Ballot notes

556 will change from a single document to an eight-part document. The subtitles for the eight parts of the proposed eight parts are listed below. The first six parts are dedicated to fired heaters designed to burn fuel gas. Part 7 is an overlay of Parts 1 through 6 for fired heaters designed to burn fuel oil. Part 8 is an overlay of Parts 1 through 6 for steam methane reforming furnaces.

Part 1 – Instrumentation
Part 2 – Control
Part 3 – Protective Functions
Part 4 – Flue Gas Analyzers
Part 5 – Main Burner Ignition Criteria
Part 6 – Tube Skin Thermocouples
Part 7 – Overlay for Oil Fired Heaters (planned publication in March 2028)
Part 8 – Overlay for Steam Methane Reforming Furnaces (planned publication in March 2028)

This "Comment-Only" ballot is on the proposed API Std 556-5, 1st Edition, which contains requirements and background information on main burner ignition criteria for fired heaters:

a. designed according to API 560 5th edition including addenda 1 and 2,
b. containing burners that are designed and tested as per API 535 4th edition,
c. have instrumentation in accordance with API 556 Part 1,
d. have controls in accordance with API 556 Part 2, and
e. have protective functions as specified in API 556 Part 3.

Instrumentation, Control, and Protective Systems for Gas Fired Heaters – Part 5: Main Burner Ignition Criteria

API Standard 556-5
1st Edition

Ballot Draft

Contents

0 Introduction

Second edition of API RP 556 allowed for a fuel accumulation up to 25% of the lower flammability limit in the firebox. For fired heaters with forced draft fans or induced draft fans, this can be an acceptable criterion as the firebox approaches a will-stirred system to a reasonable extent. In case of a natural draft heater however, the firebox does not behave as a well-stirred system at all.

API 556 commissioned a working group to study fuel accumulation and dispersion in a firebox during the trial-for-ignition period and formulate burner ignition criteria to assure smooth burner ignition. A brief report on the work done is added to Annex A of API 556-5. The main part of API 556-5 specifies requirements for main burner ignition.

Users of this Standard should be aware that further or differing requirements may be needed for individual applications. This Standard is not intended to inhibit a supplier from offering, or the purchaser from accepting, alternative equipment or engineering solutions for the individual application. This may be particularly applicable where there is innovative or developing technology. Where an alternative is offered, the supplier should identify any variations from this standard and provide details.

In API Standards, the SI system of units is used. In this standard, where practical, US Customary (USC) units are included in brackets for information.

A bullet (●) at the beginning of a clause or sub-clause indicates that either a decision is required, or further information is to be provided by the purchaser. This information should be indicated on the purchaser's checklist (see Annex B) or stated in the inquiry or purchase order.

1 Scope

API 556-5 contains requirements, recommended practices, and background information on main burner ignition for new gas Fired Heaters designed, operated, and safeguarded in accordance with API standards.

API 556-5 focusses solely on the main burner ignition procedure and assumes that:

- The fired heater has been built to the normative API standards as per section 2.
- Only conventional gas fuels are fired, being a mixture of gaseous hydrocarbons with hydrogen, a small amount of inerts, and traces [ppm level] of sulfur and nitrogen bearing components.
- The fired heater has been prepared for main burner ignition as stipulated in API 556-2, e.g. firebox is adequately purged prior to introducing ignition sources in the firebox, all start-up permissives for main burner ignition are satisfied, pilot burners (if fitted) are in service, etc.

2 Normative References

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

API Standard 535, *Burners for Fired Heaters in General Refinery Services*

API Standard 560, *Fired Heaters for General Refinery Service*

3 Terms, Definitions, and Abbreviations

3.1 For the purposes of this document, the following terms and definitions apply.

3.1.1
air factor
The ratio of air flow supplied to a burner over the stoichiometric air demand for the fuel that is supplied to the burner.

3.1.2
dilution criterion
A method for determining the minimum air factor to the first burner at start-up such that the burner ignites without a significant pressure rise.

3.1.3
stoichiometric air
The chemically correct amount of air required for complete combustion (meaning that all the fuel molecules are oxidized to CO_2 and water) with the resulting products containing no unused fuel or air.

3.1.4
reduced energy

The amount of fuel energy that can contribute to a deflagration in case of delayed ignition and is a fraction of the total energy density.
EXAMPLE: if the reduced energy is q% of the total energy density, then a delayed ignition of all the fuel in the firebox (= total energy density) can generate the same deflagration pressure as in an experiment where q% of the total energy density is concentrated in a stoichiometric air/fuel mixture balloon which is surrounded by still air in a larger volume of 1 m³.

3.1.5
trial-for-ignition interval
The maximum time interval in which fuel is permitted to flow into the firebox during an ignition attempt before the presence of flame is confirmed (visually or electronically).

3.1.6
total energy density
The total amount of fuel chemical energy [kJ or BTU] that has entered the firebox during the burner ignition attempt, divided by the total firebox volume [m³ of ft³].

3.2 Abbreviations

For the purposes of this document, the following abbreviations and symbols apply.

Symbol	*Meaning*
AF_{ST}	Stoichiometric air-to-fuel ratio on a volumetric basis. The ratio of moles of air needed for stoichiometric combustion of 1 mole of fuel.
E_{tot}	Total energy contained in the enclosure stated as fuel lower heating value
LFL	Lower Flammability Limit. Lowest volumetric concentration of fuel in air (at a specified temperature) that can be ignited.
LHV	Lower Heating Value of fuel [MJ/Nm³] or [BTU/scf]
n	Number of burners in the firebox
p	Pressure [mbar] or [psi] or [mmH2O] or [inH2O]
$P_{ignition}$	Burner ignition rate [kW] or [MMBTU/h]
Δp_{peak}	Peak pressure registered in a transient experiment [mbar] or [psi]
S_T	Lower volumetric heating value of the fuel at temperature T
S_{T0}	Lower volumetric heating value of the fuel at reference temperature of 0 °C - so equal to energy per standard volume
S_{fuel}	Lower heating value of the fuel

SG	Specific gravity - the ratio of the molecular mass of the fuel to molecular mass of air
T	Temperature [°C] or [ºF]
T_{FB}	Firebox temperature [°C] or [ºF]
TFI	Trial-for-ignition time [s]: elapsed time after opening the fuel valve and deciding that the burner ignition attempt is unsuccessful
t	Time [s]
t_{cd}	Critical delay time [s] being the time at which the fuel concentration reached the critical concentration
v_i	Volume of sub cell “i” in CFD simulation
$\dot{V}_{air}$	Volumetric air flow [m³/s] or [ft³/s]
V_{encl}	Volume of the enclosure [m³] or [ft³]
$\dot{V}_{fuel}$	Volumetric fuel flow [m³/s] or [ft³/s]
V_{fuel}	Partial volume of fuel inside the closure [m³] or [ft³]
V_{FB}	Firebox volume [m³] or [ft³]
W	Wobbe index of the fuel expressed on a (normalized) volumetric basis and lower heating value [MJ/Nm³]: $W = \frac{S_{fuel,Nm^3}}{\sqrt{SG_{fuel}}}$
x	Volumetric concentration as fraction of the total volume that is occupied by a specific component
x_{LFL}	Volumetric concentration at lower flammability limits

Symbol	*Meaning*
ξ	Volumetric energy density [kW/m³] or [Btu/h-ft³] Amount of fuel energy (as lower heating value) is available in the volume, filled with fuel and air (or flue gas).
$\xi_{t=\infty}$	Volumetric energy density after an infinite time

λ	Air factor - the ratio of the supplied amount of air over the stoichiometric air requirement for the fuel $\lambda = \frac{\dot{V}_{air}}{\dot{V}_{Fuel} \cdot AF_{ST}}$
χ	The ratio of actual volumetric fuel concentration to the fuel lower flammability volumetric concentration $\chi = \frac{x}{x_{LFL}}$
χ_∞	The ratio of actual volumetric fuel concentration to the fuel lower flammability volumetric concentration, based on the supplied air flow and fuel flow to the burners. In well-stirred-reactor terminology, this will be the fuel concentration in the firebox after an infinite time of fuel and air flow.

4 Main Burner Ignition Requirements

4.1 Main burners shall be designed and tested in accordance with API 535.

4.2 Fired heaters shall be designed in accordance with API 560.

4.3 If the purchaser specified an automated burner ignition sequence, the minimum air flow to all burners during main burner ignition shall be determined using equation 1 or 2 (depending on the units of measure).

<table>
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$$\dot{V}_{Air} \geq \dot{V}_{fuel} \cdot MAX\left(1.2 \cdot n \cdot AF_{ST},\ \frac{n}{MIN(4,n)} \cdot \left(\frac{0.185 \cdot P_{ignition}}{x_{LFL} \cdot V_{FB}} - 1\right)\right)$$

with:

- $\dot{V}_{air}$ the volumetric air flow to all burners during burner ignition - in the same unit as $\dot{V}_{fuel}$
- $\dot{V}_{fuel}$ the volumetric fuel flow to one burner during burner ignition, in the same unit as $\dot{V}_{air}$
- *n* the number of main burners in the firebox
- $P_{ignition}$ the burner ignition heat release (kW). Use the burner heat release at design fuel pressure during burner ignition (typically just above the low fuel pressure trip set point).
- AF_{ST} the stoichiometric air-to-fuel ratio on a volumetric basis (m³ air/m³fuel)
- V_{FB} the firebox volume (m³)
- x_{LFL} the fuel lower flammability limit in air, taking into consideration the fuel composition and the expected (design) air and fuel temperature during burner ignition. Unitless (i.e. number between 0 and 1 for example the lower flammability limit for methane is 0.05).

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Eq. 1

Calculation in SI-units

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$$\dot{V}_{Air} \geq \dot{V}_{fuel} \cdot MAX\left(1.2 \cdot n \cdot AF_{ST},\ \frac{n}{MIN(4,n)} \cdot \left(\frac{1916.6 \cdot P_{ignition}}{x_{LFL} \cdot V_{FB}} - 1\right)\right)$$

with:

- $\dot{V}_{air}$ the volumetric air flow to all burners during burner ignition - in the same unit as $\dot{V}_{fuel}$
- $\dot{V}_{fuel}$ the volumetric fuel flow to one burner during burner ignition, in the same unit as $\dot{V}_{air}$
- *n* the number of main burners in the firebox
- $P_{ignition}$ the burner ignition heat release (MMBTU/h). Use the burner heat release at design fuel pressure during burner ignition (typically just above the low fuel pressure trip set point).
- AF_{ST} the stoichiometric air-to-fuel ratio on a volumetric basis (ft³ air/ft³ fuel)
- V_{FB} the firebox volume (ft³)
- x_{LFL} the fuel lower flammability limit in air, taking into consideration the fuel composition and the expected (design) air and fuel temperature during burner ignition. Unitless (i.e. number between 0 and 1 for example the lower flammability limit for methane is 0.05).

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Eq. 2

Calculation in USC-units

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Note 1: Equations 1 and 2 incorporate a safety factor of 2 on the expected pressure peak after delayed ignition. See Annex A.

Note 2: Equations 1 and 2 are valid for 5 seconds trial-for-ignition time and assume 50 mbar maximum pressure rise after delayed ignition. See Annex A

Note 3: Operators can manually ignite burners with lower fuel flows than what can typically be achieved by a control system. See Annex C.

Note 4: For natural draft heaters, air flow can be inferred from available draft in the firebox, in conjunction with field operator confirming all air registers are open and confirming air is flowing into the burners. Feeling the air flow by the draft that is created when holding a hand, glove, or paper in front of an air register is sufficient.

Note 5: See Annex D for examples and calculation sheets.

4.4 For the application of 4.3, the fuel composition with the highest expected hydrogen content during burner ignition shall be used.

4.5 Before main burner ignition of natural draft or induced draft heaters (without air flow measurement), a firebox pressure of -1.27 mmH_2O (-0.05 inH_2O) shall be established.

Note 1: See API 556 Part 2 for options to establish draft in natural draft heaters.

Note 2: Although the minimum required air for burner ignition (as calculated in4.3) can be achieved with a firebox pressure closer to 0 than -1.27 mmH2O (-0.05 inH2O), draft instruments will not be able to reliably indicate the actual pressure lower than this minimum pressure.

Note 3: For natural draft and induced draft heaters, the air flow as calculated in 4.3 can be translated to a minimum required draft as shown in Annex D.

ANNEX A

(informative)

Historical work on burner ignition and potential firebox pressures from delayed ignition with comments on applicability to current fired heaters

A.1 Summary

During the 1960-1970's, the British Gas council conducted multiple experiments examining potential pressure spikes from delayed ignition during burner ignition. This work was the original foundation for different industrial combustion safety standards such as EN 746, AS-1375, ASA-standards, etc. The two key approaches considered in these papers were the total energy concept and the dilution criteria for safe ignition. This was later supplemented by an intermediate case. The conclusions from these experiments are summarized in this Annex and compared against API fired heaters and operating practices together with a CFD analysis. This CFD analysis focused on a single burner and a four-burner heater, designed according to API standards and following burner ignition practice as per API 556 Part 2. Results of this review, together with supplementary CFD work, indicate that:

- Except for the first seconds of fuel admittance, the total energy input criterion suggests a significantly higher peak pressure from delayed ignition of fuel than what actually occurs in a fired heater. This is because the total energy input criterion fails to acknowledge that fuel disperses into the firebox quickly and mixes with air below the fuel flammability limits. Air supply through adjacent burners can be accounted for in the dilution calculations.
- Fired heaters designed to API standards can typically handle internal pressures of 50 mbar (0.7 psi) or higher (depending on structural design) without any damage.
- Reduced energy accumulation slows down with higher dilution. This underpins the advantage of having either a forced air flow or an induced draft (with all burner air registers open) during burner ignition.
- API practices as laid out in API 560, API 535, and API 556 (Part 2) are sufficient to provide a safe ignition of burners without a need to perform more detailed assessments.

A.2 Introduction

API 556 second edition refers to a 25% of LFL fuel accumulation as a not-to-exceed fuel concentration in the firebox during ignition. This number is also stated in NFPA standards.

An inherent assumption in stating the maximum fuel concentration threshold is that the gas of the firebox is well-stirred, i.e. the fuel concentration inside the firebox is independent of the location in the firebox. When the burner ignition is performed under high draft and/or air flow into the firebox, or there are already burners commissioned, then the firebox behavior will be close to a well-stirred reactor.

During burner ignition under less-well-stirred conditions however, a flammable air/fuel mixture develops on the burner tips. The fuel concentration reduces with the distance from the burner tip as air gets entrained by the fuel jet. A fuel concentration gradient develops, creating a flammable mixture in proximity of the burner tips. The further away from the burner tip, the more air mixes with the fuel resulting ultimately in a fuel concentration dropping below the lower flammability limit.

The research paper from Aris, Hancock, and Moppet [1] presents a worst-case pressure rise after delayed ignition of the fuel quantity filling the firebox up to 25% of LFL, assumed all fuel being concentrated in a stoichiometric air/fuel mixture: a pressure of approximately 0.8-1.0 barg (12-14.5 psig) results for regular hydrocarbons as present in typical fuel gas. As a comparison, with pure hydrogen, the worst-case pressure rise of the amount of fuel equivalent to 25% of LFL would yield 0.2 barg (2.9 psig).

To further understand this topic, papers from the 1970's have been reviewed and have been supplemented by a CFD analysis to investigate the development of flammable air/fuel mixtures in a single burner and a four-burner firebox. The result of the paper review and the CFD analysis is documented in the next sections.

This document also reviews the structural strength of a fired heater since this defines a realistic expectation for a performance criterion for burner ignition. In section C.4, the work mainly from the British Gas Council is summarized and put in perspective for fired heaters that are designed according to API standards using the total energy and dilution criteria.

A.3 Structural Strength of a Firebox

As described in API 560, the structural design of fired heaters is typically governed by gravity, wind, snow, thermal, and seismic loads. If the heater is shipped in modules, then the loads during lifting and shipment will also be considered in the structural design.

Although the internal firebox pressure is not a default design specification, a fired heater structure as designed per API 560 will inherently have the capability to withstand internal pressures that occur during normal operation. Under normal operating conditions, the internal pressure in a firebox is in the order of -1 mbar (-0.014 psi or -0.40 inH_2O).

Discussions with fired heater engineering companies reveal that, depending on heater geometry and heater style, fireboxes can inherently withstand internal pressures in the order of 50-150 mbar (0.7-2.2 psi). Atkinson, Marshall, & Moppet, (Nov 1967) reported on experiments for paint drying box ovens that could withstand up to 300 mbar (4.4 psi) (considered in their paper as the weakest assets). However, to prevent minor damage, the pressure should be maintained below 70 mbar (1 psi). The Australian Standard AS 1375 [8] reports that fireboxes may suffer damage (depending on geometry and structural design) at 50-100 mbar internal pressure. AS 1375,2013 further notes that while pressures of 70 mbar may cause damage to the structure, these pressure events "generally unlikely would endanger personnel".

In conclusion, a fired heater in general refining service can typically handle an internal pressure of 50 mbar (0.7 psi) with no damage.

A.4 Historical work on burner ignition and potential firebox pressures from delayed ignition

During the 1960-1970's, several research institutes conducted experimental work to establish criteria for safe burner ignition. In particular, the work done at the British gas council [1, 4] was used in the development of safety standards for gas fired equipment. The experimental work focused on two extreme cases and one intermediate case:

1. Total energy input case: Determination of the pressure built-up when a balloon filled with a stoichiometric mixture of air and fuel ignites inside a larger volume which is filled with air. The larger volume has still air and an adjustable exit opening on one end.
2. Dilution case: Evaluation of pressure rise from delayed ignition of a burner, when the burner is under continuous (high) air flow in an enclosure that represents a firebox.
3. Intermediate case where the peak pressure from delayed ignition is studied for air/fuel mixtures that are within flammability limits and the actual development flammable region at the burner throat is considered.

A.4.1 Total Energy Input Case

The British Gas Council conducted a series of experiments and reported them in GC166 [1]. The experimental setup is shown in Figure 1. A balloon is filled with a stoichiometric mixture of fuel and air. The balloon sits inside a larger enclosure which is filled with still air. The opening at the left-hand side of the enclosure is variable in the experiment. The air/fuel mixture in the balloon is ignited by a spark and the pressure rise inside the larger volume (0.14 m^3 / 5 ft^3) is recorded by a high-speed pressure transducer (at the right-hand side of the figure). The highest recorded pressure peak Δp_{peak} is reported.

The main variables in the experiments were:

- Fuel composition
- Size of vent opening
- Vent opening replaced by different pipe sizes and lengths (mimicking a stack)
- Having a very dilute fuel-in-air surrounding the balloon (below the lower flammable limit)

A key parameter that was shown to be important in the results was the total energy density inside the enclosure. The name for this case is the "Total Energy Input" case. The total energy density ξ [kJ/m³] inside the enclosure is determined as follows:

- The energy content E_{tot} of all the fuel inside the balloon [kJ] or [Btu] (as lower heating value S_m [kJ/kg] or [Btu/lb] times the fuel quantity m_{fuel} [kg] or [lb] inside the balloon - heating value and fuel quantity can equally be used on a volumetric basis)
- Divided by the total volume of the enclosure V_{encl} [m³] or [ft³]
  - 0.14 m^3 or 5 ft^3 for the experiment
  - Or the total firebox volume when extrapolating to a fired heater case.

Written as an equation:

$$\xi = \frac{E_{tot}}{V_{encl}} = \frac{m_{fuel} \cdot S_m}{V_{encl}} = \frac{V_{fuel} \cdot S_V}{V_{encl}} \qquad \text{Eq. 3}$$

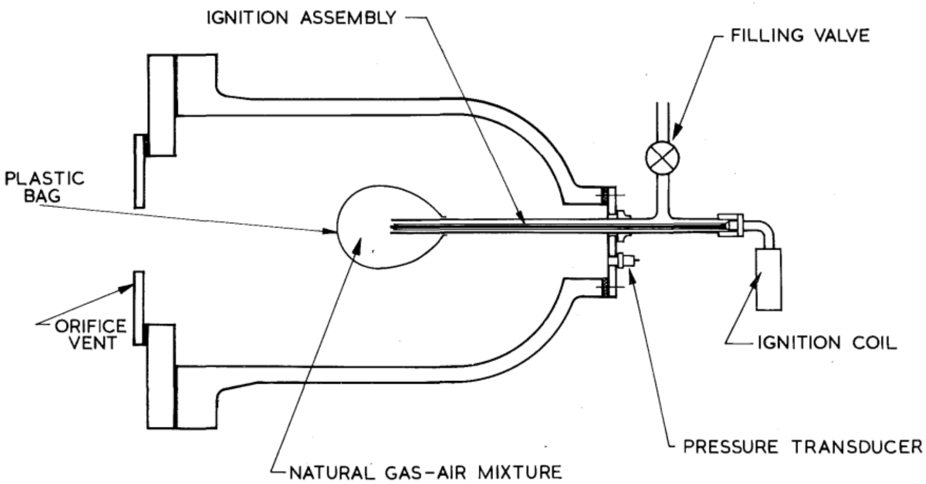


Figure 1: Basic experimental setup as used in [1]

Outcomes from these experiments are:

- When plotted against the total energy density of fuel ξ in the enclosure, the peak pressure is independent of the fuel composition.
- Up to an energy density of 186.3 kJ/m³ (5 BTU/ft³), the peak pressure from ignition in a closed volume is proportional to the total energy density: see Figure 2. The linear correlation is given by Eq. 4 and Eq. 5 (depending on the unit system used).

$$\Delta p_{peak}[mbar] = 1.85 \cdot \xi \left[\frac{kJ}{m^3}\right] \quad \text{Eq. 4}$$

$$\Delta p_{peak}[psi] = \xi \left[\frac{BTU}{ft^3}\right] \quad \text{Eq. 5}$$

The thermodynamical background explaining this proportional relation can be found in multiple sources, e.g. Atkinson, Marshall, & Moppet [2], AS 1375 [8].
Relating Eq. 4 with the finding that fired heaters in general refining service can handle pressures of 50 mbar without significant damage as noted in the previous section, shows that a reasonable threshold for the energy accumulation is 27 kJ/m³ (0.72 BTU/ft³). This assumes that the fuel occupies a small "pocket" of the firebox in perfect stoichiometric ratio with air. The fuel energy in that pocket corresponds to 27 kJ times the firebox volume.
Note that the distribution of the energy over the volume plays a key role in the actual pressure peak. The worst-case scenario, yielding the highest pressure peak, is when all the energy is concentrated in a stoichiometric mixture with air around the ignitor.

- Opening the enclosure on the left-hand side (i.e. using an orifice instead of closing off the enclosure) does reduce the peak pressure gradually with increasing opening, although only when the area of the vent opening exceeds 2 percent of the cross-sectional area of the enclosure. With less than 2 percent free venting area, the peak pressure is of the same magnitude as for a fully closed volume
- Adding a pipe on the vent area (mimicking a stack) increases the peak pressure compared to having no pipe at all. From a certain length of pipe (7.5-10 m / 25-33 ft in the experimental setup), the peak

pressure matched the closed enclosure peak pressure again and did not increase after adding even longer pipes.

- When the gas surrounding the balloon in the enclosure contains fuel in air below the LFL of that specific fuel, then the surrounding fuel does contribute to the peak pressure: see Figure 3. Noting that the LFL of methane in air is 5% on a volumetric basis, the contribution of gas below ~ 50% of LFL (2.5%(v) of fuel concentration on the abscissa) barely contributes to a pressure rise. However, the contribution to a pressure rise increases rapidly when 50% of LFL is exceeded.
  In the GC166 paper [1], the authors added an errata to the original article, stating that the fuel concentrations in the experiment (X-axis) where approximately 1.7x higher than shown on the chart. The x-axis on Figure 3 in this paper includes the correction. The results of the corrected Figure 3 were used for the CFD study to account for fuel below LFL in the firebox. See C.6.1.

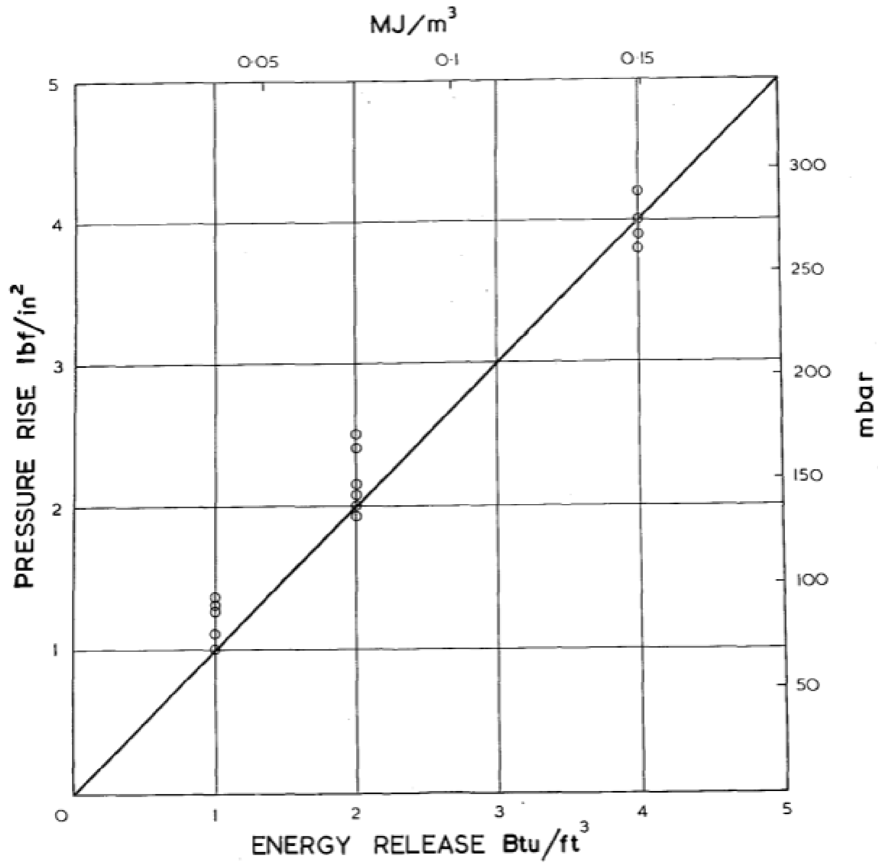


Figure 2: Pressure rise due to the explosion of pockets of stoichiometric air/gas mixture within a 5 ft³ closed steel vessel [1]

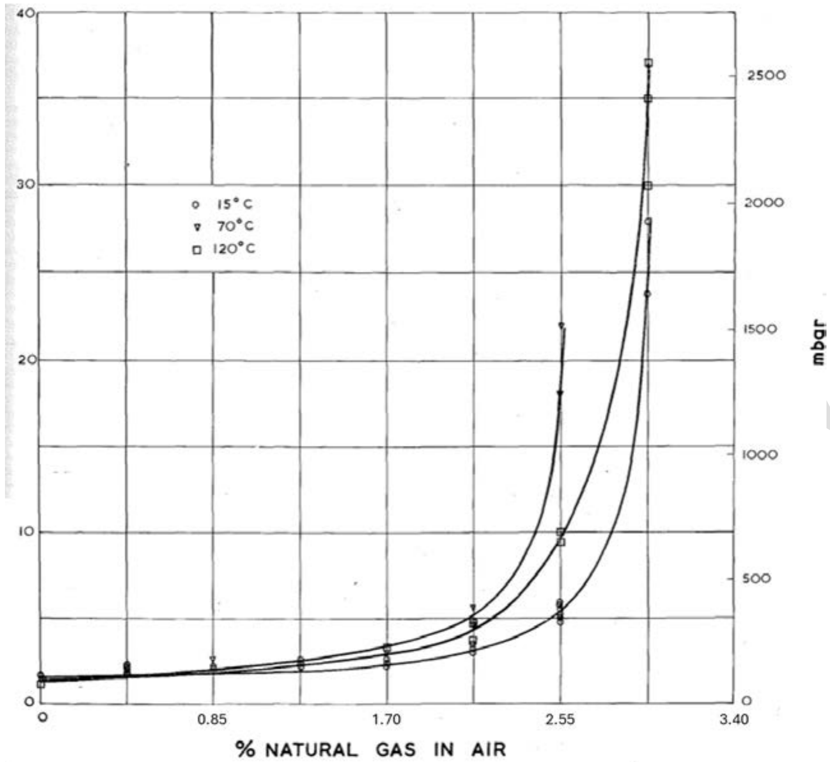


Figure 3: Effect of a below-flammability limits or air/fuel mixture surrounding the balloon. [1] Note that the chart above is corrected for the error mentioned in the original paper by the authors.

It will be shown in A.5 and A.6 that the total energy input criterion based on the total fuel supply divided by the firebox volume is not representative for an actual fired heater, with the exception of the first 2-5 seconds of burner ignition. The total energy input criterion used as such yields higher pressures than those that can occur in real situations.

A.4.2 Dilution Case

In the dilution case, experiments were conducted to determine the potential pressures from delayed ignition during burner ignition, with continuous air flow to the burner [3, 4]. Specific for this case is that the bulk fuel flow divided by bulk air flow results in a fuel concentration below the lower flammability limit of the fuel.

Figure *4* shows a typical result from the dilution case experiments that encompassed a virtually infinite trial-for-ignition, i.e. the air and fuel mixture was flowing to the burner for about three minutes before an ignition source was introduced. The air and fuel mixture inside the test enclosure evolved to a steady state. The sharp upturn in pressure occurs when the fuel concentration in the supplied air/fuel mixture exceeded the lower flammability limit. This concentration was called the “critical concentration”.

When an air/fuel mixture below the lower flammability limit is flowing to a burner, then the bulk concentration inside the enclosure will also be below the flammability limits. Only in the vicinity of the fuel tips through which the fuel is injected into the air stream, a region will develop where the mixture is within flammable limits. At ignition, the volume containing the flammable mixture ignites and contributes to a pressure increase.

With lower fuel-to-air ratios, the flammable region reduces in size as well as in energy content, resulting in a reduction in the pressure after delayed ignition.

Conversely, the higher the fuel concentration, the larger the flammable volume becomes and the more energy it contains. When the lower flammability limit is exceeded, the total volume of the enclosure is flammable and will contribute to a pressure increase.

Comparing the magnitude of pressure rise in Figure 4 with the typical structural strength of fireboxes (see C.3) reveals that fuel/air mixtures that are leaner than the critical concentration, can be easily contained with no damage. Once the critical concentration is exceeded, the pressure rise quickly increases up to the strength of fireboxes. From this experimental review it can be concluded that if the trial for ignition period is not restricted, the bulk fuel-to air mixture must be leaner than the fuel lower flammability limit to have an uneventful ignition. For higher fuel concentrations, limiting the time of fuel admittance during a burner ignition becomes imperative: see intermediate case.

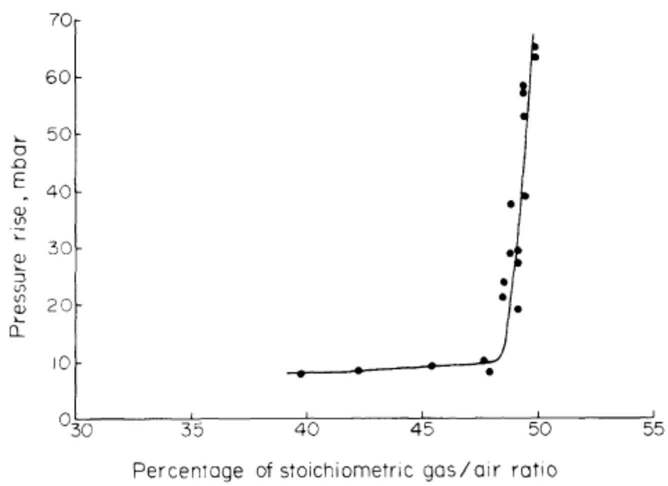


Figure 4: Effect of ratio of burner gas and air flows on pressure rise for natural gas *(Hancock , Spittle, & Ward, Nov 1976)*

From the curve shown in Figure 4 knowing that the experiment was done with natural gas (methane), the energy density in the bulk air to fuel mixture can be calculated:

$$\xi = \frac{E_{tot}}{V_{encl}} = \frac{\dot{V}_{fuel} \cdot S_{fuel}}{V_{encl}} = \frac{\dot{V}_{fuel}}{\dot{V}_{fuel} + \dot{V}_{air}} \cdot S_{fuel} \qquad \text{Eq. 6}$$

This is the asymptotic energy density that will be reached in the firebox after an infinite time, considering that the firebox as a well-stirred volume. In practical terms, 95% of this energy density is reached after three volume changes of the firebox and 99.3% is reached after five volume changes. When time further progresses, the fuel density cannot exceed ξ from Eq. 6.

The dilution criterion dictates that in order to keep the trial for ignition out of the equation, the bulk air-to-fuel mixture must have a fuel concentration below the critical concentration. This must be achieved for the actual ignition fuel concentration, air temperature, etc.

Assuming that the critical concentration is 90% of the lower flammability limit, based on combustion stoichiometry and using Le Chatelier to derive the lower flammability limit of an air/fuel mixture, it can be shown that this criterion for a methane/hydrogen mixture at ambient temperature results in a minimum air factor given by Eq. 7

$$\lambda_{min,dilution} = 2.33\ \frac{4.82 - x_{CH4}}{(1 + 3\ x_{CH4})} \qquad \text{Eq. 7}$$

Stating in terms of the hydrogen volumetric concentration:

$$\lambda_{min,dilution} = 2.333\ \frac{3.82 + x_{H2}}{(4 - 3\ x_{H2})} \qquad \text{Eq. 8}$$

Note: The CFD analysis (see C.6) reveal that 90% of LFL is not sufficient dilution to achieve a real infinite trial-for-ignition possibility. This is because sub-flammability mixtures still contribute to a pressure rise provided that a flammable pocket is also available (see Figure 3). A well-stirred reactor calculation ignores differences in concentration inside the "reactor" volume.

A.4.3 Intermediate case

In their 1976 paper, the researchers from the British Gas Midlands Research Station [4] acknowledge that the total energy input criterion and the dilution criterion "undoubtedly contributed to safety (…)". It was recognized that startup rates may be overly conservative. To allow greater flexibility without compromising safety, an additional piece of work called the intermediate case was developed.

The experiments for the intermediate case considered the impact of delayed ignition time in combination with the fuel-to-air ratio for bulk fuel-air concentrations exceeding the lower flammability limit of the fuel: see Figure 5.

The time dependency of a growing flammable pocket during the trial for ignition period is now correlated with the fuel to air bulk concentration during ignition in a single burner application. The peak pressure from delayed ignition increases rapidly with the ignition delay time. On the 55.1% of stoichiometry curve, a sharp increase occurs after seven seconds of ignition delay. In the original paper, the ignition delay at which the upturn in pressure occurs, was called the "critical delay time". When the ignition delay time is less than the critical delay time, the pressure increase was limited to approximately 20 mbar (0.3 psi). This is well below the internal pressure capability of practical fireboxes (see C.3).

For methane the lower flammability limit at ambient condition is 5.0%$_{vol}$ fuel, which is at 50% of the stoichiometric fuel to air ratio. The curve with 55.1% of stoichiometric gas-to-air ratio resembles a 109% of LFL-mixture. The combined conclusion from Figure *4* and Figure 5 is that as long as the bulk fuel in air mixture at ignition remains below the flammability limit (critical concentration), the pressure rise after a delayed ignition remains below 20 mbar irrespective of the ignition time delay. If the bulk fuel in air mixture exceeds the lower flammability limit of the fuel at actual temperature, it becomes imperative to limit the ignition delay time.

Caution is advised in translating this statement to practical situations: the theory and experiments may agree that under certain well-defined conditions, very long "trial-for-ignition times" cause no harm. In reality, several crucial variables are not well-defined nor controlled during burner ignition, e.g. the exact fuel composition, exact fuel supply quantity, exact timing of the trial-for-ignition, etc. Therefore, the default trial-for ignition is specified at 5 seconds even if the theory/experiments suggest that a longer trial-for-ignition time could be allowed for a certain situation.

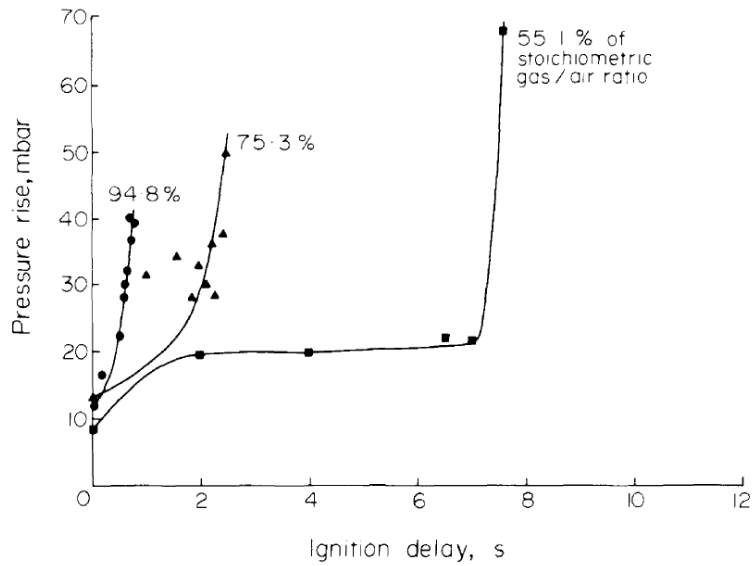


Figure 5: Effect of ignition delay on pressure rise [4]

A simplified mathematical model was developed to calculate the critical delay time for a single burner cylindrical firebox, assuming a well-stirred reactor model [4]. The relevant firebox volume to be considered for the model, was derived assuming an expanding round jet of air/fuel mixture coming from the burner with a half angle of 7°: see Figure 6. The maximum mixing length L_{mix} becomes:

$$L_{mix} = \frac{D-d}{2 \cdot \tan(7°)} = 4.072 \cdot (D-d) \qquad \text{Eq. 9}$$

With D the diameter of the firebox and d the throat diameter of the burner.

Resulting in a relevant mixing volume (for a cylindrical firebox) V_{FB} of (purple region Figure 6)

$$V_{FB} = \frac{\pi \cdot D^2}{4} \cdot L = \frac{\pi \cdot D^2}{4} \cdot 4.072 \cdot (D-d) = 3.198 \cdot D^2 \cdot (D-d) \qquad \text{Eq. 9}$$

When the firebox is shorter than the mixing length ($L_{FB} < L_{mix}$), the relevant volume is restricted to the firebox volume alone:

$$V_{FB} = \frac{\pi \cdot D^2}{4} \cdot Min(L_{mix}, L_{FB}) = Min\left[\frac{\pi \cdot D^2}{4} \cdot L_{FB}, 3.198 \cdot D^2 \cdot (D-d)\right] \qquad \text{Eq. 11 10}$$

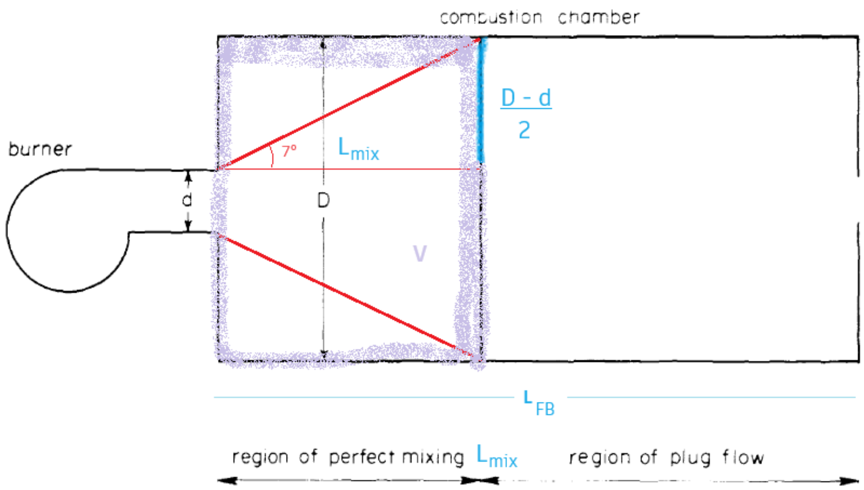


Figure 6: Simplified model to derive a relation between the critical delay time and the fuel-air bulk concentration at ignition for a single burner application [4]

Assuming a well-stirred reactor model for the volume V_{FB}, starting with a fuel-free volume, gives the following relationship between the time *t* [s] and the volumetric fuel concentration *x* [volume of fuel/total volume] inside the volume V_{FB}:

$$t = \frac{V_{FB}}{\dot{V}_{fuel} + \dot{V}_{air}} \cdot \ln\left(\frac{1}{1 - (1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}}) \cdot x}\right)$$

Eq. 12
11

which can be rearranged to:

$$x = \frac{1 - e^{-\frac{\dot{V}_{fuel} + \dot{V}_{air}}{V_{FB}} t}}{1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}}}$$

Eq. 13

With:

- $\dot{V}_{fuel}$: the fuel volumetric flow into the burner [m³/s] at actual flowing pressure and temperature
- $\dot{V}_{air}$: the air volumetric flow into the burner [m³/s] at actual flowing pressure and temperature
- *t*: the elapsed time after the start of air + fuel flowing into the burner [s]
- V_{FB}: the mixing volume [m³] as per Eq. 11.

The critical delay time t_{cd} coincides with a critical volumetric fuel concentration

$$x_{cc} = 0.97 \cdot x_{LFL} \qquad \text{Eq. 12}$$

With x_{LFL} being the lower flammability limit of the fuel "LFL" (LFL expressed in volumetric fraction):

$$t_{cd} = \frac{V_{FB}}{\dot{V}_{fuel} + \dot{V}_{air}} \cdot \ln\left(\frac{1}{1 - (1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}}) \cdot 0.97\, x_{LFL}}\right) \qquad \text{Eq. 13}$$

This can also be rearranged to introduce the parameters:

- AF_{ST} as the air-to-fuel stoichiometric volume ratio (i.e. m³ of air needed for a stoichiometric reaction per m³ of fuel)
- λ as the air factor such that the total actual air flow $\dot{V}_{air} = \lambda \cdot AF_{ST} \cdot \dot{V}_{fuel}$

To get:

$$t_{cd} = \frac{V_{FB}}{\dot{V}_{fuel} \cdot (1 + AF_{ST} \cdot \lambda)} \cdot \ln\left(\frac{1}{1 - (1 + AF_{ST} \cdot \lambda) \cdot 0.97\, x_{LFL}}\right) \qquad \text{Eq. 14}$$

This equation can be used irrespective of the fuel composition – provided that the proper AF_{ST} and x_{LFL} are determined based on the fuel composition. When the burner ignition is performed with hot air, the impact of temperature on the lower flammability limit shall be considered as well (see Supplement 1).

The relation between the fuel volumetric concentration *x* and the air factor, is:

$$\lambda = \frac{\dot{V}_{air}}{\dot{V}_{fuel} \cdot AF_{ST}} \qquad \text{Eq. 15}$$

and the volumetric fraction of fuel is given by

$$x = \frac{\dot{V}_{fuel}}{\dot{V}_{fuel} + \dot{V}_{air}} \qquad \text{Eq. 16}$$

such that

$$\lambda = \frac{1 - x}{x \cdot AF_{ST}} \qquad \text{Eq. 17}$$

The critical concentration x_{cc} is reached when the air factor equals the minimum air factor $\lambda_{min,dilution}$:

$$\lambda_{min,dilution} = \frac{1 - x_{cc}}{x_{cc} \cdot AF_{ST}} \qquad \text{Eq. 18}$$

so that Eq. 16 becomes:

$$t_{cd} = \frac{V_{FB}}{\dot{V}_{fuel} \cdot (1 + AF_{ST} \cdot \lambda)} \cdot \ln\left(\frac{1}{1 - \frac{1 + AF_{ST} \cdot \lambda}{1 + AF_{ST} \cdot \lambda_{min,dilution}}}\right) \qquad \text{Eq. 19}$$

From this equation, it can be concluded:

- If the air factor exceeds $\lambda_{min,dilution}$, then t_{cd} has no meaning (logarithm of a negative number). In this case, the dilution criterion is fulfilled meaning that even after an infinite time, the fuel cannot accumulate above the critical concentration - which is only valid for a perfectly stirred firebox.
- When the air factor equals $\lambda_{min,dilution}$, then t_{cd} is infinite (logarithm of infinity = infinity)
- When the air factor is less than $\lambda_{min,dilution}$, t_{cd} will have a defined value.

The researchers from the British gas council [4] validated the simplified model with experimental data. See Figure 7. The results from the theoretical model align with the experimental data.

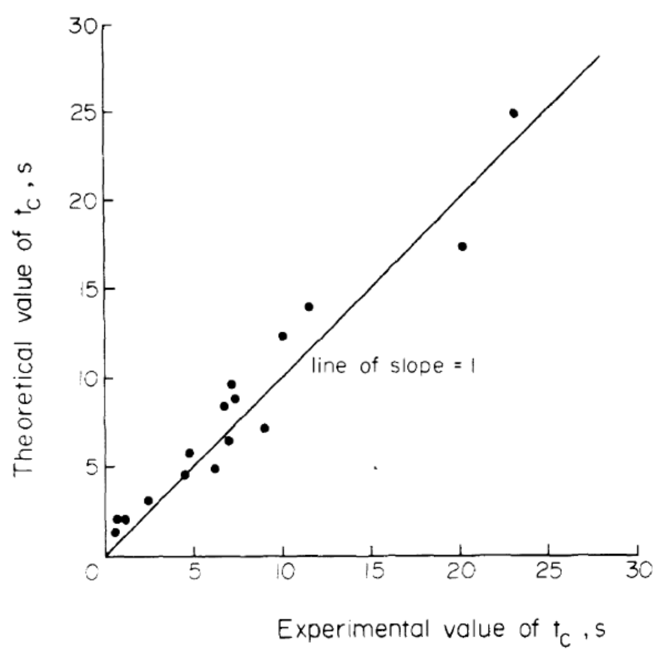


Figure 7: Comparison of theoretical and experimental critical delay times [4]

Other conclusions from the work by the British Gas Council [4]:

- The critical concentration is independent of the size of the firebox
- The critical concentration is independent of the full rate firing density [fired duty / volume of enclosure]
- The critical concentration is independent of the ignition rate of the burner
- The critical concentration remains around x_{LFL} regardless of the fuel composition.
  Note that the original authors did report a strong dependency of the critical concentration with hydrogen content of the fuel. This finding is true when the critical concentration is expressed as a fraction of the stoichiometric fuel in air concentration. However, when the critical concentration is related to the lower flammability limit of the fuel, the dependency vanishes and the critical concentration is found to be $x_{cc} \approx x_{LFL}$.
  Note that for hydrogen, the lower flammability limit is at 13.5% of the stoichiometric concentration while for regular hydrocarbons, the lower flammability limit is at around 50-55% of the stoichiometric concentration.
- The critical concentration reduces with increasing combustion air temperature. The ratio of critical concentration at 300 °C over the critical concentration at ambient condition was found to be approximately 0.75. This corresponds to the effect of elevated temperature on the lower flammability limit (see Annex B)
  The critical concentration reduced with increasing firebox wall temperature in the same magnitude as for the air temperature.
- If the fuel is injected with minimal mixing (tested by deliberately making a bad burner), the critical concentration slightly reduces, and the pressure rise for mixtures that are leaner than the critical concentration is about double compared to a normal burner.
- The critical delay time reduces with increasing full rate firing density and increasing fuel concentration. An important factor is that the critical delay time only becomes relevant when the critical concentration is exceeded.

The paper concludes [4]: "The essential feature (…) is that the bulk gas-to-air ratio within plant combustion chamber never exceeds the LFL." This would result for natural gas at ambient condition, in a 49.5% of stoichiometry-concentration. For practical reasons, safety margins were introduced to extend range of applicability without complicating the assessment too much and the paper concluded that 25% of stoichiometry should be used as the maximum natural gas accumulation in a firebox which corresponds to a safety factor of 2. Using this safety factor extends the range of applicability up to 400 °C ignition temperature, 20% hydrogen addition to natural gas and a fouled burner nozzle which hampers the fuel/air mixing process. This safety margin, when expressed as fraction of LFL, equals 50% of LFL.

These conclusions were further translated into a simplified set of ignition criteria, which can be summarized as: the bulk fuel gas-to-air ratio in the firebox shall never exceed 50% x_{LFL}.

Although the critical concentration is proven to be at around x_{LFL}, taking a 50% lower threshold allows for inaccuracies and potential deviations from the ideal model:

- Fuel gas composition is not exactly known.
- Air temperature may be higher than used to calculate the lower flammability limit.
- The walls of the firebox may be hotter, widening the flammability range of the fuel.
- Timing of the fuel admittance, fuel flow, and air flow may deviate.

The paper further concludes [4]: "(…) in order to avoid unnecessarily long times, a limit of 5 seconds has been retained in the new Code."

Eq. 16 can be further developed to get the maximum burner ignition fuel quantity $Q_{max,ignition}$ [MJ/m³]. This is the multiplication of the burner ignition rate [MW] with the critical delay time [s], per m³ of available firebox volume (from Eq. 11).

$$Q_{max,ignition} = \frac{\dot{V}_{fuel} \cdot t_{cd} \cdot S_{fuel}}{V_{FB}} = \frac{S_{fuel}}{(1 + AF_{ST} \cdot \lambda)} \cdot \ln\left(\frac{1}{1 - (1 + AF_{ST} \cdot \lambda) \cdot 0.97\, x_{LFL}}\right) \qquad \text{Eq. 20}$$

If the minimum air factor $\lambda_{min,dilution}$ is introduced, the equation becomes:

$$Q_{max,ignition} = \frac{\dot{V}_{fuel} \cdot t_{cd} \cdot S_{fuel}}{V_{FB}} = \frac{S_{fuel}}{(1 + AF_{ST} \cdot \lambda)} \cdot \ln\left(\frac{1 + AF_{ST} \cdot \lambda_{min,dilution}}{AF_{ST} \cdot \left(\lambda_{min,dilution} - \lambda\right)}\right) \qquad \text{Eq. 21}$$

The result from Eq. 23 (using 90% of LFL for the minimum dilution air factor) for methane, propane, and butane is plotted in Figure 8. For these fuel species - as for typical hydrocarbons - the lower flammability limit coincides with an air factor of approximately 2.0. When the fuel/air mixture to the burner becomes leaner than the critical concentration, the critical delay time becomes infinite. In that scenario, the time delay for ignition becomes irrelevant and the pressure from delayed ignition will be well below the firebox capability.

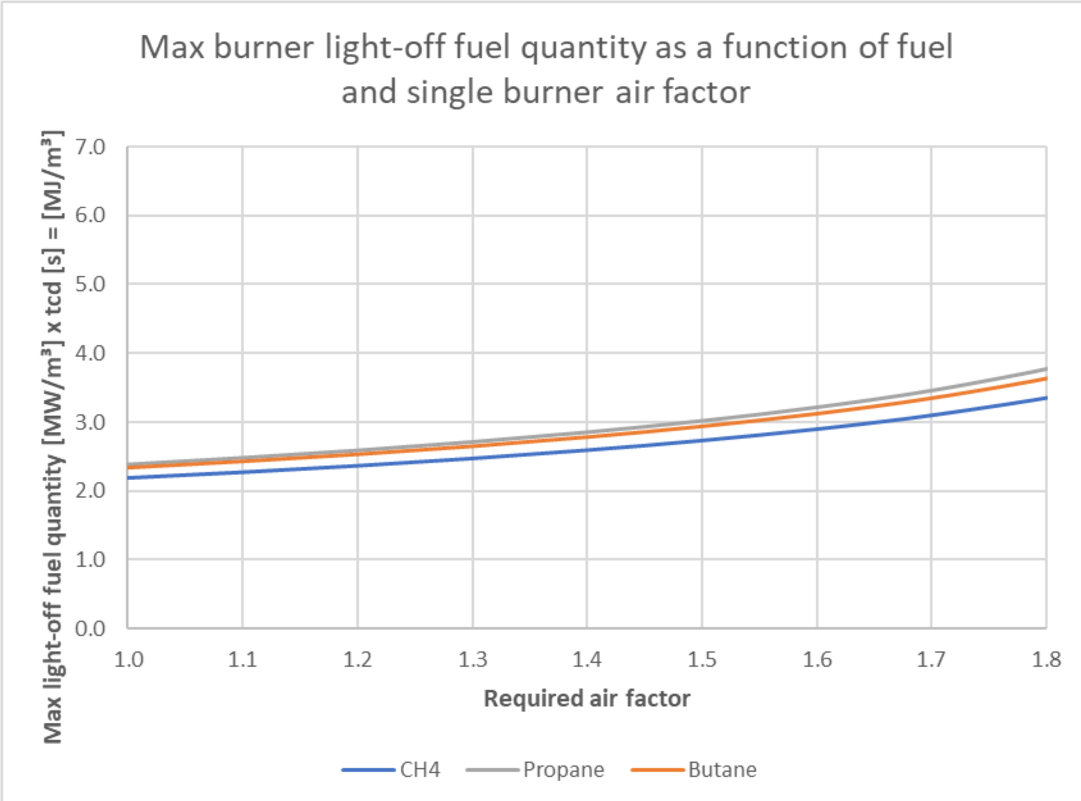


Figure 8: maximum fuel quantity for methane, propane, and butane as a function of the air factor, to maintain the fuel concentration in the firebox below the critical concentration. Curves valid for ambient air temperature and cold firebox

A.5 Burner ignition intensity for heaters built to API 560, API 535 and API 556

Heater designs following the API standards have design characteristics which are relevant for the discussion on safe burner ignition. These characteristics follow from the combination of specific design limitations:

a) API 560 4th edition specified that the volumetric heat release for gas fired heaters shall not exceed 165 kW/m^3 (16000 BTU/h-ft^3)

b) In API 560 5th edition, this clause was restated to limit the maximum floor firing density of 950 kW/m^2 (300000 Btu/h-ft^2). The limitation of 165 kW/m^3 volumetric heat release in API560 4th edition matches the floor firing density limit of API 560 5th edition for heaters with a firebox height of 5.75 m (19 ft). Heaters taller than 5.75m - which is practically the case for almost all refinery fired heaters - that obey the 950 kW/m^2 floor firing density, will have less than 165 kW/m^3. More typical, API560 fired heaters will be more in the range of 50-100 kW/m^3 (4830-9660 Btu/h-ft^3).
To put this into perspective of the experimental work from the British gas council in the 1970's: those experiments were done with full rate firing densities of 200-1500 kW/m^3 (19,300-144,900 Btu/h-ft^3) [4]

c) The burner ignition practice as described in API 556 and API 535 (4th edition ballot version), requires ignition at or below the minimum burner heat release and up to 50% of the burner capacity.

d) The number of burners in refinery fired heaters built to API standards do not follow specific standard requirements. In practice though, the burner sizes that are most used in these fired heaters built to API 535 range from 0.3 to 5 MW (1.0 to 17 MMBTU/h) heat release range for natural draft while allowing up to 20 MW (68 MMBTU/h) for forced draft applications.
A minority of the heater applications will have a single burner and is typically for packaged units (smaller utility steam boilers, hot oil packaged heaters) or in special services like sulfur plants and incinerators.

From the aforementioned parameters for a heater with n burners, it can be derived that the burner ignition firing capacity for a heater built to API standards, will be below $\frac{165}{n}50\% = \frac{82.5}{n}$ kW/m³ ($\frac{7971}{n}$ BTU/h ft³). This is an absolute maximum value for newly built heaters designed according to API 560, API 535, and API 556. A more typical range of the volumetric ignition intensity is $\frac{10}{n}$ to $\frac{30}{n}$ kW/m³ ($\frac{966}{n}$ to $\frac{2899}{n}$ BTU/h ft³).

A.5.1 Total energy input criterion in perspective to API-fired heaters

Eq. 4 or Eq. 5 can be used to determine the maximum pressure peak that would occur in the case where the total fuel energy supplied during the ignition and prior to successful ignition, suddenly ignites in the firebox. The assumptions in the total energy input criterion are:

- The space in which the ignition occurs is a closed space, i.e. no free outflow of combustion products and no free inflow of air.
- All fuel supplied from opening the fuel valve to the burner until the fuel ignites in the firebox, mixes in a stoichiometric ratio with air and accumulates without further mixing inside the firebox.

However, in real-life fired heaters in refining and petrochemical service , none of these assumptions are true. First, the firebox has a flow path for fresh air through burners and flue gas evacuation. Thus, an actual firebox is not a closed volume. The convection section and the stack do create some obstruction for a potential pressure rise inside the firebox. In practice, the firebox during ignition is well-ventilated, i.e. air is supplied through open burner air registers, forced draft fan (if installed), available heater draft, etc.

Secondly, the fuel injected through a burner nozzle mixes with air in the firebox and dilutes rapidly below flammability limits. This happens because:

- The fuel jet entrains air from the firebox.
- Fresh air is continuously supplied through air registers using available draft.
- The burner is designed for good air and fuel mixing
- Fuel disperses into the larger air-filled space. The dispersion is enhanced by fresh air supply.

As a result, the fuel cannot accumulate during the trial-for-ignition period in a single stoichiometric air/fuel pocket. Instead, a small space of a flammable air/fuel mixture develops in the proximity of the burner nozzles. The fuel concentration in this pocket reduces with the distance to the burner nozzle as the fuel jet entrains increasing amounts of air and in less than one meter from the fuel nozzle, the mixture is below flammability limits.

Therefore, the total energy input criterion applied to the total fuel supply during the trial for ignition is not representative for actual fired heaters. It results in very conservative conclusions as it predicts a higher pressure rise from delayed ignition than what can occur realistically. However, if the ignition practice on a particular fired heater meets the total energy input criterion, then the ignition practice will not be problematic regarding destructive pressure rise from delayed ignition.

In their 1976 paper, the researchers from the British Gas Midlands Research Station [4] acknowledge that the total energy input criterion and the dilution criterion "undoubtedly contributed to safety (…). It has however long been felt, particularly by burner manufacturers, that the start-up rates permitted by these criteria have been unnecessarily low and that advantages in terms of operational reliability would accrue if the permitted rates could be increased without loss of safety."

A.5.2 The intermediate burner ignition experiments in perspective to API-fired heaters

Using Eq. 22 and Figure 8 in the perspective of practical fired heaters in general refinery and petrochemical service:

a. Eq. 22 and Figure 8 indicate that for the most conservative case with general hydrocarbon fuels at ignition, the product of the volumetric fuel heat release with the trial for ignition period shall be maintained below 2.2 MJ/m^3 (59 BTU/ft^3). This assumes that the air/fuel that is supplied at the burner is at stoichiometry.
b. To include a safety factor of 2 on the API 556 trial-for-ignition time of 5 seconds, assume that the air/fuel mixture will be shut off within 10 seconds, i.e. after 10 seconds, the fuel inside the firebox has accumulated to the critical concentration. Should this fuel ignite, then the pressure rise is still manageable for a normally designed firebox and on the order of 10-20 mbar (0.14 - 0.29 psi).
c. 10 seconds critical delay time with a 2.2 MJ/m^3 (59 BTU/ft^3) volumetric heat input corresponds to 0.22 MW/m^3 or 220 kW/m^3 (21,250 BTU/h ft^3) volumetric heat input during ignition.

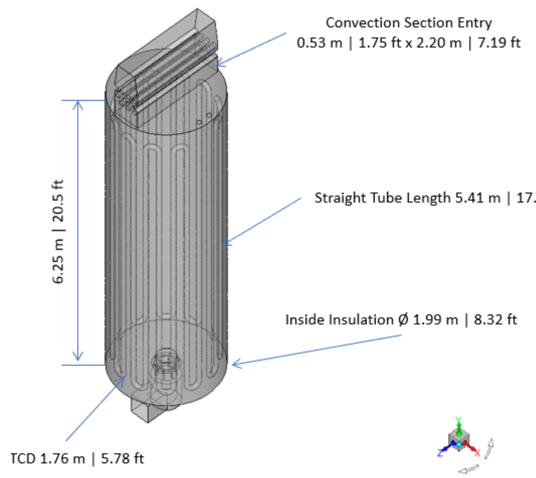
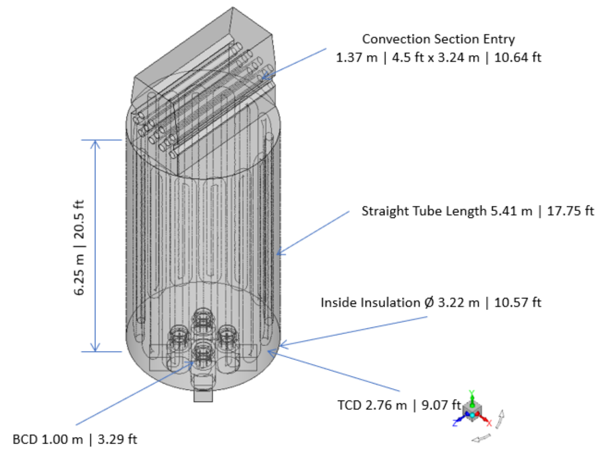
For a four-burner heater, the ignition volumetric heat release is less than $\frac{82.5}{4}$ = 21.25 kW/m^3 (2053 BTU/h ft^3). The fired heater in general refinery service, following the API 535, API 560, and API 556 design guidelines and ignition practices therefore has a comfortable margin of 220/21.25 = 10.3 for this example compared to the work from the British Gas council 1976.

To further add to the comparison, the ignition practice as described in API 556 requires that all burner air registers are in the open position at ignition such that the air flow into the firebox is *n* times higher than the single burner air flow (with *n* being the number of burners). The bulk air-to-fuel ratio during ignition will be n times higher than the single burner air/fuel ratio.

A.6 Computational Fluid Dynamics

To further investigate the process of a developing air/fuel mixture in a refinery process fired heater under ignition conditions, the API 556 working group on safe burner ignition ordered a CFD analysis. Two heater designs were assessed: a single burner vertical cylindrical heater in steady state (Figure 9) and a four-burner cylindrical heater (Figure 10) both in steady state as well as in transient condition during ignition. The heater geometries were selected and defined to:

- Align with current API 560 guidelines on floor firing density, burner to burner spacing, and burner to coil spacing.
- To have a high design volumetric heat release, which results in the most challenging conditions for burner ignition (high burner heat release per volume of firebox).

 *Figure 9: Single burner heater used in the CFD* Volumetric heat release @ design rate: 72.68 kW/m³ (7020 BTU/h ft³) Heat release at burner ignition with 10/1 turn down 7.3 kW/m³ (702 BTU/h ft³)	 *Figure 10: Four-burner heater used in the CFD* Volumetric heat release @ design rate: 92.12 kW/m³ (8900 Btu/h ft³) Heat release at burner ignition with 10/1 turn down 2.3 kW/m³ (222 BTU/h ft³)

The majority of refinery fired heaters have more than four burners and a lower volumetric heat release per burner. On this basis, CFD yields conservative results compared to the majority of the process fired heaters in refineries.

The same staged fuel ultra-low-NOx burner was used in all the CFD cases. The burner geometry used six primary and six staged gas tips with a design heat release of 1.41 MW (4.80 MMBTU/h) and a design throat air velocity of 10 m/s (33 ft/s).

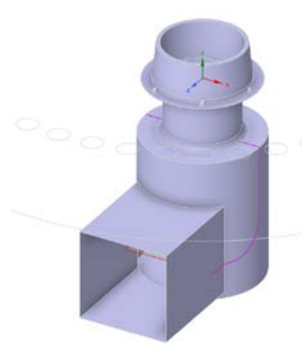


Figure 11: Burner as modelled in the CFD

A.6.1 CFD approach to determine relevant energy accumulation

The CFD study modelled the mixing of fuel with gas from the firebox atmosphere. In each mesh-cell of the firebox, the fuel concentration was calculated. In the CFD analysis, the firebox is sliced into millions of sub-cells to compute velocity vectors, mass balances, energy balances, etc. In each sub-cell of volume dV_i, the volumetric fuel concentration x_i can be computed. The total chemical energy in the form of fuel molecules in a firebox, is found through integration of the fuel concentration over the total firebox volume, and multiplying by the lower volumetric heating value of the fuel:

$$E_{tot} = S_{fuel} \int_{V_{firebox}} x_i \cdot dV_i \qquad \text{Eq. 22}$$

This is the “total accumulated energy” inside the firebox. The total accumulated energy in the firebox is expected to follow approximately the average fuel concentration that would accumulate in the firebox using a well-stirred reactor model, where the air flow through all burners (with open air registers) is considered in the well-stirred reactor model.

However, when the fuel concentration in a particular cell is less than stoichiometry and even less than flammability, the fuel (energy) in that cell will not contribute to its full extent for a pressure rise. This was shown in the work of the British Gas Midlands research Station [1] in 1970 (see Figure 3). Therefore, a method was derived based on the findings from the 1970 experiments to discount the fuel energy in cells with less than stoichiometric fuel concentration and named this the “reduced energy concept”.

This reduced energy concept introduces a weighting factor $R(x_i)$ based on the fuel concentration in each cell such that Eq. 24 is transformed to:

$$E_{red} = S_{fuel} \int_{V_{firebox}} x_i \cdot R(x_i) \cdot dV_i \qquad \text{Eq. 23}$$

With $R(x_i)$ being the reduction factor as a function of the fuel concentration in that cell and $0 \le R(x_i) \le 1$. Different approaches have been used in the CFD for the weighting factor $R(x_i)$:

- Total Energy accumulation, defined as $R(x_i) = 1.0$. In this case, every molecule of fuel in the firebox is accounted for with its full energy content regardless of the local fuel concentration.
- Energy in flammable volume, defined as
  If $x_i < x_{LFL}$ Then $R(x_i) = 0.0$ else $R(x_i) = 1.0$
- Energy available down to 75% of LFL, defined as
  If $x_i < 0.75 \cdot x_{LFL}$ Then $R(x_i) = 0.0$ else $R(x_i) = 1.0$
- Energy available down to 50% of LFL, defined as
  If $x_i < 0.50 \cdot x_{LFL}$ Then $R(x_i) = 0.0$ else $R(x_i) = 1.0$
- Energy available down to 25% of LFL, defined as
  If $x_i < 0.25 \cdot x_{LFL}$ Then $R(x_i) = 0.0$ else $R(x_i) = 1.0$

- "Reduced Energy" as derived from GC166 [1] experiments:
  $R(x_i) = \min(1, \max(0.161\ x_i/x_{LFL} + 0.0286,\ 6.047\ x_i/x_{LFL} - 2.911))$ derivation shown below.

The different methods are shown in Figure 12.

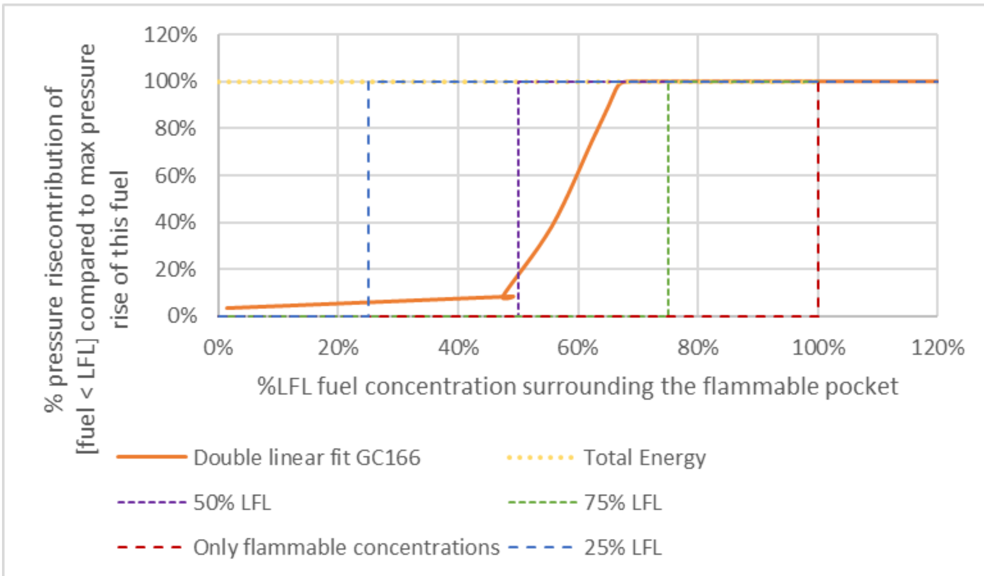


Figure 12: Different formulations of the sub-LFL energy depreciation

The reduction factor function as the "Reduced Energy" from GC166 experiments [1] is derived from Figure 3, with the intent that using E_{red} to calculate the global volumetric energy density ξ in Eq. 4 or Eq. 5 does yield the pressure indicated in the ordinate of Figure 3.

The following procedure has been used to determine $R(x_i)$ based on Figure 3:

- Calculate %LFL of the fuel in the volume surrounding the flammable pocket based on Figure 3 and the experimental description in the original 1970 paper [1].
- Calculate the energy density kJ/m³ of the <LFL fuel volume.
- Calculate the additional pressure rise from the experiment by having <LFL surrounding the flammable pocket.
- Back calculate how much energy should have been in the flammable pocket to achieve the same pressure rise as in the experiment → "equivalent accumulated energy" for the <LFL space.
- Divided the "equivalent accumulated energy" by the "total accumulated energy in the <LFL space".
- This ratio can then be used to depreciate fuel in the firebox that is < LFL and calculate the total "reduced energy content" as if the energy is all within flammable limits.

Using the parameter $\chi_i = x_i/x_{LFL}$ which is the "% of LFL" concentration of fuel in each cell, of this procedure yields the following double-linear formulation for the reduction function

$$R(\chi_i) = \min(1, \max(0.161\ \chi_i + 0.0286,\ 6.047\ \chi_i - 2.911)) \qquad \text{Eq. 24}$$

The result of this double-linear approximation is shown in Figure 13. The orange line represents 0.161 χ_i + 0.0286 while the grey line represents 6.047 χ_i – 2.911. Data points as obtained from Figure 3 are represented by the dots.

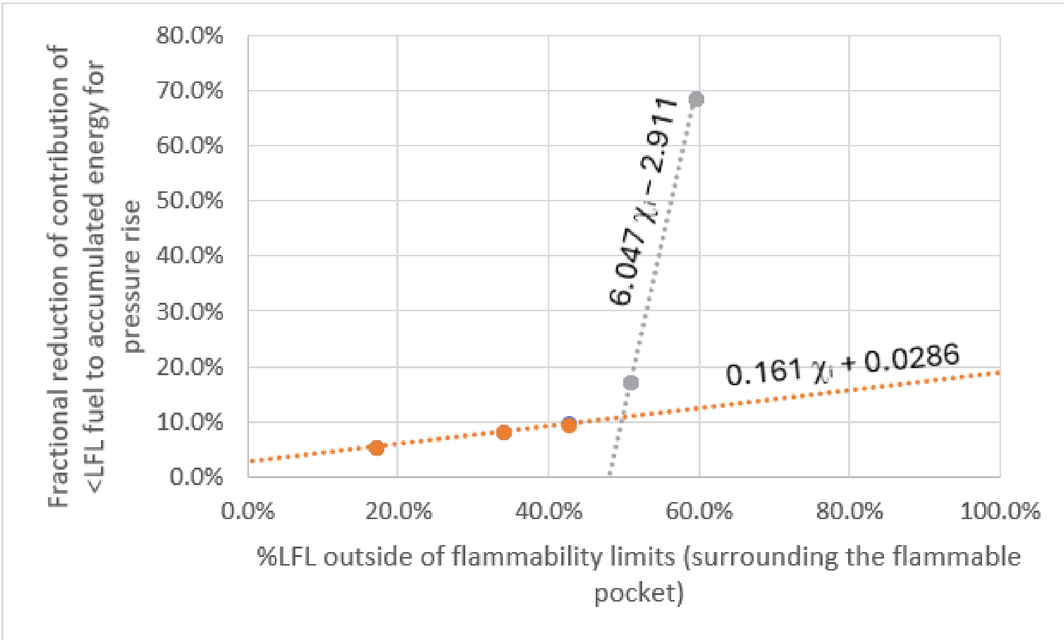


Figure 13: Energy Reduction function

Such that Eq. 25 becomes

$$E_{red} = S_{fuel} \cdot \int_{V_{firebox}} x_i \cdot min(1, max(0.161 \frac{x_i}{x_{LFL}} + 0.0286, 6.047 \frac{x_i}{x_{LFL}} - 2.911)) \cdot dV_i \qquad \text{Eq. 25}$$

Eq. 27 is used in the CFD analysis to determine the reduced energy concentration in the firebox and derive a potential pressure from delayed ignition through Eq. 4 or Eq. 5.

A.6.2 Conclusions from the CFD analysis

A transient CFD analysis (ref API CFD) showed that the region containing a flammable air-fuel mixture grows in the first two seconds of the fuel valve opening: see Figure 14. After these initial two seconds, the size and energy content of this flammable “pocket” remained at the same level. Only when the firebox internal circulation recycled earlier injected fuel into the burner area, the size and energy content of the flammable mixture grows again. Typical firebox recirculation currents take five to twenty seconds to recycle, depending on firebox geometry, draft, air flow, fuel buoyancy, firebox temperature, etc. This informs the use of a limited trial for ignition time.

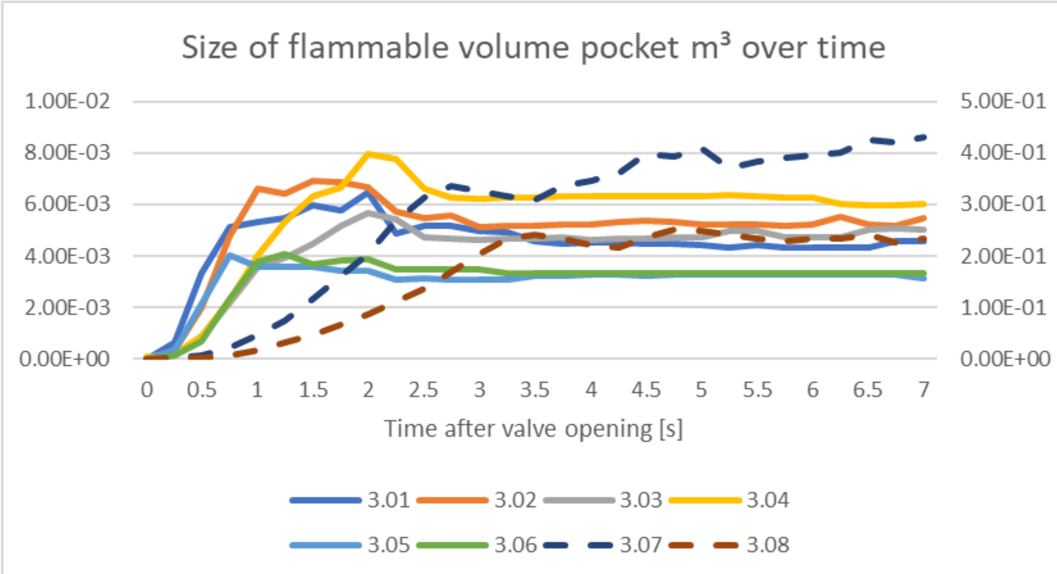


Figure 14: Size of the "flammable pocket" developing during burner ignition

Further findings from the CFD analysis are:

- The total energy accumulation in the firebox in the CFD analysis compares well with the total energy accumulation as suggested by the intermediate case: see Figure 16.
- The reduced energy in a firebox accumulates slower than the total energy due to fuel dilution and dispersion the further the distance to the gas tip.
- Air supply through adjacent burners can be accounted for in the dilution calculations.
- On a time scale of 5 to 10 seconds (see Figure 18), the reduced energy is smaller but still similar in magnitude compared to the total energy accumulation. The reduced energy accumulation in a firebox under burner ignition conditions is, by definition, lower than the total energy accumulation (see Figure 17).
- Reduced energy accumulates slower with higher dilution. This underpins the advantage of having either a forced air flow or an induced draft (with all burner air registers open) during burner ignition.
- Based on Figure 18, a correlation was found linking the ratio of reduced energy accumulation to the total energy accumulation with the bulk air-fuel concentration feeding into the burners. The correlation considered the timestamp of 5 seconds after fuel valve fully open:

$$\frac{E_{red}}{E_{tot}} = 0.51 \cdot \left(\frac{x_{fuel}}{x_{LFL}}\right)_{\infty}^{1.023} = 0.51 \cdot \left(\frac{1}{x_{LFL} \cdot \left(1+\frac{\dot{V}_{air}}{\dot{V}_{fuel}}\right)}\right)_{\infty}^{1.023} \qquad \text{Eq. 26}$$

Note that Eq. 28 is to be truncated at a maximum ratio of 1.0 since E_{red} is always less than or equal to $E_{tot.}$

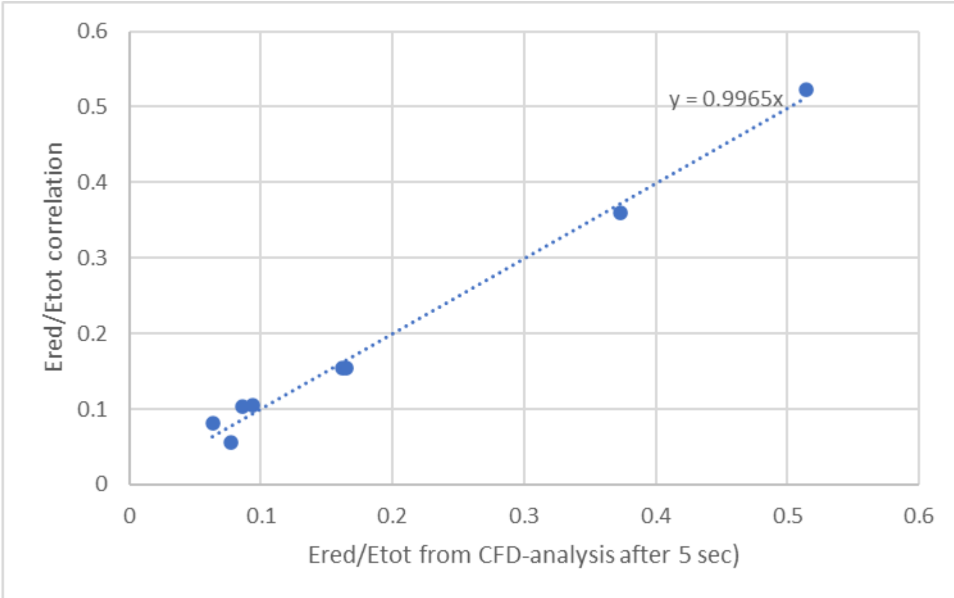


Figure 15: Correlation from Eq. 28 compared with the results from the CFD

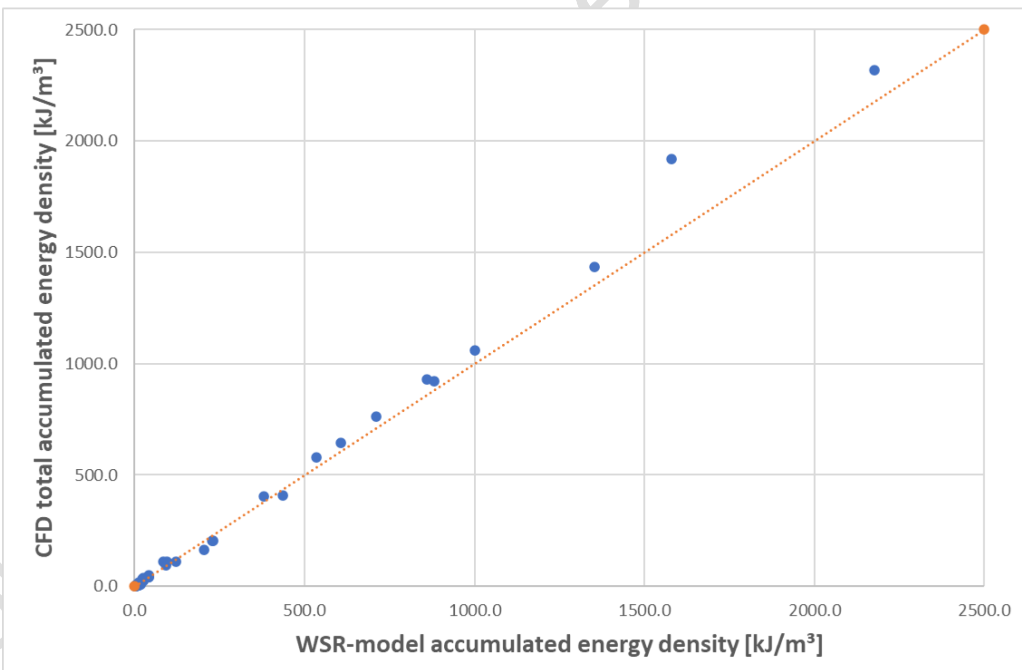


Figure 16: Comparison of the accumulated energy as per Well-stirred reactor model as per Eq. 12 and the CFD calculated total accumulated energy density. Orange line represents perfect theory (X=Y)

Figure 17 compares the discounted energy density as per CFD analysis with the accumulated energy as suggested by the well-stirred reactor model (Eq. 13). The discounted energy is as expected lower than what the well-stirred reactor suggests as $R(x_i)$ reduces the effect of energy accumulation [0…1] - see Eq. 25. The two hydrogen cases 3.7 and 3.8 come closest to the theoretical well-stirred model. The other cases result at significantly lower discounted energy numbers compared to what the well-stirred model suggest.

Figure 19 plots the reduced energy from the CFD calculations in relation to $\frac{\lambda_{min,dilution}}{\lambda}$. The 27 kJ/m³ horizontal line is also plotted. From this graph can be concluded that many of the steady-state infinite time cases that comply with the minimum required dilution air factor for the dilution criterion based on 90% of x_{LFL} ($\frac{\lambda_{min,dilution}}{\lambda} \geq$ 1.0), still have more than 27 kJ/m³ reduced energy in the firebox. The transient cases, where the fuel admittance is time limited however, are consistently below the threshold of 27 kJ/m³. To get all but one of the infinite time cases to the left-hand side of $\frac{\lambda_{min,dilution}}{\lambda}$ = 1.0, the critical concentration has to be reduced from 90% of LFL to 50% of LFL. This transforms Eq. 7 into Eq. 29 (for a methane-hydrogen mixture at ambient conditions)

$$\lambda_{min,dilution} = 4.2 \cdot \frac{4.9 - x_{CH_4}}{1 + 3 \cdot x_{CH_4}} = 4.2 \cdot \frac{3.9 + x_{H_2}}{4 - 3 \cdot x_{H_2}} \qquad \text{Eq. 27}$$

Bringing all infinite time points to the left-hand side requires a threshold of 25% of LFL.

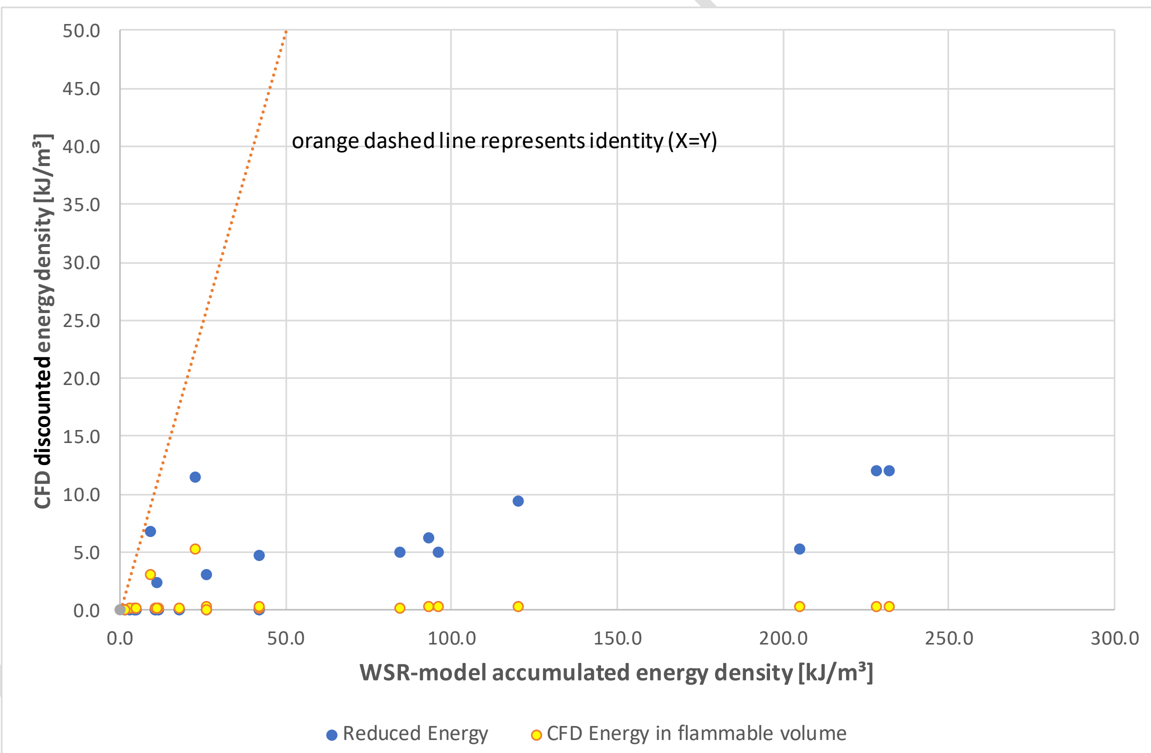


Figure 17: Comparison of CFD discounted energy densities (reduced energy and energy within flammability limits) with the accumulated energy from the thoretical well-stirred reactor model following Eq. 13 for the transient cases only.

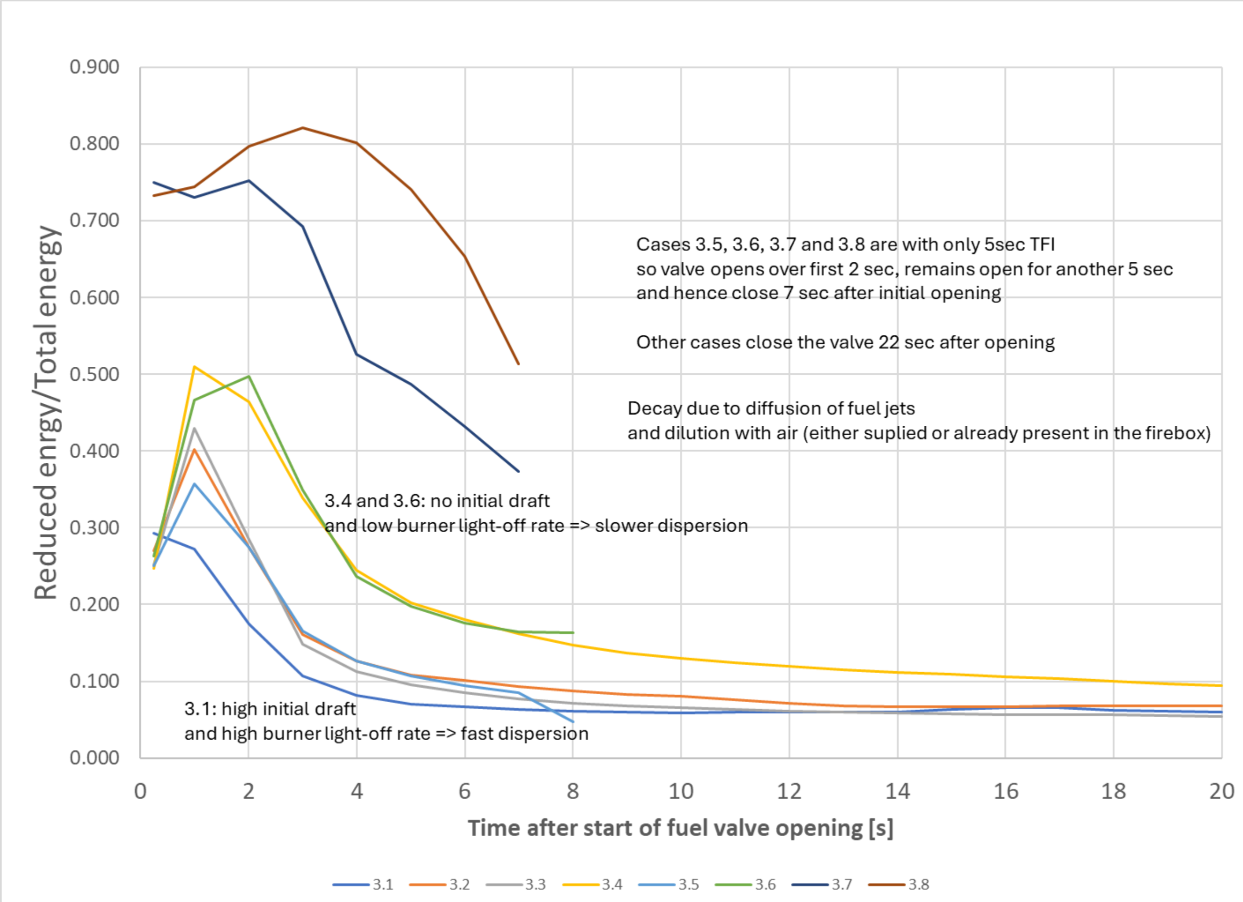


Figure 18: Ratio of reduced energy over total energy accumulation as a function of time for transient CFD cases and the four-burner heater model

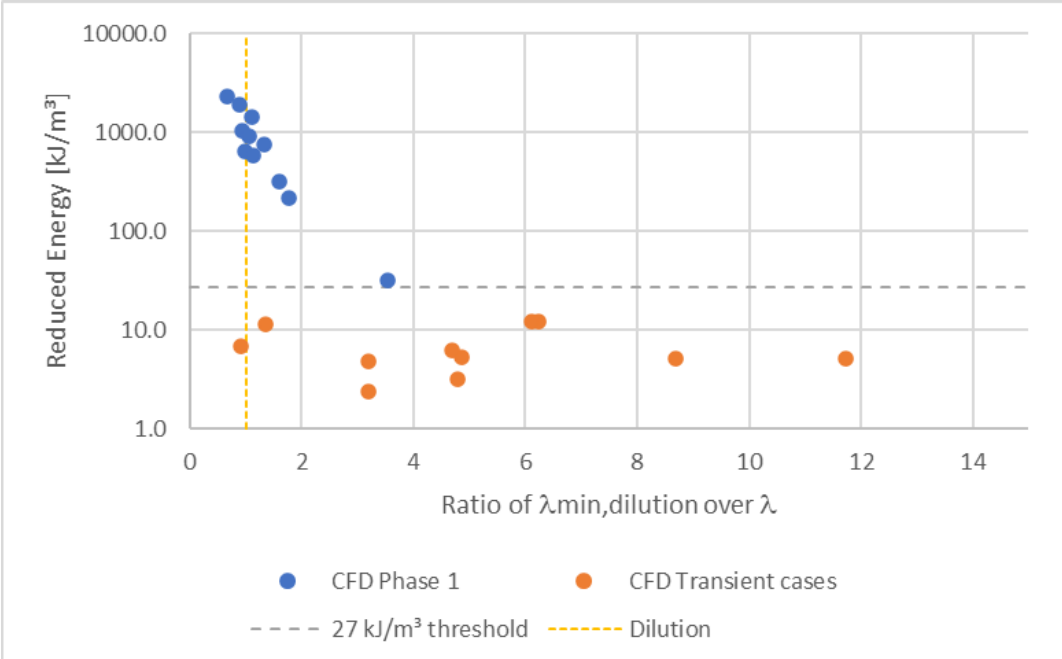


Figure 19: Reduced Energy is a function of $\frac{\lambda_{min,dilution}}{\lambda}$

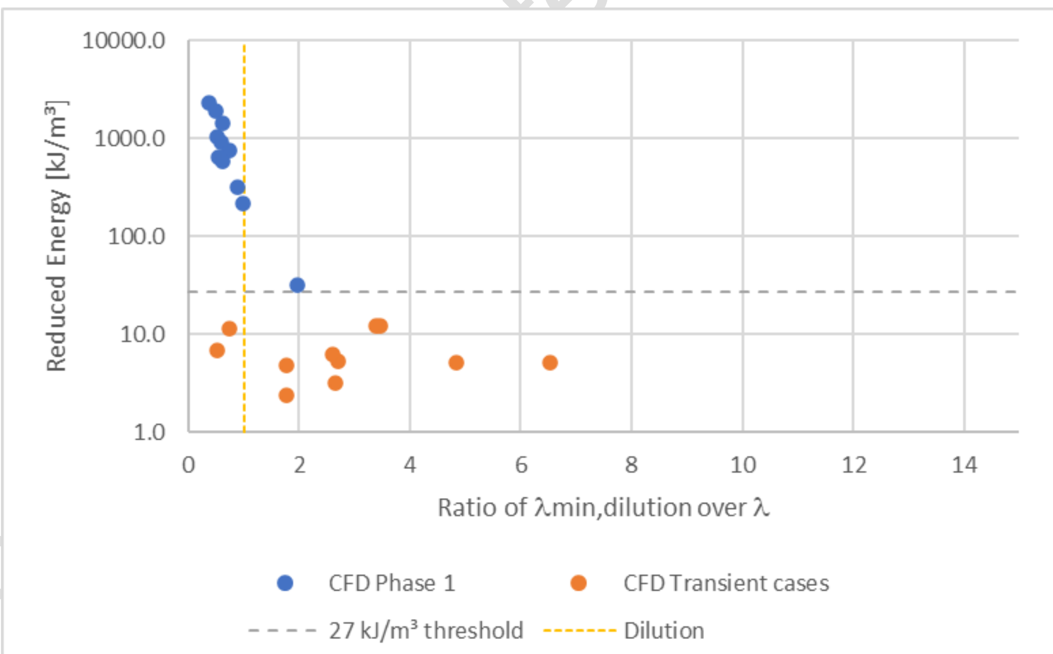


Figure 20: as Figure 19 but with $x_{cc} = 0.50\ x_{LFL}$

A.7 Deriving a burner ignition criterion from the CFD study

Taking the following findings into consideration (derivation are in SI-units):

- During the initial time of fuel admittance to the firebox, the difference between total fuel admittance since the fuel valve opening and the fuel accumulation based on a well-stirred reactor model is negligible:

$$E_{tot,WSR} = \frac{\dot{V}_{fuel}}{\dot{V}_{fuel} + \dot{V}_{air}} \cdot V_{FB} \cdot S_{fuel} \cdot \left(1 - e^{-\frac{t \cdot (\dot{V}_{fuel} + \dot{V}_{air})}{V_{FB}}}\right)$$

$$E_{tot} = \dot{V}_{fuel} \cdot t \cdot S_{fuel} = P_{ignition} \cdot t \qquad \text{Eq. 28}$$

When $t \ll \frac{V_{FB}}{\dot{V}_{fuel} + \dot{V}_{air}} \Rightarrow E_{tot,WSR} \approx E_{tot}$

- Using the correlation represented in Eq. 28 and ignoring the power of 1.023 as this is approximately the same as the value to the power of 1.0.

- Eq. 4 or Eq. 5 to correlate the reduced energy density with the firebox strength

$$\frac{0.51}{x_{LFL} \cdot \left(1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}}\right)} \cdot \frac{P_{ignition}}{V_{FB}} \cdot t \le 0.54 \cdot \Delta p_{peak} \qquad \text{Eq. 29}$$

It should further be acknowledged that the scope of the CFD analyses was limited to no more than a four burners firebox. The energy reduction by dilution as derived in Eq. 28 therefor should not be extended to more than four burners, which turns Eq. 31 into:

$$\frac{0.51}{x_{LFL} \cdot \left(1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}} \cdot min\left(1, \frac{4}{n}\right)\right)} \cdot \frac{P_{ignition}}{V_{FB}} \cdot t \le 0.54 \cdot \Delta p_{peak} \qquad \text{Eq. 30}$$

With n the number of burners in the firebox.

Assume a safety factor of ~2 (2 · 0.51 ≈ 1) , yields Eq. 33

$$\frac{1}{x_{LFL} \cdot \left(1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}} \cdot min\left(1, \frac{4}{n}\right)\right)} \cdot \frac{P_{ignition}}{V_{FB}} \cdot t \le 0.54 \cdot \Delta p_{peak} \qquad \text{Eq. 31}$$

Since it is further proven that the total energy input by itself is conservative, the result of the first fraction can be limited to a maximum value of 1.0:

$$min\left(1, \frac{1}{x_{LFL} \cdot \left(1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}} \cdot min\left(1, \frac{4}{n}\right)\right)}\right) \cdot \frac{P_{ignition}}{V_{FB}} \cdot t \leq 0.54 \cdot \Delta p_{peak} \qquad \text{Eq. 32}$$

- Assuming a trial-for-ignition time of 5 seconds (t = 5 s) and a maximum reduced energy density of 27 kJ/m³ corresponding to a firebox inherent strength for 50 mbar internal pressure then finally results in:

$$min\left(1, \frac{1}{x_{LFL} \cdot \left(1 + \frac{\dot{V}_{air}}{\dot{V}_{fuel}} \cdot min\left(1, \frac{4}{n}\right)\right)}\right) \cdot \frac{P_{ignition}}{V_{FB}} \leq 5.4 \qquad \text{Eq. 33}$$

- From this, the minimum required air flow for a safe burner ignition, considering all points above, becomes

$$\dot{V}_{Air} \geq \dot{V}_{fuel} \cdot \frac{n}{MIN(4, n)} \cdot \left(\frac{0.185 \cdot P_{ignition}}{x_{LFL} \cdot V_{FB}} - 1\right) \qquad \text{Eq. 34}$$

It is further good practice to assure that the burner being ignited has a certain amount of air in excess for the fuel that is being supplied during burner ignition. Assume a minimum air factor of 1.2 (20% excess air) as a minimum, completes the requirement for minimum air for burner ignition:

$$\dot{V}_{Air} \geq \dot{V}_{fuel} \cdot MAX\left(1.2 \cdot n \cdot AF_{ST},\ \frac{n}{MIN(4, n)} \cdot \left(\frac{0.185 \cdot P_{ignition}}{x_{LFL} \cdot V_{FB}} - 1\right)\right) \qquad \text{Eq. 35}$$

A.8 Relevance of available draft during burner ignition

Of special note is that the CFD analysis had three cases where there was no initial draft and no initial air flow. The air registers and stack damper are however fully open at initial burner ignition such that air can flow freely. However, in the CFD analysis it was clearly shown that in those cases, air flow was induced by the jet-effect of the fuel and the buoyancy effect of the fuel.

Conversely, if there is no draft induction or air flow during burner ignition, then the fuel that is injected during a burner ignition attempt, does not effectively leave the firebox. The fuel does disperse by thermal diffusion and buoyancy, but without active ventilation the fuel remains in the firebox for an extended time. Even if it is dispersed far below the lower flammability limit, the fuel will still have a certain contribution to a pressure rise would ignition occurs on a next burner ignition attempt. Therefore, active ventilation of the firebox during burner ignition is key to reduce waiting time after a failed burner ignition attempt to a practical time frame of a few minutes.

A practical solution to this is to install a draft inducing device on new API-fired heaters, that is capable of providing an initial draft of at least -1.27 mmH_2O (-0.05 inH_2O) in a still and cold firebox. Advantages of this:

- This gives operators the possibility to estimate the initial air flow and apply any of the dilution formulas that are needed.
- The draft induction device allows for a controlled and verifiable[1] firebox purge prior to introducing ignition sources (lighting of pilots).
- -1.27 mmH_2O (-0.05 inH_2O) is a measurable level of draft.
- It is worth noting that the practice of lighting all pilots prior to ignition of main burners provides draft. The use of a draft inducing method is then still required for the initial preparation of the heater for any ignition source.

The available draft in the firebox can be used to estimate air flow. On the burner datasheet the air pressure loss is available in Pa or inH_2O is available. The design pressure drop $\Delta p_{air,design}$ is for a burner in service, with wide open air register and measures the static pressure drop between air inlet of the windbox to the floor of the firebox. The design air side pressure drop corresponds to the design air flow $\dot{m}_{air,design}$ under design air pressure and temperature. This can also be found in the burner datasheet. The air flow $\dot{m}_{air,2}$ with a different available draft Δp_2, with full open air register can then be found as

$$\dot{m}_{air,2} = \dot{m}_{air,1} \cdot \sqrt{\frac{\Delta p_2}{\Delta p_1}} \qquad \text{Eq. 36}$$

The actual air flow $\dot{m}_{air,2}$ without a flame will be higher than calculated with Eq. 38 because without the flame, the combustion back pressure is not present.

[1] Verifiable by a draft measurement in the firebox, in conjunction with the field operator validating that he can feel air flow being induced into the burner air inlet.

Annex B

(informative)

Temperature dependency of the lower flammability limits

Le Chatelier derived a formula to calculate the lower flammability limit of a fuel mixture composed of hydrogen and hydrocarbon molecules:

$$\frac{1}{x_{LFL,mix}} = \sum \frac{x_{HC_i}}{x_{LFL,HC_i}} \qquad \text{Eq. 37}$$

See NFPA HAZ01[5] for the individual component's flammability limit x_{LFL,HC_i}.

Zabetakis[7] developed the following estimation of the LFL concentration of a regular hydrocarbon molecule with formula C_cH_h having c ≥ 1 :

$$x_{LFL,ch,20\,°C} = \frac{0.55}{1 + 1.193\,(4c + h)} \qquad \text{Eq. 38}$$

Note that the stoichiometric fuel concentration for a regular C_cH_h can be found as

$$x_{ST} = \frac{1}{1 + 1.190\,(4c + h)} \qquad \text{Eq. 39}$$

Combining Eq. 40 and Eq. 41 concludes that the for regular hydrocarbons, the fuel concentration at the lower flammability limit is around 55% of the stoichiometric concentration - which is indeed the case.

At different temperatures, the lower flammability limit can be recomputed for alkanes and alkenes as [7]:

$$x_{LFL,ch@T} = x_{LFL,ch@20} \cdot (1 - 0.000721\ \cdot (T[°C] - 20)) \qquad \text{Eq. 40}$$

For hydrogen in air, the lower flammability limit can be approximated as from Schröder[6] up to a temperature of 400 °C:

$$x_{LFL,H2} = 4.1\ \cdot (1 - 0.00157\, T[°C]) \qquad \text{Eq. 41}$$

The calculation methodology becomes:

1. Calculate the lower flammability limit of each hydrocarbon component at 20 °C using Eq. 40 or specific data in published tables for instance in NFPA HAZ01 [5].

2. Calculate the lower flammability limit of each hydrocarbon component at desired temperature T [°C] using Eq. 42.
3. Calculate the lower flammability limit of hydrogen at T °C using Eq. 43.
4. Combine the results from the previous steps into Eq. 39 to get the global mixture lower flammablity limit at temperature T °C.

ANNEX C

(informative)

Best Practices for manual burner ignition on heaters built in accordance with API standards and specifications

C.1 The following best practices and considerations are recommended for safe burner ignition:

C 1.1 Install manual valves beside a viewport that gives direct line-of-sight to the related burner. The operator now has a direct visual on the burner that is being ignited while he also controls the fuel supply and trial-for-ignition-time with the manual burner valve. This allows the operator for the best possible control of the burner ignition process.
For heaters where the view on a burner closest to the viewport is obstructed (for instance by a tube or where the burner fires against the wall), other viewports can be used by a second operator that keeps close communication with the field operator that manipulates the fuel valve.

Past incidents with firebox deflagrations showed that the floor of a heater is likely to yield first in case the internal pressure exceeds the firebox strength. Therefore, it is better to design the heater such that operators can perform burner ignition while not standing underneath the heater floor.

C.1.2 Burners are typically capable of igniting with lower fuel pressure than the low fuel pressure trip set point. At these lower fuel pressures, CO emissions may be excessive, but the burner does ignite, and a stable flame can be sustained.

Some of the reasons for setting the low pressure trip above the absolute minimum required pressure are related to:

- reliable and repeatable detection of a low fuel pressure, instrument capability
- control valve capability.

C.1.3 An operator is capable to carefully open the manual fuel valve to a burner while supervising burner ignition. Even if the initial fuel pressure in the line up to the manual valve is high, the operator will be capable to adjust the valve position.

C.1.4 During a manual burner ignition, the field operator opening the fuel valve visually confirms timely ignition and burner flame stabilization. The field operator’s capability to confirm even small flames in local sections of the burner exceeds the capability of flame scanners which are set-up with a limited viewing field and signal sensitivity.

C.1.5 Should the purchaser specify the need for an automated burner ignition, flame scanners to confirm timely main burner ignition are typically installed for each burner with automated ignition capability. Flame scanners have their limitations when comparing to operators though:

- The viewing field from a flame scanner is narrower than what an operator can see. Flame scanners are pointing towards a small section in the burner. If in the initial ignition stage, flame is not present in that particular area, the flame scanner will not detect any flame and cause a failed ignition attempt. In contrast, an operator can interpret small flames on burner tips and adjust the fuel supply based on his experience to get the burner going.

- Flame scanners act on specific wavelengths of flames whereas operators see the whole visible lightspectrum.

- For floor fired burners in particular, dirt collects on the flame scanners lens, despite purge air being added to the flame scanner set-up. Failed automated burner ignition attempts can often be related to dirt on the scanner.

C.1.6 Sniff testing of a firebox with floor fired burners: pay attention to probing close to the floor, below the top of the tile. Heavier-than-air fuel components may collect at the floor and are not as easily removed by purging as the space above the burner tiles.

C.1.7 Sniff testing of a firebox for top fired burners: pay attention to probing close to the roof. Lighter-than-air fuel components may collect at the roof and are not as easily removed by purging as the space below the burner tiles.

ANNEX D

(informative)

Calculation sheet for automated burner ignition, including examples

D.1 Calculation table in SI-units

Fuel Parameters	μ_i = mole fraction of component i in the fuel	
Specific Gravity versus air	$SG = \sum \mu_i \cdot SG_i$	-
Lower Heating Value	$S = \sum \mu_i \cdot S_i$	MJ/Nm³
Wobbe-number	$Wb = \frac{S}{\sqrt{SG}}$	MJ/Nm³
Stoichiometric Air demand	$AF_{ST} = \sum \mu_i \cdot AF_{ST,\,i}$	m³ air/m³ fuel
Lower Flammability limit 20°C	$x_{LFL,20} = \frac{1}{\sum \frac{\mu_i}{x_{LFL,20,i}}}$	-
Assume Temperature for LFL correction	$x_{LFL} = \frac{1}{\sum \frac{\mu_i}{x_{LFL,i}}}$	-
Air temperature	T_a	°C
Firebox Volume	V_{FB}	m³
Burner ignition heat release (1 burner)	$P_{ignition}$	kW
Number of burners	n	
Stoichiometric air flow to all burners	$\dot{V}_{air,ST} = 3.6 \cdot \frac{P_{ignition}}{S} \cdot AF_{ST} \cdot n$ $\dot{V}_{air,ST} = 3.6 \cdot \dot{V}_{fuel,ignition} \cdot AF_{ST} \cdot n$	Nm³/h

Minimum air flow to the burners during ignition - based on excess air for the igniting burner	$\dot{V}_{air,\lambda=1.2} \geq 1.2 \cdot \dot{V}_{air,ST}$ Assuming at a minimum 20% excess air on the burner that is being ignited.	Nm³/h
Minimum air flow to the burners during ignition, based on dilution from air through adjacent burners	$\dot{V}_{air,dil} \geq \dot{V}_{fuel} \cdot \left(\frac{n}{MIN(4,n)} \cdot \left(\frac{0.185 \cdot P_{ignition}}{x_{LFL} \cdot V_{FB}} - 1 \right) \right)$	Nm³/h
Minimum air flow to the burners during ignition - RESULT	$\dot{V}_{air,ignition} = MAX(\dot{V}_{air,\lambda=1.2}, \dot{V}_{air,dil})$	Nm³/h
Single firebox volume exchange	$t_{FB} = 60 \cdot \frac{V_{FB}}{\dot{V}_{air,ignition}} \cdot \frac{273 + T_a}{273}$	minutes
Fuel in air concentration (air dilution from max 4 burners)	$x_f = \frac{1}{1 + \frac{\dot{V}_{air} \cdot S}{3.6 \cdot P_{ignition}} \cdot min\left(1, \frac{4}{n}\right)}$	vol%
Fuel in air concentration as a fraction of LFL	$\chi = \frac{x_f}{x_{LFL}}$	-

Additionally for natural draft and induced draft burners: Use the information from the burner datasheet on the design case. The numbers are for a one burner design.

Burner air-side pressure drop	$\Delta p_{air,design}$	mmH_2O
Burner Design heat release	P_{design}	kW/burner
Burner design air factor	λ_{design} = 1+ "Excess Air" Example: If datasheet mentions 15% excess air, then λ_{design} = 1.15	-
Fuel LHV for burner design case	S_{design}	MJ/Nm³
Stoichiometric air demand	$AF_{ST,vol,design}$	m³ air/m³ gas
Air flow for design case	$\dot{V}_{air,design} = 3.6 \cdot \frac{P_{design}}{S_{design}} \cdot AF_{ST,vol,design} \cdot \lambda_{design}$	Nm³/h per burner

Minimum Required floor draft for burner ignition	$\Delta p_{air,ignition} = \Delta p_{air,design} \cdot \left(\frac{\dot{V}_{air}}{n \cdot \dot{V}_{air,design}}\right)^2$	mmH_2O

D.2 Calculation table in USC-units

Fuel Parameters	μ_i = mole fraction of component i in the fuel	
Specific Gravity versus air	$SG = \sum \mu_i \cdot SG_i$	-
Lower Heating Value	$S = \sum \mu_i \cdot S_i$	BTU/scf
Wobbe-number	$Wb = \frac{S}{\sqrt{SG}}$	BTU/scf
Stoichiometric Air demand	$AF_{ST} = \sum \mu_i \cdot AF_{ST,\,i}$	ft^3 air/ft^3 fuel
Lower Flammability limit 20°C	$x_{LFL,20} = \frac{1}{\sum \frac{\mu_i}{x_{LFL,20,i}}}$	-
Assume Temperature for LFL correction	$x_{LFL} = \frac{1}{\sum \frac{\mu_i}{x_{LFL,i}}}$	-
Air temperature	T_a	°F
Firebox Volume	V_{FB}	ft^3
Burner ignition heat release	$P_{ignition}$	MMBTU/h
Number of burners	n	
Stoichiometric air flow to all burners	$\dot{V}_{air,ST} = 10^6 \cdot \frac{P_{ignition}}{S} \cdot AF_{ST} \cdot n$ $\dot{V}_{air,ST} = \dot{V}_{fuel,ignition} \cdot AF_{ST} \cdot n$	scf/h
Minimum air flow to the burners during ignition - based on excess air for the igniting burner	$\dot{V}_{air,\lambda=1.2} \geq 1.2 \cdot \dot{V}_{air,ST}$ Assuming at a minimum 20% excess air on the burner that is being ignited.	scf/h

Minimum air flow to the burners during ignition, based on dilution from air through adjacent burners	$\dot{V}_{air,dil} \geq \dot{V}_{fuel} \cdot \left(\frac{n}{MIN(4,n)} \cdot \left(\frac{1916.6 \cdot P_{ignition}}{x_{LFL} \cdot V_{FB}} - 1 \right) \right)$	scf/h
Minimum air flow to the burners during ignition - RESULT	$\dot{V}_{air,ignition} = MAX(\dot{V}_{air,\lambda=1.2}, \dot{V}_{air,dil})$	scf/h
Single firebox volume exchange	$t_{FB} = 60 \cdot \frac{V_{FB}}{\dot{V}_{air}} \cdot \frac{460 + T_a}{518.6}$	minutes
Firebox internal pressure strength	Δp_{peak}	psi(g)
Fuel in air concentration (air dilution from max 4 burners)	$x_f = \frac{1}{1 + \frac{\dot{V}_{air} \cdot S}{10^6 \cdot P_{ignition}} \cdot min\left(1, \frac{4}{n}\right)}$	vol%
Fuel in air concentration as a fraction of LFL	$\chi_\infty = \frac{x_f}{x_{LFL}}$	-

Additionally for natural draft and induced draft burners: use the information from the burner datasheet on the design case. The numbers are for a single burner.

Burner air-side pressure drop	$\Delta p_{air,design}$	inH_2O
Burner Design heat release	P_{design}	MMBTU/h per burner
Burner design excess air factor	λ_{design} = 1+ *"Excess Air"* Example: If datasheet mentions 15% excess air, then $\lambda_{design} = 1.15$	-
Fuel LHV for burner design case	S_{design}	BTU/scf
Stoichiometric air demand	$AF_{ST,vol,design}$	ft^3 air/ft^3 gas
Air flow for design case	$\dot{V}_{air,design} = 10^6 \cdot \frac{P_{design}}{S_{design}} \cdot AF_{ST,vol,design} \cdot \lambda_{design}$	scf/h per burner
Minimum Required floor draft for burner ignition	$\Delta p_{air,ignition} = \Delta p_{air,design} \cdot \left(\frac{\dot{V}_{air}}{n \cdot \dot{V}_{air,design}} \right)^2$	inH_2O

D.3 Examples

For all examples below, the following fuel composition is used:

		Blend %mol
N2	Nitrogen	0.00
CO	Carbon Monoxide	0.00
CO2	Carbon Dioxide	0.00
H2	Hydrogen	30.00
CH4	Methane	50.00
C2H4	Ethylene	2.00
C2H6	Ethane	15.00
C3H6	Propylene	0.00
C3H8	Propane	3.00
C4H10	n-Butane	0.00
C4H10	Isobutane	0.00
C5H12	n-Pentane	0.00
C5H12	Isopentane	0.00
C6H6	Benzene	0.00
C6H14	n-Hexanes	0.00
C6H14	Isohexanes	0.00
C7H16	Heptane	0.00
C8H18	Octane	0.00

The fuel parameters are then:

Fuel Parameters	SI	USC
Specific Gravity versus Air	0.521	0.521
Lower Heating Value	34.87 MJ/Nm³	887 BTU/scf
Wobbe-number	48.28 MJ/Nm³	1228 BTU/scf
Stoichiometric Air demand	8.98 Nm³/Nm³	8.98 scf/scf
Stoichiometric Air demand	6.04 kg/kg	6.04 lb/lb
Stoichiometric Air demand	0.332 kg/MJ LHV	772.4 lb/BTU LHV
Lower Flammability limit 20°C	4.05%	4.05%

D.3.1 Example 1: Forced draft heater with 1 burner (SI)

Air temperature	20 °C	
Firebox Volume	132	m³
Burner ignition heat release	750	kW
Number of burners	1	
Calculation Results		
Minimum air flow to all burners	801	Nm³/h
	1033	kg/h
	859	m³/h
Minimum air flow for dilution during ignition	3339	Nm³/h
	4307	kg/h
	3583	m³/h
Minimum total air flow for ignition	3339	Nm³/h
Results for info		
Air factor for igniting burner	5.0	
Single firebox volume exchange	2.2	minutes
Five firebox volume changes take…	11.1	minutes
Fuel in air concentration around igniting burner	3.7%	
Fuel in air concentration as a fraction of LFL	95.0%	

D.3.2 Example 2: Forced draft heater with 4 burners - USC units

Air temperature	60 °F	
Firebox Volume	5000	ft³
Burner ignition heat release	3	MMBTU/hr
Number of burners	4	
Calculation Results		
Minimum air flow to all burners based on single burner air factor	140133	scf/h
	10675	lb/h
	140133	ft³/h
Minimum air flow for dilution	159871	scf/h
	12179	lb/h
	159871	ft³/h
Minimum Air flow for burner ignition	159871	scf/hr
Results for info		
Air factor for igniting burner	1.4	
Single firebox volume exchange	1.9	minutes
Five firebox volume changes take…	9.4	minutes
Fuel in air concentration around igniting burner	3.4%	
Fuel in air concentration as a fraction of LFL	87.0%	

D.3.3 Example 3: As example 2 but with natural draft burners - USC units

Addition for Natural Draft Burners: Fill in data from Burner Design case		
Burner air-side pressure drop	0.5	inH2O
Burner Design heat release	15.00	MMBTU/h per burner
Burner design excess air factor	1.30	
Fuel LHV for burner design case	890.00	BTU/scf
Stoichiometric air demand	9.80	scf/scf
Air flow for design case	214719	scf/h per burner
Minimum Required floor draft for burner ignition	-0.050	inH2O

D.3.4 Example 4: FD heater with 8 burners – SI units

Air temperature	20 °C	
Firebox Volume	286	m³
Burner ignition heat release	1000	kW
Number of burners	8	
Calculation Results		
Minimum air flow to all burners	8540	Nm³/h
	11017	kg/h
	9166	m³/h
Minimum air flow for dilution during ignition	5348	Nm³/h
	6899	kg/h
	5740	m³/h
Minimum total air flow for ignition	8540	Nm³/h
Results for info		
Air factor for igniting burner	1.2	
Single firebox volume exchange	1.9	minutes
Five firebox volume changes take…	9.4	minutes
Fuel in air concentration around igniting burner	3.8%	
Fuel in air concentration as a fraction of LFL	98.9%	

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