T is critical time, Va is the volume of appliance (m^3), G is concentration of fuel in the fuel plus air supplied to the appliance (m^3/ m^3), F is fuel input rate to the appliance (m^3/s). L is concentration of fuel in a fuel/air mixture which is at the lower explosive limit (m^3/ m^3), 0.05 for NG while 0.04 for H_2, K is safety factor 2 for NG while 4 for all other fuels.

Natural gas:

Va=0.327 m3, Maximum start gas fuel flow rate is 60 SLPM ($1*10^{-3}m^3/s$) for natural gas.
G=60/(60+1100)=0.052

Therefore, for natural gas, T=0.327*0.052/($1*10^{-3}$)*ln(0.052/(0.052-0.05/2))=11.2 seconds > setting critical time which is 3 s.

w. Air dilution rate calculations for processes involving solvents or dusts (refer AS 1375).
N/A

x. Ventilation provided or to be provided in the area where the appliance is or is to be installed (refer AS 5601).
Details of the ventilation as required by AS 5601 are included in the document “Gas Installation Qualitative Risk Assessment”.

y. N/A Method of flueing and location and type of flue material.
A vertical stainless steel flue is connected to the opening of the chamber via an elbow and vents the flue gas to the extraction hood.

z. Operating instructions.
Details of the operating instructions are included in the “SOP for PC001” document.

aa. Documentation requirements for PES system.
Not applicable since have a BMS not a PES.

Type B Submission

Stainless steel combustion chamber with high-velocity nozzle-mix burner (TC001)

Name and address of appliance manufacturer.
The University of Adelaide
The University of Adelaide, North Terrace campus, Adelaide, South Australia, 5005, Australia

- Name and address of the authorised installer or conversion contractor.
  Ignite services,
  Address: 8 Cardiff Court, Cavan SA 5094
- Name and address of commissioning person.
  Terence Ricketts
  Address: ignite services, 8 Cardiff Court, Cavan SA 5094
- Name and address of organization(s) responsible for risk assessment.
  The University of Adelaide
  The University of Adelaide, Adelaide, South Australia, 5005, Australia
- Name and address of organization where appliance is or is to be installed.
  The University of Adelaide
  Building 11, 35-37 Stirling Street, THEBARTON SA 5031
- Customer contact and telephone number.
  Neil Smith
  The University of Adelaide
  0487645550
- Appliance type, operation and process description.
  Confined firing test for H_2 and natural gas blends
  - Appliance type: A stainless steel box-shaped experimental combustion chamber with a high-velocity nozzle-mix burner
  - Operation & process: Burner fired into a stainless-steel combustion chamber with blends of hydrogen/natural gas in the University’s Thebarton combustion laboratory, under constant supervision from University researchers and technical staff (note that burner has previously been proven to operate safely up to 100% H2 based on open-firing tests).
- Number of burners and type.
  One (1) high-velocity nozzle-mix style burner with Honeywell/Kromschröder controller (BCU 370 model burner).

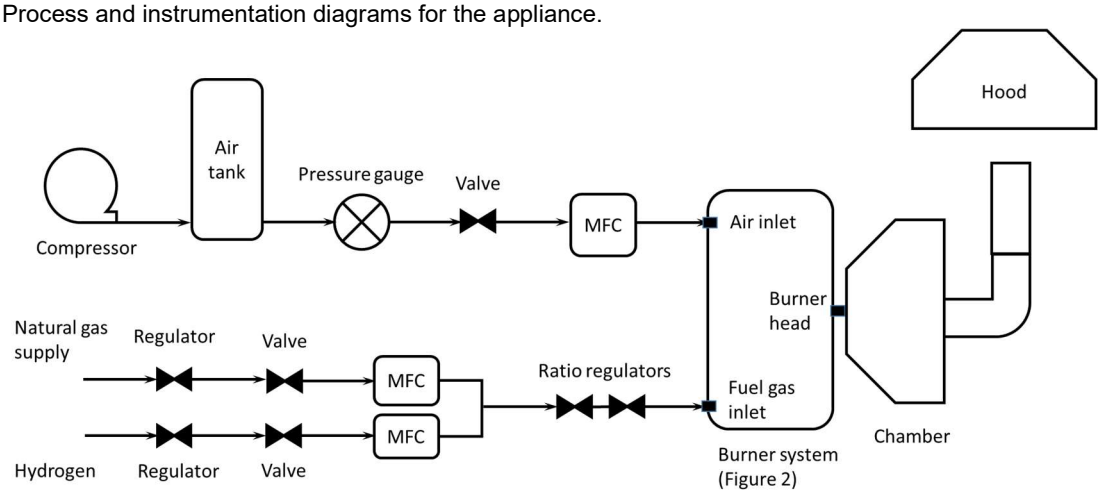
  Nominal gas consumption for total appliance and for each main burner.

  The nominal gas consumption is based on an air flowrate of 400 SLPM (24 m^3/h) and an excess air of 30% - this gives a gas consumption value of approximately 73.3 MJ/h for natural gas and about 94.3 MJ/h for H2. This represents the upper limit of gas consumption, as the burner will also be operated at lower firing rates. The upper limit of air flowrate is based on the description from the OEM, which gives a maximum of 880 scfh (415 SLPM) when burning a stoichiometric mixture (noting that the exact flowrate is sensitive to the tile pressure resulting from combustion).

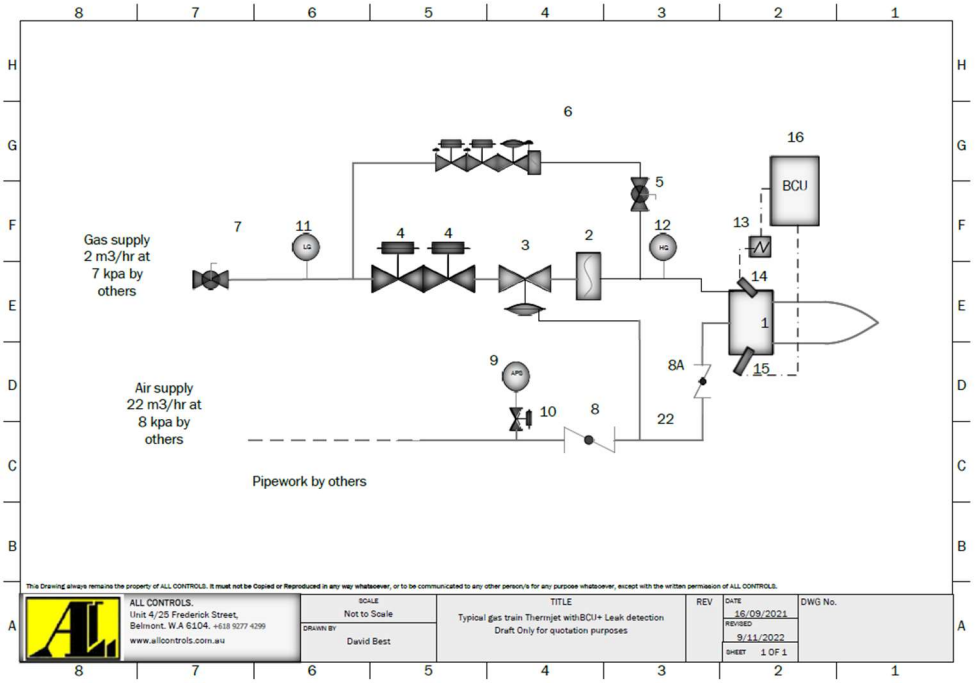
- Gas consumption at ignition for each burner.
  The start gas flow rate is controlled by adjusting the regulator on the start gas bypass to achieve reliable ignition.
  The maximum start gas rate is 50 SLPM (which is ~113 MJ/h) for natural gas and 35 SLPM (which is about 25.4 MJ/h) for H2.
  Note: The start gas rates of 50 SLPM and 35 SLPM are the maximum gas flows which occur when combustion does not occur (i.e. with the ignitor/spark plug disabled) – due to the back-pressure on the combustion tile, the actual flow during a successful ignition is significantly lower than these maximum values.
- Air flow rate at ignition for each burner.
  400 SLPM
- Volume of each combustion chamber.
  Only one combustion chamber with the volume of 0.26m^3
- Total volume swept by the combustion products from the burner(s) to each flue connection.
  The elbow and flue have a total volume of 0.04 m^3. Therefore the volume swept by combustion products is 0.26+0.04=0.3 m^3
- Air flow rate during purge periods.
  400 SLPM (with low-flow error-checking)
- Details and method of operation of any combustion air or flue dampers.
  The air is supplied by the lab air compressor at a pressure of 600 kPa, which is reduced via a mass flow controller which controls the air flow rate to the appliance. Downstream butterfly valves (components 8 and 8A in valve train schematic) are used to control the pressure drop for air-flow proving and air-fuel ratio control.
- Details of any explosion reliefs including location, cross-sectional area and weight together with calculations (refer AS 1375 Appendix E).
  The strength of the enclosure is estimated at 7 kPa, based on Clause E4.2 of AS 1375 App. E note 1(c): “Ducts of normal light sheet metal construction”. As per the same clause, “it is assumed that an internal pressure below 7kpa is generally unlikely to endanger personnel, although it may damage the appliance.”
- Appliance marking plate details (refer Clause 4.1 for requirements).
  All appliances shall have clear, permanent markings that are readily accessible and easy to read when the appliance is installed.

  The OEM’s name: The University of Adelaide
  The model identification: Combustion chamber, CT001.
  The nominal gas consumption (MJ/h): 73 for natural gas and 94 for H2. Between 73 and 94 for natural gas and hydrogen blends
  The gas type: H_2 and natural gas blends
  The maximum and minimum gas supply pressure (kPa): 7 kPa (fixed supply pressure)
  Purge times (minors): 245s
  The gas pressure at burner head, for the nominal gas consumption (kPa): <5 kPa
  The combustion chamber volume (m^3): 0.26
  The maximum appliance process design temperature (°C) and pressure (kPa):
  T_{max} = 1100 °C (stainless steel)
  P_{max} = 7 kPa
  The total volume swept by the combustion products in passing from the burner to the flue connection (m^3): 0.3
  The Serial number: CT001.
  The date of manufacture: September 2023
  The purge volume for combustible atmospheres (m^3) (when applicable): 1.5m^3
  The dilution air volume m^3/min at 20 °C (where applicable): N/A
  The type and maximum quantity of the design solvent(s) (L/h): N/A

- Process and instrumentation diagrams for the appliance.



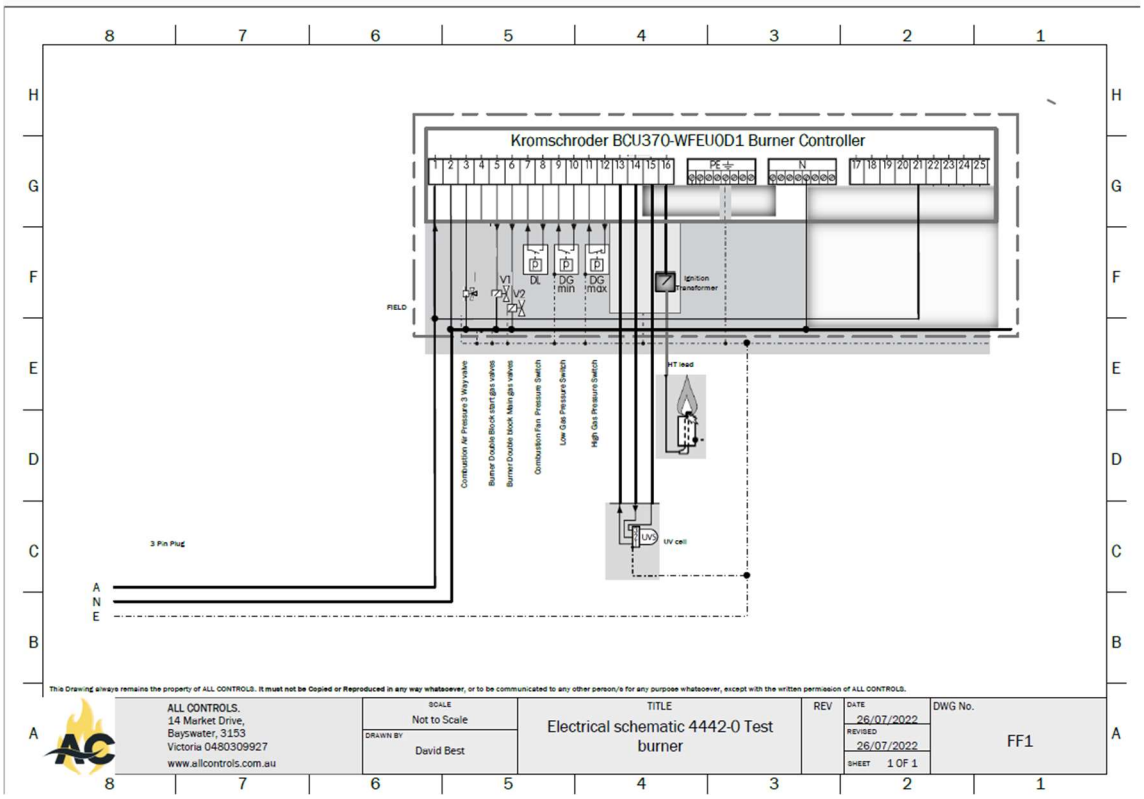
- Schematic drawing of the combustion air control system and fuel valve train specifying all valve train components (i.e. their brand, model number, size and rated working pressure) and the proposed setting of all adjustable devices



BOM for 4442-0 burner system						
	Part no.	Description	Size	Qty	Rated pressure [kPa g.]	Proposed setting
1	N4442-0	Burner 62-79 MJ/hr	3/4 NPT air	1	-	Varied over rated range
2	N1807-0	Gas limiting valve	3/4" NPT	1	172	
3	KR0802K	Ratio regulator	1/2" BSP	1	20	As required
4	B2550	GVC25 Double block valve	3/4" BSP	1	10	
5	W15M	Pilot valve	1/2" BSP	1	2100	
6	KR22070K	CG10 230v	1/2"	1	7–10	Adjusted for stable ignition

						in open-firing
7	W20M	Burner isolator	3/4"	1	2100	
8	ET-BV-MRP-1	Air limiting manual reduced port Butterfly valve	1" BSP	1	35	Fully open
8A	ET-BV-MRP-1	Burner trim Air limiting manual Butterfly valve	1" BSP	1	35	Fully open
9	KR20012T-NPT	Air pressure switch DG150	1/4"	1	60 (withstand) 15 (adj. range)	Set to 65 mbar (triggers at <370 SLPM air)
10	007SCD320A184	3 way valve	1/4"	1	>100 kPa	
11	KR20010	DG50U	1/4"	1	60 (withstand) 5 (adj. range)	
12	KR20012T-NPT	High gas pressure switch DG150	1/4"	1	60 (withstand) 15 (adj. range)	
13	B0115	Ignition transformer	-	1	-	-
14	-	Spark plug	-	1	-	-
15	KR4030D-1G1	UV cell	-	1	-	-
16	KR28268	Burner controller BCU370	-	1	-	-

- Schematic electrical wiring diagram showing the safety and control circuits including details of the brand, model number and method of operation of each major component and the proposed setting of any adjustable device.



- Purge time calculations.
  The purge air flow rate is set to be 400 SLPM which results in a required purge time of 225 seconds.

  Purge time=5*V/V_a, V is the purge volume (300 L) while V_a is the air volumetric flow rate (400 SLPM). Therefore, Purge time=5*300/400=3.75mins=**225 s slightly shorter than 245 seconds which is the setting purging time**

- Calculations of start gas rate conditions (refer Clause 3.2.3).
  In the appliance, the incoming fuel is mixed continuously with air within the appliance and with incoming air. Therefore, to calculate the start gas rate conditions, the case B method introduced in Appendix D of AS 1375 is used/referred.

$$T = \frac{V_a G}{F} \ln\left(\frac{G}{G - \left(L/K\right)}\right)$$

  Where T is critical time, Va is the volume of appliance (m^3), G is concentration of fuel in the fuel plus air supplied to the appliance (m^3/ m^3), F is fuel input rate to the appliance (m^3/s). L is concentration of fuel in a fuel/air mixture which is at the lower explosive limit (m^3/ m^3), 0.05 for NG while 0.04 for H_2, K is safety factor 2 for NG while 4 for all other fuels.

  Natural gas:

  Va=0.3 m3, Maximum start gas fuel flow rate is about 48 SLPM (set to be 50 SLPM which is $8.33*10^{-4}$$m^3$/s) for natural gas, air flow rate is 400 SLPM. G=50/(50+400)=0.11

  Therefore, for natural gas, T=0.3*0.11/($8.33*10^{-4}$)*ln(0.11/(0.11-0.05/2))=10.2 seconds > setting critical time which is 5 s.

  H2:

  Start gas flow rate is set to be 35 SLPM (which is $5.83*10^{-4}$$m^3$/s m^3/s)

  G=35/(35+400)=0.08

  T=0.3*0.08/($5.83*10^{-4}$)*ln(0.08/(0.08-0.04/4))=5.5s > 5s which is the setting critical time.

- Air dilution rate calculations for processes involving solvents or dusts (refer AS 1375).
  N/A
- Ventilation provided or to be provided in the area where the appliance is or is to be installed (refer AS 5601).
  Details of the ventilation as required by AS 5601 are included in the document “Gas Installation Qualitative Risk Assessment”.
- N/A Method of flueing and location and type of flue material.
  A vertical stainless steel flue is connected to the opening of the chamber via an elbow and vents the flue gas to the extraction hood.
- Operating instructions.
  Details of the operating instructions are included in the “SOP for TC001” document.
- Documentation requirements for PES system.
  Not applicable since have a BMS not a PES.

RP1.4-08
Milestone 7:

Medium velocity burner, site test, modelling, implications for Standards

February, 2025

Project number: RP1.4-08

Detailed assessment of Type B appliances with blends of hydrogen and natural gas and 100% hydrogen

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Australian Government
Department of Industry, Science, Energy and Resources

Business Cooperative Research Centres Program

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PROJECT INFORMATION	
Project number	RP1.4-08
Project title	Detailed assessment of Type B appliances with blends of hydrogen and natural gas and 100% hydrogen
Research Program	RP1 Future Fuel Technologies, Systems and Markets
Milestone Report Number	7
Description	Milestone Report
Research Provider	University of Adelaide
Project Leader and Team	Peter Ashman, Paul Medwell, Neil Smith, Doug Proud, Peijun Guo (University of Adelaide)
Industry Proponent and Advisor Team	Lead Industry Adviser/Team: Rob Davis, Troy Praag (AGIG) Jaga Namebeley and Richard Brewell (ATCO) Paul Beaumont (RSHQ), (OTR), Peter Kacperek (Enscope) Kevin Dwyer (ESV), Eric Bardy (SEA Gas) David Forster (GPA Engineering) Appliance and burner specialists: Anu Anand (Techrite and GAMAA) Stephen Pearce (GASCO) Vince Monsignor (GasTrain) Matthew Anderson, Peter Newman (Furnace Engineering) Terence Ricketts (Ignite Gas) David Best (All Controls)
Related Commonwealth Schedule	Milestone RP1.4.3 “Solutions for new/modified industrial equipment delivered to allow for a broader range of fuel compositions” by detailed planning of testing and other assessments.
Project start/completion date	01-Oct-2021 / 30-Dec-2024
IP Status	Confidential
Approved by	Rob Davis
Date of approval	Feb 2025

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Summary of Report

This report provides new results, showing the effects of blending hydrogen into natural gas on open firing of a medium velocity burner, over the range from natural gas to 100% hydrogen, and provides new results of firing a two-stage package burner in an industrial pyrolysis furnace, comparing natural gas with a 30 vol% H2 blend. It also provides further analysis on heat transfer and efficiency of laboratory firing of an enclosed package burner which leads into a discussion of how convective and radiant heat transfer will be affected by hydrogen addition in industrial cases.

The key finding is that blending hydrogen into natural gas at the same heat input rate and same air flowrate produces similar heat transfer, irrespective of the blend. To maintain constant heat input, the volumetric flowrate of the fuel blend needs to be slightly higher than for natural gas since the Wobbe index of blends reduces slightly from natural gas. Most practical applications control firing rate to a set temperature, so in many cases no changes would be required and in others the fuel pressure could be slightly increased to maintain constant heat input.

Open firing of a medium velocity burner with a metal cone showed that it is highly stable with hydrogen blends and for near stoichiometric conditions the flame appearance is very similar for natural gas and up to 50 vol% hydrogen. It operated well with 100% hydrogen at 30% excess air, but with a 200^{o}C increase in cone temperature.

An experimental study in an industrial pyrolysis furnace with a two stage package burner showed that operating with a 30 vol% hydrogen blend at the same inlet pressure as natural gas resulted in 7 – 8% less heat input, as expected due to the reduction in Wobbe Index. The control system had a set temperature of 400^{o}C, which was achieved by spending more time running on high rate with the blend. A second experiment illustrated that running the blend at slightly higher pressure to achieve the same heat input rate as natural gas produced similar periods on high and low rate to those for natural gas. There were insignificant differences in measured temperatures within the furnace, which were a proxy for a product quality measurement. These results should provide confidence in using blends up to 30 vol% hydrogen in industry with no changes, or minor changes to burners.

Further analysis of enclosed firing of Type B burners reported in the Milestone 6 report illustrated that as hydrogen is blended into natural gas at constant air rate there is little change in heat transfer. Heat transfer efficiency was approximately 50% and reduced with increasing air rate, as did flame temperature. This is because the constant air rate means that flame temperatures and flow velocities are relatively similar. If the air rate is dropped as hydrogen is blended in, to maintain constant excess air, the flame temperature will increase, but the velocity will drop. These two effects would tend to cancel each other out and result in little change in convective heat transfer, but the higher temperature may result in higher radiant heat transfer which may balance a reduction in gas emissivity for 100% hydrogen. For 100% hydrogen, there may be a drop radiation heat transfer in processes where it plays a role along with convection, but can be accommodated by a slight increase in fuel rate.

A table of potential burner modifications was presented for the > 30 vol% hydrogen range, whereas for 0-30 vol% hydrogen blends, natural gas burners can be operated without change or with minor changes to operation.

A comprehensive review of applicable Standards was not conducted, but the project carefully considered start gas requirements and showed that measuring oxygen in the flue during start up on 100% hydrogen may be difficult with the current procedure. Hence it is recommended to check that there is confidence in measuring 20.5% O_2 (dry) in flue gas and if not, consider whether it is possible to change the criteria for % LEL in Table 3.1 of AS3814:2018. Another recommendation is to investigate the scientific reasons that flame ionization does not work for hydrogen and if there is solution to this, other than using UV flame detectors.

1 INTRODUCTION

1.1 Background and motivation

This Milestone 7 report covers the remaining pieces of work for the project, whilst the final Milestone 8 report will compile the project reports and summarise the whole project.

The items covered in this report from Section 2.5 of the plan in the Milestone 2 report are:

- A site test in an industrial furnace with a hydrogen blend
- combustion and heat transfer model results based on experiments and using correlations for prediction of performance, failure modes and operational limits
- A “tool” for modifications to key burner types for different blend ratios and for 100% hydrogen
- General information on changes required to equipment or operating procedures
- Recommendations for updates in appliance standards for 100% hydrogen appliances.

In addition, a demonstration test on full-scale furnace with a 30vol% hydrogen blend was conducted instead of the planned task of a cost estimate for a 10vol% demonstration.

Some items in the plan were not relevant, namely, training on updates to standards can be carried out later if there are updates, and cost estimates for conversion or retrofit of appliances were limited to a description of the necessary changes for different fuel compositions. Having said that, costings for 10 vol% hydrogen only relate to the supply, storage, metering and blending of hydrogen because no changes are required to existing plant.

1.2 Scope and aims

This report aims to use an understanding of combustion and heat transfer fundamentals to translate findings from experiments with Type B burners and small furnaces to industrial applications. The scope includes reporting on:

- Open firing tests of a new medium velocity burner - stability, metal temperatures and flame appearance
- Results from a site trial in a pyrolysis furnace with 30 vol% hydrogen blended into natural gas
- Further analysis of confined firing experiments reported in the Milestone 6 report
- Simple modelling - calculations on how varying fuel composition by adding hydrogen will affect heat transfer in industrial applications
- Overview of changes in operation required for burners for various hydrogen blends and 100% hydrogen
- Implications for Standards and training

2 EXPERIMENTAL EQUIPMENT AND METHODS

2.1 Open firing a new medium velocity burner

The Thermjet TJ0015-MVATNGFR medium velocity burner shown in Figure 57 is suitable for various fuels including natural gas, propane and butane. However, its performance with natural gas/hydrogen blends has not been reported. The purpose of these tests was to assess burner ignition and stability with natural gas / hydrogen blends and to suggest a maximum hydrogen concentration that can be reliably used without making any changes to the burner.

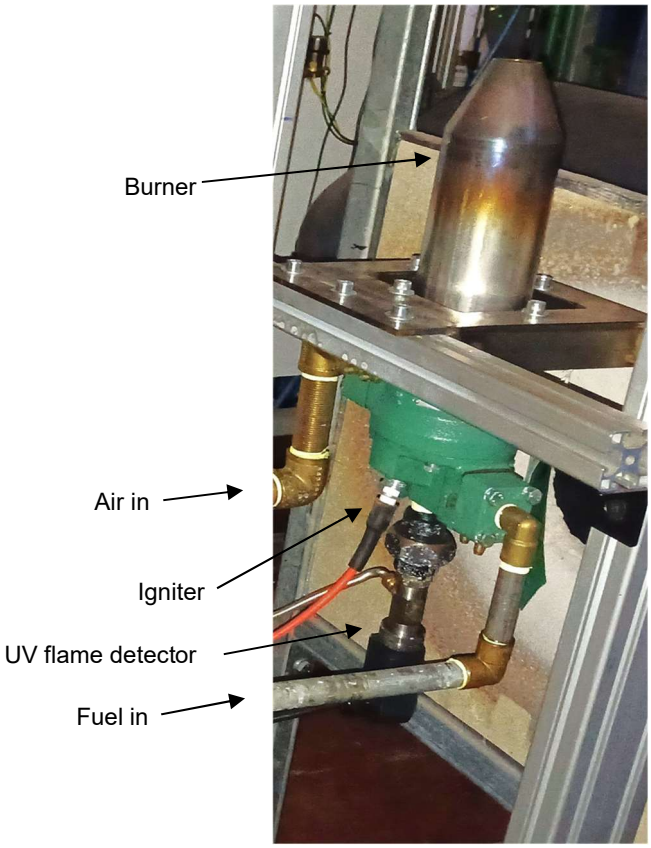


Figure 57 Thermjet TJ0015 Medium velocity burner mounted for vertical open firing

The smallest available medium velocity nozzle-mix burner was tested in open firing. The fuel and air enter a conical combustion zone separately, prior to mixing and ignition. The design produces a medium velocity flame, which when installed in a furnace, rapidly entrains surrounding furnace gas and so can cause large recirculation currents to promote even heat transfer and control NOx emissions. In this test the burner was fired vertically upwards in the open (Figure 57), rather than in a furnace, and qualitative flame photographs were taken to illustrate flame appearance as a function of varying fuel composition (natural gas and blends with hydrogen up to 100% hydrogen firing). A UV sensor was installed as the flame detector.

These medium velocity burners are typically operated in the range from slightly rich (λ= 0.9) to 30% excess air (λ= 1.3). To test these conditions, the burner was operated at 23kW with varying fuel composition and air/fuel ratio by controlling the flowrates of air and natural gas and hydrogen through the burner. Tests were performed at stoichiometric and 30% excess air (λ= 1.0 and 1.3) for four different fuel compositions, namely, natural gas, 20vol% hydrogen blend, 50 vol% hydrogen blend and 100% hydrogen.

Then to test an extreme rich condition the same fuel compositions and rates were operated with approximately 30% of stoichiometric air through the burner (λ ~ 0.3). This operating condition is unlikely unless the burner is being used for partial combustion or has a fault but was tested as a worst case scenario.

2.2 Pyrolysis Furnace

A commercial full-scale pyrolysis furnace was used to demonstrate firing of a blend of 30 vol% hydrogen in natural gas and compare it to natural gas firing. The furnace has a volume of 10.2 m^3 with a purge time of 12 minutes for 5 air changes. The furnace is used commercially to clean plastic coated or painted steel parts by operating at pyrolysis conditions using the main burner which fires horizontally into a firing tube, after which, the post combustion gases fill the furnace. A second burner in the exit flue completes combustion of the fuel and the liberated organics. The main furnace operates to a set point of 400°C measured with a thermocouple located near the top of the back wall of the furnace by switching the main burner, a two-stage package burner, between high and low fuel rate.

For the demonstration, only the main burner was used under combustion conditions at approximately 400 MJ/h with substantial excess air, rather than pyrolysis conditions. The second burner was not used. Instead of a load of plastic coated parts, eight thermocouples were located equidistant from each other and walls as illustrated in Figure 58 to measure temperature uniformity and changes for the different fuels. In addition, a single steel hook was placed inside the centre of the array of thermocouples to simulate a load and other thermocouples were located on the primary burner and the high temperature region of the burner tube.

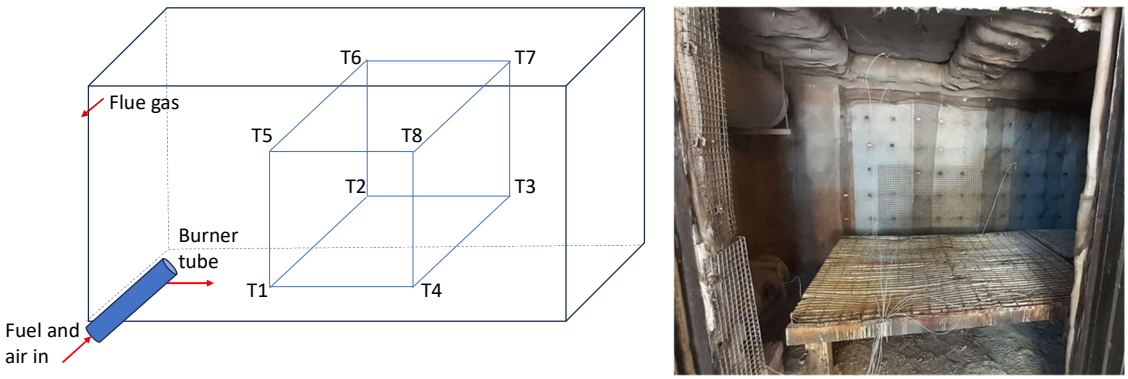


Figure 58 A schematic diagram of arrangement of the pyrolysis furnace showing burner tube and thermocouple locations, and photo with left half door open

The local natural gas composition in the network nearest the location (Table 10) was used to calculate Wobbe index and higher and lower heating values of NG and the 30% hydrogen blend, as shown in Table 11, along with values for hydrogen.

Table 10 Natural gas composition, %, and specific gravity at Springvale on March 9th 2024

methane	93.26241
ethane	4.65248
propane	0.26079
butane_i	0.02538
butane_n	0.02362
pentane_i	0.00245
pentane_n	0.00472
hexane	0.01121

nitrogen	0.99757
carbon_dioxide	0.75454
spec_gravity	0.59258

Table 11 Heating values and Wobbe index for the Pyrolysis furnace trial

		NG	H_2	NG + 30 vol% H_2
HHV	MJ/Sm^3	38.53	12.09	30.60
LHV	MJ/Sm^3	34.76	10.23	27.40
WI	MJ/Sm^3	49.90	45.83	46.23

Hydrogen was supplied from a single G sized bottle at a time through a regulator (Figure 59) previously used for the laboratory tests. A single bottle is sufficient for approximately 2 hours running at 30vol% blend at a total firing rate of 400 MJ/h.

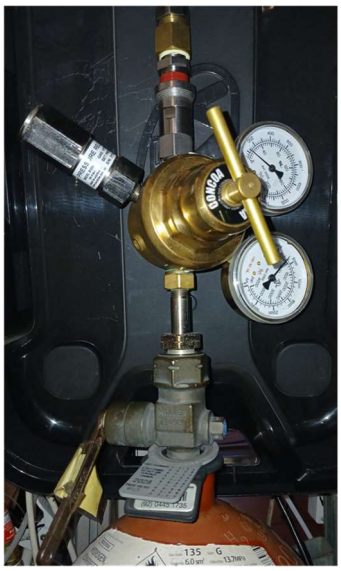


Figure 59 Hydrogen bottle regulator

The blending station from the University of Adelaide laboratory was used. Regulators were used to bring the fuel pressures down to approximately 10 kPa, and then, as shown in Figure 60, Alicat mass flow controllers and a flashback arrestor were used for each fuel before teeing together and being plumbed into the existing burner valve train via a hose. To operate with fuel blends the mass flow of one of the flows was set and a program used to measure and control of the other fuel to the desired ratio. Note that for short term trials, materials selection was not a prime consideration and a mixture of brass and stainless steel fittings were used (Figure 60).

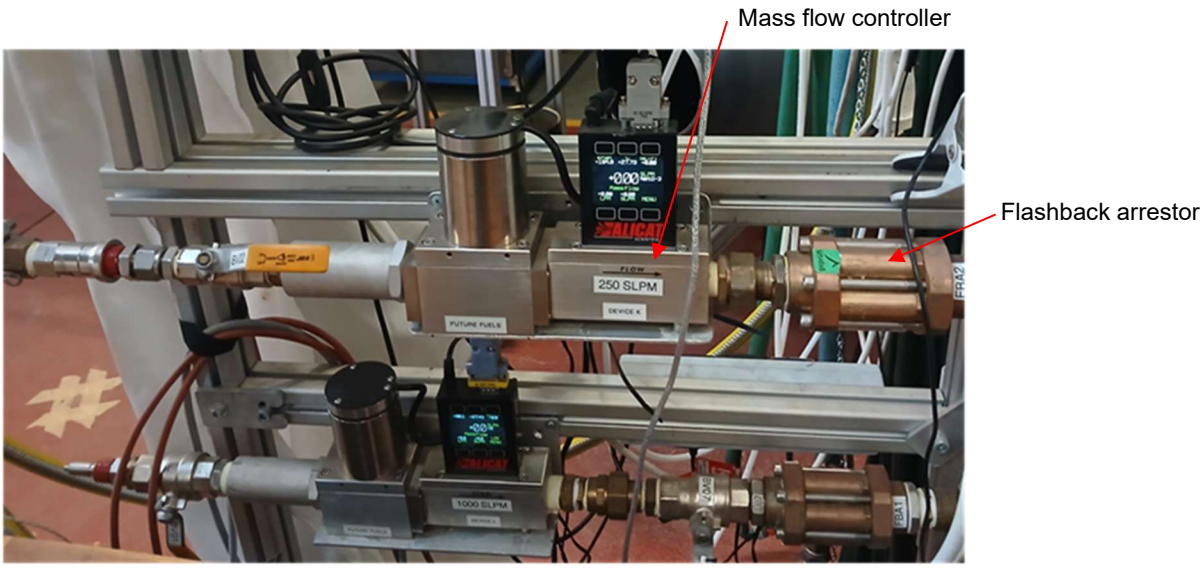


Figure 60 Mass flow controllers and flashback arrestors

2.3 Enclosed firing of a package burner

The milestone 6 report detailed experiments in which Type B burners were fired into enclosures with thin walls to allow heat loss from the enclosure to the room by radiation and natural convection from the surface. After five to ten minutes of operation at set conditions the wall temperatures were steady and results were taken. Full details are available in the Milestone 6 report so are not repeated here. The current report contains further analysis of results from enclosed 60kW firing of a package burner to illustrate how performance will change in industrial processes.

3 RESULTS - OPEN FIRING OF A MEDIUM VELOCITY BURNER

Since the medium velocity burner is a nozzle mix burner, flash-back does not occur (as it can in premixed burners). Flames were stable and operated without any flame detection issues for natural gas and all flames as hydrogen was added up to 100% hydrogen. Figure 61 presents a series of photographs of flames at a range of hydrogen concentrations with 30% excess air or stoichiometric air through the burner. In general, the stoichiometric flames are longer than the flames with 30% excess air but otherwise have similar flame appearance for the same fuel composition. The 100% hydrogen flames were less visible and red in colour due to a combination of hydrogen emission bands and excitation of sodium in the air.

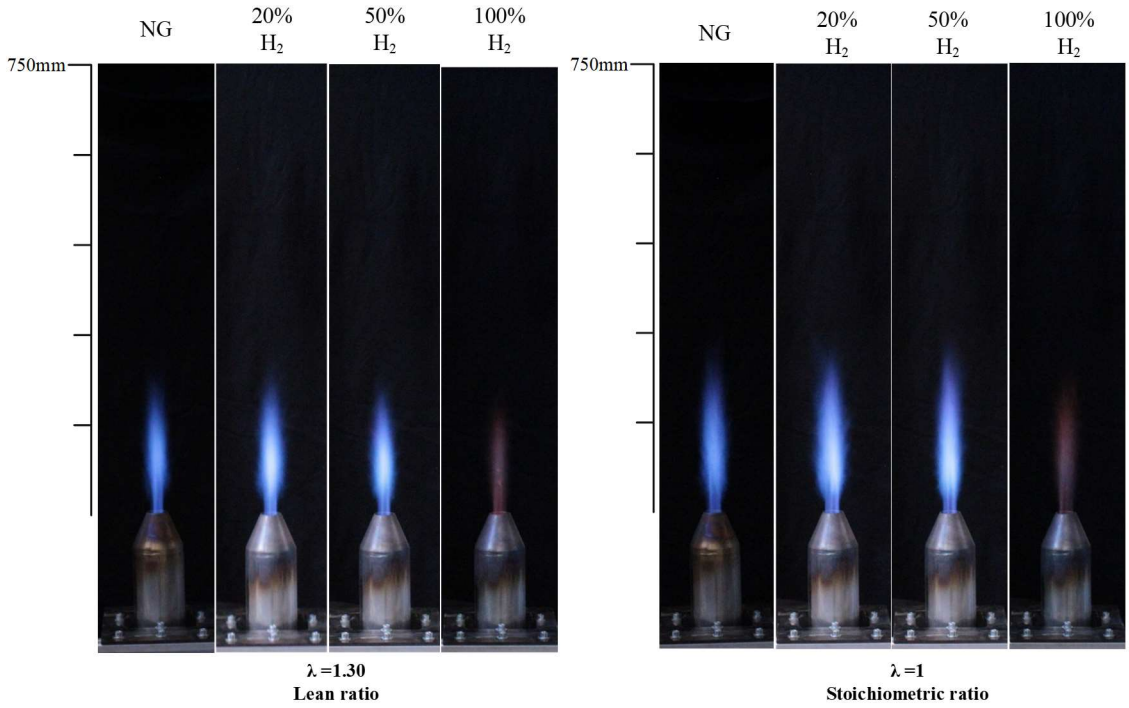


Figure 61 Qualitative photographs of 23 kW "medium velocity burner" flames with varying fuel blends with flows through the burner controlled at 30% excess air and stoichiometric. Note that the same length scale is used for both images.

Since there were no issues with stability and flame detection via the UV flame detector as hydrogen content was increased to 100%, the maximum practical hydrogen % will depend on avoiding overheating of the burner cone. Since hydrogen burns readily, a higher fraction of the fuel is burnt near the fuel exit as the hydrogen content of the blend increases and could cause overheating. However, in practice overheating will depend on the application, since these burners are used in a variety of environments with different temperatures. These open firing tests show only indicative trends in metal temperature.

To investigate burner temperature differences with different fuel compositions a thermal imaging camera was used to take images inside the combustion head, one second after flames were turned off (after running a flame for 15 minutes) for both natural gas and hydrogen flames with 30% excess air. Results, shown in Figure 62, illustrate that the 100% hydrogen flame had higher metal temperatures by approximately 200°C than the natural gas flame. The difference may be small enough for operation of these burners unmodified with 100% hydrogen in some applications which do not expose the burner to extreme temperatures provided regular maintenance procedures are followed. For other applications it may be wiser to redesign the burner for 100% hydrogen.

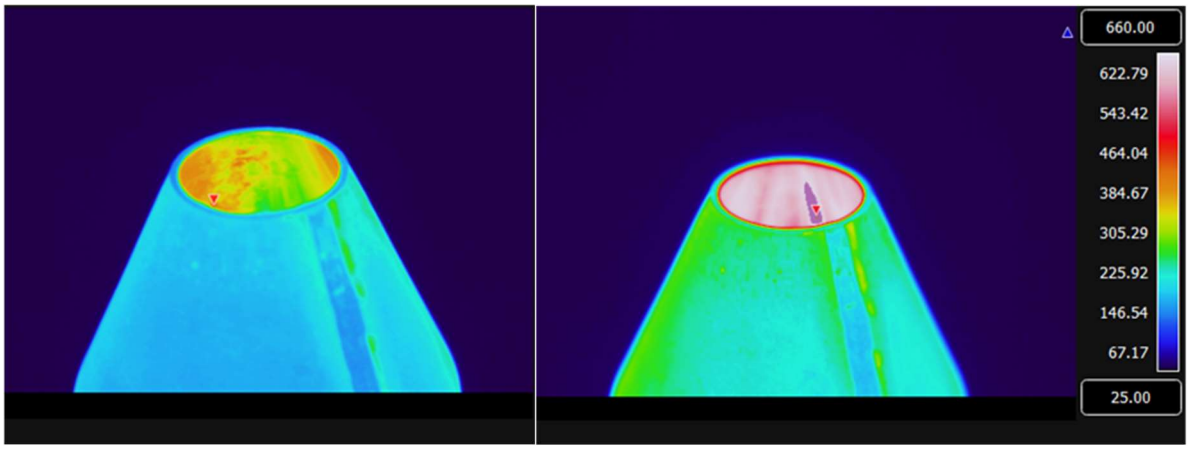


Figure 62 Burner cone thermal images for natural gas and 100% H2 with λ = 1.3, 1 s after flames were turned off. The same colour scale (°C) is used in both images.

Extremely rich 23 kW flames, with λ ~ 0.3 are shown in Figure 63. The air flow rate for all these rich flames was 100 SLPM which resulted in a slight variation in the air to fuel ratio as shown in the figure. The flame lengths are all longer than the near stoichiometric flames shown in Figure 61 because of the limited air supply, requiring diffusion of outside air into the flame after exiting the burner. The limited air supply also resulted in soot formation and hence luminous flames for natural gas and the 20 vol% blend. For 50 vol% there is much less luminosity due to reduced carbon in the fuel and for 100% hydrogen the orange colour from sodium excitation is much more pronounced than in the near stoichiometric flames.

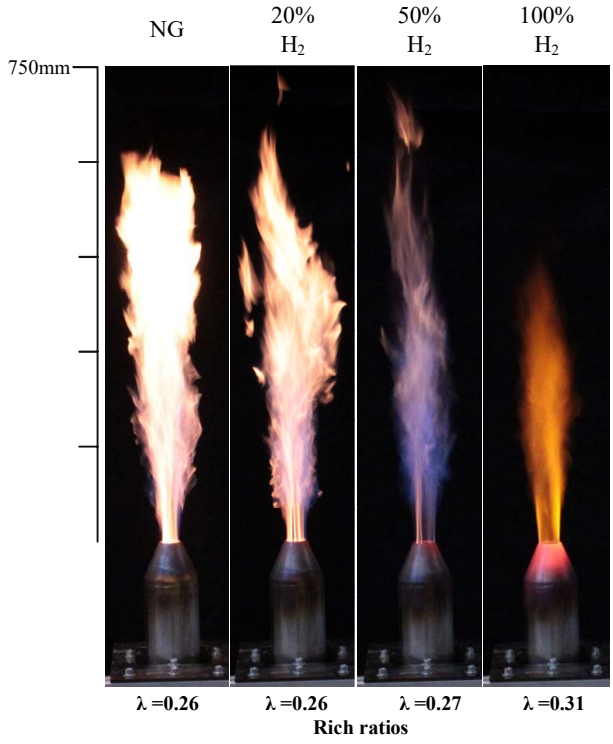


Figure 63 Qualitative photographs of 23 kW "medium velocity burner" flames with varying fuel blends with flows through the burner controlled at very fuel rich conditions. Note that the same length scale is used as in Figure 61.

For the 100% H_2 case, with λ= 0.31, the burner cone overheated but after turning off the burner the metal was discoloured without visible/physical damage. Since these rich flames are very unusual operating conditions this overheating may not occur in practice, but the images illustrates that a burner redesign for 100% hydrogen may be warranted for some applications.

NOx emissions are shown in Figure 64 for all the flames. There was little difference in NOx emissions with varying fuel composition for stoichiometric flames and those with 30% excess air. Note that open firing NOx emissions tend to be lower than confined firing because of the lower temperatures, but the trend may be preserved in confined firing. For the rich flames, the 100% hydrogen flame shows increased NOx emissions but as mentioned, this condition was tested as an impractical operating condition since it caused more combustion in the burner cone and overheating which may have contributed to higher temperatures and NOx.

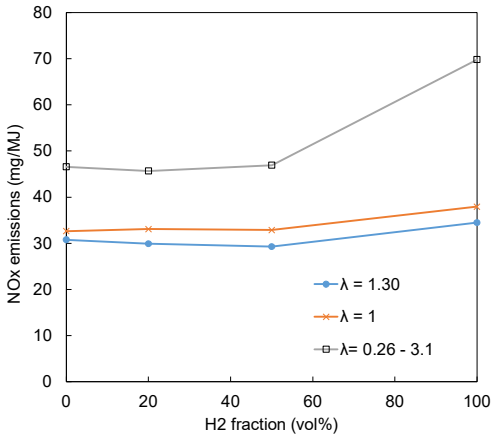


Figure 64 NOx emissions of the "medium velocity" burner flames.

4 PYROLYSIS FURNACE RESULTS

The first test compared natural gas firing to firing with a blend of natural gas and 30 vol% hydrogen at the same default burner pressure and air rate. For both cases the control system switched between high and low fuel rates to achieve 400ºC near the back top wall of the furnace. Figure 65 shows a representative example of typical changes that occurred when switching between the two fuel options. The 30 vol% blend was in operation from before 11:35 am and then a switch to natural gas was made at 11:46 a.m. There was significantly more time spent on high rate when operating on the blend.

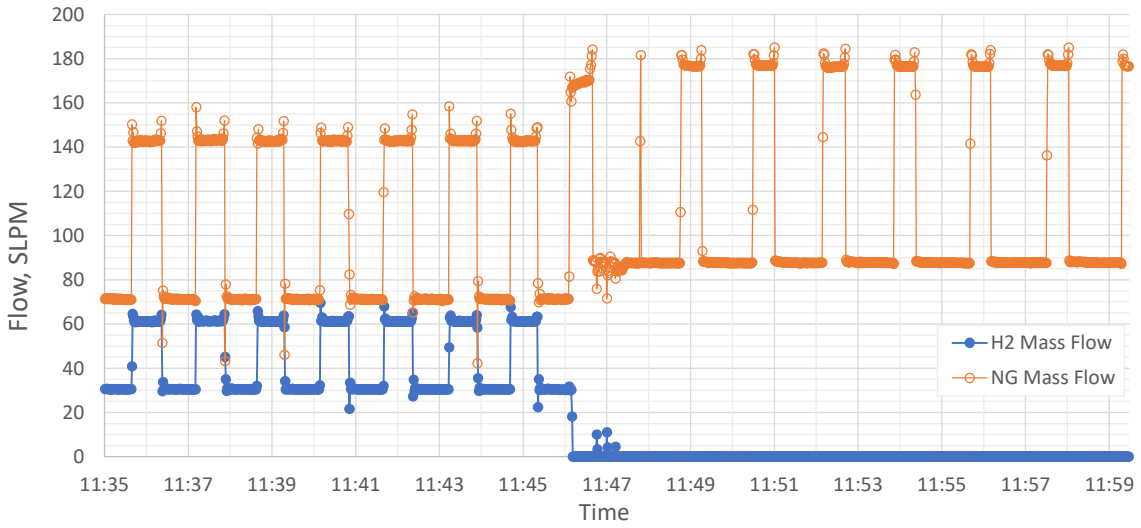


Figure 65 Flows of hydrogen and natural gas with the same burner pressure as baseline natural gas firing

The explanation for the greater fraction of time spent on high rate for the blend is its lower Wobbe index (Table 11). To illustrate how the difference in Wobbe index affected operation, Table 12 shows measured flow rates at both high and low rates for both natural gas and the 30% blend. The calculated energy input at both rates was 7 to 8% lower than for the blend than natural gas, whether based on either HHV or LHV. As a result the controller spent more time on high rate for the blend to keep the set temperature at 400ºC.

Table 12 Fuel flow rates and heat input for natural gas and the 30% blend when operated at the same pressure

		Fuel flow	Heat input	energy input change for the 30% blend based on HHV	Heat input	energy input change for the 30% blend based on LHV
		SLPM	MJ/h based on HHV	%	MJ/h based on LHV	%
NG	NG hi	174.52	403		364	
	NG lo	87.29	202		182	
30vo% blend	NG hi	142.84	330		298	
	NG lo	71.12	164		148	
	H2 hi	61.20	44.4		37.6	
	H2 lo	30.50	22.1		18.7	
	Total hi		374.6	-7.1%	335.5	-7.8%
	Total lo		186.5	-7.6%	167.0	-8.2%

Despite the change in heat input rates and times on high and low rates, Figure 66 shows that thermocouple temperatures and load temperature showed little change from firing the blend before 11:46 am to firing natural gas afterwards, relative to the effect of cycling between high and low rates (for clarity, not all of the thermocouples are shown). This is to be expected, due to the action of the temperature controller.

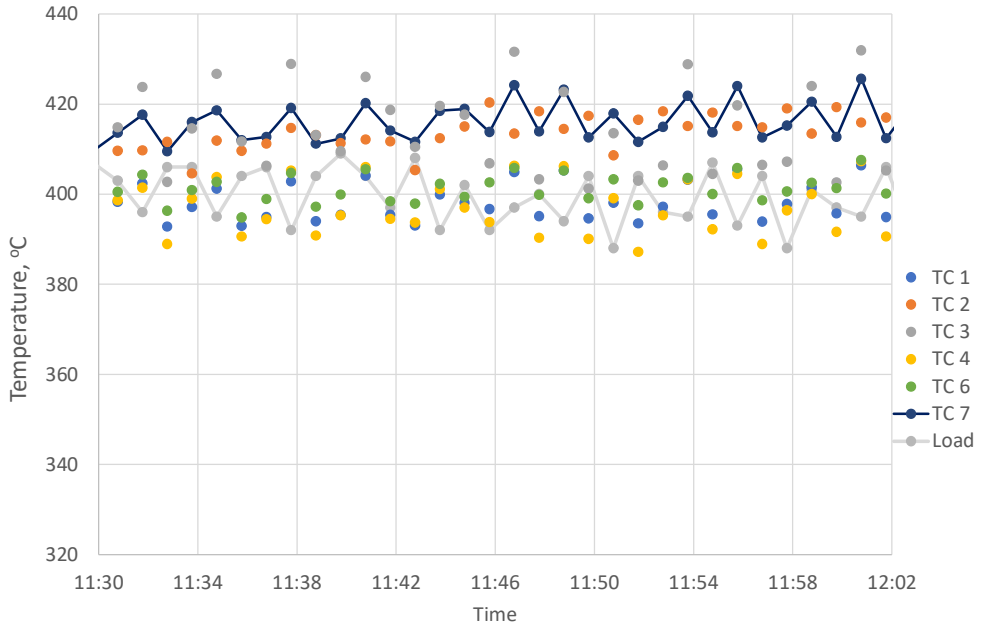


Figure 66 Temperatures sixteen minutes before and after a change from 30% hydrogen blend to natural gas at 7% higher heat input

Further tests were conducted in which the heat input of the blend and the natural gas flow were adjusted to be similar. Figure 67 shows the temperature trends for a test in which the 30% blend was operated before 10:12 am (the next day) and natural gas was operated after 10:12 am at the same heat input as the blend. As expected, there are no differences in temperatures before and after the fuel change.

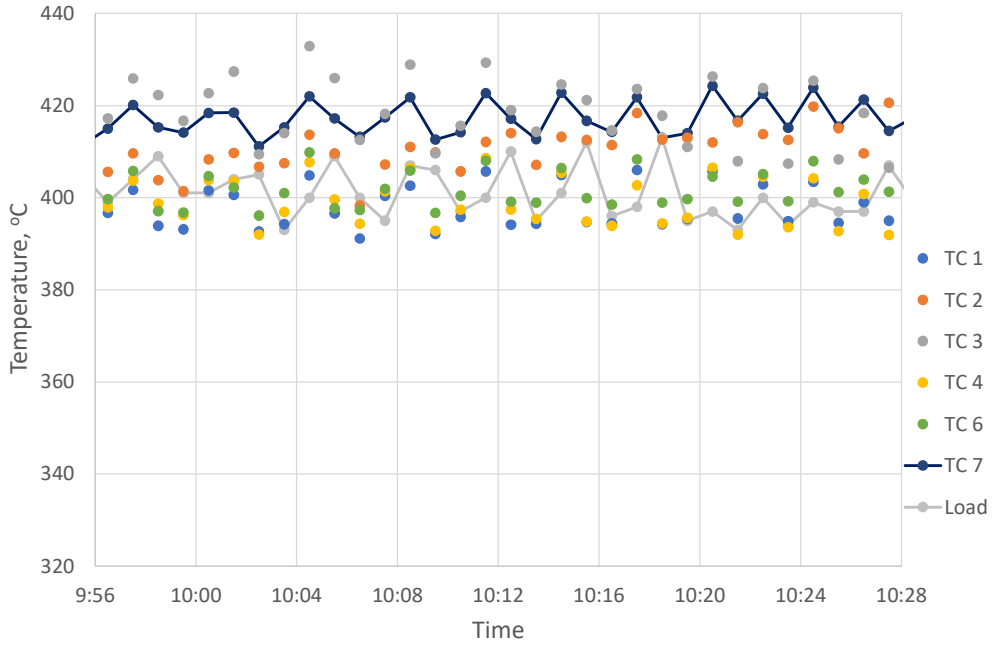


Figure 67 Temperatures sixteen minutes before and after a change from 30% hydrogen blend to natural gas at the same heat input.

Flue gas analyses show small fluctuations with switching from high to low firing rates, but the average results were consistent for both natural gas and the 30% blend, with 16.5 % oxygen, and 30 - 35 ppm CO and 16 - 20 ppm NOx for all firing rates and both natural gas and the 30 vol% blend.

In summary, the experiments showed no operational issues with the hydrogen blend. Temperature control prevented any change in process temperatures despite a reduction in energy input from the blend at equal inlet pressure. Later, an adjustment to slightly increase fuel flow for the blend allowed heat input equal that for natural gas.

The results show that 30 vol% hydrogen blends are manageable without further demonstrations. Hence there is also no need for further demonstrations with a 10 or 20 vol% hydrogen blends in processes that have temperature control since the changes are trivial, provided that small adjustments to fuel and air flowrates are possible.

5 ENCLOSED FIRING – FURTHER ANALYSIS

5.1 Significance of heat flux profiles

Further analysis of the experimental results in the Milestone 6 report is provided here to explain the heat transfer trends and illustrate how the results can inform operation of industrial appliances.

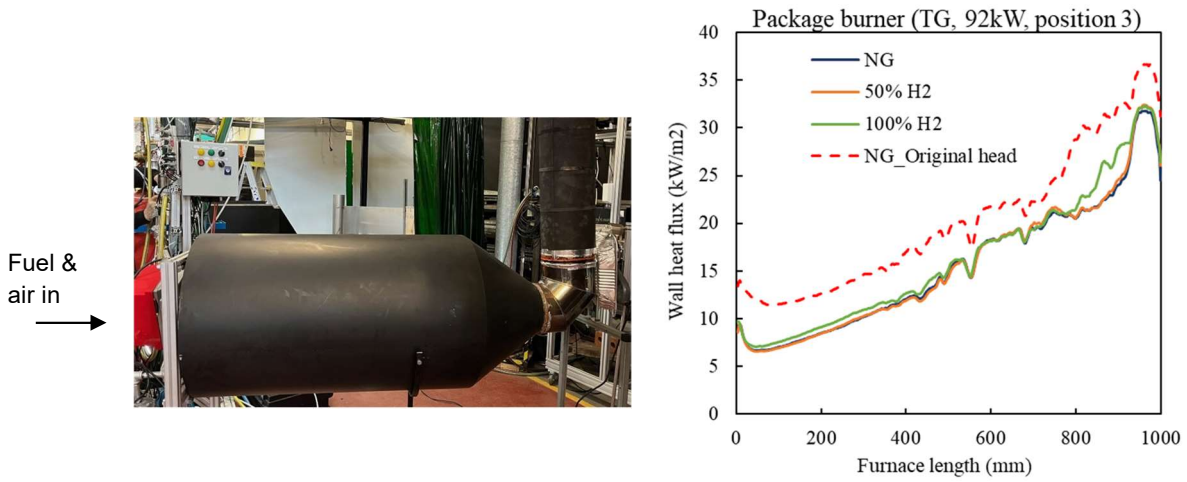
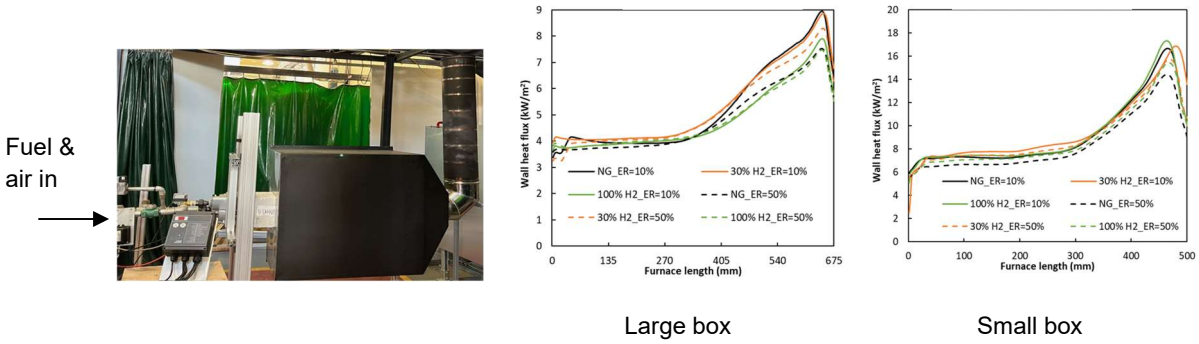


Figure 68 Package burner heat flux profiles (92 kW)

The experimental heat flux profiles shown in Figure 68 for the package burner operated with natural gas and blends up to 100% hydrogen and with two different burner heads are all not uniform and show a similar trend, increasing with distance from burner. The measurements were taken along the full length of the cylindrical section of the enclosure and did not extend into the exit cone. The broad similarity in profiles for such a wide range of fuels and for the two burners, illustrates that the geometry and flow patterns played the main role in determining the heat flux to the wall. In all cases the burner is approximately 100mm in diameter and is located on the centreline of the enclosure which has a diameter of 600mm, so the flame initially expands downstream without impacting the wall and then the hot post-flame gases likely impinge on the wall nearer the exit end and in the conical reducer with some recirculation of hot gases.

Note that the heat flux trends with length are of interest here, and the small dips in the profiles are not what is being discussed. They are probably due to variation in material thickness or differences in cleanliness both inside and outside the cylinder or imperfections in paintwork and are common to all the profiles. The final dip at 1000mm is probably due to a difference in material thickness at the weld between the cylinder and the cone.

A similar pattern of increasing heat flux with distance from the burner was found for the high velocity burner in both the large and small boxes (Figure 69). In this case the flame is short and only approximately 15 mm in diameter, so again the post-flame gases are not initially in contact with the wall and expand with distance from the burner. The boxes have a baffle at the exit end forcing the flow to the wall before exiting into the elbow and flue gas pipe, which helps to create the peak in heat flux at the exit end of the boxes.



Large box Small box

Figure 69 Heat flux profiles for confined firing of a high velocity burner in a large and small box

The significance of these heat flux profiles is to illustrate how important the specific design of a burner and a combustion chamber is to heat transfer performance. In these cases, if convection heat transfer was estimated based on average flow velocities it would be insignificant compared to radiation, whereas in fact, the profiles illustrate that convection near the exit end is dominating.

5.2 Efficiency

To estimate the efficiency of heat transfer from the flame and hot gases to the walls and then to the room, the average heat flux from the measured profile was multiplied by the total surface area of the enclosure, to calculate the total heat loss from the walls, Q, which was then divided by the fuel heat input (based on LHV), H_f.

$$Efficiency = \frac{Q}{H_f} \quad (1)$$

This method provides an estimate only, since the heat transfer rate is not uniform and was not measured for all the surfaces. The calculated package burner enclosure surface area included the front mounting plate, the main cylinder and the cone, but not the exit elbow and duct. For 60 kW flames (based on LHV), the calculated efficiency as a function of fuel composition and excess air is shown in Figure 70.

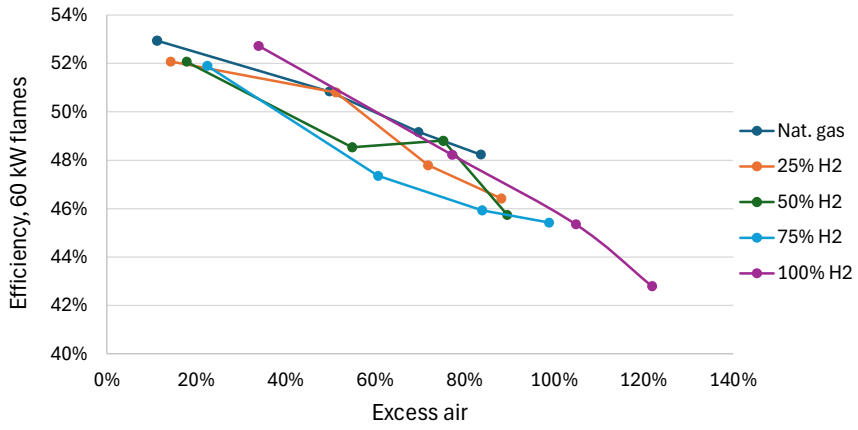


Figure 70 Efficiency of heat transfer from 60kW package burner flames enclosed flames to the room (based on LHV).

These experiments were not designed to be highly efficient. In fact, heat transfer is limited by reliance on natural convection and radiation from the enclosure to the room in addition to the convection and radiation inside the enclosures. So, it is not surprising that efficiencies range from 42 to 53%. The trends are of more interest than the overall efficiencies. There is a clear trend of reducing efficiency with increasing excess air, indicating that the reduction in temperature with increasing excess air has a greater effect on reducing both radiation and convection than any enhancement in convection that may have resulted from increased velocity. However, a clear trend with fuel composition is not evident.

Flue gas temperature, T_g, was measured with a shielded thermocouple, and was used, along with flue gas specific heat, to calculate the flue gas heat loss. Figure 71 shows results for experimental cases. The left most points on each line are for damper position 1, which had the lowest air flowrate. Looking only at the left most dots, as hydrogen is added to the fuel, at constant fuel rate of 60 kW and constant air flow rate, the excess air increases because of the lower stoichiometric requirement of hydrogen. The flue gas heat loss for this damper position is virtually unchanged because the air rate is constant but shows a marginal increase for 100% hydrogen since water vapour has a higher specific heat than carbon dioxide. It follows that at constant excess air, hydrogen flames have less heat in the flue, but this is almost entirely due to the fact that less air is used. The sum of the calculated heat loss from the surface of the enclosures and the heat lost in the flue was in the range 90 – 96% of the heat input.

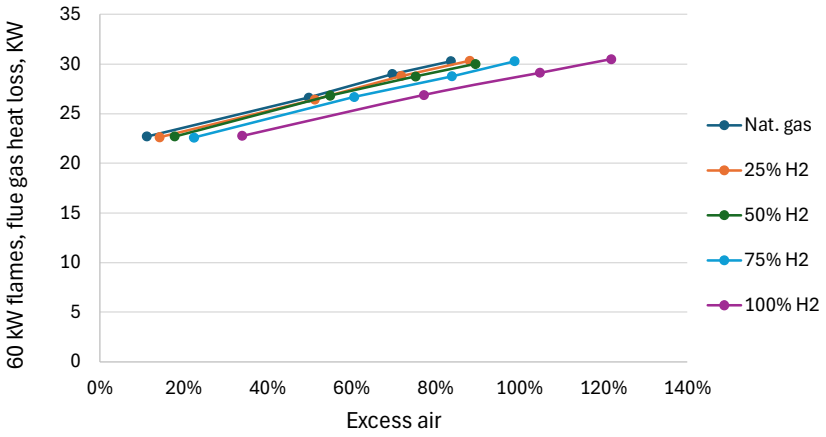


Figure 71 Calculated flue gas heat loss from 60 kW package burner enclosed flames

5.3 Flame temperature

The adiabatic flame temperature, T_{af}, is the theoretical maximum flame temperature that a given heat release, H_f, would produce if there is no heat loss and no dissociation of molecules. It has no practical application but indicates trends in actual flame temperatures, which are much lower. It can be calculated from the total mass rate of fuel and air, m, and the specific heat of the products, Cp, which varies with temperature and so requires an iterative solution. T_o is the temperature of reactants, in this case 15 °C.

$$T_{af} = \frac{H_f}{mCp} + T_o \qquad (2)$$

The calculated adiabatic flame temperatures for 60 kW enclosed package burner flames are shown in Figure 72. Again, the left most points on each line are at damper position 1, and have equal air flow rate, so the excess air increases with increasing hydrogen fraction, but T_{af} remains relatively constant for the same air flow rate. There is an overall trend of decreasing T_{af} with increasing air flow rate due to dilution. For constant excess air %, the addition of hydrogen increases T_{af} because of lower flow of air required.

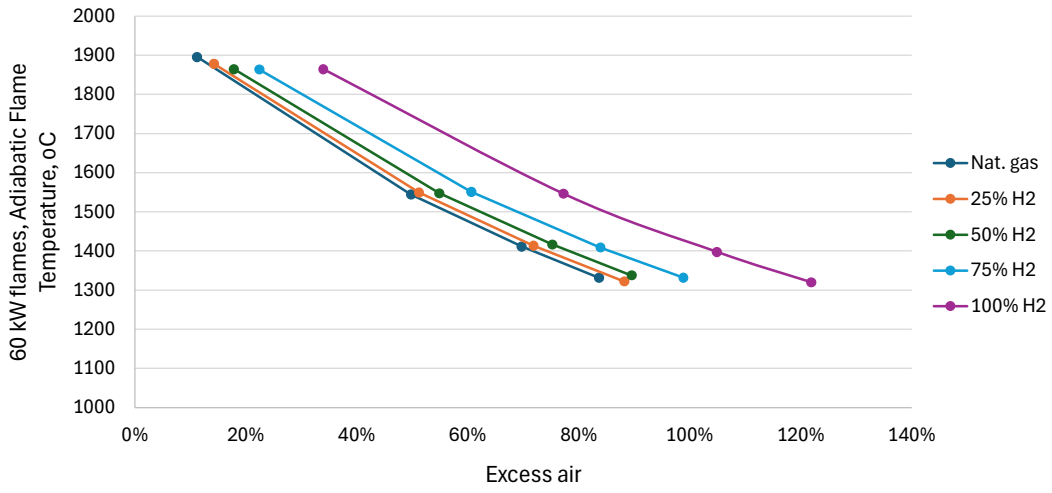


Figure 72 Enclosed package burner 60 kW flames, calculated adiabatic flame temperatures, T_{af}

An approximation of the actual flame temperature, T_f, can be made by assuming that the furnace is a simple well mixed chamber with constant properties throughout, and using the average heat loss through the walls and the measured flue gas temperature, T_g. The calculation scales up the measured flue gas temperature, by the ratio of total heat input, H_f, to the amount remaining in the flue, which is proportional to (H_f -Q), where Q is the measured heat lost through the walls. Hence,

$$T_f = \frac{T_g - T_o}{\left(\frac{H_f - Q}{H_f}\right)} + T_o \qquad (3)$$

Using this relationship the approximate flame temperatures for the 60kW enclosed package burner flames are shown in Figure 73. The temperatures are more realistic than the adiabatic flame temperature and could be used for modelling heat transfer and NOx emissions. The flame temperatures are relatively similar for the same damper position (and hence actual air rate), with a trend of reducing flame temperature with increasing excess air due to dilution. For the equal excess air, the 100% hydrogen flame has less actual air rate resulting in higher flame temperature, but the same trend was not observed for the 0-75% hydrogen blend range.

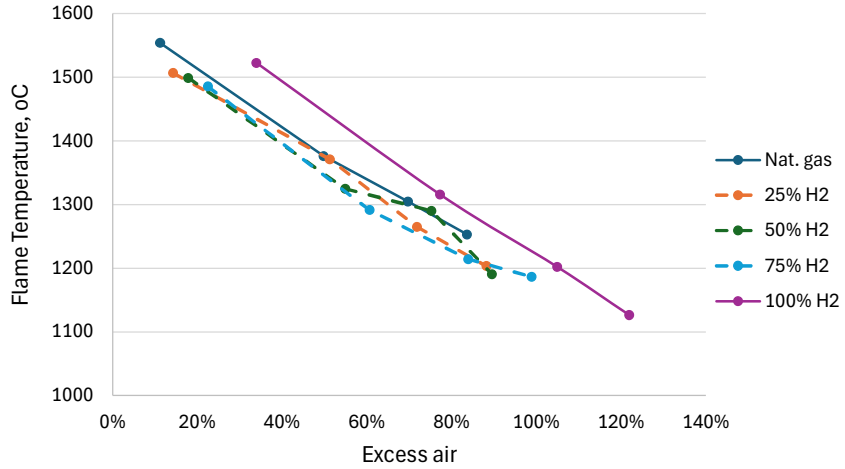


Figure 73 Approximate flame temperatures for 60kW enclosed package burner flames

6 HEAT TRANSFER MODELLING

Industrial applications use a mixture of radiant and convective heat transfer. In broad terms, convection dominates when the velocities are high, and radiation dominates when furnace volumes are large and temperatures are higher. Radiation from flame, flue gases and from hot walls can all play a role. Radiation is the sole mechanism in furnaces and ovens employing indirect heating from radiant tubes. Experimental results have shown that as hydrogen is blended into natural gas, minor heat transfer changes occur, but depend more air flow rate and burner and furnace design and hence flow patterns than on the $\%H_2$ in the blend. Other results have confirmed that for burners produce slightly less energy flow for same inlet pressure for hydrogen blends because of lower Wobbe index which can be accommodated easily by temperature control unless the burner is permanently running at full rate.

This section illustrates types of changes in heat transfer that occur if shifting from natural gas to hydrogen blends and for 100% hydrogen, using two examples. The first example has a load temperature in the low to mid temperature range (100º- 400 º C) with convection dominant by using a geometry that creates a velocity of approximately 25 m/s. The second example has a combination of radiation and convection at a higher temperature. The aim is to assess changes in heat transfer with hydrogen blends, so rather than using detailed computational modelling, the method chosen is to perform calculations of changes in key process parameters. These include changes in flue gas water vapour and carbon dioxide concentration, flue gas specific heat, density, viscosity and velocity which affect radiation and convection heat transfer.

6.1 Convection dominated heat transfer

The baseline case has a fuel input 2500 MJ/h (700 kW) with excess air of 30%, with post-flame gases flowing in 200mm steel tube. The hot gases have a temperature of 600ºC and transfer heat to the tube wall which has a temperature of 100ºC. Heat is assumed to be transferred to a liquid on the other side of the tube wall, which has a higher heat transfer coefficient than on the inside, so the gas side (inside) limits the overall heat transfer. The flue gas velocity in the tube varies with temperature, being approximately 26 m/s at 600ºC and 23 m/s at 500ºC. At these velocities and temperatures convection dominates and radiation heat transfer is negligible.

The convective heat transfer coefficient, h, is calculated using the Dittus-Boelter correlation for cooling of the fluid (as is the case here), and with fluid properties calculated at the bulk temperature (i.e. the average gas temperature):

$$Nu = hD/k = 0.023Re^{0.8} Pr^{0.3} \quad (4)$$

and can then be used to calculate the convection heat transfer:

$$Q_{conv} = hA_w(T_g - T_w) \quad (5)$$

where, A_w = Area of wall, T_g = gas temperature (ºC), and T_w = wall temperature (ºC).

The influence of fuel composition on heat transfer was calculated and results are presented in

Table 13. The first case is for the flue gas from methane combustion at 30% excess air, with (Tg-Tw) equal to 500ºC. The calculated heat transfer is just under 20 kW/m^2. For the blend cases, the addition of hydrogen, at a constant 30% excess air results in a reduction in flowrate of air which can be seen as a gradual reduction in velocity and heat transfer coefficient. The initial temperature of the hot combustion gases would be higher due to its lower mass flow, and this is represented as an increase in (Tg-Tw), which balances the reduction in heat transfer coefficient to produce very similar convection heat transfer rates for the blends. (Note that this is indicative of differences between the cases, because in all cases the gas temperature will be reducing along the tube length as heat is transferred to the wall.)

Table 13 Calculated convection heat transfer for flue gases in a 200mm tube at equal excess air

	Velocity at 500°C	Convection heat transfer coeff, h	Temperature difference, Tg-Tw	Convection heat transfer
	m/s	W/m^2K	°C	W/m^2
methane	23.4	39.5	500	19744
10%H_2	23.4	39.3	502	19737
20%H_2	23.3	39.2	504	19730
50%H_2	22.9	38.5	512	19698
70%H_2	22.4	37.6	523	19665
90%H_2	21.6	36.2	542	19612
100%H_2	20.9	34.9	561	19573

If the heat input rate and the air rate is kept constant as hydrogen is blended in, the excess air increases. Table 14 shows that the heat transfer drops marginally for 20 vol% hydrogen and 3% for 100% hydrogen. This slight drop is due to higher flue gas specific heat due to the increased moisture content, which has the effect of reducing gas temperature.

Table 14 Calculated convection heat transfer for flue gases in a 200mm tube at equal air flow rate

	Excess air	Velocity at 500°C	Convection heat transfer coeff, h	Temperature difference, Tg-Tw	Convection heat transfer
	%	m/s	W/m^2K	°C	W/m^2
methane	30	23.4	39.5	500	19744
10%H_2	32.5	23.8	39.9	494	19688
20%H_2	35.5	24.2	40.3	486	19625
50%H_2	44	25.1	41.4	470	19447
70%H_2	49.7	25.5	41.6	465	19330
90%H_2	55.4	25.3	41.0	468	19213
100%H_2	58.2	24.8	40.0	478	19157

As has been noted, many processes are controlled to a set process temperature. In these examples, (Tg-Tw) varies, which can be taken to mean that Tg varies at constant wall temperature. Tg increases with addition of hydrogen to the blend at constant excess air or decreases at equal air flow rate. In either case there is a slight

reduction in heat transfer for 100% hydrogen compared to natural gas, so fuel heat input rate for 100% hydrogen would need to be marginally higher in practise.

6.2 Radiation dominated heat transfer

In some processes requiring high levels of radiation, natural gas flames are employed with delayed mixing with air to generate highly luminous flames from soot formation. 100% hydrogen will not be able to replace these luminous flames, but blends with up to 30 vol% hydrogen can. However, most Type B applications use rapid mixing and produce short blue non-luminous flames that are much smaller than the volume of the enclosure. In these situations, most of the radiation comes from the CO_2 and H_2O in the hot post-combustion gases. It is these types of applications that are considered here, since all blends and 100% hydrogen should be able to replace natural gas in these cases, but the differences are of interest. Gas emissivity and absorptivity rely on the concentration of CO_2 and H_2O and so will change as hydrogen is blended into natural gas.

Many types of burners could be used in such applications, for example, package burners, high velocity burners, nozzle mix burners, and some premixed burners. Radiation from the flame itself only accounts for part of the radiation in furnaces with the rest coming from. These calculations only consider the portion of radiation from post flame gases, not the flame itself, and apply to all these situations.

The gas species radiate to a cooler surface, Q_{rad}, and absorb radiation from the surface, Q_{abs}, so there is a balance between them with a net radiative heat transfer, Q_{net}.

$$Q_{rad} / A_w = \varepsilon_g \sigma T_g^4 \quad (6)$$

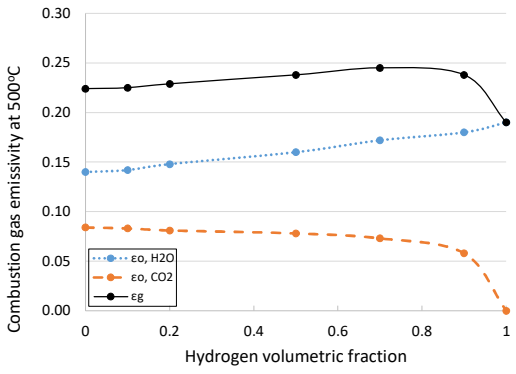
$$Q_{abs} / A_w = \alpha_g \sigma T_w^4 \quad (7)$$

$$Q_{net} / A_w = \varepsilon_g \sigma T_g^4 - \alpha_g \sigma T_w^4 \quad (8)$$

where:

- A_w = Area of wall,
- ε_g = mean emissivity of gas
- σ = Stefan-Boltzmann constant
- T_g =gas temperature (K)
- α_g =absorptivity of the gas for radiation from the wall
- T_w =wall temperature (K)

The calculations were initially done for methane / hydrogen blends fired in the experimental 600mm diameter, 1.0m long enclosure discussed in Section 5. For a relatively low mean gas temperature of 500°C and a mean wall temperature of 400°C Figure 74 shows the gas emissivity and net radiative flux (W/m2). (Absorptivity is not shown for brevity.)



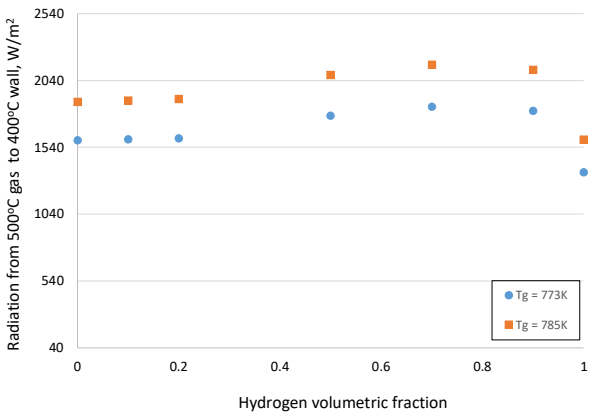


Figure 74 Calculated gas emissivity and net radiative flux for methane / hydrogen blends in a 600mm dia, 1.0m long enclosure with gas temperature of 500°C and wall temperature of 400°C

The total gas emissivity and radiation exchange both increase slightly as hydrogen is added to methane up to 70 vol% hydrogen, before dropping between 90 and 100 vol% due to the loss of CO_2 from the gas. However, a gas temperature increase of just 12 °C from 500°C and 512°C, as could occur from a small increase in fuel rate would be enough to increase the heat transfer from 100% hydrogen to match the original from methane.

For a higher gas temperature of 1500K, Figure 75 shows similar trends with much higher radiative heat transfer due to the higher gas temperature. Again, there is a drop in radiation from the 90 vol% blend to 100% hydrogen, which in this case could be cancelled out by increasing gas temperature to 1540K. However, since convection will also play a role in practical applications, the difference in total heat transfer between 90 vol% and 100% hydrogen will be less than shown. These results also show that blends of 90vol% hydrogen with 10% biomethane may be a possible renewable option where radiation is important.

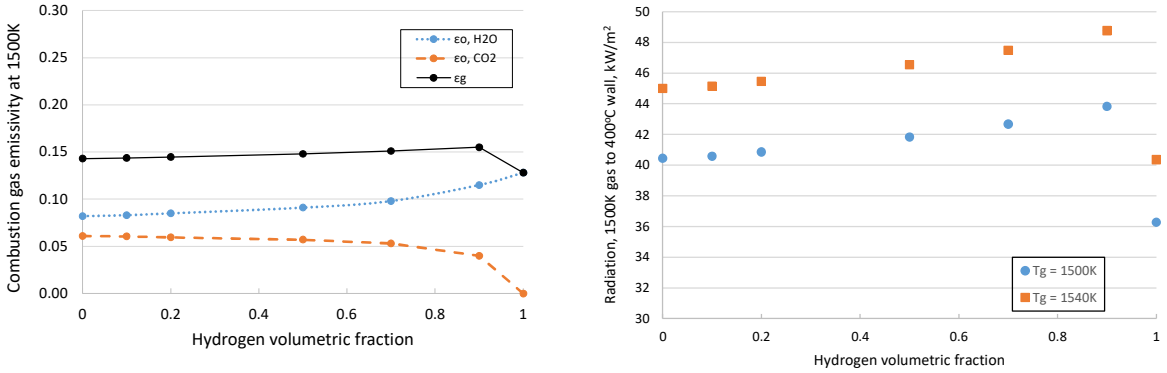


Figure 75 Calculated gas emissivity and net radiative flux for methane / hydrogen blends in a 600mm dia, 1.0m long enclosure with gas temperature of 1500K and wall temperature of 400°C

6.3 Comparison with experimental results

As mentioned earlier, the experimental heat flux profiles (e.g. Figure 68 and Figure 69) show that heat transfer was non-uniform. If radiation from hot gases to the walls dominated heat transfer would have been more uniform, which suggests convection heat transfer was mainly responsible for the peak near exit end, probably due to flow impingement and recirculation. In the Milestone 6 report, it shown that for 60 kW flames, the average wall heat flux was 12 kW/m^2, but this peaked at approximately 25 kW/m^2 near the exit end.

The modelling examples showed that radiation with T_g = 1500K produces much more heat flux than the experiment, whilst radiation with T_g = 500°C produces much less. For convection the model example with 25 m/s and (T_g-T_w) = 500°C and produces approximately 20 kW/m^2, but the velocity is much higher than in the experiment, so this rate would need to be reduced significantly. These results illustrate that a mixed gas temperature inside the furnace of somewhere in the region of 1000°C would produce the experimental results. Further refinement was not attempted. Instead, it is of interest to note the trend that addition of hydrogen has only a small effect.

6.4 Implications for industrial operation

In processes using combustion for heat the objective is normally to control to a set temperature(s) which controls product quality. In continuous processes the aim is to maintain steady temperatures, whilst in batch processes there are normally temperature targets for different parts of the cycle to control product quality.

The laboratory tests used the enclosure wall to represent the product in an industrial process, and results were taken once steady temperatures were reached in each experiment. Heat transfer rates were determined directly from wall temperatures. The 60kW results show that the actual flow rate of air had the greatest effect on heat transfer and flame temperature and hence efficiency. The fuel composition had less of an effect. The implication

is that hydrogen blends and hydrogen can be used as a fuel instead of natural gas, because the heat transfer changes from natural gas are minor for hydrogen blends and relatively small for 100% hydrogen but the flowrate of both fuel and air will need to be adjusted slightly, which will often be accommodated by existing automatic temperature control.

If hydrogen blend flames are operated with same air rate as natural gas flames, the heat transfer performance will be similar and excess air will increase. Flames become shorter or smaller due to rapid combustion of hydrogen and the additional excess air, but the effects are small up to 30vol% hydrogen. Alternatively, if a hydrogen blend is used at the same excess air as natural gas, by reducing the air rate for the same heat input, the flames become hotter and heat transfer and efficiency may improve in some cases depending on furnace design. Potential overheating of parts would need to be monitored.

For > 30vol% hydrogen blends and 100% hydrogen, further minor adjustments to rates will be required, after establishing that the burners and flame detectors are suited for the duty. Heat transfer rates are similar enough to natural gas so changes to furnaces or heat transfer surfaces are unlikely to be required in most cases.

NOx emissions have not been modelled, because they are only monitored by larger industries under the jurisdiction of the EPA. However, since hydrogen flame may produce high temperatures, the standard thermal NOx reduction methods are applicable.

7 POTENTIAL BURNER MODIFICATIONS FOR VARYING FUEL COMPOSITION

This section provides a summary of potential burner modifications. For 0-30 vol% hydrogen blends, natural gas burners can be operated without change or with minor changes to operation. For > 30 vol% hydrogen, Table 15 summarises the lessons from this project for various burner types.

It is also worth noting that manufacturers have developed some 100% hydrogen burners already. For example, the following are listed as suitable for 100% hydrogen by one supplier:

- medium velocity burners with central gas, mixed with pressurized air, partially combusting in a refractory block with a larger outlet than “high velocity” burners
- oxy-fuel burners
- some duct burners can fire on either natural gas or hydrogen
- radiant tube burners

Table 15 Potential issues and solutions for various burner types

Burner (or equipment type)	**Hydrogen blend, vol%**	**Potential Issue**	**Likely Solution**
Ionisation rod flame detectors	60 - 100	Don’t detect flames above 85% hydrogen typically	Further research on flame rods, or use UV flame detectors
Naturally aspirated, partially premixed	30 - 50	Flashback and overheating	Partially or fully close air inlet up to 50% hydrogen. For greater than 50% need redesign.
Naturally aspirated in radiant mode, e.g. pizza oven	30 - 100	Radiation too low	Replace burner with suitable radiant surface burner or radiant tube burner or electric
Partially or fully premixed radiant surface stabilised burners	30 - 100	Flashback and overheating. Air and fuel ratios are dependent on design and materials such as metal mesh and ceramic.	Guidance from manufacturer or replace with a new suitable burner
Partially premixed burner or pilot burner with forced air	30 - 50	Flashback and overheating	For 30-50% hydrogen reduce air rate. For > 50% change to a non-premixed (nozzle mix) design.
Many burners: Nozzle mix, air-heat, high-velocity, ribbon, duct – any burner where metal is contact with flame	50 -100	Higher burner metal temperatures may lead to short life	Increase fuel and axial momentum, reduce air swirl, or reduce air rate through burner and add another air source into the enclosure Some duct and high velocity burners are already available for 100% hydrogen
Package burner operated in radiant mode on natural gas	30 - 100	Unusual application, but get less radiation as blend in more hydrogen.	Replace burner with suitable radiant surface burner or radiant tube burner or electric

Radiant tube burners and recuperative burners designed for natural gas	30 -100	Overheating	Retrofit may be possible to reduce fuel / air mixing rate or new 100% hydrogen options exist.
Diffusion burners with radiation from natural gas luminosity	30-100	Reduced radiation	Limit hydrogen to 30%
100% hydrogen burners	90-100	If there is low heat transfer with 100% hydrogen	Slightly higher heat transfer may be obtained with a blend of 10% biomethane and 90% hydrogen

8 IMPLICATIONS AND RECOMMENDATIONS FOR STANDARDS

Two relevant standards are AS3814:2018 and AS1375:2023 in which detailed requirements for control, flame detection, air requirements and other operational considerations are found. These standards already apply to blends of hydrogen and natural gas with up to 50vol% hydrogen and make reference to “any other fuel” which would include any blend with over 50 vol% hydrogen and 100% hydrogen. This section only considers one aspect of the relevant standards that was explored during the project.

Tests in this project have shown that for blends with up 30 vol% hydrogen, there is little difference in combustion performance and flame appearance from natural gas, which aligns with the fact that there are only minor changes in fuel properties. Hence, it is anticipated that many burners will require no changes, whilst others may require minor adjustments to air and fuel rates. Some other equipment such as meters will need tuning for blends up to 30 vol% hydrogen. (Similar results have been found for Type A appliances, with the conclusion that gas networks may supply blends with up to 20vol% hydrogen to allow a margin.) Flame detection using ionization rods is unaffected by up to 30 vol% blends and only becomes problematic above 60 vol%. UV flame detectors work well for high levels of hydrogen and 100% hydrogen. Current burner management systems have been used in experiments with blends and with 100% hydrogen, with changes to purge times as required, and have worked well.

Naturally aspirated burners may require adjustment to limit air entrainment for > 20 vol% hydrogen blends. Most other burners wont experience problems such as overheating unless the hydrogen content is well above 30 vol% but the concentration will depend on the burner and combustor design and operation. Consequently, for > 30 vol% blends burner air fuel channels can be retrofitted to have more axial momentum, or burners can be replaced with a burner designed for the fuel. Details will become available from suppliers of new burners if these applications arise. Suppliers are also providing control equipment and valves and fittings rated for 100% hydrogen, with special attention paid to leakage and materials.

One consideration with hydrogen blends and 100% hydrogen is the difference in starting a burner and avoiding explosive atmospheres, since the LEL of hydrogen is slightly lower than methane (4% vs 5%) and the stoichiometric air requirement is much lower on a volume basis. It is worth noting that during experiments with one of the high velocity burners in this project some difficulties were encountered when lighting the burner on natural gas at very high dilution rates, but no problems were observed with hydrogen at either lean or rich conditions due to its much lower ignition energy and high upper flammability limit which means it can be easily ignited after minimal mixing with air.

How does hydrogen influence the safety calculations for staring up burners? The critical time is defined in AS3814:2018, clause 1.4.25 as the time to build up a concentration of unignited gas that would cause an explosion equal to the permissible safe internal pressure of the appliance, if ignited. This is the ignition safety time. One of the options for the allowable flame establishment period in clause 3.2.2.3 is to be less than 70% of the critical time to provide a safety margin. Start gas calculations include specifying gas and air flow rates, so these can be set to ensure that the allowed ignition time is less than 70% of the critical time.

Appendix D of AS 1375:2023 has three possible limiting conditions for determining critical time of an unignited mixture within a combustion chamber. Case A applies unless precautions have been taken to allow for either case B or C. So, the first step is to establish which situation applies. Case A is for an unignited mixture of fuel/ air that fills the chamber without further mixing with air. This case is normally applied to atmospheric naturally aspirated burners that don’t have a fan or external air supply. Case B is for forced draft burners in which the air and fuel are introduced into a chamber at rates such that the fuel concentration will at some stage reach 50% of LEL for natural gas or blends with up to 50% hydrogen, or 25% of LEL for H2 (one of the any other gases). The time take to reach that concentration is the critical time. Case C is like Case B but applies to higher air / fuel ratios so that the fuel concentration is always less than either 50% or 25% of LEL, depending on which applies, and so the critical time is taken to be infinity.

For Case A, Appendix D.3(b) of AS 1375:2023, provides an equation to calculate the critical time, T, that already applies to hydrogen and blends with natural gas.

$$T = 5PpVc/I \quad (1)$$

Where:

- Pp = the permissible safe internal pressure of an appliance
- Vc= appliance volume combustion, m^3
- I = heat input, W.

For example, a $5m^3$ volume with Pp = 7000 Pa, based on Clause E4.2 of AS 1375 App. E note 1(b), and heat input of 35 kW would have a critical time of 5 seconds.

For the thin-walled experimental enclosures used in this project, using forced draft package burners or high velocity burners, Case B or Case C apply. Type B approval was gained using Case B calculations, which can be found in the appendix of the Milestone 6 report. In these calculations the permissible safe internal pressure of an appliance was estimated at 7 kPa, based on Clause E4.2 of AS 1375 App. E. The critical time, purge time and start gas rate calculations were made for both natural gas and hydrogen. For example, the critical time for natural gas in the package burner enclosure, with a volume of 0.327 m^3, a fuel rate of 0.001 m^3/s and an air rate of 0.0183 m^3/s was 11 seconds. This was suitable since the flame establishment timing could be set for 5 seconds, after which the fuel was cut off if no flame was detected. For 10vol% or 20 vol% hydrogen blends (with the same volumetric flow rates) the critical times reduce slightly to 10.6 and 10.3 seconds respectively. However, for 100% hydrogen using the same fuel and air volumetric rates, the initial Case B calculation gave a critical time of 3 seconds. Consequently, the firing rate for starting up on hydrogen was reduced to provide a calculated critical time greater than 5 seconds.

In the package burner experiment, the fuel rate was measured, but the air was supplied by the package burner fan and not measured, so flue oxygen concentrations were used and in fact, Case C could have been applied. In practice, fuel and air flowrates are often not measured and approval and commissioning of many natural gas Type B appliances in which Case C applies is done with start gas and air settings to ensure the flue O2% concentration is greater than 16.2% as specified in Table 3.1 of AS 3814:2018. This ensures that the fuel concentration is always below 50% of LEL. Table 3.1 of AS 3814:2018 also shows the values of excess air and corresponding percentages of stoichiometric fuel and the minimum flue concentrations of oxygen and CO_2 that correspond to the same dilution rate. The same data is given for some other common hydrocarbon fuels but not hydrogen blends and hydrogen. Using the existing criteria, the values can be recalculated for hydrogen / methane blends and for 100% hydrogen as shown in Table 16. The results show that if commissioning is being done with a 10vol% or 20 vol% hydrogen blend, the required flue concentrations of oxygen are noticeably higher than for natural gas, at 16.7 vol% and 17.1 vol%. The new values could be included in an update of Table 3.1 of AS 3814.

Table 16 Values to meet clause 3.2.3a of AS3814:2018

Type of fuel	allowable % of LEL	fuel volume fraction	Excess air	% of stoich fuel	vol% O2 in dry flue	vol % CO2 in dry flue
methane	50%	0.025	309%	24%	16.2%	2.63%
10% H2	50%	0.025	351%	22%	16.7%	2.32%
20% H2	50%	0.024	401%	20%	17.1%	2.01%
50% H2	50%	0.023	628%	14%	18.4%	1.17%
51% H2	25%	0.011	1334%	7%	19.7%	0.59%
90% H2	25%	0.010	3012%	3%	20.4%	0.10%
100% H2	25%	0.010	4048%	2%	20.5%	0.00%
100% H2	50%	0.020	1953%	4.9%	20.1%	0.00%

If blends with greater than 20 vol% hydrogen are used, Table 16 shows that much higher flue oxygen concentrations are required to meet the safety criteria because of hydrogen's low stoichiometric air requirement. For > 50 vol% hydrogen fuels, the 25% of LEL criteria results in further increases in required flue oxygen concentration. Despite this, experimental experience has been that high excess air hydrogen flames are easy to ignite and very stable. For 90 and 100% hydrogen flames, a minimum of 20.4% and 20.5% oxygen in the dry flue gas is required respectively, which may be difficult to measure accurately and distinguish from fresh air.

It is recommended that Standards committees could consider the accuracy of flue oxygen meters and if there is confidence that 20.5% O_2 can be reliably measured in practice. If not, consider whether it is possible to relax the criteria for 100% H_2 to 50% of LEL. If that were the case the last row of Table 16 applies and the oxygen level needs to be kept above 20.1%. There may be some benefit in extending Table 3.1 of AS3814, and in allowing the 50% of LEL criteria for Case C start gas to blends with greater than 50% hydrogen and for 100% hydrogen so that there is a measurable difference in flue O2% from air. Another option is to use air and fuel flow metering, but many applications do not have this capability.

Another recommendation is to investigate the scientific reasons that flame ionization does not work for hydrogen and if there is solution to this, other than using UV flame detectors.

Notwithstanding the recommendations above, this project did not assess all revisions that may be appropriate for AS 3814 to be fully applicable to 100% hydrogen appliances. For example, flame ionization detectors will not be suitable for hydrogen appliances in their current form. Another example is conformance issues with AS 3814 (in some cases it mandates that components must conform to specific standards that are not applicable to hydrogen).

9 CONCLUSIONS

Open firing of a new Thermjet medium velocity burner at near stoichiometric conditions showed similar flame appearance for natural gas, 20vol% hydrogen and 50 vol% hydrogen blends. The burner operated well with 100% hydrogen but with approximately 200ºC higher metal temperature, which may become an issue in some environments.

A demonstration of use of 30 vol% hydrogen blend was conducted in an industrial pyrolysis furnace with temperature control at 400ºC, using a two stage package burner. A 30 vol% hydrogen blend operated at the same inlet pressure as natural gas resulted in 7 – 8% less heat input due to the reduction in Wobbe Index. A second experiment showed that operating at the same heat input produced similar periods on high and low rate. In all cases temperature uniformity throughout the furnace was good independent of the fuel in use.

Further analysis of previous experiments with a package burner with natural gas, blends and 100% hydrogen, in a thin walled enclosure was presented. Heat transfer efficiency was approximately 50% and reduced with increasing air rate, as did flame temperature. The air rate had a larger effect than fuel composition on heat transfer.

The heat transfer inside the enclosure was a combination of radiation and convection due to impingement and recirculation. Calculations were used to show how both radiation and convection can contribute. For industrial applications, the calculations show that there is very little change in heat transfer in the 0 - 30vol% hydrogen blending range, and any change can be accommodated by temperature control. For 100% hydrogen, there may be a drop radiation heat transfer in processes where it plays a role along with convection, but can be accommodated by a slight increase in fuel rate.

A section on potential burner modifications concentrated on the > 30 vol% hydrogen range because for 0-30 vol% hydrogen blends, natural gas burners can be operated without change or with minor changes to operation.

A comprehensive review of applicable Standards was not conducted, but the project did carefully consider start gas requirements and shows that measuring oxygen in the flue during start up on hydrogen may be difficult with the current procedure. Hence it is recommended to check that there is confidence in measuring 20.5% O_2 (dry) in flue gas and if not, consider whether it is possible to change the criteria for % LEL in Table 3.1 of AS3814:2018. Another recommendation is to investigate the scientific reasons that flame ionization does not work for hydrogen and if there is solution to this, other than using UV flame detectors.

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