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E. No.: NO 939 350 675

TECHNICAL REPORT

SUBJECT/TASK (title)

Co-combustion: Biomass Fuel Gas and Natural Gas

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CLIENT(S)

IEA Bioenergy

TR NO. TRA 4734	DATE 1998-06-29	CLIENT'S REF.	PROJECT NO. 17E004.01
ELECTRONIC FILE CODE p:\prosjekt\17e004\iearep.doc		RESPONSIBLE (NAME, SIGN.) Johan E. Hustad	CLASSIFICATION Open
ISBN NO. 82-594-1296-9	REPORT TYPE	RESEARCH DIRECTOR (NAME, SIGN.) Jens Hetland	COPIES PAGES 26
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RESULT (summary)

In light of the current marked conditions (i.e. low natural gas prices, utility deregulation, etc.), biomass-co-firing projects are one of the best opportunities for near term growth in biomass utilisation. Electricity generation from natural gas is expected to increase considerably until 2010. The expected increase is in gas turbines and combined cycle plants.

Different technologies for co-combustion of biomass fuels and natural gas where the biomass fuels are gasified prior to combustion have been presented. These technologies include; co-combustion in a gas turbine, co-combustion in a boiler and co-combustion in a duct burner. Combined firing of natural gas and low calorific gas (LCV) using the LCV gas as re-burning fuel is also mentioned

Calculations show significant effects of increased methane content in the fuel gas by mixing methane with LCV gas. A substantial increase in flame temperature, laminar burning velocity and lower heating value was observed adding 25% methane to the LCV gas.

Because gasification is not as well developed as biomass combustion, there is an increased risk associated with the commercialisation of low or high-pressure gasification technology. Nevertheless, implementing biomass gasification in this way will bring about technology acceptance, which will lead the way to larger scale biomass plants.

Overall, the processing of biomass to electricity will depend on the cost of the fuel, market conditions, the conversion technology used and the income generated by the power produced.

KEYWORDS

SELECTED BY AUTHOR(S)	combustion	forbrenning
	natural gas	naturgass

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1 INTRODUCTION

In view of the growing use of biomass fuels, it is interesting to evaluate utilisation of existing technologies for production of heat and/or power. For gaseous biomass from gasification of biomass (product gas) a gas turbine system either as a simple cycle or in combination with a steam turbine (combined cycle) and a gas engine system are technologies of particular interest. The fuel gas quality that is specified for these technologies is however generally not compatible with gas produced by air-blown gasifiers. A possible solution for improving the product gas quality is to enrich the gas with natural gas. By co-combustion of biomass (product gas) and natural gas available technologies can be utilised and in light of the current marked situation (i.e. low natural gas prices, utility deregulation, etc.), biomass co-firing projects are one of the best opportunities for near term growth in biomass utilisation.

Biomass is the collective term for all forms of plant material including forest and agro residues. Despite biomass being a divers fuel, it's a logical choice for co-firing in a natural gas fired system. Biomass has environmental benefits, offers fuel flexibility or can be used as supplemental low cost fuel, especially residues, for use in combustion systems, i.e. gas turbines, gas engines and gas burners.

Natural gas is an ideal fuel source for power generation, but there is controversy in some countries about its use for this purpose because it is a fossil fuel. There are several advantages by using natural gas:

- easily transported and readily available on site;
- few handling difficulties
- contains little or no sulphur
- contains little or no particles
- high hydrogen fuel which yields less carbon dioxide than other fossil fuels

Electricity generation from natural gas is expected to increase by over 60 % between 1993 and 2010 (Spath et.al) . The expected increase is in gas turbines and combined cycle plants. Gas fired units have a lower specific capital cost and a shorter construction time than other electricity generating system.

Development trends have concentrated on increased efficiency and power output from gas turbines, which has resulted in an increased interest for combined cycle plants. The basic components of the combined cycle are the gas turbine, heat recovery steam generator (HRSG), and steam turbine. A fuel (usually natural gas) is used to heat up compressed air to a high temperature, which then drives the gas generator and power turbine. The turbine drives the first electrical generator. The flue gas from the gas turbine passes through the heat recovery for steam production which is fed to a steam turbine, which then drives a second electrical generator. Combined cycles are a very efficient processes reaching net electrical efficiencies of 45 – 58 % (LHV), depending on the inlet temperature of the gas turbine and steam conditions.

There are several ways to utilise biomass in a co-firing scenario with a gas turbine combined cycle system that uses natural gas as the primary fuel to generate power. Pamela Spath et al. (1996) have analysed eleven process configurations that integrate biomass fuel into a gas turbine combined cycle system. The biomass system employed can be divided into the following technology categories: combustion and gasification. Biomass gasification is a generally less developed technology than biomass combustion. The configurations analysed can be used in

retrofitting an existing boiler, in increasing the output from an existing power plant, or in the construction of a new power plant. The base case for evaluating each biomass co-firing scenario was a natural gas fired gas turbine combined cycle system consisting of a gas turbine, a heat recovery steam generator (HRSG), and a steam turbine. Different combustion configurations were used to produce additional exhaust gas for the boiler, or a separate boiler was used to produce additional steam for the steam turbine. The gasification configuration describe the production of a low calorific value (LCV) gas for combustion in a gas turbine in addition to a natural gas fired gas turbine or for simultaneous combustion of LCV gas and natural gas in a gas turbine. Simultaneous combustion in a gas turbine is in the following described as co-combustion.

Because gasification is not as well developed as biomass combustion, there is an increased risk associated with the commercialisation of atmospheric or pressurised gasification technologies. Nevertheless, implementing biomass gasification in this way will bring about technology acceptance, which will lead the way to larger scale biomass plants.

Overall, the processing of biomass to electricity will depend on the cost of the fuel, market conditions, the conversion technology used and the income generated by the heat and power produced.

2 CO –COMBUSTION SYSTEMS

Co-combustion of biomass with fossil fuels can be performed in various systems and arrangements. Most common is co-combustion of solid biomass fuels with coal, but it is also interesting to look at co-combustion of biomass fuels with natural gas where the biomass fuels are gasified prior to combustion. The co-combustion can then be performed in the gas turbine and/or in the boiler. Simultaneous combustion of LCV gas and natural gas in a duct burner is also an interesting option.

Several types of biomass gasifiers have been developed over the years including updraft, downdraft, bubbling fluid bed, circulating fluid bed, and entrained flow. Gasifier pressures varies from atmospheric to 2.4 MPa with typical operating temperature in the range of 750°C to 1000 °C.

Gasification of biomass, using air, generates a low calorific value gas (LCV gas) with a lower heating value (LHV) in the range of 4-6 MJ/Nm³. Using steam or oxygen for gasification generate a medium calorific value gas with a LHV in the range of 9-13 MJ/Nm³. Alternatively the LHV can be improved by mixing the LCV gas with natural gas, which contains mainly methane.

2.1 Co –combustion in a gas turbine

A natural gas/biomass combined cycle system utilising low-pressure biomass gasification for turbine fuel is shown in Figure 1. The biomass product gas is fed to the gas turbine in conjunction with natural gas. In order to accommodate this processing concept, the product gas must go through the following additional steps after hot gas cleanup: cooling, quenching and compressing. The gas is cooled by heat exchangers to a point above alkali condensation followed by additional cooling and alkali removals with a conventional quench step. Any alkali that is not entrained in the ash coming out of the cyclone will be removed with the wastewater in the quench step. The heat exchanger is used to recover some of the sensible heat in the gas stream as well as to reduce the amount of quench water required. The quench step can be used to raise the temperature of another stream elsewhere in the process, if required, or to produce steam that is sent to the HRSG resulting in the generation of additional electricity or a combination of these two concepts may be employed.

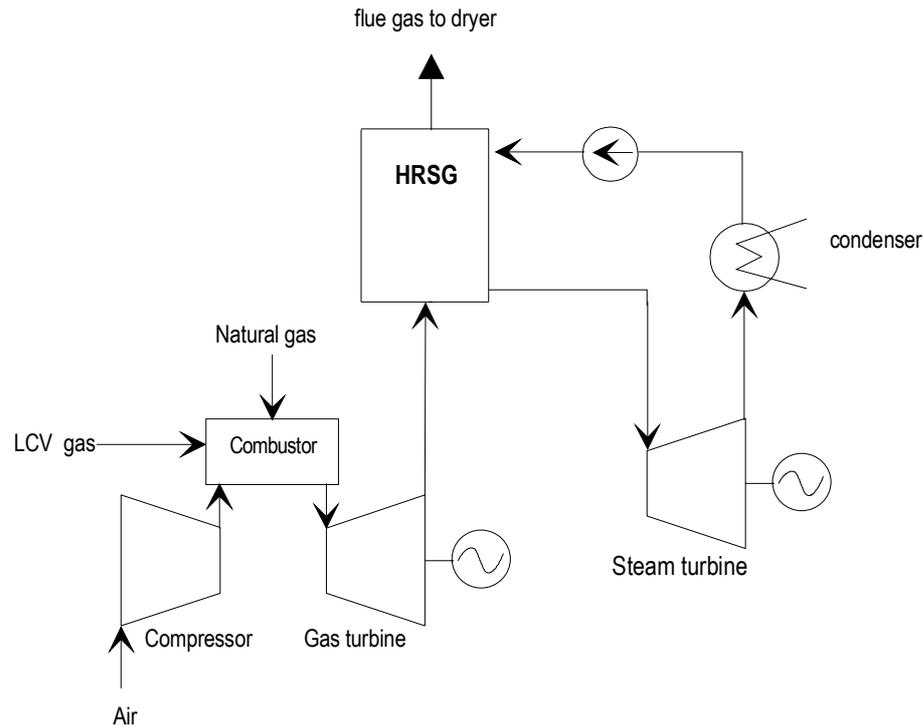


Figure 1. Combined cycle with co-combustion in the gas turbine.

For a natural gas/biomass combined cycle system utilising high pressure biomass gasification for turbine fuel, the product gas exiting the hot gas clean-up section can be fed directly to the gas turbine combustor without any of the additional processing steps, which are required for the low-pressure schematic.

2.2 Co-combustion in a boiler

For co-combustion in a boiler the low calorific gas is either mixed with the gas prior to combustion in a HRSG (Figure 2a), or LCV gas and natural gas are burning as separate flames in the a HRSG (Figure 2b). Gas burners for boiler systems are more robust and the combustion system is technically not as complex as for a gas turbine or a gas engine, hence product gas cleaning is not so critical. Figures 2a and 2b depicts co-combustion in a boiler integrated with a gas turbine. This concept is also feasible for boiler systems for production of hot water or steam.

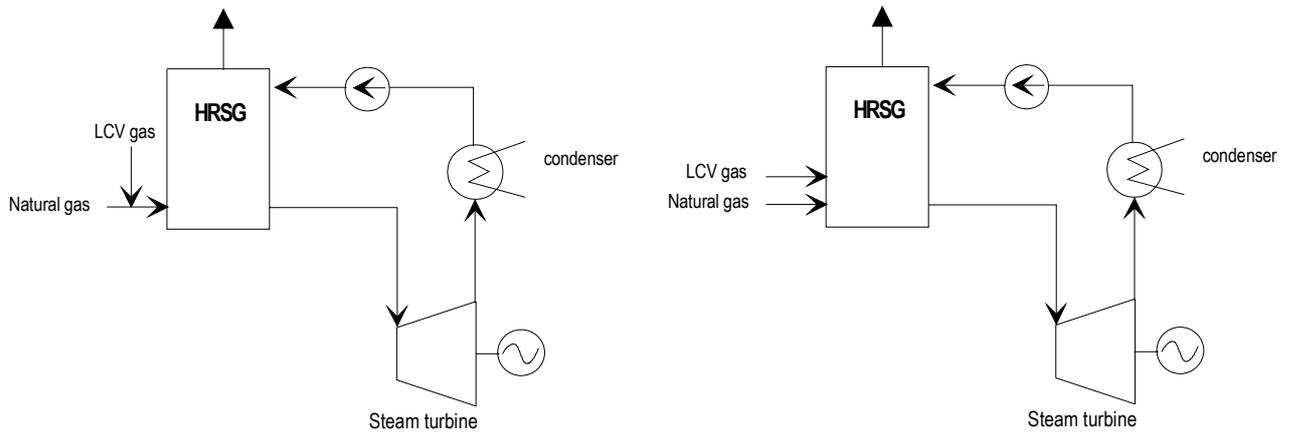


Figure 2. Co-combustion in a boiler:

- a) premixed fuels prior to combustion b) parallel combustion

Another step towards applying biomass in power generation could be combined combustion of natural gas and low calorific value gas (LCV) using the LCV gas as a re-burning fuel. Re-burning is a technology primarily relevant for large-scale applications since this system requires additional equipment inside the boiler. The aim of re-burning is to reduce the emissions of NO_x . The configuration shown in Figure 2b can also be arranged for re-burning.

2.3 Co-combustion in a duct burner

Simultaneous combustion of LCV gas and natural gas in a duct burner prior to the boiler is another option for co-combustion. The gases are either premixed prior to combustion in the duct burner (Figure 3a) or injected separately into the duct burner (Figure 3b).

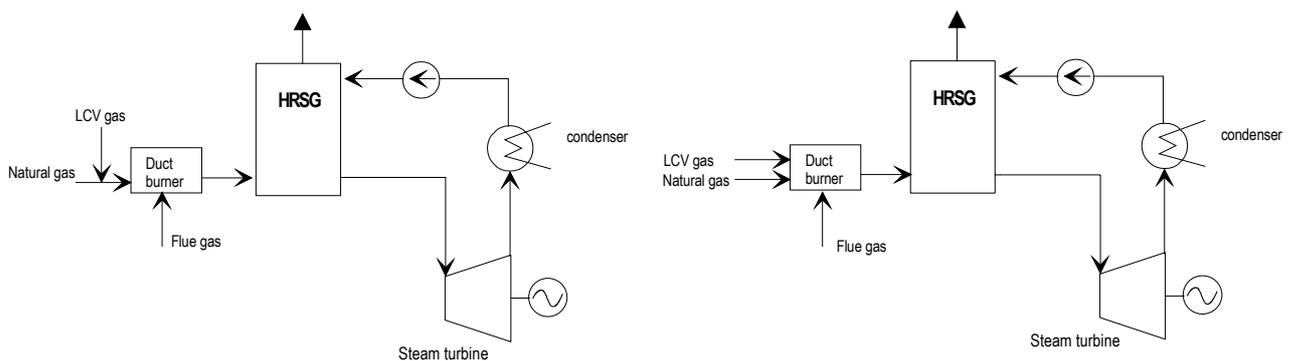


Figure 3. Co-combustion in a duct burner:

- a) premixed fuels prior to combustion b) parallel combustion

For a combined cycle configuration the hot turbine exhaust gas can be used as oxygen source for the combustion process in the duct burner as shown in Figure 4.

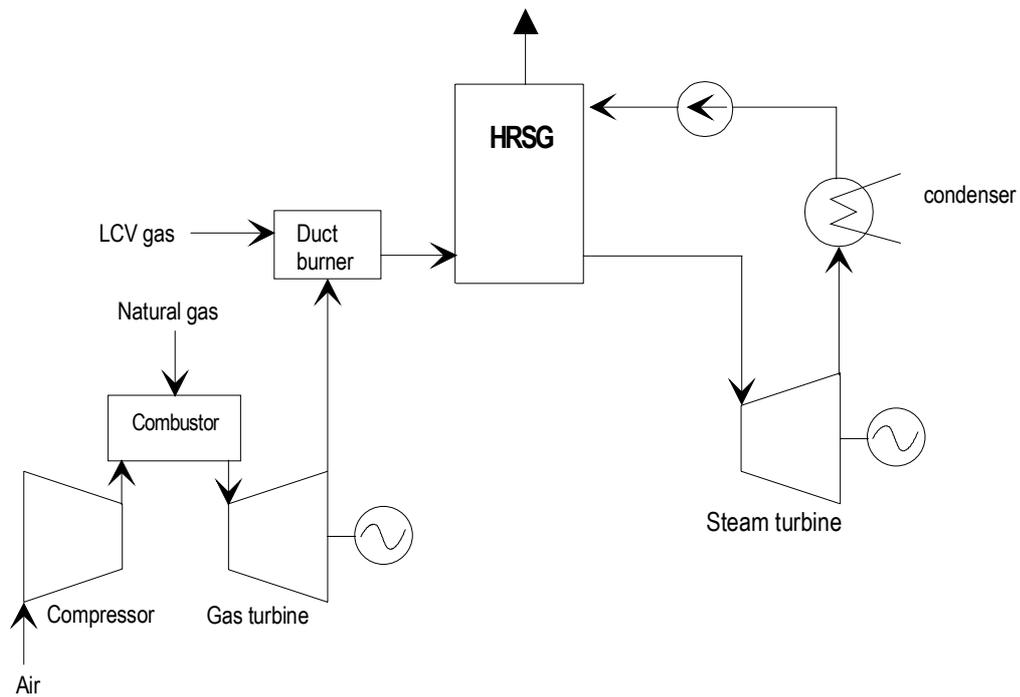
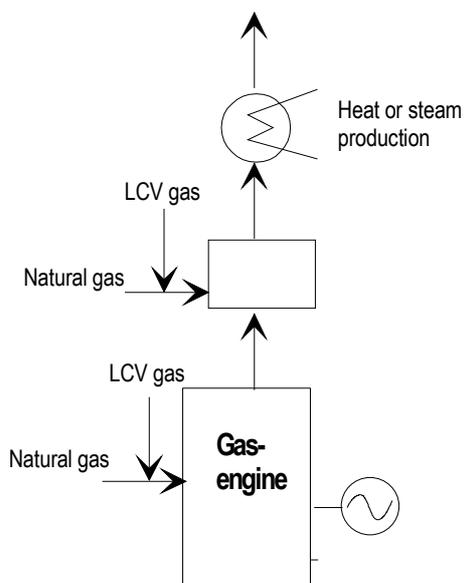


Figure 4. Combined cycle with hot turbine exhaust gas as oxygen source in a duct burner.

2.4 Co-combustion in a gas engine

In a gas engine system for CHP production co-combustion of product gas and natural gas may include both fuel gas for the gas engine and/or a fired heater downstream of the gas engine, as depicted in Figure 5. For fuel flexibility the gas engine and the fired heater should also be operated on natural gas only.



One of the major advantages of the gas engine over the gas turbine is that the fuel can be introduced at atmospheric pressure or even below, whereas the gas turbine require fuel pressure by boosting significantly above the highest compressor pressure.

Figure 5. Co-combustion in gas engine system for CHP production.

3 LOW CALORIFIC VALUE GASES

Commercially, biomass gasification is in use for replacing oil and natural gas in boilers or other furnaces. The applications are based on atmospheric pressure gasification and the incentives for using this technology are often based on the availability of low cost biomass to replace more expensive fuel in special situations or avoiding some environmental investments or taxes.

Pressurised gasification of biomass is mainly of interest in gas turbine integrated systems. Gasification is usually carried out at such a pressure, that after the cleaning process and a possible cooling stage the fuel gas can be fed directly into the combustion chamber of the gas turbine without additional compression. Gasifiers operated at atmospheric pressure must include a gas compressor stage before the product gas is led into the combustion chamber of the gas turbine.

Gasification of biomass converts the solid material into a gaseous fuel as the main product and ash and non-reacted char as rest products. The gas composition is influenced by the quality of the feedstock, gasifier process parameters and the gasifying agent. Air-blown gasifiers produce a gas where the major combustible components are CO, H₂ and CH₄. The lower heating value (LHV) for these gases is typically in the range of 4-6 MJ/Nm³, hence the gases are characterised as low calorific value gases (LCV) gases. The use of oxygen and/or steam as a gasifying agent reduces the fraction of N₂ in the gas, which gives a medium calorific value gas with a lower heating value in the area of 9-13 MJ/Nm³.

Commercial gasification technologies can be divided in two major groups; fixed bed reactors and fluid bed reactor. In addition technologies such as multiple heart, rotating reactor etc are found used. The characteristics of the different technologies are summarised in Table 1.

Table 1. Typical characteristics for air-blown gasifiers (Bridgwater, 1995).

Technology	Process temp.(°C)	Gas exit temp. (°C)	Tar	Particles	Turn-down	Max cap. (t/h)
Fixed bed						
<i>Downdraft</i>	1000	800	very low	moderate	good	0,5
<i>Updraft</i>	1000	250	very high	moderate	good	10
<i>Cross-current</i>	900	900	very high	high	fair	1
Fluid bed						
<i>Single reactor</i>	850	800	fair	high	good	10
<i>Fast fluid bed</i>	850	850	low	very high	good	20
<i>Circulating bed</i>	850	850	low	very high	good	20
<i>Entrained bed</i>	1000	1000	low	very high	poor	20
<i>Twin reactor</i>	800	700	high	high	fair	10
Moving bed						
<i>Multiple heart</i>	700	600	high	low	poor	1
<i>Horizontal moving bed</i>	700	600	high	low	fair	1
<i>Sloping heart</i>	800	700	low	low	poor	0.5
<i>Screw/auger kiln</i>	800	700	high	low	fair	0.5
Other						
<i>Rotary kiln</i>	800	800	high	high	poor	2
<i>Cyclone reactors</i>	900	900	low	very high	poor	1

In addition to tar and particles the product gas also contains alkalis and nitrogen components (ammonia and hydrogen cyanide) originating from the nitrogen in the biomass.

As can be observed from Table 2 the quality of the raw product gas in terms of the content of tar and particles is very much depending of the gasification process. Selection of a suitable gasification technology will depend on feed stock quality, capacity, specified turn-down ratio and also which system is planned for utilising the product gas. In general, for low capacity systems, an updraft fixed bed reactor is selected when the product gas is aimed at heat production only while a downdraft fixed bed reactor is more suitable for systems including power generation. Fluid bed reactors are generally selected for larger capacities and for plants where the feedstock quality varies.

Due to content of impurities in the raw product gas cleaning is often necessary in order to meet both the environmental emission limits and on the other hand fuel quality specifications in order to be acceptable as a fuel for gas turbines, gas engines or gas burners.

The sulphur content of biomass is typically very low and sulphur removal is therefore usually not required.

During the gasification a significant part of the fuel nitrogen may be converted to ammonia, hydrogen cyanide and other nitrogen compounds. Catalytic high-temperature gas cleaning is one alternative to remove ammonia and other nitrogen compounds from the gas before burning it in the gas turbine.

One of the major problems is the formation of tars. Most of the tar components are not problematic, if a hot gas filtration is applied. During the gasification high gasification temperature and the catalytic effects of the bed material promote the cracking of tars.

3.1 Characteristics of LCV gas

The characterisation of fuel gases includes the following properties:

- Gas composition
- Lower (higher) heating value
- Adiabatic flame temperature
- Flammability limits
- Laminar burning velocity
- Wobbe index

Normally these properties are stated as values for pure fuel gases. However, in premixed combustion systems the equivalent values for the mixture of fuel gas and air are of interest.

In addition, also flame stability and structure of laminar flames are important parameters for evaluating the combustion characteristics of gaseous fuels.

3.1.1 Composition of LCV gas

The major combustible components in the product gas from air blown gasification of biomass are hydrogen, carbon monoxide and methane. In addition the product gas will contain small amount

of higher hydrocarbons. The gas is diluted with inert components, nitrogen from the air, carbon dioxide as a combustion product and water vapour. The range in composition of product gas from biomass gasification is given in Table 2.

Table 2. Typical range in product gas composition.

Component	Fraction (% vol.)
CO	15 – 25
H ₂	10 – 20
CH ₄	1 – 5
Hydrocarbons	< 1
N ₂	45 – 55
CO ₂	10 - 20

3.1.2 Lower heating value

The heating value for a gaseous fuel is specified as the higher heating value (HHV) or the lower heating value (LHV). Heating values are normally given as MJ/Nm³ or MJ/kg of fuel.

The higher heating value or gross heating value is defined as the total heat obtained from combustion of a specified amount of fuel and its stoichiometrically correct amount of air both at 25 °C when combustion starts, and the combustion products being cooled down to 25 °C before the heat release is measured.

The lower heating value or net heating value is defined as the higher heating value minus the latent heat of vaporisation of the water formed by combustion of the hydrogen in the fuel. For fuels with no hydrogen, the lower and higher heating value will be equal.

In this report, the heating value of referred fuel will be in terms of the lower heating value.

For gas mixtures the lower heating value is calculated according to the following expression:

$$\text{LHV} = \sum_i y_i \text{LHV}_i$$

where y_i : volume fraction of combustible gas component (-)

LHV_i : lower heating value of combustible gas component (MJ/Nm³)

The lower heating values for the combustible components in LCV gases are:

CO: 12.6 MJ/Nm³

H₂: 10.7 MJ/Nm³

CH₄: 35.8 MJ/Nm³

Based on typical LCV gas compositions as described in Table 2 the lower heating values for LCV gases are in the range of 4-6 MJ/Nm³.

3.1.3 Flame temperature

Flame temperature is important because it affects not only the combustion rate, but also the amount of heat that can be recovered in a given furnace.

Adiabatic flame temperature at stoichiometric conditions for the combustible components in LCV gases (carbon monoxide, hydrogen and methane) are in the range of 2220 – 2320 K. Inert component in a gas mixture will act as a heat sink in a combustion process and hence reduce the adiabatic flame temperature.

For LCV gases having a total content of inert in the range of 50-60% (vol.) adiabatic flame temperatures will be in the range of 1700 K depending on gas composition.

3.1.4 Flammability limits

The flammability limits for a specified fuel defines the maximum and minimum volumetric concentration (% vol.) of fuel in a fuel-air mixture that propagates a self-sustaining flame after ignition. The maximum and minimum concentrations are referred to as the upper and lower flammability limit respectively.

Flammability limits for major combustible components in LCV gases are given in Table 4.

Table 4. Flammability limits for gases.

Component	Lower flammability limit (% fuel in air)	Upper flammability limit (% fuel in air)
CO	12.5	74.2
H ₂	4.0	74.2
CH ₄	5.0	15.0

Flammability limits for gas mixtures are either determined by experiments or by using empirical equations such as Le Chatelier's law which estimate the flammability limits as a function of the volumetric fraction and flammability limits for gas components in the mixture.

As the major combustible components in LCV gases are hydrogen and carbon monoxide the flammability limit typically is in the range of 20 – 60 % (vol. fuel in air).

3.1.5 Wobbe index

The Wobbe index or Wobbe number is defined as the heating value (lower or higher) divided by the square root of the relative density of the fuel gas

$$\text{Wobbe index} = \frac{\text{LHV}}{\sqrt{\rho_{\text{rel}}}}$$

where LHV: lower heating value (MJ(Nm³))

Ψ_{rel} : the relative density of the fuel compared with air (-)

The Wobbe index is primarily used to evaluate interchangeability of fuel gases with respect to equal heat input rate and fluid handling capability of burners, piping, valves, controls etc.

With a lower heating value in the range of 4-6 MJ/Nm³ and a relative density in the same area as air the Wobbe index for LCV gases will typically be equal to the heating value.

3.1.6 Laminar burning velocity

The burning velocity is an important parameter used to characterise premixed mixtures of fuel and oxidiser (air). The laminar burning velocity is the rate of propagation of a combustion wave through quiescent gas or as the velocity at which unburned gas move through the combustion wave in the direction normal to the wave surface.

The laminar flame velocity for single gas components premixed with air is well documented and Table 5 show typical values for the major components in LCV gases at stoichiometric conditions

Table 5. Laminar burning velocities at stoichiometric conditions (from Glassmann, 1987).

Component	Laminar burning velocity (cm/s)
H ₂	170
CO	28.5 ¹⁾
CH ₄	38.3

¹⁾ For equivalence ratio = 1.1.

It should be noticed that laminar burning velocities referred in literature show relatively large variations.

The laminar burning velocity for LCV gases will be controlled by the gas composition. As can be observed from Table 5, the content of hydrogen will strongly affect the burning velocity of a gas mixture. However, the content of inert in LCV gases will cause a significant reduction of the burning velocity.

3.1.7 The laminar flame structure

Illustration of the laminar flame structure is made by a simple example: a flame anchored on a top of a single Bunsen burner as shown in Figure 5. The fuel gas entering the burner induces air into the tube from its surroundings. As the fuel and air flow up the tube, they are mixed and before the top of the tube is reached, the mixture is completely homogeneous. The flow velocity in the tube is considered to be laminar and the velocity across the tube is parabolic in nature. Therefore, the flow velocity near the tube wall is very low. This low flow velocity is a major factor in stabilising the flame at the top.

The dark zone designated in Figure 5 is simply the unburned premixed gas before they enter the area of the luminous zone where reaction and heat release take place. The luminous zone is less than 1 mm thick. The luminous zone is that portion of the reacting zone in which the temperature is the highest. Much of the reaction and heat release take place in this zone. The colour of the luminous zone changes with fuel-air ratio.

It is possible to characterise the flame as consisting of three zones: a preheat zone, a reaction zone and a recombination zone. The general structure of the reaction zone is made up of early pyrolysis reactions and a zone in which the intermediates CO and H₂ are consumed. The actual characteristics of the reaction zone and the composition changes throughout the flame are determined by the convective flow of unburned gases toward the flame zone and the diffusion of radicals from the high temperature reaction zone against the convective flow into the preheat region. H atoms dominate this diffusion, so a substantial amount of HO₂ forms in the lower-temperature preheat region. At these temperatures, the chain branching step (H+O₂→OH+O) does not occur, then, this HO₂ forms hydrogen peroxide. Since the peroxide does not dissociate at the temperature in the preheat zone, it is convected into the reaction zone where it forms OH at the higher temperatures that prevail there. Owing to this large concentration of OH relative to O and H in the early part of the reaction zone, the OH attack on the fuel is the primary reason for the fuel decay.

The latter part of the reaction zone forms the region where the intermediate fuel molecules are consumed and where CO is converted to CO₂. The CO conversion results in the major heat release in the system and is the reason for the peaks of the rate of heat curve near the maximum temperature. This curve falls off quickly because of the rapid disappearance of CO and the remaining fuel intermediates.

The recombination zone falls into the burned gas or post-flame zone. Although recombination reactions are very exothermic, the radicals recombining have such low concentrations that the temperature profile does not reflect this phase of the overall flame system.

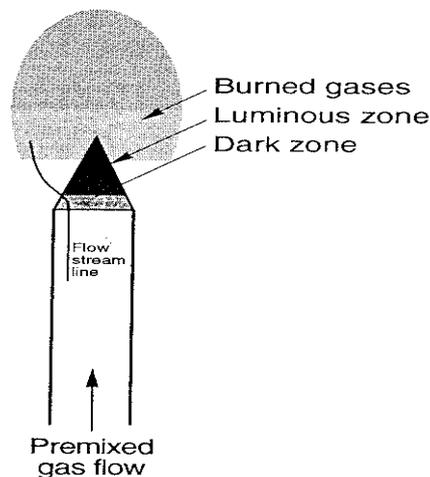


Figure 5 General description of laminar Bunsen burner flame.

The CO-H₂-O₂ is a system of combustion that is well known. The system includes 11 active species (H₂, O₂, CO, H₂O, CO₂, O₃, H, OH, H₂O₂, HO₂ and HCO) and well over 60 reactions (Wetsbrook et al., 1984). The presence of methane in the LCV gas complicates the modelling of their combustion chemistry considerably. For combustion of methane itself, the number of species reaches 30, and the number of elementary steps is in the range of several hundred. In our simulations, a reduced kinetic will be used, from 176 reactions to 8 reactions (Kaufman, 1983).

Flame stability is the most complex problem of LCV gas combustion. Flame stability may be achieved by generating a zone in the combustor in which the flow velocity is equal to the burning velocity or, in a stirred zone, with intense back mixing of products into the fresh mixture or by pre-heating reactants.

4 CALCULATIONS OF PROPERTIES FOR MIXTURES OF LCV GASES AND NATURAL GAS

The composition of natural gas varies depending on the geographical origin. The main combustible component in natural gas is methane and thus the calculations presented in this section are based on mixtures of a selected LCV gas and methane.

Calculations include the following parameters:

Lower heating value (LHV)
 Wobbe index
 Flammability limits
 Flame temperature
 Laminar burning velocity

4.1 Composition of gas mixtures

The composition of the LCV gas is selected based on typical range in composition as referred to in Table 2. The LCV gas is mixed with different fractions of methane (25%, 50%, and 75%). Calculations also include 100% methane gas as a reference to natural gas. Gas mixture compositions are shown in Table 6.

Table 6. Gas mixture composition

Components	LCVG	LCVG + 25% CH ₄	LCVG + 50% CH ₄	LCVG + 75% CH ₄	100% CH ₄
CH ₄ (%vol.)	4	28	52	76	100
CO (%vol.)	16	12	8	4	0
H ₂ (%vol.)	10	7.5	5	2.5	0
N ₂ (%vol.)	53	39.75	26.5	13.25	0
CO ₂ (%vol.)	17	12.75	8.5	4.25	0

4.2 Lower heating value and Wobbe-index

The LHV, density, and Wobbe-index for each gas mixture are given in Table 7.

Table 7 LHV, density and Wobbe-index for the gas mixtures

Different mixtures	LHV (MJ/Nm ³)	Density (kg/dm ³)	Wobbe-index
LCV gas	4.54	1.236	4.35
75% LCV gas + 25% CH ₄	12.37	1.107	13.37
50% LCV gas + 50% CH ₄	20,21	0.977	23.26
25% LCV gas + 75% CH ₄	28,04	0.847	34.68
100% CH ₄	35,88	0.717	47.12

The values given in Table 7 are specified for pure fuel gas only. However in premixed combustion systems, the lower heating value and Wobbe index for the premixed mixture of fuel gas and air is also of interest. Figure 6 and 7 depict the lower heating value and Wobbe index as a

function of the equivalence ratio Φ . The Wobbe index is calculated based on the lower heating value and relative density of the premixed mixture of fuel gas and air.

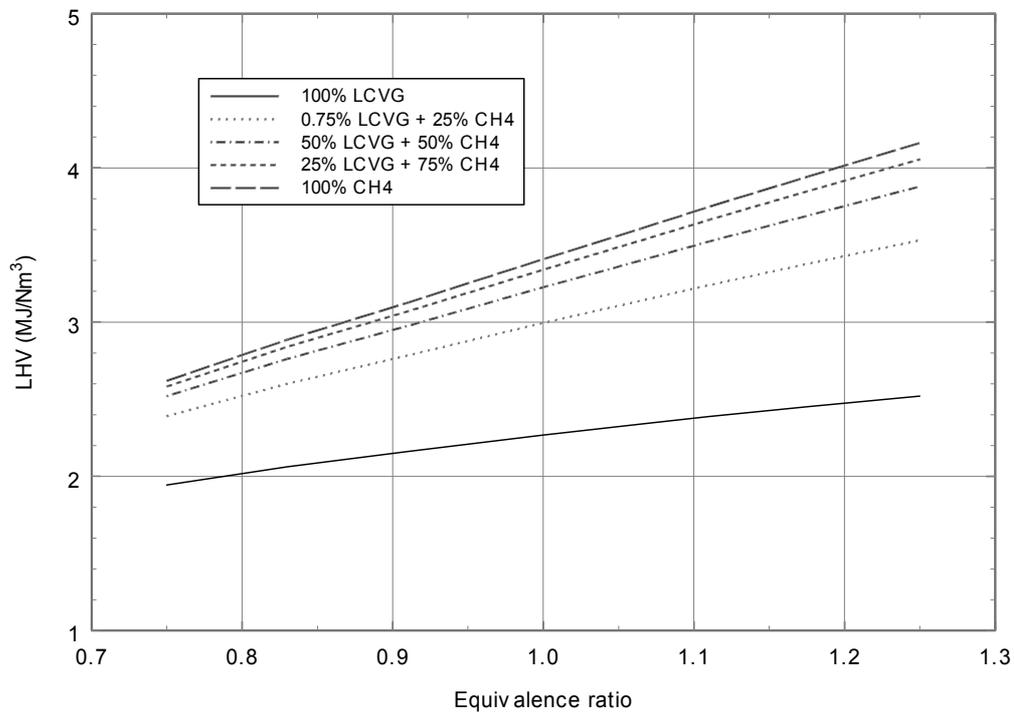


Figure 6. Lower heating value as a function of equivalence ratio (Martin-Blanc, 1997).

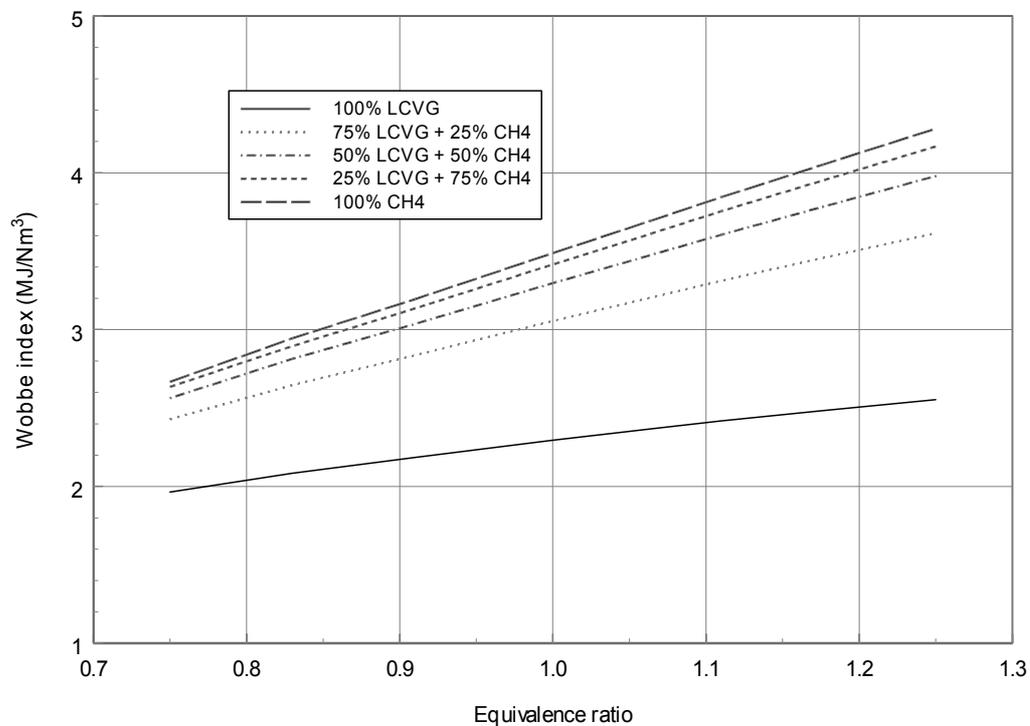


Figure 7. Wobbe index as a function of equivalence ratio (Martin-Blanc, 1997).

4.3 Flammability limits

Le Chatelier's law states that a mixture at the lower limit of flammability mixed with other mixtures, which are also at the lower limits of flammability, will yield a resulting mixture at the lower limit of flammability. A formula for estimating the lower limit of any mixture of combustible gases that obeys Le Chatelier's law is:

$$L = \frac{100 \%}{\sum_{i=1}^n \frac{C_i}{L_i}} \quad U = \frac{100 \%}{\sum_{i=1}^n \frac{C_i}{U_i}}$$

where C_i is the volumetric concentration of the i^{th} combustible gas, L_i is the lower limit of flammability for the i^{th} combustible, U_i is the upper limit of flammability for the i^{th} combustible gas and n is the total number of combustible gases present.

A practical way to determine if Le Chatelier's law is applicable to a given mixture is to look at the work by Coward and Jones (1952). It contains a compilation of many mixtures, and indicates the applicability of Le Chatelier's law (Figure 8). When inert gases such as nitrogen and carbon dioxide are present, the mixtures must be dissected into binary mixtures, each containing a single flammable gas and part, all or none of an inert, and then Figure 8 can be used. It is important to know that the mixture of gases must be without air.

Applying Le Chatelier's law in recombining the several mixtures may complete the last step in the calculation of the flammability limits of the mixture.

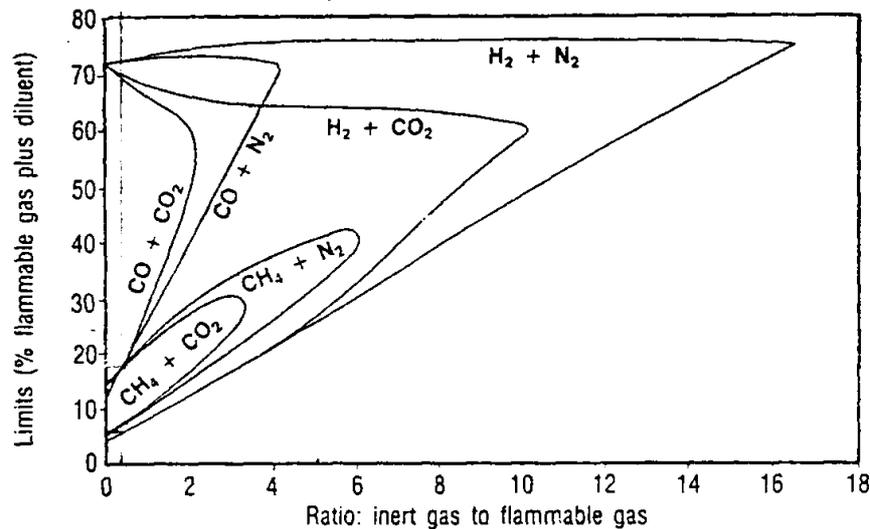


Figure 8. Flammability limits for different ratio inert gas to flammable gas (from Coward and Jones, 1952).

Heffington and Gaines (1981) describe a graphical method for flammability calculations based on flammability limits found in Figure 8. The results from this graphical method applied on the gases treated here are depicted as flammability limits as a function of methane content in Figure 9.

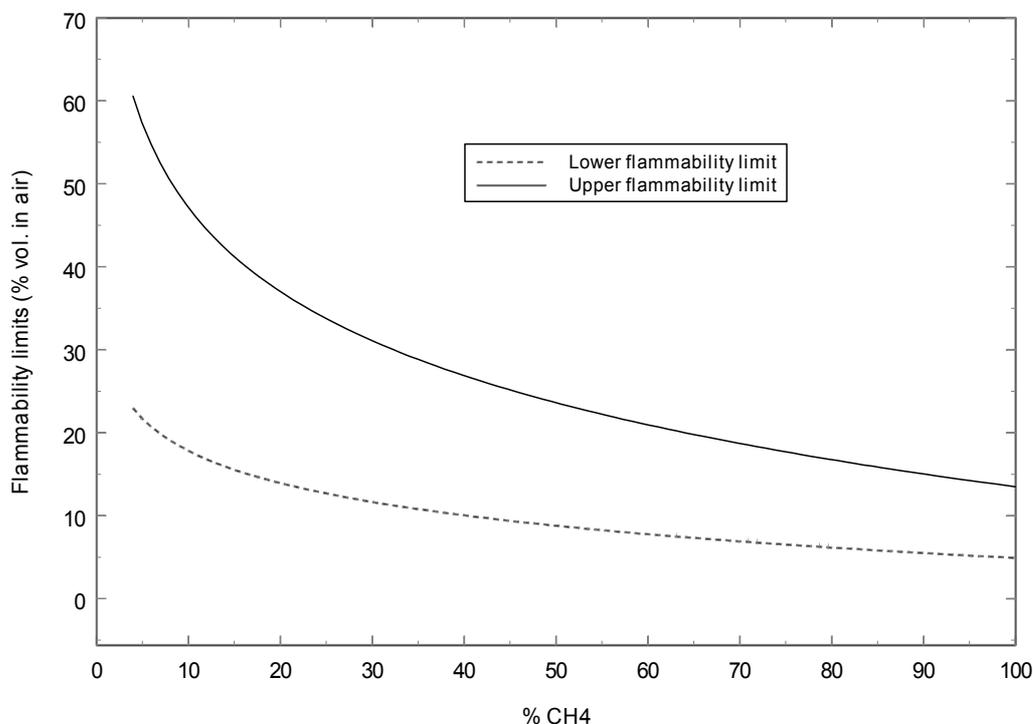


Figure 9. Flammability limits as a function of methane content in the gas mixture (Martin-Blanc, 1997).

4.4 Flame temperature

Adiabatic flame temperatures are calculated using the equilibrium program included in CHEMKIN software package. This program is based on the Stanjan –III program which is integrated in the CHEMKIN program.

The results from the equilibrium calculations in terms of adiabatic flame temperature for the actual gas mixtures are given in Figure 10 as a function of the equivalence ratio.

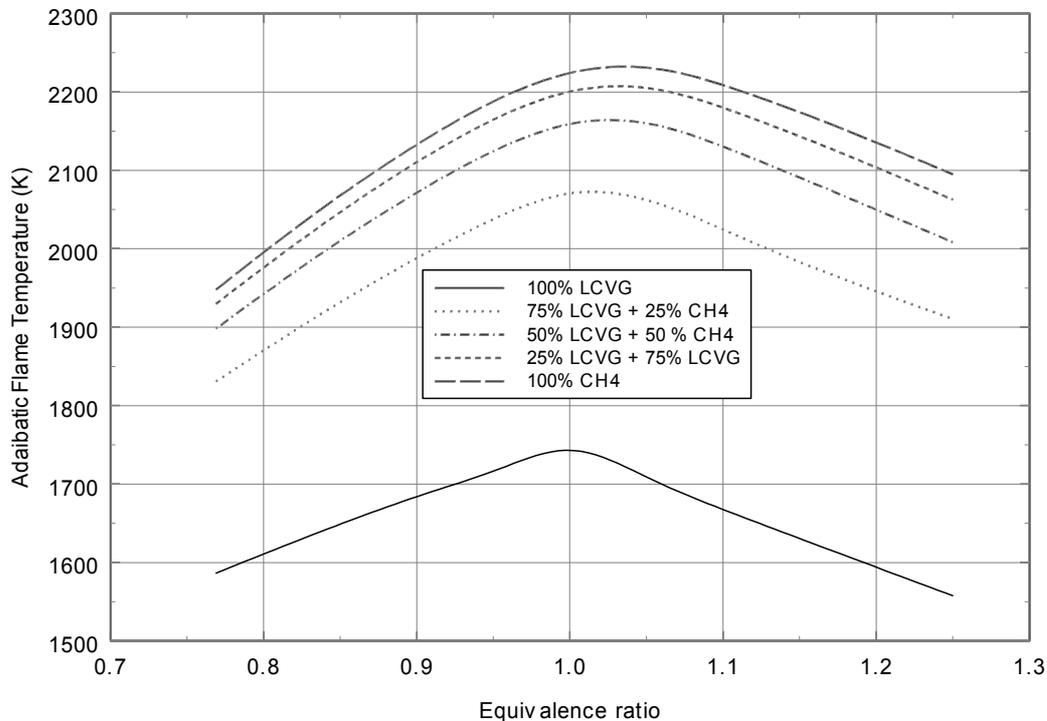


Figure 10. Adiabatic flame temperature as a function of equivalence ratio (Martin-Blanc, 1997).

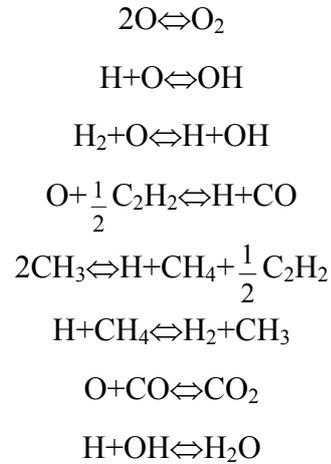
4.5 Laminar burning velocity

Laminar burning velocities are calculated using the Premix code in CHEMKIN-II which is a program for modelling steady laminar one-dimensional premixed flames. Reaction mechanisms used in the Premix code must include the combustible components in LCV gases, H_2 , CO and CH_4 . Reaction mechanisms for methane include also hydrogen and carbon monoxide as intermediate species. As an alternative to using a detailed reaction scheme for methane, which may include more than 170 reactions, a reduced mechanism developed by Chang and Chen was applied.

The eight-step mechanism from Chang and Chen is derived for combustion of methane, but the carbon monoxide and hydrogen are produced and consumed in the combustion processes of methane as intermediate species. This mechanism is developed from a skeleton mechanism, which included 176 reactions (GRI-MECH).

Reduced kinetic models are derived analytically from skeleton mechanisms by making order of magnitude considerations to justify that species may be assumed to be steady state and some elementary reactions may be assumed to be in partial equilibrium thus simplifying the problem significantly.

The eight-step mechanism contains the reactions:



where the production rates are complicated algebraic equations described in subroutines taken from Chang and Chen and appended the Premix CHEMKIN-II computer code. This reduced mechanism includes 12 species: H_2 , H , O_2 , OH , H_2O , CH_3 , CH_4 , CO , CO_2 , C_2H_2 , and N_2 .

The species assumed as steady states are C_2H_5 , C_2H_6 , C_2H_3 , C_2H , CH_3O , $\text{CH}_2(\text{S})$, CH_2OH , C , HCO , C_2H_4 , HCCOH , CH , HCCO , CH_3OH , CH_2CO , CH_2 , H_2O_2 , CH_2O , and HO_2 .

Calculated laminar burning velocities are depicted in Figure 11.

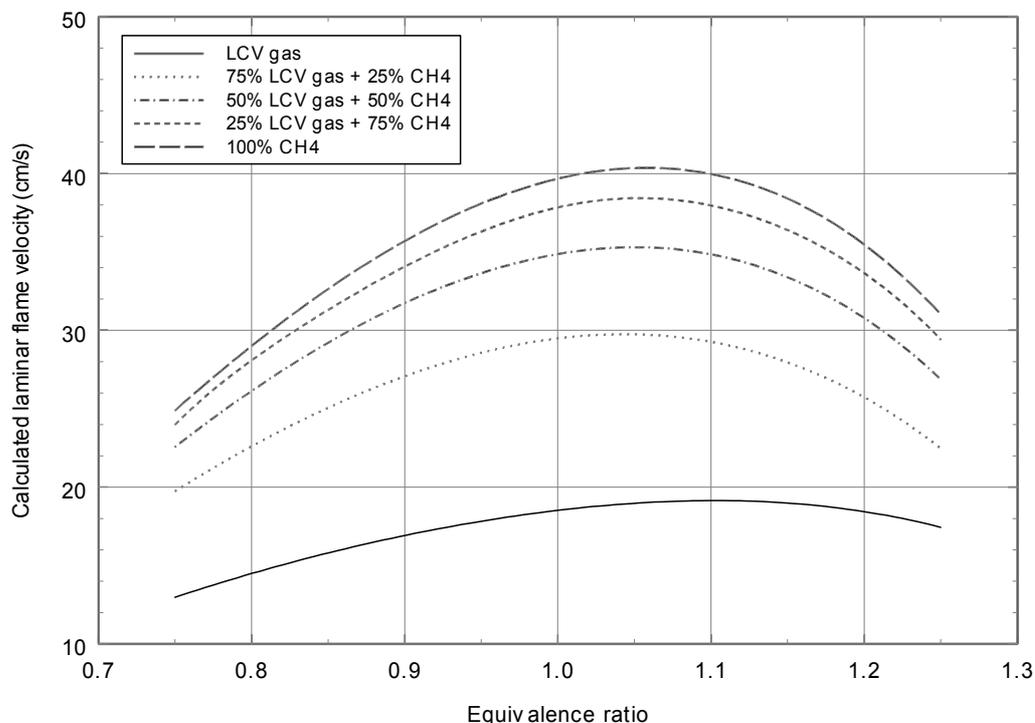


Figure 11. Calculated laminar burning velocity as a function of equivalence ratio (Martin-Blanc, 1997).

5 DISCUSSIONS

5.1 Lower heating value and Wobbe-index

As can be seen from Table 7 the lower heating value and Wobbe index for the LCV gas is about 10% of the same values for methane. However from Figure 6 and 7 which describe the lower heating value and Wobbe index for a mixture of fuel gas and air, one can observe that the values for LCV gas are in the range of 55 – 75% of the values for methane. These values increase with increasing values of the stoichiometric ratio.

Figure 6 and 7 also demonstrate the increasing demand for combustion air as the methane content in the gas increases. The lower heating value for LCV gas is 4.54 MJ/Nm^3 and the similar value at a stoichiometric ratio of 0.9 is 2.2 MJ/Nm^3 , a decrease of about 50%. For methane with a lower heating value of 35.88 MJ/Nm^3 a decrease of about 90% is observed compared to the lower heating value at a stoichiometric ratio of 0.9. Similar effect is also observed for the Wobbe index.

The difference in Wobbe index between LCV gas and methane show that it is not straight forward to replace methane, or natural gas, with LCV gas. Enrichment of LCV gas with methane will however improve the interchangeability and as can be seen from figure 6 and 7 the relative increase in lower heating value and Wobbe index is most significant when adding 25% methane into the LCV gas compared to further increasing the methane content.

5.2 Adiabatic temperature

The calculated values of adiabatic flame temperatures show similar effect of increasing methane content in the gas mixture as the lower heating value and Wobbe index. As can be seen from Figure 10 the relative increase in adiabatic flame temperature is most significant when adding 25% of methane to the LCV gas while the relative increase in temperature is less noticeable for further enrichment of methane in the gas. The differences in temperature between LCV gas and methane are increasing as the stoichiometric ratio is increased.

Maximum adiabatic flame temperature for the LVC gas is low, about 1750 K, compared to methane with a maximum temperature of about 2200 K.

Figure 10 also show a transition in the equivalence ratio for maximum flame temperature from 1 for the LCV gas to the area of 1.05 for methane. Many fuels have a maximum flame temperature for equivalence ratios in the range of 1.0 to 1.1. This is due to the lower mean specific heats of the richer products. As the methane concentration in the gas is increased the volumetric fraction of the products will become more significant compared to inert components in the fuel gas and thus the flame temperature will show a peak in the fuel rich area.

5.3 Laminar burning velocity

As can be observed from Figure 11 the maximum laminar burning velocity (S_L°) for the LCV gas is about 19 cm/s. The laminar flame velocity increases with increasing methane content in the gas similar to the lower heating value and the flame temperature, that is, the relative increase in burning velocity is most significant for 25% methane added into the LCV gas, while further addition of methane has less significant effects. For pure methane, maximum burning velocity is found to be 40.4 cm/s.

Figure 11 indicates a change in the equivalence ratio for maximum laminar burning velocity. For LCV gas the peak value is found to occur at an equivalence ratio of approximately 1.1 while the peak value for methane and also for mixtures of LCV gas and methane is found for an equivalence ratio of about 1.05.

According to Glassman (1987) maximum laminar burning velocity for methane is 44.8 cm/s at an equivalence ratio of 1.08 and similar values for hydrogen and carbon monoxide are 325 cm/s at an equivalence ratio of 1.80 and 52.0 cm/s at an equivalence ratio of 2.05 respectively. Figure 11 shows that the methane content in the gas will be determining in terms of profile of the laminar burning velocity related to the equivalence ratio.

6 CONCLUSION

In light of the current marked conditions (i.e. low natural gas prices, utility deregulation, etc.), biomass-co-firing projects are among the best opportunities for near term growth in biomass utilisation. Electricity generation from natural gas is expected to increase until 2010. The expected increase is in gas turbines and combined cycle plants.

Different technologies for co-combustion with biomass fuels and natural gas where the biomass fuels are gasified prior to combustion have been presented. These technologies include; co-combustion in a gas turbine, co-combustion in a boiler, co-combustion in a duct burner and co-combustion in a gas engine. Combined firing of natural gas and low calorific gas (LCV) using the LCV gas as re-burning fuel is also mentioned, but it is only relevant for large scale application.

Calculations of lower heating value and Wobbe index for the gas mixture described in Table 7 show a linear increase for these properties proportional to the methane content in the gas mixture. Based on values for fuel gas these values show that conversion from natural gas to LCV gas will cause a relatively large de-rating of the combustion system unless this is compensated in terms of larger volume flows. Similar values for premixed mixtures of fuel gas and air show that in premixed combustion systems operating in the fuel lean area conversion from natural gas to LCV gas induces a de-rating of approximately 30%.

Mixing methane (natural gas) into the LCV gas will improve the gas quality in terms of higher values for the lower heating value and the Wobbe index. The effect of methane enrichment is found most significant for a mixture of 75% LCV gas and 25% methane while the relative effects of further increasing fraction of methane is less noticeable. Conversion from methane to a mixture of 75% LCV gas and 25% methane reduces the de-rating down to approximately 10%.

Calculations of adiabatic flame temperature also demonstrate the relative effects of increased addition of methane into LCV gas. The difference in flame temperature between methane and LCV gas at an equivalence ratio of 1.0 are in the range of 450 – 500 K, while the similar difference between methane and a mixture of 75% LCV gas and 25% methane is about 150 K.

Mixing methane into LCV gas will improve the heat transfer properties which is of importance in fired heating units or boilers. Increased flame and flue gas temperature increases temperature differences and hence reducing the heat transfer area. Increased methane content will also increase thermal radiation from the flame which also improves the total heat transfer.

Maximum laminar burning velocity of LCV gas at stoichiometric conditions is found to be about 50% of the similar value for methane while the corresponding value for a mixture of 75% LCV gas and 25% methane is about 30%. Again the effect of 25% addition of methane is more substantial compared to the effect of further addition of methane. Although combustion systems in general operates with turbulent combustion the laminar burning velocity provides qualitative information describing the reactivity and stability properties of fuels and the results as shown in Figure 11 clearly prove that addition of a relatively small volumetric fraction of methane improves the characteristics of LCV gases.

In view of existing combustion equipment, LCV gases can be and are utilised in gas burners for boilers or other heat production units. Mixtures of natural gas and LCV gas will however stabilise the gas quality and hence the performance of the gas burner will not be so sensitive as related to variations in the LCV gas quality. For gas turbines, the main challenge in combustion of LCV

gases are the tars, particles and alkalis in gases produced from gasification of biomass. As for gas burners co-combustion of LCV gas and natural gas will reduce problems related to stability and variations in the product gas quality from the gasifier and the increased lower heating value and Wobbe index will reduce the need for extensive modification of piping and burner design. Gas reciprocating engines are commercially available for gases with medium to high calorific values, such as landfill gas, coke oven gases etc. Gas engines for LCV gases are dominated by one manufacturer, Jenbacher. By mixing LCV gas with natural gas a mixture can be adjusted to meet the specifications for gas engines available for medium calorific value gases. Landfill gas has a lower heating value in the range of 17-20 MJ/Nm³ and a similar heating value can be obtained with a mixture of 60% LCV gas and 40% methane based on the LCV gas composition used for calculations in this report.

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