DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

EFFICIENCY AND ENERGY LOSSES FOR COMBI BOILERS

by
Erdem KESDOĞAN

August, 2006
İZMİR

EFFICIENCY AND ENERGY LOSSES FOR COMBI BOILERS

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M.Sc THESIS EXAMINATION RESULT FORM

We have read the thesis entitled **"EFFICIENCY AND ENERGY LOSSES FOR COMBI BOILERS"** completed by **ERDEM KESDOĞAN** under supervision of **Prof. Dr. İSMAİL HAKKI TAVMAN** and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

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Erdem KESDOĞAN

EFFICIENCY AND ENERGY LOSSES
FOR
COMBI BOILERS

ABSTRACT

In this thesis study, the potential for efficiency improvement methods for gas-fired combi boilers are investigated. Efficiency and energy losses are experimentally determined, while the performance of a gas-fired combi boiler is evaluated using energy and exergy analysis.

This thesis consists of mainly five chapters. In the first chapter, main description and contribution of the gas-fired combi boilers for domestic and commercial in the overall energy balance worldwide are given. The second chapter includes a classification of the combi boilers according to different categories, while main components of room sealed flue combi boiler are briefly explained. In the following chapter, efficiency measurement techniques based on the related standards, detailed combustion equations, exergy analysis and energy loss sources are examined. A brief description of condensing boilers and the state of the technological improvement for these boilers are also presented. The fourth chapter contains experimental studies on efficiency and energy losses, which have been performed at the laboratories of Bosch Heating Systems Research and Development department, in Manisa. In the last chapter; the test results obtained are assessed, while main topics of the thesis are given briefly. Average energy and exergy efficiency values for the combi boilers studied are found to be 91 and 13.6%, respectively.

Keywords: combi boiler, efficiency, exergy, heating systems, energy losses.

GAZLI KOMBİLERDE VERİM VE ENERJİ KAYIPLARI

ÖZ

Bu tez çalışmasında; gazlı kombilerde verimi artırma yöntemleri incelenmiştir. Verim ve enerji kayıpları deneysel çalışmalarla irdelenip; enerji ve ekserji analizi göz önüne alınarak gazlı kombilerin performansı üzerinde çalışılmıştır.

Tez çalışması 5 ana bölüme ayrılmıştır. Birinci bölümde gazlı kombilerin genel tanımı ve dünyadaki kullanım alanları verilmiştir. İkinci bölümde; değişik kategorilere göre kombiler sınıflandırılmış ve genel ekipmanları tanıtılmıştır. Üçüncü bölümde; ilgili standartlara uygun olarak verim ölçüm teknikleri, ayrıntılı yanma denklemleri, kayıpların kaynakları, ekserji analizi ve günümüzde en gelişmiş teknoloji olan yoğuşmalı cihazlar üzerinde durulmuştur. Dördüncü bölüm; kombilerde verim ve enerji kayıplarıyla ilgili olarak, BOSCH Isıtma Ürünleri San. Tic. A.Ş. Ürün Geliştirme Departmanı laboratuarlarında yapılan deneysel çalışmalar ve bu çalışmaların sonuçlarının değerlendirilmesini içermektedir.

Son bölümde; bulunan sonuçlar değerlendirilmiş ve tezin temel amacı sunulmuştur. Ortalama enerji ve ekserji verim değerleri %91 ve %13.6 olarak hesaplanmıştır.

Anahtar sözcükler: kombi, verim, ekserji, ısıtma sistemleri, enerji kayıpları

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CHAPTER ONE

INTRODUCTION

The use of gas for domestic and commercial applications represents about 7% of the total gross energy worldwide (22nd World Gas Conference, Tokyo), and this share is growing as gas tends to gain a foothold in the markets of electricity in developed countries or traditional energy sources (e.g. coal) in developing countries. The exact proportion of gas is, in fact, larger, because some of the energy forms that count in the statistics used are heat (e.g. district heating), which is partly produced with gas. The overall trend worldwide is an increase of the gas share in replacement of solid fuel, electricity and fuel oil. (see figure 1.1)

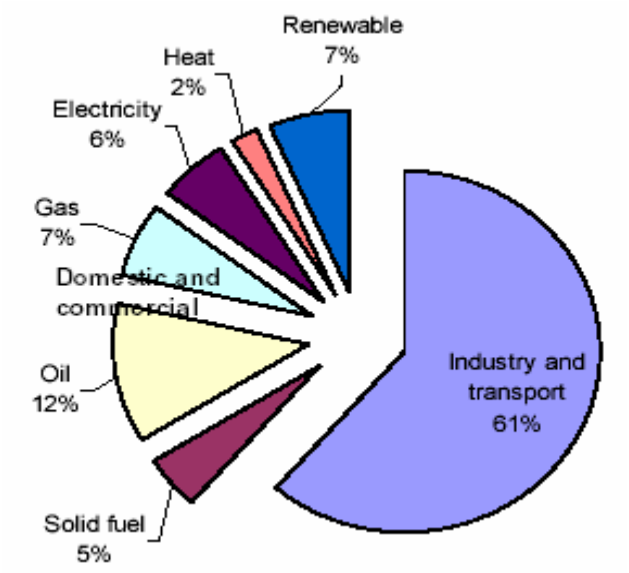


Figure1.1 Share of gas in the domestic and commercial sector in the world energy consumption (22nd World Gas Conference – June 1-5, 2003)

It might assumed that the overall contribution of gas for domestic and commercial in the overall energy balance worldwide is today probably about 10% and increasing (considering that the figure of 7% is from 1997 and that part of the "heat" is produced by NG). Therefore, the impact of the improvement of the gas technology is of major importance for energy savings and CO_2 reduction worldwide.

There is no doubt that gas technology has a significant impact on tomorrow's world environment. Not only sophisticated technologies are needed, but also simple and cheap ones that the new consumers from countries in transition or development can afford. The development of the gas market is today one of the most effective solutions for saving energy. Today's general trend is an increase of the domestic and commercial gas markets, even if some appliances are declining (e.g. water heaters are increasingly replaced by combi-boilers).

The energy crisis and the increase of natural gas consumption, especially in Europe, during the 1970s draw the attention to higher efficiency performances and new technologies like condensing appliances. Efficiency ratings of gas boilers are becoming a very important feature in all markets, regarding energy costs, environmental and marketing aspects. Therefore there is a big necessity to produce more efficient appliance that gives more convenience to the consumers with respect to developing technology and also competition in the heating market.

Neverthless the energy sources are decreasing and with respect to this, the energy costs are increasing. In the future, the gas market will expand in the sector of domestic and commercial use. The magnitude of this expansion will greatly depend on the capability of the gas industry to bring an answer to the identified technical challenges.

Heating is indeed the largest energy consumer in the domestic and commercial sectors for most of the countries as seen in Figure 1.2. Therefore, heating plays a main role and increasingly combined with hot water production (22nd World Gas Conference – June 1-5, 2003).

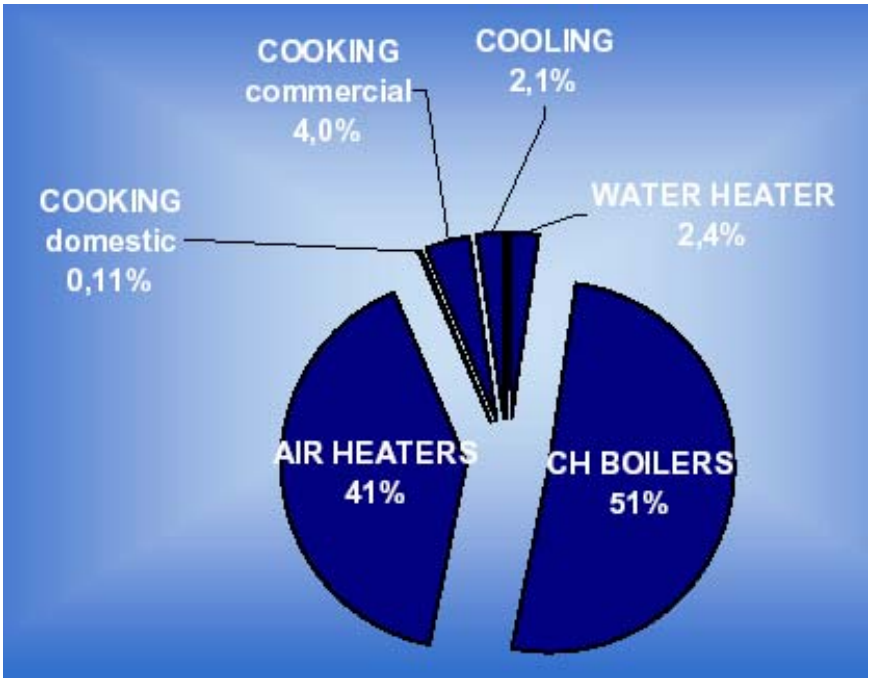


Figure1.2 Share of gas consumption by appliance

Natural gas is today used in all types of fuel consuming processes and one may easily point at both environmental and efficiency advantages in comparison with other fuels. In the sector of space heating, properties of both the fuel natural gas and the building can be noted for a potential of very efficient energy utilization. In this context are the properties divided into two categories, combustion and process. Combustion of natural gas generates a larger amount of water compared to oil or coal combustion. Only when moist fuels, for example wood and peat, or hydrogen are used the flue gases will have a water content equivalent to that of natural combustion or higher. The high water content involves a high water dew point, approximately 50-60ºC. The exact temperature is

mainly dependent on the combustion air excess ratio. The negligible content of sulphur and heavy metals combined with the possibility to control the combustion in order to minimize the CO and NOx emissions make natural gas an environmentally attractive fuel. The main objective of the gas market is trying to find the most effective solutions for saving energy and reducing the CO_2 and CO emission values. Especially reducing CO emission is very important point for human health and also environment because of its toxic effect.

The process in which the fuel is used determine how far the flue gases can be cooled, i.e. the highest obtainable efficiency. The temperatures in space heating systems, more precisely the return temperature from the heating system, are often low enough to allow condensation of the water vapour in flue gases. Natural gas can thus be used to obtain very high efficiencies. Furthermore, the condensate from natural gas flue gases is less aggressive than the condensate from oil flue gases. Altogether this have lead to the development of condensing gas boilers at attractive prices for the consumers.

Flue-loss efficiency (also referred to as “thermal efficiency” in the older literature; the proposed ASHRAE Standard 90.1-1989R correctly uses thermal efficiency to refer to boiler efficiency) is defined as the ratio of input energy minus the energy lost through the flue in the exhausted combustion products to input energy. Boiler efficiency is the percentage of energy transferred to the hot water from input energy. It takes into account the amount of the energy that is lost through the flue and other parts of the water heater, such as jacket and fitting losses. Therefore, boiler efficiency is always less than flue-loss efficiency for a given water heater.

Efficient operation of any combustion equipment (like boiler) is highly dependent on a proper air-to fuel ratio. Due to the mechanics of combustion, it is necessary to provide more air than would be required to provide exactly the right quantity of oxygen (O_2) to burn all the fuel without any O_2 left over. Because air is comprised of approximately 21% O_2 and 79% nitrogen (N_2), in delivering the right amount of O_2, nearly four times

as much N_2 is also delivered. Nitrogen absorbs heat and carries it out the stack, resulting in a loss to the system. Minimizing excess air, consistent with complete combustion, minimizes this heat loss.

The topic of this thesis will be investigating losses of room sealed gas combination boilers which are finally defining heating efficiency. This will cover more specifically; defining kinds and weights of energy losses of boiler, ideas for energy usage optimisation and possible energy conversion systems to convert the losses to useful efficiency.

CHAPTER TWO

GAS-FIRED COMBI BOILER AND CONDENSING TECHNOLOGY

Gas-fired combination boilers (so called "combination boiler" or "combi-boiler") are appliances that generate heat for two functions: space heating and domestic hot water production. Therefore, unlike a conventional system, a combination boiler does not store hot water. Instead it heats water directly from the cold mains, as needed it. They can both either heat water in a storage tank or directly heat the water when the user opens the tap. In that case the heater is called “instantaneous water heater”.

The major technical components of a combi are as follows:

- Gas valve
- Fan
- Burner
- Heat exchanger
- Pump (for floor standing type boilers often a separate component)
- Control system

2.1 Classification of the Gas-Fired Combi Boilers

2.1.1 Classification According to the Heat Exchanger Type

We can classify gas fired combination boilers with respect to different kinds of heat exchangers which the water heater function can be realized;

A. a water-to-water (plate) heat-exchanger (in which central heating water heats the tap water);
B. a small storage tank with a spiral shaped water containing pipe;
C. an air-to-water heat exchanger, integrated in the air-to-water heat-exchanger for the boiler function.

In the first two types the central heating water is used as an heating source. First the central heating water is heated by the burner in the primary heat-exchanger; then the domestic water is heated by the central heating water in a secondary heat-exchanger. In the last type of combi (C), the domestic water is directly heated by the burner in the primary heat exchanger (see figure 2.1) (SAVE II Action).

2.1.1.1 A and B Type Combi Boilers

When domestic hot water is demanded, the three-way valve opens the by-pass to the secondary heat exchanger. The heated water of the heating system is in the primary heat exchanger and flowing through the secondary heat exchanger. At the end of the tapping period, the valve closes the bypass and opens the heating circuit again. When the heating demand is over then burner is turned off. The “A” type combi boilers are equipped with a modulating burner, in which the gas supply is minimized if the water flow is below a nominal level (using pressure difference of temperature control). The “B” type combi boilers may use both on/off burners and modulating burners.

2.1.1.2 C Type Combi Boilers

In the integrated heat exchanger both the water of the heating system and the domestic hot water are heated. When domestic hot water is required, the heating system pump is turned off. At the end of the tapping the boiler pump is released and the burner is turned off if there is no heat demand.

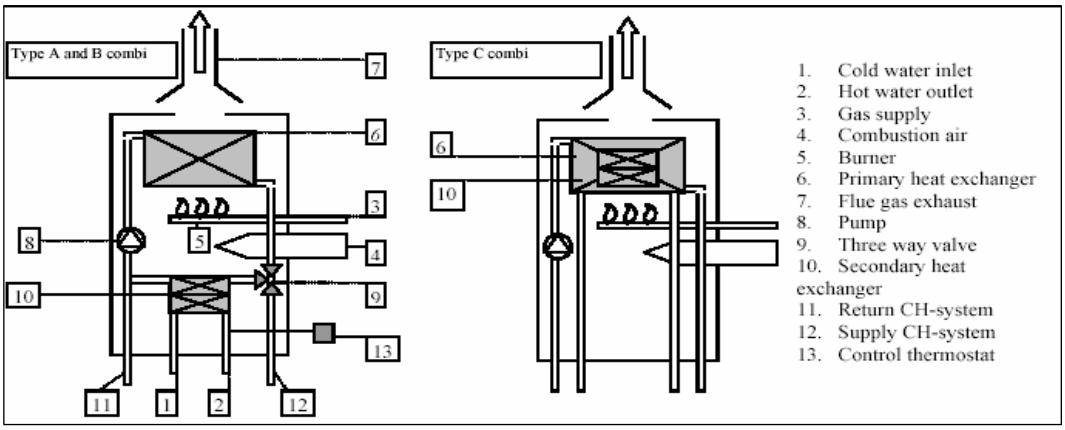


Figure 2.1 Classification of combi boilers according to the heat exchanger type (SAVE II Action)

2.1.2 Classification According to Burners

All burner types can be applied: atmospheric, ventilation and pre-mix. For atmospheric burners, the combustion air comes from the boiler room. Forced ventilation burners are often applied with an air inlet originating from outside of the dwelling. Modern pre-mix burners may have constant gas/air-ratio control. For burner ignition both a pilot flame (atmospheric burners) or electronic ignition (all types of burners) may be used. All boilers use auxiliary electric power (pump, controls, fan). The load ranges from 11 kW up to 40 kW and more. The most distinctive parameter is the function of the heat generator. In its most simple form the heat generator is used for space heating.

2.1.3 Classification According to the Gasses Capable of Being Used

Combi boilers are classified into categories defined according to the gasses and the pressures for which they are designed. Due to the conditions concerning the marketing of these boiler categories in each country and the corresponding supply pressures.

The gasses are classified into three families, possibly divided into groups depending on the “Wobbe Index”. Wobbe Index is the ratio of the calorific value of the gas per unit volume to the square root of the density under the same reference

conditions 15°C, 1013.25 mbar. The Wobbe Index is called gross or net according to whether the gross or the net calorific value is used (BSI – EN483).

Calorific value is the quantity of heat produced by the combustion at a constant pressure of 1013.25 mbar of unit volume or mass of gas, the constituents of the combustible mixture being brought to 15°C and 1013.25 mbar, the combustion products being brought to the same conditions. Two types of calorific value are identified:

Gross calorific value (H_s); water produced by combustion is taken to condensed. Net calorific value (H_i); water produced by combustion is taken to be in the vapor state.

2.1.4 Classification According to the Expansion System

Combi boilers are classified according to the expansion system of the central heating circuit use as:

- Open vented system: A boiler intended for a central heating system with an open expansion vessel
- Sealed system: A boiler intended for a central heating system with an open or sealed expansion vessel (BSI – EN483).

2.1.5 Classification According to the Water Side Operating Pressure

Combi boilers are classified according to their maximum water side operating pressure (PMS) as: (BSI – EN483).

- Pressure class 1 : PMS =1 bar
- Pressure class 2 : PMS = 3 bar
- Pressure class 3 : 3 bar < PMS < 6 bar.

2.1.6 Classification According to the Mode of Air Supply and Evacuation of the Combustion Products

Combi boilers are classified into several types according to the mode of evacuation of the combustion products and admission of the combustion air (BSI – EN483).

Open flue (OF) appliance: A combi boiler intended to be connected to a flue evacuating the combustion products outside the room, with the combustion air being drawn directly from the room where the boiler is installed. This type of combi boilers are called Type B.

Room sealed flue (RSF) appliance: A combi boiler which discharges its combustion products and draws in combustion air from adjacent points, outside the room in which it is installed, such that wind effects are effectively balanced. This type of combi boilers are called Type C.

The air supply and the combustion products evacuation ducts and the terminal or the fitting piece which is used to connect the boiler to a chimney or duct system are part of the boiler. They admit fresh air from outside the inhabitable part of the building to the burner as well as discharge the products of combustion to the outside. When classifying of RSF combi boilers flue system, it is used two subscript number like Cxy. The first subscript number is based upon the possible installation of the boiler with respect to the mode of air supply and evacuation of the combustion products. The second subscript number is based upon the presence and position of an integral fan in the boiler (see figure 2.2 and 2.3) (BSI – EN483).

Type C_{x1} $\Rightarrow$ Natural draught appliance (no fan)
Type C_{x2} $\Rightarrow$ fan downstream of the combustion chamber
Type C_{x3} $\Rightarrow$ fan upstream of the combustion chamber

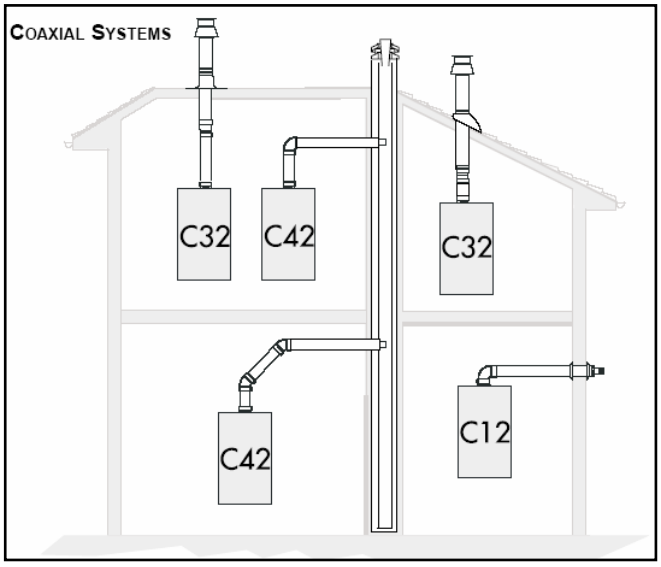


Figure 2.2 Evacuation types of RSF boilers

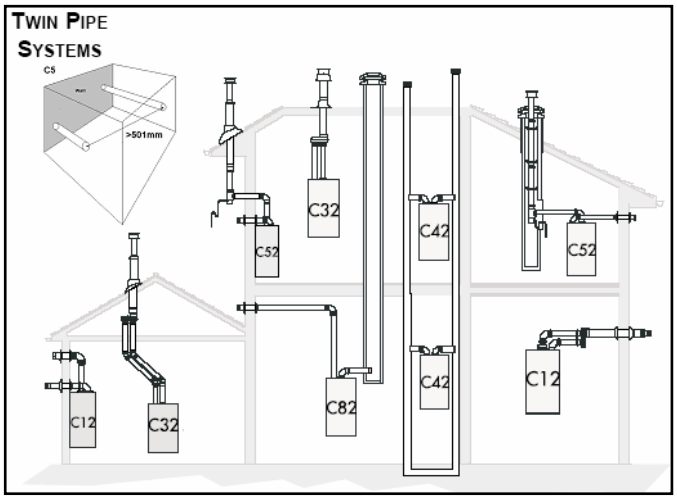


Figure 2.3 Evacuation types of RSF boilers - twin pipe system

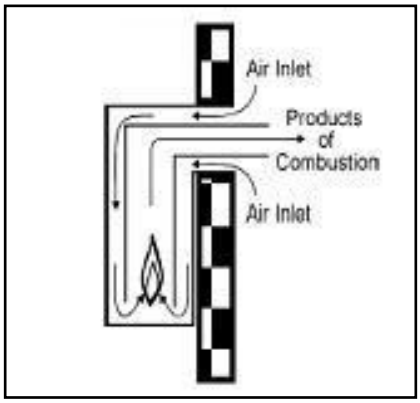
In this thesis; Room Sealed Flue (RSF) appliances are studied mostly, while condensing appliances are briefly explained.

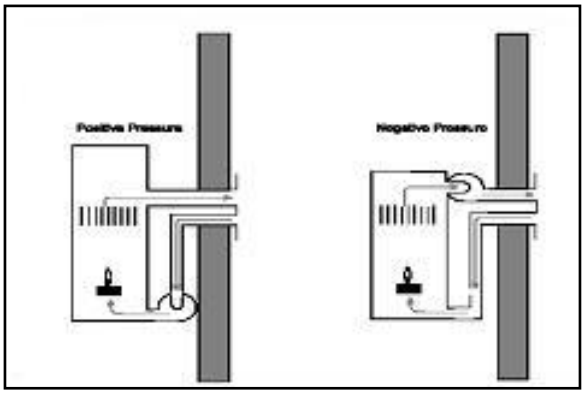
Room sealed appliances provide the solution to a number of problems associated with open flue systems. Appliances of this type are purchased as a complete unit, which incorporates the flue system. They remove the need for a potentially costly

and elaborate flue system. They also provide an extra level of customer safety being room sealed.

There are two main types of room sealed appliance:

- Balanced Flue - Natural Draught
- Balanced Flue - Fan Draught





Natural Draft Balanced Flue ***Fan Draft Balanced Flue***

Figure 2.4 Types of RSF boilers (BSI – EN483)

Balanced flue appliances should always be used in preference to an open flue appliance, where practicalities allow for this. There are a number of reasons why balanced flue appliances are preferred to open flue systems;

- Considered to be a safer appliance, as products of combustion cannot spill into the room unless the case seal is defective or incorrectly fitted. Removes the need to take air for combustion from the room.
- Minimizes condensation in the flue system can be installed in any room of a dwelling and easier to install. Usually looks neater when installed combustion air is not contaminated by household fluff, lint or pet hair

The function scheme and main components of RSF combi boilers are shown in Figure 2.5 and Figure 2.6 (BOSCH Thermotechnique).

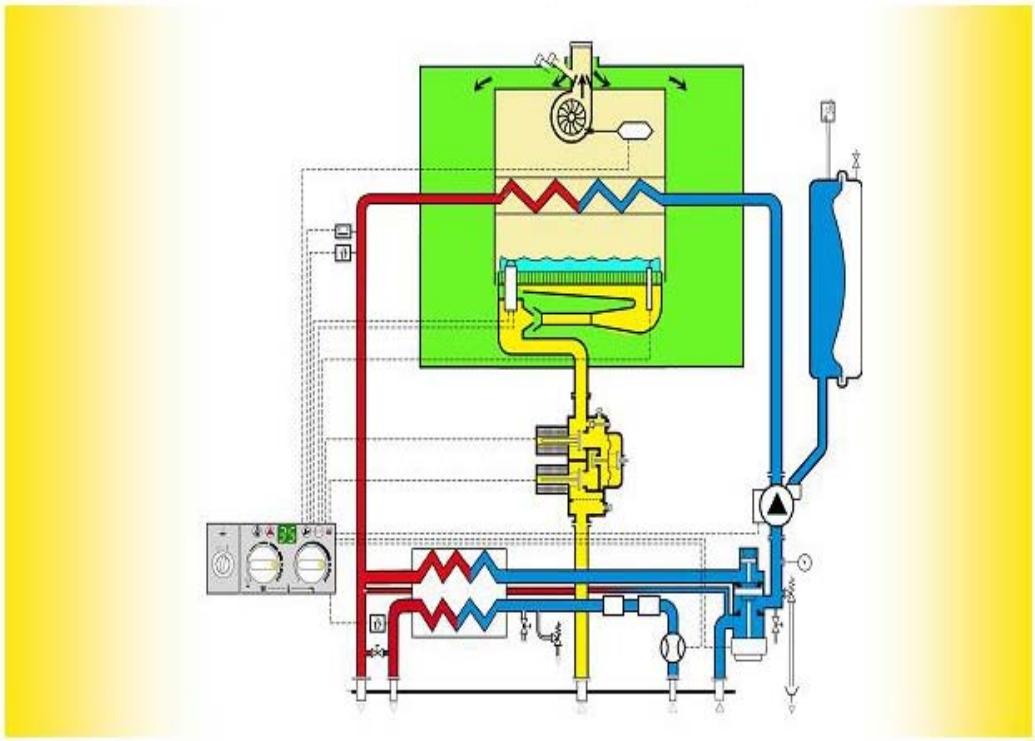


Figure 2.5 Bosch Euromaxx combi function scheme

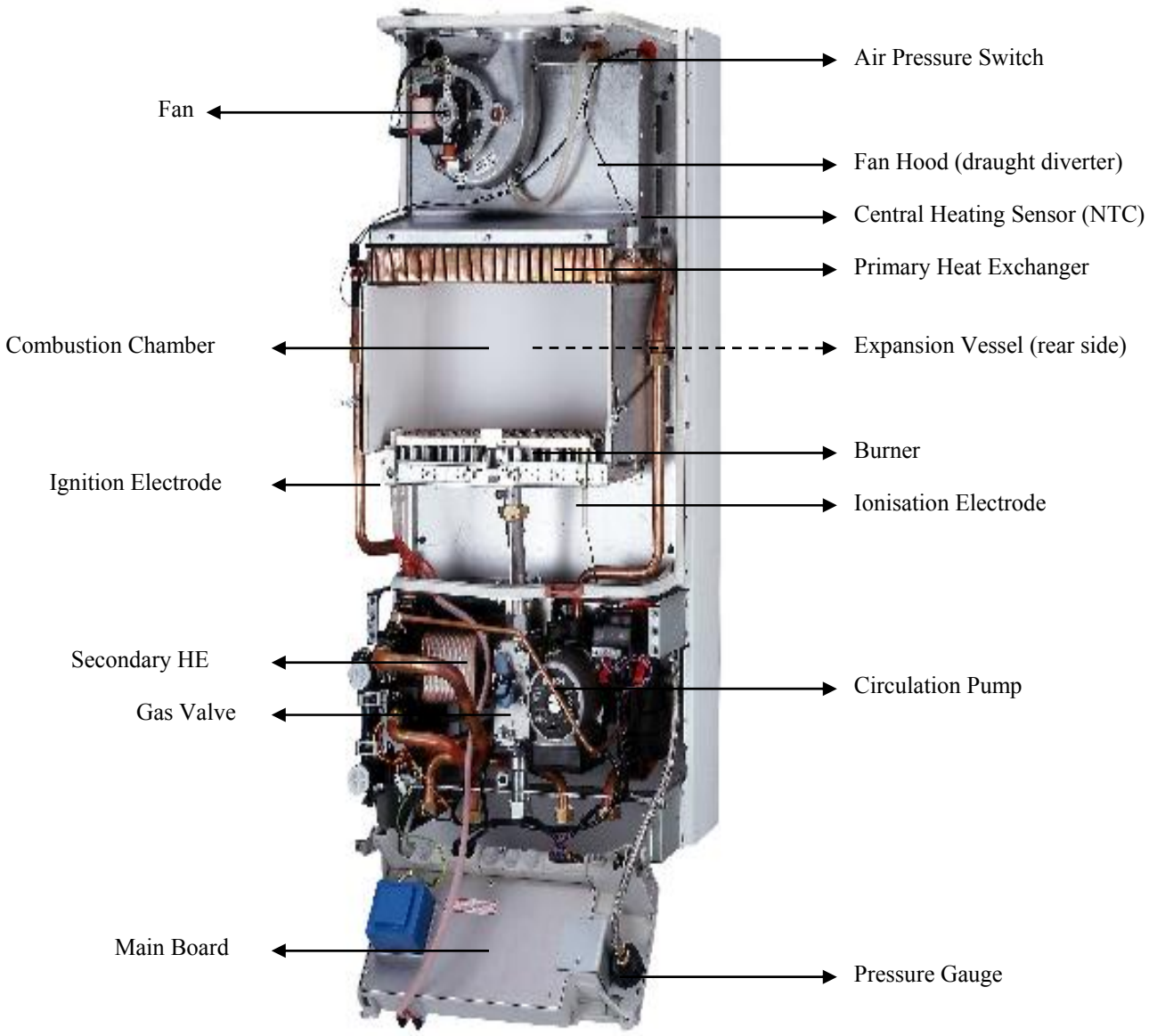


Figure 2.6 Main components of RSF combi boiler

2.2 The Parts of RSF Combi Boiler

2.2.1 Air Pressure Switch

- prevents working of burner in case of absence or insufficient air flow.

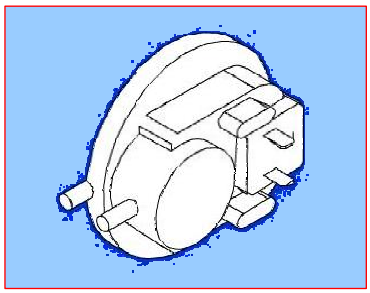


Figure 2.7 Air pressure switch

2.2.2 Fan Hood (Draught Diverter)

- collects the combustion products
- diverts combustion products to the inlet of the fan
- a device for preventing conditions in the secondary flue from interfering with the combustion performance of the boiler

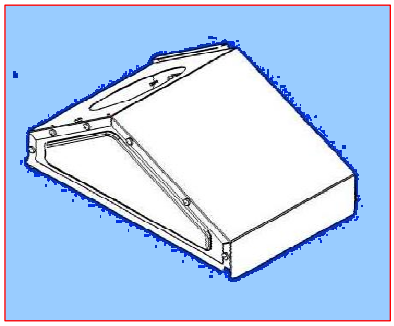


Figure 2.8 Fan hood

2.2.3 Fan

- gets fresh air for combustion
- blow the combustion products out of the hermetic chamber

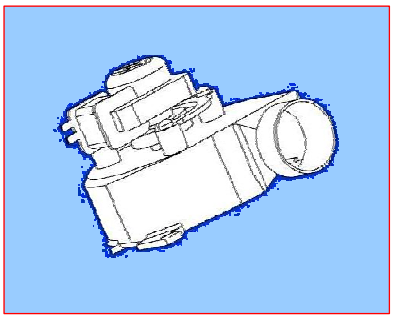


Figure 2.9 Fan

2.2.4 Combustion Chamber

- the closed and insulated chamber
- placed just above the burner
- where combustion takes place

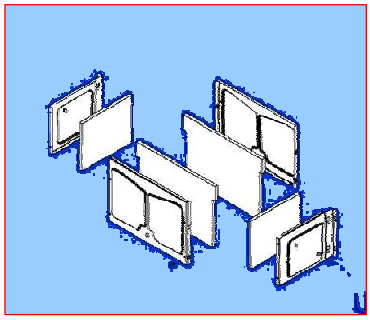


Figure 2.10 Combustion chamber

2.2.5 Primary Heat Exchanger

- Transfer the heat energy of the flame to the water in the Central Heating circuit.

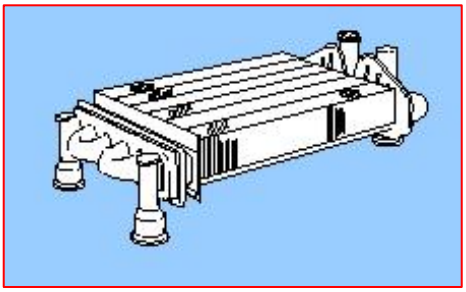


Figure 2.11 Primary heat exchanger

2.2.6 Ignition Electrode

- Its function is to light the burner
- at the time of heating request, ignition module (integrated within the mainboard) is fed by 230 V and it sends a very high voltage which causes an electric arc between the two electrodes. This arc light the gas.

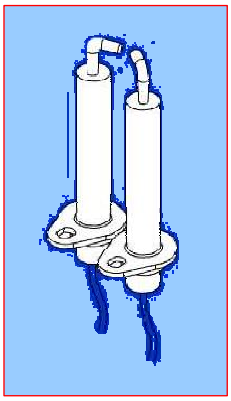


Figure 2.12 Ignition electrode

2.2.7 Ionisation Electrode

- Its function is to check the presence of flame in the burner
- In the flame, there are fractionation of molecules of the fuel with release of electrons and ions (atom that misses one or more electrons).
- The principle of control is to subject the flame to an electric field and to collect the current which passes through this flame.
- The current generated between the burner and the probe is extremely weak but it's sufficient to function the ionisation electrode.

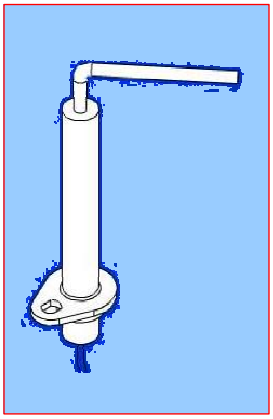


Figure 2.13 Ionisation Electrode

2.2.8 Burner

- It's built up of burner blades that mix air and the fuel coming from the manifold in order to prepare a mixture ready for combustion.
- Combustion takes place just above the burner.

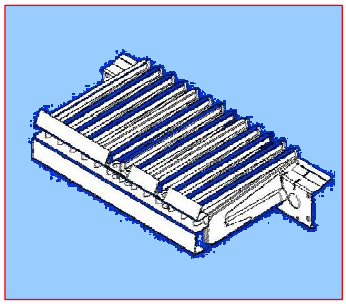


Figure 2.14 Burner

2.2.9 Central Heating Sensor (NTC)

- the NTC is placed in contact against a pipe.
- when the temperature of the tube increases, resistance decreases and conversely
- with the help of the CH sensor it's possible to control the CH water temperature

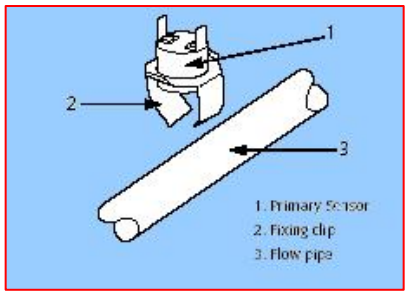


Figure 2.15 CH sensor

2.2.10 Automatic Air Vent (Air Purger)

- automatically discharges the air in the central heating circuit.
- placed in the body of the circulation pump or on the top of the expansion vessel.

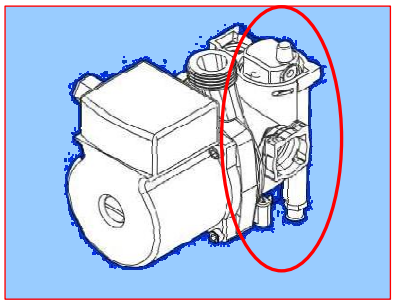


Figure 2.16 Air purger

2.2.11 Circulation Pump

- it circulates the Central Heating Water between combination boiler and radiators.
- The pumps may have modulation or different speeds and according to these speeds, the performance of the pump changes

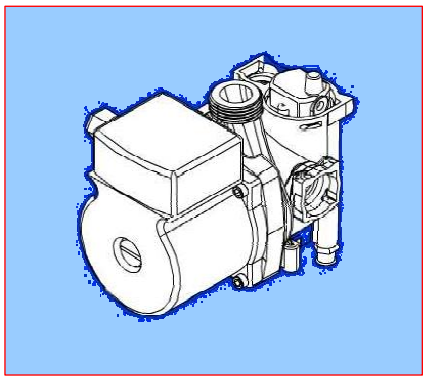


Figure 2.17 Circulation pump

2.2.12 CH Safety Valve

- if the system pressure exceeds 3,5 bar, it discharges CH water to decrease the system pressure.

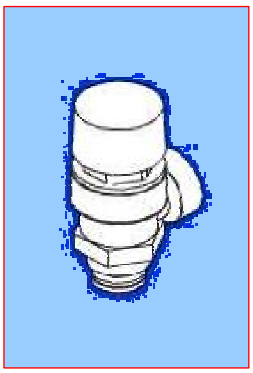


Figure 2.18 Safety valve

2.2.13 Gas Valve

- It opens or closes the gas channel and by increasing or decreasing the gas flow rate, regulates the burner pressure.

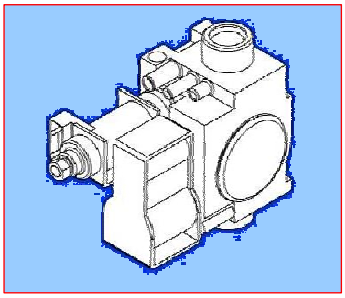


Figure 2.19 Gas valve

2.2.14 Secondary Heat Exchanger

- It transfers the heat energy of the water in the Central Heating circuit to the water in the Domestic Hot Water circuit.

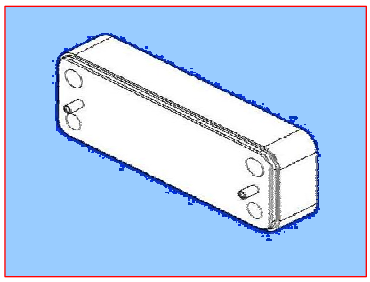


Figure 2.20 Secondary HE

2.2.15 Three-port valve

- It changes the direction of hot water flow from CH circuit to plate heat exchanger in case there is demand for DHW.

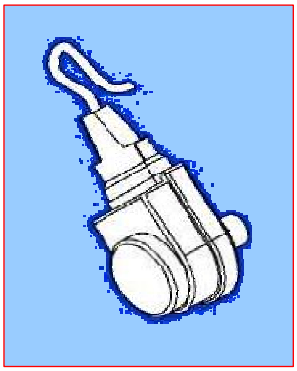


Figure 2.21 Three port valve

2.2.16 DHW Sensor

- The NTC is immersed to the body of the hydrolic kit.
- When the temperature of DHW increases, resistance decreases and conversely.
- With the help of DHW sensor it is possible to control the DHW temperature.

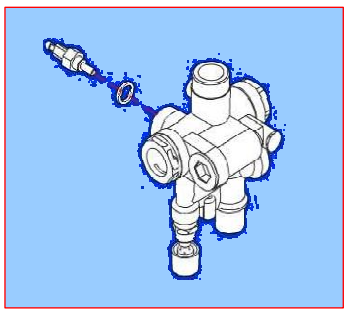


Figure 2.22 DHW sensor

2.2.17 Expansion Vessel

- During water heating, water increases in volume.
- 1 kg water at 20°C occupies a volume of 1.0018 dm^3 .1 kg water at 70°C occupies a volume of 1.0228 dm^3. It means 2% expansion in volume.
- It is thus necessary to absorb this expansion of volume to avoid breakdown the system

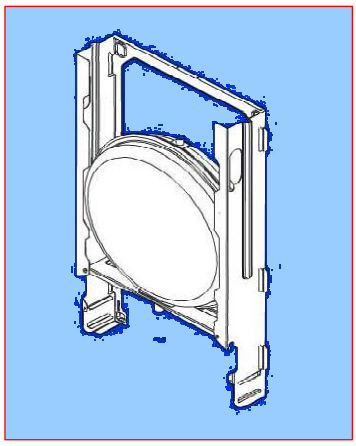


Figure 2.23 Expansion vessel

2.2.18 Discharge Valve

- It's placed in the body of the pump and is used to discharge the water in the CH circuit of the boiler.

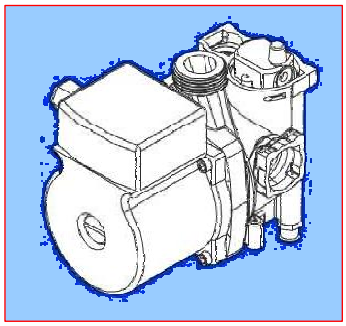


Figure 2.24 Discharge valve

2.2.19 Pressure Gauge

- it shows the system pressure
- nominal working range is between 1-3 bar

Figure 2.25 Pressure gauge

2.2.20 Turbine

- provides connection to mainboard about domestic hot water demand

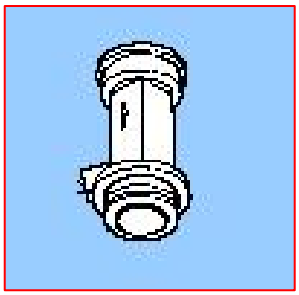


Figure 2.26 Turbine

2.2.21 Mainboard

- the main control component of the appliance
- every signal evaluates on the mainboard

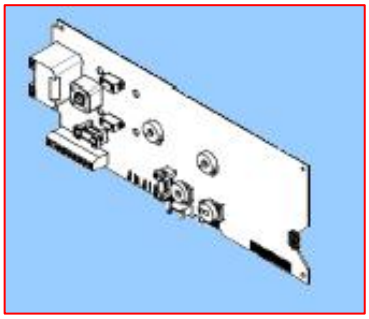


Figure 2.27 Mainboard

2.3 Standards Used for Gas-Fired Combi Boilers

Mainly, there are two relative standards that specify the requirements for Gas-fired combi boiler.

The European Standard EN297 – Gas-fired central heating boilers – Type B_{11} and B_{11BS} boilers fitted with atmospheric burners of nominal heat input not exceeding 70kW.

The European Standard EN483 – Gas-fired central heating boilers – Type C boilers fitted with atmospheric burners of nominal heat input not exceeding 70kW.

Namely, The European Standard EN483 is related with RSF combi boilers which is studied mostly in this thesis. And it specifies the requirements and test methods concerning, in particular, the construction, safety, fitness for purpose, the rational use of energy, as well as the classification and marking of gas-fired central heating boilers that are fitted with atmospheric burners, fan assisted atmospheric burners or premixed burners.

2.3.1 Efficiency Measurements according to EN 483: 1999/prA2

This standard and calculation methods applies to;

- Gas and oil burning hot water boilers of output 4 kW to 400 kW
- Low temperature boilers operating with return of 35 - 40°C
- Gas condensing boilers

And not applied to;

- Solid fuel boilers
- Instantaneous water heaters
- Boilers using fuels not commonly marketed
- Cookers designed mainly to heat the room where they are installed
- Gravity circulators < 6 kW
- On-off boilers

There are two determination with different operation modes;

“Full Load” operation

- Operating at nominal heat input - P_n
- Average water temperature of 70°C

“Part Load” operation

- Operating at 30% of nominal heat input – 30% P_n
- Average water temperature varies with appliance type

2.3.1.1 Useful Efficiency at the Nominal Heat Input (Full Load Operation)

The useful efficiency is determined at:

- The nominal heat input for boilers without a range rating device.
- The nominal heat input and the aritmetic mean of the maximum and minimum heat input for boilers with a range rating device.

First of all it is needed to determine nominal heat input value for a boiler;

The boiler is supplied with each of the reference gases for the boiler category at the normal pressure for this test. For boilers with a fixed output the adjustment shall not be changed for this test. Any adjusters shall be set to the position stated by the manufacturer. The volumetric flow rate of the gas V_{gas} obtained under these conditions (p_a, p_g, t_g, ρ) shall be corrected as if the test had been carried out under the reference test conditions (1013.25 mbar, 15°C, dry gas, and the corrected heat input is calculated using the following formulae:

If the volumetric flow rate of the gas, $\dot{V}_{gas}$ is measured:

$$Q_c = H_i * \frac{10^3}{3600} * \dot{V} * \sqrt{\frac{1013.25 + p_g}{1013.25} * \frac{p_a + p_g}{1013.25} * \frac{288.15}{273.15 + t_g} * \frac{\rho_{tgas}}{\rho_r}} \qquad (2.1)$$

whence:

$$Q_c = \frac{H_i * \dot{V}}{214.9} * \sqrt{\frac{(1013.25 + p_g) * (p_a + p_g)}{273.15 + t_g} * \frac{\rho_{tgas}}{\rho_r}} \qquad (2.2)$$

if the mass flow rate *m* is measured:

$$Q_c = H_i * \frac{10^3}{3600} * \dot{m}_{gas} * \sqrt{\frac{1013.25 + p_g}{p_a + p_g} * \frac{273.25 + t_g}{288.15} * \frac{\rho_r}{\rho_{tgas}}} \quad (2.3)$$

whence:

$$Q_c = \frac{H_i * \dot{m}_{gas}}{61.1} * \sqrt{\frac{(1013.25 + p_g) * (273.15 + t_g)}{(p_a + p_g)} * \frac{\rho_r}{\rho_{tgas}}} \quad (2.4)$$

where:

Q_c : is the corrected heat input (1013.25 mbar, 15°C, dry gas) relative on the net calorific value expressed in kW

V_{gas} : is the measured volumetric flow rate expressed under the humidity, temperature and pressure conditions at the meter, in m^3/h

m_{gas} : is the measured mass flow rate, in kg/h

H_i : is the appropriate, the net calorific value of dry reference gas at 15°C, 1013.25 mbar, in MJ/m^3 or MJ/kg

t_g : is the gas temperature at the meter, (°C)

ρ_{tgas} : is the density of the test gas

ρ_r : is the density of the reference gas

p_g : is the gas pressure at the meter, in mbar

p_a : is the atmospheric pressure at the time of the test, in mbar

And the heat input obtained under these conditions shall not differ by more than 5% from the nominal heat input value of the boiler. If this 5% is less than 500 W, a tolerance of 500 W is acceptable.

Useful efficiency at the nominal heat input

The boilers are installed in accordance with the technical instructions in a well-ventilated, draught free room (air speed less than 0.5 m/s) which has an ambient temperature of about 20°C. The boiler is protected from direct solar radiation.

Wall-mounted boilers are installed on a vertical test panel of plywood, or of a material with the same thermal characteristics, in accordance with the information in the manufacturer's instructions. The plywood panel shall be (25 ±1) mm thick and painted matt black; the panel dimensions are at least 50 mm greater than the corresponding dimensions of the boiler.

Except where otherwise stated, the boiler is connected to the shortest flue pipe with the smallest pressure loss stated by the manufacturer in his installation instructions. The boiler is supplied with the reference gas for the boiler category.

The measurement of the efficiency may begin once the boiler, eith the control thermostat put out of action, is at thermal equilibrium and the return and flow temperatures are constant. The hot water is passed into a vessel placed on scales (suitably tared before commencement of the test) and at the same time measurement of the gas rate (reading the meter) is started. Readings of the water return and flow temperatures are taken periodically so as to obtain a sufficiently accurate average.

Mass m_1 of water is collected during 10 minutes of the test. A further 10 minutes wait is required in order to evaluate the evaporation corresponding to the test period. Mass m_2 is obtained. $m_1 - m_2 = m_3$, the quantity of which note has to be taken in order to increase m_1 by the value corresponding to the evaporation, whence the corrected water mass $m = m_1 + m_3$.

The quantity of heat transferred by the boiler to the water collected in the vessel is proportional to the corrected mass m and to the difference between temperatures t_1 at the cold water inlet and t_2 at the boiler outlet.

The useful efficiency is determined from the formula below:

$$\eta_u = \frac{4.186 * m * (t_2 - t_1) + D_p}{10^3 * V_{r(10)} * H_i} * 100 \quad (2.5)$$

where:

η_u : is the useful efficiency (%)

m : is the corrected quantity of water expressed in kg

$V_{r(10)}$: is the gas consumption in m^3 measured during the test corrected to 15°C, 1013.25 mbar

H_i : is the net calorific value of the gas used (MJ/m^3) at 15°C, 1013.25 mbar, dry gas

D_p : is the heat loss from the test rig corresponding to the mean water flow temperature, expressed in kJ, taking into account the heat loss from the circulation pump.

The measurement uncertainties are chosen in a way which ensures a total uncertainty in the efficiency measurement of ±2 %.

Under these test conditions, the useful efficiency at the nominal heat input or at the Full Load operation shall be at least the value that has given in Table 2.1 and also the range of full load efficiency is given in Figure 2.28.

Table 2.1 Minimum efficiency values at the full load operation

	Minimum Efficiency at the Full Load operation	
	Average Water Temperature	**Minimum Net Efficiency (%)**
Standard boiler	70°C	$84 + 2\log P_n$
Low temperature boiler	70°C	$87,5 + 1,5\log P_n$
Gas condensing boiler	70°C	$91 + \log P_n$

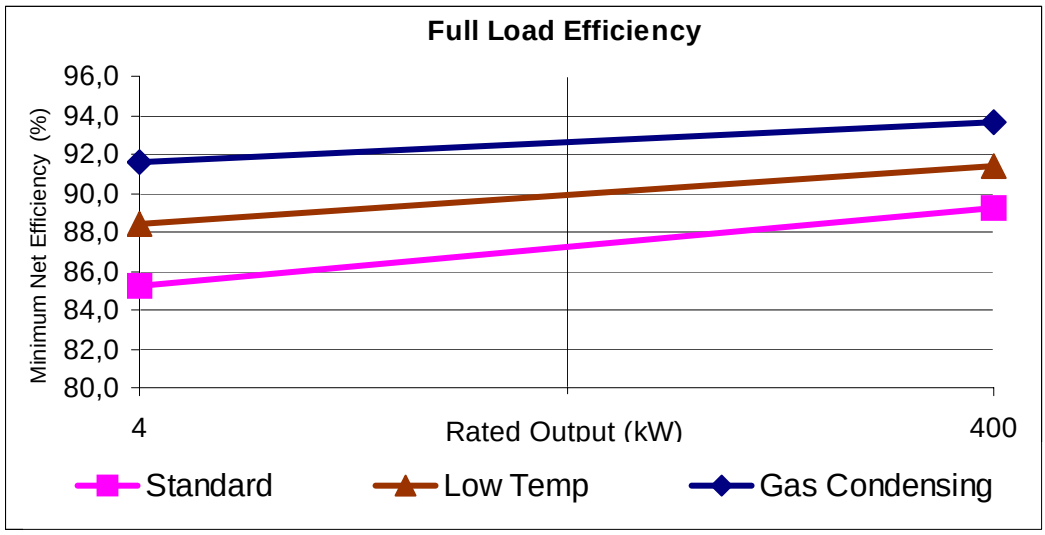


Figure 2.28 Full load efficiency range

Minimum full load efficiencies:

Standard boiler: 85.2% (4kW) to 89.2% (400kW)

Low Temperature: 88.4% (4kW) to 91.4% (400kW)

Gas Condensing: 91.6% (4kW) to 93.6% (400kW)

2.3.1.2 Useful Efficiency at Part Load Operation

First of all it has to known that determination of part load efficiency is too complicated and has much uncertainity. There are two methods to determine the useful efficiency at a load corresponding to 30 % of the nominal heat input. Both of the methods have points under consideration.

- Direct Method
- Indirect Method

2.3.1.2.1 Direct Method. The boiler is connected to the insulated test rig and supplied with one of the reference gases as for the determination of the useful efficiency. Throughout the test, the water volume rate is maintained constant within ±1 %, taking into account temperature variations and the pump operates continuously.

The boiler return temperature is held constant at (47 ± 1)°C, with a maximum variation in this temperature of ±1 K during the measurement period. The average water temperature shall be no less than 50°C. If the boiler control does not permit operation at a return temperature that is low enough, the test is carried out at the lowest return temperature compatible with the operation of the boiler.
A timepiece is fitted to the ambient temperature thermostat to obtain a working cycle of 10 minutes. The temperatures are measured continuously as close as possible to the flow and at the return of the boiler when (30 ± 2) % of the nominal input is drawn through the heat exchanger.

The boiler is considered to be in thermal equilibrium when the efficiency measurement of three consecutive cycles, combining any two results from three, does not vary by more than 0.5 %. In this case, the result is equal to the average value of at least three consecutive measurement cycles. For any other case, the average value shall be calculated from at least ten consecutive cycles. The respective gas and water

consumption's over complete cycles are measured. The shutdown and operating times are calculated as indicated in Table 2.3

The efficiency is determined using the following relation which may be also used for full load operation.

$$\eta_u = \frac{4.186 * m * (t_2 - t_1) + D_p}{10^3 * V_{r(10)} * H_i} * 100 \qquad (2.6)$$

A variation of ±1 %, with respect to the 30 % of the nominal heat input is permitted. For variations up to ±2 % it is necessary to carry out two measurements, one above and one below 30 % of the nominal heat input. The efficiency corresponding to 30 % is determined by interpolation.

2.3.1.2.2 Indirect Method. The test of "Full Load Operation" at the nominal heat input is repeated with a flow temperature of (60 ± 2)°C and a return temperature (40 ± 1)°C. The measured value η1 is noted.

If the boiler is fitted with a control system incorporating a main burner reduced rate (Q_2 = 0,3 Q_n), a test is carried out at the minimum heat input allowed by the control for a water flow temperature of (55 ± 2) °C and a return temperature of (45 ± 1) °C. The measured value is designated η_2.

If the boiler is fitted with a control system incorporating two main burner reduced rates, in which one has a heat input greater than 30 % of the nominal heat input and the other has a heat input less than 30 % of the nominal heat input, the efficiencies corresponding to the two inputs are determined.

The measured values are designated by:

- η_{21} for the larger heat input; $Q_{21} > 0{,}3\ Q_1$
- η_{22} for the smaller heat input; $Q_{21} < 0{,}3\ Q_1$

The symbols and quantities needed to calculate the efficiency at part load have given below:

Table 2.2 The symbols and quantities for part load efficiency

Operational phases of the main burner	Heat input (kW)	Operational time (sec)	Efficiency (%)
Full rate	Q_1	t_1	η_1
Reduced rate = $0.3Q_1$	Q_2	t_2	η_2
Reduced rate > $0.3Q_1$	Q_{21}	t_{21}	η_{21}
Reduced rate < $0.3Q_1$	Q_{22}	t_{22}	η_{22}

The efficiency is calculated from the ratio of the useful energy to the energy supplied by the gas during a ten minute cycle. Depending on the means of control, the following operating cycles can be identified, which correspond to the formula in Table 2.3

- Permanent operation with $Q_2 = 0{,}3\ Q_1$ (fixed reduced rate or modulating);
- Full rate/reduced rate operation (one or several reduced rates where the minimum heat input $Q_{22} < 0{,}3\ Q_1$);
- Operation with two reduced rates (where $Q_{21} > 0{,}3\ Q_1$ and $Q_{22} < 0{,}3\ Q_1$).

The efficiency is calculated as indicated in Table 2.3

Table 2.3 Part load efficiency calculation formulas

Conditions of operation	Heat input	Cycle time (s)	Meas	Useful efficiency (%)
30 % reduced rate	$Q_2 = 0.3*Q_n$	$t_2 = 600$	η_2	$\eta_u = \eta_2$
Full rate	$Q_1 = Q_n$	$t_1 = (180Q_1 - 600Q_{22}) / (Q_1 - Q_{22})$	η_1	$\eta_u = \frac{\frac{\eta_1}{100}Q_1t_1 + \frac{\eta_{22}}{100}Q_{22}t_{22}}{Q_1t_1 + Q_{22}t_{22}} * 100$
Reduced rate	$Q_{22} < 0.3*Q_n$	$t_{22} = 600 - t_1$	η_{22}	
Reduced rate 1	$Q_{21} > 0.3*Q_n$	$t_{21} = (180Q_{21} - 600Q_{22}) / (Q_{21} - Q_{22})$	η_{21}	$\eta_u = \frac{\frac{\eta_{21}}{100}Q_{21}t_{21} + \frac{\eta_{22}}{100}Q_{22}t_{22}}{Q_{21}t_{21} + Q_{22}t_{22}} * 100$
Reduced rate 2	$Q_{22} < 0.3*Q_n$	$t_{22} = 600 - t_{21}$	η_{22}	

Under these test conditions, the useful efficiency for a load corresponding to 30% of the nominal heat input or at the "Part Load Operation" shall be at least the value that has given in Table 2.4. And also the range of part load efficiency is given in Figure 2.29.

Table 2.4 Minimum efficiency values at the part load operation

	Minimum Efficiency at the Part Load operation	
	Average Water Temperature	**Minimum Net Efficiency (%)**
Standard boiler	> 50°C	$80 + 3\log P_n$
Low temperature boiler	40°C	$87.5 + 1.5\log P_n$
Gas condensing boiler	30°C (Return)	$97 + \log P_n$

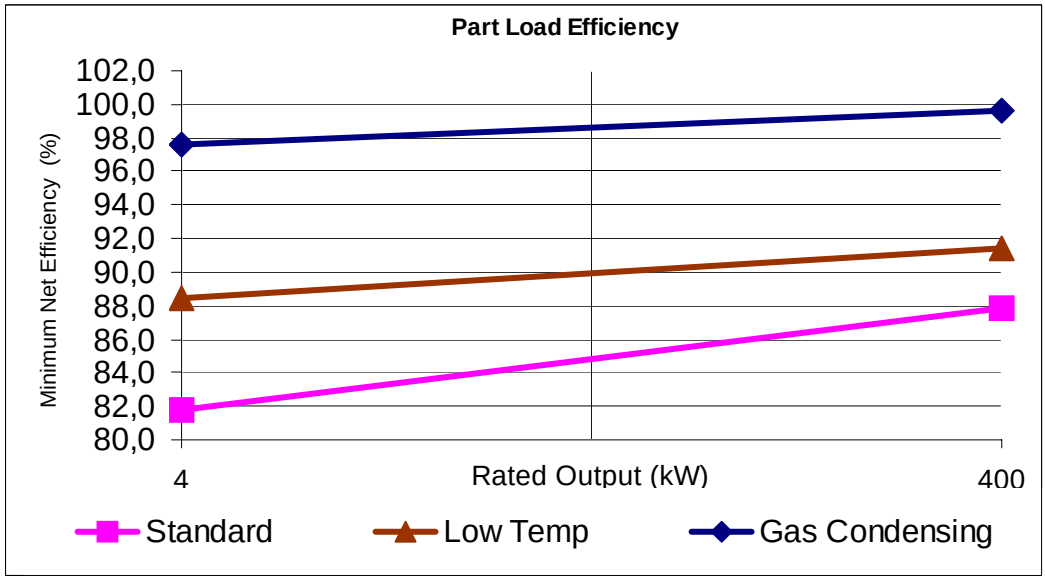


Figure 2.29 Part load efficiency range

Minimum Part Load Efficiencies:

Standard boiler: 81.8% (4kW) to 87.8% (400kW)

Low Temperature: 88.4% (4kW) to 91.4% (400kW) – Same as Full Load

Gas Condensing: 97.6% (4kW) to 99.6% (400kW)

2.3.2 Star Rating according to Efficiency

The labeling scheme works with stars " * ". One star is awarded to boilers with an efficiency at rated output higher than or equal to 84 + 2 log Pn, where Pn is the rated output in kW. Also the boiler has to have -at part load- an efficiency of 80 + 3 log Pn. For every 3 percentage points higher than these efficiency values an extra star is awarded. For instance, at two stars the boiler has an efficiency of 87 + 2 log Pn (rated output) and 83 + 3 log Pn (part load), etc.

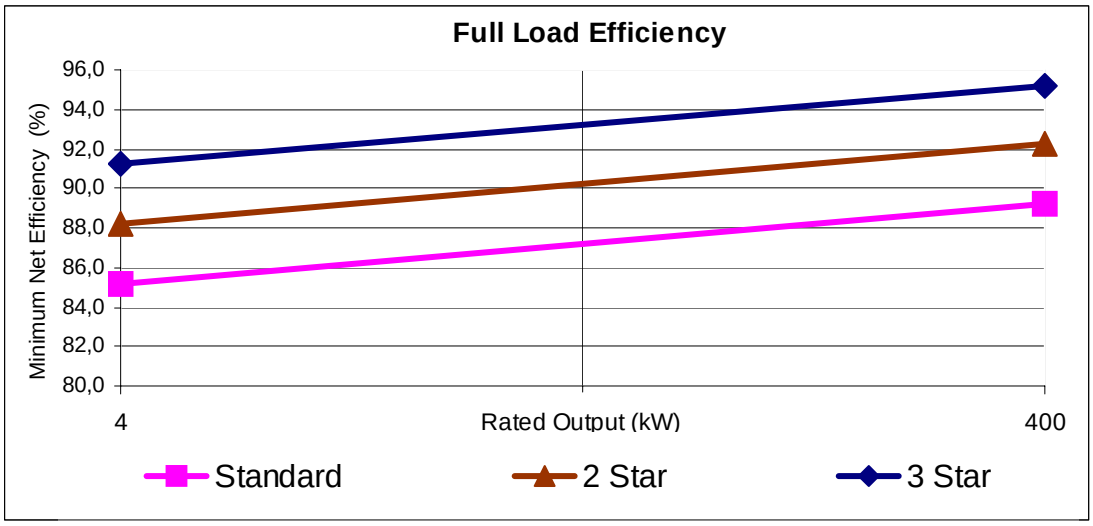


Figure 2.30 Star rating according to efficiency

2.4 Fuels Used for Gas-fired Combi Boiler

2.4.1 Classification

Gases are classified into three families, each family may be divided into groups as a function of the Wobbe index, according to the values given in Table 2.5 (BSI-EN437). Wobbe index is the ratio of the calorific value of a gas per unit volume and the square root of its relative density under the same reference conditions. The Wobbe index is said to be gross or net (W_s and W_i) according to whether the calorific value used is the gross or net calorific value (H_s and H_i).

1st family gases have the lowest Wobbe No, natural gases are 2nd family, and 3rd family gases have the highest Wobbe No (LPG). Gas families are subdivided for convenience of use in various gas distribution systems. The appliance category indicates those gas families (or subdivisions of families) which the appliance is designed to burn.

Table 2.5 Summary of gas families and groups as a function of the Wobbe indices

Gas Family	Groups	Gross Wobbe Index [MJ/m^3]	
		min	max
1st Family	Group a	22.4	24.8
2nd Family	Group H	45.7	54.7
	Group L	39.1	44.8
	Group E	40.9	54.7
3rd Family	Group B/P	72.9	87.3
	Group P	72.9	76.8
	Group B	81.8	87.3

2.4.2 Test Gases

The composition and principal characteristics of the reference gases corresponding to the gas families or groups are given in Table 2.6 (BSI-EN437).

Table 2.6 Characteristics of the reference gases – gas dry at 15 ˚C and 1013.25 mbar

Gas Family	Gas Group	Designation	Composition by volume [%]	W_i [MJ/m^3]	H_i [MJ/m^3]	W_s [MJ/m^3]	H_s [MJ/m^3]	d
1st family	Group a	G110	CH_4 = 26 H_2 = 50 N_2 = 24	21.76	13.95	24.75	15.87	0.411
2nd family	Group H	G20	CH_4 = 100	45.67	34.02	50.72	37.78	0.555
	Group L	G25	CH_4 = 86 N_2 = 14	37.78	29.25	41.52	32.49	0.612
	Group E	G20	CH_4 = 100	45.67	34.02	50.72	37.78	0.555
3rd family	Group B/P	G30	n-C_4H_{10} = 50 i-C_4H_{10} = 50	80.58	116.09	87.33	125.81	2.075
	Group B	G30	n-C_4H_{10} = 50 i-C_4H_{10} = 50	80.58	116.09	87.33	125.81	2.075
	Group P	G31	C_3H_8 =100	70.69	88.00	76.84	95.65	1.55

2.4.2.1 Natural Gas (NG)

Natural gas is a naturally occurring mixture of hydrocarbon and nonhydrocarbon gasses found in porous formations beneath the earth's surface, often in association with crude petroleum. The principal constituent of most natural gases is methane. Minor constituents are heavier hydrocarbons and certain nonhydrocarbon gases such as N_2, CO_2, H_2S and He. Oxygen and argon may be present but the amounts are usually very small. Some natural gases contain major proportions of saturated hydrocarbons heavier than methane. A few are devoid of heavier hydrocarbons, and many contain significant quantities of one or more of the nonhydrocarbons mentioned above. The natural gas of commerce, supplied to fuel gas markets, usually contains from 80 to 95% methane and lesser amounts of ethane and propane. Most of the remainder is nitrogen (Ringler & Segeler, 1965)

Analyses of natural gas distributed in Manisa is given in Appendix Table A-1.

2.4.2.2 Liquefied Petroleum Gases (LPG)

The term liquefied petroleum gas is applied to those hydrocarbons the chief components of which consist of propane, propylene, butane, butylene, and iso-butane, or mixtures thereof in any ratio or with air. These hydrocarbons can be liquefied under moderate pressure at normal temperatures, but they are gaseous under normal atmospheric conditions. Liquefied petroleum gases derived from natural gas are saturated hydrocarbons, whereas those derived from oil refinery gases may contain varying low amounts of unsaturated (olefin) hydrocarbons. Average content of propylene in propane is between 5-10 %. Butylene is used for various chemical purposes and is usually kept under 5 % in commercial butane gas.

2.5 Condensing Technology

2.5.1 Gas Condensing Boilers

During the burning process, heaters produce exhaust gases which contain a considerable proportion of steam. In conventional burners these fumes are released into the environment through the chimney. The exit flue gas temperature of a conventional gas fired boiler is usually high and a great amount of heat energy is lost to the environment.

In the case of gas condensing boilers, the remaining heat contained in the exhaust fumes is recovered via the condensation of the steam. If both sensible heat and latent heat can be recovered by adding a condensing heat exchanger, the efficiency of the boiler can be increased.

Theoretically, an additional energy content of 11% is made available in the case of natural gas condensate will be formed from the water vapor in the hot gas, if the temperature on the walls, on the hot gas side, of the heating surfaces falls below the water vapor dew point. (Figure 2.31) (Viessmann - Condensing Technology)

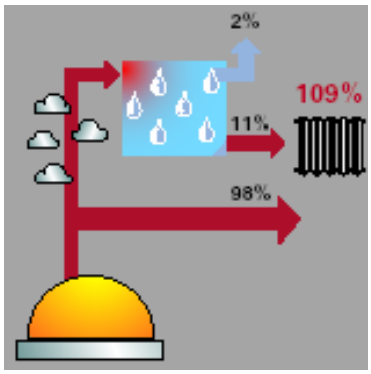


Figure 2.31

On the other hand the waste gas condenses, producing an “acidic” condensate (with a pH value of approx. 4.1 to 4.4). Since the condensate is acidic, additional standards have to be met by the chimney (it must be moisture-proof) and by the condensate collection device. Because of their specific technological features, condensing boilers usually have an efficiency of up to 107%. Over the next few years, an increase to up to 109% can be expected owing to the use of matrix burners. Even if condensing boilers are in most cases advertised as providing an efficiency of

107%, this figure must be expected to be lower in practice. On the one hand, the 107% mentioned above refers to combustion efficiency, without taking stand-by losses etc. into consideration.

Furthermore, a condensing boiler is often used in situations that do not guarantee optimum efficiency. This means that in most cases the heating system is not designed to provide the required low-temperature system. It has to be pointed out, however, that exhaust gases are cooled down and condensed in this case as well – though only partially. This means that, compared to conventional gas boilers, additional energy can be saved. According to expert opinion, given optimized utilization, one can today assume an annual utilization rate of approx. 94% for atmospheric burners and 106% for condensing boiler (laboratory values). The high annual utilization rate of condensing boilers results from the fact that these boilers show even higher efficiency when operated at part load. In practice, however, the values are lower. For this reason, one has to expect efficiency rates of approx. 90% for atmospheric burners and of 98-102% for condensing boilers (Viessmann - Condensing Technology).

The exit flue gas temperature of a conventional boiler is usually higher than 100 °C, sometimes as high as 150 °C, to avoid low temperature corrosion. At such temperatures, the water vapor entrained in the flue gases does not condense, and the latent heat cannot be reclaimed, which leads to a considerable heat loss.

Since the 1970s, condensing boilers have been developed and have found wide applications in the world. In such a boiler, the exit flue gas is reduced to such a low temperature that the water vapor can be condensed, and the latent heat released can be recovered. As such, the thermal efficiency of the boiler can be significantly increased. If the low heating value is still taken as the calculation basis, the efficiency can be as high as, or higher than 100%. Previous research has shown that SOx, NOx, dust and soot, etc., which are the constituents of the flue gas, can be partially, even totally, dissolved in the condensed water, and the pollutants emitted to

the environment can be noticeably reduced. Therefore, it is of great significance both to energy saving and environmental protection to utilize condensing boilers.

The potentially high operating efficiencies offered by condensing boilers can be achieved in practice, and this will ensure that for both new and replacement central heating installations, the condensing boiler will provide substantial savings in running costs when compared to the more conventional boiler. Most of the energy saving benefit of using condensing boilers can be achieved without recourse to excessive additional heat emitter surface or sophisticated controls. However, further developments in systems and controls for these appliances should be performed, and optimization of such schemes is necessary (Che, Gao & Liu, 2004)

The investigations show that many parameters of design and installation influence the performance of condensing boilers in the field. These parameters include pipe work design, controls, hot water cylinder design and boiler and system sizing, but the system designs for high efficiency condensing boilers do not need to be very different from current good practice for existing non-condensing boilers. The performance of condensing boilers was examined with water flow and return temperatures of 60 and 40 °C, respectively, in addition to the 80 and 60 °C used for a non-condensing unit. The results from the field study confirmed that substantially greater annual efficiencies were obtained with condensing boilers than with traditional boilers. A simple relationship was proposed that shows how boilers and radiator sizing affect annual efficiency. A payback of between 3 and 4.5 years for the installation of a condensing boiler can be expected.

While natural gas is used as the fuel of the boiler, as high as a 20% volumetric fraction of water vapor in the combustion products will be generated, which is much higher than LPG is used as the fuel. More water vapor in the flue gas means that more latent heat can be recovered, and the thermal efficiency of the boiler can be more greatly improved by decreasing the exit flue gas temperature. Thus, it is more profitable to recover the latent heat by condensing the water vapor in the flue gases

of a gas fired boiler. As is well known, hydrocarbons are the dominant components of natural gas. The volumetric content of natural gas is given below (see Table 2.7).

Table 2.7 The volumetric content of natural gas

Component	**CH_4**	**C_2H_6**	**C_3H_8**	**C_4H_{10}**	**CO_2**	**N_2**
Content (vol %)	**90**	**5,3**	**1,0**	**0,4**	**0,6**	**2,7**

2.5.1.1 Net and Gross Calorific Value

The net calorific value (Hi) describes the energy released during complete combustion, if the water created in the process is removed as vapor. The gross calorific value (Hs) defines the energy released during complete combustion including the evaporation heat contained within the hot gas water vapor (Viessmann - Condensing Technology).

In the past, the evaporation heat could not be utilized, since the relevant technical prerequisites were not yet available. Therefore, the net calorific value (Hi) was used as reference for all efficiency calculations. Referring to Hi and utilizing the additional evaporation heat can thus lead to standard efficiencies above 100 %. Because of guidelines, standard efficiencies in heating technology continue to refer to the net calorific value (Hi).

For example the higher heating value and the lower heating value of the natural gas are;

Hs (MJ/m3) = 37.78 and Hi (MJ/m3) = 34.02, respectively, with a difference of 3.76 (MJ/m3)

It means that the boiler efficiency can reach theoretically maximum value of;

37.78 / 34.02 * 100% = 111% based on lower heating value.

The different chemical consistencies of natural gas and fuel oil result in different water vapor temperatures, at which the hot gas water vapor condenses. According to the partial pressure of the water vapor in the flue gas, the dew point temperature is generally 56–60°C depending on the excess air ratio. The exit flue gas temperature can be reduced below the dew point temperature by a cooling medium to recover both the sensible heat and latent heat (Figure 2.32) (Viessmann – Condensing Technology).

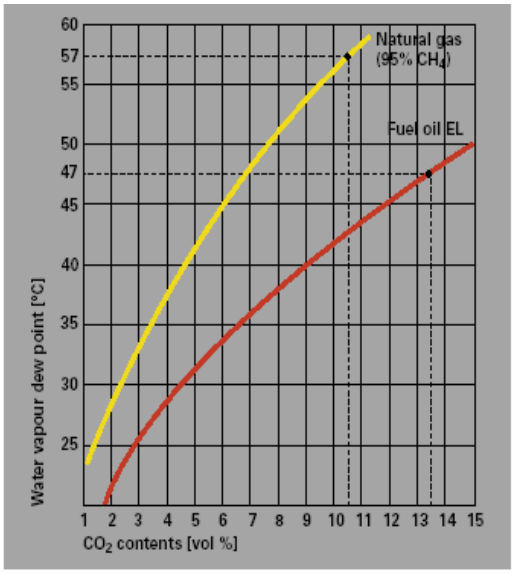


Figure 2.32 Water vapor dew point

The boiler efficiency based on lower heating value at various exit flue gas temperatures is given in Figure 2.33. Excess air ratio has taken its nominal value, 1.05

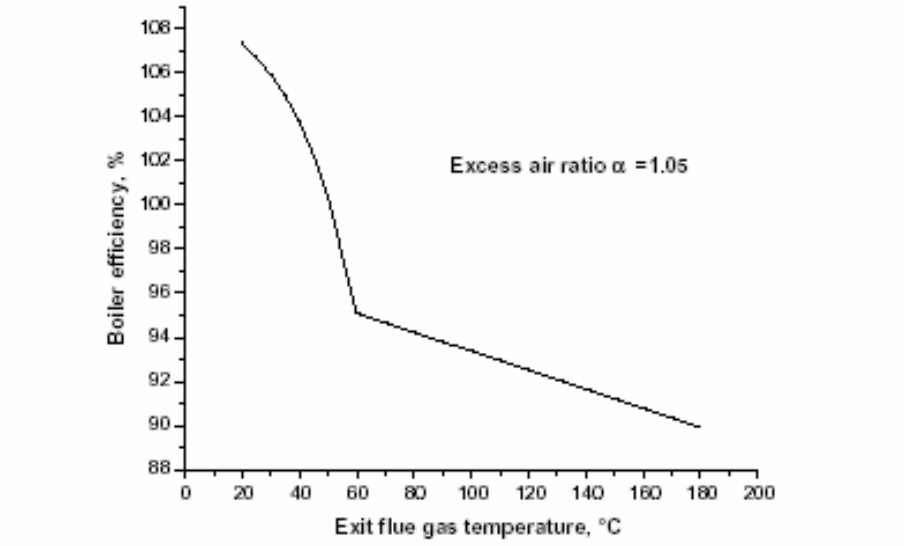


Figure 2.33 Boiler efficiency at various exit flue gas temperatures (Che, Gao & Liu, 2004)

It is easily seen from the figure that the boiler efficiency curve can be divided into two considerably different regions. The boiler efficiency varies relatively gradually in the temperature range of 60 - 180°C, and it is changed very rapidly in the temperature range of 20 - 60°C, which is mainly because the latent heat loss takes a greater role than the sensible heat. If the exit flue gas temperature is decreased to 20°C, the boiler efficiency based on lower heating value may reach over 107% theoretically.

The results show that the most economical exit flue gas temperature is 40 - 55°C when a conventional natural gas fired boiler is retrofitted into a condensing boiler simply by adding a condensing heat exchanger. It is feasible to use the return water of a heating system as the cooling medium of the condensing heat exchanger because the return temperature varies with the ambient temperature and is lower than the dew point of the water vapor in the flue gas in most periods of a heating season in some regions, which has been verified by retrofitted case.

Condensing technology is an efficient method of converting natural gas or fuel oil into useful energy by combustion. As with low-temperature technology, it follows the principle of operating the boiler only with that temperature, which is required to cover the current heating demand. While with low-temperature boilers, condensing hot gases and subsequent wetting of the heating surfaces must be avoided, condensing technology operates to quite different rules: the condensing of hot gases is actually highly desirable and is needed to turn the latent energy contained in water vapor, in addition to the sensible flue gas energy, into useful heat. Also, the residual heat exhausted via the flue will be substantially reduced, since, compared to low-temperature boilers, the flue gas temperature can be substantially reduced.

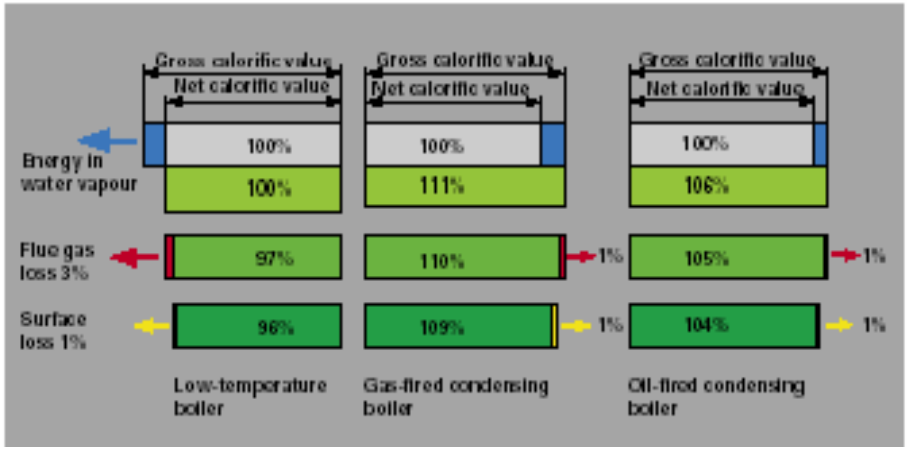


Figure 2.34 The comparison of the losses

In Figure 2.34, the comparison of the losses for low-temperature boilers, gas-fired condensing boiler and oil-fired condensing boiler has shown (Viessmann - Condensing Technology).

Through the reaction with the air component oxygen (O_2), the combustion of fuel oil or natural gas, both of which primarily consist of carbon (C) and hydrogen (H) compounds, creates carbon dioxide (CO_2) and water (H_2O). For natural gas (methane CH_4), the following simplified combustion equation applies:

$$CH_4 + 2\ O_2 \rightarrow 2\ H_2O + CO_2 + \text{energy}$$

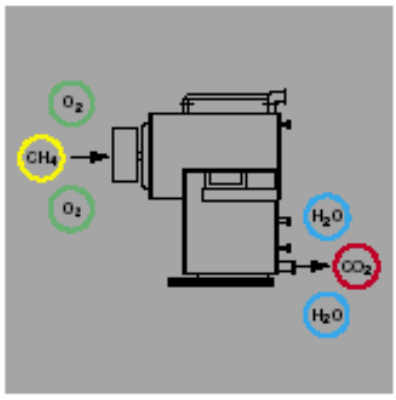


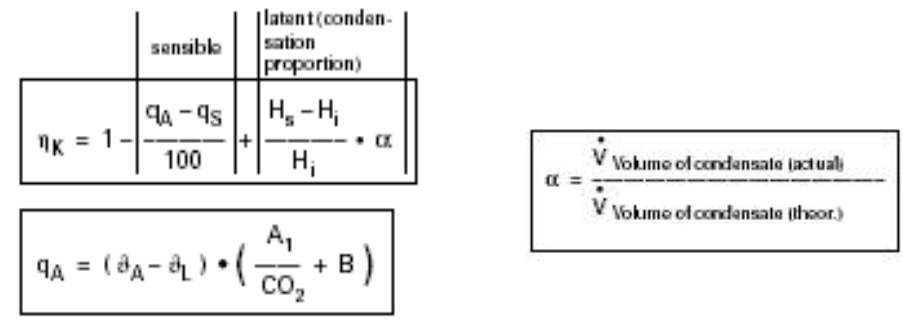
Figure 2.35

2.5.2 Influencing Variables for Condensing Technology

The energy gain between a condensing boiler and a low-temperature boiler is not just the result of the condensing energy gain, but is, to a large extent, a consequence of the low flue gas loss resulting from the low flue gas temperatures.

A basic energy assessment can be made using the boiler efficiency.

2.5.2.1 Boiler Efficiency η_K of Condensing Boilers



(2.7)

Influencing variables:

$\upsilon_A \rightarrow$ Flue gas temperature for condensing boilers, unlimited

$CO_2 \rightarrow CO_2$ concentration combustion quality subject to burner design

$\alpha \rightarrow$ Condensation value subject to boiler design and system

Key

η_K = Boiler efficiency [%]

υ_A = Flue gas temperature [°C]

υ_L = Air temperature [°C]

A_1 = Fuel correction value

B = Fuel correction value

CO_2 = Carbon dioxide content [%]

q_A = Flue gas loss [%]

q_S = Radiation loss [%]

α = Condensate value

Hs = Gross calorific value

Hi = Net calorific value

Table 2.8 Constant values

	Fuel oil EL	Natural gas	Town gas	Coking gas	LPG and LPG-air mixtures
A_1	0.5	0.37	0.35	0.29	0.42
A_2	0.68	0.66	0.63	0.60	0.63
B	0.007	0.009	0.011	0.011	0.008

Compared with a conventional boiler, the boiler efficiency formula is expanded by the condensation proportion. The condensation proportion is defined through the fuel-specific constant values Hs and Hi, as well as the variable condensation α. It provides the ratio between the actual volume of condensate in a condensing boiler and the theoretically possible volume of condensate (Viessmann - Condensing Technology).

The higher the actual volume of condensate, the more effective the condensing boiler. The lower the flue gas temperature, the greater the volume of condensate and therefore the condensate α. At the same time, the lower flue gas temperature, for example compared to a low-temperature boiler, also reduces the flue gas loss. This means that condensing boilers achieve an additional improved energy utilization through lowering the flue gas losses, as well as gaining condensation energy.

2.5.3 Productivity of the Condensing Technology

Condensation energy can be utilized not only with reduced loads, i.e. low heating system temperatures. Even with heating systems designed for 75/60 °C, the actual temperature in the return falls below the dew point when operating at load levels up to 90 % and higher or outside temperatures as low as –11 °C, so that hot gas water vapor can condense. This enables the system to function inside the condensing range for more than 90 % of its operation, even with a high design temperature of 75/60 °C acc. to Figure 2.36 (Viessmann - Condensing Technology). Even better are the circumstances of low- temperature heating systems such as under-floor heating systems (40/30 °C), where condensing operation is achieved all the year round.

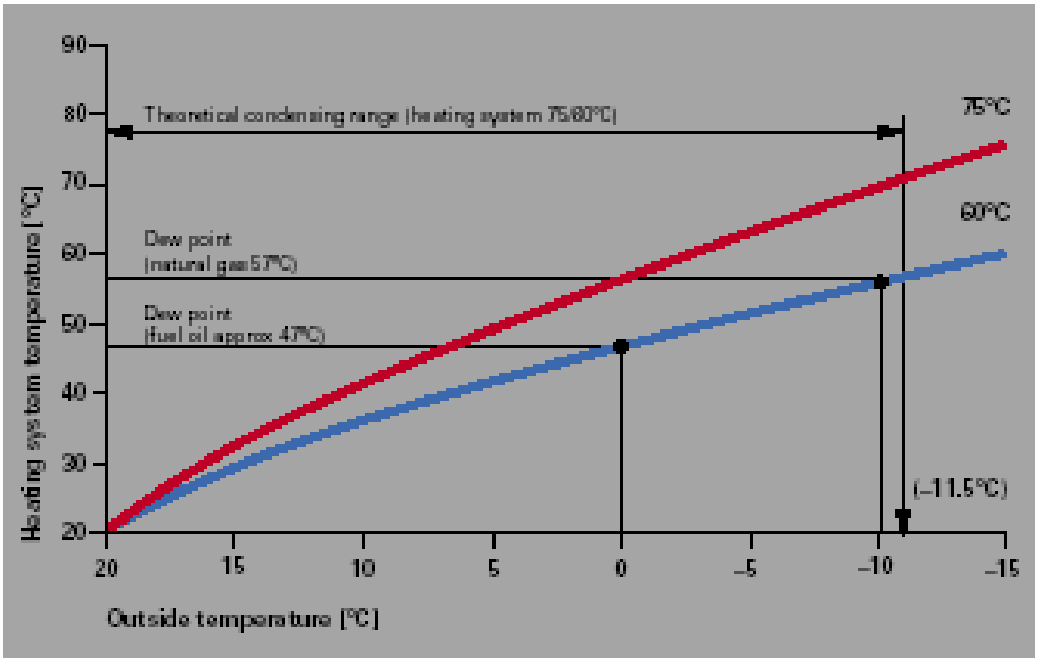


Figure 2.36 Flow / return temperature, subject to outside temperature

Experience shows, that older buildings are frequently equipped with oversized radiators. This over sizing is partly due to an over-generous design during the initial

installation, and also to the thermal insulation measures introduced over the passage of time: Retrofitted doubled glazed windows, cladding and roof insulation have substantially reduced the heating demand. However, the radiator size has remained unchanged. This enables the flow and return temperatures to be significantly reduced from the original design (e.g. 90/70 °C).

The boiler design should ensure that at the lowest likely outside temperature, the heat demand can be fully covered. However, such low temperatures are only rarely reached during daytime operation, hence the boiler must only provide its full output for a few days each year. For the remainder of its operating time, only a fraction of its rated output needs to be provided. So the part load efficiency of the boiler becomes more important than full load efficiency.

The advantages offered by condensing technology are particularly significant in the case of low average loads: the boiler operating at a constant temperature leads to substantial losses with reducing loads, since the boiler temperature must be maintained at a high level, even if the heating system temperature only demands a low output. One result is a substantially higher proportion of radiation losses as part of the overall energy requirement, hence a reduced level of efficiency.

Condensing boilers, on the other hand, provide an especially high level of efficiency in cases of low average loads, because the condensing effect is particularly successful due to the low temperature level of the heating water.

Figure 2.37 illustrates a comparison of the efficiencies for various types of boiler (Viessmann - Condensing Technology).

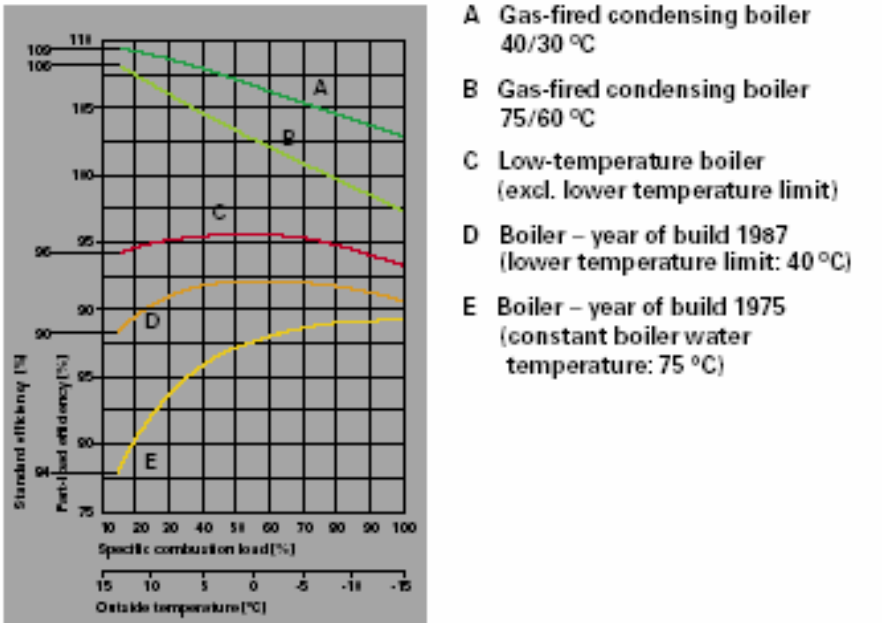


Figure 2.37 Standard efficiency for various boiler designs

2.5.4 Important Criterias for Optimum Gain

2.5.4.1 Boiler Design

Utilization of condensing technology improved with increasing condensation of the water vapor contained in hot gases. Only this enables the latent energy in hot gas to be converted into useable heating energy. Conventional boiler designs are unsuitable for this task.

In conventional boilers, the heating surfaces are designed to prevent condensation of hot gases inside the boiler. Condensing boiler design is quite different: Hot gases are guided downwards as near as possible to the return connection, leading to maximum cooling. Hot gas and heating water should flow in a counter current pattern in order to utilize the low temperature of the return water to provide maximum cooling of the exhausted hot gases.

The selection of suitable materials should ensure, that the condensate created will not cause the boiler to suffer corrosion damage. During combustion, constituents of the fuel (fuel oil or natural gas) and of the combustion air create compounds, which shift the pH value (degrees of alkalinity or acidity) of the condensate up to acid levels. Carbon dioxide can form from the CO_2 created during combustion, and the nitrogen N_2 contained within the air reacts to become nitric acid. Therefore, all heat exchanger surfaces which come into contact with condensate, must be made from materials which remain unaffected by the chemical attack of the constituents of condensate (Viessmann - Condensing Technology).

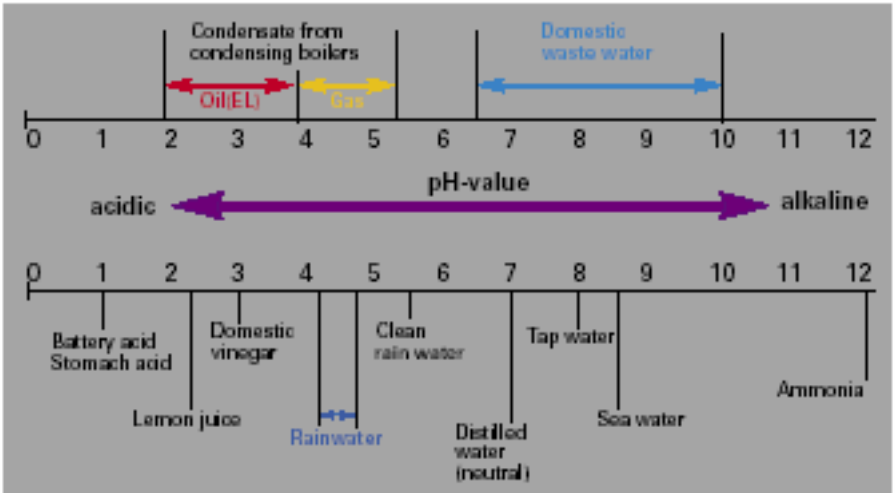


Figure 2.38 pH Value of Different Substances

For some years now, stainless steel has proved to be the ideal material for this purpose. For fuel oil and natural gas, different stainless steel alloys are available (alloying elements are, amongst others, chromium, nickel, molybdenum, titanium), which were matched to the characteristics of condensate. This enables these materials to withstand the corrosive attack of condensate without further treatment.

2.5.4.2 CO_2 Content

For the efficient utilization of condensing technology it is important, that combustion takes place with a high CO_2 content or a low level of excess air, since the dew point is influenced by the CO_2 content of the hot gases . The dew point should be kept as high as possible, to allow condensation to be achieved even in heating systems operating with a high return temperature. Therefore, the designer's aim should be a CO_2 content in the hot gases – in other words little excess air – which is as high as possible.

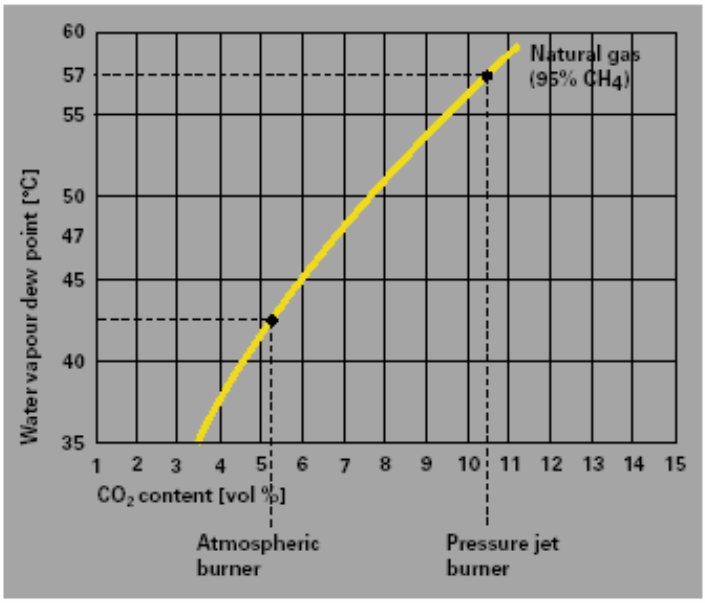


Figure 2.39 Water vapor dew point subject to CO_2 content

For this reason, atmospheric burners should not be utilized, as these – because of their high level of excess air – tend to operate with low CO_2 values, which lead to low condensation temperatures within the hot gas. Due to thermal up-draught, the residual energy in the flue gas at flue gas temperatures of 50 °C or below is generally insufficient to ensure the function of the chimney or the flue gas system by natural draught. In this context it is important, that fans for modulating devices are speed controlled to enable the air volume to be matched to the gas volume flow. Only by these means can the high CO_2 content be ensured in modulating operation too. (Fig 2.39) (Viessmann - Condensing Technology).

2.5.4.3 Condensate Treatment

Condensate created in the boiler during heating must be discharged. Subject to the return temperature, a certain flue gas temperature .A results, which in turn influences the condensation α. α becomes 1, when the total theoretical condensate volume is created (full condensation) (Fig 2.40) (Viessmann - Condensing Technology).

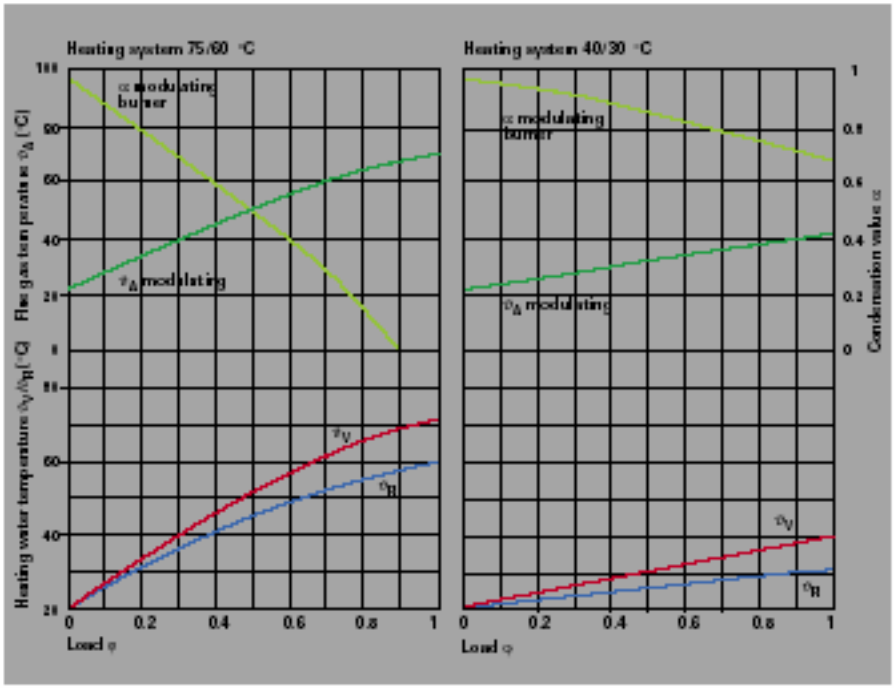


Figure 2.40 Condensate volume

For gas-fired condensing boilers below 25 kW, condensate may be directly drained into the public sewer. The proportion of condensate of the total waste water is so small, that domestic waste water provides sufficient dilution.

2.5.5 Evaluation about Condensing Technology

In all cold and moderate EU climate zones the 4 star condensing boiler or the low temperature boiler might be economical. But only in regions with a short heating season of 2 to 4 months, (like some parts of Turkey) there may be a case to fall back on so-called 'improved efficiency boilers', but already now the price difference between an 'improved efficiency boiler' (> 84 + 2 log Pn % efficiency) and a condensing boiler is small.

As a result we can say that; condensing boilers are ideal for cold climate zones and also large properties, properties with extensive heating systems or large heat load requirements, e.g. swimming pools. If there is a large difference between the boiler flow and return temperatures, then a condensing boiler could be considered. It would be working at its peak efficiency for long operating periods. Condensing boilers cannot and should not be used for every application.

2.5.5.1 Physical Restrictions of Condensing Boilers

Most condensing boilers are of the balanced flue type and as such they need to be sited on an external wall. Due to the condensate produced from the flue gases they must be permanently connected to a foul drain, this should run inside the property to prevent the condensate freezing.

The main feature of condensing boilers is that they sometimes produce a plume of water vapor around the flue terminal. Although this is in no way harmful it can be perceived as a nuisance. Therefore the sitting of the flue terminal is important so that the plume does not discharge over neighbouring properties and preferably not within full view of windows.

The other reasons why people are not choosing condensing boilers might be:

- lack of information/training of the installers (and consumers)
- in retrofitting applications the existing chimney is often not fit and it may not always be possible to find an economical solution for the ‘wet’ flue gases from a condensing boiler
- in new housing with a very critical cost factor (e.g. social housing) the extra cost for a condensing boiler may still be a factor of importance.

CHAPTER THREE

THEORETICAL ANALYSIS

3.1 Combustion

Combustion processes have been, are and will be for the near future, the prime generator of energy in our civilization, which is burning fuels and gases at an ever-increasing rate. The processes must be managed well for the sake of the environment and the sustainability of civilization.

In a gas appliance, flammable gas is burnt in a controlled manner. Other forms of combustion process are possible, for example explosive combustion. In most cases, however, this will be uncontrolled combustion and more than likely also an undesirable form of combustion. Although the process of explosive combustion may appear different from that of controlled combustion, there are substantial similarities.

Conventional fuels consist mainly of two elements – carbon and hydrogen. Burning any flammable substance requires oxygen. During combustion, they combine with oxygen to produce heat. This oxygen is generally obtained from the atmosphere. Another requirement is a source of ignition in order to initiate the combustion process. Combustion is accompanied by flames, and sensible heat is liberated. This heat is carried in the products of the combustion process. Generally speaking, combustion of a flammable gas involves an exothermic chemical process that can be represented as follows (Saxon, 2003):

Flammable gas + Oxygen $\Rightarrow$ Combustion products + Heat

In order to control the combustion process, there are some important topics which playing a part in that process. They also will be detailed in following parts of this thesis.

3.1.1 Combustion Reaction Equation

Combustion is the process that takes place when a substance is broken down in the presence of oxygen. Solid fuels, like wood and coal, first have to be made gaseous. The gaseous components liberated by this process are then enable to combine with oxygen. The combustion process liberates heat. Since this process takes place in a very short space of time, the liberated heat is at a high temperature. The heat is therefore useful, for example, for heating processes. This is the distinction between the process of combustion and the familiar process of corrosion, which also involves the combination of a substance with oxygen and the production of heat. The corrosion process, however, generally proceeds many times more slowly, with the result that the heat which is liberated is at a low temperature and is not useful for other purposes. The combustion process can be represented in a reaction or chemical equation, with the original substances, i.e. fuel and oxygen or air, on the left and the combustion products on the right (Saxon, 2003).

In writing the reaction equation, it should be borne in mind that no material is lost and no new material is formed during the reaction process. In other words, the numbers of atoms of a substance in the left-hand half of the equation must be equal to the numbers of atoms of the same substance in the right-hand half or, putting it another way, the reaction equation must balance.

The reaction equation for methane (the main component of natural gas) looks like this:

$$CH_4 + 2\ O_2 \Rightarrow CO_2 + 2\ H_2O \qquad (3.1)$$

The equation (3.1) shows that 1 molecule of methane combines with 2 molecules of oxygen and that the reaction produces 1 molecule of carbon dioxide (CO_2) and 2 molecules of water vapor (H_2O).

It is not grossly inaccurate to say that 1 m^3 of gas of whatever kind will always contain approximately the same number of molecules. Thus in 1 m^3 of CH_4, there are the same number of molecules as in 1 m^3 of O_2. It follows from this that, if the molecular ratio in a reaction equation is 1 : 2.

The above reaction equation can be also written as:

$$1\ m^3\ CH_4 + 2\ m^3\ O_2 \Rightarrow 1\ m^3\ CO_2 + 2\ m^3\ H_2O \qquad (3.2)$$

As already mentioned, the oxygen needed for combustion is generally drawn from the atmosphere. Dry air contains around 20.95% O_2, the remainder being almost entirely made up of N_2 (nitrogen). From the reaction equation immediately above, we can see that to burn 1 m^3 of CH_4 we need 2 m^3 of O_2. Therefore, in order to obtain this volume of oxygen from the atmosphere, it is needed:

$$2 / 0.2095 = 9.55\ m^3$$

The volume of N_2 in this air is than obviously $9.55 - 2 = 7.55\ m^3$. Using these figures, now the combustion equation for CH_4 and air can be written as:

$$CH_4 + 2\ O_2 + 7.55\ N_2 \Rightarrow CO_2 + 2\ H_2O + 7.55\ N_2 \qquad (3.3)$$

Unfortunately, natural gas, and also landfill gas, for example, is not made up of CH_4 alone but a mixture of hydrocarbons and inert components (non-flammable substances). Hydrocarbon compounds are represented by the chemical formula CxHy, in which x is the number of carbon atoms and y is the number of hydrogen atoms in the molecule. Therefore, in order to write the complete combustion equation for natural gas, we have to write the combustion equations for all separate components and add them together according to the proportion in which they are present.

To do this, we can also make use of the following generally applicable combustion equation for hydrocarbons and dry air (in m^3) (Saxon, 2003).

$$C_xH_y + \underbrace{\left(x+\frac{y}{4}\right)(O_2+3.76N_2)}_{\text{Theoretical air requirement}} \Rightarrow x.CO_2 + \frac{y}{2}.H_2O + 3.76\left(x+\frac{y}{4}\right)N_2 \qquad (3.4)$$

The theoretical air requirement is the stoichiometric quantity of air, which is precisely required amount of air as an oxidizer for a complete combustion. Ideally, combustion breaks down the molecular structure of the fuel; the carbon oxidizes to carbon dioxide (CO_2) and the hydrogen to water vapor (H_2O). But an incomplete process creates undesirable and dangerous products. To ensure complete combustion, even modern equipment with many features must operate with excess air. That is, more air (carrying about 21% oxygen by volume) is passed through the burner than is chemically required for complete combustion. This excess air speeds up the mixing of fuel and air. On one hand, this process ensures that nearly all the fuel receives the oxygen it needs for combustion before it is chilled below combustion temperatures by contact with heat exchange surfaces. It also prevents fuel that is not burned completely from exploding within the boiler.

On the other hand, excess air wastes energy by carrying heat up the stack. A fine line exists between combustion efficiency and safety in ensuring that as little excess air as possible is supplied to the burner.

3.1.1.1 Oxygen Requirements

For natural gas, commercial propane and butane/air; we can show oxygen requirement as seen in Table 3.1, 3.2 and 3.3 (Saxon, 2003).

Table 3.1 Oxygen requirement for natural gas

Constituent	**% by volume**	**Chemical Equation for Combustion**	**Volumes of Oxygen Required to Burn**
N_2	2.7	-----	
CO_2	0.6	-----	
CH_4	90.0	$CH_4 + 2O_2 = CO_2 + 2H_2O$	(90/1)*2 = 180.0
C_2H_6	5.3	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	(5.3/2)*7 = 18.6
C_3H_8	1.0	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	(1/1)*5 = 5.0
C_4H_{10}	0.4	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	(0.4/2)*13 = 2.6
Totals	100		**206.2**

So 1 m^3 of natural gas requires 2.062 m^3 of oxygen – near enough 2 m^3 .

Table 3.2 Oxygen requirement for commercial propane

Constituent	**% by volume**	**Chemical Equation for Combustion**	**Volumes of Oxygen Required to Burn**
C_3H_8	85.9	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	(85.9/1)*5 = 429.5
C_3H_6	12.0	$2C_3H_6 + 9O_2 = 6CO_2 + 6H_2O$	(12/2)*9 = 54
C_2H_6	1.5	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	(1.5/2)*7 = 5.25
C_4H_{10}	0.6	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	(0.6/2)*13 = 3.9
Totals	100		**492.65**

So 1 m^3 of propane requires 4.926 m^3 of oxygen – near enough 5 m^3 .

Table 3.3 Oxygen requirement for butane/air

Constituent	% by volume	Chemical Equation for Combustion	Volumes of Oxygen Required to Burn
O_2	17.01	-----	-17.01
N_2	63.99	-----	
C_3H_8	2.5	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	(2.5/1)*5 = 12.5
C_4H_{10}	16.5	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	(16.5/2)*13 = 107.25
Totals	100		**102.74**

So 1 m^3 of butane requires 1.0274 m^3 of oxygen – near enough 1 m^3 .

Since the atmosphere consists of 21% oxygen the air requirements for the complete combustion of 1 m^3 of gas are;

Natural Gas

2.06 x (100 / 21) = 9.8 m^3 of air roughly 10 m^3

Commercial propane

4.926 x (100 / 21) = 23.45 m^3 of air roughly 24 m^3

Butane/Air

1.027 x (100 / 21) = 4.89 m^3 of air roughly 5 m^3

3.1.1.2 Products of Combustion

It does not follow that because 1 m^3 of natural gas requires 9.8 m^3 of air, the volume of products will be 1 + 9.8 = 10.8 m^3 of products of combustion.

For example,

C_3H_8	+	$5O_2$	=	$3CO_2$	+	$4H_2O$
1 volume	**+**	**5 volumes**	**=**	**3 volumes**	**+**	**4 volumes**
		6 volumes	=	7 volumes		

Obviously the volumes are not equal whether we use O_2 or air volumes. But what the equation does show is that 1 m^3 of propane combined with 5 m^3 of oxygen will produce 3 m^3 of carbon dioxide and 4 m^3 of water vapor. The amount of the products of combustion produced in each case can be calculated from the combustion equations.

For natural gas, commercial propane and butane/air; combustion products can be seen in Table 3.4, 3.5 and 3.6 (Saxon, 2003).

Table 3.4 Products of combustion for natural gas

Const.	**% by volume**	**Chemical Equation for Combustion**	**Products of Combustion**	
			Carbon dioxide	**Water vapor**
N_2	2.7	-----	-----	-----
CO_2	0.6	-----	0.6	-----
CH_4	90.0	$CH_4 + 2O_2 = CO_2 + 2H_2O$	(90/1)*1 = 90.0	(90/1)*2 = 180.0
C_2H_6	5.3	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	(5.3/2)*4 = 10.6	(5.3/2)*6 = 15.9
C_3H_8	1.0	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	(1/1)*3 = 3.0	(1/1)*4 = 4.0
C_4H_{10}	0.4	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	(0.4/2)*8 = 1.6	(0.4/2)*10 = 2.0
Totals	100.0		**105.8**	**201.9**

Table 3.5 Products of combustion for commercial propane

Const.	% by volume	Chemical Equation for Combustion	Products of Combustion	
			Carbon dioxide	Water vapor
C_3H_8	85.9	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	(85.9/1)*3 =257.7	(85.9/1)*4 = 343.6
C_3H_6	12.0	$2C_3H_6 + 9O_2 = 6CO_2 + 6H_2O$	(12/2)*6 = 36	(12/2)*6 = 36
C_2H_6	1.5	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	(1.5/2)*4 = 3.0	(1.5/2)*6 = 4.5
C_4H_{10}	0.6	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	(0.6/2)*8 = 2.4	(0.6/2)*10 = 3.0
Totals	100.0		**299.1**	**387.1**

Table 3.6 Products of combustion for butane/air

Const.	% by volume	Chemical Equation for Combustion	Products of Combustion	
			Carbon dioxide	Water vapor
O_2	17.01	-----	-----	-----
N_2	63.99	-----	-----	-----
C_3H_8	2.5	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	(2.5/1)*3 = 7.5	(2.5/1)*4 = 10.0
C_4H_{10}	16.5	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	(16.5/2)*8 = 66.0	(16.5/2)*10 = 82.5
Totals	100.0		**73.5**	**92.5**

Total volumes; to provide the 2.06 m^3 of oxygen required to burn 1 m^3 of natural gas, it is needed 9.8 m^3 of air. The difference, 9.8 – 2.06 = 7.74 m^3, is nitrogen.

So, when 1 m^3 of natural gas is burned, the total volume of gases which are leaving the combustion chamber of the appliance is:

Carbon dioxide	1.058
Water vapor	2.019
Nitrogen from gas	0.027
Nitrogen from air	7.74
Total	10.844 m^3 or roughly 11 m^3

For commercial propane the total volume of product is:

Carbon dioxide	2.991
Water vapor	3.871
Nitrogen from air	18.524
Total	25.386 m^3 or roughly 26 m^3

For butane/air the total volume of product is:

Carbon dioxide	0.735
Water vapor	0.925
Nitrogen from gas	0.639
Nitrogen from air	3.863
Total	6.162 m^3 or roughly 6 m^3

3.1.2 Excess Air

Stable combustion requires three inputs: fuel, oxygen, and a source of ignition. If the combustibles themselves can provide this third element as they burn, the source of ignition can be turned off. In complete combustion, the carbon in the fuel is converted to carbon dioxide, hydrogen in the fuel to water vapor, and sulfur and nitrogen in the fuel and in the air supplied for combustion are converted to their oxides. In theory, there is a precise and predictable amount of oxygen necessary to completely burn a given amount of fuel, called stoichiometric air. In practice, however, burning conditions are never ideal. Therefore, more air must be supplied to completely burn the fuel. The amount of air above the theoretical requirement is referred to as "excess air" (Figure 3.1) (Dockrill & Friedrich, 2001).

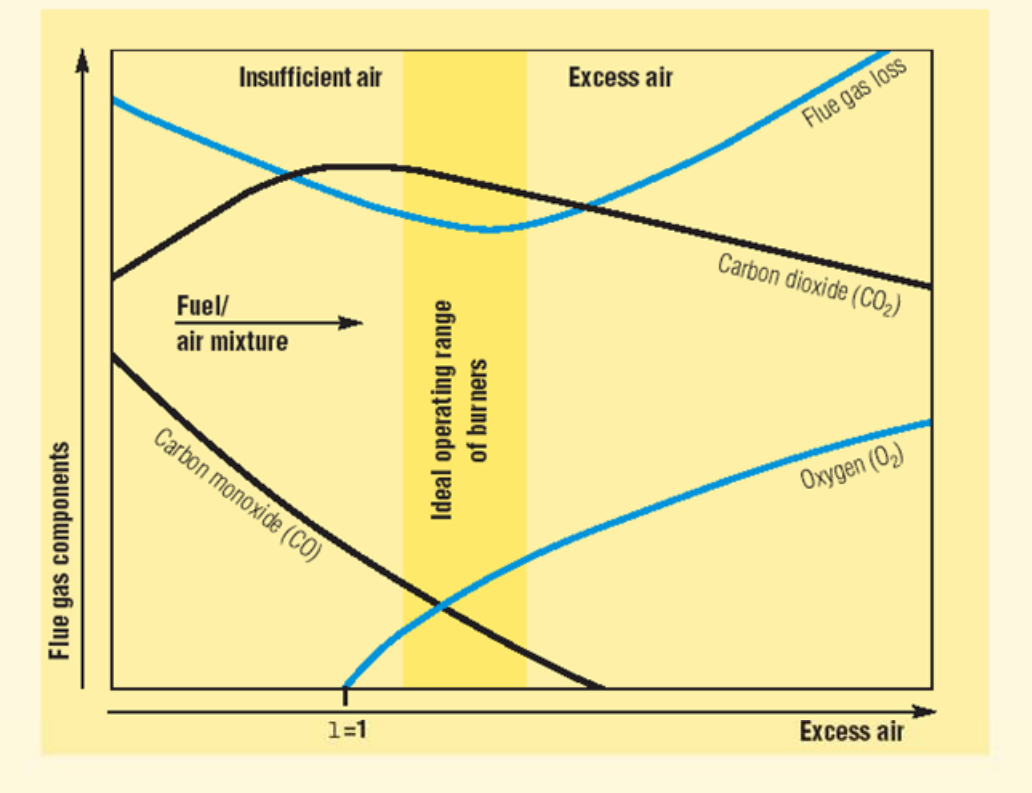


Figure 3.1 Combustion diagram for ideal operating ranges

If an insufficient amount of air is supplied to the burners, unburned fuel, soot and smoke, and carbon monoxide (the incomplete conversion to carbon dioxide) appear in the exhaust from the boiler stack. These can result in heat transfer surface fouling, pollution, lower combustion efficiency, flame instability (i.e., the flame blows out), and the potential for an explosion. To avoid these costly and potentially unsafe conditions, boilers are normally operated at excess air levels. This excess air level also provides operating protection from an insufficient oxygen condition caused by variations in fuel composition, and "operating slop" in the fuel/air control system on the boiler.

Considerable energy is lost through flue exhaust in the heat exhausted by the products of combustion, and by unburnt combustibles due to incomplete combustion. Generally, losses due to unburnt combustibles are negligible. More air than the theoretical minimum requirement for complete combustion is usually supplied for

the following reasons. To ensure stable combustion and prevent the formation of carbon monoxide (CO) which is highly poisonous and explosive.

To allow for variations in the required air-to-fuel ratio due to combustion air temperature, pressure and humidity changes. To allow for slight variations in the chemical composition of the fuel gas and its supply pressure. To provide good air–fuel mixing in order to ensure complete combustion over the operating range of the burner. To allow for operating range inconsistencies of fuel-to-air ratio control equipment, such as valves, linkages and regulators.

By operating the boiler with a minimum amount of excess air, it would decrease stack heat losses and increase combustion efficiency.

Large, unnecessary amounts of excess air can occur because of:

- Burner/control system imperfections
- Variations in boiler room temperature, pressure, and *relative humidity*
- Control system “slop”
- Need for burner maintenance
- Changes in fuel composition

3.1.2.1 Excess Air Calculation

Excess air value can be determined easily according to the combustion product, with equation (3.5) (EN483).

$$\lambda = 1 + 0{,}9 * \left[\left(CO_{2\max} / CO_2 \right) - 1 \right] \tag{3.5}$$

CO_{2max}: the maximum carbon dioxide content of the dry,
air-free combustion products in percent.

CO_{2max} for G20 (it can also be taken for natural gas) = 11.7 %

CO_{2max} for G30 (LPG) = 14.0 %

3.1.2.2 Controlling of Excess Air

Controlling excess air is the most important tool for managing the energy efficiency and atmospheric emissions of a boiler system. It is important to keep in mind that the air-to-fuel ratio is based on mass, not volume. The mass of air supplied to the mass of fuel being used (e.g. on a kilogram-to-kilogram basis) must be controlled. The density of air and gaseous fuels changes with temperature and pressure, in fact that must be taken into account in controlling the air-to-fuel ratio. For example, if pressure is fixed, the mass of air flowing in a duct will decrease when the temperature increases. The controls should therefore compensate for seasonal temperature variations and, optimally, for day and night variations too (especially during the spring and fall, when daily temperature variations are substantial).

As can be seen in Figure 3.2 , the effect of air temperature on excess air in the flue gas can be dramatic (Dockrill & Friedrich, 2001)

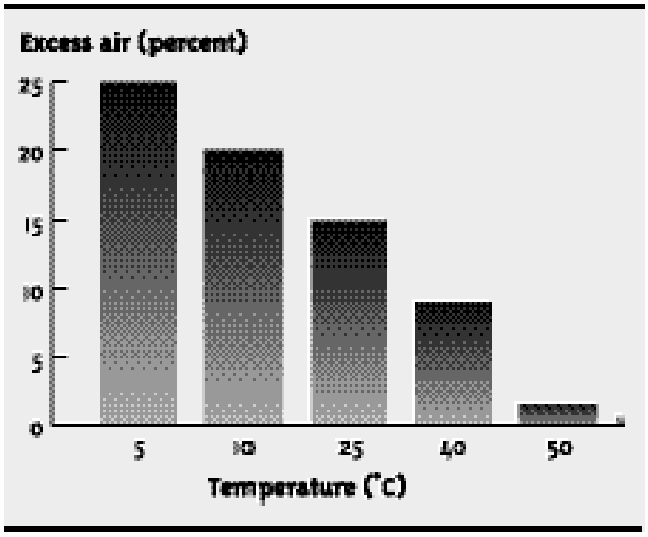


Figure 3.2 Effects of air temperature on excess air level

3.1.3 Incomplete Combustion

In the reaction zone of the flame, the constituent gases were broken down and recombined to form other compounds before being burned to the final products of combustion, CO_2 and H_2O, or carbon dioxide and water vapor. These intermediate substances included carbon, carbon monoxide, alcohols and aldehydes.

If combustion is interrupted at this incomplete stage, then we have “incomplete combustion” and these products can be released from the flame. Of the products of incomplete combustion, carbon monoxide is the most dangerous. It’s a toxic gas. A concentration of only 0.4% in air can be fatal within a few minutes. Unfortunately carbon monoxide combines much more readily with the hemoglobin in the blood stream than does oxygen, which is what the blood needs. The blood normally picks up oxygen from the lungs and carries it to the tissues of the body, collecting the waste products in return. If this hemoglobin becomes saturated with carbon monoxide the tissues cannot take it in or get rid of their waste matter. The body is quickly poisoned and the person dies. Even if death does not occur there may be very serious brain damage due to the lack of oxygen. The negative effects of carbon monoxide are shown in Table 3.7 (Saxon, 2003).

Table 3.7 The negative effects of carbon monoxide (Saxon,2003)

% Saturation of Hemoglobin with CO	**Symptoms**
0 to 10%	No symptoms
10 to 20%	Tightness across the forehead.
20 to 30%	Flushed skin, headache, breathlessness and palpitation on exertion
30 to 40%	Severe headache, dizziness, nausea, weakness of the knees, irritability, possible collapse.
40 to 50%	Symptoms as above with increased respiration and pulse rates
50 to 60%	Loss of consciousness, coma
60 to 70%	Coma, weakened heart and respiration
70% and above	Respiratory failure and death
% Vol. Of CO in Air	**% Saturation of Hemoglobin with CO**
0,01%	4% in 1½ hours About 15% maximum with indefinite exposure
0,03%	10% in 1½ hours 20% in 4 to 5 hours
0,05%	20% in 1½ hours 40% in 4 to 5 hours
0,4%	60% in a few minutes

Carbon itself is not dangerous. It can be seen in the flame as a yellow tip or a yellow zone. If this part of the flame touches a cold surface the carbon is deposited either as hard lumps on the surface or in the form of fine soot throughout the combustion chamber. Soot is a particular nuisance because it makes the servicing of appliances difficult. It is often experienced in the form of a fine dust, which blows about the house as soon as an attempt is made to remove it. It is difficult to get it off from hands or clothes. The formation of soot in narrow passages within a combustion chamber can restrict the flow of the products of combustion which in turn, will reduce the flow of fresh air to or through the flames. The shortage of air may result

in a more serious deposit of soot and carbon monoxide can be formed. Alcohols are quickly oxidized to become aldehydes which have a characteristic smell and often give the indication that incomplete combustion is taking place. Although they are poisonous they are a nuisance rather than a danger because they cause irritation of the eyes which becomes unbearable before a dangerous concentration is reached. The amounts produced by incomplete combustion would not, in any case, be dangerous.

3.1.4 The Negative Effects of Combustion on the Environment

Emissions released to the atmosphere that contribute to global warming have received much attention in recent years. This issue is addressed in the Kyoto Protocol (1997). The purpose of this agreement is reducing emissions between 2008 and 2012 by six percent of 1990 levels. Climate change resulting from global warming is one of the greatest challenges facing the world. Managing combustion processes better and improving the efficiency of energy generation and use are two of the key strategies for reducing atmospheric emissions (see Table 3.8 for a list of some emissions from combustion systems and their effects).

Table 3.8 Emissions from combustion systems and their effects (Dockrill & Friedrich, 2001)

Emission	**Source**	**Effect**
CO_2 (carbon dioxide)	Complete combustion of carbon in fuel	Global warming
CO (carbon monoxide)	Incomplete combustion of carbon in fuel	Smog
SO_2 (sulphur dioxide)	Combustion of sulphur in fuel	Smog, acid rain
NO_x (nitrogen oxides)	By-product of most combustion process	Acid rain
N_2O (nitrous oxide)	**By-product of some combustion process**	Global warming
CH_4 (methane)	Principal component of natural gas; leakage from gas wells, pipelines…	Global warming
H_2O (water vapor)	Combustion of hydrogen in fuel	Localized fog
Particulates (dust, soot)	Unburned or partially burned carbon and hydrocarbons; also ash and dirt in fuel	Smog
Trace elements	Impurities in fuel	Potential carcinogens

3.1.5 Causes of Incomplete Combustion

Almost anything that interferes with the combustion process may result in the release of products of incomplete combustion from the flame. The two main causes, however, are chilling the flame and starving it of oxygen (Fundamentals of combustion, 2004).

3.1.5.1 Chilling

Chilling occurs when a flame touches a cold surface. This does not mean a cold surface in the ordinary sense. It means a surface which is cold compared to the flame. If the surface cools the gases in the flame to below their ignition temperature, some of them will not be completely burned and may be released as incomplete combustion products.

The effect can be particularly bad if the inner cone of a pre-aerated flame is "broken" by the cold surface. This is because this inner cone contains unburned gases. Chilling can also result if the flame is exposed to a stream of cold air or to a draught. This may also tend to lift the flame off the burner. If the flame is under-aerated and yellow-tipped and touches a cold surface, soot will be deposited and, if the conditions are not corrected, the deposit will continue to grow, causing more interference.

3.1.5.2 Lack of Oxygen

Lack of oxygen produces carbon monoxide and may be caused by restricting the amount of air to the burner or by a shortage of oxygen in the air. The air supply to the burner may be restricted either by a blockage of the air inlet to the appliance or by a blockage of the flue outlet. Both have the same effect on the flow of air over the burner.

Even when the total amount of air required can enter the appliance the required amount may not be able to pass through to the flames if there are deposits of dirt, fluff or soot on or in the burner or in the mixing tube.

3.1.5.3 CO / CO_2 Ratio

Because of the dangers of poor combustion, minimum standards of combustion have been laid down by the British Standards Institution. All approved appliances must conform to these specifications, BS 5258, BS EN 483 and BS EN 625.

British and European Appliance manufacturing standards require that the permitted maximum ratio of amount of carbon monoxide produced to the amount of carbon dioxide produced is 0.02 to 1 (20 parts of CO to 1000 parts of CO_2 or 20 divided by 1000 equals 0.02). This ratio figure applies to all domestic appliances burning natural gas and liquefied petroleum gas (LPG). A ratio shows the relative amounts of a substance within a mixture of a liquid, solid or gas.

*3.1.5.3.1 How the CO/CO_2 Ratio is Calculated. It's calculated a*s followed;

$$CO/CO_2 = \frac{\%\ vol\ CO\ concentration}{\%\ vol\ CO_2\ concentration} \tag{3.6}$$

or

$$CO/CO_2 = \frac{ppm\ CO}{\%\ vol\ CO_2 * 10000} \tag{3.7}$$

- A full service was required if the CO/CO_2 ratio was 0.004 or higher
- An appliance was unsafe at a CO/CO_2 ratio figure of 0.008* or higher

* 0.008 is equivalent to 8 parts CO in 1000 parts CO_2, still substantially lower than 0.02 stated in British and European boiler performance standards.

3.1.6 The Logic of Combustion Efficiency Tests

The boiler burns fuel efficiently if it satisfies these conditions:

- it burns the fuel completely
- it uses as little excess air as possible to do it
- it extracts as much heat as possible from the combustion gases.

The combustion efficiency test analyzes the flue gases to tell how well the boiler meets these conditions. The test is essentially a test for excess air, combined with a flue gas temperature measurement. The only purpose of bringing air into the boiler is to provide oxygen for combustion. Bringing in too much air reduces efficiency because the excess air absorbs some of the heat of combustion, and because it reduces the temperature of the combustion gases, which reduces heat transfer. The temperature of the flue gas indicates how much energy is being thrown away to the atmosphere (Fundamentals of combustion, 2004).

• The amount of oxygen in the flue gas indicates directly the amount of excess air that is flowing through the boiler. The atmosphere contains about 21% oxygen. All this oxygen would be consumed if combustion were perfect. Therefore, the presence of oxygen in the flue gas indicates that more air is being used than is necessary for perfect combustion.

• The volume of flue gas is known from the assumed chemical composition of the fuel and the amount of excess air. Knowing the volume of the flue gas and its temperature indicates how much energy is contained in the flue gas. This energy is wasted.

• Knowing how much energy is being produced and how much is being lost gives the combustion efficiency. Combustion efficiency can be also tested by measuring the oxygen content instead of measuring the carbon dioxide content of the flue gases. The principles of the carbon dioxide test are different. Carbon dioxide content

indicates both the amount of fuel burned and the amount of energy released because it is a principal residue of fuel combustion. (The other major residue is water vapor.) If there were no excess air, the amount of carbon dioxide in the flue gases would be a certain amount, determined by the chemical composition of the fuel. A lower concentration of carbon dioxide therefore indicates excess air. In other words, the oxygen test indicates the amount of excess air directly, whereas the carbon dioxide test indicates excess air indirectly, based on an assumption about the chemical composition of the fuel.

3.1.7 Tests for Incomplete Combustion

No boiler is capable of burning fuel without some amount of excess air, although the percentage of excess air may be small. As the excess air is reduced toward zero, there is some fuel that is not burned completely. This partially burned fuel creates smoke, leaves deposits on firesides, and creates environmental problems.

Unburned fuel may also represent a significant waste of energy. The amount of waste depends on the energy content of the unburned fuel components. One source estimates that each 0.1 % of unburned combustibles in flue gas typically represents between 0.3 and 0.6 % of the energy content of the fuel. This waste of energy is not measured by the combustion efficiency test. Therefore, optimum efficiency requires some positive amount of excess air, even though the combustion efficiency test indicates maximum efficiency with no excess air. If the boiler is working properly, the optimum amount of excess air is small. To fine-tune the excess air, it will be needed an additional test that detects small amounts of incompletely burned combustion products. The most common test for this purpose is carbon monoxide in the flue gas (Fundamentals of combustion, 2004).

3.1.7.1 Carbon Monoxide Test

The carbon monoxide content of flue gas is a good indicator of incomplete combustion with all types of fuels, as long as they contain carbon. Carbon monoxide in the flue is minimal with ordinary amounts of excess air, but it rises abruptly as soon as fuel combustion starts to be incomplete. This makes it an excellent indicator when making final adjustments of the air-fuel ratio. An excessive level of carbon monoxide that occurs in the normal region of the air-fuel ratio indicates trouble within the boiler. Carbon monoxide rises excessively if any defect in the boiler causes incomplete combustion, even with excess air. This makes carbon monoxide testing an excellent tool for discovering combustion problems. For example, the carbon monoxide test might reveal a fouled burner. It might also point toward a more subtle problem, such as a poor match of the burner assembly to the combustion chamber, causing a portion of the flame to strike a surrounding surface. (Cooling the flame interrupts the combustion process, leaving carbon monoxide and other intermediate products of combustion in the flue gases.) Carbon monoxide also forms if there is a great excess of air. This is not a matter of practical significance. Once the air-fuel ratio is set properly, the carbon monoxide content falls into the proper range if there are no other problems (Fundamentals of combustion, 2004).

3.1.7.2 Flame Appearance

The practice of adjusting burners on the basis of the color and shape of the flame is no longer acceptable, now that more accurate tests are readily available. However, the color and shape of the flame should checked periodically to verify that there are no combustion problems, such as a clogged burner or water in the fuel.

3.1.7.3 Air Infiltration

Excess air is defined as the air supplied to the burner beyond the air required to theoretically assure complete combustion. Excess air infiltration interferes with the efficiency of the fuel burning process. Excess air can find its way through the burner.

Any test to determine if there is excessive air leakage should be performed when there is a planned maintenance session. Leakage is more likely on older tube and tile constructed furnaces, although welded wall furnaces can leak air at seal areas, such as corner seams. Sealing the furnace at these potential leak sites will allow operating at lower excess air conditions and improve efficiency. Air infiltration can also occur at the seals between the burner and flue hood. Any apparent leakages should be repaired. As always, the combustion equipment manufacturer's instructions for proper maintenance should be followed. In general, leak checking should be done anytime O_2 levels or flue gas volumes increase, or exhaust temperatures decrease (Zeitz, 1997).

3.1.7.4 Combustion Air Temperature

Another common area that affects all types of combustion equipment is combustion air temperature. It can have a dramatic effect on boiler efficiency. Changes in combustion air temperature directly affect the amount of combustion air supplied to the boiler and may increase or decrease the excess level.

High excess air levels mean additional energy losses, increased stack temperatures and reduced boiler efficiency. Changes in flame shape, color and sound are often early indicators. An outside air temperature increase or drop will most certainly affect combustion air temperatures. And also seasonal changes in temperature can cause the excess air levels to change, as can changes in the boiler setting, burners that are out of service and tramp air infiltration (Zeitz, 1997).

3.1.7.5 Fuel and Air Linkages

Changes in the fuel and air linkage settings can also affect boiler efficiency by changing the fuel to- air ratio. These changes can also act on excess air levels.

3.1.8 Maintaining of Combustion Equipments

Gas burner systems consist of a gas metering device, burner nozzles, an air delivery and metering system and a flame ignition and management system. If nitrogen oxide (NOx) control is part of the system, there may be staged combustion, low NOx burner adjustments or flue gas recirculation. As with everything in the combustion system, proper maintenance and operation of each of these systems is essential to an efficient and safe combustion process. Here are some specific items relative to gas burner systems that should receive routine attention.

Boilers with multiple burners should have the burner use patterns set for maximum efficiency where load conditions do not require the use of all burners at all times. Again, consider seasonal outside temperatures in setting these patterns. The operation of the gas modulating control valve should be checked to make sure it is responding to signals from the controls. Nozzle gas pressure should be correct. The operation of the air control damper should be checked to make sure it's responding properly to signals from the controls. The fuel/air ratio should be adjusted for the most efficient operation for the type of load the boiler carries. Settings may be different for base-loaded boilers as opposed to a boiler that follows load swings from a heating or process load. Boilers having swinging loads might not be able to operate at as low an excess air setting as a base- loaded boiler (Zeitz, 1997).

The burner management system should be maintained for peak performance, in accordance with manufacturer recommendations. Flame detection devices should be adjusted to allow minimum excess air operation safely. Burners with NOx controls should have the dampers for staged combustion or flue gas recirculation free and adjusted to follow the signals from the controls. The rest of the NOx and burner controls should be adjusted for maximum NOx control at maximum efficiency and minimum carbon dioxide or volatile organic compound production. Gas pressure is critical to proper burner operation and efficient combustion. Irregular pressure may cause fuel-rich combustion, high carbon dioxide (CO) levels and soot buildup that might reduce boiler efficiency.

3.2 Efficiency and Energy Losses

3.2.1 Conservation of Energy

One of the laws of physics is that energy is always conserved. In combustion, chemical energy is being converted to thermal energy, but nothing is lost. For instance, if one added up the energy in the water, the energy in the flue gas, the energy radiated from the boiler and all the other forms of energy leaving the boiler, they will exactly equal the energy in the gas burned. Boiler efficiency is the percentage of the gas energy which is converted to heat energy (Zeitz,1997).

The heat-loss method is the most accurate way of determining boiler efficiency. To do this, one simply measures the individual heat losses (expressed as a percent of heat input) and subtract them from 100% (representing the fact that energy is never lost).

A simplified method for quantifying boiler efficiency uses the following equation;

Boiler Efficiency (%) = (Output / Input) * 100 (3.8)

with Output = Input - Heat Losses

Alternatively, a simple formula will show:

Boiler Efficiency (%) = 100 % - Heat Losses (3.9)

3.2.2 Heat Loss – Heat Loss Sources

"Heat Loss" is the terminology used to describe wasted energy. It is helpful to understand how boiler efficiency is determined by the measurement of heat losses.

3.2.2.1 Stack Gas (Flue Gas)

The biggest energy loss in a conventional boiler goes "up the chimney"; that is, out the stack. The size of the heat loss depends on the temperature and the volume of gas leaving the boiler, therefore, reducing either one of these will reduce the heat loss. Some stack gas heat loss is unavoidable, but to eliminate this loss altogether, the stack gas temperature would have to be reduced to the air temperature around the boiler. Practically speaking, this is impossible. The practical gas flue temperature limit is about 150 °C. This can vary, depending on the fuel used. Lower temperatures make the small amount of sulphuric acid vapor in the gas condense on cold metal surfaces and cause severe corrosion.

The basic strategies for minimizing stack gas heat loss are:

- Minimizing excess air
- Keeping heat transfer surfaces clean
- Adding flue gas heat recovery equipment where justified;
- Controlling air infiltration

With reduced excess air, stack gas volume is also reduced. Stack gas temperature is also reduced because gas velocities are reduced, allowing the gas to spend more time inside the boiler where the heat can be absorbed. The economics are attractive. As a rule of thumb, boiler efficiency can be increased one percent for each 15 percent reduction in excess air; 1.3 percent reduction in oxygen content or 4-5 °C reduction of stack gas temperature (Zeitz, 1997).

3.2.2.2 Combustible Heat Losses

This is the second largest source of heat loss and is caused by unburned fuel. The most common combustible gas that often enters the flue gas unburned is carbon monoxide. It forms when excess air is too low, or when the flame strikes a cold water wall if the air properly adjusted, this heat loss can be minimized. A carbon monoxide concentration of below 1000 ppm (0.1%) should be the goal.

3.2.2.3 Radiation Heat

A portion of the heat from combustion escapes from the walls of the combustion chamber without being absorbed by the boiler water. Some of this is unavoidable. Heat loss from this occurrence is controlled through proper insulation techniques and the maintenance of insulation layers (Zeitz, 1997).

3.2.3 Calculation of Combustion Efficiency

Combustion efficiency is a measure of how much of the chemical energy of the fuel is converted into useful thermal energy. Combustion efficiency can be determined by reference to flue gas loss and is expressed as:

$$\% \text{ Combustion efficiency} = 100 - \% \text{ Flue gas loss} \tag{3.10}$$

It needs to be remembered that combustion efficiency does not allow for leaks, radiation and other losses from the boiler and so is not the same as the overall thermal efficiency of the boiler. It gives an indication of the quality of the combustion process as a chemical reaction (Dockrill & Friedrich, 2001).

Therefore the overall thermal efficiency is as follows:

η(%)=100-(Flue gas loss+Radiation and convection loss+Unaccounted loss) (3.11)

3.2.3.1 Flue Gas Losses

Flue gas loss is usually the largest single boiler inefficiency component. The range of efficiency losses are 5 to 20 % of their fuel energy input up the gas.

The primary determinants of flue losses are;

- The % of CO_2 or O_2 in the flue gas
- The temperature of the flue gas
- The temperature of the air going into the boiler

The flue losses are determined by the use of following simplified formula: (BSI – EN483).

$$q_c = \left(a + \frac{b}{CO_2} \right) * \frac{\left(T_f - T_a \right)}{100} \tag{3.12}$$

Where,

q_c are the flue losses of the heat input, in %;

a and b are the coefficients according to gas type; (see Table 3.9)

CO_2 is the carbon dioxide content in the dry products of combustion, in percent;

T_f is the temperature of the products of combustion, in °C;

T_a is the ambient temperature, in °C.

Table 3.9 Coefficients “a” and “b” according to gas type

Reference Gas	G110	G25	G20 (also for NG)	G30 (LPG)
a	1,05	0,85	0,86	0,65
b	23,2	36	36,6	42,5

Unfortunately, there is not much solutions to reduce it. Since most of the lost heat is in the heat of vaporization, measures that reduce the flue gas temperature have only a small effect. One measure, particularly effective when firing natural gas, is to pass the flue gas through a condensing heat exchanger, which recovers the heat of vaporization by converting the water vapor back to liquid form.

3.2.3.2 Loss due to Convection and Radiation

This loss occurs from the external surfaces of an operating boiler. For any boiler at operating temperature, the loss is constant. Expressed as a percentage of the boiler’s heat output, the loss increases as boiler output is reduced. Hence, operating the boiler at full load lowers the percentage of loss. Since the boiler’s surface area relates to its bulk, the relative loss is lower for a larger boiler and higher for a smaller boiler

Both of them can also calculate by using simplified heat transfer equations; (Kakac & Yuncu, 1999).

$$Q_{conv} = \alpha * A * (T_s - T_a) \quad (3.13)$$

Q_{conv} : Conventional heat transfer rate, (W)

α : coefficient of convection, ($W / m^2 * K$)

A : Surface area, (m^2)

T_s : Temperature of surface of the boiler, K

T_a : Ambient temperature, K

$$Q_{rad} = \varepsilon * \sigma * A * (T_s^4 - T_a^4) \quad (3.14)$$

Q_{rad} : Radiational heat transfer rate, (W)

ε : emittance, $0 < \varepsilon < 1$

σ : Stefan-Boltzmann constant, $5.6704 * 10^{-8}$ ($W / m^2 * K^4$)

A : Surface area, (m^2)

T_s : Temperature of surface of the boiler, K

T_a : Ambient temperature, K

It will examined more specifically in the following chapter for 24kW RSF combi boiler.

3.2.3.3 Unaccounted Losses

For combustion systems with natural gas or fuel oil fired, many of the losses such as unburned fuel, moisture in fuel and in air etc. are either non-existent or very small. They can be handled conveniently, and usually with sufficient accuracy, by assuming a value for the total as "unaccounted" losses. Commonly used figures are 0.1% for natural gas-fired systems and 0.2% for oil-fired systems (Dockrill & Friedrich, 2001).

3.2.4 Energy Losses in the Steady-State Heat Exchange Process

Every heating appliance is designed to transfer heat from an energy source to the space. After a certain period of time, a heating appliance in operation reaches a point, where the fraction of input energy that is transferred, remains constant. This is the point were the mass of the involved heater components (pipes, heat exchanger, pump, jacket, control, etc.) has reached the operating temperature, and the heat exchanger reaches its maximal efficiency. Unfortunately this fraction is in most cases much lower than the ideal figure of 100%. Depending on the efficiency of the heat exchanger itself, the heat losses through flue ducts and the radiation- and convection-

losses of the appliance, these steady-state energy losses can vary from approximately 5% to in worst cases 40 – 50% (Figure 3.3) (SAVE II Action).

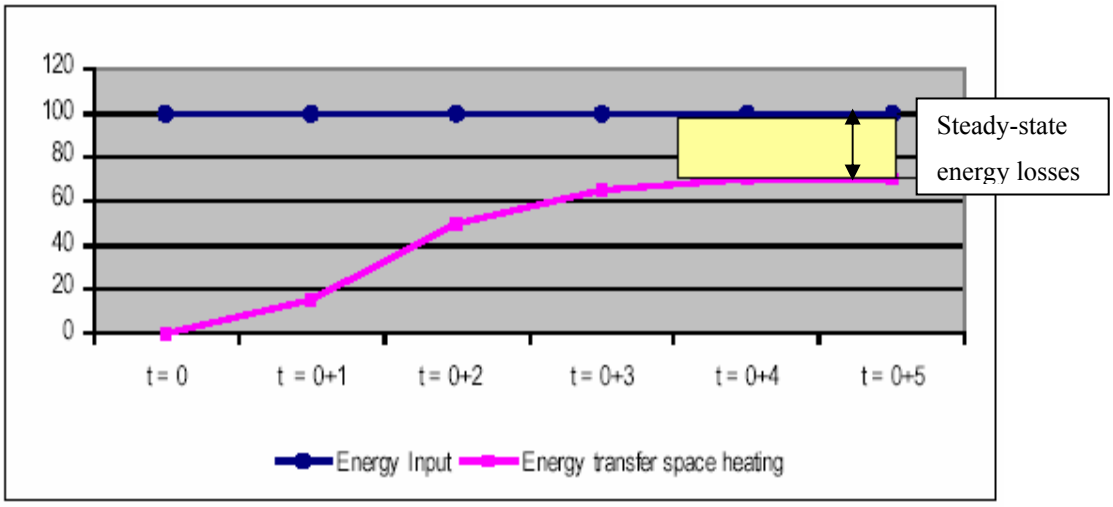


Figure 3.3 Steady-state energy losses (SAVE II Action)

Important parameters:

- water temperature
- water amount
- heat exchanger efficiency
- heat conductivity
- heat capacity
- emission coefficient
- insulation
- heat traps
- flue duct construction.

3.2.4.1 Standing Losses

Standing losses refer to the heat losses that occur when the heating appliance is not used. Standing losses are caused by the fact that heating appliance transfers heat

to its surroundings through convection and radiation. Since space heating appliances in general do not incorporate heat storage (e.g. combi and solar boilers excluded) standing losses are little. Another type of standing losses is caused be the 'pilot flame'.

Some heating appliances use pilot flames for the ignition of the burner. These pilot flames continuously consume energy.

Important parameters:

- ignition system;
- pump control;
- insulation;
- flue duct construction.

3.2.4.2 Start / Stop Losses

Start/stop losses are the losses that are caused by the heating up and cooling down of the involved appliance-components. Before the "steady-state" in heat transfer is reached, various components are heated up. Depending on the type of appliance this can be components like pipes, fittings, heat-exchanger, flue ducts, pump, valves, fan, jacket, etc. The heat-capacity of these components/materials determine how much energy will be absorbed, before the steady-state situation is reached. The quality of the insulation determines the speed with which this absorbed heat can be lost through convection and radiation. Gas fired boilers may have increased start/stop losses, when flue-ducts are used without any heat-block provisions. The flue-duct then works as a chimney. (Figure 3.4) (SAVE II Action).

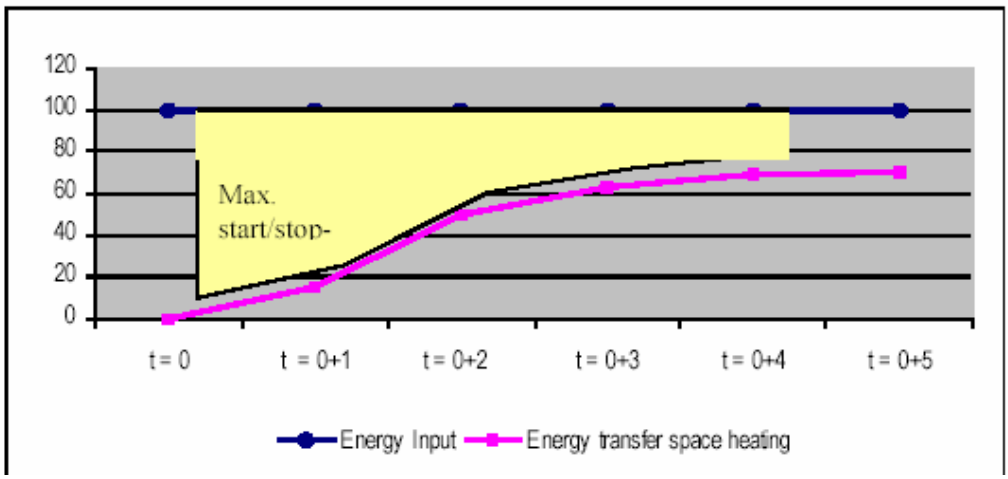


Figure 3.4 Start - Stop losses (SAVE II Action)

In the graph above the start/stop-losses are illustrated. If the time between two heating periods is long enough to cool down the heating system and its components, than the indicated area really is the amount of energy that is lost. If heating periods are done with short intervals, only a fraction of the accumulated heat will be lost. The appliance is still warm when the next heating period starts. The actual start/stop losses will therefore depend upon:

- the heat-capacity (the max. amount of heat that can be stored in the heating system during a heating period).
- the daily/yearly heating pattern (the real amount of heat that is stored in the heating system).
- the insulation (the *speed* with which the heated appliance cools down when it is not used); (there might be an adverse effect here because of the heat-capacity of the insulation)

Start/stop losses can be minimized by optimizing the insulation (jacket, heat traps in pipes and flue ducts) and/or by minimizing the heat capacity of the appliance (weight and specific heat of the material used).

Important parameters

- water temperature;
- water amount;
- heat exchanger efficiency;
- heat conductivity;
- heat capacity;
- weight;
- emission coefficient;
- insulation;
- heat traps;
- flue duct construction

3.2.4.3 Fan and Burner System Efficiency (Modulating Burners and Fans)

The burner system does not only concern the burner itself, also the gas and air supply (gas valve and fan), the mixing of the two and the burner (gas valve and fan) control are important matters.

In earlier years, CH-boilers were equipped with on/off type burner control. Burners of on/off heating systems have only two settings “off” or “full power”. The last decade the burner control system has gone through quite some developments. Starting with multi-stage type controls, nowadays continuous burner control systems (modulating burner control) tends to become the standard.

The major advantage of modulating systems is that the power can be adjusted to the desired send temperature. The consequence is less temperature fluctuation of the CH-water and therefore less fluctuation in air temperature in the house. Besides the comfort aspect there are also energetic advantages: (SAVE II Action).

- A modulating system burns longer on low power. There will be less overshoot (less water temperature fluctuations). Moreover there will be less methane-slip (during start-up not all the gas will immediately ignite)
- The return water temperature will be lower as well. This improves condensation of flue gasses and will therefore improve efficiency
- There will be less losses during start-up (for heating up the boiler).

The "optimal" burner power can be based on:

- CH (send) water temperature
- difference between CH send and return water temperature
- slow increase of burner temperature
- difference between set and actual temperature of room thermostat (it is necessary to have communication between room thermostat and boiler e.g. with Open Therm)
- outside temperature sensor (weather dependent control system).

3.2.4.4 Heat Exchanger Efficiency and Losses

In the past cast iron was a frequently used material for heat exchangers. Nowadays Aluminum and stainless steel are the common materials. Both materials show good qualities to withstand the aggressive condense in combi boilers. Especially aluminum has some clear advantages concerning energy efficiency and is becoming the most popular heat exchanger material (SAVE II Action). Aluminum shows good heat conductivity (7 times higher than stainless steel). Aluminum heat exchangers can therefore be very compact and lightweight. A compact heat exchanger also means less water content. As long as the control strategy of the boiler is able to prevent overheating of the system, e.g. when all radiators are closed, a boiler with a very low heat capacity can be designed. Furthermore Aluminum has a low emission coefficient and therefore low radiation losses. The most important parameters for the energy efficiency of the CH-boiler are the start/stop-losses, the steady-state efficiency, and the auxiliary power consumption.

3.2.5 Auxiliary Energy Losses

3.2.5.1 System Description

Besides the energy that is needed for heating, many appliances use electricity for the control- functions. Most of these control- components are switched off when there is no heat demand. But some of them may be designed in a way that they constantly use electricity. Like standing losses, this '*stand by electricity consumption*' can be important because of its continuous and ever lasting nature. A considerable number of heating appliances use electric components in order to fulfil their function as space heater. These are components like pumps, fans, controls, and electronic ignition. The electricity consumption of these components has to be taken into account when something is to be said about the energy efficiency of heating appliances. On the total primary energy needed for the fan, the pump and the control system of the boiler varies 1 – 3 % (An example for 24kW appliance is in Table 3.10) (SAVE II Action).

Table 3.10 Auxiliary energy losses

Conventional Combi Boiler	**[W]**
Pump	150
Fan	150
Controls	150
Total Aux. Energy	450
Total Heating Energy	24000
% of Aux. Energy Loss	1.875%

3.2.5.2 Reducing Auxiliary Power

The total electric load for components like pump, controls and fan mounts up to 110 – 150 Watts. When no fan is used the load is reduced to 70 – 80 Watt. With efficient pumps, fans and controls the electricity consumption can be reduced with maximal 50% (selection of the right pump, improvement of electrical efficiency of the motors (fan & pump)). Also the use of reciprocating pumps should be investigated. For combi boilers there will also be an effect on the efficiency of water heating but of course the savings will have a greater impact on the power consumption for boiler function, where the average operating times are approximately 2000 – 2500 hours per year versus 300 – 400 minutes for water heating.

3.2.5.3 Stand-by Power Consumption

All CH-appliances must be installed with an electrical connection. Most of these appliances use power in their stand-by mode, because of the controls. When LED's are used (e.g. to indicate the operation mode of the appliance) the standby power consumption is approximately 8 Watts. Per year this is approximately 70 $kWh_{el.}$ with proper electronic design this can be reduced with 50 to 60%.

The lists of various kinds of losses for nine different boiler types can be seen in Appendix Table A-2 (SAVE II Action).

3.2.6 Distribution and Emitters

3.2.6.1 System Description

Depending on building-practice and –regulations, the heat generator is placed in the attic, the cellar, the scullery, the kitchen and sometimes the bathroom (combustion air from outside). Because there are strict regulations in most countries concerning the position of outlet of the flue-duct (through walls and roofs), it is not always easy to install the product in the most profitable place (= short distance from taps). All combi boilers need electric power for the pump, the controls and the fan. The efficiency of the piping network is mainly determined by heat losses through uninsulated pipes where the pipes run through non-habitable spaces in the house. This could be an attic, a cellar, etc. This is not uncommon, because the attic and cellar are also popular places for installers to put the CH ring-network from which the vertical feed of the radiators starts. Insulation of the pipes could save up to 10%. Hydronic central heating systems use water as a medium to transport and emit heat from the heat generator to the individual rooms. Figure 3.5 shows a categorization of the emitter systems (SAVE II Action).

CENTRAL HEATING EMITTER SYSTEMS

1. *SEPARATE FROM BUILDING CONSTRUCTION*

• radiators	• high temperature system • low temperature system
• convectors	• high temperature system • low temperature system

2. *INCORPORATED IN BUILDING CONSTRUCTION*

• floor heating systems	• low temperature system
• wall heating systems	• low temperature system
• ceiling heating systems	• low temperature system

Figure 3.5 Categorization of emitter systems (SAVE II Action)

For hydronic central heating systems so called ‘radiators’ are the most popular emitters. Floor heating however is becoming more popular. According to the heating temperature the systems are classified as;

- High temperature systems (HTS): Flow / return temperature 90 / 70°C
- Medium temperature systems (MTS): Flow / return temperature 65 / 55°C
- Low temperature systems (LTS): Flow / return temperature 55 / 35°C

3.2.6.2 Conventional Radiators / Convectors

3.2.6.2.1 Radiators. This type of distributers can be classified as:

• multiple section radiators (ribbed, sectional radiators);
• panel radiators (single/double).

*3.2.6.2.2 Convectors. A*re made out of thin metal tubes surrounded with a range of narrow spaced thin metal fins. Convectors can be placed in a shaft/pit (e.g. in front of sliding doors) or in a casing (placed e.g. under a window). Because the emitter is completely enclosed, there is hardly any radiation. Convector systems have little water content and therefore have short heating times (See Table 3.11) (SAVE II Action).

Table 3.11 Heat emitters, radiation versus convection

	Radiation (in %)	Convection (in %)
Sectional radiator	30	70
Single panel radiator	50	50
Double panel radiator	20	80
Convector	0	100

3.2.7 Low Temperature Heating (LTH)

3.2.7.1 Floor Heating

Typically floor-heating systems use plastic tubes, which are poured in a layer of concrete (heavy systems). It is important that under and around this layer insulation material is used (floating system). Besides “heavy systems” there are also “light systems” for use in existing houses and for new-built timber frame houses. The temperature of the floor can be maximum 29°C; the maximum send temperature is therefore 45°C. In direct systems the water heated by the heat generator flows directly through the tubes. If the send temperature is too high, an indirect system can be used. The latter is less efficient.

3.2.7.2 Wall Heating

Wall heating systems show a lot of similarities with floor heating. Plastic tubes are placed in into the wall milled channels or into specially formed bricks. The walls are finished with plaster. There are also “light systems”. The maximum wall temperature is 40°C, the maximum send temperature is 55°C. Also in wall heating systems a distinction is made between direct and indirect systems. Ceiling heating can also be used but offers a less comfortable climatisation when compared to floor or wall heating (higher vertical temperature gradient). One advantage of ceiling heating is the easy combination with ceiling cooling (which is regarded as highly comfortable).

Table 3.12 Extra costs of LTH systems (SAVE II Action)

Emitter System	Floor System	Wall Heating	LT - radiators	LT - convectors
Reference	radiators 90 / 70˚C	radiators 90 / 70˚C	radiators 90 / 70˚C	convectors 90 / 70˚C
Extra Costs (in €)	500 to 1600	500 to 1600	250 to 500	250 to 500

3.2.7.3 Advantages and Disadvantages

The main advantages of LTH are higher boiler efficiencies for condensing boilers (the low return temperature increases the effect of condensing). Combination with heat pumps are also possible since the temperature difference between source and delivered heat is smaller. Low temperature heating also opens up possibilities for making use of renewable (solar) energy. (SAVE II Action).

Low temperature systems especially score better on thermal comfort and air quality, thanks to higher amount of heat transfer through radiation, less turbulence, less dust burn. Savings for HR central heating systems 3 to 9%. With low temperature floor heating the room thermostat can be set 2 to 3°C lower than with conventional radiator heating.

A disadvantage of LTS is the longer heating time. For new, well-insulated houses this need not be a problem because setting a lower night temperature is not necessary. Another drawbacks for floor and wall heating are the higher transmission and distribution losses.

3.2.8 Efficiency Increasing Methods

With the growing popularity of combi-boilers and the increased insulation of new houses, the dimensioning of the boiler with respect to the space heating demand has become a problem. For hot water heating the minimum capacity of a combi-boiler should be around 22-24 kW. However, in a well insulated house the space heating – at its peak—may require less than 8-10 kW. And in spring and autumn –in a moderate climate—the space heating requirement may on average be only 1 to 2 kW. Especially for a house where space heating demand is in the same range as capacity needed for hot water heating (22 kW and upwards), there is an advantage of a combi-boiler over two single appliances if the combi-boiler is a storage device. In that case, after the hot water demand has been fulfilled the three way valve can switch to the

space heating circuit and utilize any remaining heat in the water and heat exchanger. And the estimated gain is about 3 to 5 % by this operation.

There are ways to overcome problems caused by the discrepancy between the capacity of the burner needed for heating and hot water production and the capacity required for solely heating. These solutions affect the feed temperature (or boiler water temperature).

On-off control is the least expensive but also typically the least efficient because of the losses associated with each cycle. Often boilers are oversized causing frequent on-off cycling. The associated pre and post-purge losses can be substantial. When a boiler is cycled on or off, a purging of the combustion chamber takes place before and after firing to prevent explosions of combustible gases that may have accumulated in the boiler. This process wastes heat; in addition, frequent cycling of a boiler decreases its useful life because of the thermal stresses associated with each on-off cycle.

Low-high-low combustion control is also fairly inexpensive but more efficient and effective than on-off control. The boiler starts and stops with the typical pre and post purge and then cycles between a low-fire and max-fire condition to meet the heating load.

Modulating control is generally more expensive, but also the most efficient and most effective at handling varying heating loads. Modulating control turns down the firing rate of the burners to meet the heating load. At low fire conditions the flow of combustion gases through the boiler is less and more heat can be captured before the combustion gases exit the boiler. The modulating burners can regulate the burner capacity to at least 40% of nominal value. This is now becoming the standard for new houses. With a bit more expensive components on the part of the gas valve and boiler fan values of 30% modulation could be reached. In case of segmented burners, even 10-20% modulation is now possible.

An interesting combination is a modulating burner with a ‘modulating’ (room) thermostat. The modulating thermostat ‘senses’ the difference between the set and actual room temperature and regulates burner modulation according this difference (the lower the difference, the more modulation occurs). This helps to prevent temperature overshoots. Another way to prevent temperature overshoots is applying a weather-dependent boiler temperature control. This control tunes the boiler temperature to the outside temperature. E.g. in spring the boiler could run at a 30/50°C regime, whereas in winter it needs 40/60°C. This does avoid some of the temperature overshoot, but does little for the boiler malfunction. The main gain of modulating and weather dependant systems is in heating comfort and avoiding boiler malfunction. And there is also a gain in energy efficiency on the fuel side. There are penalties on the part of the electricity consumption of the fan (35-40W) and the pump (60-80W, in a typical residential boiler) that now have to run non-stop during the whole heating system.

Stack gas economizer can be implemented on a conventional boiler to capture some of the heat of the exhausted combustion gases. Stack gas economizers are heat exchangers that capture heat from the exhausting gases and transfer it to the return water going into the boiler.

Recuperators increase the efficiency of a boiler by transferring some of the heat from the exhaust combustion gases to the intake combustion air and may increase boiler efficiency by 2.5 %.

Boiler tube turbulators can be installed in fire tube boilers to increase the heat transfer of the combustion gases. Boiler tube turbulators are placed inside the fire tubes and may increase boiler efficiency by 1.5 %.

Whether these solutions actually represent an efficiency increase remains to be seen. In fact, from that point of view, it would probably be better to split water heating and space heating as before in two separate devices and dimension the space

heating device according to need, despite the fact that there is some efficiency gain to be had from a combi.

Preliminary summary of the design options effects on the efficiency can be found in Appendix Table A-3.

When looking to maximize boiler efficiency by considering a condensing boiler; condensing boilers will capture more heat from the combustion gases than conventional boilers. An important factor in maximizing condensing boiler efficiency is maintaining a relatively low return water temperature (30–40°C) whenever possible. A lower return water temperature allows the boiler to extract more of the heat of combustion because of the larger temperature difference between the return water and the combustion gases in the boiler. Running at a lower return water temperature can be accomplished by using supply water temperature reset to lower the supply water temperature. Modifications like stack gas economizers or recuperators can often be retrofitted to a current conventional boiler to increase the efficiency. (See Table 3.13) (SAVE II Action).

Table 3.13 Comparison between the efficiency of conventional and condensing boilers

Efficiency and emissions. Average values calculated from the answers to the questionnaires.		Water heaters	CH boilers
Net efficiency (*) of units older than 15 years			
Range from	Net %	63	79
Range to	Net %	75	88
Average	Net %	72	84
Net efficiency (*) of new units NON-CONDENSING			
Range from	Net %	72	87
Range to	Net %	83	95
Average	Net %	80	90
Net efficiency (*) of new units CONDENSING (given for water temperature set of 30/40)			
Range from	Net %		94
Range to	Net %		106
Average	Net %		100
NOx emission for units older than 15 years			
Range from	mg/kWh	207	156
Range to	mg/kWh	293	256
Average	mg/kWh	238	204
NOx emission for new units			
Range from	mg/kWh	167	46
Range to	mg/kWh	250	123
Average	mg/kWh	166	89

(*) efficiency under standard test conditions (eg. 60/80 for CH boilers)

3.2.9 Sankey Diagram

The Sankey diagram is very useful tool to represent an entire input and output energy flow in any energy equipment or system such as boiler. The diagram visualizes in numbers and thickness of arrows the relative importance of each energy input, output and losses associated with the system based on a energy balance calculation. Another advantage of Sankey diagrams is that the width of the arrows communicates visually the idea that energy is conserved – the total amount of energy is the same at beginning and end.

In Figure 3.6, the Sankey diagram has given for 24kW combi boiler regarding to the test results that were performed in the lab and presented in following chapter.

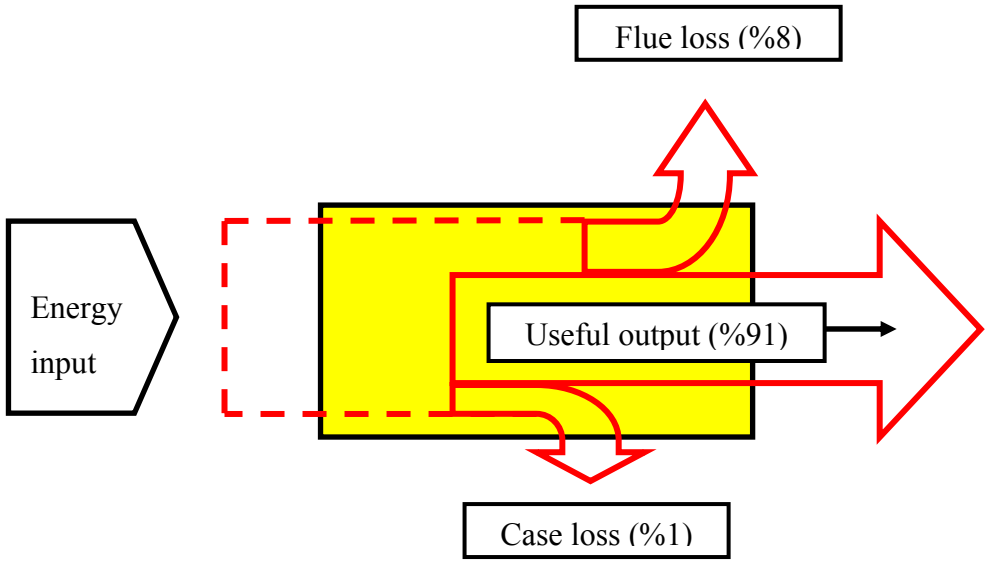


Figure 3.6 Sankey diagram

3.2.10 Exergy Analysis

Exergy is defined as the maximum amount of work which can be produced by a system or a flow of matter or energy as it comes to equilibrium with a reference environment. Exergy is a measure of the potential of the system or flow to cause change, as a consequence of not being completely in stable equilibrium relative to the reference environment. Unlike energy, exergy is not subject to a conservation law (except for ideal, or reversible, processes). Rather, exergy is consumed or destroyed, due to irreversibilities in any real process. The exergy consumption during a process is proportional to the entropy created due to irreversibilities associated with the process (Dincer & Cengel, 2001).

For exergy analysis, the state of the reference environment, or the reference state, must be specified completely. This is commonly done by specifying the temperature, pressure and chemical composition of the reference environment. The results of exergy analyses, consequently, are relative to the specified reference environment, which in most applications is modelled after the actual local environment (Dincer & Cengel, 2001).

Exergy analysis is a method that uses the conservation of mass and conservation of energy principles together with the 2^{nd} law of thermodynamic for the analysis, design and improvement of energy and other systems. The exergy method is useful for furthering the goal of more efficient energy-resource use, for it enables the locations, types, and true magnitudes of wastes and losses to be determined. In general, more meaningful efficiencies are evaluated with exergy analysis rather than energy analysis, since exergy efficiencies are always a measure of the approach to the ideal case. Therefore, exergy analysis can reveal whether or not and by how much it is possible to design more efficient energy systems by reducing the inefficiencies in existing systems.

For the analysis of exergy for 24kW RSF combi boiler;

- it is assumed that combustion is complete
- fuel air ratio (λ) for the combustion reaction has taken 1.2
- inlet conditions for reactants (CH_4 and air) are taken 25ºC and 1atm
- combustion products are liberated to the environment and the temperature of the products will cool to environment temperature, 25ºC, at the end.

For fuel air ratio, λ = 1.2

Combustion reaction equation is;

$$CH_4 + 2.4O_2 + 9.02N_2 \Rightarrow CO_2 + 2H_2O + 9.02N_2 + 0.4O_2 \quad (3.15)$$

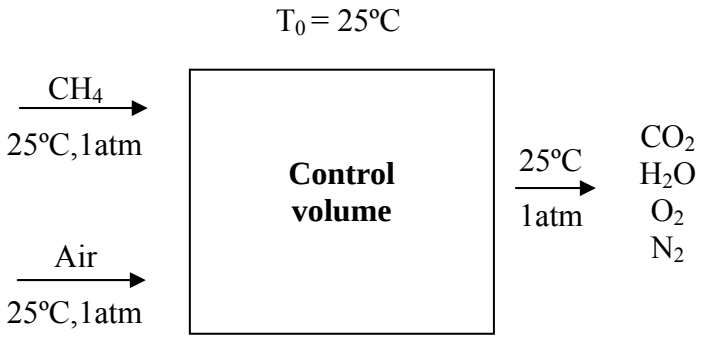


Figure 3.7 Inlet and outlet conditions

Temperature of the products will be the same as the environment temperature after cooling and some of the vapor will be condensed. Amount of condensed water (n_l) could be calculated.from the following equations;

$$\frac{n_{v-H_2O}}{n_{total-gas}} = \frac{P_{v-H_2O}}{P_{total}} \tag{3.16}$$

$$n_{l-H_2O} = n_{H_2O} - n_{v-H_2O} \tag{3.17}$$

P_v = 3.169 kPa (for 25ºC saturated water pressure from Appendix - Table A-4)

P_{total} = 101.325 kPa

$n_{total\text{-}gas} = 1 + 9.02 + 0.4 + n_{v\text{-}H_2O}$

from equation (3.16) and (3.17);

$n_{v\text{-}H_2O}$ = 0.34 kmol and $n_{l\text{-}H_2O}$ =2 – 0.34 = 1.66 kmol

3.2.10.1 Calculation of Heat Transfer to the Environment

Throughout combustion process, heat transfer to the environment is calculated by using following equation;

$$Q_{env} = \sum n_{out}\left(\bar{h}_f^{\,o} + \bar{h} - \bar{h}^{\,o}\right)_{out} - \sum n_{in}\left(\bar{h}_f^{\,o} + \bar{h} - \bar{h}^{\,o}\right)_{in} \qquad \text{[kJ/kmol]} \tag{3.18}$$

Inlet and outlet temperatures have been taken as the same, 25 ºC, ($\bar{h} = \bar{h}^{\,o}$)

Then;

Heat transfer to the environment can be simplified as follows;

$$Q_{env} = \sum n_{out}\,\bar{h}_{f,out}^{\,o} - \sum n_{in}\,\bar{h}_{f,in}^{\,o} \tag{3.19}$$

The enthalpy of formation of the molecules have been taken from the Appendix - Table A-5.

$$\bar{h}^{o}_{f\,CO_2} = -393520\ kJ/kmol$$

$$\bar{h}^{o}_{f\,H_2Ogas} = -241820\ kJ/kmol$$

$$\bar{h}^{o}_{f\,H_2Oliq} = -285830\ kJ/kmol$$

$$\bar{h}^{o}_{f\,CH_4} = -74850\ kJ/kmol$$

$$\bar{h}^{o}_{f\,O_2} = 0\ kJ/kmol$$

$$\bar{h}^{o}_{f\,N_2} = 0\ kJ/kmol$$

and heat transfer to the environment has been calculated from equation (3.19)

Q_{env} = -875400 kJ / kmol

3.2.10.2 Calculation of Entropy Generation (S_{gen})

Table 3.14 Entropy calculation for reactants

Reactants	n_i	y_i	$\bar{s}^{o}_{i}(T, 1\ atm)$	$-R_u \ln y_i P$	$n_i s_i$
CH_4	1	1.00	186.16	-	186.16
O_2	2.4	0.21	205.04	12.98	523.25
N_2	9.02	0.79	191.61	1.96	1746

In Table 3.14;

n_i : number of moles

y_i : proportion of molecules

R_u : universal gas constant , 8.31434 kJ/kmol K

$\bar{s}_i^{\,o}$: absolute entropy for ideal gas (at 25 ºC and 1 atm , from Appendix - Table A-5)

$[S_i]_{react}$: entropy of reactants [kJ / kmol K]

$$[S_i]_{react} = n_i \bar{s}_i(T, P_i) = n_i \left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P \right] \tag{3.20}$$

$$[S_i]_{CH4} = n_i \left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P \right]_{CH_4} = 1[186.16 - 8.314 \ln 1] = 186\, kJ / kmolK$$

$$[S_i]_{O_2} = n_i \left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P \right]_{O_2} = 2.4[205.04 - 8.314 \ln 0.21] = 523.25\, kJ / kmolK$$

$$[S_i]_{N_2} = n_i \left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P \right]_{N_2} = 9.02[191.61 - 8.314 \ln 0.79] = 1746\, kJ / kmolK$$

$$\sum [S_i]_{react} = 2455.41\, kJ / kmolK$$

Table 3.15 Entropy calculation for combustion products

Products	n_i	y_i	$\bar{s}_i^{\,o}(T, 1\,atm)$	$-R_u \ln y_i P$	$n_i s_i$
H_2O_{liq}	1.66	1.0000	69.92	-	116.07
H_2O_{vap}	0.34	0.0316	188.83	28.72	73.97
CO_2	1	0.0929	213.80	19.76	233.56
O_2	0.4	0.0371	205.04	27.39	92.97
N_2	9.02	0.8383	191.61	1.47	1741.58

In Table 3.15;

n_i : number of moles

y_i : proportion of molecules

R_u : universal gas constant , 8.31434 kJ/kmol K

$\bar{s}_i^{\,o}$: absolute entropy value for ideal gas (at 25 ºC and 1 atm , from Appendix - Table A-5)

$[S_i]_{prdct}$: entropy of products [kJ / kmol K]

$$[S_i]_{prdct} = n_i \bar{s}_i(T, P_i) = n_i\left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P\right] \tag{3.20}$$

$$[S_i]_{H_2O_{liq}} = n_i\left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P\right]_{H_2O_{liq}} = 1.66\left[69.92 - 8.314 \ln 1\right] = 116.07\, kJ/kmolK$$

$$[S_i]_{H_2O_{vap}} = n_i\left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P\right]_{H_2O_{vap}} = 0.34\left[188.83 - 8.314 \ln 0.0316\right] = 73.97\, kJ/kmolK$$

$$[S_i]_{CO_2} = n_i\left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P\right]_{CO_2} = 1\left[213.80 - 8.314 \ln 0.0929\right] = 233.56\, kJ/kmolK$$

$$[S_i]_{O_2} = n_i\left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P\right]_{O_2} = 0.4\left[205.04 - 8.314 \ln 0.0371\right] = 92.97\, kJ/kmolK$$

$$[S_i]_{N_2} = n_i\left[\bar{s}_i^{\,o}(T, P_o) - R_u \ln y_i P\right]_{N_2} = 9.02\left[191.61 - 8.314 \ln 0.8383\right] = 1741.58\, kJ/kmolK$$

$$\sum [S_i]_{prdct} = 2258.15\, kJ/kmolK$$

$$\Delta S_{system} = \sum [S_i]_{prdct} - \sum [S_i]_{react} = 2258.15 - 2455.41 = -197.26\, kJ / kmolK \quad (3.21)$$

$$\Delta S_{env} = Q_{env} / T_o = 875400 / 298 = 2937.58\, kJ / kmolK \quad (3.22)$$

Finally entropy generation is;

$$S_{gen} = \Delta S_{system} + \Delta S_{env} = 2740.32\, kJ / kmolK \quad (3.23)$$

$$I = W_{rev} = S_{gen} \text{ x } T_o = 2740.32 \text{ x } 298 = 816615 \text{ kJ/kmol} \quad (3.24)$$

I : total irreversibility

W_{rev} : reversible work

When 1 kmol CH_4 is burnt; it must be possible to get 816615 kJ available energy. But in this chemical reaction; there is no work production therefore reversible work is equal to total irreversibility.

To calculate reversible power ($\dot{W}_{rev}$);

$$\dot{Q} = \dot{m}_{CH_4} H_i \quad (3.25)$$

In this equation;

H_i , is the net calorific value of methane and expressed as;

34.02 MJ/m^3 of dry gas under the reference conditions; or 47.628 MJ/kg of dry gas.

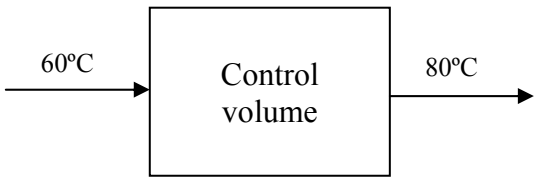
Then

$$\dot{m}_{CH_4} = \frac{\dot{Q}}{H_i} = \frac{24000J/\sec}{47.628MJ/kg} = 5.04*10^{-4}\,kg/\sec$$

and finally;

$$\dot{W}_{rev} = \dot{m}_{CH_4} W_{rev} = 5.04*10^{-4}\,\frac{816615kJ/kmol}{16kg/kmol} = 25.72kW$$

3.2.10.3 Calculation of Available Energy (Useful Work) (Wu)



Useful work (available energy) can calculate by using following formula;

$$w_u = \psi_{80} - \psi_{60} = \left[\left(h_{80} - h_{60}\right) - T_o\left(s_{80} - s_{60}\right)\right] \qquad (3.26)$$

w_u : available energy, [kJ/kg]

Ψ : exergy (availability) [kJ/kg]

h : enthalpy [kJ/kmol] , from Appendix - Table A-4

s : entropy [kJ/kmolK] , from Appendix - Table A-4

$$w_u = \left[\left(334.91 - 251.13\right) - 298\left(1.0753 - 0.8312\right)\right] = 11.724\,kJ/kg$$

$$\dot{m}_{H_2O} = \frac{Q}{c_p \Delta T} = \frac{24000 J/\sec}{4180 J/kgK \; 20K} = 0.287 kg/\sec \tag{3.27}$$

$$\dot{W}_u = \dot{m} w_u = 0.287 \times 11.724 = 3.36 \; kW \tag{3.28}$$

and finally 2nd Law Efficiency is calculated as follows:

$$\eta_{II} = \frac{\dot{W}_u}{\dot{W}_{rev}} = \frac{3.36}{25.72} = \%13.06 \tag{3.29}$$

Apparently, the 1st Law of Thermodynamic refers to the energy analysis which only identifies losses of work and potential improvements or the effective use of resources. However, the 2nd Law of Thermodynamic, i.e., exergy analysis takes the entropy portion into consideration by including irreversibilities (Dincer & Cengel, 2001).

CHAPTER FOUR

EXPERIMENTAL STUDY

4.1 Study on the Flue Length Effect on the Flue Loss Efficiency

For this purpose a series of tests had been carried out on the appliance with different flue lengths and different fraction of the nominal heat input.

4.1.1 Test Conditions

Flue Length **:** 300 / 1000 / 2000 / 3000 / 4000 mm
Direction **:** Right (the worst direction for that appliance)
Test Gas **:** Natural Gas (with H_i =33.839 MJ/m^3, Rel. Density: 0.569)
Inlet Pressure **:** 20 mbar
Amb. Temperature **:** 23°C
Fan Voltage **:** 230 V
Test Date **:** 21/09/2004
Test performed by **:** E. Kesdogan

Table 4.1 Flue loss efficiency with various flue lengths

300mm flue									
Percentage of Rate (%)	52,2	62,7	70,6	79	97,4	111,2	113,5	116,4	118,3
Flue Loss Efficiency (%)	85,9	86,9	88	88,8	89,9	90,7	90,8	90,8	91

1000mm flue									
Percentage of Rate (%)	51,6	58,2	63,3	72,4	81,7	97,3	112,7	116,8	119,6
Flue Loss Efficiency (%)	86,6	87,5	88,1	89,1	89,7	90,8	91,7	91,8	91,7

2000mm flue									
Percentage of Rate (%)	47	52,1	57,3	62,9	71,9	79,4	98,1	112,9	120,3
Flue Loss Efficiency (%)	87,5	88,5	89,4	90,1	90,9	91,2	92,3	93	93,2

3000mm flue									
Percentage of Rate (%)	45,6	56,4	65,1	78,1	97,3	110,9	113,5	115,5	117,7
Flue Loss Efficiency (%)	88,8	90,9	91,8	92,4	92,9	93,4	93,5	93,5	93,7

4000mm flue									
Percentage of Rate (%)	47,1	53,2	58,3	63,8	72,9	79,9	97,7	112,2	119,9
Flue Loss Efficiency (%)	90,7	91,6	92	92,3	93	93,2	94	94,5	94,6

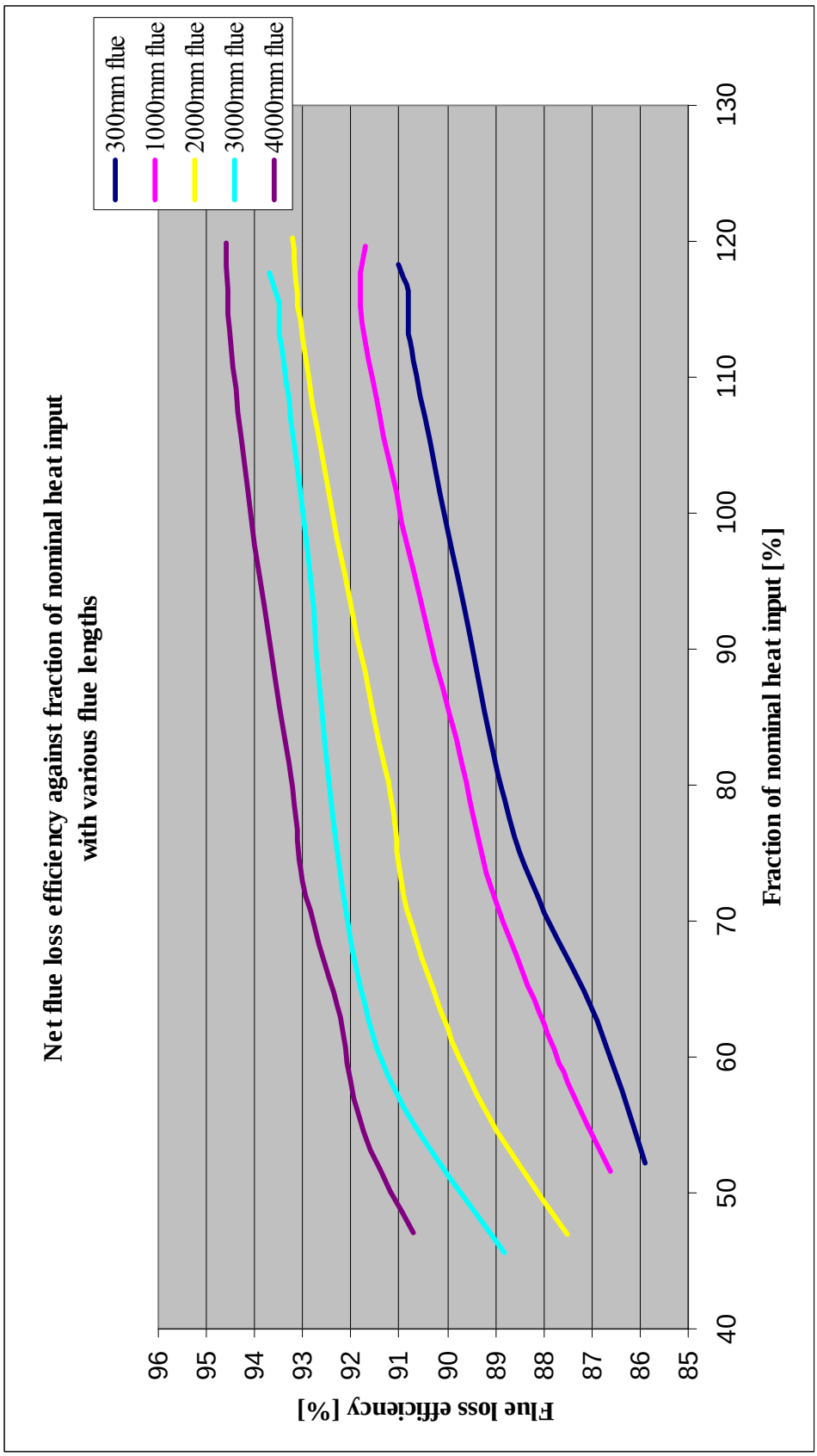


Figure 4.1 Flue loss efficiency with various flue lengths

4.1.3 *Evaluation of the Test Results*

The primary determinants of flue losses are;

- The % of CO_2 or O_2 in the flue gas
- The temperature of the flue gas
- The temperature of the air going into the boiler

$$\eta_{flue\ loss} = 100 - \left(a + \frac{b}{CO_2} \right) * \frac{\left(T_f - T_a \right)}{100} \tag{4.1}$$

a and b are the coefficients according to gas type;

CO_2 is the carbon dioxide content in the dry products of combustion, in percent;

T_f is the temperature of the flue gas, in °C;

T_a is the ambient temperature, in °C (BSI – EN483).

Due to the flue length increases; the temperature of the flue gas has been decreasing and the concentration of CO_2 has been increasing.

The effect of the flue gas temperature decrease, flue losses are decreasing and the flue loss efficiency is increasing. For more efficient appliance the flue gas temperature should be less as possible.

4.2 Study on Convective and Radiative Heat Losses

4.2.1 Calculation of the Convective Heat Loss

Convective heat losses that related with natural convection processes were calculated by using some experimental heat transfer equation;

First of all, the coefficient of convection (α) has to be calculated.

For natural convection on vertical isothermal surfaces, (front, left and right side of the combustion chamber) the following equations were used (Kakac & Yuncu, 1999). The thermal properties should all be evaluated at film temperature ($T_f = (T_w + T_\infty) / 2$) except for β, which is to be evaluated at T_∞.

$$Gr_L = \frac{g\beta(T_w - T_\infty)L^3}{\nu^2} \tag{4.2}$$

Gr_L : Grashof number
g : gravity [m^2/s]
β : coefficient of thermal expansion [K^{-1}]
T_w : Temperature of the surface [ºC]
T_∞ : Temperature of the ambient [ºC]
ν : Kinematic viscosity [m^2/s]

$$\bar{Nu} = 0.64(Gr_L)^{1/4}\,\mathrm{Pr}^{1/2}(0.861 + \mathrm{Pr})^{-1/4} \tag{4.3}$$

Nu : Nusselt number
Gr_L : Grashof number
Pr : Prandtl number

coefficient of convection α is;

$$\bar{\alpha} = \bar{Nu}\, x \frac{\lambda}{L} \tag{4.4}$$

For natural convection on horizontal isothermal surfaces, (bottom and top side of the combustion chamber) the following equations were used (Kakac & Yuncu, 1999).

$$Ra_L = Gr_L \,\mathrm{Pr} = \frac{g\beta(T_w - T_\infty)L_{ch}^{\,3}}{\nu^2}\mathrm{Pr} \tag{4.5}$$

λ : Coefficient of conduction [W/mK]

L_{ch} : Characteristic length [m] (for rectangular surface; L= area / perimeter)

Ra_L : Rayleigh number

if $Ra_L \leq 10^9$ laminar flow , $Ra_L \geq 10^9$ turbulent flow)

$$\bar{Nu} = C(Ra_L)^m \tag{4.6}$$

if the fluid flow above a hot plate (as the top of the combustion chamber)

C = 0.54 , m = ¼

If the fluid flow below a hot plate (as the bottom of the combustion chamber)

C = 0.27 , m = ¼

finally heat transfer rate Q_{conv} is;

$$Q_{conv} = \bar{\alpha} * A * (T_s - T_\infty) \tag{4.7}$$

Q_{conv} : Convectional heat transfer rate, [W]

α : Coefficient of convection, [W / m^2 * K]

A : Surface area, [m^2]

T_s : Temperature of surface of the combustion chamber, [ºC]

T_∞ : Ambient temperature, [ºC]

For determining the convective losses, the sides of the combustion chamber had been taken as control volume and divided into squares. The average temperature of each side was measured (see figure 4.2)

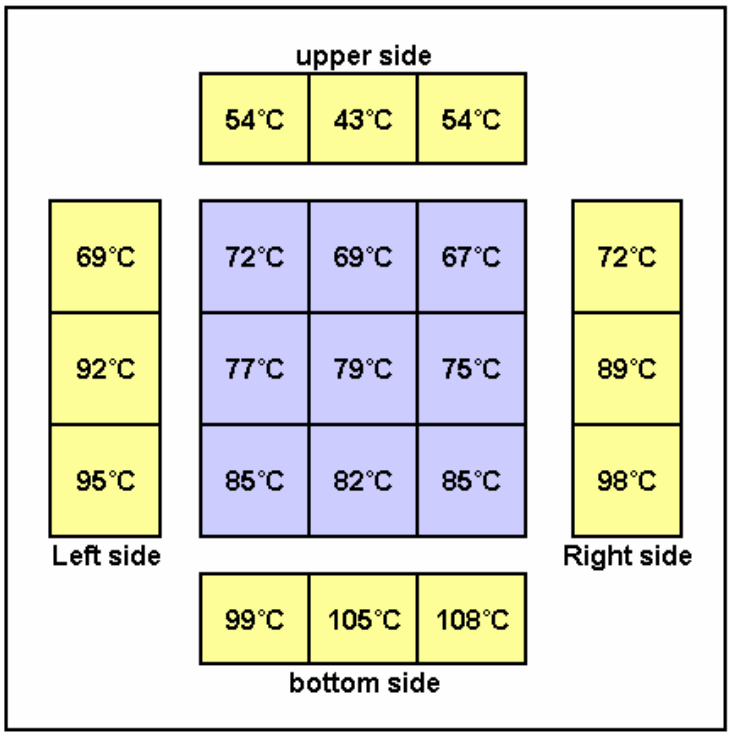


Figure 4.2 Temperature of the sides of combustion chamber

Average temperatures that were used for calculation;

Front side of combustion chamber: 80˚C
Left side of combustion chamber: 90˚C
Right side of combustion chamber: 90˚C
Bottom side of combustion chamber: 105˚C
Left side of combustion chamber: 50˚C

Front side of the combustion chamber;

g : 9.81 m^2/s

β : $3.43x10^{-3}$ K^{-1} (evaluated at T_∞)

T_w : 80 ºC

T_∞ : 20 ºC

T_f : 50 ºC

L : 0,5 m

ϒ : $1.787x10^{-5}$ m^2/s (see from Appendix – Table A-6)

Pr : 0.710 (see from Appendix – Table A-6)

λ : 0.0278 W/mK (see from Appendix – Table A-6)

$$Gr_L = \frac{g\beta(T_w - T_\infty)L^3}{\nu^2} = \frac{9.81*3.43*10^{-3}*(80-20)*0.5^3}{(1.787*10^{-5})^2} = 7.9*10^8$$

$$Ra_L = Gr_L \Pr = 5.61*10^8 \le 10^9 \Rightarrow laminar\ \ flow$$

$$\bar{Nu} = 0.64(Gr_L)^{1/4} \Pr^{1/2}(0.861+\Pr)^{-1/4} = 80.76$$

$$\bar{\alpha} = \bar{Nu}\frac{\lambda}{L} = 80.76*\frac{0.0278}{0.5} = 4.49 W/m^2K$$

$$Q_{front} = \bar{\alpha}*A*(T_s - T_\infty) = 59.27W$$

Left and right sides of the combustion chamber;

g : 9.81 m^2/s

β : $3.43x10^{-3}$ K^{-1}

T_w : 90 ºC

T_∞ : 20 ºC

T_f : 55 ºC

L : 0,5 m

w : 0.3 m

ϒ : $1.84x10^{-5}$ m^2/s (see from Appendix – Table A-6)

Pr : 0.709 (see from Appendix – Table A-6)

λ : 0.028 W/mK (see from Appendix – Table A-6)

$$Gr_L = \frac{g\beta(T_w - T_\infty)L^3}{\nu^2} = \frac{9.81*3.43*10^{-3}*(90-20)*0.5^3}{(1.84*10^{-5})^2} = 8.7*10^8$$

$$Ra_L = Gr_L \Pr = 6.17*10^8 \le 10^9 \Rightarrow la\min ar \ \ flow$$

$$\bar{Nu} = 0.64(Gr_L)^{1/4} \Pr^{1/2}(0.861+\Pr)^{-1/4} = 82.67$$

$$\bar{\alpha} = \bar{Nu}\frac{\lambda}{L} = 80.76*\frac{0.0278}{0.5} = 4.63 W/m^2K$$

$$Q_{left} = \bar{\alpha}*A*(T_s - T_\infty) = 48.61W$$

$$Q_{right} = \bar{\alpha}*A*(T_s - T_\infty) = 48.61W$$

Bottom side of the combustion chamber;

g : 9.81 m^2/s

β : 3.43x10^{-3} K^{-1}

T_w : 105 ºC

T_∞ : 20 ºC

T_f : 62.5 ºC

L : 0,44 m

w : 0.3 m

L_{ch} : 0.44 x 0.3 / (2 x (0.44 + 0.3)) = 0.0891 m

ϒ : 1.915x10^{-5} m^2/s (see from Appendix – Table A-6)

Pr : 0.709 (see from Appendix – Table A-6)

λ : 0.0288 W/mK (see from Appendix – Table A-6)

C : 0.27

m : 1/4

$$Ra_L = Gr_L \Pr = \frac{g\beta(T_w - T_\infty)L_{ch}^{\ 3}}{\nu^2}\Pr = 3.91*10^6 \le 10^9 \Rightarrow la\min ar \ \ flow$$

$$\bar{Nu} = C(Ra_L)^m = 0.27\left(3.91*10^6\right)^{1/4} = 12$$

$$\bar{\alpha} = \bar{Nu}\frac{\lambda}{L} = 12*\frac{0.0288}{0.0891} = 3.88 W/m^2K$$

$$Q_{bottom} = \bar{\alpha}*A*(T_s - T_\infty) = 43.53W$$

Upper side of the combustion chamber;

g : 9.81 m^2/s

β : 3.43x10^{-3} K^{-1}

T_w : 50 ºC

T_∞ : 20 ºC

T_f : 35 ºC

L : 0,44 m

w : 0.3 m

L_{ch} : 0.44 x 0.3 / (2 x (0.44 + 0.3)) = 0.0891 m

ϒ : 1.65x10^{-5} m^2/s (see from Appendix – Table A-6)

Pr : 0.711 (see from Appendix – Table A-6)

λ : 0.0268 W/mK (see from Appendix – Table A-6)

C : 0.54

m : 1/4

$$Ra_L = Gr_L \Pr = \frac{g\beta(T_w - T_\infty)L_{ch}^{\ 3}}{\nu^2}\Pr = 1.86*10^6 \le 10^9 \Rightarrow la\min ar \ \ flow$$

$$\bar{Nu} = C(Ra_L)^m = 0.54\left(3.91*10^6\right)^{1/4} = 20$$

$$\bar{\alpha} = \bar{Nu}\frac{\lambda}{L} = 20*\frac{0.0288}{0.0891} = 6W/m^2K$$

$$Q_{upper} = \bar{\alpha}*A*(T_s - T_\infty) = 23.76W$$

$$Q_{conv\,total} = Q_{front} + Q_{right} + Q_{left} + Q_{bottom} + Q_{upper} = 223.78\,W$$

4.2.2 Calculation of the Radiative Heat Loss

For determining the radiative losses, some temperature measurements had been taken on the cover of the appliance that divided into squares.(see figure 4.3)

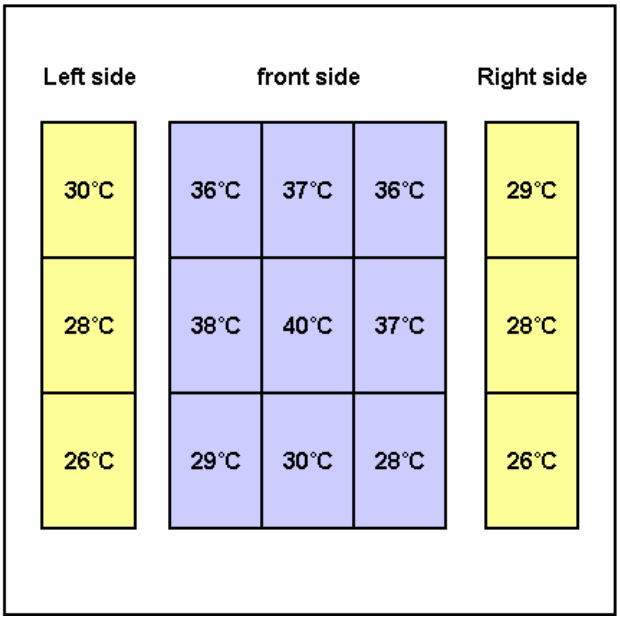


Figure 4.3 Temperature distribution on the cover of the appliance

The radiation heat losses were calculated by using simplified heat transfer equation; (Kakac & Yuncu, 1999).

$$Q_{rad} = \varepsilon * \sigma * A * (T_s^4 - T_a^4) \quad (4.8)$$

Q_{rad} : Radiational heat transfer rate, (W)

ε : Emittance factor, $0 < \varepsilon < 1$, (see Appendix – Table A-7)

σ : Stefan-Boltzmann constant, $5.6704 * 10^{-8}$ ($W / m^2 * K^4$)

A : Surface area, (m^2)

T_s : Temperature of surface of the boiler, K

T_a : Ambient temperature, K , (296,4 K)

Mantel surface of the combi boiler is painted white so emittance of the surface has taken 0.9 regarding to surface properties (see Appendix - Table A-7).

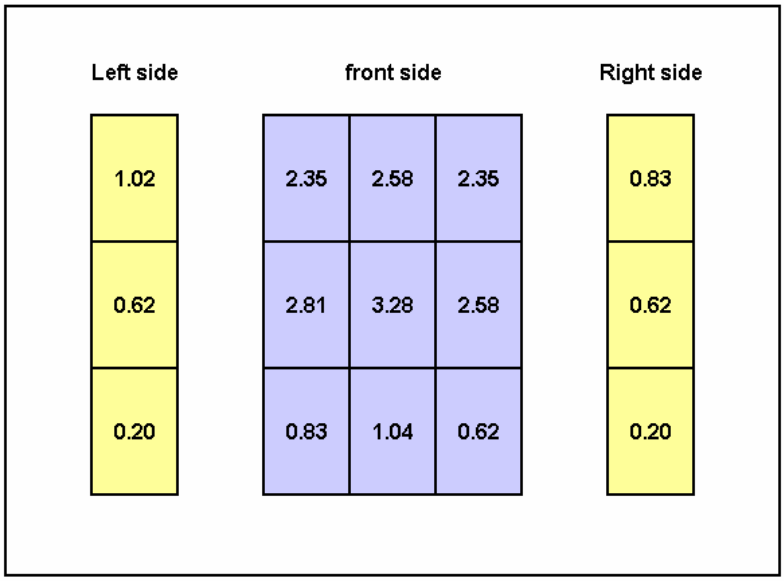


Figure 4.4 Radiational Losses

Total radiative losses are;

$$Q_{rad\ total} = Q_{front} + Q_{right} + Q_{left} = 21.95\,W$$

And finally; total heat loss from the cover of the appliance is;

$$Q_{total} = Q_{conv} + Q_{rad} = 245.73\,W$$

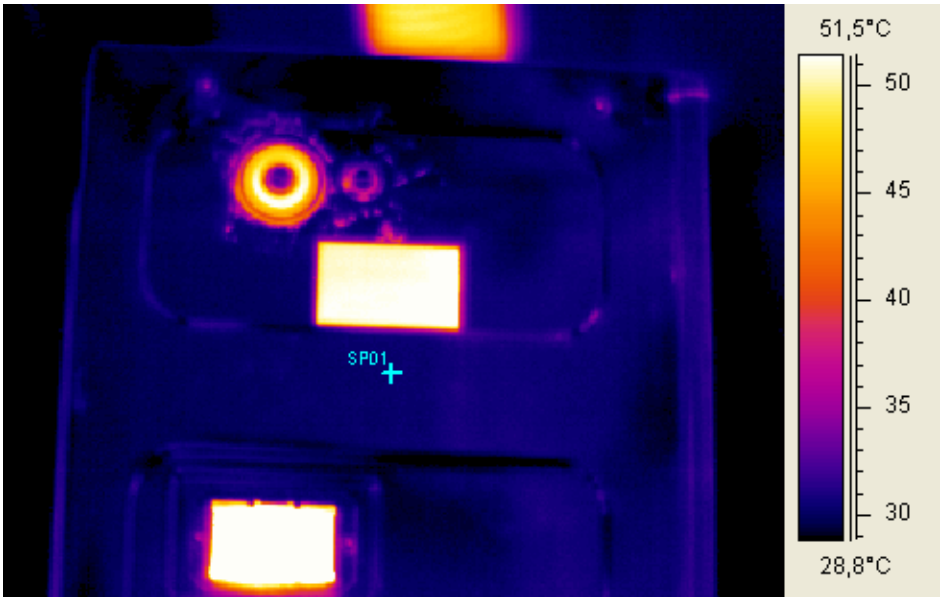
And the total % of the case loss for 24kW appliance is;

245.73 / 24000 = **%1.02**

4.3 Thermal Camera Photos

To see the temperature distribution on the surfaces of appliance it was taken some photos with thermal camera.

4.3.1 Front View of the Appliance (3a AME – 24 kW)



same photo with different color range

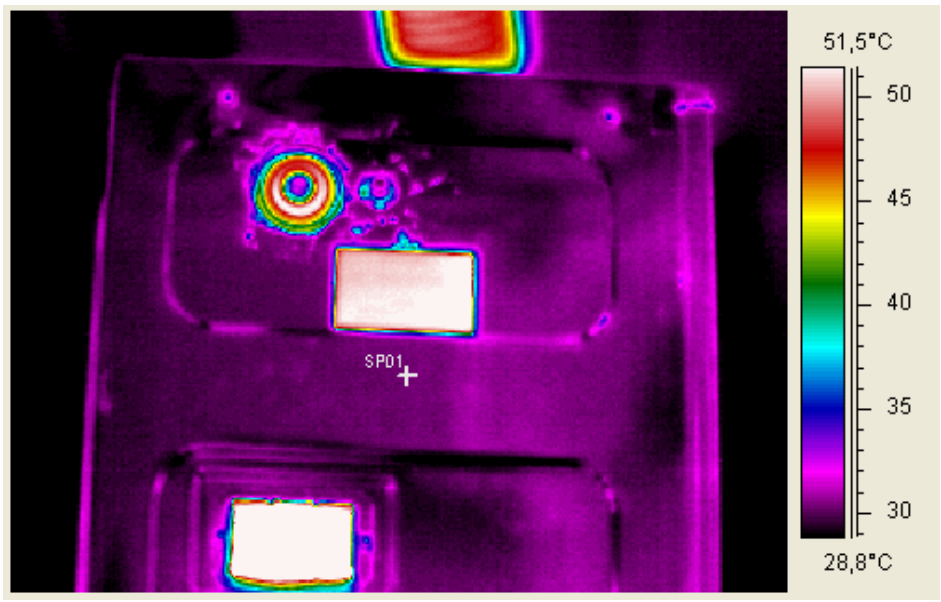
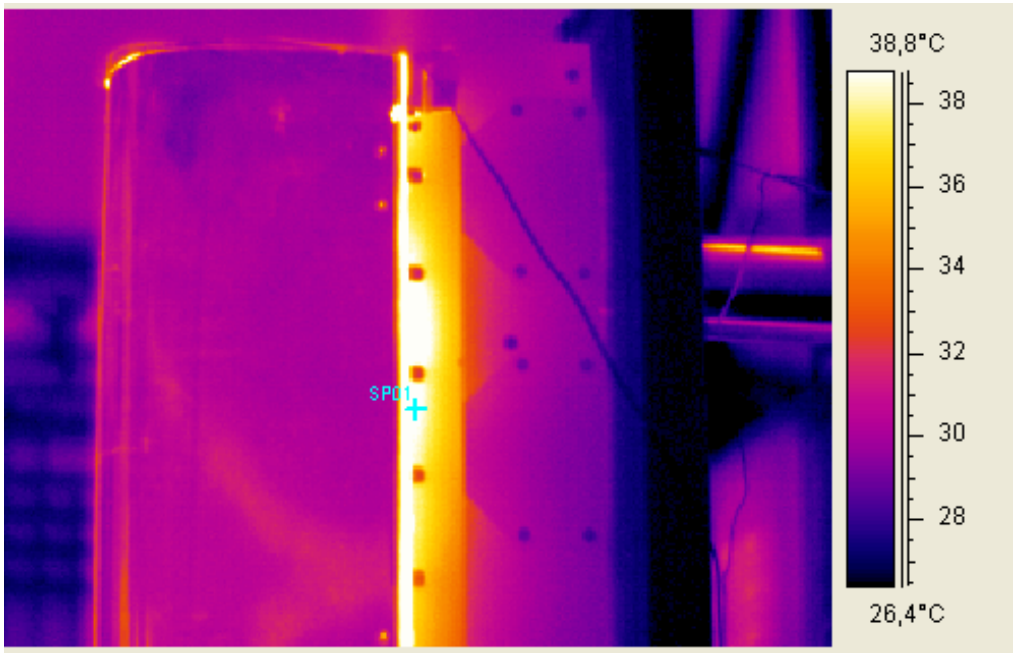


Figure 4.5 Thermal camera photos (front side)

4.3.2 Right Side of the Appliance



with different color range

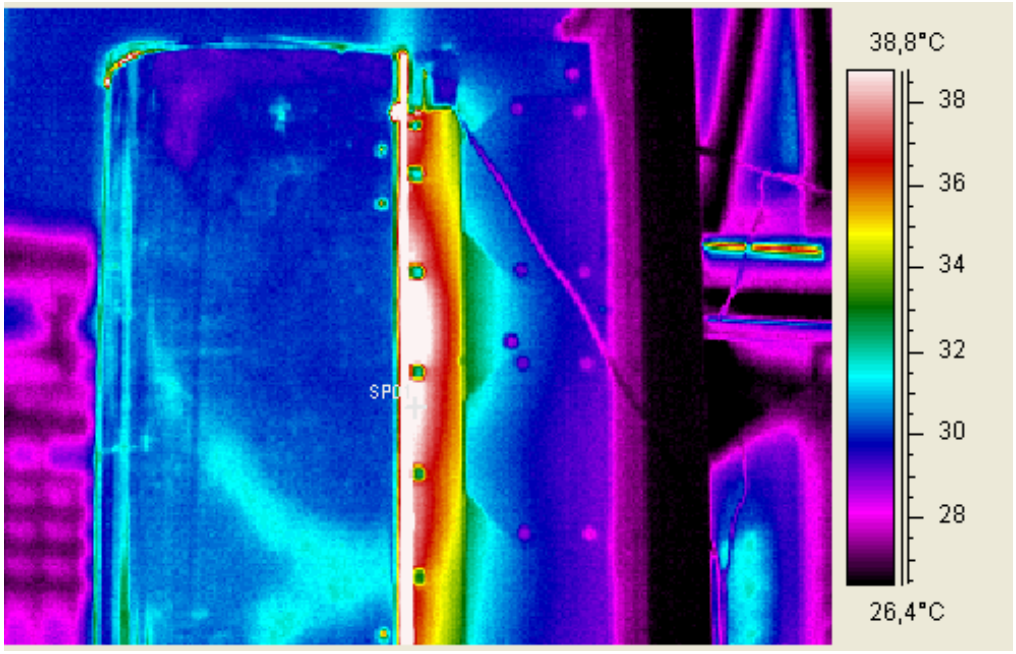
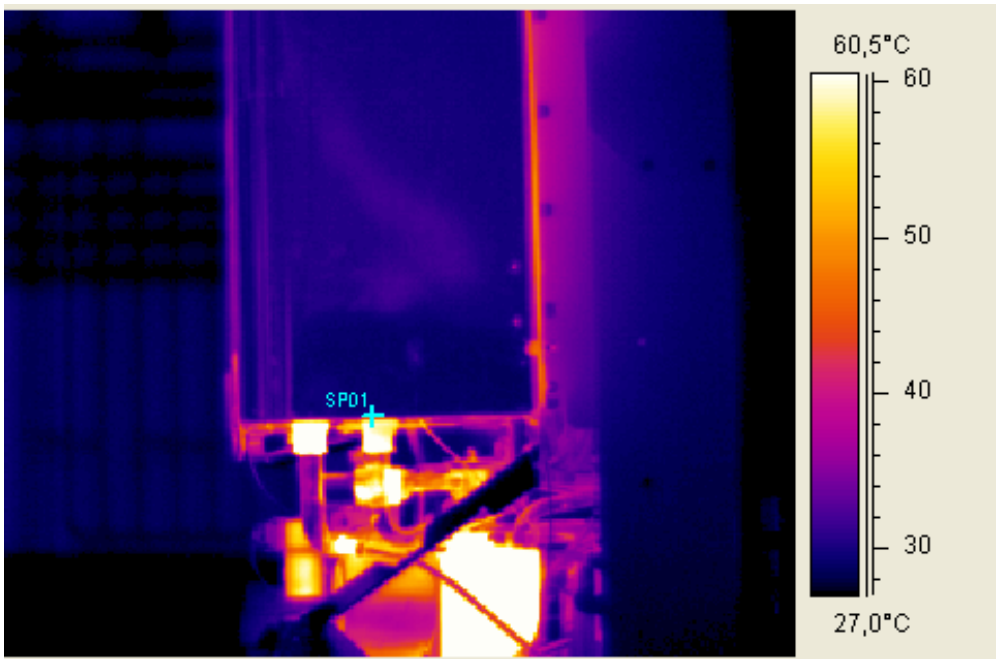


Figure 4.6 Thermal camera photos (right side)

4.3.3 Pipe Connections of the Appliance



with different color range

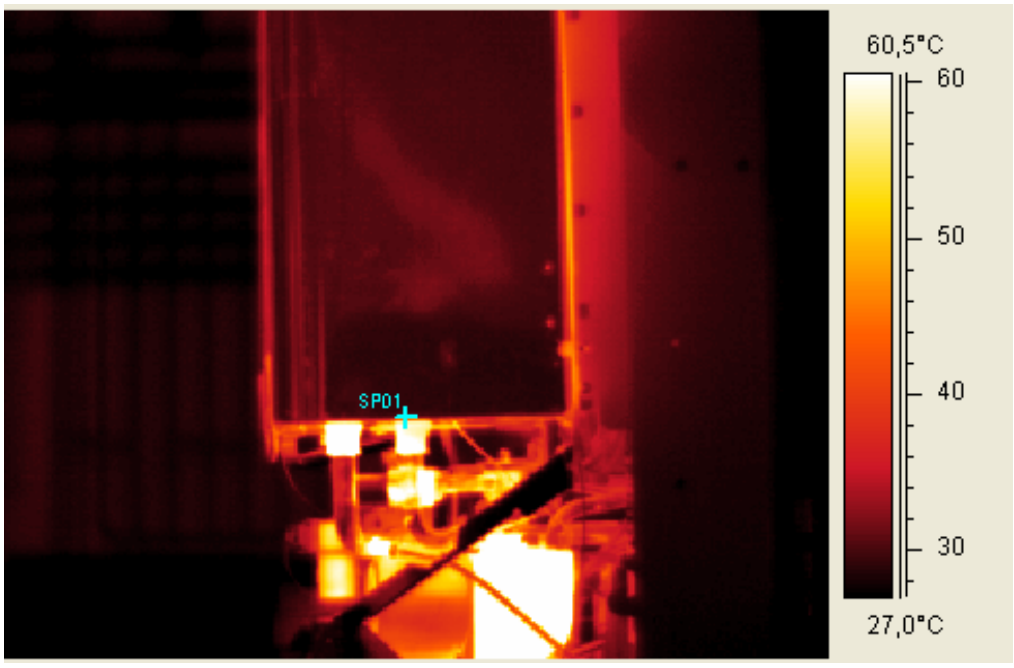


Figure 4.7 Thermal camera photos (pipe connection)

4.4 Study with Modulated Fan

It was carried out a serious of tests with and w/o modulated fan to observe the effects of modulated fan on the flue loss efficiency.

4.4.1 Works Done

Firstly; the fan speed was fixed at 2400 rpm by using external cable connection. And the combustion curve was constituted. The flue loss efficiency curve was also constituted with various heat input values.

All works were repeated by using modulated fan with a range of 1800 – 2400 rpm. And also a comparison has given between the flue loss efficiencies with and w/o modulation fan.

4.4.2 Test Conditions

Project **:** 3a AME
Appliance **:** ZWN24-6 AME
Flue Length **:** 600mm
Direction **:** Left (the worst direction)
Test gas **:** Natural Gas (with H_i =33.809 MJ/m^3 and Rel. density: 0.569)
Inlet pressure **:** 20 mbar
Fan speed range **:** 1800 – 2400 rpm for modulation test
Test performed by **:** E. Kesdogan

Table 4.2 Combustion values w/o modulated fan (fan speed was fixed 2400 rpm)

Project	3a AME					Date	20.09.2004			
Appliance	ZWN24-6AME									
Engineer	E.Kesdogan									
Test/Clause	Combustion without modulation fan									
Test Number										
Flue Length	mm	600	600	600	600	600	600	600	600	600
Direction		Left	Left	Left	Left	Left	Left	Left	Left	Left
Restrictor	mm	75	75	75	75	75	75	75	75	75
Inlet Pressure	mbar	20	20	20	20	20	20	20	20	20
Fan Speed [rev/min]	X	2400	2400	2400	2400	2400	2400	2400	2400	2400
Injector Marked	X	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*145
Code Plug		130	130	130	130	130	130	130	130	130
Test Gas		Line Gas	Line Gas	Line Gas	Line Gas	Line Gas	Line Gas	Line Gas	Line Gas	Line Gas
CV (net)	MJ/m³	33,809	33,809	33,809	33,809	33,809	33,809	33,809	33,809	33,809
S.G.		0,569	0,569	0,569	0,569	0,569	0,569	0,569	0,569	0,569
Atmos. Press.	mbar	1005,5	1005,5	1005,5	1005,5	1005,4	1005,3	1005,3	1005,3	1005,3
Amb. Temp.	°C	23,4	23,4	23,5	23,4	23,4	23,3	23,4	23,5	23,5
Burner Pressure	mbar	19	18,2	16	14	11	8,4	7	5	4
Meter Pressure	mbar	21	21	21	21	21	21	21	21	21
Meter Temp.	°C	23,4	23,1	22,7	23,5	23,7	23,8	23,8	23,9	23,9
Volume	dm³	100	100	100	100	100	50	50	50	60
Time	secs	118,08	120,75	127,4	137,7	155,5	85,24	93,8	108	140,13
Gas Rate	dm³/hr	3048,78	2981,37	2825,75	2614,38	2315,11	2111,68	1918,98	1666,67	1541,43
Meter Factor		0,985	0,985	0,984	0,985	0,985	0,985	0,985	0,985	0,985
S.V.P.		N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
G.V.F.		0,985	0,986	0,987	0,984	0,984	0,983	0,983	0,983	0,983
Corr. Gas Rate	dm³/hr	2958	2895,54	2744,39	2533,96	2243,9	2044,64	1858,06	1613,76	1492,5
Heat Input	kW	27,8	27,21	25,79	23,82	21,09	19,22	17,46	15,17	14,03
Nominal	kW	26,4	26,4	26,4	26,4	26,4	26,4	26,4	26,4	26,4
Input	%	105,3	103,1	97,7	90,2	79,9	72,8	66,1	57,5	53,1
Voltage	V	230	230	230	230	230	230	230	230	230
CO2	%	6,8	6,61	6,16	5,76	5,2	4,49	4,1	3,72	3,3
CO	ppm	41	38	31	26	19	19	24	35	44
AFCO	ppm	71	67	59	53	43	50	68	110	156
Flue Gas Temp.	°C	166,6	164,2	159	154,7	147,8	140	133	125,6	122,5
Flue Loss (net)	%	8,94	9,01	9,22	9,47	9,83	10,52	10,73	10,92	11,83
Flue Loss (gross)	%	9,92	10	10,23	10,51	10,91	11,68	11,91	12,12	13,13
Flue Loss Efficiency	%	91,06	90,99	90,78	90,53	90,17	89,48	89,27	89,08	88,17

Table 4.3 Combustion values with modulated fan (fan speed varies between 1800 - 2400 rpm)

Project	3a AME					Date	22.09.2004	
Appliance	ZWN24-6AME							
Engineer	E.Kesdogan							
Test/Clause	Combustion with modulation fan							
Test Number								
Flue Length	mm	600	600	600	600	600	600	600	600
Direction		Left	Left	Left	Left	Left	Left	Left	Left
Restrictor	mm	75	75	75	75	75	75	75	75
Inlet Pressure	mbar	20	20	20	20	20	20	20	20
Fan speed (rpm)		2400	2400	2320	2160	2100	1920	1800	1800
Injector Marked	X	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144	2*150+6*144
Code Plug		130	130	130	130	130	130	130	130
Test Gas		Line Gas	Line Gas	Line Gas	Line Gas	Line Gas	Line Gas	Line Gas	Line Gas
CV (net)	MJ/m³	33,809	33,809	33,809	33,809	33,809	33,809	33,809	33,809
S.G.		0,569	0,569	0,569	0,569	0,569	0,569	0,569	0,569
Atmos. Press.	mbar	1005,5	1004,3	1004,2	1004,3	1004,4	1004,3	1004,3	1004,4
Amb. Temp.	°C	23,4	21,4	21,4	21,6	21,2	21,3	21,6	22
Burner Pressure	mbar	19	16	15	13,5	11	8	6	5
Meter Pressure	mbar	21	21,5	21,4	21,4	21,6	21,8	21,8	21,9
Meter Temp.	°C	23,4	20,6	21,4	21,7	21,8	21,9	22	22,1
Volume	dm³	100	100	100	100	100	100	100	100
Time	secs	118,08	127	131,3	140	152,5	178,9	210,5	222,6
Gas Rate	dm³/hr	3048,78	2834,65	2741,81	2571,43	2360,66	2012,3	1710,21	1617,25
Meter Factor		0,985	0,987	0,986	0,985	0,985	0,984	0,984	0,983
S.V.P.		N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
G.V.F.		0,985	0,993	0,99	0,99	0,989	0,989	0,989	0,989
Corr. Gas Rate	dm³/hr	2958	2778,21	2676,39	2507,53	2299,67	1958,32	1664,34	1572,27
Heat Input	kW	27,8	26,11	25,16	23,57	21,61	18,41	15,64	14,78
Nominal	kW	26,4	26,4	26,4	26,4	26,4	26,4	26,4	26,4
Input	%	105,3	98,9	95,3	89,3	81,9	69,7	59,2	56
Voltage	V	230	230	230	230	230	230	230	230
CO2	%	6,8	6,68	6,42	6	5,64	5	4,5	4,1
CO	ppm	41	33	28	20,2	16	15	17	17
AFCO	ppm	71	58	51	39	33	35	44	49
Flue Gas Temp.	°C	166,6	162	155	148	141,3	129	118,2	112,4
Flue Loss (net)	%	8,94	8,91	8,77	8,8	8,83	8,81	8,69	8,85
Flue Loss (gross)	%	9,92	9,89	9,73	9,77	9,8	9,78	9,64	9,82
Flue Loss Efficiency	%	91,06	91,09	91,23	91,2	91,17	91,19	91,31	91,15

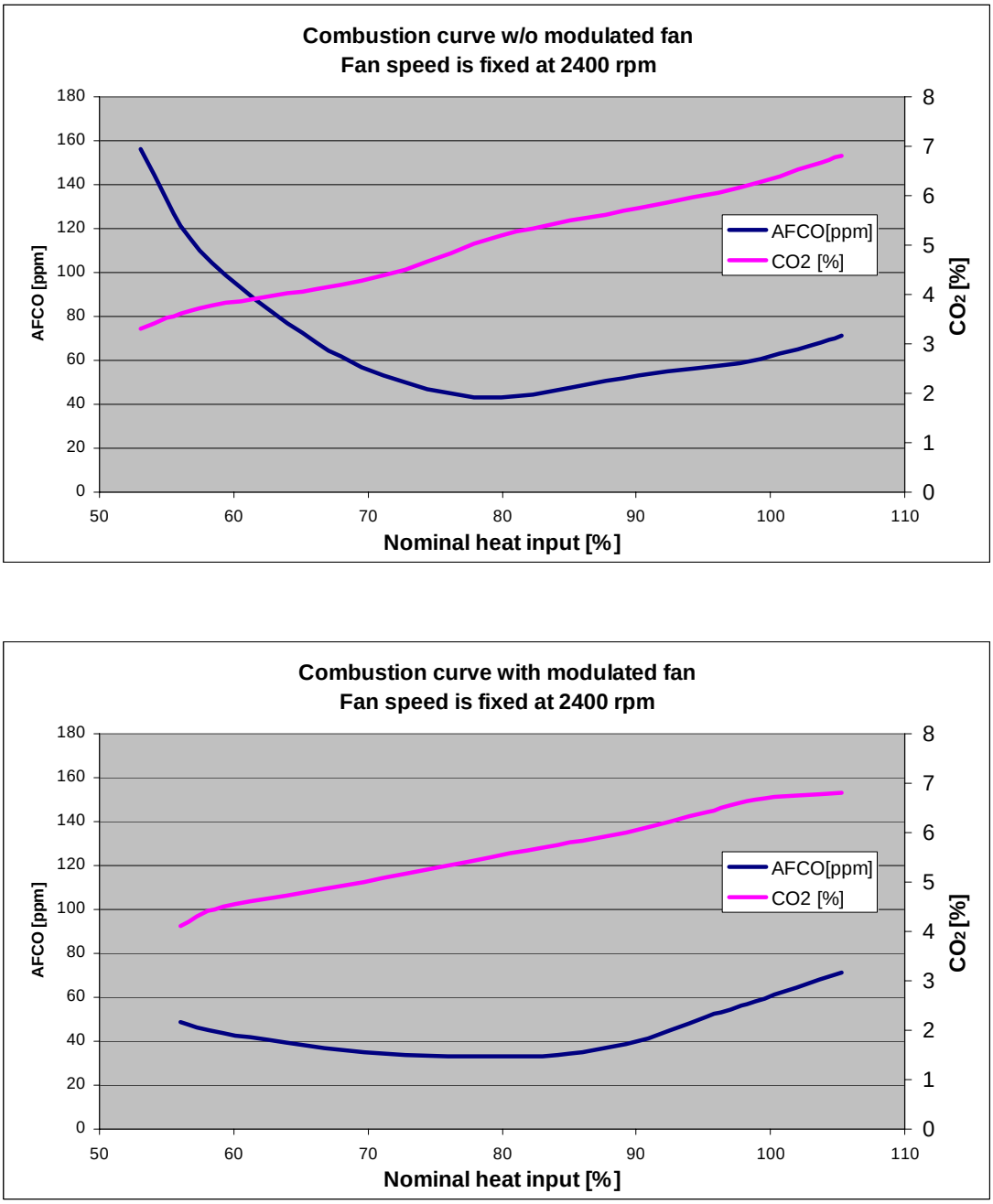


Figure 4.8 Combustion curve with and w/o modulated fan

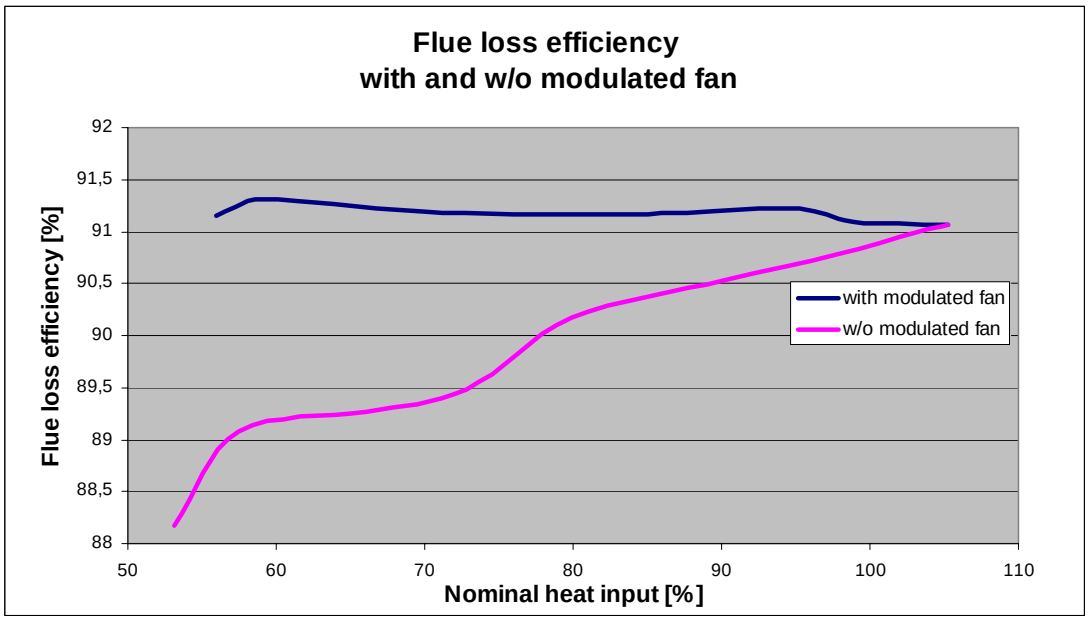


Figure 4.9 Flue loss efficiency with and w/o modulated fan

4.4.4 Evaluation of Test Results

For hot water heating the minimum capacity of a combi boiler should be around 22-24 kW. However, in a well insulated house the space heating -at its peak- may require less than 8-10 kW. It means that the appliance -with modulated fan and gas valve- is more feasible than w/o modulated fan and gas valve.

With modulated fan; the flue loss efficiency varies between % 91.06 and % 91.31, almost constant, w/o modulated fan (fixed at 2400 rpm); the flue loss efficiency varies between % 88.17 and % 91.06 due to the nominal heat input.

Test results show that the modulated fan gives more satisfactory results than fan with fixed speed.

4.5 Study with Low Flow Temperature

4.5.1 Test Conditions

For that purpose; appliance was operated at maximum and minimum load with different flow temperatures to see the flue loss efficiency variations.

Flow temperatures:

- 30 °C – 50 °C
- 40 °C – 60 °C
- 50 °C – 70 °C
- 60 °C – 80 °C
- 70 °C – 90 °C

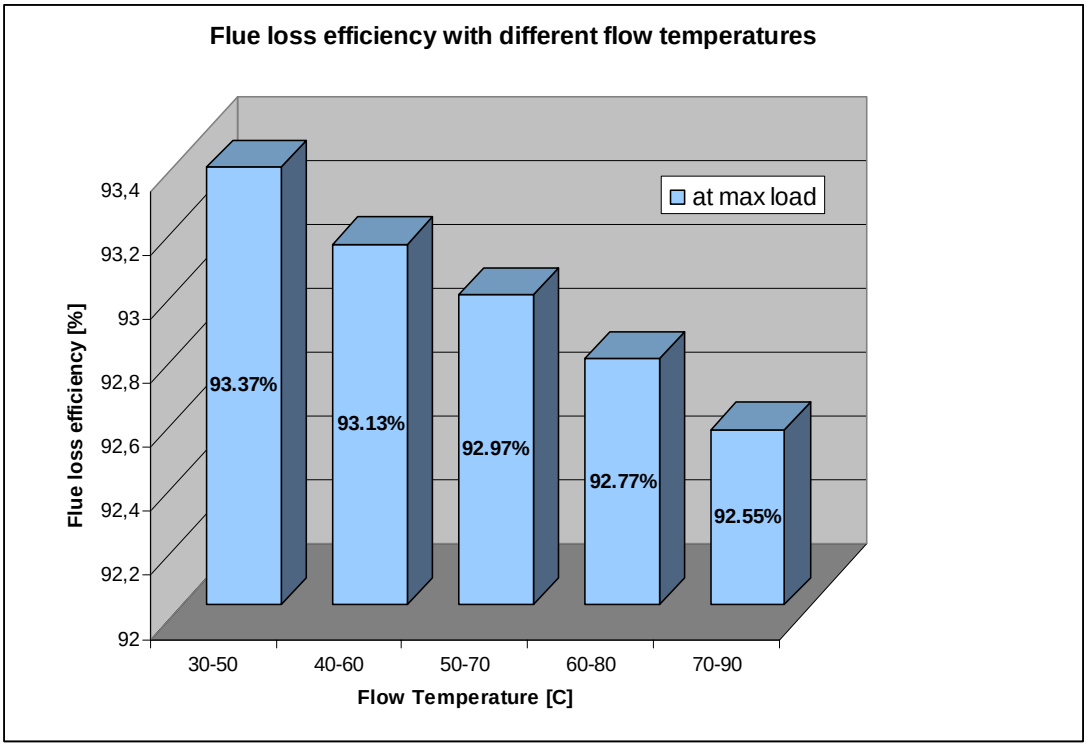


Figure 4.10 Flue loss efficiency with different flow temperatures at max load

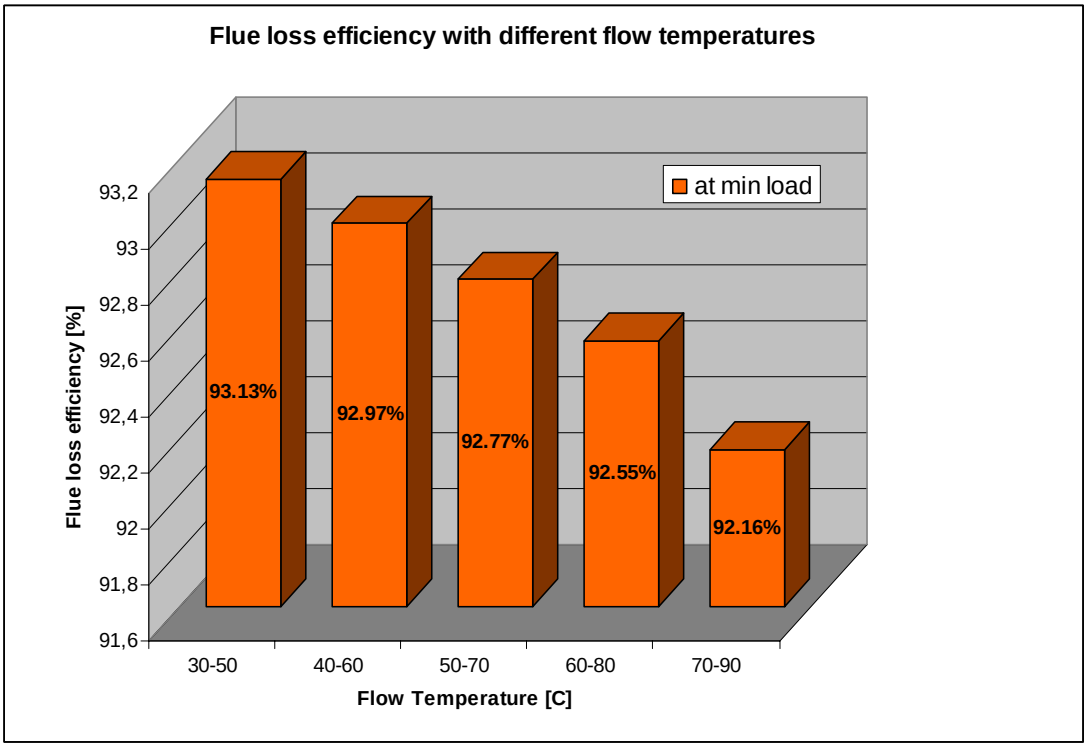


Figure 4.11 Flue loss efficiency with different flow temperatures at min load

4.5.2 Evaluation of test results

The flue gas temperatures are decreasing when working low temperature flows. Therefore flue loss efficiency is increasing with flow temperature decreases.

As a result of this test we can say that; with respect to flue loss efficiency, low temperature heating systems are more efficient than normal systems. For example; floor heating systems, wall heating systems, condensing appliances etc…

4.6 Efficiency Measurement according to EN 483

Table 4.4 Efficiency measurements according to EN 483

Project	Gzt 3a				
Appliance	Gzt3a RSF 24 kW				
Technician	E.Kesdogan				
Test/Clause	Efficiency test				
Test Number		1	2	3	
Flue Length	mm	300	300	300	
Direction		Right	Right	Right	
Restrictor Diameter	mm	N/A	N/A	N/A	
Inlet Pressure	mbar	20	20	20	
Injector(s) Marked	X	2*150+6*144	2*150+6*144	2*150+6*144	
Code Plug		130	130	130	
Test Gas		NG	NG	NG	
CV (net)	MJ/m³	33,62	33,62	33,62	
S.G		0,573	0,573	0,573	
Atmos. Press	mbar	1005	1004	1004,5	
Amb. Temp.	ºC	23,6	23,5	23,6	
Burner Pressure	mbar	16,7	16,7	16,7	
Meter Pressure	mbar	24,87	24,86	24,86	
Meter Temp.	ºC	25,3	25,9	26,36	
Volume	dm³	400	400	400	
Time	secs	484,88	485,1	484,47	
Gas Rate	dm³/hr	2969,81	2968,46	2972,32	
Meter Factor		0,983	0,983	0,983	
G.V.F		0,982	0,979	0,978	
Corr. Gas Rate	dm³/hr	2866,78	2856,72	2857,51	
Heat Input	kW	26,79	26,7	26,71	
Nominal	kW	26,4	26,4	26,4	
Input	%	101,5	101,1	101,2	
CO_2	%	6,6	6,7	6,6	
CO	ppm	25	30	30	
AFCO	ppm	44	52	53	
Flue Gas Temp.	ºC	160	161,12	161,16	
Flue Loss (net)	%	8,74	8,70	8,81	
Flue Loss (gross)	%	9,7	9,66	9,78	
Flow Temp.	ºC	59,56	59,68	60	
Return Temp.	ºC	79,56	79,67	80,07	
Inlet Temp.	ºC	21,11	21,32	21,7	
Outlet Temp.	ºC	47,91	48,13	48,64	
Water	kg	103	107,5	106,12	
Time	secs	499	520	517	
Rig Loss	kW	1	1	1	
Output	kW	24,16	24,21	24,15	
Efficiency (net)	%	90,18	90,67	90,42	
Efficiency (gross)	%	81,25	81,69	81,47	
		Average efficiency = 90.43 %			

4.7 Study on the Excess Air Distribution on the Heat Exchanger

Excess air is so important for combustion quality. Ideally it should be between 1 and 1,4. It is calculated as a function of CO_2 or O_2 .

$$\lambda = 1 + 0{,}9 * \left[\left(CO_{2\max} / CO_2 \right) - 1 \right] \tag{4.8}$$

4.7.1 Test Conditions

Appliance **:** 3a AME 24 kW
Flue Length **:** 3000mm. (the longest flue length for this appliance)
Direction of flue **:** Right (the worst direction for this length)
Injectors **:** 2*150 + 6*144
Test gas **:** Natural gas (H_i = 33.856 , Rel. Density = 0.569)
Inlet pressure **:** 20 mbar
Burner pressure **:** For max load P = 16.4 mbar , for min load P = 5.3 mbar
Flow/Return Temp. **:** 80 / 60 °C
Fan speed **:** For max load N = 2400 rpm , for min load N = 1800 rpm
Supply voltage **:** 230 V
Test performed by **:** E. Kesdogan
Date **:** 02-03/06/2004

The width of heat exchanger was divided into 7 equal parts with a distance of 55 mm between each part. The depth of heat exchanger was also divided into 7 parts with a distance of 27 mm between each part. Therefore there were 7*7 = 49 measurement points. Around the upper surface of heat exchanger it was taken CO_2 and CO emission values by using a measurement probe from each measurement point.

4.7.2 Test Results

Table 4.5 Emission and excess air distribution on the heat exchanger for maximum load

CO_2 [%]		Distance from left side [mm]						
		30	85	140	195	250	305	360
Distance from the front side [mm]	10	0.90	7.19	9.00	9.58	8.67	7.70	0.93
	37	2.90	10.09	10.36	10.33	9.82	7.94	1.75
	64	3.02	10.68	10.52	10.04	9.82	8.20	2.65
	91	2.66	10.81	10.62	10.50	10.39	5.66	3.77
	118	2.89	10.71	10.73	10.40	10.30	6.38	3.26
	145	2.72	9.11	10.17	9.00	9.21	7.11	1.16
	172	0.86	2.45	3.15	2.30	2.90	3.34	0.44

λ		Distance from left side [mm]						
		30	85	140	195	250	305	360
Distance from the front side [mm]	10	11.83	1.57	1.27	1.20	1.32	1.47	11.45
	37	3.74	1.15	1.12	1.12	1.18	1.43	6.13
	64	3.60	1.09	1.10	1.15	1.18	1.39	4.08
	91	4.07	1.08	1.09	1.11	1.12	1.97	2.90
	118	3.75	1.09	1.08	1.12	1.12	1.75	3.34
	145	3.98	1.26	1.14	1.27	1.25	1.58	9.20
	172	12.38	4.41	3.45	4.69	3.74	3.26	24.09

Table 4.6 Emission and excess air distribution on the heat exchanger for minimum load

CO_2 [%]		Distance from left side [mm]						
		30	85	140	195	250	305	360
Distance from the front side [mm]	10	0.17	3.05	5.29	3.69	3.40	5.24	0.25
	37	0.56	6.17	9.40	6.43	6.46	5.13	0.53
	64	0.76	6.88	10.51	10.00	9.56	3.14	0.80
	91	1.15	7.58	10.56	10.25	9.94	3.24	1.09
	118	1.95	9.36	9.00	9.82	9.71	2.38	0.90
	145	1.02	5.16	4.47	3.74	5.02	3.05	0.30
	172	0.30	1.73	1.12	0.93	1.71	1.86	0,15

λ		Distance from left side [mm]						
		30	85	140	195	250	305	360
Distance from the front side [mm]	10	62.20	3.56	2.10	2.96	3.21	2.11	42.33
	37	18.95	1.81	1.22	1.74	1.73	2.16	20.02
	64	13.99	1.63	1.10	1.16	1.20	3.46	13.30
	91	9.28	1.49	1.10	1.13	1.16	3.36	9.79
	118	5.51	1.23	1.27	1.18	1.19	4.54	11.83
	145	10.45	2.15	2.46	2.92	2.20	3.56	35.29
	172	35.29	6.20	9.53	11.45	6.27	5.78	70.48

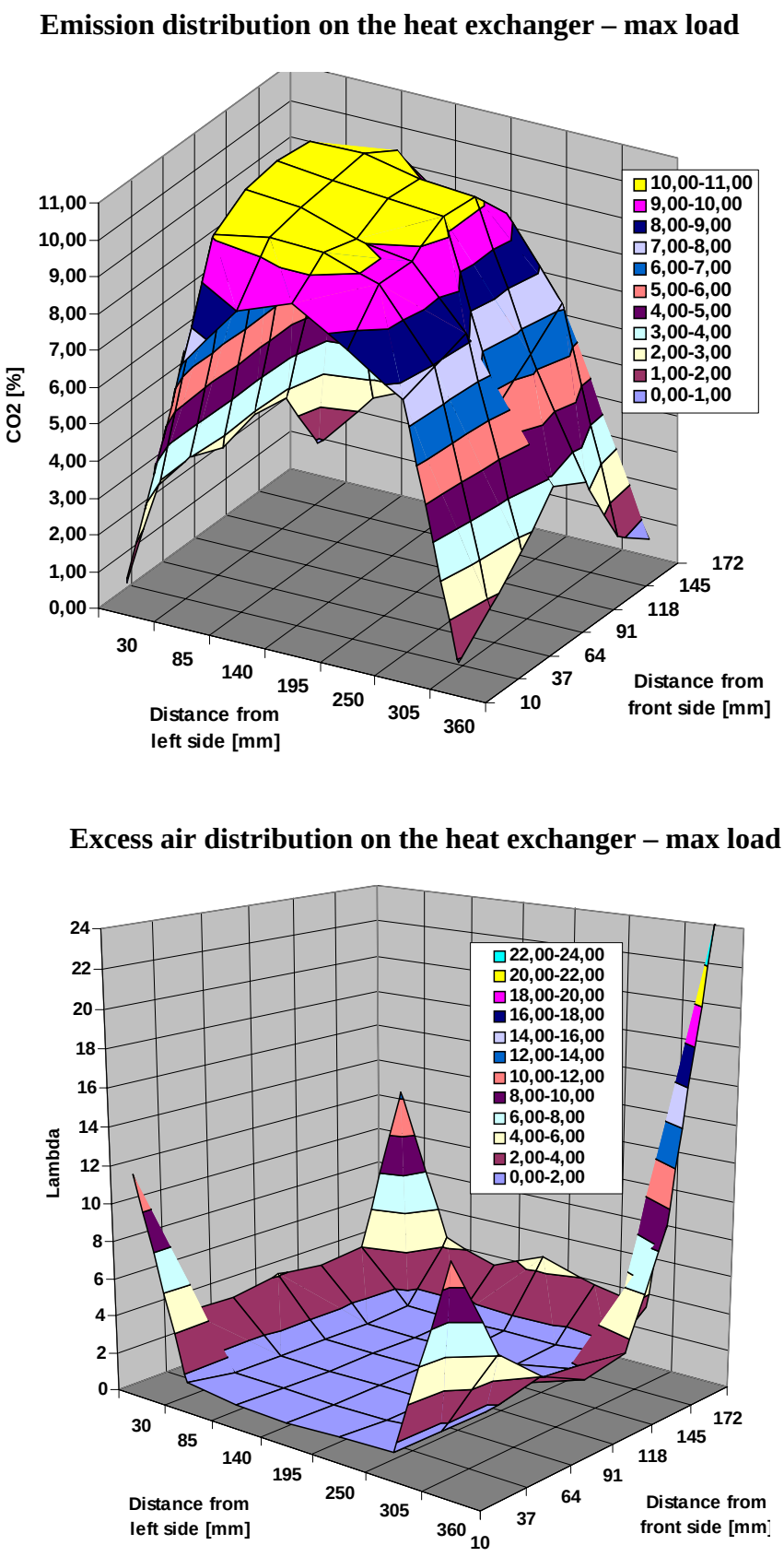


Figure 4.12 Emission and excess air distribution on the heat exchanger - max load

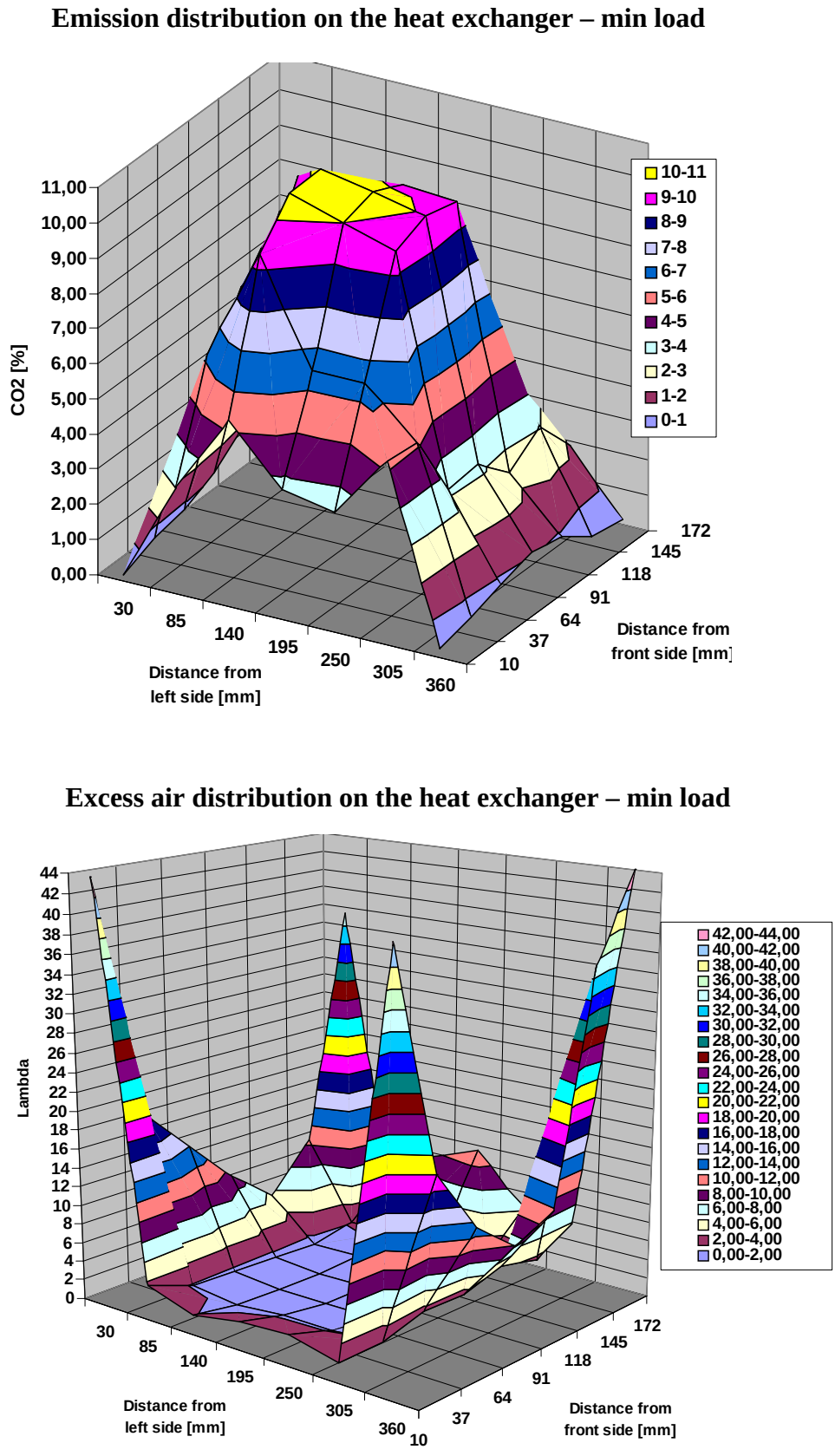


Figure 4.13 Emission and excess air distribution on the heat exchanger - min load

CHAPTER FIVE

CONCLUSIONS

For utilizations like central heating boilers, the non-condensing technology has reached maturity and it's close to the maximum theoretical nominal efficiency that can be obtained on boilers. But still there are possibilities to realize savings by optimizing some improvement techniques.

The evaluation of this study;

- As presented in chapter four; the convective and radiative heat loss from the cover of the appliance is approximately %1 of nominal output. By optimizing the insulation of the combustion chamber and also using of other improvement techniques; this loss might be saved a little more.

- Furthermore; efforts need to be made, not only in the direction of technology improvement, but also the development of design tools. As presented in previous chapter, modulated fan usage makes the non-condensing boiler more efficient. And also there are large savings potentials in the replacement of older technologies with the newest available. The replacement of low-efficient boilers would also offer the opportunity of having safer appliances installed.

- The exit flue gas temperature of a conventional boiler is usually higher than 150 ºC. At such temperatures, the water vapor entrained in the flue gases does not condense, and the latent heat cannot be reclaimed, which leads to a considerable heat loss. Therefore condensing boilers have been developed and have found wide applications in the world. It is the most efficient method of converting natural gas or fuel oil into useful energy by combustion. But it is not seem so feasible issue for warm temperate zone like Turkey.

- Another effective solution for decreasing the efficiency loss is utilization of the low temperature technology like wall heating or floor heating systems. As with low temperature technology, it follows the principle of operating the boiler only with that temperature, which is required to cover the current heating demand. The energy gain between a condensing boiler and a low temperature boiler is not just the result of the condensing energy gain, but is, to a large extent, a consequence of the low flue gas loss resulting from the low flue gas temperatures.

- In general, emissions are not an issue for the consumers, but emissions are important for the image of the gas industry, especially considering the competition with other energy sources. The consumer will see the benefit of high-efficiency boilers, but the advantages of low emission appliances are more for the overall society than for the individual consumer.

- The gas technologies shall also open to combination with renewable energy sources. Not only has the consumer more and more environmental concern, also the future energy policies of the countries may make the renewable technology economically feasible.

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Table A-1 Analyses of natural gas distributed in Manisa

Date	Time	ICV	Rel. Density	C3H8 (mol%)	i-C4H10 (mol%)	n-C4H10 (mol%)	neo-C5H12 (mol%)	i-C5H12 (mol%)	n-C5H12 (mol%)	N2 (mol%)	CH4 (mol%)	CO2 (mol%)	C2H6 (mol%)	Gas Density (kg/m3)	Wobbe Index (MJ/m3)	Compressibility Factor	Oven Temperature (C)	Carrier Gas Pressure (kPa)
20.Mar.06	10:53:12	35,19644	0,58491	0,863982	0,193212	0,174136	0	0,009862	0,006348	0,728995	95,41054	0,052426	2,560505	0,716756	51,04134	0,997785	58,04429	168,7299
20.Mar.06	11:03:12	35,1908	0,58475	0,865802	0,194436	0,174704	0	0,008196	0	0,729676	95,42122	0,04828	2,557688	0,716556	51,04063	0,997787	57,98864	168,1749
20.Mar.06	11:13:13	35,20193	0,58482	0,869041	0,192748	0,17185	0	0,011008	0,006254	0,732114	95,41748	0,03835	2,561156	0,716645	51,05319	0,997785	58,00352	165,4344
20.Mar.06	11:23:12	35,18864	0,5847	0,86699	0,192208	0,171528	0	0,009188	0	0,736858	95,41773	0,043298	2,562193	0,716501	51,03953	0,997787	58,02825	171,9213
20.Mar.06	11:33:12	35,19967	0,585	0,866266	0,194608	0,174495	0	0,008266	0,007575	0,744389	95,38593	0,044514	2,573963	0,716871	51,04168	0,997785	58,00161	169,3543
20.Mar.06	11:43:12	35,20152	0,58502	0,870555	0,195083	0,17397	0	0,010561	0,008934	0,742253	95,39899	0,04437	2,555274	0,716884	51,04384	0,997785	58,01936	166,6138
20.Mar.06	11:53:12	35,19885	0,58505	0,868884	0,194333	0,175943	0	0,007408	0,005318	0,752222	95,37039	0,042954	2,582541	0,716921	51,03863	0,997785	58,0339	165,9894
20.Mar.06	12:03:12	35,18988	0,58509	0,867188	0,195172	0,173628	0	0,008022	0,006342	0,763628	95,36519	0,04872	2,572106	0,716977	51,02376	0,997786	57,99647	172,2682
20.Mar.06	12:13:12	35,18842	0,58511	0,865313	0,194098	0,17555	0	0,009873	0	0,759567	95,35435	0,054055	2,587191	0,716995	51,02104	0,997786	58,02825	169,8747
20.Mar.06	12:23:12	35,1887	0,58539	0,864255	0,193929	0,178513	0	0,011145	0,006421	0,787122	95,31942	0,053263	2,585925	0,717339	51,00875	0,997785	58,02825	167,2036
20.Mar.06	12:33:12	35,19118	0,58534	0,864615	0,193154	0,17784	0	0,010824	0,005103	0,776205	95,32457	0,054627	2,593055	0,717282	51,01443	0,997784	57,98061	166,7873
20.Mar.06	12:43:12	35,18332	0,58532	0,858544	0,192824	0,180283	0	0,008608	0,007759	0,816139	95,30469	0,036103	2,595047	0,717252	51,0041	0,997786	57,98558	163,9427
20.Mar.06	12:53:12	35,18502	0,58537	0,857418	0,193713	0,181434	0	0,009182	0,006878	0,812422	95,29949	0,040195	2,599267	0,717322	51,00396	0,997786	58,02825	170,2909
20.Mar.06	13:03:12	35,16996	0,58545	0,854009	0,193586	0,182109	0	0,008022	0,005799	0,845438	95,27043	0,04081	2,599798	0,717416	50,97898	0,997787	57,98598	167,6198
20.Mar.06	13:13:12	35,1705	0,58549	0,856835	0,193877	0,180583	0	0,007453	0,006992	0,855892	95,25806	0,035826	2,60447	0,717464	50,97794	0,997787	57,9646	164,8447
20.Mar.06	13:23:12	35,14623	0,58545	0,841728	0,189552	0,182199	0	0,00801	0,005529	0,887083	95,24268	0,041363	2,601866	0,717421	50,94482	0,99779	57,98848	163,9081
20.Mar.06	13:33:13	35,15201	0,58546	0,847064	0,189635	0,179727	0	0,009174	0,006452	0,884374	95,24455	0,03665	2,602377	0,717425	50,95289	0,997789	58,02028	170,4991
20.Mar.06	13:43:12	35,08262	0,5856	0,836888	0,188019	0,182607	0	0,009023	0	0,968973	95,20832	0,07034	2,535824	0,717598	50,84753	0,997795	57,98848	167,6892

Table A-2 Lists the various kinds of losses for nine different boiler types (SAVE II Action)

	Major specifications	Efficiency (%)	Jacket loss (W/K)	Draught loss (W/K)	Flue gas loss (W/K)	Total losses (W/K)
1.	non condensing forced draught burner no fan open air supply on/off burner	91	1,34	1,1	8,09	10,5
2.	non condensing atmospheric burner no fan open air supply on/off burner	81	2,99	0,36	7,02	10,4
3.	condensing atmospheric burner fan assisted open air supply on/off burner	102	2,59	1,9	5,85	10,3
4.	non condensing atmospheric burner fan assisted open air supply on/off burner	88	2,04	0,53	3,96	6,53
5.	non condensing atmospheric burner no fan open air supply on/off burner	84	1,72	0,67	4,43	6,82
6.	non condensing atmospheric burner fan assisted closed air supply on/off burner	91	1,27	1,08	3,31	5,66
7.	condensing atmospheric burner fan assisted closed air supply on/off burner control	97	1,94	1,36	7,57	10,87
8.	condensing atmospheric burner fan assisted closed air supply modulating burner	97	1,2	1,11	3,63	5,94
9.	condensing burner? fan assisted air supply? on/off burner	103	1,65	1,17	4,24	7,06

Table A-3 Overview design options effect on the efficiency (SAVE II Action)

Table 9.3: Overview design options - dwelling based heating systems

				Efficiency on lhv	Indication on savings
Dwelling Based Heating System (Central Hydronic Heating System)					
	Design options related to generator				
		Improve steady state efficiency			
			- Conventional boiler (gas/oil)	85%	*Reference*
			- Improved efficiency boilers (gas/oil)	90%	6%
			- Condensing boilers (gas/oil)	107%	26%
			- Gas fired heatpump with additional condensing boiler	140%	65%
			- Electric heat pump (efficiency based on primary energy)	140%	65%
		Reduce standing losses			
			- Replacement pilot flame by electronic ignition		< 550 kWh_{pe}/yr
			- Storage heaters (coal & wood) : Improve insulation		< 350 kWh_{pe}/yr
			- : Install flue damper		< 350 kWh_{pe}/yr
		Reduce start/stop losses			
			- Improve appliance insulation		< 200 kWh_{pe}/yr
			- Install flue damper		< 300 kWh_{pe}/yr
			- Reduce heat capacity of generator (important for on/of burners)		< 400 kWh_{pe}/yr
		Reduce auxiliary power consumption			
			- Use pump with permanent magnet motor, controlled by the boiler (also relevant for sub-systems such as floor heating)		< 400 kWh_{pe}/yr
			- Reduce stand-by power consumption		< 40 kWh_{pe}/yr
		Improve generator control (boiler water temperature)			
			- Boiler water temp. controlled by max. thermostat in boiler (90°C), combined with conventional on/of room thermostat		*Reference*
			- Modulating burner control		<5%
			- Boiler water temp. weather controlled (or controlled by modulating room thermostat)		<3%
	Design options related to distribution system				
			- No insulation of pipes / 90-70 temp. regime		*Reference*
			- Insulation distribution pipes		...
			- Hydraulically evened out system ("Ausgleichen")		...
	Design options related to emitter system				
			- High temperature / small surface emitter (>90°C)		*Reference*
			- Average temperature / small surface emitter (90-70°C)		<5%
			- Low temperature / large surface radiators (30 – 50°C)		<10%
	Design options related to control system (room temperature)				
			- Clock room thermostat (for modulating room thermostat - see above)		<5%
			- Thermostatic radiator valves		<3%
			- Behavioural Feed Back Control Unit combined with Room Thermostat		<10%

Table A-4 Thermodynamic property of saturated water (H_2O), SI units

Saturated Water (H2O)--Temperature Table									
		Spec. Volume		Internal Energy		Enthalpy		Entropy	
deg-C	kPa	m^3/kg		kJ/kg		kJ/kg		kJ/kg*K	
Temp.	Sat. press.	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor
T °C	P	v_f	v_g	u_f	u_g	h_f	h_g	s_f	s_g
0.01	0.6113	0.001000	206.14	0.00	2375.3	0.01	2501.4	0.0000	9.1562
5	0.8721	0.001000	147.12	20.97	2382.3	20.98	2510.6	0.0761	9.0257
10	1.2276	0.001000	106.38	42.00	2389.2	42.01	2519.8	0.1510	8.9008
15	1.7051	0.001001	77.93	62.99	2396.1	62.99	2528.9	0.2245	8.7814
20	2.339	0.001002	57.79	83.95	2402.9	83.96	2538.1	0.2966	8.6672
25	3.169	0.001003	43.36	104.88	2409.8	104.89	2547.2	0.3674	8.5580
30	4.246	0.001004	32.89	125.78	2416.6	125.79	2556.3	0.4369	8.4533
35	5.628	0.001006	25.22	146.67	2423.4	146.68	2565.3	0.5053	8.3531
40	7.384	0.001008	19.52	167.56	2430.1	167.57	2574.3	0.5725	8.2570
45	9.593	0.001010	15.26	188.44	2436.8	188.45	2583.2	0.6387	8.1648
50	12.349	0.001012	12.03	209.32	2443.5	209.33	2592.1	0.7038	8.0763
55	15.758	0.001015	9.568	230.21	2450.1	230.23	2600.9	0.7679	7.9913
60	19.940	0.001017	7.671	251.11	2456.6	251.13	2609.6	0.8312	7.9096
65	25.03	0.001020	6.197	272.02	2463.1	272.06	2618.3	0.8935	7.8310
70	31.19	0.001023	5.042	292.95	2469.6	292.98	2626.8	0.9549	7.7553
75	38.58	0.001026	4.131	313.90	2475.9	313.93	2643.7	1.0155	7.6824
80	47.39	0.001029	3.407	334.86	2482.2	334.91	2635.3	1.0753	7.6122
85	57.83	0.001033	2.828	355.84	2488.4	355.90	2651.9	1.1343	7.5445
90	70.14	0.001036	2.361	376.85	2494.5	376.92	2660.1	1.1925	7.4791
95	84.55	0.001040	1.982	397.88	2500.6	397.96	2668.1	1.2500	7.4159
T °C	P	v_f	v_g	u_f	u_g	h_f	h_g	s_f	s_g
100	0.10135	0.001044	1.6729	418.94	2506.5	419.04	2676.1	1.3069	7.3549
105	0.12082	0.001048	1.4194	440.02	2512.4	440.15	2683.8	1.3630	7.2958
110	0.14327	0.001052	1.2102	461.14	2518.1	461.30	2691.5	1.4185	7.2387
115	0.16906	0.001056	1.0366	482.30	2523.7	482.48	2699.0	1.4734	7.1833
120	0.19853	0.001060	0.8919	503.50	2529.3	503.71	2706.3	1.5276	7.1296
125	0.2321	0.001065	0.7706	524.74	2534.6	524.99	2713.5	1.5813	7.0775
130	0.2701	0.001070	0.6685	546.02	2539.9	546.31	2720.5	1.6344	7.0269
135	0.3130	0.001075	0.5822	567.35	2545.0	567.69	2727.3	1.6870	6.9777
140	0.3613	0.001080	0.5089	588.74	2550.0	589.13	2733.9	1.7391	6.9299
145	0.4154	0.001085	0.4463	610.18	2554.9	610.63	2740.3	1.7907	6.8833
150	0.4758	0.001091	0.3928	631.68	2559.5	632.20	2746.5	1.8418	6.8379
155	0.5431	0.001096	0.3468	653.24	2564.1	653.84	2752.4	1.8925	6.7935
160	0.6178	0.001102	0.3071	674.87	2568.4	675.55	2758.1	1.9427	6.7502
165	0.7005	0.001108	0.2727	696.56	2572.5	697.34	2763.5	1.9925	6.7078
170	0.7917	0.001114	0.2428	718.33	2576.5	719.21	2768.7	2.0419	6.6663
175	0.8920	0.001121	0.2168	740.17	2580.2	741.17	2773.6	2.0909	6.6256
180	1.0021	0.001127	0.19405	762.09	2583.7	763.22	2778.2	2.1396	6.5857
185	1.1227	0.001134	0.17409	784.10	2587.0	785.37	2782.4	2.1879	6.5465
190	1.2544	0.001141	0.15654	806.19	2590.0	807.62	2786.4	2.2359	6.5079
195	1.3978	0.001149	0.14105	828.37	2592.8	829.98	2790.0	2.2835	6.4698
200	1.5538	0.001157	0.12736	850.65	2595.3	852.45	2793.2	2.3309	6.4323
205	1.7230	0.001164	0.11521	873.04	2597.5	875.04	2796.0	2.3780	6.3952

210	1.9062	0.001173	0.10441	895.53	2599.5	897.76	2798.5	2.4248	6.3585
215	2.104	0.001181	0.09479	918.14	2601.1	920.62	2800.5	2.4714	6.3221
220	2.318	0.001190	0.08619	940.87	2602.4	943.62	2802.1	2.5178	6.2861
225	2.548	0.001199	0.07849	963.73	2603.3	966.78	2803.3	2.5639	6.2503
230	2.795	0.001209	0.07158	986.74	2603.9	990.12	2804.0	2.6099	6.2146
235	3.060	0.001219	0.06537	1009.89	2604.1	1013.62	2804.2	2.6558	6.1791
240	3.344	0.001229	0.05976	1033.21	2604.0	1037.32	2803.8	2.7015	6.1437
245	3.648	0.001240	0.05471	1056.71	2603.4	1061.23	2803.0	2.7472	6.1083
250	3.973	0.001251	0.05013	1080.39	2602.4	1085.36	2801.5	2.7927	6.0730
255	4.319	0.001263	0.04598	1104.28	2600.9	1109.73	2799.5	2.8383	6.0375
260	4.688	0.001276	0.04221	1128.39	2599.0	1134.37	2796.9	2.8838	6.0019
265	5.081	0.001289	0.03877	1152.74	2596.6	1159.28	2793.6	2.9294	5.9662
270	5.499	0.001302	0.03564	1177.36	2593.7	1184.51	2789.7	2.9751	5.9301
275	5.942	0.001317	0.03279	1202.25	2590.2	1210.07	2785.0	3.0208	5.8938
280	6.412	0.001332	0.03017	1227.46	2586.1	1235.99	2779.6	3.0668	5.8571
285	6.909	0.001348	0.02777	1253.00	2581.4	1262.31	2773.3	3.1130	5.8199
290	7.436	0.001366	0.02557	1278.92	2576.0	1289.07	2766.2	3.1594	5.7821
295	7.993	0.001384	0.02354	1305.20	2569.9	1316.30	2758.1	3.2062	5.7437
300	8.581	0.001404	0.02167	1332.00	2563.0	1344.00	2749.0	3.2534	5.7045
305	9.202	0.001425	0.019948	1359.30	2555.2	1372.40	2738.7	3.3010	5.6643
310	9.856	0.001447	0.018350	1387.10	2546.4	1401.30	2727.3	3.3493	5.6230
315	10.547	0.001472	0.016867	1415.50	2536.6	1431.00	2714.5	3.3982	5.5804
320	11.274	0.001499	0.015488	1444.60	2525.5	1461.50	2700.1	3.4480	5.5362
330	12.845	0.001561	0.012996	1505.30	2498.9	1525.30	2665.9	3.5507	5.4417
340	14.586	0.001638	0.010797	1570.30	2464.6	1594.20	2622.0	3.6594	5.3357
350	16.513	0.001740	0.008813	1641.90	2418.4	1670.60	2563.9	3.7777	5.2112
360	18.651	0.001893	0.006945	1725.20	2351.5	1760.50	2481.0	3.9147	5.0526
370	21.03	0.002213	0.004925	1844.00	2228.5	1890.50	2332.1	4.1106	4.7971
374.14	22.09	0.003155	0.003155	2029.60	2029.6	2099.30	2099.3	4.4298	4.4298

Saturated Water (H_2O)--Temperature **Table**. Spec. Volume, Internal Energy, Enthalpy, Entropy. Retrieved June 2006, from kahuna.sdsu.edu/testcenter/testhome/Test/solve/basics/tables/tablesPC/TSatH2O.html

Table A-5 Formation enthalpy, Gibbs function and enthalpy for 25ºC and 1atm

Madde	Kimyasal formülü	$\bar{h}_f^\circ$ kJ/kmol	$\bar{g}_f^\circ$ kJ/kmol	$\bar{s}^\circ$ kJ/(kmol · K)
Karbon	$C(k)$	0	0	5.74
Hidrojen	$H_2(g)$	0	0	130.68
Azot	$N_2(g)$	0	0	191.61
Oksijen	$O_2(g)$	0	0	205.04
Karbon monoksit	$CO(g)$	−110,530	−137,150	197.65
Karbon dioksit	$CO_2(g)$	−393,520	−394,360	213.80
Su buharı	$H_2O(g)$	−241,820	−228,590	188.83
Su	$H_2O(s)$	−285,830	−237,180	69.92
Hidrojen peroksit	$H_2O_2(g)$	−136,310	−105,600	232.63
Amonyak	$NH_3(g)$	−46,190	−16,590	192.33
Metan	$CH_4(g)$	−74,850	−50,790	186.16
Asetilen	$C_2H_2(g)$	+226,730	+209,170	200.85
Etilen	$C_2H_4(g)$	+52,280	+68,120	219.83
Etan	$C_2H_6(g)$	−84,680	−32,890	229.49
Propilen	$C_3H_6(g)$	+20,410	+62,720	266.94
Propan	$C_3H_8(g)$	−103,850	−23,490	269.91
n-Bütan	$C_4H_{10}(g)$	−126,150	−15,710	310.12
n-Oktan	$C_8H_{18}(g)$	−208,450	+16,530	466.73
n-Oktan	$C_8H_{18}(s)$	−249,950	+6,610	360.79
n-Dodekan	$C_{12}H_{26}(g)$	−291,010	+50,150	622.83
Benzen	$C_6H_6(g)$	+82,930	+129,660	269.20
Metil alkol	$CH_3OH(g)$	−200,670	−162,000	239.70
Metil alkol	$CH_3OH(s)$	−238,660	−166,360	126.80
Etil alkol	$C_2H_5OH(g)$	−235,310	−168,570	282.59
Etil alkol	$C_2H_5OH(s)$	−277,690	−174,890	160.70
Oksijen	$O(g)$	+249,190	+231,770	161.06
Hidrojen	$H(g)$	+218,000	+203,290	114.72
Azot	$N(g)$	+472,650	+455,510	153.30
Hidroksil	$OH(g)$	+39,460	+34,280	183.70

Source : JANAF, Thermochemical Tables, Dow Chemical Co.; Selected Values of Chemical Thermodynamic Properties, NBS Technical Noote 270-3.

Table A-6 Thermophysical properties of air at atmospheric pressure

T (K)	ρ (kg/m^3)	c_p (J/kg·K)	μ (kg/m·s)	ν (m^2/s)	k (W/m·K)	α (m^2/s)	Pr
			Air				
100	3.605	1039	0.711×10^{-5}	0.197×10^{-5}	0.00941	0.251×10^{-5}	0.784
150	2.368	1012	1.035	0.437	0.01406	0.587	0.745
200	1.769	1007	1.333	0.754	0.01836	1.031	0.731
250	1.412	1006	1.606	1.137	0.02241	1.578	0.721
260	1.358	1006	1.649	1.214	0.02329	1.705	0.712
270	1.308	1006	1.699	1.299	0.02400	1.824	0.712
280	1.261	1006	1.747	1.385	0.02473	1.879	0.711
290	1.217	1006	1.795	1.475	0.02544	2.078	0.710
300	1.177	1007	1.857	1.578	0.02623	2.213	0.713
310	1.139	1007	1.889	1.659	0.02684	2.340	0.709
320	1.103	1008	1.935	1.754	0.02753	2.476	0.708
330	1.070	1008	1.981	1.851	0.02821	2.616	0.708
340	1.038	1009	2.025	1.951	0.02888	2.821	0.707
350	1.008	1009	2.090	2.073	0.02984	2.931	0.707
400	0.8821	1014	2.310	2.619	0.03328	3.721	0.704
450	0.7840	1021	2.517	3.210	0.03656	4.567	0.703
500	0.7056	1030	2.713	3.845	0.03971	5.464	0.704
550	0.6414	1040	2.902	4.524	0.04277	6.412	0.706
600	0.5880	1051	3.082	5.242	0.04573	7.400	0.708
650	0.5427	1063	3.257	6.001	0.04863	8.430	0.712
700	0.5040	1075	3.425	6.796	0.05146	9.498	0.715
750	0.4704	1087	3.588	7.623	0.05425	10.61	0.719
800	0.4410	1099	3.747	8.497	0.05699	11.76	0.723
850	0.4150	1110	3.901	9.400	0.05969	12.96	0.725
900	0.3920	1121	4.052	10.34	0.06237	14.19	0.728
950	0.3716	1131	4.199	11.30	0.06501	15.47	0.731
1000	0.3528	1142	4.343	12.31	0.06763	16.79	0.733
1100	0.3207	1159	4.622	14.41	0.07281	19.59	0.736
1200	0.2940	1175	4.891	16.64	0.07792	22.56	0.738
1300	0.2714	1189	5.151	18.98	0.08297	25.71	0.738
1400	0.2520	1201	5.403	21.44	0.08798	29.05	0.738
1500	0.2352	1211	5.648	23.99	0.09296	32.64	0.735

Source: Lienhard, J.H. (2001). Heat transfer textbook (3rd ed.). Massachusetts

Table A-7 Total emittances for a variety of surfaces

Surface	*Temp.* (°C)	ε	*Surface*	*Temp.* (°C)	ε
Aluminum			Asbestos	40	0.93–0.97
Polished, 98% pure	200–600	0.04–0.06	Brick		
Commercial sheet	90	0.09	Red, rough	40	0.93
Heavily oxidized	90–540	0.20–0.33	Silica	980	0.80–0.85
Brass			Fireclay	980	0.75
Highly polished	260	0.03	Ordinary refractory	1090	0.59
Dull plate	40–260	0.22	Magnesite refractory	980	0.38
Oxidized	40–260	0.46–0.56	White refractory	1090	0.29
Copper			Carbon		
Highly polished electrolytic	90	0.02	Filament	1040–1430	0.53
Slightly polished to dull	40	0.12–0.15	Lampsoot	40	0.95
Black oxidized	40	0.76	Concrete, rough	40	0.94
Gold: pure, polished	90–600	0.02–0.035	Glass		
Iron and steel			Smooth	40	0.94
Mild steel, polished	150–480	0.14–0.32	Quartz glass (2 mm)	260–540	0.96–0.66
Steel, polished	40–260	0.07–0.10	Pyrex	260–540	0.94–0.74
Sheet steel, rolled	40	0.66	Gypsum	40	0.80–0.90
Sheet steel, strong rough oxide	40	0.80	Ice	0	0.97–0.98
Cast iron, oxidized	40–260	0.57–0.66	Limestone	400–260	0.95–0.83
Iron, rusted	40	0.61–0.85	Marble	40	0.93–0.95
Wrought iron, smooth	40	0.35	Mica	40	0.75
Wrought iron, dull oxidized	20–360	0.94	Paints		
Stainless, polished	40	0.07–0.17	Black gloss	40	0.90
Stainless, after repeated heating	230–900	0.50–0.70	White paint	40	0.89–0.97
Lead			Lacquer	40	0.80–0.95
Polished	40–260	0.05–0.08	Various oil paints	40	0.92–0.96
Oxidized	40–200	0.63	Red lead	90	0.93
Mercury: pure, clean	40–90	0.10–0.12	Paper		
Platinum			White	40	0.95–0.98
Pure, polished plate	200–590	0.05–0.10	Other colors	40	0.92–0.94
Oxidized at 590°C	260–590	0.07–0.11	Roofing	40	0.91
Drawn wire and strips	40–1370	0.04–0.19	Plaster, rough lime	40–260	0.92
Silver	200	0.01–0.04	Quartz	100–1000	0.89–0.58
Tin	40–90	0.05	Rubber	40	0.86–0.94
Tungsten			Snow	10–20	0.82
Filament	540–1090	0.11–0.16	Water, thickness ≥0.1 mm	40	0.96
Filament	2760	0.39	Wood	40	0.80–0.90
			Oak, planed	20	0.90