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THE COMBUSTION AND THE THERMAL-ENERGETIC BEHAVIOR OF AN OIL-FIRED CONDENSING BOILER

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Summary

The purpose of this Master’s Thesis is to analyze an oil-fired condensing boiler (Vaillant IcoVit 246-7) describing the aspects concerning the combustion (atomization and droplet combustion theory), condensation latent heat recovery from the combustion products (Colburn and Heugen theory) and the heat transfer phenomena occurring in the device.

Furthermore will be described the regulatory aspects that defines calculation methods and laboratory tests to assess the energetic performance of a boiler. In particular: the combustion efficiency assessment for tuning the boiler’s burner; the laboratory tests used for the determination of the useful efficiency and the heat losses of the boiler that will be used in the calculation of the boiler’s seasonal efficiency.

This Master’s Thesis was made within the Erasmus Program, under the supervision of Prof. Klaus Sommer, of the Institut für Technische Gebäudeausrüstung, Fakultät für Anlagen, Energie-und Maschinensysteme, Fachhochschule Köln.

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Introduction

The residential space heating is one of the most important contributors on the energy consumption in the European Union. It was estimated that the household sector is responsible of around the 25% of the final energy consumption in the EU-27 area, contributing to greenhouse emission for around the 8,5% [1].

In the residential buildings the most of the energy is used to space heating and hot domestic water production (80%) [2]. Thus, improving the domestic space heating boilers' technology is a fundamental objective to achieve the energetic and environmental goals that the European Union has recognized by ratifying the Kyoto Protocol.

The highest performances in the field of combustion space heaters are achieved by the Condensing Boilers.

Commonly, condensation boilers are fed with natural gas, but when is not available a gas distribution grid, liquid fuels (heating oil, also known as No.2 fuel oil) or LPG are used.

Comparing the performance of oil-fired and gas-fired condensing boiler, we can find the following results [3]:

	Gas	***Oil***
Full load	98,0 %	97,9 %
Part load (30%)	108,0 %	103,9 %

Tab. 1Calculated maximum efficiency of condensing boilers
(on Lower Heating Value basis)

We can recognize that the oil-fired condensing boiler technology has lower performances than gas-fired condensing boilers, because of the chemical composition of the fuel. This topic will be discussed in the following.

The European Industry continues to invest in R&D for oil-fired boilers, with aim of improving the combustion process, reducing the pollutant emissions and increasing the combustion efficiency, and enhance the heat exchange mechanisms, trying to exploiting the heat generated by combustion in the best way.

Chapter1

Combustion

The starting point for the description of the oil-fired condensing boiler is the combustion process. At first are provided some general information about the “fuel oil”, and will be discussed the principal features of a combustion process.

1.1 LIQUID FUELS

1.1.1 REFINERY

The basic source of liquid fuels is crude oil. Gas, liquid and semi-solid materials are at the well head, but the liquid fuels which are burned in practice are first processed in refinery.

The most important process in refinery is distillation: the crude oil is flashed into a column. The main part of the column is at atmospheric pressure with a vacuum section producing the heavier products. The vaporized oil travels up to the column which has a vertical temperature gradient and the top of the column is the coolest part. The fraction of the oil which vaporizes in the column will condense out at the appropriate level in the column. Bubble caps and trays are used to facilitate this process.

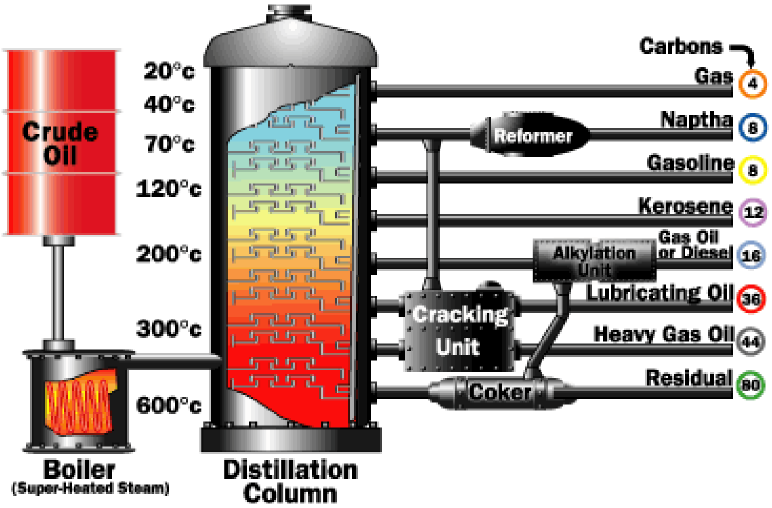
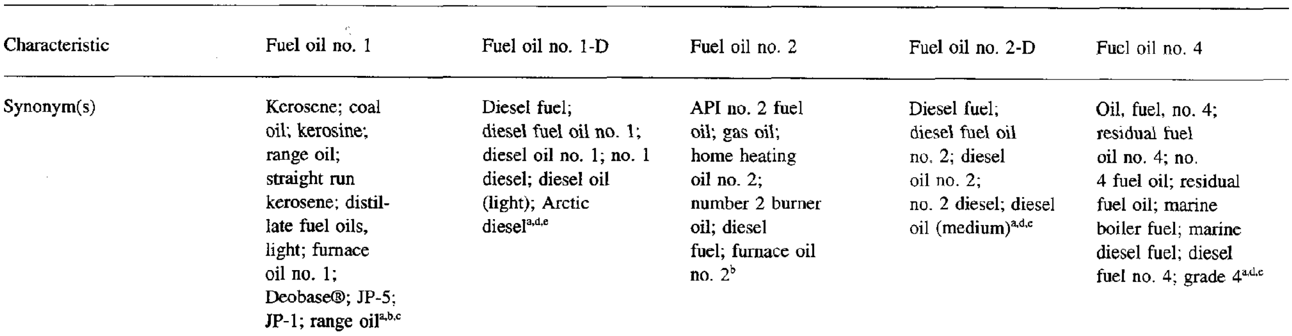


Fig.1.1 Distillation of crude oil

Whit this system the most volatile fractions are extracted on the top of the column, since they have a lower dew point, and at the top of the column there is the lower temperature.All products which have been condensate in the column are defined as “distillate oils” , whereas the feedstock’s components that did not evaporate are called “residual fuel products”.

1.1.2 PROPERTIES OF FUEL OIL

[4] provides this classification of fuel oils:



Characteristic	Fuel oil no. 1	Fuel oil no. 1-D	Fuel oil no. 2	Fuel oil no. 2-D	Fuel oil no. 4
Synonym(s)	Kerosene; coal oil; kerosine; range oil; straight run kerosene; distillate fuel oils, light; furnace oil no. 1; Deobase®; JP-5; JP-1; range oil[a,b,c]	Diesel fuel; diesel fuel oil no. 1; diesel oil no. 1; no. 1 diesel; diesel oil (light); Arctic diesel[a,d,e]	API no. 2 fuel oil; gas oil; home heating oil no. 2; number 2 burner oil; diesel fuel; furnace oil no. 2[b]	Diesel fuel; diesel fuel oil no. 2; diesel oil no. 2; no. 2 diesel; diesel oil (medium)[a,d,c]	Oil, fuel, no. 4; residual fuel oil no. 4; no. 4 fuel oil; residual fuel oil; marine boiler fuel; marine diesel fuel; diesel fuel no. 4; grade 4[a,d,c]

Tab.1.1 Fuel oil definitions

All the fuel oil classes refined from crude petroleum may be categorized as either a distillate fuel or a residual fuel depending on the method of production.

Fuel oils no. 1 and no. 2 are distillate fuels which consist of distilled process streams.

Residual fuel oils such as fuel oil no. 4 are residues remaining after distillation or cracking, or blends of such residues with distillates (IARC 1989).

Diesel fuels are approximately similar to fuel oils used for heating (fuel oils no. 1, no. 2, and no. 4).

All fuel oils consist of complex mixtures of aliphatic and aromatic hydrocarbons.

The liquid fuels used in house heating systems are:

- Fuel oil no. 1 (kerosene) is a light distillate which consists primarily of hydrocarbons in the C9 ÷ C16 range [4]. A light liquid distillate with distillation range of about 325 to 570 °F.
  Is used generally in vaporizing "pot-type" burners for space heaters, but is not recommended for wick burners. [5].
- fuel oil no. 2 is a heavier, usually blended, distillate with hydrocarbons in the C11-C20 range[4]. A slightly heavier distillate with a maximum distillation temperature of 675 °F.
  Is used for mechanically atomizing type burners and where sediment in fuel and preheating are prohibited or limited. This is the general fuel for automatic oil-fired heating equipment. It is also used by utilities for gas enrichment purposes and by industry for many purposes such as baking, evaporating, annealing, and drying, etc. [5].

Since the complexity of fuel oil composition, to determine the chemical composition and thermo-physical properties, is required a laboratory investigation for the specific fuel oil considered. Thermo-physical properties are needed in order to choose the burner's technical characteristics and are related to safety matters.

The chemical composition of a fuel oil is given in form of "ultimate analysis" , which gives the percentage by mass of each element present in the fuel. These data are fundamental in order to write combustion equations.

Focusing our attention on Domestic Fuel Oil (DFO), that belongs to category “fuel oil n. 2”, we can find in literature some typical values for the main chemical and thermo-physical properties, as shown in Tab.1.2:

Properties	***Reference [51]***	***Reference [52]***	***Reference [53]***
C (%)	*86,89*	*86,89*	*-*
H (%)	*12,98*	*12,98*	*-*
O (%)	*0,45*	*0,45*	*-*
N (%)	*<0,30*	*<0,3*	*-*
S (%)	*0,20*	*<0,2*	*0,2*
Density at 15°C [kg m^{-3}]	*844*	*854*	*815-865*
Kinematic viscosity at 37,8 °C [mm^2s^{-1}]	*3,3*	*3,3*	*-*
Pour point [°C]	*-9*	*-*	*-9*
Flash point [°C]	*63*	*-*	*>55*
Low heating value [kJ kg^{-1}]	*43700*	*43741*	*>41868*

Tab. 1.2 Domestic Fuel Oil Ultimate Analysis

Where the chemical-physical properties have the following definitions:

Density: is the ratio between the weight of the fuel and its volume, measured at a specific temperature (15°C). SI unit for this parameter is [kg/m^3]

Kinematic viscosity: is the measurement of the resistance to flow at a certain temperature, and is therefore very significant as far as boiler installation are concerned; it reflects the energy required to pump the oil through the pipelines and it has an important bearing on the atomization process in burners. The SI unit of kinematic viscosity is [m^2/s], but since this unit is so large is preferred to use [cm^2/s], called stokes [St], or [mm^2/s], called centistokes [cSt]. The equivalence between this units is the following:

$$1\ m^2/s = 10{,}000\ cm^2/s\ [stokes] = 1{,}000{,}000\ mm^2/s\ [centistokes] \qquad (1.1)$$

Pour Point: is complementary to viscosity in that it gives an indication of the temperature at which the oil will start to flow freely.It is an indication on the temperature at which oil can be pumped in pipes.

*Flash point :*is the lowest temperature at which a volatile material can vaporize to form an ignitable mixture in air. It is important to underline that this mixture ignites only with an external energy input (ignition). At the flash point, the vapor may cease to burn when the source of ignition is removed. This parameter is not to be confused with the autoignition temperature. This is an important parameter for safety aspects.

Calorific Value: in the case of liquid fuels is measured in a bomb calorimeter, which measures directly the gross calorific value at constant volume.

Sulfur content: sulfur is present in all liquid fuels, in significant quantity in residual fuel oil. When burnt, sulfur forms sulfur dioxide (SO_2) and sulfur trioxide (SO_3) which are major sources of air pollution. These products have important consequences for thermal plants installations: the flue must be designed to provide acceptable concentrations of SO_x at the ground level, and when SO_x reacts with water it gives sulfuric acid, a strong corrosive agent that could attack the plant's surfaces.

1.2 STOICHIOMETRY

Since the chemical composition of a liquid fuel is known, by the ultimate analysis, in terms of percentage by mass (w_i), all the chemical equations will be written in terms of mass.
The species that we can find in a liquid fuel are: carbon, hydrogen, nitrogen, oxygen, sulfur. Only three of them can burn, i.e. carbon, hydrogen and sulfur, according the following relations expressed by mass:

$$12kgC + 32kgO_2 \longrightarrow 44kgCO_2 \qquad 1kgC + 2{,}67kgO_2 \longrightarrow 3{,}67kgCO_2 \qquad (1.2a)$$

$$2kgH_2 + \frac{1}{2}\,32kgO_2 \longrightarrow 18kgH_2O \qquad 1kgH_2 + 8kgO_2 \longrightarrow 9kgH_2O \qquad (1.2b)$$

$$32kgS + 32kgO_2 \longrightarrow 64kgSO_2 \qquad 1kgS + 1kgO_2 \longrightarrow 2kgSO_2 \qquad (1.2c)$$

Thus, for stoichiometric combustion:

- 2,67 kg of oxygen are required to burn 1 kg of carbon, producing 3,67kg of carbon dioxide;
- 8 kg of oxygen are required to burn 1 kg of hydrogen, producing 9kg of water;
- 1 kg of oxygen is required to burn 1 kg of sulfur, producing 2kg of sulfur dioxide;

Complex hydrocarbons contain other species in their molecules, such as nitrogen and oxygen, which are not fuels. The oxygen present in the fuel is considered to be available for the combustion, then the amount of oxygen in the fuel reduces the oxygen requirement for combustion.The nitrogen in the fuel is taken to appear as gaseous nitrogen in the Combustion Products.
Referring to the composition of our fuel oil, we can easily obtain the oxygen request to burn 1 kg of this fuel and the amount of species in the products mixture:

$$Fuel + aO_2 \longrightarrow bCO_2 + cH_2O + dSO_2 + eN_2 \qquad (1.3)$$

Where:

$a = w_C \cdot 2{,}67 + w_{H_2} \cdot 8 + w_S - w_{O_2}$
$b = w_C \cdot 3{,}67$
$c = w_{H_2} \cdot 8$
$d = w_S \cdot 2$
$e = w_{N_2}$

1.2.1 COMBUSTION IN DRY AIR

Generally, combustion are in air, not in pure oxygen. Now we will consider the common air properties.

If we consider the model of dry air used in technical calculations, we will represent it as an ideal mixture of nitrogen and oxygen:

$$O_{2_{vol\%}} = 21\%$$

$$N_{2_{vol\%}} = 79\%$$

As we consider air as ideal mixture of ideal gases, the volumetric percentage of each component corresponds to its molar fraction in the mixture. Thanks to this consideration, we can convert the gas composition from molar basis to mass basis.

A given quantity of air contents a 21% of oxygen. This means that the 21% of the molecules in the mixture are oxygen. Each oxygen molecule has a molecular weight of 32 kg/kmol. The same we can say about the nitrogen: the71% of our air sample has a molecular weight of 28 kg/kmol. The molecular weight of the mixture is given from:

$$M_{air} = M_{O_2}x_{O_2} + M_{N_2}x_{N_2} = 32 \cdot 0{,}21 + 28 \cdot 0{,}79 = 28{,}84\ \frac{kg}{kmol} \quad (1.4)$$

To obtain the mass fraction of oxygen and nitrogen in the mixture:

$$w_{O_2} = \frac{M_{O_2}x_{O_2}}{M_{air}} = \frac{32 \cdot 0{,}21}{28{,}84} = 0{,}233\ \frac{kg_{O_2}}{kg_{air}} \quad (1.5)$$

$$w_{N_2} = \frac{M_{N_2}x_{N_2}}{M_{air}} = \frac{28 \cdot 0{,}79}{28{,}84} = 0{,}767\ \frac{kg_{O_2}}{kg_{air}} \quad (1.6)$$

In other words, 1 kg of dry air contains 0,233 kg of oxygen and 0,767 kg of nitrogen, orlooking this from the opposite point of view, to collect 1 kg of oxygen we need $\frac{1}{0{,}233} = 4{,}29$ kg of dry air, that will carry 3,29 kg of nitrogen as well.

With these considerations, we are now able to calculate the air request for a stoichiometric combustion of a particular fuel in Dry Air:

$$m_{dry\ air} = a \cdot 4{,}29\ kg_{dry\ air} \quad (1.7)$$

The composition of the combustion products is obtained by modifying the equation (1.3):

$$Fuel + a(O_2 + 3{,}28N_2) \longrightarrow bCO_2 + cH_2O + dSO_2 + eN_2 \quad (1.8)$$

Where:

$$a = w_C \cdot 2{,}67 + w_{H_2} \cdot 8 + w_S - w_{O_2} \quad (1.8a)$$
$$b = w_C \cdot 3{,}67 \quad (1.8b)$$

$c = w_{H_2} \cdot 8$ (1.8c)

$d = w_S \cdot 2$ (1.8d)

$e = a \cdot 3{,}29 + w_{N_2}$ (1.8e)

At this point, is useful to remember that the Products of Combustion are gaseous. Thus, for an easier treatment, we should transform their composition in volume (molar)basis.

Recalling some definitions:

$$w_i = \frac{m_i}{m_{tot}} \qquad n_i = \frac{m_i}{M_i}$$

$$x_i = \frac{n_i}{n_{tot}} = \frac{V_i}{V_{tot}} \qquad M_{tot} = \sum M_i \cdot x_i$$

1.2.2 COMBUSTION IN WET AIR

Till now we have considered air as dry air. But in real combustions we use wet air, which contains some water vapor. This water content will be present in the flue gas as well, thus the volumetric composition of the combustion gases will be modified.

Wet air is assumed to be a binary homogeneous mixture of dry air (21% O_2 and 79% N_2) and water vapor in variable percentage.

To quantify the moisture content in a sample of wet air, two parameters are used: Absolute Humidity (X) or Relative Humidity (φ)

The definition of Absolute Humidity is: the ratio between water vapor mass and dry air mass of a sample of dry air.

$$X = \frac{m_v}{m_{dryair}} \tag{1.9}$$

The reason for not using the total mass but the mass of dry air is that air and water are very dissimilar substances, and for the range of temperatures and pressures envisaged, one of the components (dry air) may be thought as permanently in the gas phase, while water vapor can be easily changed, and thus it is advantageous to refer concentrations, enthalpies, and other thermodynamic functions, to the conservative mass of dry air.

Assuming dry air and water vapor as perfect gases we can write:

$$p_i V = R_i T \tag{1.10}$$

$$X = \frac{m_v}{m_a} = \frac{\rho_v V}{\rho_a V} = \frac{p_v}{R_v T}\frac{R_a T}{p_a} = \frac{R_a p_v}{R_v p_a} = \frac{R}{M_a}\frac{M_v}{R}\frac{p_v}{p_a} = \frac{M_a}{M_v}\frac{p_v}{p_a} = 0{,}622\frac{p_v}{p - p_v} \tag{1.11}$$

Where:

v: vapor;
a: dry air;
V: volume of the sample;
T: temperature of the mixture;

p_i: partial pressure of the component "i" in the mixture;
p: total pressure of the mixture;
M_v=18,01534 kg / kmol: water vapor molar mass;
M_a=28,97 kg / kmol: dry air molar mass;

We can see that the absolute humidity of dry air depends only on partial pressure of vapor in the mixture, for a known total pressure of the mixture. Remembering the Gibbs-Dalton law of partial pressures, we know that the numerical value of the partial pressure of one component in a mixture is equal to its molar fraction in the mixture, for a total pressure equal to 1 atm.

$$\frac{p_v}{p} = \frac{n_v}{n_{tot}} = x_v \qquad (1.12)$$

Relative Humidity is defined as the ratio between the mass of water vapor content in a volume of wet air, and the maximum mass of water vapor in a saturated sample of wet air that occupies the same volume, and at the same temperature.

$$\varphi = \left.\frac{m_v}{m_s}\right]_T = \left.\frac{\frac{p_v V}{R_v T}}{\frac{p_s V}{R_v T}}\right]_T = \left.\frac{p_v}{p_s}\right]_T \qquad (1.13)$$

Where $p_s$is the saturation pressure of water vapor at the temperature T. This parameter can vary from 0 or 0% (dry air) to1 or 100% (saturated air).
Knowing the Relative Humidity, the temperature T and the total pressure p of the mixture, is easy to obtain the molar fraction of the water vapor content in the mixture:
Using tables of water properties or relations, from the temperature value of the wet air we can find the saturation pressure of water vapor, and from this parameter we apply the following equation:

$$p_v = \varphi \cdot p_s \qquad (1.14)$$

Then:

$$X = \frac{m_v}{m_a} = 0{,}622\frac{p_v}{p - p_v} \Rightarrow m_v = m_a \cdot 0{,}622\frac{p_v}{p - p_v} \qquad (1.15)$$

The last equation is written in case of wet air, but the approach to obtain it is general. In fact, if we want to calculate the water content of flue gases we just have to modify the numerical value 0,662, that represents the ratio of molecular weight of dry air and water vapor:

$$\frac{M_a}{M_v} = 0{,}662$$

Now we have a complete picture of the stoichiometric combustion in air of a liquid fuel, and we can write what happens when we burn 1 kg of fuel in Wet Air.
With these considerations, we are now able to calculate the air request for a stoichiometric combustion of a particular fuel in Wet Air:

$$m_{wet\ air} = m_{dry\ air} + m_v \qquad (1.16)$$

1.2.3 EXCESS AIR

A fundamental parameter in the combustion processes is the ratio between the mass of combustion air used to burn the mass unit of fuel. This parameter is called “Air-to-Fuel Ratio”, AFR, and is defined as:

$$AFR = \frac{m_{air}}{m_{fuel}} \qquad (1.17)$$

On dry air base.

When the combustion process takes place with theoretical amount of air, calculated with equation (1.8a), will refer to this ratio as the “stoichiometric Air-to-Fuel ratio” (ATF_{st}). This parameter is a characteristic of the fuel burned, so, know the fuel, ATF_{st} is also known.

In real combustions we need to provide more air than for a stoichiometric reaction is required, due to the imperfect mixing of fuel and oxidizer.

To characterize the amount of combustion air used for the real combustion under analysis, is used the Air-to-Fuel equivalence ratio, λ:

$$\lambda = \frac{AFR}{AFR_{st}} \qquad (1.18)$$

For stoichiometric combustion $\lambda = 1$, if less air is provided we will have $\lambda < 1$ (rich mixture), if more than the stoichiometric air is provided we will have $\lambda > 1$ (lean mixture).

Sometimes the relation between stoichiometric air needed and actual air required for the combustion is given as “excess of air” (ε). This parameter is defined as follows:

$$\varepsilon = \frac{(m_a)_{actual} - (m_a)_{st}}{(m_a)_{st}} = \frac{(m_a/m_f)_{actual} - (m_a/m_f)_{st}}{(m_a/m_f)_{st}} = \frac{(AFR)_{actual} - (AFR)_{st}}{(AFR)_{st}} = \lambda - 1 \qquad (1.19)$$

If we don’t provide a sufficient air mass in a real combustions, with a certain excess of air, we will obtain a “sub-stoichiometric” reaction localized in some parts of the combustion zone . This situation is to be avoided for the following reasons:

1. If there is not enough oxygen reacting with the fuel, a part of this will not release its energy, thus we are wasting energy;
2. If there is not enough oxygen reacting with the fuel, in the Combustion Products we will find dangerous compounds (i.e. carbon monoxide);

Combustion of a fuel with less than the stoichiometric air requirement is a complex situation which involves chemical equilibriums and reaction kinetics; however a simplified mechanism is conventionally assumed which enables an estimation of the products of combustion[6]. The mechanism is the following:

1. The available oxygen burns all the hydrogen in the fuel to water vapor;
2. All the carbon in the fuel is then burned to carbon monoxide;
3. The remaining oxygen is consumed by burning carbon monoxide to carbon dioxide;

The described model is applicable only there is enough oxygen to complete steps 1 and 2.

Considering a sub-stoichiometric combustion, we can see that the volume fraction of CO in the Combustion Products rapidly increases, for small decrements of oxygen delivered ($\lambda < 1$) as shown in Fig. 1.2.

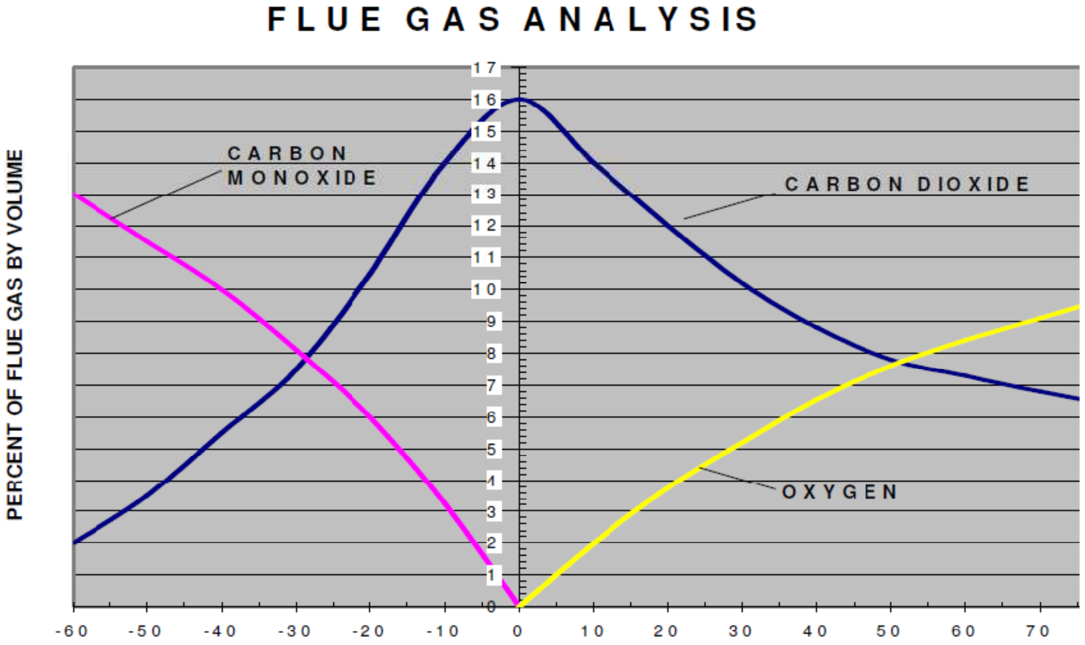


Fig.1.2 Flue gas composition vs. the Air-to-Fuel equivalent ratio

Commonly, for heating applications, is used an value of λ from 1 to 1.5, depending on the burner characteristics, we can say that a more the burner is advanced, more the λ value used will be close to 1.

Increasing λgives as a result, aside from a better chemical reaction, a heat loss: the Flue Loss.

Increasing the air supply means an increasing mass of nitrogen flowing in the combustion zone, in which it will be heated, without giving any energetic contribution to the reaction. In practice: thinking of a given heating device, that works with a certain λ. The device has a defined heat exchange surface area, which can extract an amount of heat, Q , from the combustion of 1 kg of fuel. If now we provide a grater quantity of air (higher λ) we will find that:

- The flame temperature is decreased, because the excess of air cools down the combustion process;
- The flue gas temperature is decreased;
- The heat extract from the heat exchanger, Q , is decreased, because the temperature difference between Combustion Products and carrier fluid (cold fluid in the heat exchange) is decreased;

So we reduced the thermal efficiency of our process.

1.3 THERMOCHEMISTRY

1.3.1 FIRST LAW OF THERMODYNAMICS

Considering a control volume, in which the fluid may flow across the boundary, assuming steady-state and steady-flow, we can write the energy balance as:

$$\dot{Q}_{cv} - \dot{W}_{cv} = \dot{m}e_o - \dot{m}e_i + \dot{m}(p_o v_o - p_i v_i) \tag{1.20}$$

Where:

$\dot{Q}_{cv}$: *is the rate of heat transferred across the boundary of the system;*
$\dot{W}_{cv}$: *is the rate of all work done by the control volume;*
$\dot{m}$: *is the mass flow rate of the fluid that across the control volume;*
e: *is the specific energy of the stream in the considered point;*
$\dot{m}e$: *is the rate of energy going out of the system (subscript "o") or entering in the system (subscript "i");*
$\dot{W}_{cv}$: *is the rate of all work done by the control volume;*
$\dot{m}$: *is the mass flow rate of the fluid that across the control volume;*
e: *is the specific energy of the stream in the considered point;*
$\dot{m}e$: *is the rate of energy going out of the system (subscript "o") or entering in the system (subscript "i");*
$\dot{m}(p_o v_o - p_i v_i)$:*is the rate of work associated with the pressure forces where the fluid crosses the control surface, p_x and v_x are, respectively, the pressure and the specific volumeof the stream in the point "x".*

The specific energy of the stream (e) is the sum of internal energy, kinetic energy and potential energy:

$$e = u + \frac{1}{2}v^2 + gz \tag{1.21}$$

Where:

u: *is the internal energy;*
v: *is the velocity of the fluid;*
z: *is the elevation.*

We can write the sum of internal energy and the pressure-specific volume associated to the flow work, that represents the Enthalpy, h, of the fluid in the specific condition:

$$h = u + pv \tag{1.22}$$

Finally we obtain the expression of energy conservation in a control volume:

$$\dot{Q}_{cv} - \dot{W}_{cv} = \dot{m}\left[(h_o - h_i) + \frac{1}{2}(v_o^2 - v_i^2) + g(z_o - z_i)\right] \tag{1.23}$$

That on a mass-specific basis becomes:

$$\dot{q}_{cv} - \dot{w}_{cv} = \left[(h_o - h_i) + \frac{1}{2}(v_o^2 - v_i^2) + g(z_o - z_i)\right] \tag{1.24}$$

Considering the case of combustion we can make some simplifications:
No work is exchanged across the system's boundary with the outside and the variation of kinetic and potential energy can be neglected. So, we obtain :

$$\dot{Q}_{cv} = \dot{m}(h_o - h_i) \tag{1.25}$$

or in mass-specific basis:

$$\dot{q}_{cv} = (h_o - h_i) \tag{1.26}$$

1.3.2 ABSOLUTE ENTHALPY AND ENTHALPY OF FORMATION

For any species we can define an Absolute Enthalpy that is the sum of an enthalpy that takes into account the energy associated with the energy of the chemical bonds, the Enthalpy of Formation h_f, and an enthalpy that is associated only with the temperature, the Sensible Enthalpy change, Δh_s.

The Molar Absolute Enthalpy for the specie "i" is defined as:

$$h_i(T) = h_{f,i}^0(T_{ref}) + \Delta h_{s,i}(T_{ref} \to T) \tag{1.27}$$

To use this equation we need to define a Standard Reference state. Commonly this state is represented by:

$$T_{ref} = 25°C$$
$$p_{ref} = 1\ atm$$

The convention is that Enthalpy of Formation is zero for the elements in their naturally occurring state at the Reference State temperature and pressure.
If we assume that the species that we are considering have an ideal gas behavior, the following equations are consistent:

$$h_i(T) = h_{f,i}^0(T_{ref}) + \int_{T_{Ref}}^{T} c_{p,i}(T)dT = h_{f,i}^0(T_{ref}) + \left[c_{p,i}\right]_{T_{Ref}}^{T}(T - T_{ref}) = h_{f,i}^0(T_{ref}) + \overline{c_{p,i}}(T - T_{ref}) \tag{1.28}$$

Where:

$c_p(T)$ is the constant-pressure specific heat of the specie "i", that is a function of the temperature;

$\left[c_{p,i}\right]_{T_{Ref}}^{T} = \overline{c_{p,\iota}}$ *is the mean value of the specific heat of the specie "i", in the temperature interval considered;*
$h_{f,i}^{0}(T_{ref})$ *is the standard enthalpy of formation of the specie "i".*

1.3.3 ENTHALPY OF COMBUSTION AND HEATING VALUES

Writing the energy equation for a combustion process, assuming a simple steady-flow reactor in which a stoichiometric mixture of reactants enters and products exits, both at the Standard Reference state ($T_{ref} = 25°C, p_{ref} = 1\ atm$). The combustion is assumed to be complete, we obtain:

$$q_{cv} = \left[H_o(T_{ref}) - H_i(T_{ref})\right] = H_{prod}(T_{ref}) - H_{reac}(T_{ref}) \tag{1.29}$$

Where:

$H_o(T_{ref}) = H_{prod}(T_{ref})$ *is the enthalpy of the products mixture at the reference temperature;*
$H_i(T_{ref}) = H_{reac}(T_{ref})$ *is the enthalpy of the reactant mixture at the reference temperature;*

Since we imposed that the products of combustion have to be at reference state (T_{ref}), we need to cool them. The heat that we extract cooling the combustion products to the reference temperature is defined Enthalpy of Reaction or Enthalpy of Combustion (per mass of mixture),Δh_R.

$$\Delta h_R = q_{cv} = H_{prod}(T_{ref}) - H_{reac}(T_{ref}) \tag{1.30}$$

$$\Delta h_R = \left(H_f^0(T_{ref}) + \Delta H_s(T_{ref})\right)_{Prod} - \left(H_f^0(T_{ref}) + \Delta H_s(T_{ref})\right)_{Reac} \tag{1.31}$$

$$\Delta h_R = \sum\nolimits_{Prod} w_i \left(h_{f,i}^0(T_{ref}) + \overline{c_{p,\iota}}(T_{ref} - T_{ref})\right) - \sum\nolimits_{Reac} w_i \left(h_{f,i}^0(T_{ref}) + \overline{c_{p,\iota}}(T_{ref} - T_{ref})\right) \tag{1.32}$$

$$\Delta h_R = \sum\nolimits_{Prod} w_i \cdot h_{f,i}^0(T_{ref}) - \sum\nolimits_{Reac} w_i \cdot h_{f,i}^0(T_{ref}) \tag{1.33}$$

where $w_i$is the mass fraction of the element "i" in the mixture;

If we write the previous equation considering to burn the mass-unit of fuel we obtain the standard higher heating value (HHV) , that is defined as the heat release to the ambient when fuel, oxidizer and products flow at 25°C, 1 atm and species-separated, and computed by adding the latent heat of the water-vapor.
From this definition we notice that the HHV is a positive value, for the usual conventions used for heat transfer direction, and we see also that the enthalpy of combustion is a positive value. So the heating value has the same numerical value of the enthalpy of combustion with the signchanged, [7].

$$HHV = -\Delta h_R \tag{1.34}$$

We can estimate the Higher Heating Value of a fuel knowing its compositions by mass, writing the following equation, derived from the basic combustion reactions of Carbon, Hydrogen and Sulfur[8]:

$$HHV \approx 34{,}03 w_C + 244{,}42 w_H + 10{,}88 w_S \left[\frac{MJ}{kg_{fuel}}\right] \tag{1.35}$$

When the higher heating value of a fuel is determined experimentally, we are referring to the standard practical heating value PHV, that is defined as: the heat transfer to cooling water in a Junkers-type calorimeter, i.e. a flow calorimeter with inputs (fuel and air mixture, and cooling water) and output approaching standard conditions (25°C and 1 atm). The fuel is burnt with excess air for complete combustion, since the heating value is independent of excess air, while the cooling flow-rate is traded off for best resolution.

When we are talking about heating values of fuels, is important to define the physical state of all reactants and products, especially:

- physical state of water in the products (liquid or vapor);
- state of aggregation of the fuel (gas, liquid or solid);

In the definition of High Heating Value we assume that the water content in the products is in condensate state. If we refer to a combustion with products at standard conditions, but with water in vapor phase, we have the Lower Heating Value LHV.
Obviously LHV is lower than the HHV, since is not taken in account the latent heat of vaporization (h_{fg,H_2O}) of the water vapor as an available heat source.

$$LHV = HHV - h_{fg,H_2O} \tag{1.36}$$

Formally, the Latent Heat of Vaporization or Enthalpy of Vaporization, h_{fg}, is defined as the heat required in a constant pressure process to completely vaporize the unit mass of liquid at a given temperature, i.e. :

$$h_{fg}(T,p) = h_{vapor}(T,p) - h_{liquid}(T,p) \tag{1.37}$$

Where T and p are the saturation temperature and pressure respectively.
Values of h_{fg} are tabulated.

If the fuel is in gas phase, its HHV will be higher than the fuel was in liquid phase, due to the Latent Heat of Vaporization required by the liquid fuel to change phase. In Fig.1.2 all the above consideration are graphically shown.

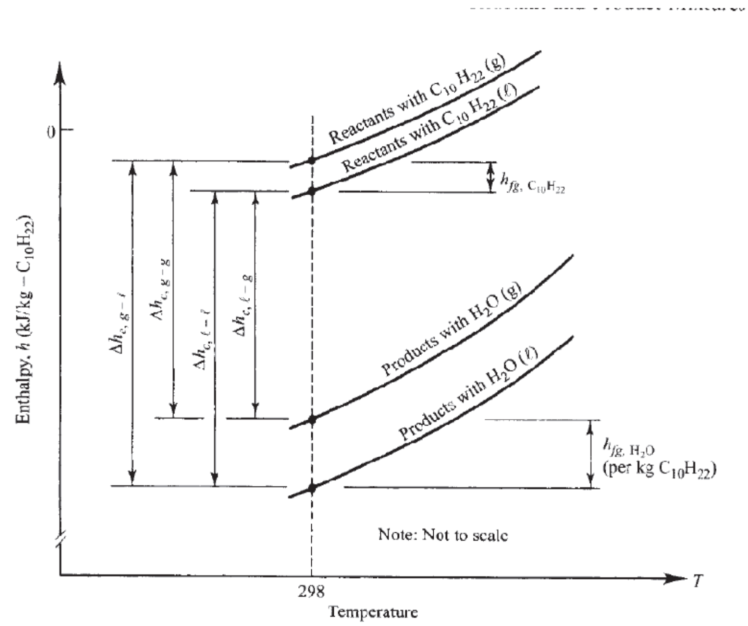


Fig. 1.2 Representation of the reactant's and product's enthalpies

1.3.4 ADIABATIC COMBUSTION TEMPERATURE

Till now we considered combustion were the reactants temperature at the beginning of the process was equal to the products temperature after the thermo-chemical reaction. This means that we had to export heat from the system, who allowed the heat transfer across its boundaries. Now we want to analyze an adiabatic system, in which the energy released by the chemical transformation of reactants into products is used only to rise up the products temperature, and no heat transfer is admitted across the reactor boundaries. In this case we will obtain another important parameter for combustion processes: the Adiabatic Combustion Temperature.

The adiabatic temperature is a thermodynamic state variable that results from the conversion of internal chemical energy to internal thermal energy after the combustion process takes place.

It can be measured at the exhaust of an adiabatic combustor, or just after a flame, but the practical difficulties in ensuring minimal heat losses, particularly from the thermometer probe, render a theoretical computation more precise that the actual measurement [9]

Writing the equation (1.8) under this condition we obtain:

$$1kgFUEL + AFRkgAIR = \sum_{Prod} w_i kg \cdot PROD_i \tag{1.38}$$

$$0 = \left[h^0_{f,Fuel} + \bar{c}_{p_{Fuel}}(T_{in} - T_{Ref})\right] + AFR \cdot \left[h^0_{f,Air} + \bar{c}_{p_{Air}}(T_{in} - T_{Ref})\right] - \sum_{Product} w_i \cdot \left[h^0_{f,i} + \bar{c}_{p_i}(T_{ad} - T_{Ref})\right] \tag{1.39}$$

$$h^0_{f,Fuel} + AFR \cdot h^0_{f,Air} - \sum_{Prod} w_i \cdot h^0_{f,i} = LHV \tag{1.40}$$

We use the LHV because in this situation the water in the products is in vapor phase.

$$0 = LHV + \left(\bar{c}_{p_{Fuel}} + AFR \cdot \bar{c}_{p_{Air}}\right) \cdot \left(T_{in} - T_{Ref}\right) - \sum_{Prod} w_i \cdot \bar{c}_{p_i}\left(T_{ad} - T_{Ref}\right) \tag{1.41}$$

Thus, the Adiabatic temperature is:

$$T_{ad} = T_{Ref} + \frac{LHV + \left(\overline{c_p}_{Fuel} + AFR \cdot \overline{c_p}_{Air}\right) \cdot (T_{in} - T_{Ref})}{\sum_{Prod} w_i \cdot \overline{c_p}_i} \qquad (1.42)$$

Where:

T_{ad}: *is the temperature of product mixture exiting the system, i.e. the Adiabatic Combustion Temperature;*
T_{in}: *is the temperature of the reactants mixture entering the system;*
AFR is the air-to-fuel ratio
$\overline{c_p}_i$: *is the mean constant-pressure specific heat for the specie "i"*

We can understand see that factors that influence the Adiabatic Temperature of Combustion are:the lower heating value (LHV), the air-to-fuel ratio (AFR), the inlet temperature of reactants (T_{in}) and the products composition (w_i).

We can think that higher is the heating value of the fuel, higher will be the Adiabatic Temperature of Combustion. But in reality things are not going this way. In fact the fuels with the highest heating values are those rich in hydrogen, or with a high carbon-to-hydrogen ratio (methane C:H=1.3). The combustion of a hydrogen-rich fuel will generate more water vapor in its products. Water vapor has the higher specific heat than the other combustion product, thus, looking at the equation (1.42), we can understand that the Adiabatic Temperature of Combustion will be lower.

The highest value of Adiabatic Temperature of Combustion is obtained with stoichiometric air. Any increasing of air delivered to the combustion will lower the Adiabatic Temperature of Combustion, since it will be increased the amount of gases to be heated, i.e. it will be higher the denominator in equation (1.42). In Fig.1.3 is shown what just said:

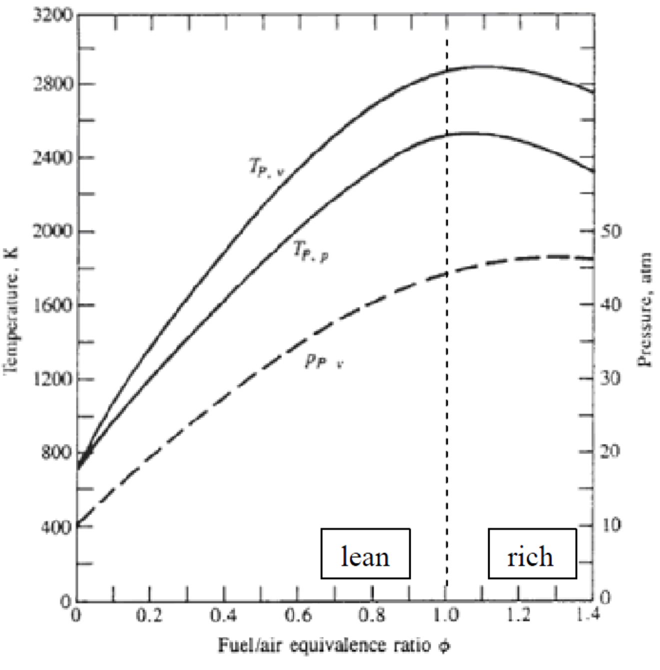


Fig 1.3 Adiabatic flame temperature and Fuel-to-Air equivalence ratio

In many burners the incoming air is preheated, usually by a heat exchanger which uses the heat content of flue gases when they leave the last section of the heating device (as in our case). Increasing the reactants' temperature corresponds to increase the numerator in equation (1.42), so the Adiabatic Temperature of Combustion will increase.

All that we said represent a simplification of what actually happens, but it can be used as an upper bound to the Adiabatic Temperature of Combustion. In practice there are a number of reasons why the Adiabatic Temperature will be lower than the value predicted by our approach, and the most significant is the dissociation of products in reactants or other highly reactive species due to the temperature achieved in the flame. This kind of transformation is endothermic, so the dissociation process will be accompanied by an absorption of energy, hence reducing the flame temperature. Admitting the products dissociation in our model, corresponds to remove the hypothesis of mono-directionality for the chemical reactions.

1.3.5 DROPLET COMBUSTION

When in a combustion process the fuel and the air are initially separated before they reach the flame zone, we are referring to "non premixed flames" or "diffusive flames". Focusing on our case, an oil burner, the fuel is a liquid. It enters the combustion chamber as a spray. Since combustion occurs in the gas phase, is important to understand the vaporization mechanism of the spray droplets and the mixing process of the fuel vapor with the oxidizer, to avoid low combustion efficiency and pollutant emissions in the atmosphere. In fact in a non premixed combustion, overall mixing can be stoichiometric or with a certain excess of air, but is possible that in some areas of the combustion chamber the combustion occurs in sub-stoichiometric conditions, this phenomenon is the cause of combustion ineffectiveness (presence of unburned hydrocarbons) and pollutants formation.

Liquid fuel combustion is usually done by injecting liquid fuel into the gas phase combustion environment. Turbulence within the liquid flow is created inside the injector. This turbulent fluid flow gets out of the nozzle in the form of complex mix of strings and then crushes into small drops and appears as dense clouds of droplets that launches through the gas into the flame zone. Heat transfer to drop, increases the vapor pressure and therefore the fuel evaporates into the gas, so burning of gas phase is began. Non-premixed flame surrounds the group drops or droplets and eventually fuel vapor burns. This set of processes is called combustion spray[10].

If we consider that each droplet of the spray is surrounded by a spherical flame that burns the fuel vapor released by the evaporating droplet, we talk about "single droplet combustion". This situation is pretty rare to find in usual combustion processes, but it is easy to study (especially if we assume that the droplet exists in a quiescent medium).

1.3.5.1 SINGLE DROPLET COMBUSTION MODEL

Analytical and simplified models to describe this process were developed by several authors.
Fuel droplets models are used to describe the influence of droplet size and ambient conditions on fuel devices such as oil-fired boilers. The simple quasi-steady model by [9 and 11] is widely accepted as a theoretical model of fuel droplet combustion.

The assumption of this model are:

1. The burning droplet is surrounded by a spherically symmetric flame, and exists in a quiescent, infinite medium. There are no interactions with any other droplets, and the effects of convection are ignored;
2. The burning process is quasi-steady;
3. The fuel is a single component liquid with zero solubility for gases. Phase equilibrium prevails at the liquid-vapor interface;
4. Pressure is uniform and constant;
5. The gas phase consists in only three "species": fuel vapor, oxidizer and combustion products.
6. The gas phase is divided into two zones. The "inner zone" between the droplet surface and the flame, which contains only fuel vapor and combustion products, and the "outer zone" starting from the flame to the infinite, which contains only oxidizer and combustion products;
7. Fuel and oxidizer react in stoichiometric proportions at the flame. Chemical kinetics is assumed to be infinitely fast, so that the flame is represented as an infinitesimally sheet;
8. The Lewis number is unity ($Le = \frac{\alpha}{\mathcal{D}} = \frac{k_g}{\rho c_{pg} \mathcal{D}} = 1$, that means that the thermal and mass diffusivity have the same magnitude)
9. Radiation heat transfer is neglected;
10. The values of $k_g, c_{pg}, \rho\mathcal{D}$ are constant;
11. The liquid fuel droplet is the only condensed phase; soot or liquid water are not present.

With these assumptions, a graphical sketch valid for the model is the following:

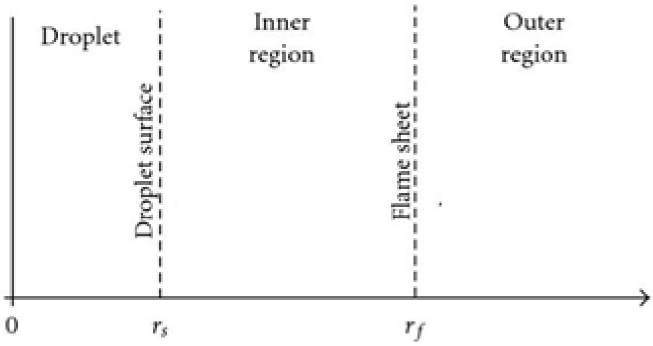


Fig. 1.4 Representation of the model

Where *r* is the radial distance from the droplet center.

Implementing this model we will be able to find as main result the droplet burning rate, $\dot{m}_F$, for a given initial droplet diameter, D_0, and for given environmental conditions, far from the droplet, i.e. the temperature, T_∞, and the oxidizer mass fraction $Y_{ox,\infty} = 1$.

To achieve this main result, we will obtain other interesting information such as:

1. temperature and species profiles in each region;
2. the flame radius,r_f;
3. the flame temperature, T_f;
4. the droplet surface temperature,T_s;
5. the fuel vapor mass fraction at the droplet surface, $Y_{F,s}$.

Summarizing we seek expressions to evaluate five parameters: $\dot{m}_F$,$Y_{F,s}$,T_s,$T_f$and r_f. Thus we will write five equations:

1. the fuel vapor distribution in the inner region;
2. the oxidizer distribution in the outer region;
3. energy balance at the droplet surface;
4. energy balance at the flame sheet;
5. the phase-equilibrium at the liquid-vapor interface using the Clausius-Clapeyron relationship.

Mass conservation

Since is assumed a quasi-steady process, the mass flow rate is a constant, independent by the radius:

$$\dot{m}(r) = \dot{m}_F = const. \tag{1.43}$$

And we can say that the mass flow rate is everywhere identical to the fuel flow rate, that is the Burning Rate.

1. *Species conservation in the inner region*

We assumed that in the inner region there are only fuel vapor and combustion products that are diffusing in the gas phase. Using the Fick's law, under the following boundary condition:

$$Y_F(r_s) = Y_{F,s}(T_s) \tag{1.44}$$

we obtain species distributions in the inner region:

Fuel:

$$Y_F(r) = 1 - \frac{(1 - Y_{F,s})e^{-\frac{Z_F \dot{m}_F}{r}}}{e^{-\frac{Z_F \dot{m}_F}{r_s}}} \tag{1.45}$$

Combustion Products:

$$Y_{Pr}(r) = 1 - Y_F(r) \tag{1.46}$$

We can find a relation between $Y_{F,s}$,$\dot{m}_F$ and r_f if we consider the following boundary condition:

$$Y_F(r_f) = 0 \tag{1.47}$$

And it results:

$$Y_{F,s} = 1 - \frac{e^{-\frac{Z_F \dot{m}_F}{r_s}}}{e^{-\frac{Z_F \dot{m}_F}{r_f}}} \tag{1.48}$$

Where:

$$Z_F = \frac{1}{4\pi\rho\mathcal{D}} \tag{1.49}$$

2. *Species conservation in the outer region*

We assumed that in the outer region there are only oxidizer and combustion products that are diffusing in the gas phase.

At the flame the oxidizer and the fuel vapor combine in stoichimetric proportions:

$$1kgFuel + \upsilon kgOxidizer = (1 + \upsilon)\ kgProducts \tag{1.49}$$

Using the Fick's law, under the following boundary condition:

$$Y_{Ox}(r_f) = 0 \tag{1.50}$$

we obtain species distributions in the outer region

$$Y_{Ox}(r) = \upsilon \left[\frac{e^{-\frac{Z_F \dot{m}_F}{r}}}{e^{-\frac{Z_F \dot{m}_F}{r_f}}} - 1\right] \tag{1.51}$$

The combustion products mass fraction distribution is:

$$Y_{Pr}(r) = 1 - Y_{Ox}(r) \tag{1.52}$$

We can find a relation between $\dot{m}_F$ and r_f if we use the following boundary condition:

$$Y_{Ox}(r \to \infty) = Y_{Ox,\infty} = 1 \tag{1.53}$$

And it results:

$$e^{+\frac{Z_F \dot{m}_F}{r_f}} = \frac{\upsilon + 1}{\upsilon} \tag{1.54}$$

Energy conservation balances

With the hypothesis that we stated for this model, we can adopt the Shvab-Zeldovich energy equation to write the following energy balances. The differential expression of the energy conservation for our case is:

$$\frac{d\left(r^2\frac{dT}{dr}\right)}{dr} = Z_T\dot{m}_F\frac{dT}{dr} \tag{1.55}$$

Where:

$$Z_T = \frac{c_{pg}}{4\pi k_g} = \frac{1}{4\pi\rho\mathcal{D}} = Z_F \tag{1.56}$$

Because of the assumption of unity of Lewis number.

To obtain the *temperature profile in the inner region*, we need to integrate the equation (1.55) with the following boundary conditions:

$$T(r_s) = T_s \tag{1.57a}$$
$$T(r_f) = T_f \tag{1.57b}$$

Obtaining the expression:

$$T(r) = \frac{(T_s - T_f)e^{-\frac{Z_T\dot{m}_F}{r}} + T_f e^{-\frac{Z_T\dot{m}_F}{r_s}} - T_s e^{-\frac{Z_T\dot{m}_F}{r_f}}}{e^{-\frac{Z_T\dot{m}_F}{r_s}} - e^{-\frac{Z_T\dot{m}_F}{r_f}}} \tag{1.58}$$

Considering the outer zone, the boundary conditions are:

$$T(r_f) = T_f \tag{1.59a}$$
$$T(r \to \infty) = T_\infty \tag{1.59b}$$

And the *temperature profile in the outer region* is:

$$T(r) = \frac{(T_f - T_\infty)e^{-\frac{Z_T\dot{m}_F}{r}} + T_\infty e^{-\frac{Z_T\dot{m}_F}{r_f}} - T_f}{e^{-\frac{Z_T\dot{m}_F}{r_f}} - 1} \tag{1.60}$$

3. *Energy balance at the droplet surface*

Heat generated at the flame is conducted through the gas phase to the droplet surface. Some of this heat is used to vaporize the fuel and the remaining is conducted into the droplet interior:

$$\dot{Q}_{g-i} = \dot{m}_F h_{fg} + \dot{Q}_{i-l} \tag{1.61}$$

$\dot{Q}_{i-l}$ is the heat conducted into the droplet interior, and one common approach to handle this quantity is adopting the “onion skin” model for the droplet. It consist in thinking that the droplet is composed of two zones: the interior region existing uniformly at its initial temperature, T_0; and a thin surface layer at the surface temperature, T_s. Thus we can write:

$$\dot{Q}_{i-l} = \dot{m}_F c_{pl}(T_s - T_0) \tag{1.62}$$

$$q_{i-l} = \frac{\dot{Q}_{i-l}}{\dot{m}_F} = c_{pl}(T_s - T_0) \tag{1.63}$$

This heat is the energy required to increase the liquid temperature from the initial value to the temperature at which the liquid starts to vaporize.

The heat conducted to the droplet surface from the flame can be evaluated using the Fourier's law, and using the temperature distribution of the inner region:

$$-\left[-k_g 4\pi r^2 \frac{dT}{dr}\right]_{r_s} = \dot{m}_F\left[h_{fg} + q_{i-l}\right] \tag{1.64}$$

The temperature gradient is obtained from equation (1.58) and, calculating its expression at the droplet surface, i.e. when $r = r_s$, we can write the following correlation between the four unknown $\dot{m}_F$,T_f,T_s and r_f:

$$\frac{c_{pg}(T_f - T_s)}{h_{fg} + q_{i-l}} \cdot \frac{e^{-\frac{Z_T \dot{m}_F}{r_s}}}{e^{-\frac{Z_T \dot{m}_F}{r_s}} - e^{-\frac{Z_T \dot{m}_F}{r_f}}} + 1 = 0 \tag{1.65}$$

4. *Energy balance at the flame sheet*

The flame temperature is the highest in the considered system, thus, heat is conducted both toward the droplet,$\dot{Q}_{g-i}$, both away to infinity $\dot{Q}_{f-\infty}$.
The energy released by the combustion process is evaluated by using absolute enthalpy fluxes for the fuel, oxidizer and products. The surface energy balance at the flame sheet is:

$$\dot{m}_F h_F + \dot{m}_{Ox} h_{Ox} - \dot{m}_{Pr} h_{Pr} = \dot{Q}_{g-i} + \dot{Q}_{f-\infty} \tag{1.66}$$

Defining the heat of combustion as:

$$\Delta h_c(T_{ref}) = h^0_{f,F} + \upsilon h^0_{f,Ox} - (\upsilon + 1) h^0_{f,Pr} \tag{1.67}$$

And substituting it in equation (1.66), we obtain:

$$\dot{m}_F \Delta h_c + \dot{m}_F c_{pg}\left[(T_f - T_{ref}) + \upsilon(T_f - T_{ref}) - (\upsilon + 1)(T_f - T_{ref})\right] = \dot{Q}_{g-i} + \dot{Q}_{f-\infty} \tag{1.68}$$

Assuming that c_{pg} is constant mean that Δh_c is independent from temperature, so, if we think that $T_f = T_{ref}$, we obtain:

$$\dot{m}_F \Delta h_c = \dot{Q}_{g-i} + \dot{Q}_{f-\infty} \tag{1.69}$$

That states that "the rate at which chemical energy is converted to thermal energy at the flame is equal to the rate at which the heat is conducted away from the flame.
Relying on Fuorier's law, as above, we can estimate $\dot{Q}_{f-\infty}$:

$$\dot{m}_F \Delta h_c = k_g 4\pi r_f^2 \left[\frac{dT}{dr}\right]_{r_f^-} - k_g 4\pi r_f^2 \left[\frac{dT}{dr}\right]_{r_f^+} \tag{1.70}$$

Where the temperature gradients are given differentiating the temperature profile of the inner region (-) and for the outer region (+), and valuating them for $r = r_f^-$ and $r = r_f^+$.

The flame sheet energy balance is finally expressed by:

$$\frac{c_{pg}}{\Delta h_c}\left[\frac{(T_f-T_s)e^{-\frac{Z_T\dot{m}_F}{r_f}}}{e^{-\frac{Z_T\dot{m}_F}{r_s}}-e^{-\frac{Z_T\dot{m}_F}{r_f}}}-\frac{(T_f-T_\infty)e^{-\frac{Z_T\dot{m}_F}{r_f}}}{1-e^{-\frac{Z_T\dot{m}_F}{r_f}}}\right]-1=0 \qquad (1.71)$$

5. Liquid vapor equilibrium

The model assumes equilibrium between the liquid and the vapor phase of fuel at the droplet surface. we can write the Clausius-Clapeyron equation to obtain the last equation we need to provide the closure to the problem.

The partial pressure of the fuel at the liquid-vapor interface is given by:

$$p_{F,s}=Ae^{-\frac{B}{T_s}} \qquad (1.72)$$

Where A and B are constants obtained from the Clausius-Clapeyron equation, and depends on the particular fuel considered.

The partial pressure of the fuel is related to the mole fraction of the fuel vapor and mass fraction:

$$x_{F,s}=\frac{p_{F,s}}{p} \qquad (1.73)$$

And

$$Y_{F,s}=x_{F,s}\frac{MW_F}{x_{F,s}MW_F+(1-x_{F,s})MW_{Pr}} \qquad (1.74)$$

Than we obtain from (1.68) and (1.71):

$$Y_{F,s}=\frac{A\cdot MW_F\cdot e^{-\frac{B}{T_s}}}{A\cdot MW_F\cdot e^{-\frac{B}{T_s}}+\left[p-A\cdot e^{-\frac{B}{T_s}}\right]\cdot MW_{Pr}} \qquad (1.75)$$

Now we have five equation with five unknown, and we are able to solve the problem.

	Equation	***unknown***
Fuel species conservation in the inner zone	$Y_{F,s}=1-\frac{e^{-\frac{Z_F\dot{m}_F}{r_s}}}{e^{-\frac{Z_F\dot{m}_F}{r_f}}}$	$\dot{m}_F$, r_f,$Y_{F,s}$
Oxidizer species conservation in the outer zone	$e^{+\frac{Z_F\dot{m}_F}{r_f}}=\frac{\upsilon+1}{\upsilon}$	$\dot{m}_F$, r_f
Energy balance at the droplet liquid-vapor interface	$\frac{c_{pg}(T_f-T_s)}{h_{fg}+q_{i-l}}\cdot\frac{e^{-\frac{Z_T\dot{m}_F}{r_s}}}{e^{-\frac{Z_T\dot{m}_F}{r_s}}-e^{-\frac{Z_T\dot{m}_F}{r_f}}}+1=0$	$\dot{m}_F$, r_f,T_f,T_s

Energy balance at the flame sheet	$\frac{c_{pg}}{\Delta h_c}\left[\frac{\left(T_f - T_s\right)e^{-\frac{Z_T\dot{m}_F}{r_f}}}{e^{-\frac{Z_T\dot{m}_F}{r_s}} - e^{-\frac{Z_T\dot{m}_F}{r_f}}} - \frac{\left(T_f - T_\infty\right)e^{-\frac{Z_T\dot{m}_F}{r_f}}}{1 - e^{-\frac{Z_T\dot{m}_F}{r_f}}}\right] - 1 = 0$	$\dot{m}_F$, r_f,T_f,T_s
Liquid vapor phase equilibrium at interface (Clausius-Clapeyron)	$Y_{F,s} = \frac{A \cdot MW_F \cdot e^{-\frac{B}{T_s}}}{A \cdot MW_F \cdot e^{-\frac{B}{T_s}} + \left[p - A \cdot e^{-\frac{B}{T_s}}\right] \cdot MW_{Pr}}$	T_s,$Y_{F,s}$

The species properties for inner and outer region are estimated as follows:

$$\bar{T} = 0{,}5\left(T_s + T_f\right) \tag{1.76a}$$
$$c_{pg} = c_{pF}(\bar{T}) \tag{1.76b}$$
$$k_g = 0{,}4k_F(\bar{T}) + 0{,}6k_{Ox}(\bar{T}) \tag{1.76c}$$
$$\rho_l = \rho_l(\bar{T}) \tag{1.76d}$$

as suggested by Williams and Law [12].

The main results of this model are the following:

- Fuel mass flow rate [kg/s]:

$$\dot{m}_F = \frac{4\pi k_g r_s}{c_{pg}} ln(1 + B) \tag{1.77}$$

Where B is the Transfer number, defined as:

$$B = \frac{\frac{\Delta h_c}{\upsilon} + c_{pg}(T_\infty - T_s)}{q_{i-l} + h_{fg}} \tag{1.78}$$

- Flame temperature [K]:

$$T_f = \frac{\left(q_{i-l} + h_{fg}\right)}{c_{pg}(1+\upsilon)}[\upsilon B - 1] + T_s \tag{1.79}$$

- Flame to droplet radius ratio:

$$\frac{r_f}{r_s} = \frac{ln(1+B)}{ln\left(1+\frac{1}{\upsilon}\right)} \tag{1.80}$$

- The $D^2\ law$ and the Droplet lifetime are:

$$D^2(t) = D_0^2 - K \cdot t \tag{1.81}$$

and

$$t_d = \frac{D_0^2}{K} \tag{1.82}$$

Where:

r_s: *is the droplet radius* $[m]$;

k_g: *is the thermal conductivity of the gas phase* $\left[\frac{W}{mK}\right]$;

c_{pg}: *is the specific heat of the gas phase* $\left[\frac{J}{kgK}\right]$;

B: *is the Transfer Number, or Spalding Number;*

Δh_c: *is the enthalpy of combustion* $\left[\frac{J}{kg}\right]$;

υ: *is the oxidizer-to fuel stoichiometric ratio* $\left[\frac{kg}{kg}\right]$;

T_∞: *is the free-stream temperature* $[K]$;

T_s: *is the droplet surface temperature* $[K]$;

q_{i-l}: *is the heat per unit mass conducted into the droplet interior* $\left[\frac{J}{kg}\right]$;

h_{fg}: *is the latent heat of vaporization* $\left[\frac{J}{kg}\right]$;

t: *is time* $[s]$;

D_0: *is the initial droplet diameter* $[m]$;

K: *is the evaporation rate constant* $\left[\frac{m^2}{s}\right]$;

$$K = \frac{8k_g}{\rho_l c_{pg}} ln(1+B) = \frac{2\dot{m}_F}{\pi \rho_l r_s} \tag{1.83}$$

ρ_l: *is the liquid density* $\left[\frac{kg}{m^3}\right]$;

In the following figure are shown temperature and species profiles in the inner and outer regions.

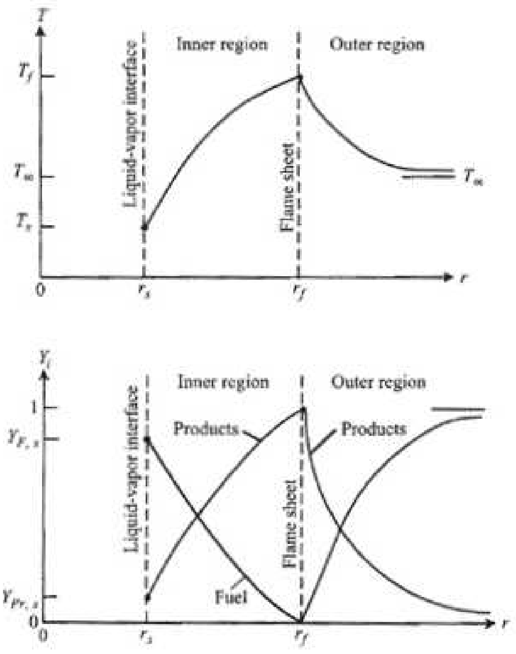


Fig.1.5 Temperature and species profiles

1.3.5.2 FILM THEORY-CONVECTIVE ENVIRONMENT

This simple quasi-steady model assumes that there is no relative velocity between the droplet and the free stream, no buoyancy. This situation occurs if the droplet is immersed in a stagnant medium and in micro gravity conditions.

In order to make more close to real cases of interest the droplet burning model, we should think now that the droplet is in a convective environment. There are several approaches to include convection in the droplet burning model. The simplest one is the so-called “film theory”. The essence of this new model is to move the heat- and mass- transfer boundary conditions from the infinity, inward, to the so-called film radius: δ_M for species and δ_T for energy.

The film radii are defined in terms of the Nusselt number, Nu, for heat transfer and the Sherwood number, Sh, for mass transfer.

$$\frac{\delta_T}{r_s} = \frac{Nu}{Nu-2} \tag{1.84}$$

$$\frac{\delta_M}{r_s} = \frac{Sh}{Sh-2} \tag{1.85}$$

The Nusselt number, Nu, is the ratio of convective to conductive heat transfer across the boundary.

The Sherwood number, Sh, represents the ratio of convective to diffusive mass transport.

With the hypothesis of unity of Lewis number, we can understand that $Sh = Nu$, thus $\delta_T = \delta_M$.

Feath [7] recommends to use the following correlation in case of droplet burning with forced convection:

$$Nu = 2 + \frac{0{,}555\, Re^{1/2} Pr^{1/3}}{\left[1+1{,}232/\left(Re\, Pr^{4/3}\right)\right]^{1/2}} \tag{1.86}$$

Where the Reynolds number is based on the droplet diameter and the relative velocity. Thermo-physical properties are evaluated at the mean temperature.

The presence of a convective environment modifies only the mass and energy balances in the outer region (oxidizer distribution in the outer region and temperature distribution in the outer region), and the energy balance at the flame sheet.

Introducing the film theory boundary conditions we obtain:

For species conservation:

$$Y_{Ox}(r \to \infty) = 1 \xrightarrow{becomes} Y_{Ox}(\delta_M) = 1 \tag{1.87}$$

The new relation between the burning rate,$\dot{m}_F$, and the flam radius,r_f , is:

$$\frac{e^{-\frac{Z_F \dot{m}_F}{r_s\, Nu/(Nu-2)}}}{e^{-\frac{Z_T \dot{m}_F}{r_f}}} = \frac{\upsilon+1}{\upsilon} \tag{1.88}$$

For energy conservation:

$$T(r \to \infty) = T_\infty \xrightarrow{becomes} T(\delta_T) = T_\infty \tag{1.89}$$

The new outer region temperature distribution is:

$$T(r) = \frac{(T_f - T_\infty)e^{-\frac{Z_T \dot{m}_F}{r}} + T_\infty e^{-\frac{Z_T \dot{m}_F}{r_f}} - T_f e^{-\frac{Z_T \dot{m}_F (Nu-2)}{r_s\, Nu}}}{e^{-\frac{Z_T \dot{m}_F}{r_f}} - e^{-\frac{Z_T \dot{m}_F (Nu-2)}{r_s\, Nu}}} \tag{1.90}$$

Whit the above equation we can now write the energy balance at the flame sheet as:

$$\frac{c_{pg}}{\Delta h_c}\left[\frac{(T_f - T_s)e^{-\frac{Z_T \dot{m}_F}{r_f}}}{e^{-\frac{Z_T \dot{m}_F}{r_f}} - e^{-\frac{Z_T \dot{m}_F}{r_s}}} - \frac{(T_f - T_\infty)e^{-\frac{Z_T \dot{m}_F}{r_f}}}{e^{-\frac{Z_T \dot{m}_F}{r_f}} - e^{-\frac{Z_T \dot{m}_F (Nu-2)}{r_s\, Nu}}}\right] - 1 = 0 \tag{1.91}$$

The new equation system to solve is:

	Equation	***Unknown***
Fuelspecies conservation in the inner zone	$Y_{F,s} = 1 - \frac{e^{-\frac{Z_F \dot{m}_F}{r_s}}}{e^{-\frac{Z_F \dot{m}_F}{r_f}}}$	$\dot{m}_F, r_f, Y_{F,s}$
Oxidizer species conservation in the outer zone-MODIFIED	$\frac{e^{-\frac{Z_F \dot{m}_F}{r_s\, Nu/(Nu-2)}}}{e^{-\frac{Z_T \dot{m}_F}{r_f}}} = \frac{\upsilon + 1}{\upsilon}$	$\underline{\dot{m}_F, r_f}$
Energy balance at the droplet liquid-vapor interface	$\frac{c_{pg}(T_f - T_s)}{h_{fg} + q_{i-l}} \cdot \frac{e^{-\frac{Z_T \dot{m}_F}{r_s}}}{e^{-\frac{Z_T \dot{m}_F}{r_s}} - e^{-\frac{Z_T \dot{m}_F}{r_f}}} + 1 = 0$	$\dot{m}_F$, r_f, T_f, T_s
Energy balance at the flame sheet-MODIFIED	$\frac{c_{pg}}{\Delta h_c}\left[\frac{(T_f - T_s)e^{-\frac{Z_T \dot{m}_F}{r_f}}}{e^{-\frac{Z_T \dot{m}_F}{r_f}} - e^{-\frac{Z_T \dot{m}_F}{r_s}}} - \frac{(T_f - T_\infty)e^{-\frac{Z_T \dot{m}_F}{r_f}}}{e^{-\frac{Z_T \dot{m}_F}{r_f}} - e^{-\frac{Z_T \dot{m}_F (Nu-2)}{r_s\, Nu}}}\right] - 1 = 0$	$\dot{m}_F$, r_f, T_f, T_s
Liquid vapor phase equilibrium at interface (Clausius-Clapeyron)	$Y_{F,s} = \frac{A \cdot MW_F \cdot e^{-\frac{B}{T_s}}}{A \cdot MW_F \cdot e^{-\frac{B}{T_s}} + \left[p - A \cdot e^{-\frac{B}{T_s}}\right] \cdot MW_{Pr}}$	$T_s, Y_{F,s}$

In this case we obtain the following formulation that provides the burning rate of a fuel droplet in a convective environment:

$$\dot{m}_F = \frac{2\pi k_g r_s Nu}{c_{pg}} ln(1 + B) \tag{1.92}$$

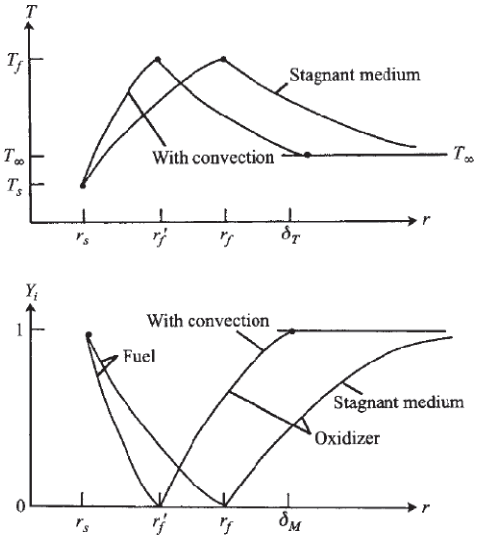


Fig.1.6 Temperature and species profile according the "Film theory"

We can see that according the film theory, the droplet combustion in a convective environment reduces the flame radius to droplet radius ratio.

This simple quasi-steady droplet combustion model is best at predicting fuel mass flow rate, $\dot{m}_F$, but is not accurate at predicting the flame temperature, T_f (lower than observed values), and the flame to droplet radius,$\frac{r_f}{r_s}$ (higher than observed values)[13].

In the simplified quasi-steady model all properties are treated as constants, and the selection of the mean values is used to provide agreement with experimental observations. In reality, many of the properties have a strong temperature and/or composition dependency. To consider this aspect [13] proposes a modification of this simple quasi-steady model that predicts more accurately results, assuming discontinuity in heat transfer and transport coefficients across the flame sheet and non unity of the Lewis number for both inner and outer region.

The unrealistically large flame stand-off distance predicted by the D^2 law is shown by Law et al. [7] to be a result of neglecting the unsteady effect of fuel vapor accumulation between the droplet surface and the flame.

The simple quasi-steady model shown above requires that the liquid fuel consists of a single component, when fuel oil is a multi-component mixture. Is required to assume as a fuel a mono-component fuel, that could well represent the real fuel.

Finally, is important to consider the issue of the interactions among multiple droplets as would be present in a fuel spray.

CHAPTER 2

FLUE GAS ANALYSIS

The composition of the flue gases is important for many reasons:

1. Assessment of pollutant emissions;
2. Assessment of air-to-fuel ratio;
3. Valuation of the combustion efficiency;
4. Calculation of the dew point temperature of the flue gases;

2.1 POLLUTANT EMISSIONS

Beside the main products that a combustion process produces, i.e. CO_2 an H_sO, there other kinds of compounds in the emissions from a fossil fuel-fired heat generator. These can be classified by their effects on the environment (Environmental Impact) or by their origin.

Generally accepted Environmental Impacts categories are the following:

- Global Warming Potential (GWP):in this category are included CO_2 ,CO and CH_4 .These emissions are considered to be responsible of the Greenhouse Effect.
- Acidification Potential(AP): in this category are included the sulfur and nitrogen oxides, SO_x snd NO_x,
- Volatile Organic Compounds (VOC) in this category are included the emissions of hydrocarbon compounds, C_xH_y, from oil-fired burners.

There are essentially three systems that can be adopted to reduce the pollutants:

Preventive systems, by acting on the fuel before subjecting it to combustion, trying to reduce the amount of polluting agents. These systems involve the control of the “Unavoidable products from the combustion reaction” and the “Pollutants that are unavoidable because they are already contained in the fuel”.
A typical case is represented by liquid fuels: a good option to reduce the combustion emissions is to chose a fuel with low sulfur and nitrogen content.

Primary systems, by acting on the process and combustion equipment (burner), so that combustion takes place under the best conditions thus reducing the formation of pollutants. These systems involve the control of the “Emissions that are consequence of incomplete combustion” and the “Emissions that don’t involve the fuel, but are connected with the chemical reaction between air molecules triggered by the specific combustion conditions”.

Secondary systems, by acting on the combustion gases, to break down the polluting components before they are expelled into the atmosphere. These systems involve the control of all the above emissions groups. This option is taken into account only in the case of large thermal plants.

The Primary Systems will be discussed, since are the most interesting from the heat generator's design characteristics.
To understand the strategies for the emission control is better to explore the mechanisms and causes that origin these emissions.
For our purpose, a more interesting categorization of the heat generator emissions is by their origin.
Is possible to define the following groups:

- *Unavoidable products from the combustion reaction*: a combustion reaction of an hydrocarbon will always be accompanied with a carbon dioxide and water production. The amount of CO_2 generated is totally depending on the carbon content in the fuel.
  Selected a certain fuel, there is no way to avoid or reduce the generation of these two compounds unless reducing or not executing the combustion process.

- *Pollutants that are unavoidable because they are already contained in the fuel*: this is the case of sulfur oxides, SO_x , which are generated from the oxidation of the sulfur content initially in the fuel. To reduce or avoid its generation has to be chosen a fuel with a low content of sulfur. In the case of liquid fuels, also some part of Nitrogen Oxides, NO_x, are produced from the oxidation of the nitrogen content in the fuel.

- *Emissions that are consequence of incomplete combustion*: basically all the compounds of carbon, except the carbon dioxide are included in this category, CO, C_xH_y and particulate matter, PM, i.e. soot. The causes of incomplete combustion are principally:

  - Lack of oxygen to complete the combustion of the fuel, and
  - Fuel temperature too low to permit oxidation to occur. This phenomenon might happen as a result of flame impingement with the metal walls of the combustion chamber, that determines a cooling of the flame that could reach a temperature below the burn temperature of the fuel. In the case of liquid fuel the incomplete combustion can occur also for problems in the atomization-vaporization phase.
  - During the start and stop of combustion the unburned gases (in pre purge and after purge phases) are sent to the chimney. This causes a considerable amount of unburned emissions.

- *Emissions that don't involve the fuel, but are connected with the chemical reaction between air molecules triggered by the specific combustion conditions*: this is the case of nitrogen oxides. NO_x, that are produced from the reaction between the oxygen and nitrogen in the air. The conditions that allow the NO_x are: enough air in the combustion zone and temperature sufficiently high (above 1200°C).

We will focus on the NO_x , CO and Particulate Matter (PM).

2.1.1 NITROGEN OXIDES

In the combustion of fuels that contain no nitrogen, nitric oxide, NO, is formed by three chemical mechanisms that involve the nitrogen from the air:

- The thermal or Zeldovich mechanism that dominates in high temperature combustion over a fairly wide range of equivalence ratios;
- The prompt or Fenimore mechanism that is particular important in fuel-rich combustions;
- The N_2O-intermediate mechanism, plays an important role in the production of NO in very fuel-lean, low-temperature combustion processes.

Three models of nitric oxide formation exist, which lead to the formation of different types of nitric oxide (different by type of origin but not by chemical composition); respectively they are:

- Thermal nitric oxides (thermal NOx);
- Prompt nitric oxides (prompt NOx);
- Fuel nitric oxides (fuel NOx);

Thermal nitric oxides are formed by the oxidation of atmospheric nitrogen (contained in combustion supporter air) under high temperature (T>1500 K) and high oxygen concentration conditions, and represent the majority of nitric oxides in the case of gaseous fuels (methane and LPG) and in general in

fuels which do not contain nitrogenous compounds. The mechanism proposed by Zeldovich is represented by the following chain reactions (extended Zeldovich mechanism):

$$N2 + O \leftrightarrow NO + N$$
$$N + O2 \leftrightarrow NO + O$$
$$N + OH \leftrightarrow NO + H$$

Prompt nitric oxides are formed by means of the fixation of atmospheric molecular nitrogen by hydrocarbon fragments (radicals) present in the flame area; this method of forming oxides is extremely rapid thus giving rise to the name prompt. Fenimore characterized this mechanism with the following chain reactions:

$$CH + N_2 \longleftrightarrow HCN + N$$
$$C + N_2 \leftrightarrow CN + N$$
$$HCN + O \leftrightarrow NH + H$$
$$NH + H \leftrightarrow N + H_2$$
$$N + OH \leftrightarrow NO + H$$

Their formation essentially depends on the concentration of radicals in the first stage of the flame; for oxidative flames (combustion with excess of oxygen), their contribution is negligible, while in the case of rich mixtures and for low-temperature combustion, their contribution may reach 25% of the full nitric oxides total.

Nitric oxides from fuel form by means of oxidation of the nitrogenous compounds contained in the fuel within the flame area, and their production is significant when the fuel's nitrogen content exceeds 0.1% in weight, essentially only for liquid and solid fuels.

The relative importance of the three NO formation mechanisms, for different fuels, is shown in the following graph:

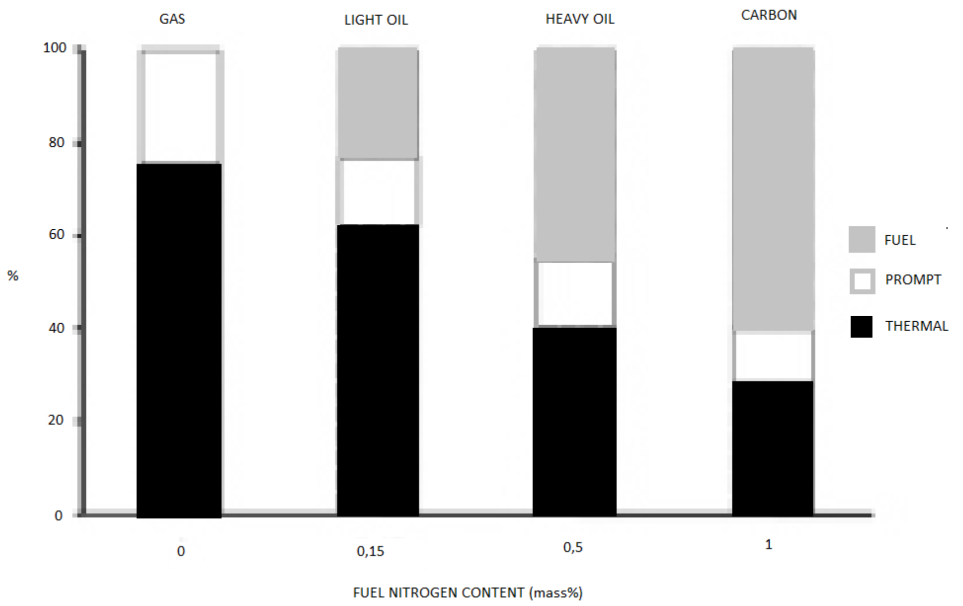


Fig. 2.1 Percentage of NO_x formation mechanisms

From Fig.2.1 we can observe that the most important mechanism source of NO is the Thermal mechanism that in some cases could constitute up to the 80% of the total NO_x emissions.

There are three basic principles of primary NOx control to reduce NOx formation

- Reduction of high combustion/flame temperature since more NOx will be formed at higher temperatures under thermodynamic equilibrium conditions

- Reduction of residence time at high combustion temperature to resist the NOx formation approaches thermodynamic equilibrium concentration.

- Reduction of oxygen concentration and hence the nascent oxygen concentration in the high temperature zone.

The primary NOx control technologies involve either or both of the following:

- Modification of fuel/air delivery-burner system.

- Modification of gas burner.

2.1.2 CARBON MONOXIDE

Carbon monoxide is a result of a partial oxidation of the carbon present in the fuel. It appears when the combustion is carried out with too low air than the stoichiometric requirements, so that the oxygen is not sufficient to complete the oxidation of carbon, obtaining carbon dioxide. The presence of carbon monoxide is an indicator of low Combustion Efficiency . The carbon oxide production in heating systems is usually avoided performing the combustion with a sufficient air excess.

2.1.3 TOTAL SUSPENDED PARTICLES

The total particulate matter, in the case of liquid-fired heat generators (the principal category of PM producers) is primarily constituted by Soot.
The formation of Soot can be regarded as a intrinsic property of most non premixed flames.
Soot is formed in the fuel-rich regions and is consumed in fuel-lean regions. The chemistry and physics of soot formation is really complex, but it could be figured as a four steps sequence:

1. Formation of precursors species: formation of the polycyclic aromatic hydrocarbons, from reactions who start with the original fuel;
2. Particle inception: formation of small particles of a critical size from growth by both chemical means and coagulation;
3. Surface growth and particle agglomeration: these particles continue to be exposed to the bath of species, traveling through the flame;
4. Particle oxidation: at some point this soot particle will pass through an oxidizing region of the flame;

If all the soot particles formed in the flame are not oxidized in the flame, the residual part wil be emitted in atmosphere.

Soot is a hazardous combustion sub-product for human health, and also a problem for the combustion devices, causing surface fouling.

In the technical field the measure of the soot emission by flames is given in terms of Bacharach Number, an empiric procedure that consists in filtering a known amount of flue gas with a proper white filter. This will capture the suspended particles in the flue gas, changing color. By comparing the filter color with a proper reference, is obtained the Bacharach Number, that is a representation of the actual Soot content of the flue gas.

In order to reduce the production of these components, it is necessary to increase the period spent in the combustion chamber, for liquid fuels has to be guaranteed a good atomization and vaporization process, and guarantee the fuel an adequate excess of oxygen and enhance an intimate mixing [14].

2.2 DEW POINT TEMPERATURE OF FLUE GAS

The Combustion Products of all Hydrocarbon combustion contain a variable quantity of water. If this reaction product is not condensate, the Combustion Products can be seen as a binary homogeneous mixture of dry gases and water vapor, likewise wet air.

The quantity of water vapor in the Combustion Products has a relevant importance in Condensation Boilers, since it determines:

- The extra heat (latent heat) that these devices can extract in comparison to the Non-Condensing Boilers;
- The temperature at which this extra heat recovery can take place.

Focusing on the last point, we can say: is possible to recover the latent heat of water vapor content in a given flue gas mixture, if this water vapor is carried at the condensation temperature (dew point temperature T_{dew}, which depend exclusively on the partial pressure of water vapor in the considered mixture sample.

In other words, T_{dew}represents the temperature at which the water vapor has a saturation pressure equal to the partial pressure of the water vapor in the mixture. If the temperature of the flue gases is carried to T_{dew}, this mixture will be saturated of water $(p_v = p_s)$, and if the mixture temperature falls, the partial pressure of water vapor in the mixture will be greater than the water vapor saturation $(p_v > p_s)$. So the extra water vapor will condensate in liquid water, to restore the equilibrium $(p_v = p_s)$.

We can estimate the dew point temperature T_{dew} by knowing the total pressure of the mixture p, th temperature of the mixture T and the partial pressure of water vapor in the mixture p_v.

The partial pressure of the water vapor in the mixture is given by:

$$p_v = p\frac{n_i}{n_{tot}} = px_v \tag{2.1}$$

The saturation temperature of the mixture, T_s, can be found in properties tables of water, imposing a saturation pressure $p_s = p_v$, or with specific equations. The value of the dew temperature is simply:

$$T_{dew} = T_s \tag{2.2}$$

Two issues influence the dew temperature of the flue gas:

- Fuel composition;
- Excess air and its moisture;

Water vapor production in a combustion process depends exclusively on the hydrogen content of the fuel, since the only reaction that provides water as a product is the hydrogen oxidation. Higher is the hydrogen content in the fuel, higher will be the water vapor production. Among hydrocarbons , the maximum water vapor production is obtained with the combustion of methane (CH_4), which has the lower carbon-to-hydrogen ratio (C:H=3 by mass). Increasing the number of carbon atoms in the fuel’s molecule , this ratio creases, and decreases the water production from combustion as well.

Decreasing the C:H ratio of the fuel, for a stoichiometric combustion, the water steam content will decrease. This determines a lower concentration of water vapor in the flue gas, then a lower partial pressure of water vapor in mixture, thus the dew point temperature of the flue gas falls down.

Comparing the most common fuels in domestic heating, Methane and Fuel Oil, we can find the following characteristics:

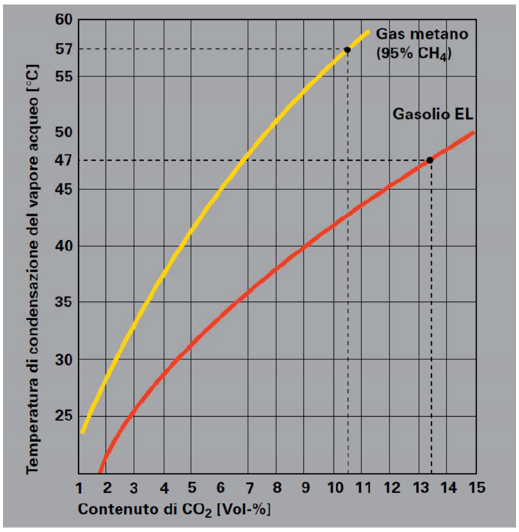


Fig 2.2 Dew point Temperature vs. CO_2 % in the flue gases for Gasoil and Methane

The curves represent the dew point temperature of the Combustion Products as a function of the volumetric percentage of CO_2 in the mixture. Volumetric concentration of CO_2 is a measurable parameter that provides an indication of the fuel-to-air ratio λ.

Stoichiometric combustion ($\lambda = 1$) is realized for a CO_2 volumetric fraction that is a fix value for each kind of fuel.

As we can see, in stoichiometric combustion of Fuel Oil is obtained a dew point temperature of flue gas of about 10°C lower than the Methane Combustion Products.

If now we consider the effect of excess of air on the dew point temperature, we can see that, decreasing the volumetric fraction of CO_2 content in the flue gas, i.e. we provide more than the stoichiometric amount of air ($\lambda > 1$), the dew point temperature will decrease, since the water vapor’s volumetric fraction will be lower, thus the partial pressure of vapor is lower as well, thus is required a lower temperature to reach the saturation condition for water vapor.

We have to underline that lower is the dew point temperature, more difficult will be to reach the condensation conditions in a condensing device. An talking about boilers, a low value of the dew point temperature determines the request to the heating system to work with low temperatures, as will be discussed later.

Obviously, when we burn a fuel in air, we should consider the moisture content carried by air, because this will be added to the water vapor content of the flue gases.

2.3 CONDENSATION OF FLUE GAS

Condensation of flue gas is desired in order to increase the heat recovered from the combustion of the fuel. This is possible only if the internal wall temperature of one of the section of the flue gas path is lower than the dew point temperature of the flue gas. This temperature depends on the type of fuel and on the excess air.

Thus, if we measure a return water temperature higher than the flue gas dew temperature, we can state that no condensation will occur (inside the boiler). Therefore, the return water temperature is a key factor in achieving the condensing conditions.

The condensation of pure water vapor is a well known mechanism. The first studies about this topic were done by Nusselt. A lot of works were made to analyze experimentally and theoretically the condensation heat transfer of water vapor in mix with small quantities of non-condensable gases. On the other hand, when we talk about the phenomenon of water vapor condensation of flue gas, the situation is quite different. In this case the volumetric percentage of non-condensable gases is as high as 80% ÷ 85%, and the theories above mentioned do not refer to this case.

To study this situation, Colburn and Hougen introduced the “film model”, a simple physical description of the phenomena that occur in vapor condensation with high non-condensable gas percentage processes.

To explain what the Colburn and Heugen theory say, let’s refer to the following situation: the flue gas with water vapor at a superheated state flows touching the tube walls. The water vapor entrained in the non condensable gases, that are dominant in volume, condenses into liquid water while the tube wall temperature of the heat exchanger is below the saturation temperature of the water vapor, the tube wall will be partially or totally wetted by water film.

The water film is surrounded by the gas boundary layer around the tube, the condensation of water vapor occurs at the interface of the water film and the boundary layer due to the diffusion driven by the vapor concentration difference across the layer.

The heat transferred through the film includes two parts, which are the sensible heat due to the forced convection of the gases and the latent heat due to the vapor condensation respectively.

Along the flow path of flue gases the water vapor content is gradually decreased because of its being condensed, which gives rise to a continuous reduction of corresponding partial pressure, thus saturation temperature.

In [15] was proposed a theoretical analysis of the process of condensation of moistened gas.

When non-condensable gases are absent, the bulk vapor temperature $T_{b,pc}$ and the liquid–vapor interface temperature $T_{i,pc}$ are identical and equal to the saturation temperature of pure vapor T_{sat};

when non-condensable gases exist, there is a drop between the vapor saturation temperature in the bulk and the temperature at the liquid–vapor interface, which is due to the reduction in vapor partial pressure$(p_{b,v} > p_{i,v})$ and can be considered as a penalty in the condensation process.

It can be imagined that the vapor must diffuse through the non-condensable gas boundary layer for it to attain the liquid–vapor interface, so that the vapor concentration difference between the bulk and the interface is the driving force of mass transfer.

In addition, there is a drop from bulk gas temperature T_g to the interfacial temperature T_i, which is the driving force of convection heat transfer.

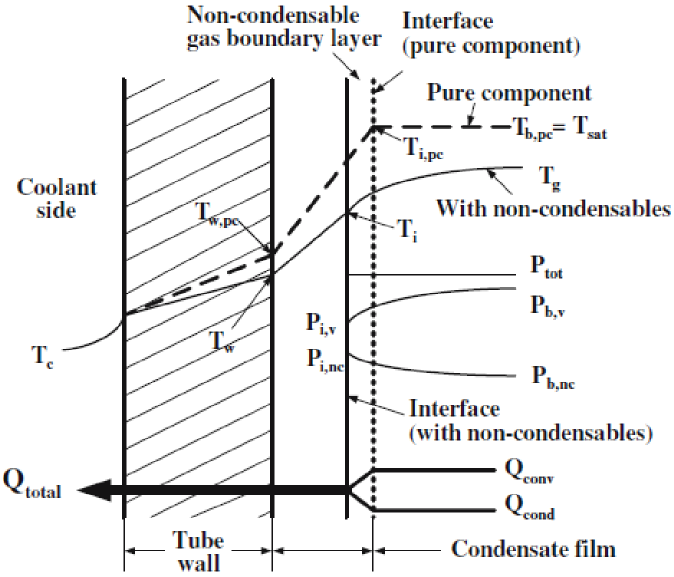


Fig.2.3 The Film Model

Based on this model, a heat balance at the liquid–gas interface between the heat transfer through the non-condensable gas boundary layer and the heat transfer through the condensate film can be obtained. As the heat transfer though non-condensable gas boundary layer consists of two components: the sensible heat flux q_{conv} due to temperature difference between bulk vapor– gas mixture and interface, and the latent heat flux q_{cond} given up by the condensing vapor, therefore:

$$q_{tot} = q_{conv} + q_{cond} \tag{2.3}$$

Where:

$$q_{conv} = h_{cv}(T_g - T_i) \tag{2.4}$$

$$q_{cond} = m''_{cond} H_{fg} \tag{2.5}$$

h_{cv}: *forced convection heat transfer coefficient due to sensible heat;*
T_g: *bulk vapor-gas mixture temperature;*
T_i: *interface temperature;*

m''_{cond}: *interfacial condensate flux;*
H_{fg}: *latent heat of vaporization of the vapor under the corresponding partial pressure;*

the overall heat transfer coefficient $h_{overall}$ can be determined by the following:

$$h_{overall}(T_g - T_i) = h_{cv}(T_g - T_i) + m''_{cond}H_{fg} \tag{2.6}$$

Forced convection heat transfer can be described by the Nusselt number and forced convection mass transfer can be described by the Sherwood number as:

$$Nu = CRe^{m}Pr^{1/3} \tag{2.7}$$
$$Sh = C'Re^{m'}Sc^{1/3} \tag{2.8}$$

The Nusselt number is defined as:

$$Nu = \frac{hd}{\lambda} \tag{2.9}$$

The Sherwood number is defined as:

$$Sh = \frac{g_m d}{\rho D} \tag{2.10}$$

The Prandtl number is defined as:

$$Pr = \frac{\mu c_p}{\lambda} = \frac{\rho v c_p}{\lambda} \tag{2.11}$$

The Schmidt number is defined as:

$$Sc = \frac{\mu}{\rho D} = \frac{v}{D} \tag{2.12}$$

The Reynolds number is defined as:

$$Re = \frac{\rho u d}{\mu} = \frac{ud}{v} \tag{2.13}$$

Where:

h: *heat transfer coefficient* $\frac{W}{m^2K}$
λ: *thermal conductivity* $\frac{W}{mK}$
d: *characteristic length* m
g_m: *mass transfer coefficient* $\frac{kg}{m^2s}$
ρ: *density* $\frac{kg}{m^3}$

D: *diffusion coefficient* $\frac{m}{s}$
μ: *dynamic viscosity* $\frac{kg}{ms}$
c_p: *specific heat* $\frac{kJ}{kgK}$
υ: *kinematic viscosity* $\frac{m^2}{s}$
u: *flow linear velocity* $\frac{m}{s}$

A close analogy exists between convective heat and convective mass transfer owing to the fact that conduction and diffusion in a fluid are governed by physical laws of identical form, that is, Fourier's and Fick's laws, respectively.

As a result, in many circumstances the Sherwood can be obtained in a simple manner from the Nusselt number for the same flow conditions.

Indeed, in most gas mixtures Sh is nearly equal to its heat transfer counterparts. For dilute mixtures and solutions and low mass transfer rates, the rule for exploiting the analogy is simple: The *Sherwood is obtained by replacing the Prandtl numberby the Schmidt number in the appropriate heat transfer correlation.[16]*

For this reason the coefficients and the exponents in the previous correlations should be equal:$C = C'; m = m'$.

Therefore:

$$\frac{Nu}{Sh} = \left(\frac{Pr}{Sc}\right)^{1/3} \tag{2.14}$$

$$\frac{h}{g_m} = \left(\frac{D}{a}\right)^{1/3} \frac{\lambda}{D} = \rho c_p \left(\frac{a}{D}\right)^{2/3} = \rho c_p Le^{2/3} \tag{2.15}$$

Where $a = \frac{\lambda}{\rho c_p}$ is the thermal diffusivity and Le is the Lewis number, the thermal to mass diffusivity ratio.

The vapor must diffuse through the non-condensable gas boundary layer to reach the liquid-vapor interface. The mass balance at the interface yields:

$$m''_{cond} = \left(-\rho D \frac{\partial W_v}{\partial y}\right)_i + W_{v,i}\left(m''_{tot}\right)_i \tag{2.16}$$

$$m''_{nc} = \left(-\rho D \frac{\partial W_{nc}}{\partial y}\right)_i + W_{nc,i}\left(m''_{tot}\right)_i \tag{2.17}$$

Where:

m''_{cond}: *interfacial condensate flux;*
m''_{nc}: *interfacial non condensable flux;*

m''_{tot}: *imaginary compensatory mixture flow at the interface to account for the situation that the diffusion process is in one direction only*
W: *vapor (subscript "v,i") and gas (subscript "nc,i") mass fraction at the interface*

As the condensate surface is impermeable to the non-condensable gas , m''_{nc}=0 than:

$$W_{nc,i}\left(m''_{tot}\right)_i = \left(\rho D \frac{\partial W_{nc}}{\partial y}\right)_i \qquad (2.18)$$

Since the system is composed of two components: condensing vapor and non condensable gas, the mass fractions follow the equation:

$$W_{nc}+W_v = 1 \qquad (2.19)$$

Therefore:

$$\frac{\partial W_{nc}}{\partial y} + \frac{\partial W_v}{\partial y} = 0 \qquad (2.20)$$

Combining eq. (2.16),(2.18) and (2.20) and simplifying, we obtain:

$$m''_{cond} = \frac{\left(-\rho D \frac{\partial W_v}{\partial y}\right)_i}{1-W_{v,i}} = g_m \frac{\left(W_{v,b}-W_{v,i}\right)}{\left(1-W_{v,i}\right)} = g_m B \qquad (2.21)$$

Where:

g_m:*mass transfer coefficient* $\frac{kg}{m^2 s}$;
$B = \frac{\left(W_{v,b}-W_{v,i}\right)}{\left(1-W_{v,i}\right)}$: *represents the driving force of mass transfer;*
$W_{v,b}$:*is the vapor mass fraction in the bulk mixture;*
$W_{v,i}$:*is the vapor mass fraction in the interface;*

The condensation heat transfer coefficient h_{cd} accounting the latent heat transfer is defined as:

$$h_{cd}\left(T_g - T_i\right) = m''_{cond} H_{fg} \qquad (2.22)$$

Therefore:

$$h_{cd} = \frac{m''_{cond} H_{fg}}{\left(T_g - T_i\right)} = g_m B H_{fg} \frac{1}{\left(T_g - T_i\right)} \qquad (2.23)$$

Defining the Nusselt number for condensation:

$$Nu_{cd} = \frac{h_{cd} d}{\lambda} = g_m B H_{fg} \frac{d}{\left(T_g - T_i\right)\lambda} \qquad (2.24)$$

Recalling the Sherwood number’s definition:

$$Sh = \frac{g_m d}{\rho D} \tag{2.25}$$

$$Nu_{cd} = \frac{g_m d}{\rho D}\frac{Pr}{Sc}B\frac{H_{fg}}{c_p(T_g - T_i)} \tag{2.26}$$

In the last equation we can recognize the following factors:

$$\frac{g_m d}{\rho D} = Sh \tag{2.27}$$

$$\frac{H_{fg}}{c_p(T_g - T_i)} = Ja \tag{2.28}$$

Ja , the Jacob number, represents the phase change energy to the sensible heat transfer ratio;

$$\frac{Pr}{Sc} = Le \tag{2.29}$$

So that the equation for the condensation Nusselt number becames:

$$Nu_{cd} = ShBLeJa \tag{2.30}$$

Applying equation (2.8) to the previous one, we obtain:

$$Sh = CRe^m Sc^{1/3} \tag{2.31}$$

$$Sc = \frac{Pr}{Le} \tag{2.32}$$

$$Sh = CRe^m \left(\frac{Pr}{Le}\right)^{1/3} \tag{2.33}$$

$$Nu_{cd} = CRe^m \left(\frac{Pr}{Le}\right)^{1/3} LeBJa = CRe^m Pr^{1/3} Le^{2/3} BJa \tag{2.34}$$

The overall heat transfer is due to the convective heat transfer and to the condensation heat transfer

$$q_{tot} = q_{conv} + q_{cond} \tag{2.35}$$

Thus the overall Nusselt number can be obtained as:

$$Nu_{overall} = \frac{h_{overall} d}{\lambda} = \frac{(h_{cv} + h_{cd})d}{\lambda} = Nu_{cv} + Nu_{cd} \tag{2.36}$$

Nu_{cv} is the Nusselt number due to convection and it can be calculated using a correlation valid for the case of interest, such as:

$$Nu_{cv} = CRe^m Pr^{1/3} \tag{2.37}$$

Nu_{cd} is the Nusselt number due to condensation is the one the was obtained in the previous discussion:

$$Nu_{cd} = CRe^{m}Pr^{1/3}Le^{2/3}BJa \qquad (2.38)$$

$$Nu_{overall} = Nu_{cv} + Nu_{cd} = CRe^{m}Pr^{1/3} + CRe^{m}Pr^{1/3}Le^{2/3}BJa \qquad (2.39)$$

$$Nu_{overall} = CRe^{m}Pr^{1/3}\left(1 + Le^{2/3}BJa\right) \qquad (2.40)$$

The equation above is the normalized correlation of convection-condensation heat transfer, in which C and m are constants to be determined by experimental data.

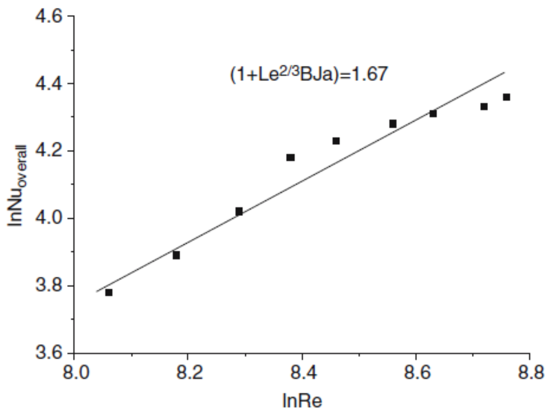


Fig.2.4 Relationseep between Nusselt number and Reynolds Nuber Based on the experimental data, [Liang et al.]

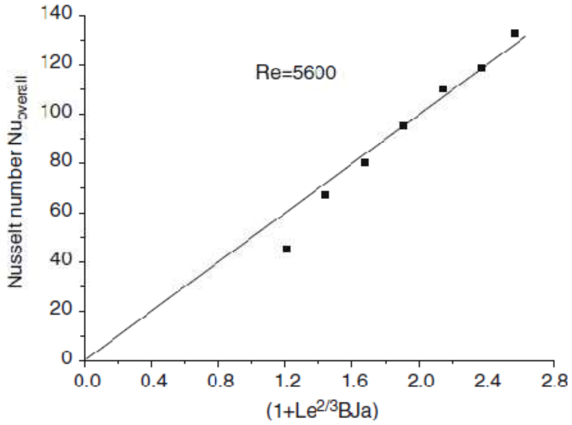


Fig.2.5 Relationseep between Nusselt number and $(1 + Le^{2/3}BJa)$Based on the experimental data, [Liang et al.]

If Re is raised, the sensible heat transfer is enhanced. it can be found that for a given value of $(1 + Le^{2/3}BJa)$, the increase of sensible heat transfer leads to the increase of total convection–condensation heat transfer.

In the expression of the augmentation factor $(1 + Le^{2/3}BJa)$, $Ja = \frac{H_{fg}}{c_p(T_g - T_i)}$is the ratio of the phase change energy to the sensible heat transfer$B = \frac{(W_{v,b} - W_{v,i})}{(1 - W_{v,i})}$: is the driving force of mass transfer, and its value increases as the vapor mass fraction increases.

In the case of higher vapor mass fraction (i.e., higher value of B), more water vapor can be condensed and more latent heat can be released, the overall heat transfer coefficient is higher. Therefore, higher value of $(1 + Le^{2/3}BJa)$, means higher value of $Nu_{overall}$.

[17]studied the same problem as above, but for another configuration: condensation of wet flue gas inside of a vertical straight pipe.

2.4 COMBUSTION EFFICIENCY

A general definition of Combustion Efficiency is given by the formula [18]:

$$\eta = \frac{\mathrm{H_{react}(T)} - \mathrm{H_{prod}(T)}}{m_{fuel}LHV} \qquad (2.41)$$

That considers the Lower Heating Value since is assumed that water content in the products is in liquidphase.

The reactants enthalpy, $\mathrm{H_{react}(T)}$, involves not only the chemical energy content in the fuel (LHV) and air (that is zero), but also the sensible enthalpy of them. So, is taken in account the preheating of reactants.

The products enthalpy, $\mathrm{H_{prod}(T)}$, is referred to the real composition of the products mixture, considering all species content, such as unburned fuel (CO,H_2 and C_xH_y) due to a not-perfect mixing of oxygen and fuel. Higher is this value (high temperature of product mixture) lower is the heat extract form the fuel (stack loss).

The difference in the numerator represents the effective energy amount provided by the combustion process. Complete combustion efficiency would extract all the energy available in the fuel.

Combustion efficiency increases with [19]:

- increasing temperature of the reactants;
- increasing time that the reactants are in contact;
- increasing surface areas;
- increasing stored chemical energy;

The specific heat of combustion is a chemical property that refers to the amount of energy that can theoretically be extracted from a fuel at 100 % combustion efficiency. The Lower Heating Value is a more realistic term and does not include the condensation of the water vapor produced. It is thus more easily applied to real combustion processes.

One way of increasing the temperature of the reactants is to preheat them by a heat exchanger, that uses a low grade heat available from flue gas, before being injected into the combustion chamber.

Increasing time and surface of contact between fuel and oxidizer is a design task.

For a liquid fuel combustion, the oil must be dispersed into the combustion air in an even spray of fine droplets. The droplets must be sufficiently small (increasing the surface of contact-to-volume of fuel ratio) to allow for the process of droplet evaporation and combustion of fuel vapor to take place in the available time within the flame.

The creation of a fuel spray before the burning process is called Atomization, and we will discuss this topic further.

2.4.1 COMBUSTION EFFICIENCY IN NON CONDENSING BOILERS

In the boiler field is not adopted this definition of Combustion Efficiency, since to measure all the parameters involved in its calculation (enthalpy of products and reactants), are required very intrusive and sophisticated measuring tools.

However there are simplified approaches [20] to determine the Combustion Efficiency, that can provide useful information related to the effective Combustion Efficiency that was given in equation (2.41). The simplified Combustion Efficiency definition is:

$$\eta_{comb} = 100\% - Q_{stack} \tag{2.42}$$

That represents the ratio between the energy extracted from the fuel and potential energy contained in the fuel.

Where Q_{stack} is Stack Loss, i.e. the amount of heat transferred by combustion gases to the environment, that is commonly calculated on the basis of the empirical "Siegert formula" [21]:

$$Q_{stack} = \left(T_{flue} - T_{amb}\right) \cdot \left(\frac{A1}{CO_2} + B\right) \tag{2.43}$$

Where:

$$A1 = CO_{2,MAX} \cdot A_{th} \cdot B \tag{2.44}$$

A_{th}: *is the theoretical air* $\left[\frac{kg_{air}}{kg_{fuel}}\right]$;
B: *is a fuel dependent parameter*$[°C^{-1}]$;
$CO_{2,MAX}$: *is the stoichiometric volume fraction of* CO_2 *in the flue gas* $[m^3/m^3]$;
T_{flue}: *is the flue gas temperature [°C];*
T_{amb}: *is the ambient temperature [°C].*

The above formula (2.42) assume that the only quantity decreasing the combustion efficiency is the stack loss. Thus, them neglect incomplete combustion losses, radiation losses, etc.
This simplification is necessary due to the inability to measure the amount of other losses with a gas analyzer, device usually adopted in efficiency analysis in the domestic boiler field.
The simplification introduced to obtain the formulae above makes the value of efficiency calculated in this way not as precise, but enough useful for the tuning operation of a boiler, since they reflect precisely the tendencies of efficiency change.
To improve the accuracy of the Combustion Efficiency, taking in account the incomplete combustion loss, is introduced the so called Loss by Incomplete Combustion,F_{inc} . It represents the percentage of energy loss caused by the presence of flammable gases (in this case mainly CO) in the combustion gases.

The loss caused by incomplete combustion is calculated on the basis of the measured CO concentration in the combustion gases according to the following formula:

$$F_{inc} = \frac{\alpha \cdot CO}{CO + CO_2} \quad (2.45)$$

Where:

CO, CO_2: are the volume concentrations (%) in the flue gas;
α: is a factor specific for a given fuel.

Calculating F_{inc} enables a correction of the previously calculated Combustion Efficiency (2.42). Then the so-called corrected efficiency η^* is given by:

$$\eta^*_{comb} = \eta_{comb} - F_{inc} \quad (2.46)$$

2.4.2 COMBUSTION EFFICIENCY IN CONDENSING BOILERS

The Combustion Efficiency is referred to the fuel's lower heating value, neglecting the potential energy contained in the fuel coming from the condensation of the water vapor in the combustion products, and represented by the higher heating value.

The reason of this choice came from the fact, that in the past, was not welcome any condensation of vapor present in the combustion products because of the condensate is characterized by a strong acidity that could damage the materials of which the old heat generators were built. With the introduction of new materials for the construction of heat generators, resistant to the corrosive action of the flue gas condensate, a new technology for boilers was born: the Condensing Boiler.

The condensing technology allows to cool the combustion products under the dew point temperature of the flue gases before they are released in the atmosphere. This determines two benefits from the energetic point of view:

- Cooling the combustion products, letting them reach a lower temperature, implies a lower sensible heat loss (Stack Loss);
- Condensing the vapor contained in the combustion products implies the recovery of the latent heat of vaporization, that is an important contribution in the case of fuels with an high hydrogen content. In the case of condensation, the value that represents the "energy source" for the system is the fuel's higher heating value , that is from the 11% (for natural gas) to the 6% (for gasoil) bigger than the lower heating value.

For the assessment of the Combustion Efficiency in the case of Condensation, has to be introduced a different procedure that takes in account the additional heat recovered by the combustion products vapor condensation. Has to be noticed that even in this case the denominator value for the Combustion Efficiency formula remains the lower heating value, in order to make possible efficiency comparisons between condensing and non condensing boilers. Thus, in the condensing cases it will not be wrong to obtain efficiency values greater than the unity, that, from the thermodynamic point of view, represents a non-sense.

From the Combustion Efficiency, written according the equation (2.42), we can write the Combustion Efficiency formula for the condensing case:

$$\eta_{comb} = 100\% - Q_{stack} + Q_{lat} \tag{2.47}$$

Where $Q_{lat}$is the value that increases the Combustion Efficiency due to the condensation of flue gases.

2.4.3 COMBUSTION EFFICIENCY ACCORDING THE EUROPEAN STANDARD(EN 15378)

This EN Standard prescribes the following measuring conditions [22]:

- Sample flue gas near boiler discharge, at least within 3 diameters.
- Sampling hole shall be tight.
- Flue gas shall be sampled in the middle of the stream. The hottest point or the point with the lowest measured
- Combustion air temperature shall be measured at burner intake.
- Simultaneous measurement of flue gas and intake air temperature is recommended.
- If there is a combustion air pre-heater, flue gas temperature and combustion air temperature shall be measured between boiler and heat recovery device.
- The flue gas properties shall be measured in the same point, preferably with a multifunction device (combined probe), using an instrument complying with EN 50379:

The combustion efficiency, η_{comb}, is given by:

$$\eta_{comb} = 100 - \alpha_{ch,on} \tag{2.48}$$

Where $\alpha_{ch,on}$ is the sensible heat of the flue gas losses, $\alpha_{ch,on}$, i.e. the sensible heat losses through the chimney with burner on, and is calculated with the following equation:

$$\alpha_{ch,on} = \left(\theta_{fg} - \theta_{air}\right) * \left(\frac{c_1}{21 - X_{O2,fg.dry}} + c_2\right) \tag{2.49}$$

where c1 and c2 are constants given in national annexes, depending on the fuel.

In the case of a heat generator that allows the recovery of the latent heat from the flue gases (condensing boilers), an additive correction term is applied to take into account the recovery of the latent heat of condensation:

$$\eta_{comb,corr} = \eta_{comb} + \alpha_{cond} \tag{2.50}$$

In this EN standard is given the procedure to calculate α_{cond} when is not available any National Standard.

The following properties of flue gas shall be measured:

- θ_{fg} flue gas temperature;
- $X_{O2,fg.dry}$ oxygen content of the dry flue gas;

The following properties of flue gas shall be measured or estimated:

- θ_{air} combustion air temperature (default value: external ambient temperature);
- x_{air} combustion air relative humidity (default value: 50%);
- x_{fg} flue gas relative humidity (default value:100%)

The recovered latent heat of condensation, α_{cond} , is defined as the mass of water vapor condensed multiplied by the specific heat of condensation of water.

The mass of water vapor condensed in the heat generator is the difference between the water vapor available (the sum of the water production due to the stoichiometric combustion and the water content in the combustion air) and the mass of water content in the flue gas exiting the heat generator.

To calculate the required quantities to assess the recovered latent heat of condensation, α_{cond}, the EN Standard proposes the following procedure.

The actual amount of dry flue gas $V_{fg,dry}$ is given by:

$$V_{fg,dry} = V_{fg,st,dry} * \frac{20{,}94}{20{,}94 - X_{O2,fg.dry}} \tag{2.51}$$

The actual amount of dry combustion air $V_{air,dry}$ is given by:

$$V_{air,dry} = \mathrm{V_{air,st,dry}} + \mathrm{V_{fg,dry}} - \mathrm{V_{fg,st,dry}} \tag{2.52}$$

Saturation humidity of air $\xi_{H2O,air,sat}$ and flue gas $\xi_{H2O,fg,sat}$ shall be calculated according to θ_{air} (combustion air temperature) and θ_{fg} (flue gas temperature), respectively, and be expressed as kg of humidity per Nm^3 of dry air or flue gas.

The EN Standard provides the saturation humidity values, as a function of temperature, to be used in the above calculations, and linear or polynomial interpolation are used for intermediate temperatures.

Is assumed that the flue gases and the combustion air have the same saturation humidity values.

θ_{air} *or* θ_{fg}	**°C**	*0*	*10*	*20*	*30*	*40*	*50*	*60*	*70*
$\xi_{H2O,air,sat}$ *or* $\xi_{H2O,fg,sat}$	**kg/Nm$^3_{dry}$**	*0,00493*	*0,00986*	*0,01912*	*0,03521*	*0,06331*	*0,1112*	*0,1975*	*0,3596*
Note*: saturation humidity is expressed as kg of water vapor per Nm^3 of dry gas (either air or flue gas)*									

Tab. 2.1 Saturation humidity values, prEN 15378

Total humidity of combustion air mH2O,air is given by:

$$m_{H2O,air} = \xi_{H2O,air,sat} * V_{air,dry} * \frac{x_{air}}{100} \qquad (2.53)$$

Total humidity of flue gas $m_{H2O,fg}$ is given by:

$$m_{H2O,fg} = \xi_{H2O,fg,sat} * V_{fg,dry} * \frac{x_{fg}}{100} \qquad (2.54)$$

The amount of condensed water $m_{H2O,cond}$ is given by:

$$m_{H2O,cond} = m_{H2O,st} + m_{H2O,air} - m_{H2O,fg} \qquad (2.55)$$

If $m_{H2O,cond}$ is negative, there is no condensation, and $m_{H2O,cond} = 0$ and $\alpha_{cond} = 0$.

The specific latent heat of condensation $h_{cond,fg}$ is given by:

$$h_{cond,fg} = 2\,500\,600 \frac{J}{kg} - \theta_{fg} * 2\,435 \frac{J}{kg°C} \qquad (2.56a)$$

Or

$$h_{cond,fg} = 694{,}61 \frac{Wh}{kg} - \theta_{fg} * 0{,}6764 \frac{Wh}{kg°C} \qquad (2.57b)$$

The condensation heat Q_{cond} is given by:

$$Q_{cond} = m_{H2O,cond} * h_{cond,fg} \qquad (2.58)$$

If the calculation is based on net calorific values, the recovered latent heat of condensation α_{cond} is given by:

$$\alpha_{cond} = 100 * \frac{Q_{cond}}{H_i} \qquad (2.59a)$$

If the calculation is based on gross calorific values, the recovered latent heat of condensation α_{cond} is given by:

$$\alpha_{cond} = 100 * \frac{Q_{cond}}{H_s} \qquad (2.59b)$$

2.4.4 COMBUSTION EFFICIENCY ACCORDING THE ITALIAN STANDARD (UNI 10389-1)

The UNI 10389-1 “Flue gases analysis and measurement on site of combustion efficiency” [23] provides the instructions to evaluate the Combustion Efficiency of a boiler and defines the procedures in order to assess the composition of the flue gases, in accordance with the European Standards.

The Combustion Efficiency, η_{comb} ,is defined as: the ratio between Conventional Heating Power, P_C, and the Burner Thermal Power, P_F .

$$\eta_{comb} = \frac{P_C}{P_F} \tag{2.60}$$

The Conventional Heating Power is defined as: the Burner Thermal Power lowered by the Stack Heat Loss, Q_s.

$$P_C = P_F - Q_S \tag{2.61}$$

The Burner Thermal Power is defined as: the product of the fuel’s Lower Heating Value (LHV) and the burned fuel flow rate.
This technical regulation states the procedure in order to measure:

1. The Burner Thermal Power;
2. The flue gases composition and temperature and the combustion air temperature;
3. Evaluation of the Combustion Efficiency;
4. Evaluation of the Smoke Index;

Burner Thermal Power assessment

Known the oil nozzle nominal flow rate and the atomization pressure, is possible to obtain the oil mass flow rate G [kg/h] using the tables or curves provided by the nozzle’s manufacturer.
To obtain the Burner Thermal Power, the value of G is multiplied by the conventional Lower Heating Value:

a- Gasoil: 11,86 kWh/kg;
b- Fuel Oil:22,47 kWh/kg;

Flue gases sampling and combustion air temperature measurement

Measurement conditions:
Each measurement has be made when the heat generator is in steady-state condition at the thermal power at which the measurement is referred.

Is assumed that the Steady-State condition is reached when the flue gases temperature is stabilized at a certain value ,i.e. it doesn't vary more than ±2°C.

Flue gases sampling and temperature measurement must be done at a proper sampling socket on the gas evacuation pipe.
If downstream the heat generator there is a device that allows the heat recovery from flue gases and heat recovered is used to pre heat the combustion air, the measurement of the flue gas temperature and the combustion air temperature must be made:

- Flue gas temperature: downstream the recovery system (after pre heating);
- Combustion air temperature: upstream the recovery system (before the pre heating);

Evaluation of the Combustion Efficiency in NON CONDENSING MODE

In the non condensing mode, i.e. when the average flue gases temperature is higher than the flue gases dew point temperature, t_{gd}, the Combustion Efficiency is obtained as follow:

a- Stack Heat Power Loss, Q_s, determination:

$$Q_s = \left(T_{flue} - T_{amb}\right) \cdot \left(\frac{A1}{21\% - O_2} + B\right) \qquad (2.62)$$

Where $A1$ and B are coefficients depending on the fuel used:

$A1$	*B*	*FUEL*
0,68	*0,50*	*Gasoil*
0,68	*0,52*	*Fuel Oil*

Tab. 2.2 coefficients $A1$ and B (UNI 10389-1)

b- The Combustion Efficiency is given by:

$$\eta_{comb} = 100 - Q_s \qquad (2.63)$$

The uncertainty on this value is assumed as $\pm 2{,}0$.

Evaluation of the Combustion Efficiency in CONDENSING MODE

In condensing mode, i.e. when the average flue gases temperature is lower than the flue gases dew point temperature,t_{gd}, the Combustion Efficiency is obtained as follow:

a- Stack Heat Power Loss, Q_s, determination:

$$Q_s = \left(T_{flue} - T_{amb}\right) \cdot \left(\frac{A1}{21\% - O_2} + B\right) \qquad (2.64)$$

b- Air index, n, determination:

$$n = \frac{21}{21 - O_2} \tag{2.65}$$

c- Determination of the dry combustion products volume at normal conditions (TPN), V_{fs}:

$$V_{fs} = n * VF \tag{2.67}$$

d- Determination of the actual combustion products volume, $V_{fs,act}$, at the flue gases temperature, t_f:

$$V_{fs,act} = V_{fs} * \frac{273 + t_f}{273} \tag{2.68}$$

e- Calculation of the actual combustion products mass corrected for the flue gases temperature, $M_{fs,act}$:

$$M_{fs,act} = V_{fs,act} * MV_f \tag{2.69}$$

f- Determination of the dry combustion air volume at normal conditions (TPN), V_{as}:

$$V_{as} = n * VA \tag{2.70}$$

g- Determination of the actual combustion air volume, $V_{as,act}$, at the combustion air temperature, t_a:

$$V_{as,act} = V_{as} * \frac{273 + t_a}{273} \tag{2.71}$$

h- Calculation of the actual combustion air mass corrected for the air temperature, $M_{as,act}$:

$$M_{as,act} = V_{as,act} * MV_a \tag{2.72}$$

i- Water mass content in combustion air, AR, with the assumption that the combustion air has relative humidity of 50%:

$$AR = M_{as,act} * X_a * 0{,}5 \tag{2.73}$$

j- Water mass content in combustion products, AF:

$$AF = M_{fs,act} * X_f \tag{2.74}$$

k- Determination of the condensed water amount, CD:

$$CD = AC + AR - AF \tag{2.75}$$

l- Determination of the energy recovered from the water condensation, ER:

$$ER = CD * H_{cond} \tag{2.76}$$

m- Calculation of the Combustion Efficiency increment, ET, due to water condensation:

$$ET = \frac{ER}{LHV} * 100 \tag{2.77}$$

n- The Combustion Efficiency is given by:

$$\eta_{comb} = 100 - Q_s + ET \tag{2.78}$$

The uncertainty on this value is assumed as $\pm 2{,}0$.

All the parameters in the procedure above described are given in the same technical law and are reported in Tab. 2.3,Tab. 2.4, Tab. 2.5.

Air index n	*1*	*1,1*	*1,2*	*1,3*	*1,5*	*2*
Dew point temperature [°C]	*52*	*50,4*	*48,7*	*45,9*	*44,6*	*39,3*

Tab.2.3 Dew point temperature v.s air index (UNI 10389-1)

Parameter	***Gasoil***
VF [$m^3_{flue\ gas}$]	*10,41*
VA [m^3_{air}]	*11,09*
AC [kg_{water}]	*1,17*
LHV [MJ]	*42,70*

Tab. 2.4 parameters (UNI 10389-1)

Temperature [°C]	***X_a and $X_f\left[\frac{kg_{water}}{kg_{dryair}}\right]$***	***MV_a and $MV_f\left[\frac{kg_{dryair}}{m^3}\right]$***	***$H_{cond}\left[\frac{MJ}{kg_{cond}}\right]$***
0	*0,00379*	*1,29299*	*2,502*
5	*0,00542*	*1,26968*	*2,490*
10	*0,00766*	*1,24719*	*2,478*
15	*0,01069*	*1,22549*	*2,466*
20	*0,01476*	*1,20453*	*2,454*
25	*0,02017*	*1,18427*	*2,443*
30	*0,02733*	*1,16469*	*2,431*
35	*0,03676*	*1,14574*	*2,419*
40	*0,04914*	*1,12740*	*2,407*
45	*0,026541*	*1,10963*	*2,395*
50	*0,08686*	*1,09242*	*2,383*
55	*0,11532*	*1,07573*	*2,371*
60	*0,15354*	*1,05955*	*2,359*

Tab. 2.5 parameters (UNI 10389-1)

If the measurements are done with a multifunction analyzer, it has to satisfy the following requests in accuracy:

Measured parameter	***Measuring range***	***Resolution***	***Accuracy***	***Minimum value***
CO	*0 ppm - 2000ppm*	*1 ppm*	*±20ppm or 5% measured value*	*20 ppm*
O_2	*0% - 21%*	*0,1%*	*±0,3%*	*0,3%*
CO_2	*0% - 20%*	*0,1%*	*±0,2%*	*0,2%*
Flue gas temperature	*0°C – 400°C*	*1°C*	*±2°C or 1,5% measured value*	
Combustion air temperature	*-20°C – 100°C*	*1°C*	*±1°C*	

Tab. 2.6 multifunction analyzer requirements (UNI 10389-1)

CHAPTER 3

OIL FIRED CONDENSING BOILER

3.1 BOILER DESCRIPTION

A heat generator is basically composed by a source of heat and a heat exchanger, that transfers the heat form the source to the heat carrier fluid. In combustion heat generators the heat source are the combustion products of a combustion. The combustion process is obtained by using a burner.

The heat exchanger of a heat generator can be usually divided into two main areas:

- the fire box, or combustion chamber, in which the combustion takes place and the metallic walls of this section are “seeing” the flame;
- the smoke tubes, or smoke passages, in which the combustion products are further cooled down.

3.1.1 THE BURNER

The term “burners” describes a series of equipment for burning various types of fuel under suitable conditions for perfect combustion.

The burner operates by sucking in the fuel and the combustion air, mixes them thoroughly together and safely ignites them inside the heat generator combustion chamber.

A burner is composed by:

- *The combustion head* which mixes the fuel and the combustion air;
- *Fuel supply*, comprising components used for regulating the fuel flow and guaranteeing the safety of the combustion system;
- *The combustion air supply*, comprising of the fan and any pipes for taking the air to the combustion head;
- *The electrical and control components* required for firing the flame, the electricity supply to the motors and thermal output regulation developed by the burner.

We can classify burners according to the strategy in delivering the thermal output:

- single-stage burners: the thermal output is invariable and the burner can be switched on or off (ON-OFF);
- multi-stage burners: usually two-stage or three-stage, are set for running at one or more reduced output speeds or at maximum output (OFF-LOW-HIGH or OFF-LOW-MID-HIGH); switchover from one stage to another can be automatic or manual;
- modulating burners: the delivered output is automatically varied continuously between a minimum and maximum value, for optimum delivery of the thermal output in relation to system requirements.

In fig. 3.1 are shown the above defined burner types.

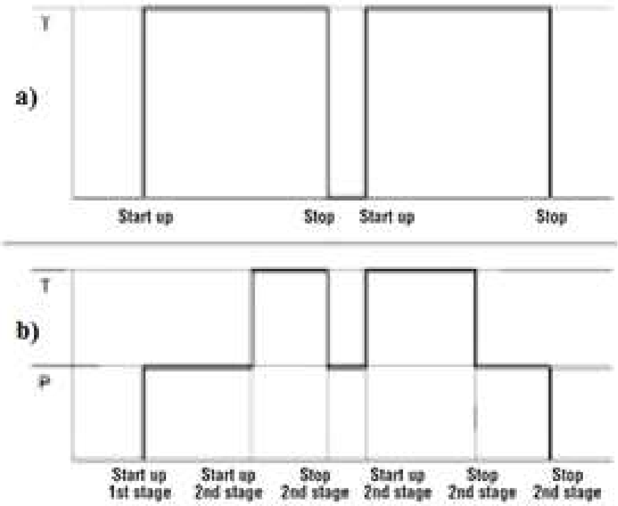


Fig.3.1 Single-stage (a) and two-stage (b) burner working strategies

The firing range of burner is a representation in the Cartesian plan of an area, showing the pressure in the combustion chamber (in mbars or in Pa) on the Y axis and the thermal output (in kW or in kg/h of fuel burnt) on the X axis; this area indicates working conditions under which the burner guarantees combustion corresponding to the thermotechnical requirements.

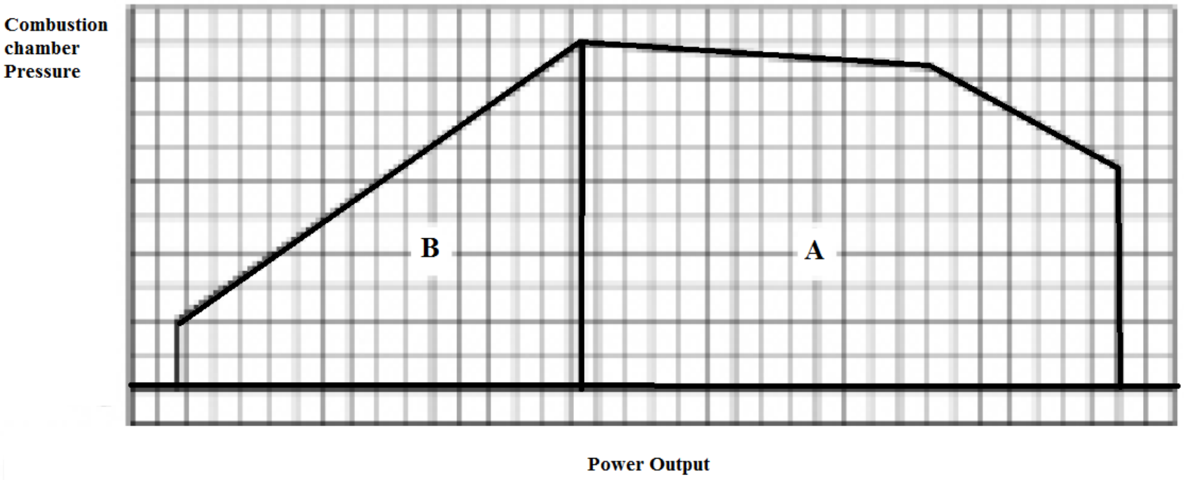


Fig. 3.2 firing range for a two-stage burner

The firing range is obtained in special test boilers according to methods established by European legislation, EN 267 standard for liquid fuel burners.
For two-stage burners (Fig. 3.2) the firing range is ideally divided into two areas, left (zone A) and right (zone B) of the vertical line traced for the point corresponding to the maximum head available. The firing point corresponding to the maximum output and, consequently, to operate in the 2nd stage, must be chosen within zone B. Zone B provides the maximum output of the burner in relation to the combustion chamber pressure.

The 1st stage output should be chosen within the minimum/maximum declared formula and normally falls within zone A. The absolute lower limit corresponds to the minimum value of zone A.

3.1.1.1 THE COMBUSTION HEAD

The combustion head is the part of the burner that mixes the combustion air with the fuel and stabilizes the flame that is generated.

The combustion head essentially comprises the following components:

The fuel metering device: nozzles for liquid fuels and orifices and distributors for gaseous fuels; oil nozzles are characterized by three parameters: output, spray angle and type of spray distribution;

The turbulator diffuser disk, which mixes the fuel and the combustion air, and stabilizes the flame to avoid it blowing back into the burner;

The flame ignition system, uses electric arcs produced by high-voltage powered electrodes directly igniting the flame or coupled with a pilot burner;

A flame sensor for motoring the flame;

The flame tube comprising made of profiled metal cylinder which defines the output speed range.

Flame tube and diffuser disk determine the geometry of the flame developed by the burner. Especially the diffuser disk determines the rotational features of the fuel and combustion air mixture flow and, consequently, the flame dimensions.

As a rule, an increase in the rotational features causes an increase in the flame diameter and a decrease in the flame length.[14]

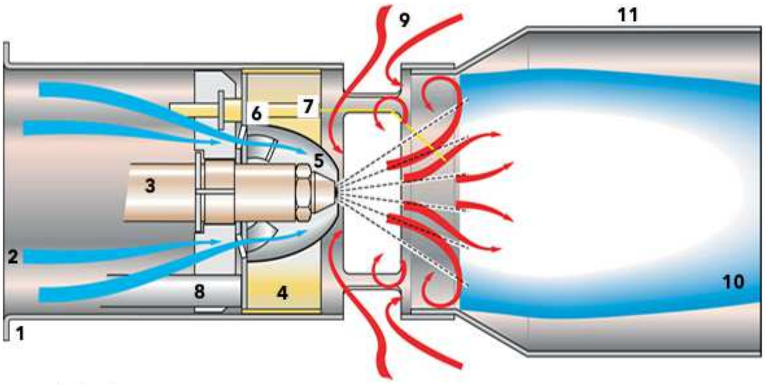
3.1.1.1.1 BLUE FLAME

The convenctional oil burners, before the introduction of “blue flame” technology, used to produce yellow or yellow-orange flames which are noisy, characterised by incomplete combustion and often deposit of soot int the flue or chimney, and discharge of particulate matter into athmosphere.
Such yellow type flame burners have localized high temperature regions that tend to produce large amounts od oxides of nitrogen and soot (which have high temperature mechanisms of formation) [24].
The “blue flame” burner is able to burn liquid fuels preventig the above seen problems.
To obtain a blue flame the recirculation of a large amount of combustion products, upstream the flame zone, is required.
The mechanism to obtain a blue flame is as follows:

- fresch mixture of atomized liquid fuel, exiting the fuel nozzle, and combustion air, exiting the air nozzle, is injected through a converging throat in the preparation zone.
- This spray of fresh mixure is travelling at high velocity through this zone, thus creating a low pressure zone.
- In correspondence of the preparation zone there are some holes or slots, which connect the preparation zone with combusion chamber, filled of combustion products. So that the low

pressure zone will drown back some of the gases contained in the combustion chamber and they will mixed with the fresh mixture.

Recirculated gases include combustion products and vaporized unburned fuel, thus being a fuel vapor rich mixture of combustion products having no significant oxygen content.



1.Burner tube
2. Combustion air
3. Nozzle holder
4. Dosing ring
5. Air nozzle
6. Helix
7. Ignition electrode
8. Flame control
9. Flue gas recirculation
10. Blue flame
11. Fire tube

Fig.3.3 Flue gas recirculation in a Blue Flame burner

By the injection of hot gases near the fuel oil outlet from the atomizing nozzle, are obtained the following benefits:

- Hot gases enhances the liquid fuel droplet vaporization, in a way that the fuel reaches the flame zone in gas phase. This considerably reduces the soot formation, and the unburned hydrocarbon amount in the flue gas. This has as a result a lower fuel consumption (and higher combustion efficiency), lower unburned emissions and a lower fouling of the combustion chamber.
  A flame that is soot-free has the characteristic color blue. In luminous flames, characterized by a yellow color, soot takes an important role in radiative heat transfer.
- The flame temperature is lowered since the flame is cooled by the hot gases, this is a key factor to have low nitrogen oxides emissions. In fact, in a combustion process nitrogen oxides start to be generated in a significant amount , when the flame temperature is higher than 1500 °C and the production reaches its maximum around 2000°C [25].

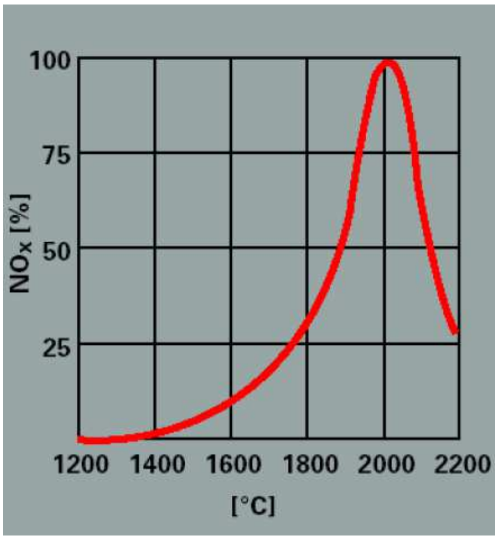


Fig.3.4 NO_x production vs. reaction temperature

3.1.1.2 FUEL SUPPLY

Hydraulic circuits on board liquid fuel burners or mixed burners, have different features and complexity, depending on the type of fuel, supplied output, load regulation logic (single-stage, multi-stage or modulating) and special standards in force.
Generally the burners are fitted with a geared pump and a single or double fuel shut off solenoid system. [14]

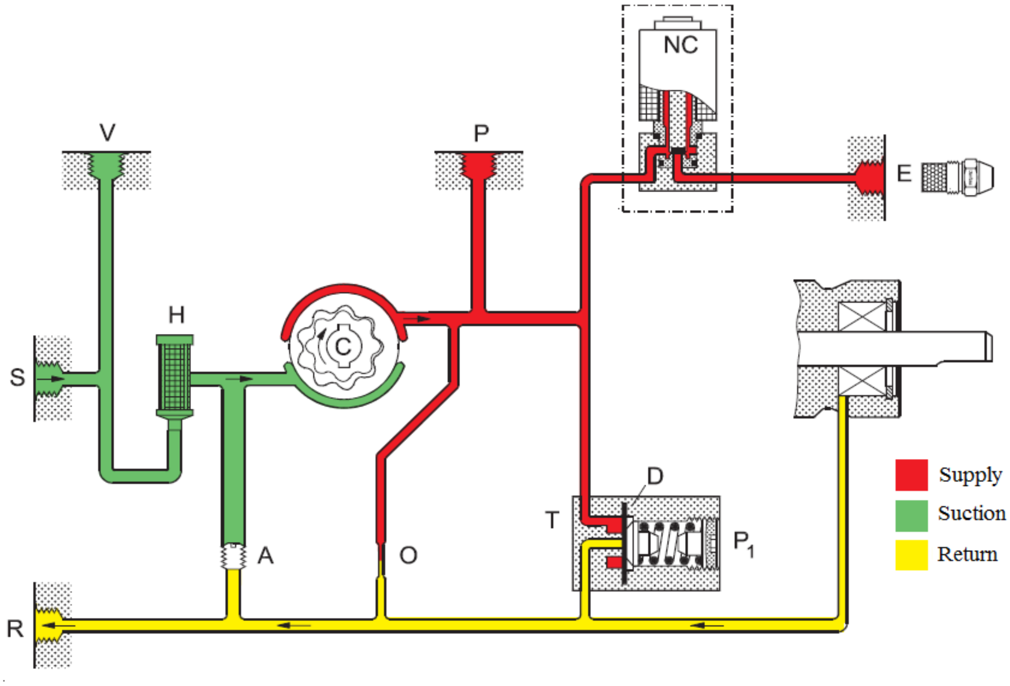


Fig.3.5 sketch of an oil pump

Modulating burners also have shut off valves on the return circuit and a pressure regulator for varying output.
The fan motor drives the pump, or it is run independently, it can have special features for using kerosene.
The oil feed systems covered in this section are a "bi-pipeline" type and namely those comprising a delivery pipeline from the tank to the burners and a return pipeline from the burner to the tank[14].
In Fig.3.5 is represented an oil pump sketch, in which we can identify three zones:

- suction (in green): the inlet (S) is connected with the oil tank, the negative pressure generated by the geared wheel, sucks the oil, which passes through a filter (H) and enters in the compression chamber;
- supply (in red) : the pressurized oil, if the electro-valve (NC) is open, is sent to the nozzle trough the outlet (E). The oil pressure is regulated by the pressure controller valve (T), set at the desired pressure (P1).
- return (in yellow): the excess oil from the pressure controller valve is sent back to the oil tank. The oil leakages are collected and sent back to the tank.[26]

The important task of the oil pump is to provide a sufficient pressure value to the oil delivered to the oil nozzle, in order to accomplish the fuel atomization.

3.1.1.2.1 THE NOZZLE

The vaporizing process is accelerated if fuel is bulk liquid oil is spilt in small droplets.
A very small droplet will be vaporize in a short period of time when exposed to high temperatures.
Separating the oil into very small droplets the surface area is increased, exposing more oil surface to contact with air. The process above described is called Atomization.
The simplest method to achieve the Atomization with light oils is by the use of nozzles, which separate the fuel into small droplets by their particular design.
The most common type of nozzle for Atomization is “high pressure atomizing type”.
In addition to breaking up the fuel into small droplets, the nozzle is expected to deliver these droplets in a specific pattern. It must be designed to deliver a specified spray angle within specified limits. It must also be designed to distribute these droplets as desired across a cross section of the spray.
The second function of a nozzle is metering the fuel. At the predetermined pressure the nozzle will deliver a definite amount of fuel. This characteristic allows the heat generation control. For example, in double stage burners, which can deliver two fixed values of thermal power, the amount of fuel burner is regulated by changing the pressure at which the fuel oil is sent to the nozzle.
Usually the flow rate is given in terms of Gallons per Hour, GPH, that in terms of Liters per Hour corresponds:

$$1\mathrm{GPH} = 3{,}7854\ \frac{\mathrm{l}}{\mathrm{h}} \tag{3.1}$$

3.1.1.2.2 FUNCTIONING PRINCIPLE

Breaking the bulk liquid into small droplets requires the application of energy. With nozzles, this energy is supplied as pressure energy, generated from a motor driven pump. In order to create droplets, this pressure energy has to be converted into velocity energy. This is done by supplying the fuel under pressure and forcing it through a set of holes or slots. The oil emerges from these slots at a very high velocity.

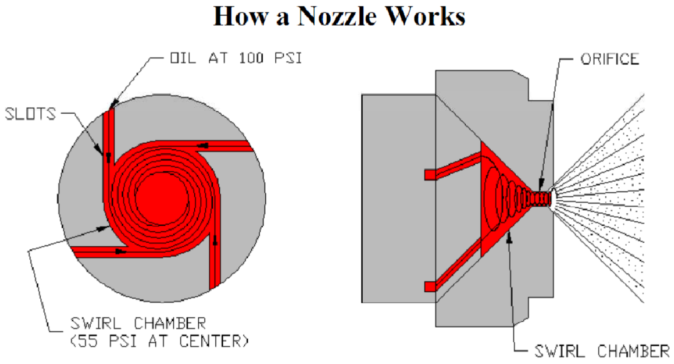


Fig.3.6 Nozzle sketch

These slots are cut tangentially into a swirl chamber. The velocity of rotation increases as the liquid approaches the center of the swirl chamber. There is placed the discharge orifice, from which the liquid exits with a so high rotational velocity that in the middle of the rotating fluid flow is created an air core. The oil then will extend into the orifice in the form of a rapidly rotating tube of oil, leaving an air core in the center. As this tube of oil is rotating it pushes outward against the walls of the orifice because of the centrifugal force developed. Not all pressure of the liquid is converted into velocity energy, but some of it remains as pressure energy and the pressure, which tends to push the liquid forward through the orifice, will be approximately one half of the applied pressure.

This pressure forces the oil to emerge from the orifice in the form of a spinning tube which, because of centrifugal force, immediately expands into a cone shaped sheet as it leaves the orifice.
This sheet of liquid emerging from the orifice stretches to the point where it ruptures and throws off droplets of liquid.
A spray is so created. The spray angle is governed by the design of the swirl chamber and the orifice.

The conditions that affect the nozzle performance are: pressure, specific gravity, viscosity, temperature of the oil supplied at the nozzle.

Pressure

1. As might be expected, an increase in pressure increases the discharge rate of the nozzle, all other factors remaining
equal. The relationship between the pressure and discharge from a nozzle is a fundamental one. The theoretical discharge from any orifice or nozzle is given by the equation:

$$\mathrm{F_{fuel}} = \mathrm{C} \cdot \mathrm{A} \cdot \sqrt{\mathrm{2gH}} \qquad (3.2)$$

Where:

C: is a dimensionless coefficient for the particular nozzle in question;
A:is the area of the nozzle orifice;
H:is the pressure head applied to the nozzle;
g:is acceleration of gravity.

This fundamental equation is modified by various factors encountered in nozzle design, but from it we arrive at a simple formula, which is of value to anyone using nozzles.

$$\mathrm{F_2} = \mathrm{F_1} \cdot \sqrt{\left(\frac{\mathrm{p_2}}{\mathrm{p_1}}\right)} \qquad (3.3)$$

In this equation:

p_1:   *is pressure at which the nozzle is calibrated;*
p_2:   *is any pressure at which it is desired to operate a nozzle;*
F1: is the calibrated flow rate at pressure p_1.
F2: is the flow rate at the desired pressure;

In Fig. 5.7 is shown what was said above.

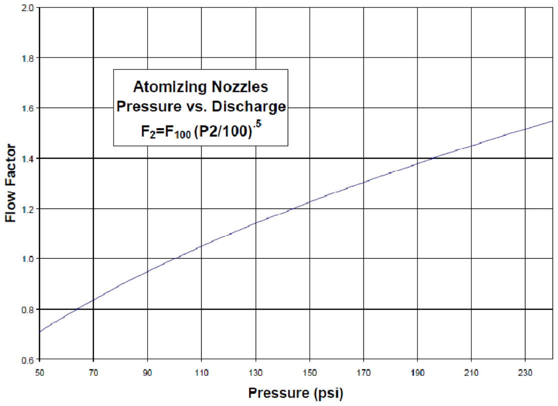


Fig.3.7 Flow Factor vs. Atomizing Pressure for a Nozzle

This relationship is approximately true for nozzles in the small sizes used for domestic oil burners but not exactly so.

The value of the exponent varies somewhat with nozzle design and flow rate and it has been found to be as low as 0.470.

2. After a true divergent spray is established, any increase in the pressure does not change the basic spray angle.

When the spray angle is measured a distance of about 10 cm in front of the nozzle operating at high pressure, there is a change in the direction the droplets are traveling, but that is not the basic spray angle of the nozzle. The reason the spray angle appears to change at a distance in front of the nozzle is also quite fundamental. With an increase in pressure, the velocity of the droplets is increased within the body of the spray. An increase in velocity in the spray causes a reduction in pressure in that area. Thus the spray might be said to aspirate air, which will tend to push into the spray. As that air moves toward the center of the spray, it carries with it the droplets which have lost their velocity to the point where the air can move them.

3. An increase in the applied pressure will produce smaller droplets in the spray. Some recent research shows that the median droplet size varies inversely as the .3 power of the pressure change. That means that if the pressure is increased from 100 psi to 300 psi, the median droplet diameter is reduced approximately 28%. An increase in pressure from 100 to 150 psi reduces the median droplet diameter by approximately 11%.

Specific Gravity

Specific gravity is normally used in flow calculations but in the petroleum industry the more common term is API gravity.

The relationship between specific gravity and API° gravity is given by the equation:

$$\mathrm{Sp.Gr.} = \frac{141{,}5}{\mathrm{API}° + 131{,}5} \qquad (3.4)$$

The effect of specific gravity on discharge rate (volumetric) is as follows:

$$F_2 = F_1 \cdot \sqrt{\left(\frac{d_2}{d_1}\right)} \qquad (3.5)$$

Where:

d_1: *is the Specific Gravity for flow* F_1;
d_2: *is the Specific Gravity for flow* F_2.

Viscosity

In small nozzles and within the limits of No. 2 fuel oil, the effects of changes in gravity are less important than the effects of changes in viscosity. The viscosity of fuel is a measure of its resistance to flow.

To measure this parameter can be used the Saybolt viscosimeter. This apparatus lets flow the oil through an orifice and counts the time required to fill a prescribed sample. The number of seconds that are taken to fill a prescribed sample is the kinematic viscosity of the fluid in SUS at that particular temperature.

The conversion relation from SUS unit to cSt unit (centistokes) is the following:

For 32<SUS<100:

$$cSt = 0{,}226 \cdot SUS - \frac{195}{SUS} \tag{3.6a}$$

For SUS>100:

$$cSt = 0{,}220 \cdot SUS - \frac{135}{SUS} \tag{3.6b}$$

A high viscosity indicates greater resistance to flow.

Fuels with higher viscosities need greater energy to move them whether the flow is through pipes or through nozzles.. For purposes of calculations it is necessary to determine the absolute viscosity. Absolute viscosity is determined by multiplying kinematic viscosity in terms of centistokes by the specific gravity of the liquid at the same temperature. It is expressed in centipoises. Changes in viscosity have several effects on nozzle performance:

1. With a pressure-atomizing nozzle of the size and type commonly used in domestic oil burners, the flow rate is affected as shown in Figure www, assuming typical values of gravity.

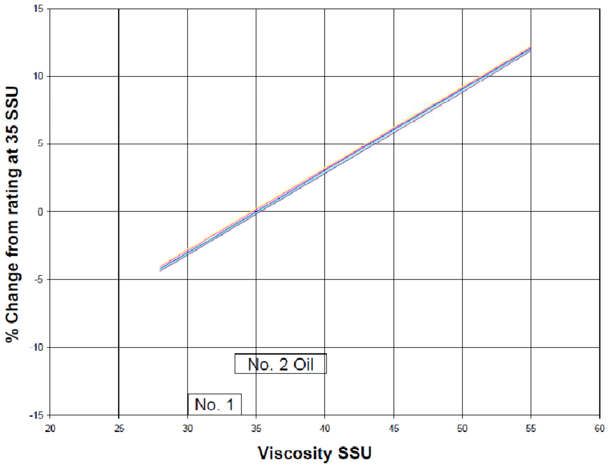


Fig.3.8 Variation on the flow rate vs. Fuel Viscosity

When viscosity is increased from 35 SSU to 45 SSU the discharge rate from the nozzle is increased. The effect of viscosity on flow rate varies with different designs of nozzles and at different supply pressures:

- Larger nozzles show less effect of viscosity on flow than small nozzles of similar design.
- The effect of viscosity is less at higher pressures.

The correlation between viscosity and flow rate through the nozzle has to be determined by experimental tests.

2. With higher viscosity fuels the spray angle would not be as stable as with low viscosity fuels. The angle of the film at the orifice is not changed appreciably, but the effective spray angle becomes narrower. The film tends to collapse, projecting the main body of droplets more toward the center of the spray. If the viscosity is high enough, the effective spray angle may collapse to the extent that a long, very narrow flame results.

3 With higher viscosity fuels the droplet size is increased. The droplet size may be increased to the point that it is impossible to maintain a steady flame front and the oil will burn off the back wall of the combustion chamber. Since an increase in viscosity requires an increase in the amount of energy provided to atomize the oil, the effects of viscosity on spray angle and droplet size can be minimized by increasing the supply pressure.

Temperature

The effect of temperature on the atomization process is just related to the viscosity changes. Lowering temperature means increasing viscosity and vice versa, thus we can repeat the same conclusion made for the variation of the viscosity value.

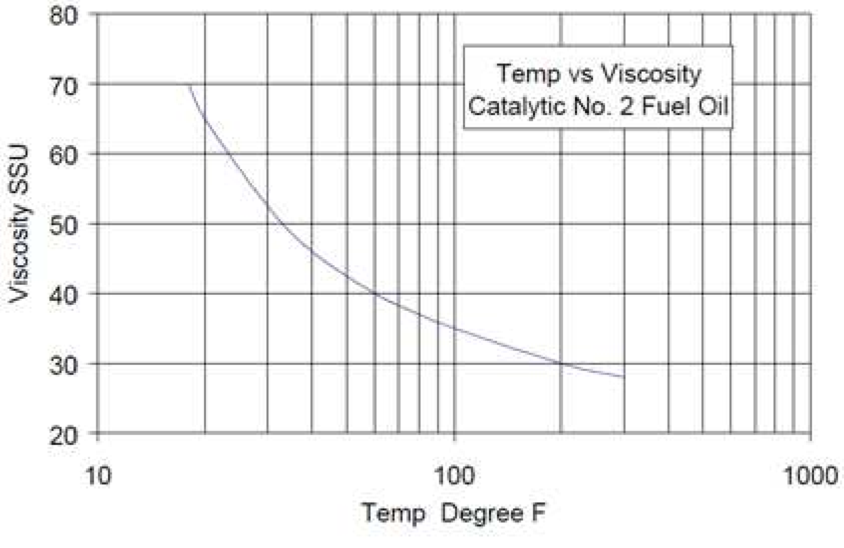


Fig. 3.9 Temperature effect on Fuel Viscosity

3.1.1.2.3 DROPLET SIZE AND SPRY PATTERN

The droplet size is important in order to improve the combustion efficiency. The droplet size is dependent upon the following factors:

1. The droplet size is usually larger in the higher discharge rates, assuming the same pressure. That means that a 10.00 GPH nozzle will have larger droplets in its spray than a 5.00 GPH nozzle with the same spray angle at the same pressure.

2. The droplet size is smaller in the wider spray angles.

3. High viscosity fuel produces larger droplets in a spray than low viscosity fuel at the same pressure.

4. A nozzle with a rough orifice finish produces larger droplet size than a nozzle with a good finish.

5. Increasing the fuel pressure on the nozzle reduces the droplet size.

6. Nozzle design is a very important determinant of droplet size in the spray. The smallest possible droplet size is not necessarily the most desirable. Good droplet size distribution to produce efficient, quiet fires is determined by nozzle design.

Nozzles for oil burner use are provided in two different general types of spray patterns, hollow cone and solid cone.
Generally speaking, it is well to use a spray type which matches the air pattern for the particular burner in question, and that can fit the geometrical characteristics of the combustion chamber.

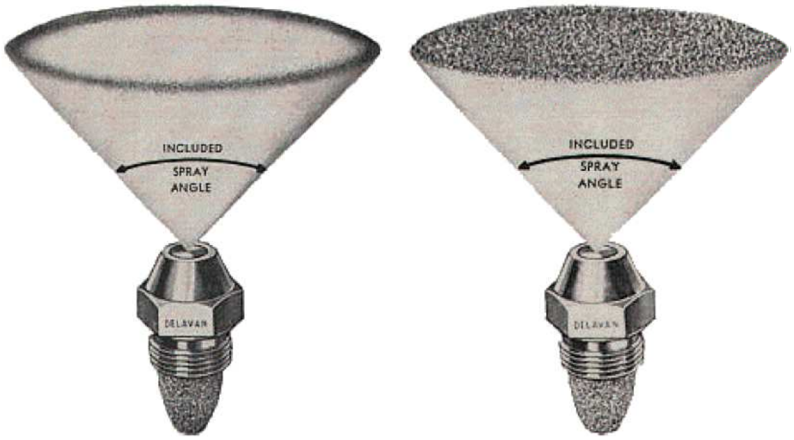
- The hollow cone is a spray in which the concentration of droplets is at the outer edge of the spray with little or no fuel in the center of the spray.
- The solid cone spray is defined as one in which the distribution of droplets is fairly uniform across the cross section of the spray.

In actual applications the use of hollow cone and solid cone nozzles seems to resolve itself into a few simple rules, to which, of course, there are some exceptions:

1. In low flow rates (up to approximately 2.00 GPH) it is common to use an hollow cone nozzle for all applications. This is because of the fact that a solid cone nozzle naturally gives a very long flame and could not be contained in a round or square combustion chamber.
   If a spray is required with more fuel spray near the center, it appears advisable to select a nozzle with a narrower spray angle rather than go to the solid cone type.
   The use of hollow cone nozzles in these small sizes also insures quietest possible operation. Hollow cone nozzles are more stable in their spray angle and droplet size distribution under adverse conditions than solid cone nozzles of the same flow rate.

2. in medium flow rates (from 2.00 GPH to 3.00 GPH) hollow or solid cone sprays may be selected to fit the air pattern in which they are to operate.

3. in high flow rates (above 3.00 GPH) it solid cone sprays are used. The main reason for this is that smoother ignition is obtained with solid cone sprays in the majority of burners, since in this range of flow rates the burner doesn’t face stability problems that can have the smaller sized burners[27].



Type “A”
Hollow Cone

Type “B”
Solid Cone

Fig.3.10 Spay Patterns representation

3.1.1.3 THE AIR FAN

Fans are machines capable of supplying energy to a fluid, by increasing pressure or speed, using a rotating element.

The fans installed on burners are generally centrifugal.
Centrifugal fans are made up of a box that contains a keyed fan wheel on a shaft supported by bearings.
The shaft can be connected directly to the electric motor using joints or, indirectly, using belts and pulleys.
The fan wheel positioned inside the box may have differing blade orientations/profiles and specifically:

- fan wheel with wing-shaped blades;
- fan wheel with reverse curved blades,
- fan wheel with radial blades;
- fan wheel with forward curved blades;

The working characteristics of a fan, similar to those for pumps, are described by the characteristic curve. The characteristic curve of a fan is represented in a Cartesian plan where the Y axis shows the pressure and the X axis shows the volumetric delivery (see Fig. 3.11)

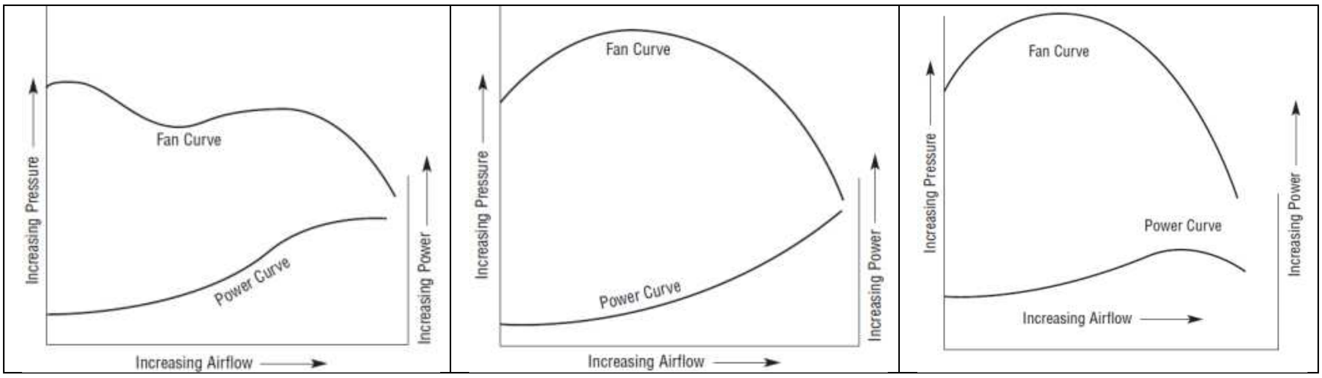


Fig. 3.11 working characteristics of air fans

The number of characteristic curves for each fan depends on the number of rotation speeds.
The working point of a fan is determined by the circuit in which it is included.
The circuit also has a characteristic curve, and the intersection of the fan characteristic curve and circuit characteristic curve, represents the working point of the system.

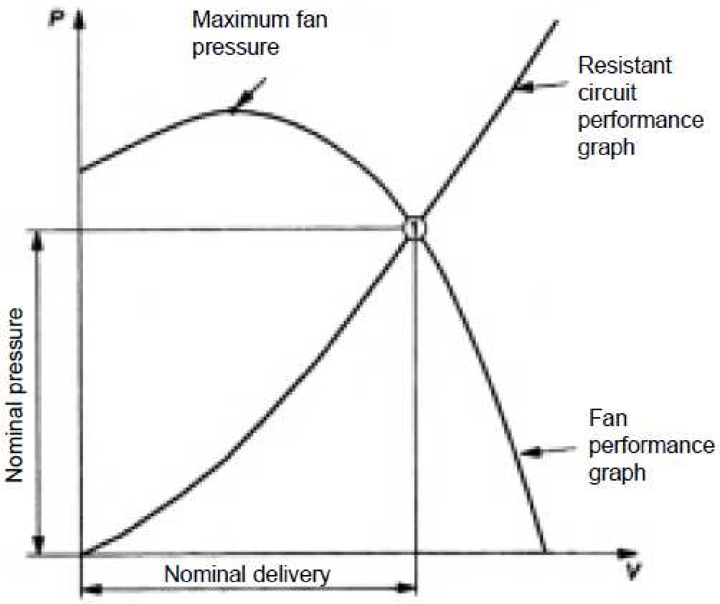


Fig. 3.12 working point of the air fan

The system characteristic curve varies in relation to the setting of the combustion head and opening degree of the air damper. Therefore, in order to determine the air flow conditions (rate rate and pressure) delivered by the fan, is requested a sufficiently precise evaluation of the pressure drop induced by the circuit (air intake pipes, burner head feeding pipes and other accessories).
The circuit pressure drops have a parabolic flow with respect to fluid speed and, consequently, delivery.

Pressure drops in an areaulic system are determined by two components:

- concentrated pressure drops;
- distributed pressure drops.

Among the concentrated pressure drops, must be taken into account:

- those introduced by the combustion head, where the air transits using a complex geometric route; furthermore, an air damper is fitted inside the burner for calibrating the delivery of combustion supporter air.
- those introduced by the presence of dampers, grids and any heat exchangers, must be calculated for the effective value of the drop introduced, which must be provided by the manufacturer of the mentioned devices.
- those introduced by the presence of circuit peculiarities, such as curves, direction and section variations, can be calculated using the following equation:

$$\Delta p_w = \xi * \rho * \frac{v^2}{2} \tag{3.7}$$

Where:

Δp_w: *is the pressure drop [Pa];*
ξ: *is the non-dimensional drop factor;*
ρ: *is the air density [kg/m^3];*
v: *is the average speed in the pipeline [m/s].*

Distributed pressure drops can be estimated by using the Darcy-Weisbach formula:

$$\Delta p_f = f * \frac{L}{D} * \frac{v^2}{2*g} \tag{3.8}$$

Where:

Δp_f: is the pressure drop due to friction [m];
f: is the friction factor;
L: is the pipeline length [m];
D: is the pipeline diameter [m];
v: is the air speed inside the pipeline [m/s];
g: is the gravity acceleration 9.81 [m/s^2].

The delivery of combustion air is proportionate to the delivery of fuel burnt, which in turn is proportionate to the required output.
For multistage and modulating burners, air provided by the fan must be changed in order to vary the delivery.
In Forced draught burners, delivery can be varied in two principal manners:

- Varying the fan firing point;
- Varying the number of fan revolutions;

In the first regulation method, the fan firing point is moved, along the characteristic curve, by varying the pressure drop of the areaulic circuit by introducing a servo-controlled damper (see Fig.3.13).

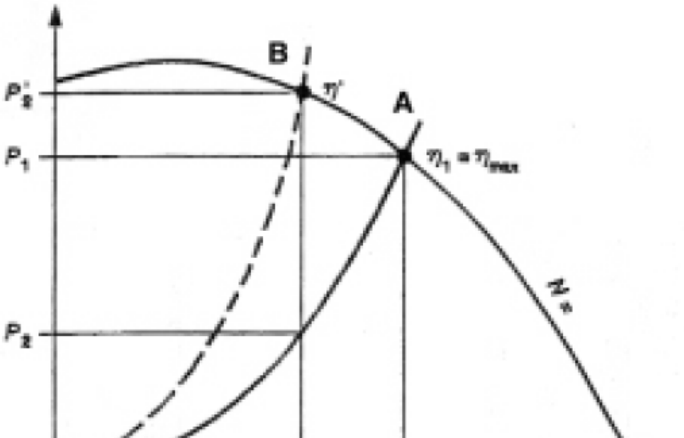


Fig.3.13 air fan regulation by introducing a pressure drop in the system

Depending on the opening degree of the damper, the various system curves are obtained.
In the case shown in Fig.5.13, the regulation damper closing determines the variation of then characteristic system curve from curve 1 to curve 2; consequently, the fan firing point moves from A to B, with the consequent increase of the fan head from P1 to P2’ and the decrease of the delivery from Q1 to Q2.

The opening degree of the damper introduces the various characteristic system curves thus determining differing delivery values.

This system is rather effective above all in centrifugal fans with forward curved blades where a delivery drop corresponds to a drop in absorbed output. In centrifugal fans with reverse curved blades, the output curve has a virtually flat trend and therefore it is not possible to obtain optimum operating performances.

The variation of the number of fan revolutions is obtained by using an inverter.

This device varies the frequency of the power supply voltage to the electric motor connected to the fan wheel. The modification in the rotation speed of the motor, due to a variation in the electric power frequency is represented by the following equation:

$$n = 120 * \frac{f}{p} \tag{3.9}$$

Where:

n: *is the motor rotation speed [rpm];*
f: *is the power supply voltage frequency [Hz];*
p: *is the number of poles of the motor.*

By regulating the number of revolutions, maximum performance operating can be obtained under the various working conditions, as the characteristic curve is translated until it coincides with the nominal firing point, as is shown in Fig. 3.14.

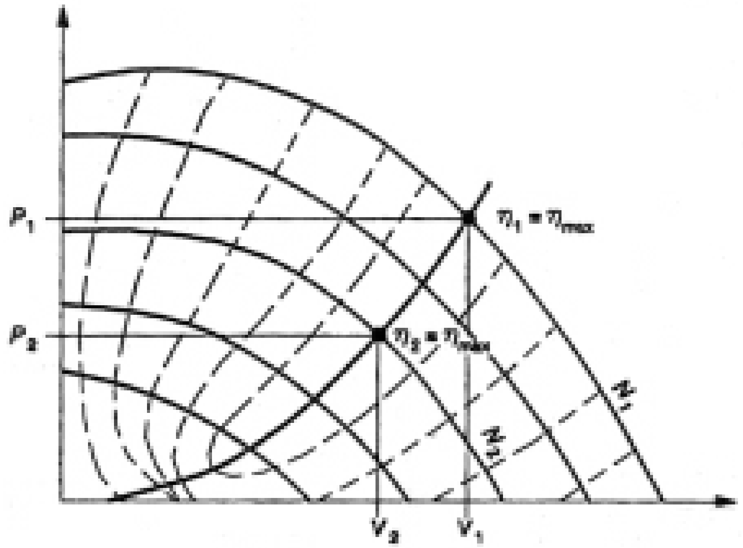


Fig. 3.14 air fan regulation by varying the fan speed

3.1.1.4 BURNER CONTROL AND AUXILIARY ENERGY

Boilers use electrical components for their operations. In the case of an oil-fired boiler we can recognize: control unit, oil pump, oil pre-heater, air fan, ignition system. These components introduces an energy consumption (electrical). A part of this electrical energy might be partially recovered as heat released to the involved fluids (i.e. the air fan heats the combustion air and the oil pump heats the oil before the combustion process). In Tab.3.1 is provided an overview of the typical power consumption for the various components, given from the "Boiler Savelec study".

Component	**Typical instantaneous power [W]**	**Consumption during system *off* mode**	**Consumption during system *on* burner *off* mode**	**Consumption during system *on* burner *on* mode**
Pump	55 – 80	Depends on type of T control system	Yes	Yes
Fan	30 – 50	No	No	Yes
Control unit	2 - 6	Yes	Yes	Yes
Gas valve	6 - 10	No	No	Yes
Stand-by consumption	5 - 15	Yes	Yes	Yes
Oil preheat	40 - 150	No	No	Yes, during 50s. for cold start only
Oil pump / atomization	75 - 200	No	No	Yes

Tab.3.1 Auxiliary energy consumption

The auxiliary electricity consumption for gas boilers over the year (space heating only) may vary from approximately 150 kWh_{el} to 450 kWh_{el} according to the control strategy of the circulator. For oil boilers the yearly electricity consumption is estimated to be 2 times higher [28] (considering also the consumption of the circulating pump.

Combustion regulation and control requires an electricity power supply to perform the following functions:

- Handling of the firing and flame safety sequence;
- Regulating the thermal load.

Observance of the safety instructions requires the use of special devices for supervising the burners. These devices must be capable to satisfy the following principles:

1. The flame must be present within a limited period of time from when the fuel is made available to the nozzle and subsequently it must burn uninterruptedly;

2. Depending on the type of burner, the maximum margin of time must be indicated during which the fuel can be discharged without the flame forming. This period of time, which is sufficiently short, is called safety time;

3. In absence of flame detection within safety time, a system lock-out takes place and fuel flow is stopped;

4. For oil burners, if during working flame goes out due to a temporary problem, it may be re established by a new firing;

5. Failure of burner devices that compromise safety, control and formation of the flame, must automatically interrupt the burner operations. This lock out, called a safety lock out, is indicated by a warning light and can only be released manually.

The equipment required to perform these functions is as follows:

- An on/off fuel system, for example an electromagnetic valve;
- An electrical firing device;
- A flame detection system which ascertains with safety the presence or absence of the flame and determines the corresponding control orders. For gas burners, the detection sensor is generally an ionization type, while for liquid fuel or mixed fuel burners the sensor is usually a photocell type;
- A timing circuit which establishes the safety time;
- A lock out circuit in case of failure In the case of single-stage regulation, in order to start-up or stop the burner all that is needed is a device that activates the heat regulation handling system.

As a rule, it is represented by the contact of a thermostat with two settings, which regulates, for example, the water temperature of a boiler. When the temperature decreases with respect to the value prescribed, the thermostat requests heat, therefore it closes the contact and starts up the burner; vice versa for an increase in temperature the contact opens and the burner is stopped.
The control panel, which differs in relation to the output and the type of burner, having received the heat request signal, establishes the running sequence of the various devices.
If it becomes necessary to supply the thermal output on two levels, one can use a two-stage type regulation. In this case, two separate thermostats called first flame and second flame control the firings. Each of the two thermostats behaves like the single thermostat in the single-stage burner, activating/deactivating the release devices of the fuel and combustion air.

The two thermostats function at different temperature levels. The thermostat that controls the first stage must be calibrated to a higher temperature than that of the second thermostat [14].

3.1.2 THE HEAT EXCHANGER

As said before, the heat exchanger is composed of two sections: the combustion chamber and the smoke tubes (in the smoke tubes-type heat generators). The Combustion chamber is nothing else than a cylinder, in which the burner shoots the flame. The temperatures in this zone are pretty high (around 1000-1200°C), so that the radiation heat transfer is not to be neglected. The treatment of the heat exchange in the combustion chamber will be discussed elsewhere.

The second part of the heat exchange is in the smoke tubes. Here the combustion products, leaving the combustion chamber, are cooled down to the required temperature for the exiting gases from the boiler.

If the characteristics of burners are almost similar among the various brands and boiler typologies, we can’t say the same for the characteristics of the heat exchangers. In fact, the design option of the heat exchanger section can vary in strong way considering different boiler models.

In the following paragraph will be described a specific architecture of heat generator, produced by Vaillant.

3.1.2.1 OIL-FIRED CONDENSING BOILER VAILLANT *ICOvit VKO 246-7*

The heat generator analyzed is an oil fired condensing floor-standing boiler, Vaillant *ICOvit exclusiv VKO 246-7*. The boiler is characterized by a two stage burner with a Burner Heating Power of 14 ÷ 23 kW.

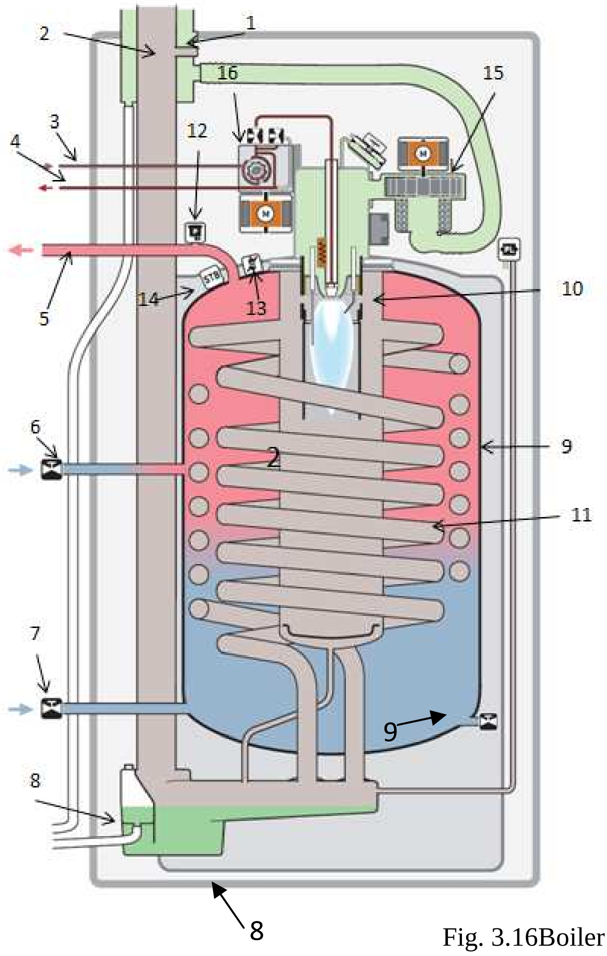
The boiler's body is cylinder in deep-drawnstainless steel with a diameter of 500 mm, approximately 700 mm of length. It contains the combustion chamber, a cylinder with a diameter of 180 mm, approximately 650 mm of length and two separated stainless steelcoils, approximately 9 m length for each coil, that runthrough the boilerbody. Combustion chamber and the coils are in thermal contact with the water content (85 l) by the boiler body.

Fig. 3.15 The BoilerVaillant*ICOvitexclusiv VKO 246-7*

The main components of the boiler studied are shown in Fig.3.16.

The combustion air is preheated in the coaxial air-to-air heat exchanger (1), and is aspirated by the air fan (15).

Oil is sucked from the pump(14), preheated and sent to the oil nozzle, atomized and mixed with combustion air.



1. Coaxial air pre-heater
2. Chimney
3. Oil to the oil pump
4. Drain oil to the oil tank
5. Supply water
6. Return water from an eventual water storage
7.Return water from heating system
8. Flue gas collector and condensate collector
9. Boiler body
10. Combustion chamber
11. Helical coil
12. Water pressure gauge
13. NTC probe
14.Safety temperaturelimiter
15. Air fan
16. Oil Pump

Fig. 3.16Boiler's sketch

The arrangement of the combustor is vertical reverse flame: the burner fires out from the top of the combustion chamber to the bottom. The hot gases reach the bottom of the combustion chamber and there they reverse their flow direction, going back, from the bottom to the top of the chamber.

Meanwhile, outside the combustion chamber, water is flowing in co current with the second passage of the gases inside the combustion chamber.

The combustion products exit the combustion chamber at the top of it. Here the gases stream splits in two equal parts each one entering one coil. The coils have opposite rotation verse. The gases go through the coils cooling down in a counter current configuration with the water outside the coils. Actually, In the case of coils it would not be possible to speak of fluids in parallel flow or counterflow. In fact, each section of the coil is hit by the fluid outside the tubes in such a way to be considered cross flow. Therefore, the coil is the sum of elements in which the fluxes are in cross flow. Usually, though, if the inside fluid enters the coils in correspondence of the outlet of the external fluid, it is customary to speak of fluids in counter flow [29].

The gases exit the coils and they are collected in the flue gas collector. Here the condensate from flue gases is collected and sent to the neutralizer, to decrease the condensate acidity that, otherwise, could not be disposed of in publicsewage.

From the flue gas collector the flue gases are sent to the stack. In the path from the boiler outlet to the chimney outlet, the combustion products are cooled down in the preheating process of the combustion air.

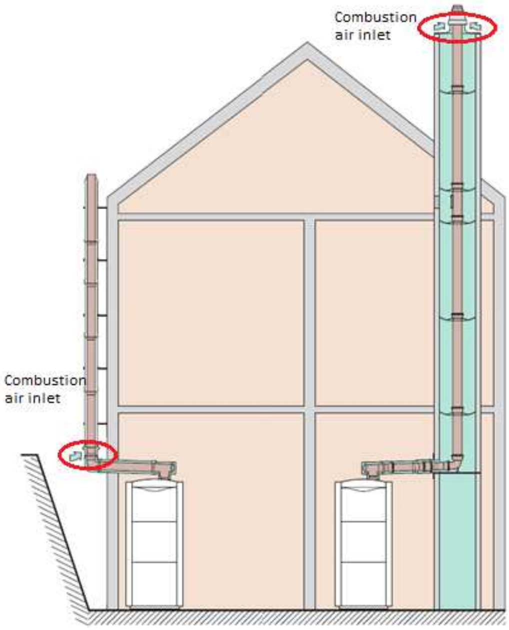
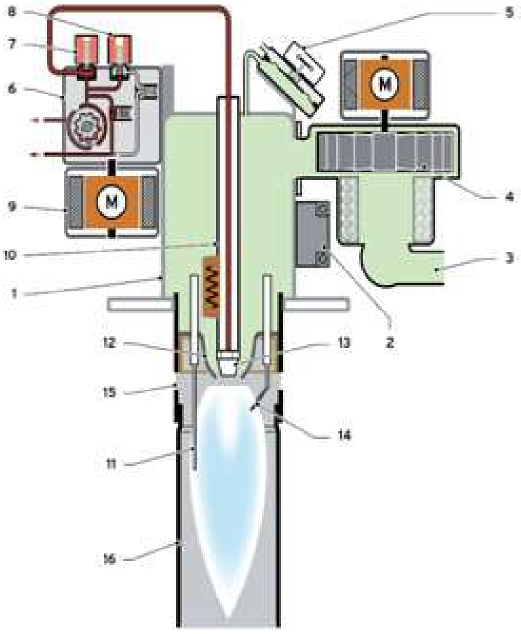


Fig. 3.17 Chimney and Combustion Air Preheating in different installation modes

The oil burner is composed of:



1. Burnercase
2. Ignition transformer
3. Combustionair inlet
4. Blowerwith silencer
5.Air differential pressure gauge
6. Two-stage oil pump
7. Solenoidvalve 1
8. Solenoidvalve 2
9. Burnermotor
10.Nozzlewith oil preheating
11.Ionization detector
12.Air nozzle
13.Oil nozzle
14.Ignition electrode
15.Recirculation port
16.Flame tube

Fig.3.18 Burner’s sketch

When a heat requirement is recorded by the room thermostat or by a weather-dependent control, the oil pre-heater(10), that is upstream of the nozzle, is turned on.

The burner motor (9) is switched on. The oil pump (6) sucks the Fuel oil from the tank and transfers it under pressure to the Solenoid valve1 (7), that is set at the first pressure level, for the First Stage of power production. After no more than 3 minutes, is reached the oil temperature of approximately 55 °C by the preheating system, and the release thermostat closes the control circuit for the oil-burner. The blower motor runs to - the pressure cell in turn <4000 rpm - and reached briefly the maximum speed.

After the fan reaches the starting speed (about 60% of maximum speed) the ignition transformer (2) is turned on and the solenoid valve 1 for the first stage is opened. The oil occurs under high pressure from the oil jet from (13) and is atomized.

The combustion air from the blower (4) passes through the air nozzle (12) that promotes the mixing with the oil mist, obtaining an ignitable mixture. This is ignited by a spark produced by the ignition electrodes (14). The flame is formed in the flame tube (16) and the Ionization electrode detectsthe flame.After astabilization period, if the heat demand from the load is not satisfied with the First Stage power, the solenoid valve 2 (8) is switched in order to achieve the Second Stage of the burner. Simultaneouslythe bloweris regulated to maintain the appropriate excess air.

Before the release of the oil supply the boiler and the exhaust system are ventilated.

This Period is called the pre-purge.

The fan increases the pressure in the combustion chamber and the still air is set in motion, before the combustion begins. Simultaneously, any existing flammable gases in the combustion chamber is removed [30].

3.1.2.2 COMBUSTION CHAMBER

Heat transmission inside the combustion chamber takes place between a mixture of moving gases and the cylinder walls. The heat transfer mechanisms that take place are: radiation and forced convection.

The starting problem to solve is to determine the hot gas-flame temperature inside the combustion chamber.

[31] propose to get the value of the hot gas-flame temperature as a function of the adiabatic flame temperature (calculated) and an experimental factor γ that is an image of the heat transferred in the combustion chamber, depending on the firing rate and the excess air, while the adiabatic temperature is only influenced by the excess air. They found the following expression:

$$\gamma = \gamma_0(\dot{M}_{fuel}) + 0{,}535\ e \qquad (3.10)$$

$$T_{flame} = \gamma T_{ad} \qquad (3.11)$$

Where γ_0 is a constant experimentally determined and e is the excess air.

The heat transfer rate from the gases and the combustion chamber walls is given by the following formula:

$$Q_{cc,g} = h_{g,r} A_p (T_g - T_w) + h_{g,c} A_p (T_g - T_w) \qquad (3.12)$$

To calculate the heat transfer from the combustion products to the chamber walls, have to be considered convective and radiative heat transfer modes. Since the fluid temperatures are not initially a known parameter, is applied the epsilon-NTU method to get converged values.

The flame is considered as cylinder with a unity view factor to the camber’s walls.

The gas to chamber’s wall radiative heat transfer is calculated with the equation :

$$h_{g,r} = \sigma * \varepsilon_t * \left(T_g^2 + T_w^2\right) * \left(T_g + T_w\right) \tag{3.13}$$

Where ε_t is the flame emissivity, calculated according the Hottel studies results (see Paragraph 3.2.31)

The gas to combustion chamber walls convective heat transfer coefficient,$h_{g,c}$, is calculated by using a Nussuelt number correlation that fits the case of study. In the conditions in the [25]study is used the Gnielinski correlation:

$$Nu_g = \frac{h_{g,c}D}{k} = \frac{\frac{f}{2}(Re_g - 1000)Pr_g}{1 - 12{,}7\sqrt{\frac{\xi}{8}}\left({Pr_g}^{2/3} - 1\right)}\left[1 + \frac{d_{cc}}{L_{cc}}^{2/3}\right]\left(\frac{T_g}{T_w}\right)^{0{,}36} \tag{3.14}$$

$$f = \left(1{,}58\,Log\left(Re_g\right) - 3{,}28\right)^{-2} \tag{3.15}$$

Valid in the following conditions:

$0.37 < \frac{Tw}{Tg} < 1$ and $2300 < Re_g < 10^4$

Of course, is required an approximate knowledge of the flow and temperature conditions in the combustion chamber, in order to select the correct Nusselt number correlation.

At the Water side, the heat transfer mechanism is convection. Depending on the specific characteristics of the heat exchanger, different correlations must be used. The heat transfer rate between the combustion chamber wall and the water is:

$$Q_{cc,w} = h_{w,,c}A_p(T_w - T_{water}) \tag{3.16}$$

A first distinction to do is if the water flow and heat transfer is characterized by forced convection or free convection. In the first case the correlations to be used are functions of the Reynolds number and Prandtl number, in the second case the correlations are functions of the Grashof number (Gr) and Prandtl,or simply the Rayleig number (Ra=Gr*Pr)

We can consider that the convection mechanism is Free convection if:

$$Gr \gg Re^2$$

In the case studied by [31]the convection is considered as Free convection. The water side heat transfer coefficient is calculated with the Kuehn and Goldstein correlation.

Kuehn and Goldstein correlation for natural convection between horizontal concentric and eccentric cylinders under quasi-steady conditions for Pr ≈ 0.7 and a large range of Rayleigh numbers ($2.2 * 10^2 \leq RaL \leq 7.7 * 107$). The overall correlation they proposed is given by equativo:

$$Nu_{water} = [(Nu_{cond})^{15} + (Nu_{conv})^{15}]^{1/15} \tag{3.17}$$

where the mean Nusselt numbers for conduction and convection are given by equations (3.18) and (3.19) respectively. The inner cylinder is the boiler firebox and the outer cylinder is the water jacket.

$$Nu_{cond} = \frac{2}{ln\left(\frac{D_o}{D_i}\right)} \tag{3.18}$$

$$Nu_{conv} = \left[\frac{1}{Nu_i} + \frac{1}{Nu_o}\right]^{-1} \tag{3.19}$$

Nu_i and Nu_o are the Nusselt numbers for the inner and outer cylinders at boundary-layer conditions:

$$Nu_i = \frac{2}{ln\left[1+\frac{2}{\left(\left(0{,}5*Ra_{Di}^{\frac{1}{4}}\right)^{15}+\left(0{,}21*Ra_{Di}^{\frac{1}{3}}\right)^{15}\right)^{\frac{1}{15}}}\right]} \tag{3.20}$$

$$Nu_o = \frac{-2}{ln\left[1+\frac{2}{\left(\left(0{,}5*Ra_{Do}^{\frac{1}{4}}\right)^{15}+\left(0{,}21*Ra_{Do}^{\frac{1}{3}}\right)^{15}\right)^{\frac{1}{15}}}\right]} \tag{3.21}$$

3.1.2.3 HELICAL COILS

After the combustion chamber we find the helical coils in which the flue gas transfer is further cooled down and enters in the region of possible condensation of water vapor.
The heat exchange in a helical coil has been studied by several authors, and lots of empirical correlations are available.
Helically coiled tubes have been introduced as one of the passive heat transfer enhancement techniques to enhance heat transfer coefficient in various application.

The easiest benefit to be recognized, when is used an helical coil heat exchanger, is its compactness, allowing a large heat exchange area, with a little volume occupation.
In addition, considering an equal heat exchange area, helically coiled tubes are superior to straight tubes, due to the presence of a secondary flow in the tube (Dean flow), normal to the axial direction, which mix the fluid particles enhancing the heat transfer rate [32].
The origin of the secondary flow is as follows:

When a fluid flows through a straight tube, the fluid velocity is maximum at the tube center, zero at the tube wall and symmetrically distributed about the axis. However, when the fluid flows through a curved tube, the primary velocity profile indicated above is distorted by the addition of secondary flow pattern. The secondary flow is generated by centrifugal action and acts in a plane perpendicular to the primary flow. Since the velocity is maximum at the center, the fluid at the center is subjected to the maximum centrifugal action, which pushes the fluid towards the outer wall. The fluid at the outer wall moves in ward along the tube wall to replace the fluid ejected outwards.This results in the formation of two vortices symmetrical about a horizontal plane through the tube center. Figure 3.19 shows a sketch of the secondary flow pattern[33].

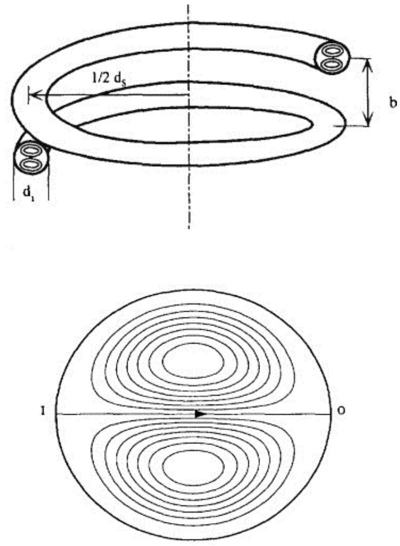


Fig.3.19 The secondary flow in a coiled tube

The heat transfer rate for the coil could be estimated by:

$$Q_{coil} = A_i * U_{coil} * (T_g - T_{water}) \tag{3.22}$$

Where:

U_{coil}: *is the global heat transfer coefficient;*

$$U_{coil} = \frac{1}{\frac{1}{h_{coil,i}} + \frac{A_i}{h_{coil,o} * A_o}} \tag{3.23}$$

$h_{coil,i}$,$h_{coil,o}$: *are respectively the inner and outer heat transfer coefficients;*
A_i,A_o: *are respectively the inner and outer heat transfer areas;*
T_g: *is a representative gas temperature in the coil;*
T_{water}: *is a representative water temperature out of the coil.*

In the following pages will be given some indication about the Nusselt number correlations for helical coils.

A helical coil can be geometrically described by the coil radius R, the pipe radius r and the coil pitch $2\pi b$:

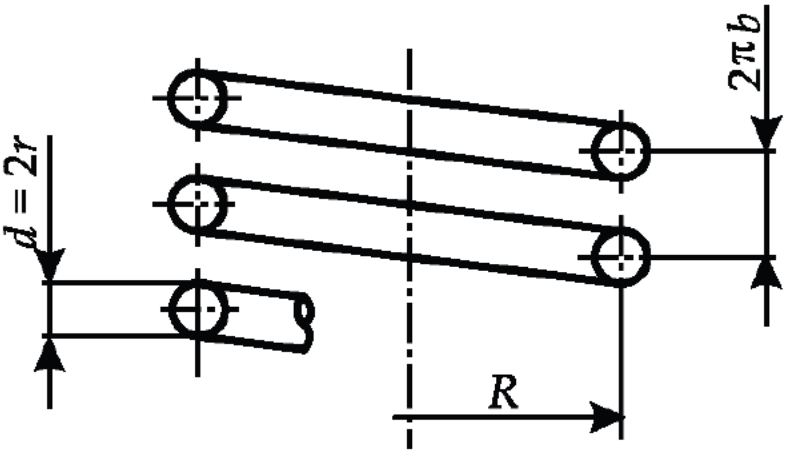


Fig.3.20 Helical coil geometry

The internal heat transfer coefficient

The Reynolds critic number for helical coiled pipes suggested by Schimidt is:

$$Re_{crit} = 2300\left[1 + 8{,}6\left(\frac{r}{R}\right)^{0{,}45}\right] \tag{3.24}$$

And, for curvature ratios, $\delta = \frac{r}{R}$,less than 1/860, the critical Reynolds number was found to correspond with that for straight pipes.

The most used correlation for the Nusselt number of helical coiled pipes are the following:

- Schmidt's correlations:
  - Laminar flow:

$$Nu = 3{,}65 + 0{,}08\left[1 + 0{,}08\left(\frac{r}{R}\right)^{0{,}9}\right] Re^{\left[0{,}5+0{,}2903\left(\frac{r}{R}\right)^{0{,}194}\right]} Pr^{1/3} \qquad 100 < Re < Re_{crit} \tag{3.25}$$

  - Turbulent flow:

$$Nu = 0{,}023\left[1 + 14{,}8\left(1 + \frac{r}{R}\right)\left(\frac{r}{R}\right)^{1/3}\right] Re^{\left[0{,}8+0{,}22\left(\frac{r}{R}\right)^{0{,}1}\right]} Pr^{1/3} \qquad Re_{crit} < Re < 22000 \tag{3.26}$$

$$Nu = 0{,}023\left[1 + 3{,}6\left(1 - \frac{r}{R}\right)\left(\frac{r}{R}\right)^{0{,}8}\right] Re^{0{,}8} Pr^{1/3} \qquad 22000 < Re < 150000 \tag{3.27}$$

- Seban and McLaughlin, for turbulent flow:

$$Nu = 0{,}023\left(\frac{r}{R}\right)^{0{,}1} Re^{0{,}85} Pr^{0{,}4} \qquad Re > Re_{crit} \tag{3.28}$$

- Seider and Tate, for turbulent flow:

  Used in the case of large property variations from the bulk flow to the wall temperature:

$$Nu = 0{,}027Re^{0{,}8}Pr^{1/3}\left(\frac{\mu_{bulk}}{\mu_{wall}}\right)^{0{,}14} \qquad \begin{array}{l} 0{,}7 < Pr < 16000 \\ Re > 10000 \\ \frac{L}{D} > 10 \end{array} \tag{3.29}$$

- Petukhov

An accurate correlation valid in fully developed turbulent flow:

$$Nu = \frac{(f/2)RePr}{1{,}07 + 12{,}7(f/2)^{0{,}5}(Pr^{2/3} - 1)}\left(\frac{\mu_{bulk}}{\mu_{wall}}\right)^{0{,}14} \qquad \begin{array}{c} 0{,}5 < Pr < 2000 \\ 10000 < Re < 500000 \end{array} \tag{3.30}$$

Where f is the friction factor that can be obtained from the Moody diagram or from the Petukhov's fiction factor correlation:

$$f = \frac{1}{1{,}58ln(Re) - 3{,}28} \qquad 3000 < Re < 500000 \tag{3.31}$$

The external heat transfer coefficient

The heat transfer coefficient in the annulus, h_o, can be calculated with the following equations [34]

$$Nu = \frac{h_o D_e}{k} = 0{,}6Re^{0{,}5}Pr^{0{,}31} \qquad 50 < Re < 10000 \tag{3.32}$$

or:

$$Nu = \frac{h_o D_e}{k} = 0{,}36Re^{0{,}55}Pr^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0{,}14} \qquad Re > 10000 \tag{3.33}$$

Another approach in order to define a correlation to evaluate the outside Nusselt number is to use correlation for tube bundles in cross flow such as the one proposed by Zukauskas. This correlation has a 15% uncertainty. More inaccuracy may result from this particular analysis, as flow over the helical coils will be of a similar, but different nature than flow over tubes in crossflow The flow over the helical coils may produce a swirling motion that tubes in crossflow would not. The swirling motion increases the movement of the water, thereby enhancing heat transfer.

$$Nu = CRe_{D,max}^{m}Pr^{n}\left(\frac{Pr}{Pr_w}\right)^{1/4} \tag{3.34}$$

$Re_{D,max}$	**C**	**M**	**n**
0 ÷ 100	*0,9*	*0,4*	*0,36*
100 ÷ 1000	*0,52*	*0,5*	*0,36*

$Re_{D,max} = \frac{u_{max} D_{t,o}}{\upsilon}$ (3.35)

Zukauskas' crossflow correlation requires a Reynolds number, $Re_{D,max}$, which is based upon the maximum fluid velocity ,u_{max} , and the diameter of the tubes in crossflow, $D_{t,o}$. The kinematic viscosity, υ, is evaluated at the average heat exchanger water temperature[35].

3.2 ENERGY BALANCE AND LOSSES

Looking at boiler as a complex system, we can describe its principal basic ideal parts as follows:

- Combustion zone
- Burner
- Primary heat exchanger
- Secondary heat exchanger
- Tertiary heat exchanger.

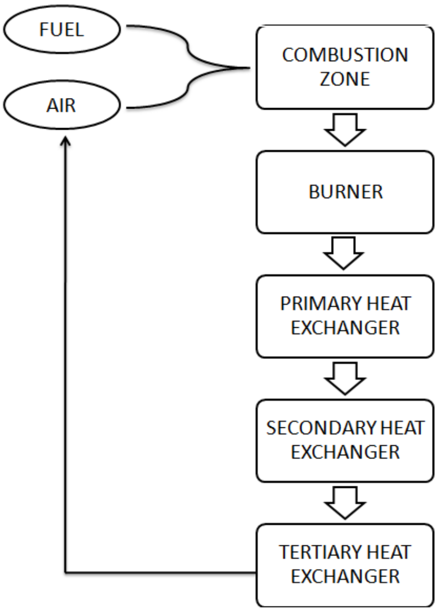


Fig. 3.21 Heat generator representation

Such a distinction is made on the basis of the basic function of the component. We can adopt the following definitions:

- *combustion*: is the place where the fuel chemical energy is converted into thermal energy (sensible radiant and convective heat and latent condensing heat) and where are determined the heat losses due to incomplete combustion and to combustion air excess ;
- *Burner*: represents the device dedicated to the combustion process. The distinction between burner and combustion zone, is not always made, since the difficulties in temperature measurements, that leads difficulties in modeling the behavior.

- *Primary heat exchanger*: radiation heat and the convention heat coming from the burner is transferred to the boiler water. Here is not allowed any latent heat recuperation, the combustion products leave this exchanger at temperature higher than their dew point temperature. This heat exchanger is the only one in a non condensing boiler
- *Secondary heat exchanger:* only in condensing boilers is present. Its function is to further cool the flue gases to a temperature level where most of the latent heat can be recovered, recovering at the same time the sensible heat remaining in the flue gases.
- *Tertiary heat exchanger:* is considered when is present in the heat generator configuration, a dedicated place in which the combustion air is pre-heat with the exiting flue gases.

Each one of the above ideal components may be physically recognizable as a dedicated appliance in the boiler system, or more than one of the above ideal components are built-in in a single physical component. For example, the combustion chamber is the physical component in which is possible to identify three ideal components: the combustion zone, the burner and a part of the primary heat exchanger.

Moreover, the ideal components may be located between two physical components, for instance, in the Vaillant boiler we can identify the primary heat exchanger between the combustion chamber and the first part of the helical coil heat exchanger.

3.2.1 COMBUSTION

During the combustion process, chemical Energy contained in the fuel is converted into three energy forms:

- Radiation energy of the flame/burner;
- Convection energy of the combustion products;
- Latent heat of the water vapor;

The total heat released by the combustion process is the Combustion Heat that is represented by the Higher Heating Value, HHV, of the fuel:

$$HHV = Q_{flame} + Q_{latent} + Q_{xsair} + Q_{fuel-loss} \tag{3.36}$$

Where Q_{flame} is the sum of radiation energy and convection energy.

If we leave out the latent heat contained in the water vapor, we obtain the Lower Heating Value, LHV:

$$LHV = Q_{flame} + Q_{xsair} + Q_{fuel-loss} \tag{3.37}$$

To calculate the flame temperature we can refer to the “calorific flame temperature” or to the “adiabatic flame temperature”. The first one can be calculated from the enthalpy of the fuel, under the assumption that all the energy is converted into hot combustion products. The temperature increase, over the ambient temperature, of the combustion products comes from:

$$LHV = mc_p\Delta T \tag{3.38}$$

Where :

m: *is the mass of the combustion products;*
c_p: *is the specific heat of the combustion products;*
ΔT: *is the temperature increase of the combustion products, from the ambient temperature.*

This value of temperature is never reached in practice, because of the dissociation effects.
Another value of temperature that takes into account the dissociation process is the “adiabatic flame temperature”. Especially for gases the adiabatic flame temperature is practically equal to the calorific flame temperature at the commonly used air factor values.

3.2.1.1 LATENT CONDENSATION HEAT

The latent condensation heatis the heat contained in the water vapor from combustion when condensing. Numerically it is the difference between Higher and Lower Heating Values (HHV and LHV)of the fuel.
In theory, the latent condensation heat can be fully recovered, if somewhere in the boiler, before the combustion products go up the chimney are cooled to the ambient temperature.
In a *non condensing boiler*the flue gases leave the boiler at a temperature between 200-300°C. In this condition the water content is still in vapor phase, so that no latent condensation heat is recovered by the boiler, and is completely lost in the atmosphere. To establish where this point of total loss is, the EN standards says that there is no condensation risk if the flue gases are leaving the boiler at a temperature above 160°C.
In a *condensing boiler*there is the possibility of recovering the latent condensation heat of the flue gases. The EN standards speak of a dedicated “condensing boiler” at flue gas temperature of lower than 80°C.
Condensing’ relates to the fact that the water vapor in the flue gas comes into contact with a cold surface of the heat exchanger and then turns into liquid, releasing the latent heat of condensation.

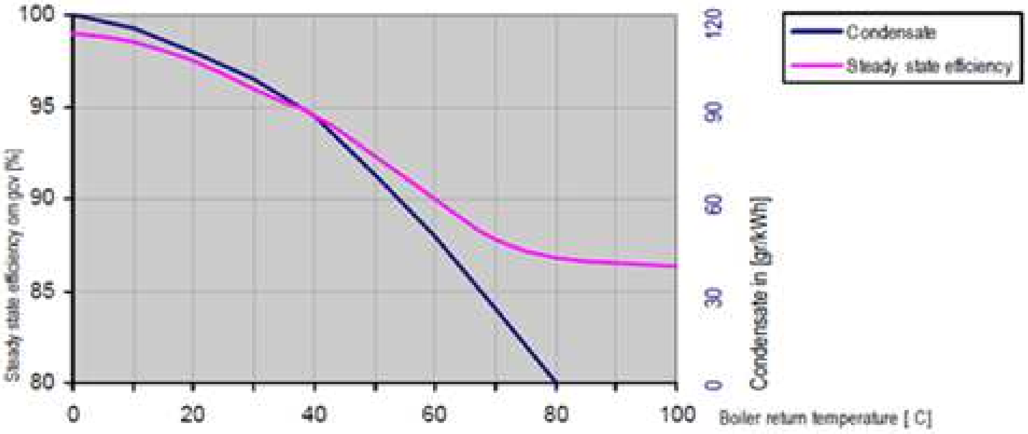


Fig. 3.22 Steady state efficiency and amount of condensate related to return temperature in gas fired boiler

3.2.1.2 HEAT LOSS IN EXCESS COMBUSTION AIR

When extra air is provided for the combustion process, beside the benefit of lowering the unburned production, enhancing combustion due to an higher amount of oxygen available, is introduced a penalty as well. In fact, if more air than the necessary is introduced in the combustion process, an extra amount of air has to be heated by the combustion, from the ambient temperature to the combustion temperature.

A "rule-of-thumbs" is that: every 1% O_2 extra results in 0,5% efficiency loss.

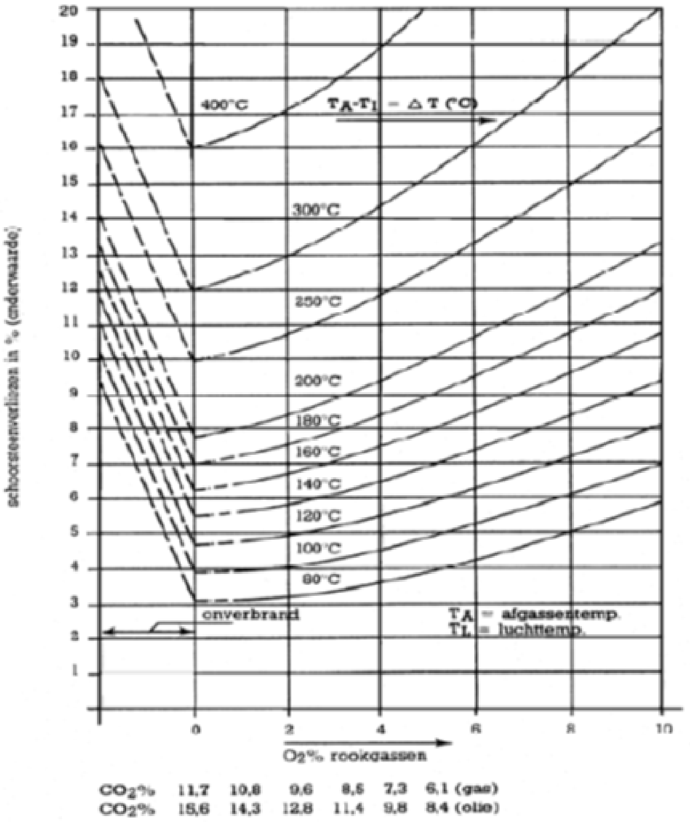
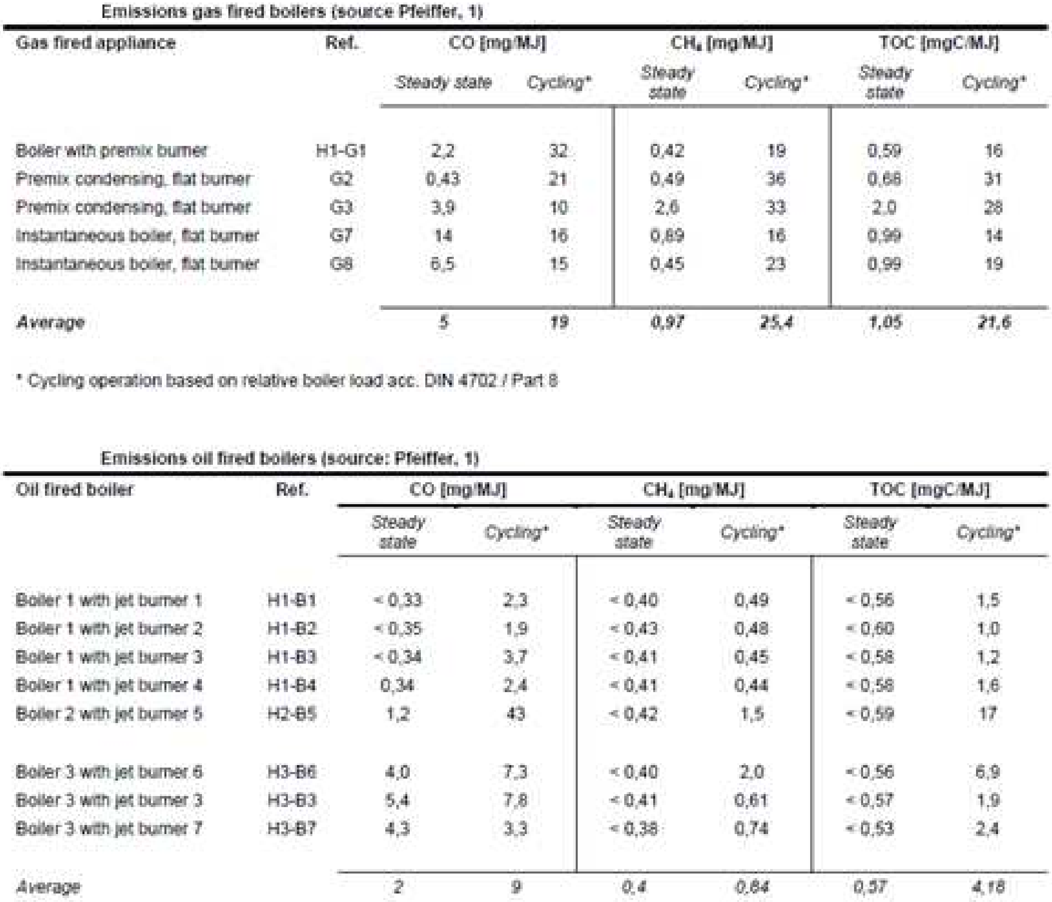


Fig.3.23 Heat losses and air excess

3.2.1.3 FUEL LOSS

Is the energy loss due to incomplete combustion that manifested by the presence of carbon monoxide (CO2) and total organic compounds (TOC).

Pfeiffer 2001 of the University of Stuttgart executed several tests on emissions of oil and gas-fired boilers, looking at the case of stationary operation (as required in EN standards) and at the case of cycling operations (according to DIN 4702-8). The results that was found are shown in the following table.



Emissions gas fired boilers (source Pfeiffer, 1)

Gas fired appliance	Ref.	CO [mg/MJ]		CH_4 [mg/MJ]		TOC [mgC/MJ]	
		Steady state	*Cycling**	*Steady state*	*Cycling**	*Steady state*	*Cycling**
Boiler with premix burner	H1-G1	2,2	32	0,42	19	0,59	16
Premix condensing, flat burner	G2	0,43	21	0,49	36	0,68	31
Premix condensing, flat burner	G3	3,9	10	2,6	33	2,0	28
Instantaneous boiler, flat burner	G7	14	16	0,89	16	0,99	14
Instantaneous boiler, flat burner	G8	6,5	15	0,45	23	0,99	19
Average		***5***	***19***	***0,97***	***25,4***	***1,05***	***21,6***

* Cycling operation based on relative boiler load acc. DIN 4702 / Part 8

Emissions oil fired boilers (source: Pfeiffer, 1)

Oil fired boiler	Ref.	CO [mg/MJ]		CH_4 [mg/MJ]		TOC [mgC/MJ]	
		Steady state	*Cycling**	*Steady state*	*Cycling**	*Steady state*	*Cycling**
Boiler 1 with jet burner 1	H1-B1	< 0,33	2,3	< 0,40	0,49	< 0,56	1,5
Boiler 1 with jet burner 2	H1-B2	< 0,35	1,9	< 0,43	0,48	< 0,60	1,0
Boiler 1 with jet burner 3	H1-B3	< 0,34	3,7	< 0,41	0,45	< 0,58	1,2
Boiler 1 with jet burner 4	H1-B4	0,34	2,4	< 0,41	0,44	< 0,58	1,6
Boiler 2 with jet burner 5	H2-B5	1,2	43	< 0,42	1,5	< 0,59	17
Boiler 3 with jet burner 6	H3-B6	4,0	7,3	< 0,40	2,0	< 0,56	6,9
Boiler 3 with jet burner 3	H3-B3	5,4	7,8	< 0,41	0,61	< 0,57	1,9
Boiler 3 with jet burner 7	H3-B7	4,3	3,3	< 0,38	0,74	< 0,53	2,4
Average		*2*	*9*	*0,4*	*0,64*	*0,57*	*4,18*

Fig. 3.24 Emissions by gas and oil boilers in steady state and cycling operations

In principle we can say that the fuel loss due to incomplete combustion can be assessed by measuring the amount of these sub products of incomplete combustion.

3.2.2 BURNER

Usually the distinction between combustion energy balance and the burner energy balance are not separated, because the actual measurements of flame temperature is very difficult without some sort of burner. But is interesting to distinguish this two ideal processes, because the interactions between the flame and the burner lead to different characteristics of the division between convective energy and radiation energy.

When the temperature of combustion products is measured at the burner, the so-called Combustion Temperature, is found a temperature value that is always approximately 100-200° lower than the adiabatic flame temperature.
Looking at the following picture (Fig. 3.25), that represents the Combustion Temperature at various burner power output, against the adiabatic and caloric temperature, we can observe the following facts:

If we assume that the power output of the burner is a measure of the flame size, we can see that for a smaller flame size (i.e. for a partial load of the same burner-10 kW) the Combustion Temperature is significantly lower than for a bigger size (i.e. the nominal load-35 kW).

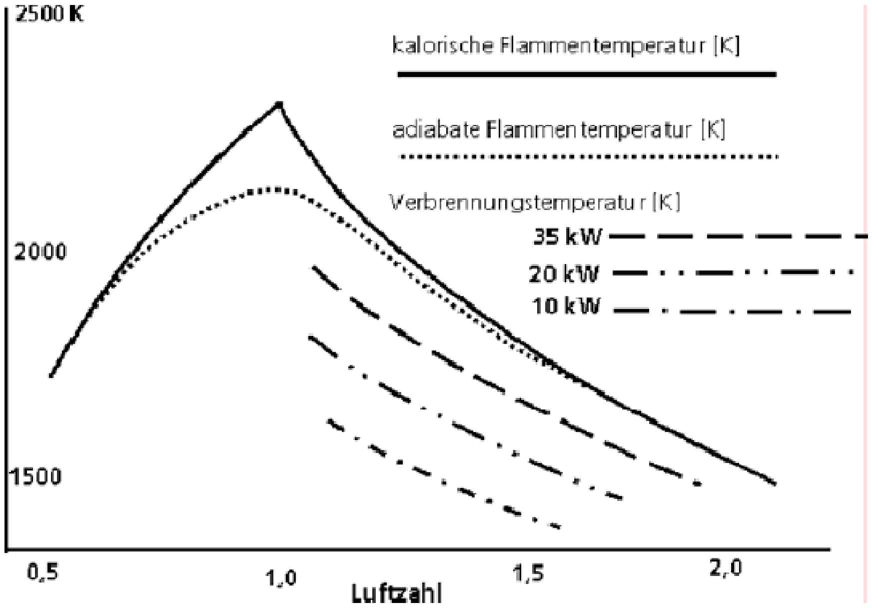
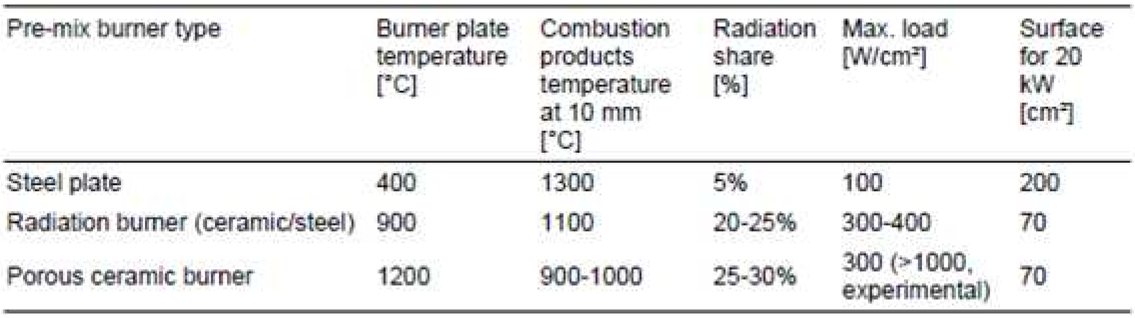


Fig. 3.25 Flame temperatures and burner power output

This fact has an influence in the share of radiation energy and convection energy produced in the burner.

Between 35 and 10 kW power the temperature difference is some 350 K. Assuming this is proportional to the temperature difference with the ambient (ca. 1700°C) this means that at 10 kW (30% load) the share of radiation energy has increased by 20% with respect of 35 kW (100% load). On average, every 10% decrease in load has yielded around 2,5-3% more radiation share.

This contra-intuitive aspect could be explained from a different point of view: with a larger burner plate (compared to its nominal capacity in W/cm²) the radiation share increases (and the convection share, i.e. the temperature of the combustion gases, decreases). This result is showed in Fig. 3.26 in which are compared different burner technologies, characterized by different combustion temperatures (and radiation energy share):



Pre-mix burner type	Burner plate temperature [°C]	Combustion products temperature at 10 mm [°C]	Radiation share [%]	Max. load [W/cm²]	Surface for 20 kW [cm²]
Steel plate	400	1300	5%	100	200
Radiation burner (ceramic/steel)	900	1100	20-25%	300-400	70
Porous ceramic burner	1200	900-1000	25-30%	300 (>1000, experimental)	70

Fig. 3.26 Estimated temperatures and loads for pre-mix burners (air factor 1,2; no preheat air)

3.2.3 PRIMARY HEAT EXCHANGER

In the primary heat exchanger the radiation heat and the convention heat coming from the burner is transferred to the boiler water.

In non-condensing boiler, the primary heat exchanger is the only heat exchanger of the system, since in this kind of heat generator is not allowed any latent heat recovery.

In the primary heat exchanger the heat transfer process is simultaneously duel to radiation and convection .

In the heat exchanger/combustion chamber there are parts that can be “seen” by the burner and are subject to radiation heat.

All parts of the heat exchanger are subject to the convection heat.

3.2.3.1 RADIATION HEAT TRANSFER

The relation between radiation and convection is quite complex, but as proposed in [36] is possible to simplify the radiation modeling in industrial burner, by applying the Stefan-Bolzman formula:

$$Q_{rad} = A * \varepsilon_{res} * \sigma_s * (T_g^4 - T_w^4) \qquad (3.39)$$

Where :

Q_{rad}: *is the radiation heat energy;*
A: *is the surface of radiation heat transfer;*
ε_{res}: *is the resulting emission factor;*
σ_s: *is the Stefan-Bolzmancostant;*
T_g, T_w: *are the temperature of gas and wall in K.*

A key factor in this energy transfer is represented by the “resulting emission factor” ε_{res}.

The flame is approximated by a cylinder with unity view factor to the chamber walls.

Assuming that gas an combustion chamber walls are behaving as a gray-Lambert surface, the total emissivity and the total absorptivity are equal. Thus is possible to calculate ε_{res} with the following formula:

$$\varepsilon_{res} = \frac{\varepsilon_{wall} * \varepsilon_{gas}}{\varepsilon_{wall} + \varepsilon_{gas} - \varepsilon_{wall} * \varepsilon_{gas}} \qquad (3.40)$$

To evaluate the flame emissivityε_{gas}, luminous (ε_s) and non luminous (ε_g) contributions have to be estimated [31].

The luminous contribution is given by the soot present in the flame. If we consider the typical (new) domestic boiler the soot production is negligible, thus we can disregard the luminous contribution.

For hydrocarbons flames, non luminous emissivity ε_g is mainly due to CO2 and H2O. It can be modeled by spectral models or analytical expressions obtained from fits of experimental data. A

simplified model has been developed by Hottel (1954) where ε_g was correlated in terms of gas temperature, gas total pressure, radiating species partial pressures and radius of the hemisphere.
The gas emissivity ε_g has been correlated in terms of the temperature T_g, the total pressure p of the gas, the partial pressure p_g of the gas species (such as water vapor, carbon dioxide, etc.), and the radius L of an equivalent hemispherical gas mass.

The water vapor emissivity $\varepsilon_{H_2O}$is presented in Fig.3.27 as a function of the gas temperature $T_g$and $p_{H2O}L_e$.

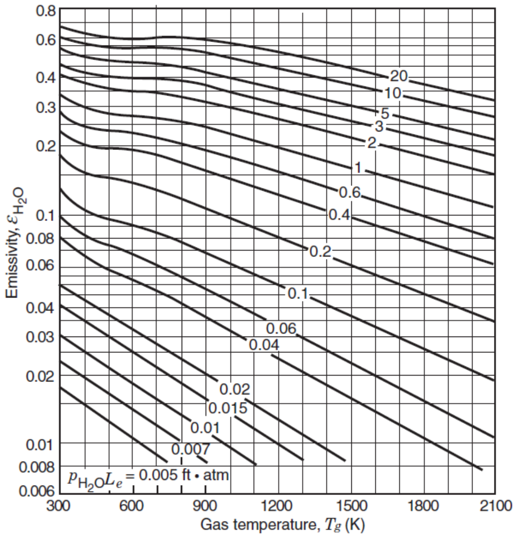


Fig.3.27 Emissivity of water vapor in a mixture with nonradiating gases at 1 atm total pressure and of hemispherical shape. (From Hottel, 1954)

Here $p_{H2O}$is the partial pressure of water vapor in the gas mixture at a total pressure of 1 atm; and L_e is the mean beam length to take into account the size and shape of the gas geometry, which is different from a hemispherical geometry.
If the total pressure of the gas is different from 1 atm, a correction factor C_{H2O} (to be multiplied to "H2O for 1 atm pressure) is obtained from Fig.3.28.

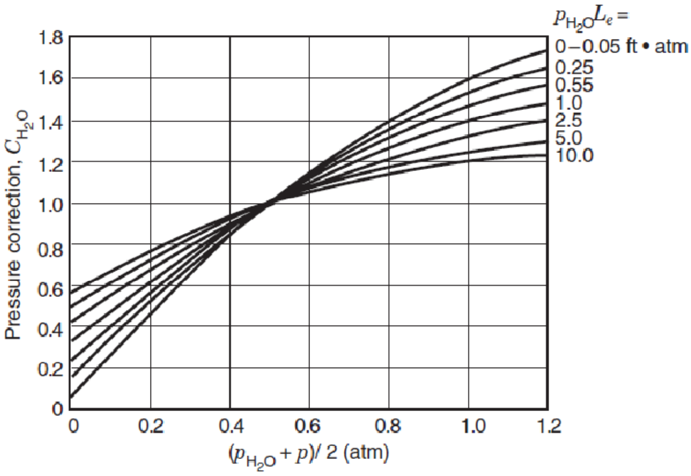
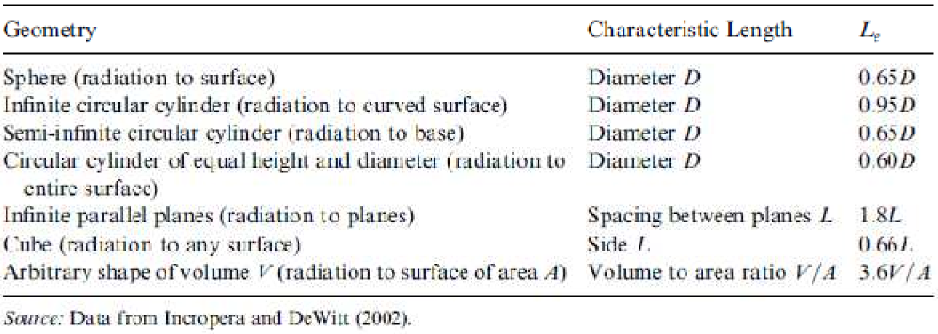


Fig.3.28 Correction factor for obtaining water vapor emissivity at pressures other than 1 atm; (From Hottel, 1954.)

The mean beam length L_e is provided in Table 3.29 for various gas geometries (Incropera and DeWitt, 2002).



Geometry	Characteristic Length	L_e
Sphere (radiation to surface)	Diameter D	$0.65D$
Infinite circular cylinder (radiation to curved surface)	Diameter D	$0.95D$
Semi-infinite circular cylinder (radiation to base)	Diameter D	$0.65D$
Circular cylinder of equal height and diameter (radiation to entire surface)	Diameter D	$0.60D$
Infinite parallel planes (radiation to planes)	Spacing between planes L	$1.8L$
Cube (radiation to any surface)	Side L	$0.66L$
Arbitrary shape of volume V (radiation to surface of area A)	Volume to area ratio V/A	$3.6V/A$

Source: Data from Incropera and DeWitt (2002).

Table 3.29 Mean Beam Lengths L_e for Various Gas Geometries

Similarly, the carbon dioxide emissivity ε_{CO_2} and corresponding correction factor C_{CO2} are provided in Figs. 3.30 and 3.31, respectively.

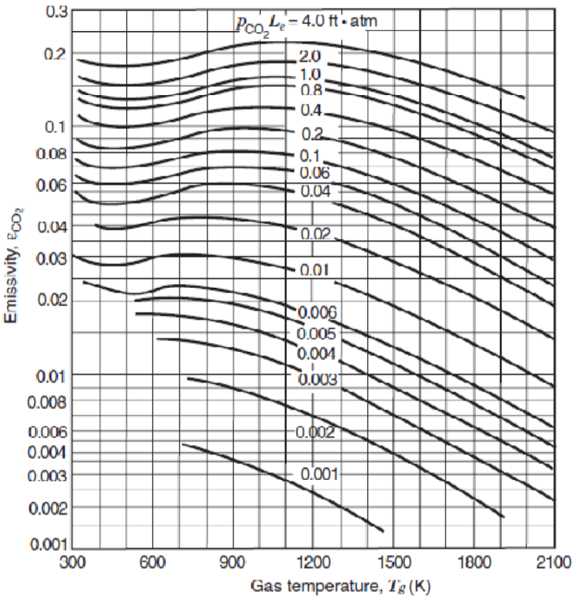


Fig. 3.30 Emissivity of carbon dioxide in a mixture with nonradiating gases at 1 atm total pressure and of hemispherical shape. (From Hottel, 1954.)

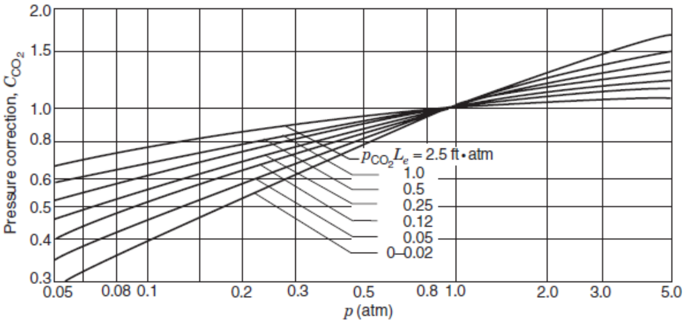


Fig. 3.31 Correction factor for obtaining carbon dioxide emissivity at pressures other than 1atm. (From Hottel, 1954.)

The emissivity value for water vapor and carbon dioxide apply only when they are alone with other non-radiating components in the gas mixture. However, when both are present in the gas with other non-radiating components, the total gas emissivity is given by:

$$\varepsilon_g = C_{H_2O} * \varepsilon_{H_2O} + C_{CO_2} * \varepsilon_{CO_2} - \Delta\varepsilon \tag{3.41}$$

where the correction factor $\Delta\varepsilon$ is given in Fig. 3.32 [36].

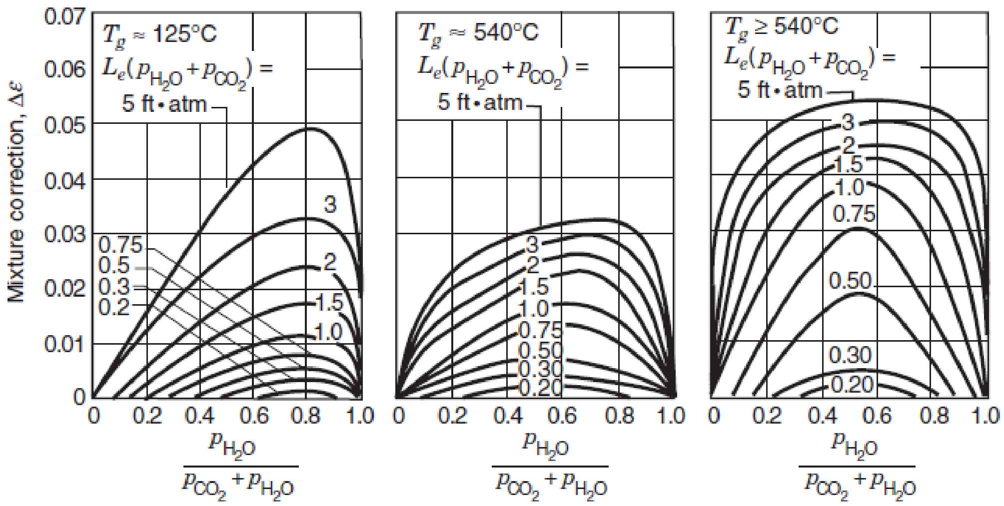


Fig. 3.32Correction factors associated with mixtures of water vapor and carbon dioxide. (From Hottel, 1954.)

Considering a particular case, representative of a common situation for an industrial boiler. For a fixed combustion chamber wall temperature and fixed flue gas composition, is shown the variation of the resulting emission factor, ε_{res}, when the gas temperature varies, for different combustion chamber heights:

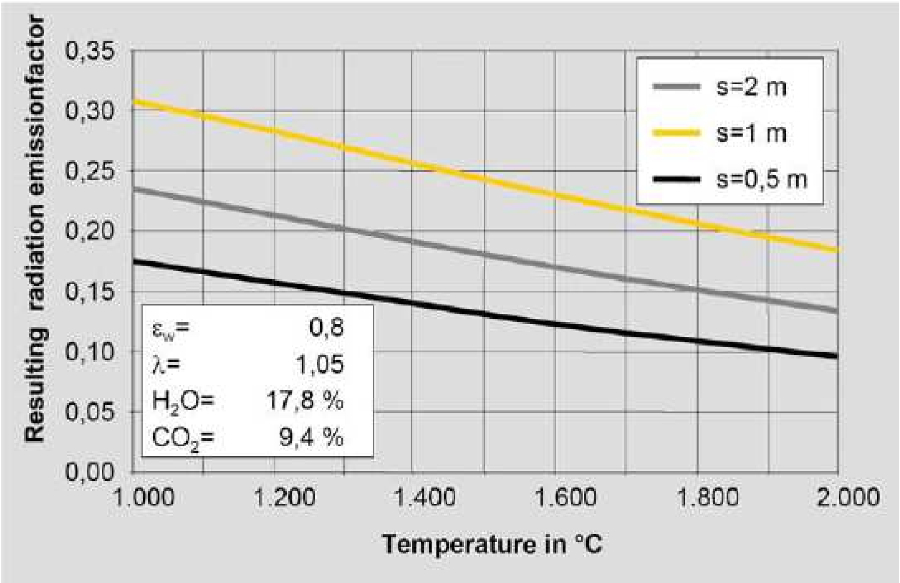


Fig.3.33 Emission factor vs. gas temperature

The graph shows two important things:

- The dimensions of the combustion chamber influences the radiation fraction of the heat transfer;
- The radiation emission factor increases at a lower temperature.

The “ERDGAS REPORT 1/03” mentions a value of ε_{res}=0,2 to 0,3 for normal burners and ε_{res}=0,6 for radiation burners[28].

3.2.3.2 CONVECTIVE HEAT TRANSFER

The convective heat transfer is depending linearly on the temperature difference. A simplified equation for the convection is:

$$\mathrm{Q_{conv}} = \mathrm{A} * \alpha * (\mathrm{T_g} - \mathrm{T_w}) \tag{3.42}$$

Where:

A: is the heat transfer surface;
α: is the heat transfer coefficient;
T_g, T_w: are the temperature of gas and wall

The heat transfer coefficient, α, depends on the flue gas velocity.In the following picture is represented the convective heat transfer coefficient (gas side) as function of the gas velocity in the heat exchanger, for a fixed wall temperature and for two values of gas temperature.

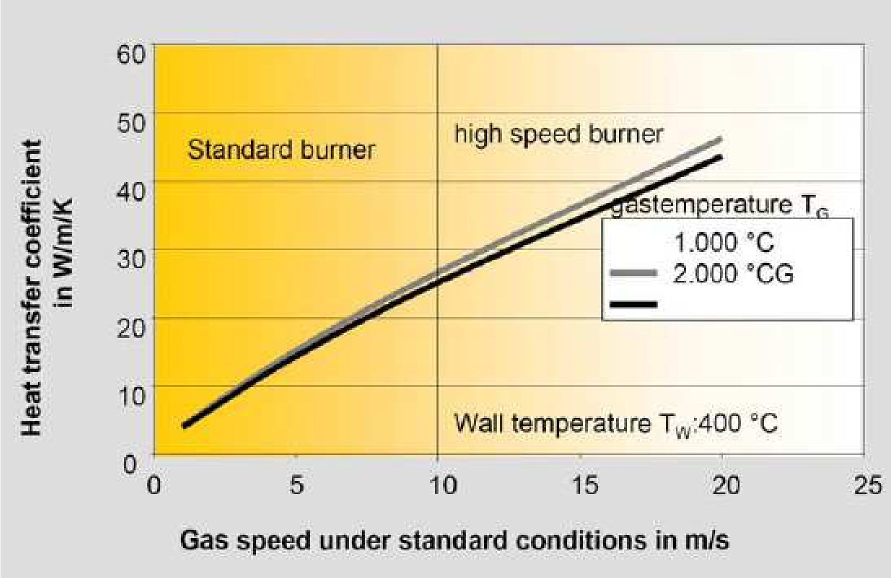


Fig.3.34 Convective heat transfer coefficient vs. gas speed

We can see that the heat transfer coefficient, gas side, is strongly dependent on the gas velocity, more than how it is influenced by the gas temperature.

3.2.4 ENERGY BALANCE OF THE COMBUSTION, BURNER AND PRIMARY HEAT EXCHANGER

If we are analyzing a non-condensing boiler, the ideal model of the heat generator is completed at the level of the Primary Heat Exchanger. In fact, the Secondary Heat Exchanger is used to represent the part of an heat generator in which the flue gases are at a temperature level that allows the condensation. Thus, at this point of the analysis (Combustion, Burner and Primary Heat Exchanger) is possible to define a system energy balance. Conversely, when we are referring to a condensing boiler, we just need to add to the model of a non condensing boiler the Secondary Heat Exchanger that introduces the possibility of a further sensible heat recuperation from the flue gases and the latent heat of condensation.

Now is introduced the energy balance of the Combustion, Burner and Primary Heat Exchanger System. The heat generator in a heating system has the important role of "following" the heat demand of the system. The heat load is not time-constant but it varies with some temporal law, depending on the boundary conditions of the system. This makes the heat generator follow a cycle made of starts and stops during the time.

The dynamic behavior of the heat generator is represented by a series of cycles in which is possible to distinguish the "ON-mode", i.e. the flame is switched on, and the "OFF-mode", i.e. the burner is switched off. Fig. 3.35 shows the energy balance of the heat generator in one cycle.

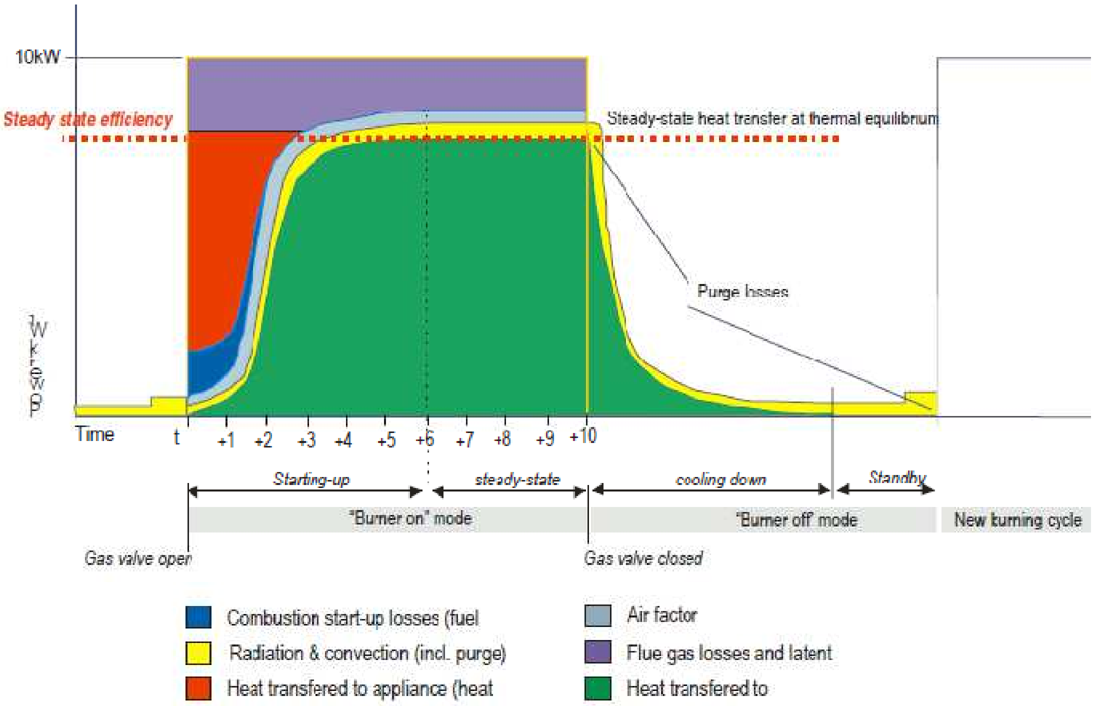


Fig. 3.35 Boiler cycle and energy balance

We can divide the cycle into four section, two for each mode:

In the ON-mode:

- *starting-up:*
    - the flame is switched on. The condition (temperatures and fuel-air mixing) in the combustion chamber in the first moment of the reaction are such that they don't allow an optimal combustion, with an higher production of unburned compounds, that constitute an energy loss.
    - The materials of the boiler structure will absorb and store some of the heat released by the combustion, subtracting heat to transit to the boiler water.
    - The latent heat and the sensible heat of the combustion product that are not recovered by the Primary Heat Exchanger are lost, if there is not any Secondary Heat Exchanger.
    - The heat transferred to the water grows while the heat transferred to the appliance and the combustion star-up loss decrease.
    - During these processes a certain amount of energy is lost through the envelope of the boiler. It depends on the superficial temperature of the boiler envelope, thus, is easy to understand that in the starting-up phase, its magnitude grows until the Steady condition is reached. In fact, in the steady state condition will be reached the highest temperature of the boiler envelope, at which corresponds the highest value of envelope losses
- *steady state:*
    - the most of the heat extracted from the fuel is transferred to the boiler water
    - a certain amount of energy is lost through the boiler casing due to Radiation and Conduction.
    - A certain amount of energy is lost at the chimney because the flue gases are at high temperature, and at the Primary Heat Exchanger is not recovered any amount of latent heat.

In the OFF-mode:

- cooling down: the burner is switched off, and the water pump is still running
    - a certain amount of the energy stored in the structure of the boiler during the starting-up is released to the boiler water;
    - the after purge, made to clean the combustion chamber, removing the combustion products from it, will enhance the heat transfer between the hot gases and the boiler water;
    - The boiler envelope will continue to transfer heat to the boiler room, but with a lower intensity.
- stand-by
    - If is not installed a valve that closes the flue gas duct or the air inlet duct during the OFF-mode, there will be a "chimney effect" that will induce an air flow through the combustion chamber to the chimney that causes a heat loss due to the loss of the internal heat;
    - The Pre Purge, made to remove any unburned trace in the combustion chamber before the new cycle, will determine an energy loss through the chimney,

- The boiler envelope will continue to transfer heat to the boiler room;

In the following paragraphs are presented the energy losses that could be localized in *the Primary Heat Exchanger.*

3.2.4.1 FLUE GAS LOSSES

The primary heat exchanger is designed to capture the radiation heat from the burner and to transfer the heat from the combustion products to the boiler water, but without condensation.

The important parameters are the heat exchange surface, the temperature difference between water and flue gas, and the convective coefficients, which depends on the configuration of the heat exchanger, normally counter flow.

After passing the primary heat exchanger the flue gases temperature are in a range of temperature of:

- 130 ÷140°C in low temperature and condensing boilers;
- 180 °C in modern atmospheric boilers;
- Up to 250 °C in older atmospheric boilers;

This means that, compared to the original 1730°C circa- at the entrance of the primary heat exchanger- and the ambient temperature, around the 6-11 % of the flue gas energy exits the primary heat exchanger, without taking into account the latent condensing heat.

In case of atmospheric boiler or low temperature boiler, this value corresponds to the final flue gas loss.

An important parameter in this context is the Combustion Efficiency that expresses a measure of the Flue Gas Losses, is Steady-State condition, as a percentage of the Combustion Power referred to the fuel lower heating value .

In the European Standard EN 15316-4-1 [37] (see Paragraph 5.1.1) is proposed the following definition for the flue gas losses: it is called "heat loss through chimney when the burner is on" $P_{ch,on}$ " and represents the percentage of the nominal combustion power of the boiler, it calculated as the complement to 100 of the Combustion Efficiency at test conditions (average boiler water temperature $\theta_{gn,test}$ and the boiler room temperature $\theta_{i,test}$). In the Appendix C of the Norm are given the following default values:

Description	$\theta_{gn,test}$ **°C**	$\theta_{i,test}$ **°C**	$P'_{ch,on}$ **%**
Atmospheric boiler	*70*	*20*	*12*
Force draught gas boiler	*70*	*20*	*10*
Liquid fuel boiler	*70*	*20*	*11*
Condensing boiler	*50*	*20*	*6*

In these values is not taken into account the probable condensing heat recovery. It will be introduced in the balance with the Secondary Heat Exchanger.

3.2.4.2 LOSSES THROUGH THE HEAT GENERATOR ENVELOPE

During the operation time, while the burner is on, the heat exchanger (that includes the combustion chamber) will transmit heat directly to the casing and to the air between the heat exchanger and the casing.

In some situations the heat transmitted to the air between the heat exchanger and the casing can be recovered-is the combustion air is picked up by the combustion air fan. Conversely the heat transmitted to the casing is not recovered. This heat is lost through radiation and convection round the envelope and through conduction.

When the burner is turned off the heat generator continues to lose heat through radiation, convection through the envelope, but in a minor amount since the temperature of the envelope surface is lower.

The radiation heat loss through the boiler envelope is given by:

$$q_{rad} = A_{env} * \varepsilon_{env} * \sigma * (T_{env}^4 - T_{blr}^4) \quad (3.43)$$

In which:

A_{env}: *is the boiler envelope surface;*
ε_{env}: *is the emission factor of the envelope (values between 0,1 and 0,9 depending on material and finishing);*
σ: *is the Stefan-Boltzmann constant;*
T_{env}: *is the boiler envelope temperature in K;*
T_{blr}: *is the boiler room temperature in K.*

The convection heat loss through the boiler envelope is given by:

$$q_{conv} = 2{,}6 * A_{env} * (T_{env} - T_{blr})^{1{,}25} \quad (3.44)$$

Heat losses through the envelope of the boiler in on-mode can be seen an determined as the difference between the Combustion Efficiency and the Net Efficiency of the boiler, and can be indicated as a percentage of the input power [28].

The heat losses through the boiler envelope in burner on-mode are influenced by:

- Combustion temperature (type of burner);
- Heat exchanger/burner configuration;
- Boiler water temperature;
- Insulation, material and finishing of the boiler envelope.

In the European Standard EN 15316-4-1[37] the Heat Losses through the generator envelope $\mathrm{P_{gn,env}}$ are calculated from laboratory tests and expressed as percentage of the nominal combustion power of the generator. Its value should be declared by the manufacturer. The following default values are provided in the Appendix C of the Norm:

$$\mathrm{P'_{gn,env}} = \mathrm{A} - \mathrm{B} * \log \Phi_{\mathrm{cn}} \tag{3.45}$$

Where Φ_{cn} is the boiler nominal combustion power [kW] and the parameters A and B depend on the boiler's insulation:

Boiler insulation type	***A***	***B***
Well insulated, high efficiency new boiler	*1,72*	*0,44*
Well insulated and maintained	*3,45*	*0,88*
Old boiler with average insulation	*6,90*	*1,76*
Old boiler, poor insulation	*8,36*	*2,2*
No insulation	*10,35*	*2,64*

The Norm also allows the calculation of $\mathrm{P'_{gn,env}}$ as the difference between the Combustion Efficiency and the Useful (net) efficiency of the boiler at test conditions.

3.2.4.3 STANDING LOSSES

When the burner is turned off the heat generator continues to lose heat through radiation, convection and conduction. This losses are:

- convection trough the chimney;
- radiation, convection and conduction through the boiler envelope.

These losses depend on:

- average boiler water temperature;
- average water flow;
- use of a flue valve;
- insulation, material and finishing of the envelope;
- operating time of the pump;
- use of a pilot flame;

if the pump is not continuously running, but is turned off after each cycle we can consider the following additional parameters:

- heat capacity of the heat generator;

- operating time of the pump after the burner is turned off;
- boiler operation periods over the day;

Pump continuously running:

The standing losses are determined according the EN 303 standards, in which are described the methods to measure the heat lost while the burner is turned off.
From this value coming from test bed is possible to determine the real standing loss by a corrective formula that takes into account the real average water temperature in the boiler and the real boiler temperature.

Pump switches off 10 minutes after each burning cycle:

In principle the standing losses will be lower if the pump is switched off after some time the burner is turned off, because the appliance is not kept continuously at a certain temperature but is allowed to cool down.
In this situation the heat capacity of the generator determines the amount of energy stored in the generator and the amount that can be lost.
The operating time of the pump after the burner is switched-off achieves that some of the stored energy in the boiler mass (heat exchanger) is transferred to the hydronic system.
During an operating period the radiation and convection losses depend on the average appliance temperature.

In the European Standard EN 15316-4-1 the standing losses are the sum of the Losses through the generator envelope $P_{gn,env}$, discussed above, and the Heat Losses through the chimney with the burner off,$P_{ch,off}$.the last one of them takes into account the stack effect of the chimney, which causes a flow of cold air through the boiler. $P_{ch,off}$is expressed as a percentage of the nominal combustion power. The following default values are given in Appendix C of the Norm:

Description	$P'_{ch,off}\%$
Liquid fuel or gas fired boiler with the fan before the combustion chamber and automatic closure of air intake with burner off: *Premixed burners*	*0,2*
Wall mounted, gas fired boiler with fan and wall flue gas exhaust	*0,4*
Liquid fuel or gas fired boiler with the fan before the combustion chamber and no closure of air intake with burner off:	
Chimney height < 10 m	*1,0*
Chimney height > 10 m	*1,2*
Atmospheric gas fired boiler:	
Chimney height < 10 m	*1,2*
Chimney height > 10 m	*1,6*

3.2.4.4 PRE-PURGE

For safety reasons, before the burner is switched-on, a certain amount of cold air (ambient air) is sent to the combustion chamber to remove the exhaust gases that could still be there since the last combustion phase. According to prEN 13836 the pre-purge period must be of 30 seconds with an air flow corresponding the nominal boiler load. Obviously this operation determines a heat loss, since the cold air will cool down the boiler mass.
Is possible a rough calculation of this heat loss by using the formula:

$$Q_{loss;purge} = t_{purge,bf} * \varphi_{fan} * \rho_{air} * c_{air} * \Delta T_{air,avg} \tag{3.46}$$

Where:

$Q_{loss;purge}$: *is the energy loss per burning cycle caused by pre-purge and after-purge;*
$t_{purge,bf}$: *is the pre-purge time in [s];*
φ_{fan}: *is the air flow in [m^3/s];*
ρ_{air}: *is the air density in [kg/m^3];*
c_{air}: *is the specific heat of air in [J/kgK].*

3.2.4.5 AFTER PURGE

The EN standards prescribe also an after-purge, at the end of each cycle, of around 10 seconds. This is imposed for safety reasons, i.e. removing residual ignitable products from the combustion chamber. Unlike the pre-purge, the after-purge is a beneficial to transfer the residual heat of the burner and the heat exchanger to the boiler water, since the flue gases are at a higher temperature than water.
All in all, the after-purge could be neglected from an energy balance.
Pre Purge and After Purge phase can be regarded as components of the Standing Losses.

3.2.4.6 START-STOP LOSSES

It shows that , depending on the heat load and the heat capacity of the boiler, is required some time before the boiler system can reach the steady state regime.
During the start-up time there are the most emission of fuel and other emissions, causing a fuel loss of about 1,5%[28].After the heating up of the boiler is reached the thermal equilibrium, and from this moment the boiler is working with the steady state efficiency as defined in EN 303.

3.2.4.7 CYCLING LOSSES

A boiler starts to cycling when the energy input is too high for the heat output realized by boiler water flow. Cycling occurs especially with boilers characterized by low water content.
For example, considering a heat generator connected with a radiator system with thermostatic valves. When several thermostatic valves are closed, the heat demand decreases and the resistance

in the heating system increases, causing a higher flow speed of the boiler water. As a result the boiler return temperature will be higher. The consequence of this is that in condensing boilers, in cycling regime, the part load efficiency will be lower than measured according to test standards (EN 303) because a higher return water temperature increases the flue gas losses and reduces the condensation potential.

3.2.5 SECONDARY HEAT EXCHANGER

If the heat generator is a condensing boiler in the representation of it we introduce a secondary heat exchanger. The function of the secondary heat exchanger is to further cool the flue gases to a temperature level where most of the latent heat can recuperated, recovering at the same time the sensible heat remaining in the flue gases. This could be seen as a reduction on the above mentioned energy losses localized at the Primary Heat exchanger.

In the reality this secondary heat exchanger is no more than a little extension of the primary heat exchanger in the case of integrated boilers with flat burner. In the case of cylindrical burner and a spiral-tube heat exchanger, the Secondary Heat Exchanger could be really a secondary spiral.

In most cases the Secondary Heat Exchanger is a flue-gas/boiler water heat exchanger. In some oil-fired burners the secondary heat exchanger is a flue gas/combustion air heat exchanger.

The EN standards identify the possibility of condensation when the boiler return water is around 30°C, and in this condition is assumed a 90% of the latent heat recovered. In reality the average return water temperature is around 45°C, that leads a latent heat recovery around the 50%.

3.2.5.1 CONDENSATION TECHNOLOGY

In condensing boiler, the highest level of water vapor condensation is required inside the boiler. This requires water to return from the plant at rather low temperatures and well-sized and thermally enhanced heat transfer surfaces so that the boiler size is not increased with respect to a traditional one with the similar capacity, arriving at a temperature difference even lower than 10 ◦C between the exhaust and the return water.

The main innovation introduced by the Condensing technology is allowing a cooling of flue gases temperature in the boiler that may trigger the condensation of vapor content in them.

This requires solving the following problems:

- the inside of the boiler requires different and more expensive materials, from aluminum to stainless steel that bear carbonic acid corrosion.
- the flue gases are cooled till saturation for all the route till the exhaust. Therefore, condensation must be expected all along the chimney as the smokes probably get cooled along this route. That is why it is necessary to provide the tubing of the chimney all along the route before the final exhaust all along the route before the final exhaust.

- the low temperature exhaust gives a very poor draft: pressurised burning system is requested that must be equipped with a burner fan for the combustion air.

The main rule is: the lower is the temperature to which the smokes are cooled,the higher the latent heat fraction of the smokes made available by a condensing boiler.

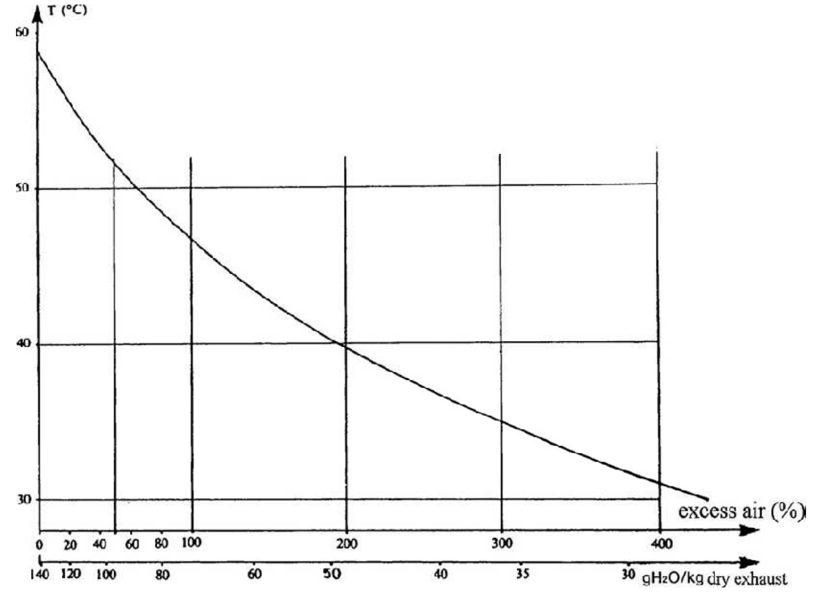


Fig.3.36Dew point temperature of methane combustion products vs. excess air.

The graph represented in Fig. 3.36gives the dew point temperature of methane combustion products as a function of excess air. The excess air dilutes the water vapor content in the combustion products and hence tends to lower the dew temperature.

In the graph is added the horizontal axis, representing the saturation water content related to the dry combustion products at different temperatures.

This value allows through the computation of the difference between the initial water content (stoichiometric water production in combustion, i.e.0% of excess air) and the water content at a given temperature to evaluate the condensate amount formed by cooling the combustion products to that temperature.
If the exhaust gas is cooled to 50 ◦C, its water content is about 85 g/kg $_{\text{dry smokes}}$, thus the condensate is given by: (140÷ 85) = 55 g/kg $_{\text{dry smokes}}$
The change of water phase from vapor to liquid releases thermal energy: the condensing heat. It is a function of temperature, but at usual building temperatures it can be considered as 2500 kJ/kg
The low methane heat value is about 35 MJ/m^3, the combustion of 1 m^3 of methane produces 1,5 kg of water vapor, that if is all condensed, provides a condensing heat of about 3750 kJ/m^3 ,that correspond to around the 10% of the low heating value. This 10% corresponds also to the difference between the higher and lower heating value of methane.
So, if the combustion products are cooled till 50°C about 40%of the water vapor is condensed (55/140). If HHV and LHV differby about 4 MJ/m3 a recovery of about 1.6 MJ/m3 is obtained. The best traditional boiler can give about 32 MJ/m3; therefore, theefficiency improvement for the condensation is 5% (1.6/32).

However,the cooling of the exhaust to a lower temperature means asensible heat recovery. This is shown in Fig.3.37, which illustrateshow the sensible loss as a function of the air excess is for differentexhaust temperatures. A 50 ◦C exhaust means a 2% loss against a6% loss for a 140 ◦C exhaust for a 20% air excess.

Therefore, the possible efficiency improvement for the considered condensing boiler operation is 9%.

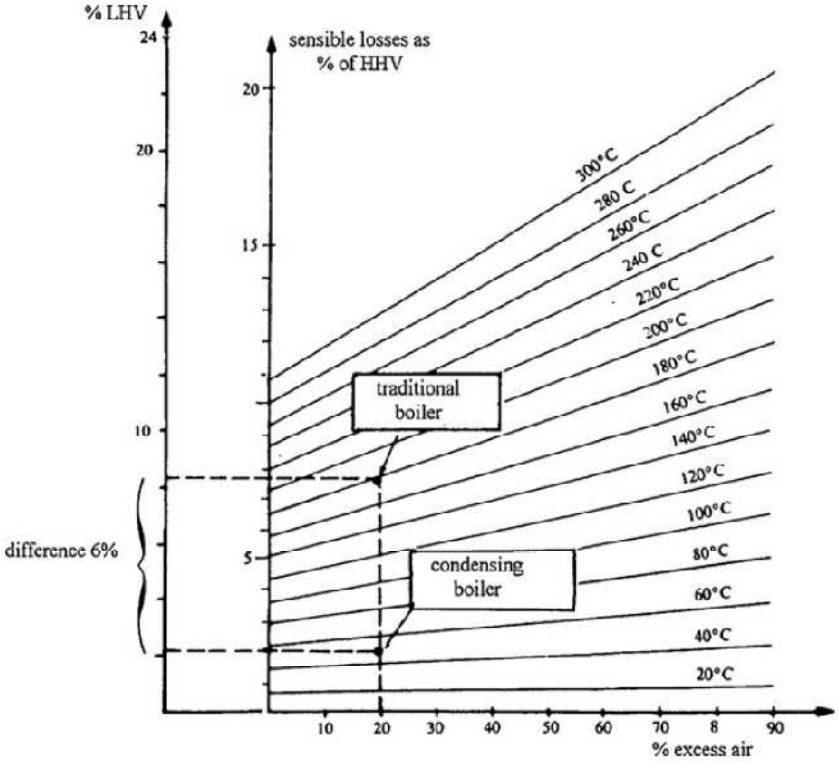


Fig.3.37 Sensible heat losses and flue gas temperature

If the exhaust were cooled till 40 ◦C, the water content would be lowered to 50 g/kg $_{\text{dry smokes}}$ with a condensation of 90 g/kg $_{\text{dry smokes}}$ and the latent heat recovery would be raised to 65%, that is 2.6 MJ/m^3 with an overall efficiency increment of 8%.

If you add to this effect the 4% recovered as sensible heat, the overall improvement with respect to the best traditional boiler is 12%. If the best traditional boiler efficiency is 92%, the condensing boiler efficiency is 104%.

In Fig.3.38 are reported three different curves: the boiler efficiency (on HHV), η, the exhaust gasses temperature, t, and the condensed production, G, as a function of the boiler water return temperature.

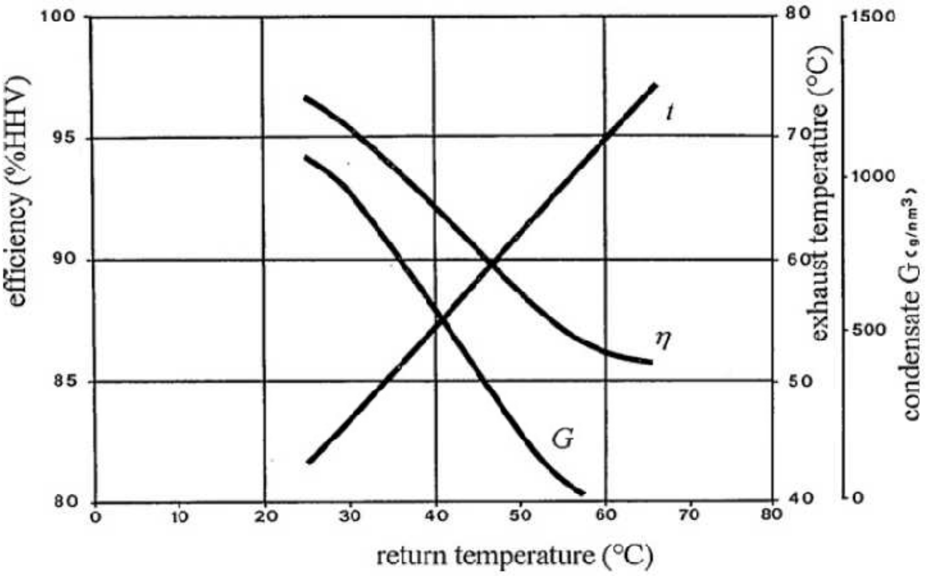


Fig.3.38 Influence of the return water temperature

Whereas the traditional boilers practically do not depend on that temperature, condensing boiler begins slowly to improve its efficiency on reducing return temperature. When an inflexion point is met in the curve, the dew point temperature is reached and from there the efficiency improvement is very rapid. The inflexion point is met at a temperature lower than 60 °C (the dew temperature for a stoichiometric combustion of methane).

A return water temperature between 45° and 55 °C is requested according to the heat exchange effectiveness of a given boiler: in fact the heat exchange between the combustion products and the water requires a temperature difference that depends on the extension and quality of the heat transfer surfaces [38].

In the case of oil-fired condensing boiler all what was said above is still true, only the numerical values are changing.

For example, the dew point temperature is around 47°C for a 20% excess air.

The average difference between the Higher Heating Value and Lower is around 6%.

3.2.6 TERTIARY HEAT EXCHANGER

The Tertiary Heat Exchanger is considered when is present in the heat generator configuration, a dedicated place in which the combustion air is pre-heat with the exiting flue gases. This heat exchange could take place in concentric flue/air ducts or in a separated plate heat exchanger.

In the case of oil-fired boiler the pre-heating of the combustion air is also functional to enhance the fuel liquid droplet vaporization.

In the tertiary heat exchanger is also possible the recuperation of another part of the latent heat of the flue gases [28].

CHAPTER 4

EFFICIENCY TESTS

A heat generator, with the combustion process, transfers energy from the fuel to the heat carrier fluid (water) which will be used downstream the heat generator to achieve different scopes, such as space heating, domestic hot water production or other industrial processes.
We can represent the heat generator's power flows as follows:

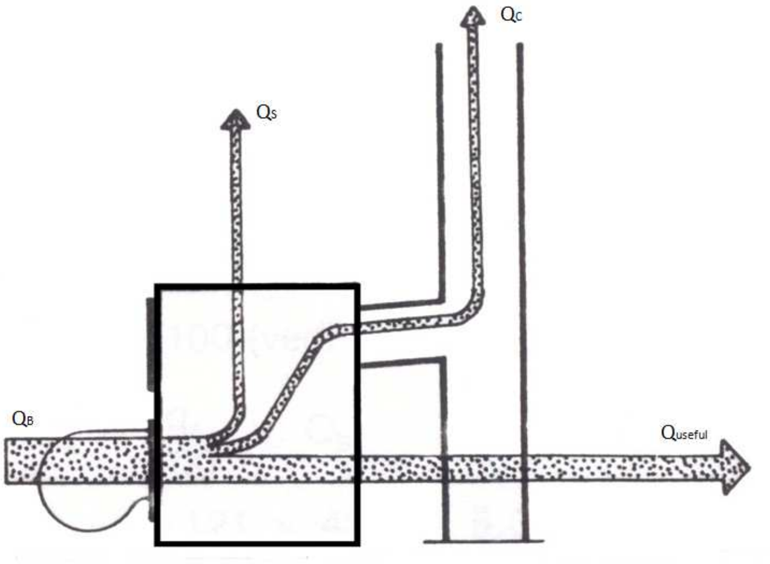


Fig. 4.1 Heat Generator's feat flows

Writing the First Principle balance:

$$Q_B = Q_C + Q_S + Q_{useful} \qquad (4.1)$$

Where:

Q_B: *is the Burner Heating Power, i.e. the energy amount, in the time unit, provided by the entering fuel. It is obtained by the fuel's lower heating power multiplied by the fuel flow rate:*

$$Q_B = \dot{m}_B * LHV \qquad (4.2)$$

Where:

$\dot{m}_B$: *is the fuel flow rate;*
LHV: *is the fuel lower heating value;*

Q_S: *is the Casing Heat Loss, i.e. the energy amount, in the time unit, transferred to the environment in which the heat generator is working, due to the temperature difference between the boiler's case and environment air;*

Q_C: *is the Chimney Heat Loss, i.e. sum of the energy amount, in the time unit, transferred to the atmosphere by the combustion products which are released at high temperature,* Q_A *and the energy loss due to incomplete combustion of the fuel,* Q_U*:*

$$Q_C = Q_A + Q_U \tag{4.3}$$

Q_{useful}*: is the Useful Heating Power, i.e. the energy amount, in the time unit, received by the heat carrier fluid.*

The Useful Heating Power value can be obtained with a direct approach, by measuring the actual thermal power received by the heat carrier fluid:

$$Q_{useful} = \dot{m}_w c_w (T_2 - T_1) \tag{4.4}$$

Where:

$\dot{m}_w$: is the water flow rate;
c_w. is the water specific heat;
$T_2 - T_1$: is the temperature difference between supply and return water.

or through an indirect approach, i.e. subtracting from the power inlet the power losses :

$$Q_{useful} = Q_B - Q_C - Q_S \tag{4.5}$$

The last approach is affected by strong uncertainness due to difficulties in measuring the parameters required.

The ratio between the Useful Heating Power and the Burner Heating Power is defined as Useful Efficiency of the heat generator, η_u:

$$\eta_u = \frac{Q_{useful}}{Q_B} \tag{4.6}$$

The Useful Efficiency qualifies the heat generator's ability in converting the fuel energy into useful thermal energy transferred to the heat carrier fluid.

4.1 USEFUL EFFICIENCY FOR CONDENSING BOILERS

As was discussed in the paragraph 2.5.2, in the past was not allowed any vapor condensation in the combustion products' path from the burner to the chimney outlet. Hence the representative parameter of the "energy source" of the system was the fuel's lower heating power. For this reason the denominator of the Useful Efficiency's formula is the fuel flow rate multiplied by its lower heating value.

In the case of Useful Efficiency evaluation for a condensing boiler, the "extra energy" recovered by the flue gas vapor condensation is already computed by an higher value of Useful Heating Power, if we consider a constant Burner Heating Power, or, from the opposite point of view, by a lower value of the Burner Heating Power, considering a constant Useful Heating Power. Hence the formula to assess the Useful Efficiency (4.6) doesn't need any modification when is considered a condensing boiler. The only change that will be seen is a greater value of the Useful Efficiency in the condensing case, with values even higher than unity.

4.2 USEFUL HEATING POWER AND EFFICIENCY ACCORDING THE UNI EN 304

In the European norm UNI EN 304 "Heating boilers- Test code for heating boilers for atomizing oil burners"[39]sets the roles in order to execute tests on a heat generator. In this regulation, among the other topics, we can find the procedures to assess:

- Boiler's Useful Heating Power and Efficiency;
- Nominal Useful Heating Power;
- Boiler's Useful Efficiency;
- Standing Loss.

4.2.1 FUEL

Tests must be made by using commercial heating gasoil, which can be measured by weight measurements or volumetric measurements.
If the fuel's lower heating value is not determined by sing a bomb calorimeter, and is unknown the ultimate analysis, the following default value should be taken:

Lower heating value:$H_u = 42{,}689\,\frac{MJ}{kg}$

Chemical mass fractions:

- Carbon $c = 0{,}86\ [\frac{kg}{kg}]$;
- Hydrogen $h = 0{,}136\ [\frac{kg}{kg}]$;
- Sulfur $s = 0{,}003\ [\frac{kg}{kg}]$;
- Nitrogen n=0
- Oxygen o=0

4.2.2 DETERMINATION OF THE COMBUSTION PARAMETERS

Stoichiometric Oxygen Requirement:

$$O_{min} = c * 1{,}86 + s * 0{,}70 + h * 5{,}55 - o * 0{,}7 \qquad [m^3/kg] \qquad (4.7)$$

Dry Air Requirement:

$$L_{min} = \frac{O_{min}}{0{,}21} \qquad [m^3/kg] \qquad (4.8)$$

Stoichiometric Combustion Products on dry basis:

$V_{Atr,min} = c * 1,85 + s * 0,68 + 0,8 * n + O_{min} * \frac{0,79}{0,21}$ [m^3/kg] (4.9)

Stoichiometric Carbon Dioxide percentage on dry basis in the Combustion Products:

$CO_{2,max} = \frac{c * 1,85}{V_{Atr,min}}$ [m^3/m^3] (4.10)

Stoichiometric Sulfur Dioxide percentage on dry basis in the Combustion Products:

$SO_{2,max} = \frac{s * 0,70}{V_{Atr,min}}$ [m^3/m^3] (4.11)

Combustion Products Actual Volume on dry basis:

$V_{Atr} = \frac{V_{CO_2} - V_{SO_2}}{(CO_2 + SO_2)_{meas} + CO_{meas}} = \frac{c * 1,85 + s + 0,70}{(CO_2 + SO_2)_{meas} + CO_{meas}}$ [m^3/kg] (4.12)

Water vapor content in the Combustion Products:

$V_w = h * 11,1$ [m^3/kg] (4.13)

4.2.3 AIR INDEX AND AIR EXCESS

The Air Index, λ, is given by:

$$\lambda = \frac{AIR\ VOLUME\ INTRODUCED\ IN\ THE\ BURNER}{L_{min}}$$

And is obtained measuring the $(CO_2 + SO_2)$ and the CO content or the O_2 content in the Combustion Products.

The Air Excess, e, is given by:

$e = (\lambda - 1) * 100$ (4.14)

Thus, if the Combustion Products don’t contain other unburned components:

$\lambda = 1 + \left[\frac{CO_{2,max} - SO_{2,max}}{(CO_2 + SO_2)_{meas} + CO_{meas}} - 1\right] \frac{V_{Atr,min}}{L_{min}}$ (4.15)

Or, if is measured the O_2content in the Combustion Products:

$\lambda = 1 + \frac{V_{Atr,min}}{L_{min}} * \frac{O_2}{21 - O_2}$ (4.16)

4.2.4 DIRECT AND INDIRECT METHODS

The regulation provides two calculation methods in order to assess the Useful Efficiency of the heat generator: a Direct Method and Indirect Method. The last one is considered just as a verification procedure for the Direct Method. In fact only the Direct Method is recognized as reliable in this regulation.

4.2.4.1 DIRECT METHOD

Consist on a direct evaluation of the heat transferred from the heat generator to the heat carrier fluid (water), i.e. the Useful Heating Power. This is done by measuring the water temperature increase between the upstream and the downstream of the heat generator, and the water mass flow rate, as was written in equation (4.4).
The measurement of this quantity can be evaluated or directly on the heat generator (short circuit), or through a heat exchanger (heat exchanger circuit).

1- *Useful Heating Power direct measurement on the heat generator*:is obtained by measuring the cold water mass flow rate entering the boiler's loop and the water temperature increment between water's inlet and outlet. Another way is to measure the water mass flow rate circulating in the boiler's loop and its temperature increase.

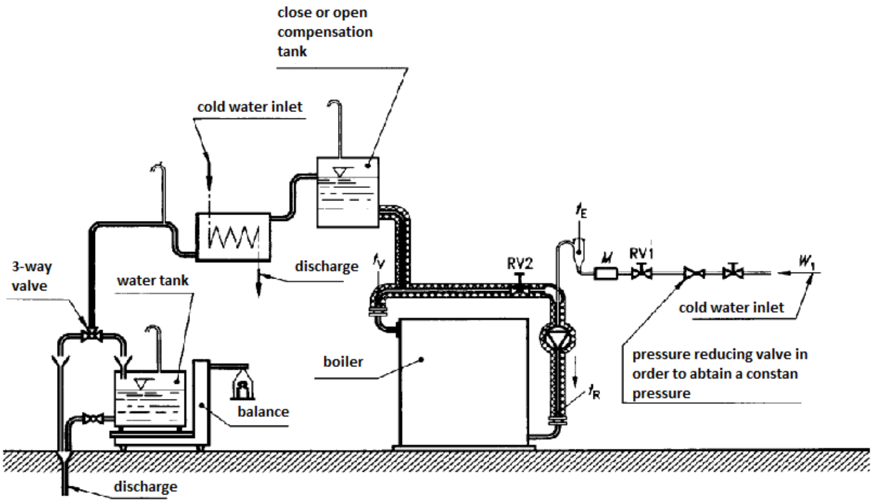


Fig.4.2 Short circuit

2- *Useful Heating Power indirect measurement through an heat exchanger*: thermal power provided by the heat generator is transferred to the cooling water through an heat exchanger. The heat power that this receives is determined by measurements of the cooling water mass flow rate and of its temperature increase. This method introduces some thermal loss (at the heat exchanger) that have to be evaluated in a preliminary test. The Useful Heating Power is the sum of the heat that the heat exchanger receives and the circuit thermal loss.

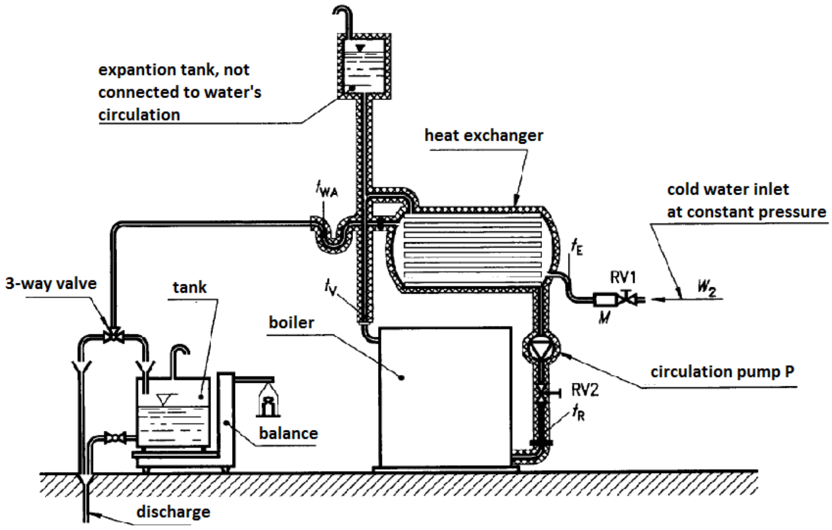


Fig.4.3 Heat exchanger circuit

The Useful Heating Power is determinate by using the following procedure:

a- If is adopted the short circuit:

$$Q_{useful} = W_1 * c_{w1} * (t_V - t_E) \qquad \text{[W]} \tag{4.17}$$

Where:

W_1: *is the cold or hot water mass flow rate [kg/s]*
c_{w1}: *is the thermal capacity on mass basis at the average water temperature [J/(kg K)]*
t_V : *cold water temperature [°C]*
t_E: *hot water temperature [°C]*

b- If is adopted the heat exchanger circuit:

$$Q_{useful} = W_2 * c_{w2} * (t_{wA} - t_E) + Q_V \qquad \text{[W]} \tag{4.18}$$

Where:

W_2: *is the cooling water mass flow rate[kg/s]*
c_{w2}: *is the thermal capacity on mass basis at the average water temperature [J/(kg K)]*
t_{wA}: *is the temperature of the cooling water [°C]*
t_E: *is the hot water temperature [°C]*
Q_V: *is the circuit's thermal loss[W]*

The Burner Thermal Power $Q_B$is calculated as follows:

$$Q_B = B * H_u \qquad \text{[W]} \tag{4.19}$$

Where:

B: *is the fuel's mass flow rate [kg/s];*
H_u: *is the fuel's Lower Heating Value [J/kg];*

The Boiler Efficiency is given by the ratio:

$$\eta_k = \frac{Q_{useful}}{Q_B} \tag{4.20}$$

4.2.4.2 THE INDIRECT METHOD

Looking at Fig. 9.1, we can see that we could determine the Useful Heating Power starting from the Burner Heating Power and subtracting from its value the Power Losses amounts that occur in the system. Hence the Useful Efficiency could be obtained by:

$$\eta_k = 1 - q_A - q_U - q_S \tag{4.21}$$

Where:

q_A: *is the loss due to the combustion products sensible heat (relative value, referred to the Burner Heating Power);*
q_U: *is the loss due to the incomplete combustion (relative value, referred to the Burner Heating Power);*
q_S: *is the loss due to heat transfer between the heat generator and the ambient (relative value, referred to the Burner Heating Power).*

- *The Combustion Products Loss,*q_A

If the combustion air and the fuel are introduced in the system at ambient conditions:

$$q_A = \frac{V_A * C_{pm,Atr}}{H_u} * (t_A - t_L) \tag{4.22}$$

Where :

V_A: *is the Combustion Products' volume per kg of fuel burned;*
t_A: *is the Combustion Products' temperature;*
t_L: *is the environment air temperature;*
$C_{pm,Atr}$: *is the Combustion Products' mass thermal capacity at the average temperature between* t_A *and* t_L *;*
H_u: *is the fuel's lower heating value.*

The Dry Combustion Product's Volume could be determined in a general way as follows, in the case of incomplete combustion and with air excess:

$$V_{Atr} = V_{CO_2} + V_{SO_2} + V_{CO} + V_{N_2} + V_{O_2} = \frac{(V_{CO_2}+V_{SO_2})+V_{CO}}{(CO_2+SO_2)_{meas}+CO_{meas}} \quad \text{[m}^3\text{/kg]} \tag{4.23}$$

Where:

V_{CO_2}: *is the carbon dioxide volume [m^3/kg];*

V_{SO_2}: *is the sulfur dioxide volume[m^3/kg];*
V_{CO}: *is the carbon oxide volume[m^3/kg];*
V_{N_2}: *is the nitrogen volume[m^3/kg];*
V_{O_2}: *is the oxygen volume[m^3/kg];*
$(CO_2 + SO_2)_{meas}$: *is the measured percentage of* CO_2 *and* SO_2 *in the dry Combustion Products [m^3/m^3]*
CO_{meas}*is the measured percentage of CO in the dry Combustion Products[m^3/m^3];*

In the combustion of fuel oil, by neglecting the value of CO_{meas}, we can write:

$$V_{Atr} = V_{Atr,min} * \frac{100}{100-4{,}76*O_2} \qquad [\mathrm{m^3/kg}] \tag{4.24}$$

Where $V_{Atr,min}$is determined by equation (9.12).

If we split the contribution of the water vapor from the one from the dry Combustion Products, equation (9.22) becomes:

$$q_A = \left(V_{Atr} * C_{pm,Atr} + V_w * C_{pm,H_2O}\right) * (t_A - t_L) * \frac{1}{H_u} \tag{4.25}$$

Where:

$C_{pm,Atr}$: *is the dry Combustion Products' average mass thermal capacity between* t_A*and* t_L*;*
C_{pm,H_2O}: *is the water vapor's average mass thermal capacity between* t_A*and* t_L*;*

The above parameters cold be calculated by using the following formulae, valid for $t_A < 500°C$:

$$C_{pm,Atr} = 0{,}361 + 0{,}008\left[\frac{t_A}{1000°C}\right] + 0{,}034\left[\frac{t_A}{1000°C}\right]^2 + \left[0{,}085 + 0{,}19\left[\frac{t_A}{1000°C}\right] - 0{,}14\left[\frac{t_A}{1000°C}\right]^2\right]\left[\frac{CO_2}{100\%}\right] + \\ + \left[0{,}3\left[\frac{t_A}{1000°C}\right] - 0{,}2\left[\frac{t_A}{1000°C}\right]^2\right]\left[\frac{CO_2}{100\%}\right]^2 \tag{4.26}$$

$$C_{pm,H_2O} = 0{,}14 + 0{,}038\left[\frac{t_A}{1000°C}\right] + 0{,}34\left[\frac{t_A}{1000°C}\right]^2 \tag{4.27}$$

- *Unburned Loss,*q_U

Considering that the only unburned present in the Combustion Products is carbon oxide:

$$q_U = CO * V_{Atr} * \frac{12{,}64}{H_u} \tag{4.28}$$

Where:

CO: is the carbon oxide quantity in the dry Combustion Products [m^3/m^3]
V_{Atr}: *is the Dry Combustion Product's Volume per kg of fuel burned [m^3/kg]*
H_u: *is the fuel's lower heating power[MJ/kg]*

The value 12,64 is the heating value of the carbon oxide [MJ/kg]

- *Casing Loss,* q_s

This heat loss is due to the thermal difference between the heat generator case and the ambient in which the heat generator is situated. The value of this heat loss could be determined by the approximation that follows:

The outer surface of the heat generator is divided into zones (partial surfaces, Q_x) which are characterized by similar temperatures. The heat lost by the partial surface is calculate as follows:

$$Q_x = F_x * \alpha * (t_m - t_L) \qquad [W] \qquad (4.29)$$

Where:

Q_x: *is the heat lost by the partial surface[W]*
F_x: *is the partial surface extension[m^2]*
α: *is the heat exchange coefficient due to radiation and convection [W/m^{2}K]*
t_m: *is the average temperature of the partial surface[°C]*
t_L: *is the ambient temperature[°C]*

The heat exchange coefficient, α , could be chosen by using the following graph:

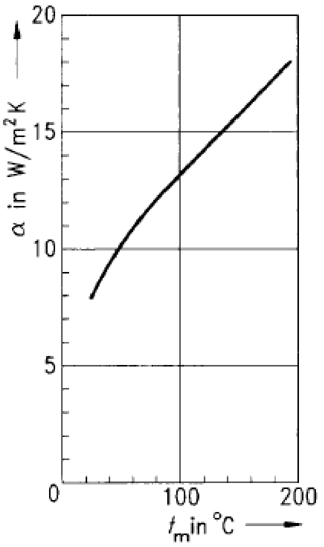


Fig.9.4 Heat transfer coefficient by natural convection and radiation for horizontal and vertical surfaces with ambient temperature $t_L = 20°C$.

The Casing Loss, q_s, is obtained by:

$$q_s = \frac{\sum Q_x}{Q_B} \qquad (4.30)$$

4.2.5 STANDING LOSS DETERMINATION

In period in which the burner is turned off, the heat generator continues to lose heat trough the envelope and the chimney. This happens because of the temperature difference between the boiler components (hot) and the air temperature of the boiler room (cold). This heat loss determines the cooling of the boiler, that will yield an extra requirement of energy to heat-up again the device, when it will be recalled to operate.

The EN 304 provides the experimental method to evaluating the Standing Loss. The procedure proposed is as described in the following.

The boiler is not providing any heat power to the heat carrier (the hydraulic loop is not working), and the boiler's thermostat is set to maintain a temperature difference of 50 K between the average water temperature in the boiler and the ambient temperature.

During the OFF-period of the boiler the circulator has to be switch off. The circulator is turned on by the thermostat together with the burner, and must work until 3 minutes after the burner is switched off.

After the heat up of the boiler, the test period starts when the burner is switched on for the first time. The test period is finishes with the next turning on of the burner. For each test period the fuel consumption must be measured.

The Standing Loss (it is a relative value) is calculated as:

$$q_s = \frac{fuel consumption during the test period}{imposed fuel flowrate during the test period} \tag{4.31}$$

CHAPTER 5

LEGISLATION ON BOILER’S PERFORMANCE

The European Union has elaborated Directives in order to meet the “20-20-20” objectives, i.e. reducing of 20% the greenhouse gas emission, increasing of 20% the renewable energy use and a reduction of 20% of the global energy use, through a more efficient energy use.

The greatest energy saving potential lies in buildings [40]. For this reason in 2002 was promulgate the 2002/91/CE, “Energy Performance of Buildings Directive”[41], then recast in the 2010/31/CE [42]. In this Directive is defined a harmonized framework for a calculation method to assess the energy performance of a building. The performance involves the whole system “envelope-systems”, from space heating to lighting appliances. By using such calculation methods, the Member States will apply minimum requirements on the energy performances of new buildings.

This European Directive proposes, also, the introduction of energy performance labels for dwellings in all European Member States. Such an energy performance label would have to be handed over when a dwelling is built, reconstructed, sold or rented out. This label shows the current energy performance status of the building divided over energy efficiency classes A to G[43].

The European Committee for Standardization, CEN, received the mandate to elaborate the standards for a methodology calculating the integrated energy performance of buildings in accordance with the EPBD.

The main scheme of Standards produced is shown in Fig. 5.1.

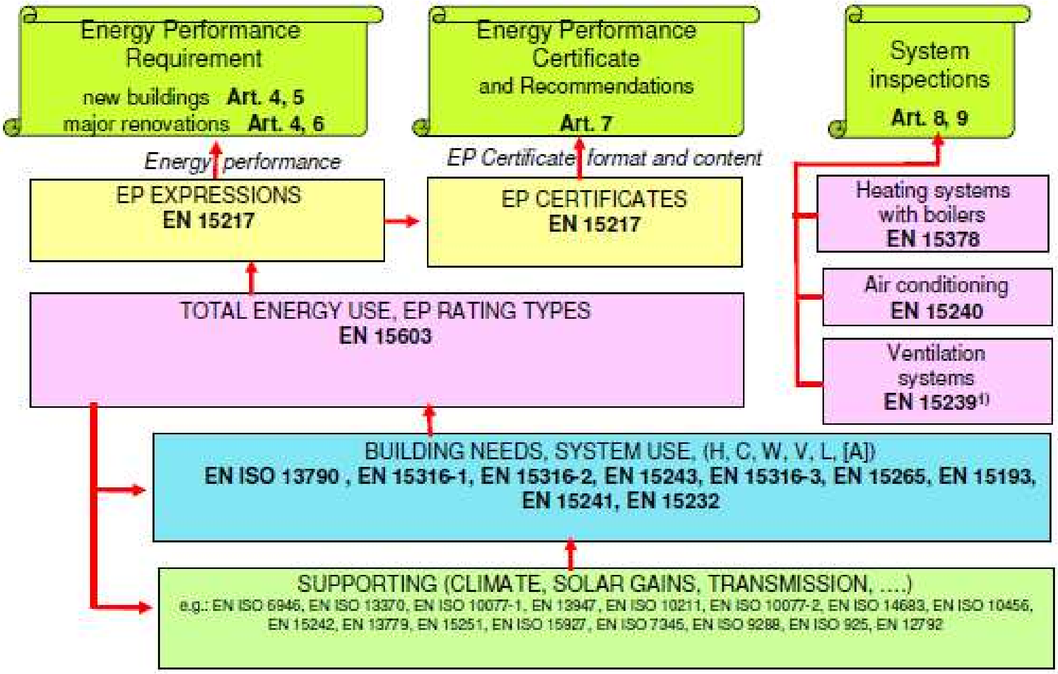


Fig. 5.1 Standards involved in the EPBD

In this wide field the interesting Standard elaborated from CEN is the one involving the heat generation sub-system: prEN15316-4-1,space heating generation systems, combustion systems, that will be discussed in Paragraph 5.2.

Another Directive adopted by the European Union, with the same aim of increasing energy efficiency, is the 2005/32/CE "Energy using Products"(EuP) [44], than recast in the2009/125/CE "Energy related Products" (ErP) [45].

The ErP Directive covers all those products that are energy consumers or that could contribute in somehow to the energy saving. All these products were divided in "Lots", from the boilers for ambient heating (Lot1) to the windows. For each Lot was produced a calculation method to assessing the energy performance, and defined minimum performance levels. The examined Energy related Product can be sold in the European market (by receiving the "CE" marc) only if the prescribed minimum performance level is achieved.

The spirit of this Directive is based on the awareness that the 80% of the environmental impact coming from the life-cycle of a Energy related Product is defined in the design phase of the product. In fact, this Directive is also known as Eco-Design Directive.

In the Paragraph 5.3 will be discussed the calculation method to assess the energy performance of space heating boilers.

5.1 SEASONAL EFFICIENCY

In the Paragraph 2.5, about the Combustion Efficiency and inChapter4 about the Useful Efficiency, were described the heat generator's performance in Stead-State conditions. The Useful Efficiency, that characterizes the boiler's attitude to transform the chemical energy in the fuel into thermal energy of the heat carrier fluid, can be measured at different Load (nominal Load, Part Load), but always in Steady State conditions.

This kind of methods had limitations, that determine a significant gap between announced performance and actual performance. A heat generator works for most of the time at partial load, and if a heat generator has good performance-in terms of efficiency- at the nominal power, could behave in a different way when it is working in other conditions.

In these EPBD and Eco-Design Directives is considered a more accurate measurement : the "seasonal efficiency" that takes into account the actual energy provided by the heat generator and the fuel and electricity consumption that allows the device to provide the useful effect (heating), over the heating season. This means that the performance of the heat generator are considered in the "real use" of the device.

$$\eta_s = \left[\frac{\mathrm{heatprovidedbythegenerator}}{\mathrm{totalenergyconsumption}}\right]_{\mathrm{overaperiod}} \tag{5.1}$$

The major changes between seasonal and nominal calculation are the use of several different working points instead of one, and integration in the calculation of operation at partial capacity instead of full capacity. Since most systems operate most of the time under a partial load, the new methodology gives a better indication of expected real-life performance. The new seasonal efficiency calculation will also take into consideration the power consumed by devices in auxiliary modes like standby mode.

When we evaluate the seasonal efficiency of the heat generator, in order to assess the building energy consumption, the load profile of the space heater depends on the actual heat demand. This is in the scope of the EPBD, discussed later.

When we want evaluate the seasonal efficiency of the heat generator, considered as a single product, before to be introduced in a heating system, the actual load profile is unknown, thus is introduced a fictitious representation of the load profile that a space heater should encounter in a typical heating system. This is in the scope of the ErP Directive.

5.2 EUROPEAN STANDARD: EN 15316-4-1

The European Standard EN 15316-4-1 [42], coming from the EPBD Directive, presents methods for the calculation of the additional energy requirements of a heat generation system in order to meet the distribution and/or storage sub-system demand. The calculation is based on the performance characteristics of the products given in product standards and on other characteristics required to evaluate the
performance of the products as included in the system.

This method can be used for the following applications:

- judging compliance with regulations expressed in terms of energy targets;
- optimization of the energy performance of a planned heat generation system, by applying the method to several possible options;
- assessing the effect of possible energy conservation measures on an existing heat generation system, by calculating the energy use with and without the energy conservation measure.

The user shall refer to other European Standards or to national documents for input
data and detailed calculation procedures not provided by this standard.

For the first method the considered calculation period is the heating season. The performance calculation is based on the data related to the boiler directive. The operation conditions taken into account (climate, distribution system connected to the generator, etc.) are approximated by typology of the considered region and are not case specific. If this method is to be applied, an appropriate national annex with the relevant
values shall be available.

The second method is also based on the data related to the boiler directive, but supplementary data are needed in order to take into account the specific operation conditions of the individual installation. The considered calculation period can be the heating season but may also be a shorter period (month, week, or the operation periods according to EN ISO 13790). The method is not limited and can be used with the default values given in informative annex B.

The third method distinguishes in a more explicit way the losses of a generator which occurs during boiler cycling (i.e. combustion losses). Some of the parameters can be measured on site. This method is well adapted for existing buildings. The calculation method to be applied is chosen as a function of the available data and the objectives of the user.

The first two methods explicitly take the measured efficiency data from the Boiler Efficiency Directive BED at 30% and 100% load measured according to the EN product standards, but also the defaults of the third method are taken from the Boiler Directive.
For the second and third method, the declared value for ‘losses at 0% load’ is assumed available.

5.2.1 BOILER CYCLING METHOD

This method well reflects the ideal analysis made in Paragraph 3.2.

The calculation is based on a subdivision of the boiler operating time between burner on-mode and burner off-mode (ton and t_{off}). The time in on-mode is the time the fuel valve is actually open, pre- and post ventilation are not considered. Heating losses through chimney are taken separately for these two time periods ($Q_{ch,on}$ / $Q_{ch,off}$).

Auxiliary energy is considered separately for components before (typically burner fan) and after (pump) the burner (W_{br} and W_{af}), each with their own recoverable energy coefficient (k_{br} and k_{af}) and thereby recovered energy (Q_{br} = kbr * W_{br} and $Q_{af}=k_{af} * W_{af}$).

The method then continues to detail each of the terms in the equation, basically by measuring several parameters under test conditions. In practice, these values are not provided by manufacturers and in order to apply this method the user would be required to do the measures him/herself (i.e. through a laboratory). Once the measured
values are available under test conditions, the method proceeds to calibrate these test values for the actual conditions using a number of equations.
When the input data are not available (i.e. performance at test conditions), and this provides interesting background information, the method proposes a number of default values in Annex C.

As was said, the method considers two operating times:

- Burner ONtime (see Fig. 5.2) flue gas losses and envelope losses are considered here;
- Burner OFF time (stand-by) (see Fig.5.3) draught losses and envelope losses are considered here.

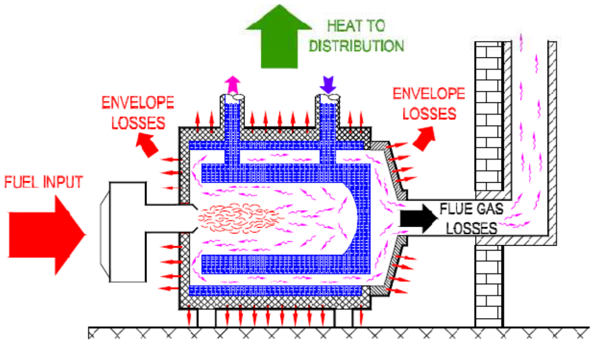


Fig.5.2 Burner ONtime

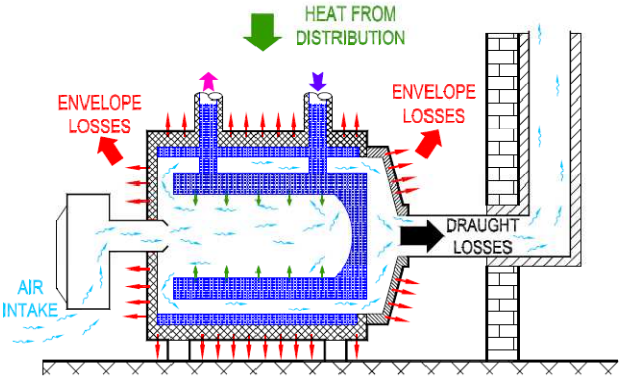


Fig. 5.3 Burner OFF time

From this time division is defined the load factor, FC, as:

$$\mathrm{FC} = \frac{\mathrm{t_{ON}}}{\mathrm{t_{gn}}} = \frac{\mathrm{t_{ON}}}{\mathrm{t_{ON}+t_{OFF}}} \tag{5.2}$$

This value can be measured on an existing heat generator, or calculated from the $\mathrm{Q_{g,out}}$.
For each calculation interval the burner-on time is calculated depending on the system heat demand $\mathrm{Q_{g,out}}$. For both on and off-time, losses are calculated using boiler characterization data, correction formulas and tabulated parameters to take into account actual operating conditions .

The required amount of fuel is obtained according to combustion power and calculated on-time.

The basic energy balance of the generation sub-system is:

$$\mathrm{Q_{g,out} = Q_{cn} + Q_{br} + Q_{af} - Q_{ch,on} - Q_{ch,off} - Q_{gn,env}} \tag{5.3}$$

Where:

The heat losses through the chimney with the burner on,$\mathrm{Q_{ch,on}}$, are:

$$\mathrm{Q_{ch,on}} = \frac{\mathrm{P_{ch,on}}}{100} * \Phi_{\mathrm{cn}} * \mathrm{t_{ON}} \tag{5.4}$$

Where:

$P_{ch,on}$: *is the actual specific heat loss through the chimney when the burner is on. Its value is calculated from the test value,*$P'_{ch,on}$, *referred to test temperature values, and then corrected to consider the actual boiler conditions:*

$$\mathrm{P_{ch,on}} = \left[\mathrm{P'_{ch,on}} + \left(\theta_{\mathrm{gn,w,av}} - \theta_{\mathrm{gn,test}}\right) * \mathrm{f_{cor}}\right] * FC^n \tag{5.5}$$

$\theta_{gn,w,av}$: *is the average water temperature in the boiler at the actual conditions;*
$\theta_{gn,test}$: *is the average water temperature in the boiler at test conditions;*
f_{cor}: *is a correction factor given in the procedure;*
FC^n: *takes into account the reduction of losses with high intermittencies, due to a lower average temperature of the flue gas (higher efficiency at start);*
Φ_{cn}: *is the combustion power of the generator.*

The heat losses through the chimney with the burner off,$Q_{ch,off}$, are:

$$Q_{ch,off} = \frac{P_{ch,off}}{100} * \Phi_{cn} * t_{OFF} \tag{5.6}$$

Where:

$P_{ch,off}$: *is the actual specific heat loss through the chimney when the burner is off. Its value is calculated from the test value,*$P'_{ch,off}$*, referred to test temperature values, and then corrected to consider the actual boiler conditions:*

$$P_{ch,off} = P'_{ch,off} * \frac{\theta_{gn,w,av} - \theta_{i,gn}}{\theta_{gn,test} - \theta_{i,test}} * FC^{p} \qquad (5.7)$$

$\theta_{i,gn}$: *is the actual temperature of the boiler room;*
$\theta_{i,test}$: *is the temperature of the test room;*
p: *is an exponent that depends on the heat capacity of the boiler.*

The heat losses through the generator envelope are:

$$Q_{gn,env} = \frac{P_{gn,env}}{100} * \Phi_{cn} * (t_{OFF} + t_{ON}) \qquad (5.8)$$

Where :

$P_{gn,env}$: *is the actual the actual total generator envelope heat losses and is calculated from test conditions:*

$$P_{gn,env} = P'_{gn,env} * k_{gn,env} * \frac{\theta_{gn,w,av} - \theta_{i,gn}}{\theta_{gn,test} - \theta_{i,test}} * FC^{m} \qquad (5.9)$$

$k_{gn,env}$: *is the reduction factor taking into account the location of the generator that determines the recovery of heat losses as a reduction of total losses.*

The total auxiliary energy required by the generation sub-system is given by:

$$W_g = \sum \Phi_{W,i} * t_{ON,i} \qquad (5.10)$$

The auxiliary energy converted to heat and recovered to the system before the generator is:

$$Q_{br} = \sum \Phi_{br,i} * t_{ON,i} * k_{br,i} \qquad (5.11)$$

Where:

$\Phi_{br,i}$: *is the electrical power consumption of auxiliary appliances (before the generator);*
$k_{br,i}$: *is the recovery factor of* $\Phi_{br,i}$

The auxiliary energy converted to heat and recovered to the system after the generator is:

$$Q_{af} = \sum \Phi_{af,i} * t_{ON,i} * k_{af,i} \qquad (5.12)$$

Where:

$\Phi_{af,i}$: is the electrical power consumption of auxiliary appliances (after the generator);
$k_{af,i}$: is the recovery factor of $\Phi_{af,i}$

Calculation procedure for a single stage generator:

1- Determine the total heat output of the generation system, that is the value of the pre determined energy required as input of the distribution system (calculated with the EN 15316-2-3)

2- Determine the total time $t_{gn}$of operation of the generator;

3- If the load factor is not known, execute the following iterative procedure (4,5,6)with a starting value FC=1, from which are determined $P_{ch,on}$,$P_{ch,off}$ and $P_{gn,env}$.

4- Determine $P_{ch,on}$,$P_{ch,off}$ and $P_{gn,env}$ with the actual FC value;

5- Determine W_g,Q_{br} and Q_{af} with the actual FC value;

6- Calculate the new load factor:

$$FC = \frac{\frac{100*(Q_{g,out}-Q_{af})}{t_{gn}*\Phi_{ref}}+P_{ch,off}+P_{gn,env}}{100*\frac{\Phi_{cn}+k_{br}*\Phi_{br}}{Q_{ref}}-\frac{\Phi_{cn}}{\Phi_{ref}}*P_{ch,on}+P_{ch,off}} \tag{5.13}$$

7- Repeat steps 4,5 and 6until FC converges;

8- Calculate the energy to be supplied to the heat generator by the fuel:

$$Q_{g,in} = \Phi_{cn} * t_{gn} * FC \tag{5.14}$$

9- Calculate the total heat losses by:

$$Q_{g,l,t} = Q_{g,in} - Q_{g,out} + Q_{br} + Q_{af} \tag{5.15}$$

10- The recoverable heat losses has been taken into account as a reduction of heat losses through the generator, then:

$$Q_{g,l,rl} = 0 \tag{5.16}$$

Multistage and modulating generators (effect of burner control) are taken into account by addition of a third reference state: burner ON at minimum continuous power. The performance of these boilers is calculated assuming that the following operating conditions are possible:

- if the power required by the distribution system is less than the minimum ON power, the boiler will cycle just as a single stage boiler;

- if the power required by the distribution system is higher than the minimum ON power, the boiler will stay ON continuously and its loss factors are calculated by interpolation between minimum load and maximum load values.

Calculation procedure for a multi stage generator:

The following procedure has to be considered if, with a multi-stage heat generator, by using the above procedure the FC value converges to a number > 1.

1- Determine the total heat output of the generation system, that is the value of the pre determined energy required as input of the distribution system (calculated with the EN 15316-2-3);

2- Calculate $P_{gn,env}$ with FC=1;

3- Calculate the heat loss at the chimney in on-mode for the maximum $P_{ch,on,max}$ and the minimum $P_{ch,on,min}$ generators power wit FC=1;

4- Set $\Phi_{cn,avg} = \Phi_{cn,min}$;

5- Calculate $P_{ch,on,avg}$ by:

$$P_{ch,on,avg} = P_{ch,on,min} + \left(P_{ch,on,max} - P_{ch,on,min}\right) * \frac{\Phi_{cn,avg}-\Phi_{cn,min}}{\Phi_{cn,max}-\Phi_{cn,min}} \quad (5.17)$$

6- Calculate $\Phi_{br,avg}$ by:

$$\Phi_{br,avg} = \Phi_{br,min} + \left(\Phi_{br,max} - \Phi_{br,min}\right) * \frac{\Phi_{cn,avg}-\Phi_{cn,min}}{\Phi_{cn,max}-\Phi_{cn,min}} \quad (5.18)$$

7- Calculate a new $\Phi_{cn,avg}$ value by:

$$\Phi_{cn,avg} = \frac{\frac{Q_{g,out}-Q_{af}}{t_{gn}}+\frac{P_{gn,env}}{100}*\Phi_{ref}-k_{br}*\Phi_{br,avg}}{1-\frac{P_{ch,on,avg}}{100}} \quad (5.19)$$

8- Repeat steps 5,6 and 7 until $\Phi_{cn,avg}$ converges.

9- Calculate the energy to be supplied to the heat generator by the fuel:

$$Q_{g,in} = \Phi_{cn,avg} * t_{gn} \quad (5.20)$$

10- Calculate the auxiliary energy by:

$$W_g = t_{gn} * (\Phi_{br,avg} + \Phi_{af}) \quad (5.21)$$

11- Calculate the recovered auxiliary energy by:

$$W_{g,rd} = t_{gn} * (\Phi_{br,avg} * k_{br} + \Phi_{af} * k_{af}) \quad (5.22)$$

12- Calculate the total heat losses by:

$$Q_{g,l,t} = Q_{g,in} - Q_{g,out} + \Phi_{br,avg} * k_{br} * t_{gn} + \Phi_{af} * k_{af} * t_{gn} \quad (5.24)$$

13- The recoverable heat losses has been taken into account as a reduction of heat losses through the generator, then:

$$Q_{g,l,rl} = 0 \quad (5.25)$$

Condensation latent heat recovery is calculated according to flue gas composition and temperature and taken into account as a "bonus" (reduction) of flue gas losses.
For a single stage boiler the procedure to be taken into account is the same as the one described above, where $P_{ch,on}$ is replaced with $P^{*}_{ch,on}$ given by:

$$P^{*}_{ch,on} = P_{ch,on} - R \quad (5.26)$$

Where R is the recovered latent heat at a nominal rate, as a percentage of the burner combustion power,Φ_{cn}.

The procedure to calculate the R factor is summarized below

- Flue gas temperature,θ_{fl}, is calculated as the sum of:
  - water return temperature, $\theta_{gn,w,r}$,which depends on the operating conditions of emitters and distribution system (heating system effect);
  - temperature difference between flue gas temperature and water return temperature (boiler effect),$\Delta\theta_{wfl}$.

- Dry flue gas actual volume:

$$V_{fl,dry} = V_{fl,st,dry} * \frac{20{,}94}{20{,}94 - O_{2\,fl,dry}} \quad (5.27)$$

- Dry air actual volume:

$$V_{air,dry} = V_{air,st,dry} + V_{fl,dry} - V_{fl,st,dry} \quad (5.28)$$

- Water amount in the combustion air:

$$M_{H2O,air} = M_{H2O,air,sat} * V_{air,dry} * \frac{HUM_{air}}{100} \quad (5.29)$$

- Water amount in the flue gas:

$$M_{H2O,fl} = M_{H2O,fl,sat} * V_{fl,dry} * \frac{HUM_{fl}}{100} \tag{5.30}$$

Where:

$M_{H2O,fl,sat}$: *is given in tables and is relative to the flue gas temperature,*θ_{fl}*;*
$M_{H2O,air,sat}$: *is given in tables and is relative to the inlet air temperature,*θ_{air}*;*
HUM_{air} *and* HUM_{fl} *are given by tables as default values.*

- Water amount condensed:

$$M_{H2O,cond} = M_{H2O,st} + M_{H2O,air} - M_{H2O,fl} \tag{5.31}$$

- Condensation specific heat:

$$H_{cond,fl} = 2500600 - \theta_{fl} * 2435 \ \text{[J/kg]} \tag{5.32}$$

- Condensation heat:

$$Q_{cond} = M_{H2O,cond} * H_{cond,fl} \tag{5.33}$$

- Recovered latent heat:

$$R = \frac{100 * Q_{cond}}{H_i} \tag{5.34}$$

The Seasonal Generation Efficiency is:

$$\eta_s = \frac{Q_{g,out}}{Q_{g,in} + f_{el} * W_g} \tag{5.35}$$

5.2.2 NUMERICAL EXEMPLE

It was considered for the calculation of the seasonal efficiency a residential building in Northern Italy. Following the calculation methodology proposed by the Italian Standard UNI-TS 11300 [46] for the evaluation of the ideal energy required for space heating, it was obtained:

The heating season is between 15th October and 15th April: $t_{gn} = 15.638.400 \ s$;

- Energy requirement for space heating for each month:

Month	*October**	*November*	*December*	*January*	*February*	*March*	*April**
Q_H [MJ]	*3218*	*11423*	*15176*	*16581*	*13200*	*11148*	*3581*

*only 15 days.

- Energy gain by sun radiation and internal heat sources:

Month	*October*	*November*	*December*	*January*	*February*	*March*	*April*
Q_G [MJ]	1510	2195	2198	2317	2551	3137	1590

Of this energy gain only one part is recovered, and reduces the heat requirements of the building, it was estimated that only the 88% is effectively recovered. Thus, the Ideal heat required for space heating, month by month is given in the next table:

Month	***October***	***November***	***December***	***January***	***February***	***March***	***April***
Q_hvs [MJ]	*1972*	*9612*	*13363*	*14669*	*11095*	*8560*	*2269*

The efficiency of the emission and regulation and distribution sub systems was chosen from the default values proposed in the UNITS 11300 [46]:

- Emission efficiency: radiator on external insulated wall, $\eta_{em} = 0{,}94$;
- Control efficiency: climatic compensation and ambient control, $\eta_{control} = 0{,}97$;
- Distribution efficiency: independent plant, built between 1977-1993, $\eta_{dis} = 0{,}98$;

The total efficiency is:

$$\eta_{tot} = \eta_{em} * \eta_{control} * \eta_{dist} = 0{,}89 \tag{5.36}$$

The energy that the boiler must provide is:

$$Q_{g,out} = Q_{dist,in} = \frac{Q_{hvs}}{\eta_{tot}} \tag{5.37}$$

Month	***October***	***November***	***December***	***January***	***February***	***March***	***April***
$\boldsymbol{Q_{g,out}}$	*2216*	*10800*	*15014*	*16483*	*12467*	*9618*	*2550*

The energy that the heat generator must provide during the heating season is:

$$Q_{g,out} = 69147 \ \ [MJ]$$

Was executed the calculations for the whole heating season, with the oil-fired condensing boiler ICOvit VKO 246-7, with the following input data:

The boiler data were chosen from both the plate data and default values provided by EN 15316-4-1[42], Appendix C:

$\boldsymbol{\Phi_{cn}}$	*18 kW*	*Plate data*
$\boldsymbol{\theta_{gn,wf}}$	*60 °C*	*Plant data*
$\boldsymbol{\theta_{gn,wr}}$	*40 °C*	*Plant data*
$\boldsymbol{\theta_{gn,test}}$	*70 °C*	*Appendix C*
$\boldsymbol{\theta_{i,gn}}$	*16 °C*	*Appendix C*
$\boldsymbol{\theta_{i,gn,test}}$	*13 °C*	*Appendix C*
$\boldsymbol{\Phi_{br}}$	*260 W*	*Plate data*
$\boldsymbol{\Phi_{af}}$	*0*	*no internal circulator*
$\boldsymbol{P'_{ch,on}}$	*11 %*	*Appendix C*
$\boldsymbol{P'_{gn,env}}$	*1,17 %*	*Appendix C*
$\boldsymbol{P'_{ch,off}}$	*1,2 %*	*Appendix C*
$\boldsymbol{k_{br}}$	*0,8*	*Appendix C*
$\boldsymbol{f_{corr}}$	*0,045*	*Appendix C*
$\boldsymbol{n, m, p}$	*0,1*	*Appendix C*
Data for condensation		*Appendix C*

The results are :

Load factor $FC = 0{,}27$

$$P_{ch,on} = 7{,}55\%$$
$$P_{ch,off} = 0{,}63\%$$
$$P_{gn,env} = 0{,}43\%$$

Auxiliary Energy: $W_g = 1106\ MJ$

$$Q_{ch,on} = 5780\ MJ$$
$$Q_{ch,off} = 1288\ MJ$$
$$Q_{gn,env} = 1205\ MJ$$

$$Q_{g,in} = 76535\ MJ$$
$$Q_{g,l,t} = 8272\ MJ$$

The seasonal efficiency was calculated with:

$$\eta_s = \frac{Q_{g,out}}{Q_{g,in} + 2{,}5 * W_g} = 0{,}87 \tag{5.38}$$

Using the same procedure and the same boiler data, it was calculated the monthly performance of the generation sub-system:

month	*October*	*November*	*December*	*January*	*February*	*March*	*April*
$Q_{g,out}$	*2216*	*10800*	*15014*	*16483*	*12467*	*9618*	*2550*
t_{gn}	*1296000*	*2592000*	*2678400*	*2678400*	*2419200*	*2678400*	*1296000*
FC	*0,11*	*0,26*	*0,34*	*0,38*	*0,32*	*0,22*	*0,13*
$Q_{g,in}$	*2566*	*11972*	*16517*	*18103*	*13743*	*10756*	*2928*
$Q_{ch,on}$	*174*	*898*	*1282*	*1420*	*1056*	*794*	*202*
$Q_{ch,off}$	*119*	*217*	*204*	*195*	*190*	*231*	*119*
$Q_{gn,env}$	*91*	*199*	*211*	*213*	*189*	*202*	*92*
W_g	*37*	*173*	*239*	*261*	*199*	*155*	*42*
η_s	*0,83*	*0,87*	*0,88*	*0,88*	*0,88*	*0,86*	*0,84*

5.3 DIRECTIVE: ENERGY RELATED PRODUCTS

Europe has long required objective measurements of equipment performance. These measurements allow the setting of minimum requirements that manufacturers must meet, and provide consumers with information on appliances performance so they can make intelligent choices when purchasing.

Has been proposed the following definition of ‘seasonal space heating energy efficiency’,ηs, as:
the ratio between the space heating demand for a designated heating season, supplied by a space heater and the annual energy consumption required to meet this demand, expressed in %[47];

To calculate the ‘seasonal space heating energy efficiency for a “boiler space heater”, according the draft regulation [48], are required the following input data, which the manufacturer will be obliged to provide in the technical documentation:

1. P_4 is the useful heat output at nominal heat output, calculated with the testing procedure given in the EN 304, in which the test conditions are: water temperatures 80/60°C, and nominal heat output;
2. P_1 is the useful heat output at 30% of the nominal heat output, calculated with the testing procedure given in the EN 304, in which the test conditions are: return water temperatures 30°C (condensing boiler), and 30% of the nominal heat output;
3. η_4 is the useful efficiency at nominal power), calculated with the testing procedure given in the EN 304, referred to P_1 condition and measured on Higher Heating Value basis;
4. η_1 is the useful efficiency at 30% of the nominal power (part load), calculated with the testing procedure given in the EN 304, referred to P_4 condition and measured on Higher Heating Value basis;
5. P_{stby} is the stand-by heat loss is calculated with the procedure described in the EN 304 (in which is indicated as “q”);
6. elmax is the auxiliary electricity consumption at full load (mainly Oil pump and Ari blower);
7. elmin is the auxiliary electricity consumption at part load (30%);
8. P_{SB} is the auxiliary electricity consumption in stand-by mode;
9. P_{ign} is the ignition burner power consumption

The representative seasonal space heating energy efficiency is given as follows:

$$\eta_s = \eta_{son} - \sum F(i) \qquad (5.39)$$

Where:

η_{son} is the “seasonal space heating energy efficiency in active mode”;
F(i) are the correction factors.

The “seasonal space heating energy efficiency in active mode”,η_{son}, is calculated as follows:

$$\eta_{son} = 0{,}85 * \eta_1 + 0{,}15 * \eta_4 \qquad (5.40)$$

Where:

η_1: *is the useful efficiency at 30% of the nominal power (part load)*
η_4: *is the useful efficiency at nominal power.*

That represents a weighted measure of the heat generator, considering that in the “heating season” it works for the 15% of the time at full load and for the 85%of the time at the 30% of the nominal load.

The correction factors F(i) are calculated as follows:

F(1) accounts a negative contribution to the “seasonal space heating energy efficiency” due to adjusted contribution of temperature controls. For space heaters, boiler combination heater and cogenerating space heaters, the correction is $F(1) = 3\%$.

F(2) accounts the negative contribution to the “seasonal space heating energy efficiency” by the “auxiliary electricity consumption”, and is calculated with:

$$F(2) = 2{,}5 * \frac{0{,}15*\mathrm{elmax}+0{,}85*\mathrm{elmin}+1{,}3*P_{SB}}{0{,}15*P_4+0{,}85*P_1} \tag{5.41}$$

Where ‘auxiliary electricity consumption’ means the annual electricity required for the designated operation of a boiler space heater, calculated from the electric power consumption at full load ($elmax$), at part load ($elmin$), in standby mode and default operating hours at each mode, expressed in kWh in terms of final energy (the factor 2,5 is the conversion factor from electricity final energy);

F(3) accounts a negative contribution to the “seasonal space heating energy efficiency” by stand-by heat losses and is given as follows:

$$F(3) = 0{,}5 * \frac{P_{stby}}{P_4} \tag{5.42}$$

Where ‘standby heat loss’ (P_{stby}) means the heat loss of a boiler space heater, in operating modes without heat demand, expressed in kW.

F(4) accounts a negative contribution to the “seasonal space heating energy efficiency” by ignition burner power consumption, i.e. when the burner is provided with a pilot flame, and is calculated as follows:

$$F(4) = 0{,}5 * \frac{P_{ign}}{P_4} \tag{5.43}$$

Where the‘ignition burner power consumption’ (Pign) is the power consumption of a burner intended to ignite the main burner, expressed in W in terms of GCV.

Once obtained the value of η_s, we can understand if the considered heating generator fulfill the European Standards, and what is the energy class of the device.

For the Vaillant boiler considered, we couldn’t find all the values required for the calculation of the seasonal space heating energy efficiency, so we tried to estimate probable values using literature and default values found in technical standards.

P_1	*5,7 kW*
P_4	*19 kW*
η_1	*100,3 %*
η_4	*92,3 %*
$elmax$	*260 W*
$elmin$	*78 W*
P_{sb}	*5 W*
P_{stby}	*50 W*
P_{ign}	*0 W*

Applying the equations described above we obtained the following representative seasonal space heating energy efficiency value:

$$\eta_s = 90\ \%$$

5.3.1 ENERGY LABEL

Increase the market share of energy-efficient heaters will contribute to achieving the 20%energy savings potential anticipated by 2020 in the Energy Efficiency Action Plan(COM(2006) 545), and confirmed in the Commission’s Communication on Energy 2020(COM(2010) 639) and Energy Efficiency Plan 2011 (COM(2011) 109).

One of the main reasons for persistent sales of low-efficiency heaters and conventionalproducts without the use of renewable energy sources or cogeneration is that end-users basetheir purchase decisions on the purchase costs of products rather than their life cycle costs, asituation not helped by the current practice of not fully including environmental costs inenergy costs.

For this reason EU decided to promulgate the 2010/30/EU Directive [47], that introduces energy labels for Energy related Products, thus for space heating boilers and other heating systems. Itintroduces the widely known A-G scale to cover the various types of conventional boilers.

Additionally, the dynamic top classes A+, A++ and A+++ are intended to promote the use ofcogeneration and renewable energy sources. Standardized product information will be madeavailable to end-users in the form of ‘fiches’ (i.e. information notices), on the internet and inadvertisements.

The proposed measure sets out new mandatory labeling and standard product informationrequirements for suppliers placing on the market and/or putting into service heaters, temperature controls, solar devices (solar-only system, solar collector, solar tank and othersolar products placed on the market separately) or passive flue heat recovery devices, and fordealers offering stand-alone heaters and packages of heaters, temperature controls, solardevices and/or passive flue heat recovery devices. The scope of the measure is aligned withthe scope of a proposed Eco-design implementing measure setting Eco-design requirements forthe energy efficiency, sound power levels and nitrogen oxide emissions of heaters.

In Fig. 5.3 is reported an example of Label.

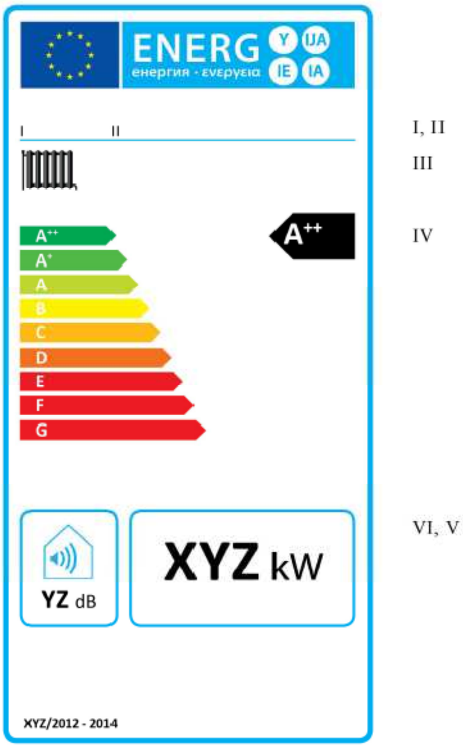


Fig.5.3 Energy Label

In the Label is possible to individuate the following areas:

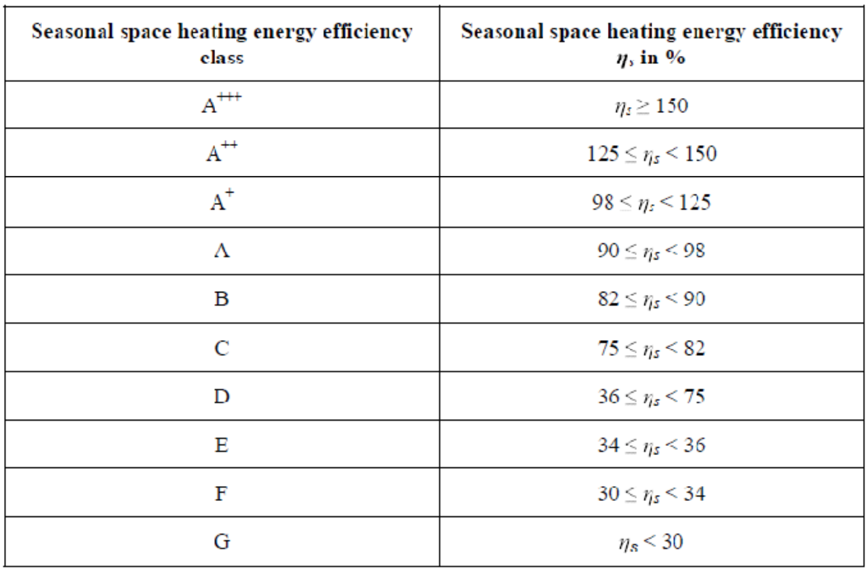
I. Supplier’s name;
II. Supplier’s model identifier;
III. The space heating function, (in this case radiator system);
IV. The seasonal space heating energy efficiency;
V. The rated heat output in kW, rounded to the nearest integer;
VI. The sound power level, indoors, in dB, rounded to the nearest integer;

The energy efficiency ranking of heaters is based on the scheme laid down in Directive2010/30/EU in having a single efficiency scale for space heating, covering boilers,cogeneration, heat pumps and their packages with further products. Two years after the entryinto force of the Delegated Regulation, a scale from G to A for conventional heaters (G-D forelectric boilers, C-B for non-condensing boilers in collective buildings, B-A for condensingboilers) with higher classes A_{+} for cogeneration and A_{++} for heat pumps will be introduced.

Four years after the entry into force of the Delegated Regulation, a further class A_{+++} will beadded on top of the labelling scale, while classes G to E will be abolished due to moreambitious Eco-design requirements. This will ensure dynamic market transformation towardhighly efficient heaters using new and renewable energy technologies[49].

The energy efficiency ranking refers to the “seasonal space heating energy efficiency”, discussed in Paragraph 5.3.

The seasonal space heating energy efficiency classes of a heater, with the exception of lowtemperatureheat pumps and heat pump space heaters forlow-temperature application, shall bedetermined on the basis of its seasonal space heating energy efficiency as set out in Table 1.



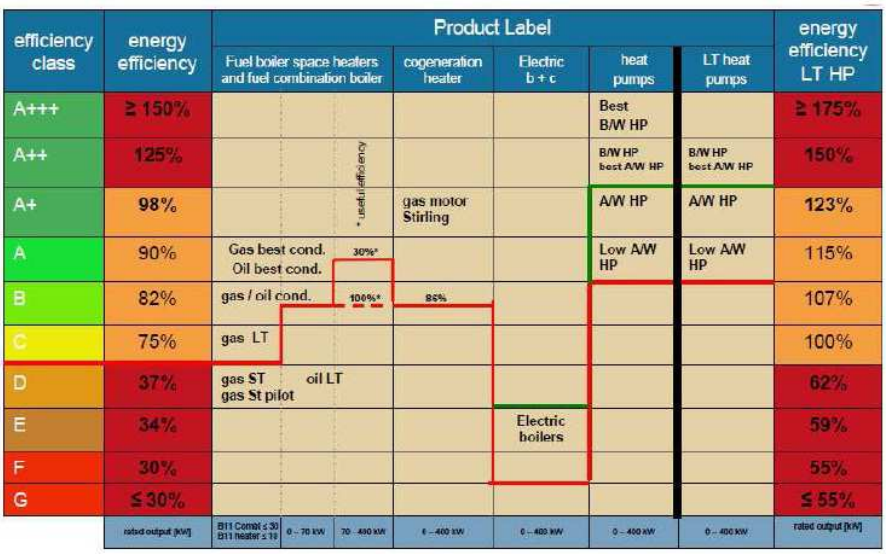
Seasonal space heating energy efficiency class	Seasonal space heating energy efficiency η, in %
A^{+++}	$\eta_s \geq 150$
A^{++}	$125 \leq \eta_s < 150$
A^{+}	$98 \leq \eta_s < 125$
A	$90 \leq \eta_s < 98$
B	$82 \leq \eta_s < 90$
C	$75 \leq \eta_s < 82$
D	$36 \leq \eta_s < 75$
E	$34 \leq \eta_s < 36$
F	$30 \leq \eta_s < 34$
G	$\eta_s < 30$

Tab. 1 Boiler space heaters in seasonal space heating energy efficiency classes A++ to G

With the calculations made in the previous section, the Vaillant oil-fired condensing boiler belongs to the efficiency class A, since the estimated seasonal space heating efficiency value is 90%.

This result, shows that the oil-fired condensing boiler analyzed belongs to the best technology currently available in the market, regarding the oil-fired space heating boilers.

In Fig. 5.4 is represented a diagram showing the subdivision in efficiency classes, for different space heating technologies [50].



efficiency class	energy efficiency	Product Label							energy efficiency LT HP
		Fuel boiler space heaters and fuel combination boiler			cogeneration heater	Electric b + c	heat pumps	LT heat pumps	
A+++	≥ 150%						Best B/W HP		≥ 175%
A++	125%			* useful efficiency			B/W HP best A/W HP	B/W HP best A/W HP	150%
A+	98%				gas motor Stirling		A/W HP	A/W HP	123%
A	90%	Gas best cond. Oil best cond.		30%*			Low A/W HP	Low A/W HP	115%
B	82%	gas / oil cond.		100%*	86%				107%
C	75%	gas LT							100%
D	37%	gas ST gas St pilot	oil LT						62%
E	34%					Electric boilers			59%
F	30%								55%
G	≤ 30%								≤ 55%
	rated output [kW]	B11 Combi ≤ 30 B11 heater ≤ 10	0 – 70 kW	70 – 400 kW	6 – 400 kW	0 – 400 kW	0 – 400 kW	0 – 400 kW	rated output [kW]

Fig.5.4 Different technologies performance and labels

CHAPTER 6

CONCLUSIONS

In this Master's Thesis were investigated the thermal and energetic characteristics of an oil-fired condensing boilerdescribing the main phenomena occurring in an oil boiler: atomization, evaporation of fuel and mixing with combustion air obtaining a ignitable mixture in the combustion chamber. Were discussed the droplet combustion model, describing the combustion mechanism when a liquid fuel is used, and the Colburn and Heugen theory to describe the water vapor condensation of the flue gases.

It was discussed the problem of pollutant emissions and the strategies for lowering them. And with that it was discussed the combustion efficiency concept, according the European Standard prEN 15378 and Italian Standard UNI 10389-1.

It was taken as example of boiler the Vaillant oil-fired condensing boiler IcoVIt 246-7, and was described its components and layout.

It was presented the concept of Useful Efficiency, and described the European test standard used for assessing this parameter.

The final part of this work is about the European Directives regarding the Seasonal Efficiency of a heat generator, the EPBD and the ErP Directives. These two Directives are consider the boiler from two different points of view: EPBD sees the boiler are a subsystem of the building, and the performance of the heat generator is influenced by the actual interaction with the building in which is installed. The ErP approach considers the heat generator as a "product" and sets the characteristics that boilers must satisfy to be sold in the European market.

By using the procedure described in these Directives was calculated the Seasonal Efficiency of the analyzed boiler. The results show that the seasonal efficiency of the boiler, calculated according both the Directives, is around 90% (on higher heating value basis). That, according the new parameters introduced by the 2010/30/EU Directive, on the Energy Labeling, corresponds to an A class.

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