Preparatory Study on

Eco-design of CH Boilers

Task 4 Report (FINAL)

Technical Analysis (incl. System Model)

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CONTENTS

page

LIST OF TABLES

page

1 INTRODUCTION

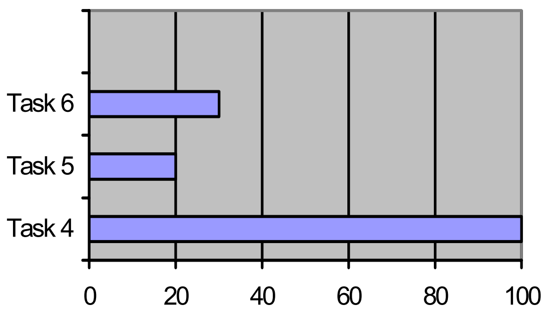
1.1 Scope

This is the *Draft Final* report on Task 4 of the preparatory study on the Eco-design of Central Heating Boilers for the European Commission, in the context of the Ecodesign of Energy-using Products directive 2005/32/EC.

The scope of Task 4 is Product System Analysis. It looks into the interaction of the EuP with the installation/system in which it operates during the use phase. Probably more than with any other Energy Using Product, the System Analysis is a vital subject in the assessment of real life energy efficiency and the improvement potential of boilers. This task therefore includes a functional analysis of the system to which the product belongs, including a rough estimate of the overall impacts and an assessment of how the integration of the product into the system and its design can improve its overall environmental performance.

For practical reasons this Task 4 report covers also a full technical analysis of boiler components and heating system components, describing technical product features and best available technologies. As such the report covers not only Task 4 but also parts of Task 5 and Task 6.

Figure 1-1.
Coverage Task 4 report



It must be clear that it is outside the scope of the EuP directive to look at measures other than those related to the design and installation (see Annex VII.4 of directive 2005/32/EC) of the EuP. Obviously, a boiler designer/ manufacturer cannot influence

- Climate characteristics (heating season, degree-days, etc.)
- Building shell characteristics (insulation heat losses, ventilation losses, orientation)
- Internal load (humans, appliances, solar, heat capacity of inventory)
- Heating system characteristics (radiator, convector, low temperature wall/floor heating, tubing, valves; etc.)

However, there are certain characteristics of the boiler design that determine how well the boiler interacts with (variations in) the above parameters. These characteristics, in earlier studies referred to as acting on the 'control efficiency', determine a significant part of the energy consumption and the emissions of pollutants of the product.

As mentioned in the Task 1 report, these characteristics are so far not captured in the EN product test standards (e.g. EN 303). Partly they are addressed in the new draft building performance standards (e.g. prEN 15316), but they are then not translated back at product level where they could be incorporated in CE-marking. For instance, the extend to which a boiler is equipped to follow the weather conditions, adjusting the boiler water temperature to the need, is not (fully) incorporated in the product test standard. Yet it is clearly a product feature. The fact whether a boiler is equipped with a flue valve, is clearly another boiler feature reducing the flue gas losses when the boiler is 'off', which again is not necessarily picked up in the product test standards. The extend to which a 'condensing' boiler can control the system parameters so it can actually achieve these condensing operation conditions in practice is not incorporated. The control of the hydraulic parameters (pump, mixing valves) to achieve optimal heat exchanger efficiency is yet another item.

Task 4 sets out to identify and evaluate these boiler features (and more) on a systematic basis. Partly it builds on the data assessed in the SAVE Heating study 2002, mentioned earlier. In this study VHK already indicated and quantified a methodology for the heating system as a whole. Also the annex I in the ECCP-report is a valuable input in this respect. Finally, for the actual modelling the methodology that is developed in the EN standards for the Energy Performance of Buildings will play an important role.

Extending on these sources a mathematical model is developed that on one hand is linked to the stock model from subtask 2.2 and on the other hand should incorporate the technical design options in the following tasks. The focus of the model is on the boiler characteristics and how they interact with the heating systems throughout the EU. The other characteristics (climate, housing stock and building shell, internal load, heating systems) are largely a given for the current (most recent) situation, but –e.g. for the sake of the impact analysis in Subtask 5.2— should be expandable to make future projections (2010, 2020) and design-solutions.

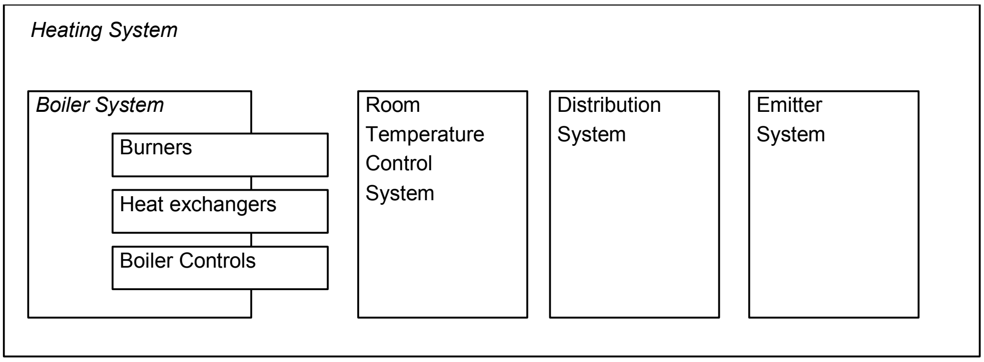
It is not the intention of VHK to duplicate research work and modelling that is an ongoing activity in many research institutes. Rather it aims to harmonise what has been developed in the energy performance of building standards (prEN 15316) and the product test standards (e.g. EN 303), taking it one step further where necessary. Once this has been achieved, the technical modelling can be linked to the market and stock model from Task 2 and will allow modelling of design options in Task 6.

1.2 Approach

For practical reasons, this Task 4 Report not only covers the Product System Analysis, but also a full Technical Analysis of the boiler and its components and of the individual heating system components. Obviously, analysing boilers, boiler components heating system components and their mutual relations, it is more practical to directly include the analysis of their technical product features and their best available technologies. As such this report also provides vital information on the Best Available Technology for boiler and system components and on the product specific data needed for the Definition of the Base-Case. Subsequently this Task 4 Report not only reports on the activities mentioned in Task 4 of the contract, but it also contains detailed information for the requested deliverables of Task 5.1 and Task 6.1 & 6.2. As a part of the activities following under these three subtasks are already performed in this Task 4, the remaining activities for these subtasks will mainly focus on a logical chategorization and representation of data. In addition task 6.1 and 6.2 will give a further analysis on the BAT-options that are not described in this task report, being heatpumps and solar systems and on costs.

The boilersystem and heating system components that are analysed and described in this Task 4 are respresented with the following figure.

Figure 1-2.
Graphic representation of the boiler and its components and the overall heating system with its components.



The data and knowledge coming from the analysis of these indivual components are used for a System Analysis of the overall Heating System Efficiency on the basis of which a mathematical model is developed.

1.3 Report structure

This preliminary Task 4 draft report contains 11 chapters. After this introductory chapter, the next chapters process the following topics:

- Chapter 2 gives an introduction of the mass- and energy balance of the boiler and describes in a general way the interactions of the boiler with the other boiler components. This Chapter is a basis for a more extended discussion of the system analysis in chapter 10.
- Chapter 3 discusses the emissions from oil- and gas-fired combustion processes, their nature, impact and current design measures. This Chapter is a basis for a more extended discussion of the burner technology in Chapter 4.
- Chapters 4 (burners) and 5 (heat exchangers) describe the state of the art in vital components and contain information that can be used when the boiler design options are discussed (task 6).

Apart from the energy losses of the boiler and its components, the heating system as a whole will have additional losses through suboptimal boiler controls, losses caused by the room temperature controls system, losses in the heat distribution system and losses caused by emitters. The chapters dealing with these heating system components clearly show, that the boiler influences the heat losses caused by these components. In other words, the boiler and its controls can help minimising the energy losses of the heating system. The nature of these system losses will therefore be further explained in the following chapters.

- Chapter 6 deals with boiler controls and explains the basic principles and the developments related to power input control, fuel air ratio control, pump control and feed temperature control. It is explained that proper feed temperature control can only be achieved in combination with adequate room temperature control systems.
- In the next chapter 7 a technical analysis is given on the room temperature control systems that are predominantly used. Estimates are given on the energy losses related to the various room temperature control systems, as well as options for further improvement.
- Chapter 8 describes the nature and magnitude of the heat distribution losses, and explains how boiler controls could contribute in minimising these losses.
- Chapter (9) describes what the influences are of the emitter system on energy losses, on feed temperature control and on generator efficiency.
- In Chapter 10 a compact technical analysis is given on heat pump boiler generator systems, based on public available information that could be gathered and studied within the constaints of this Ecodesign study.

The final chapter 11 contains the System Analysis of the overall Heating System Efficiency on the basis of which the ECOBOILER mathematical model is developed. This work is mainly documented as an Excel file and in the Annexes:

ANNEX A: Documentation of the Excel file for the ECOBOILER model

ANNEX B: Intermediate model review by boiler experts and iTG

ANNEX C: Notes for the separate boiler model

ANNEX D: Reference material

ANNEX E: Emissions according toGEMIS 4.2

ANNEX F: List of references

2 BASIC ENERGY AND MASS BALANCE

2.1 Introduction

In most energy policy studies on heating appliances, the combustion process and the detailed energy- and mass balance of central heating boilers are not explained. The scientific background is not very easy and in general it is not needed for readers in the policy field to go beyond the level of the CH boiler being a 'black-box' with a certain efficiency level according to a product test standard.

Yet, this approach has also led to a number of notions, myths and half-truths regarding boiler efficiency and emissions in practice which can only be understood (and partially denied) when looking inside the black-box. For this reason we have made an attempt, as part of the system analysis, to provide some guidance for policy makers regarding the basics of the energy and mass balance with a boiler. We have taken methane, the main component of natural gas but also a fuel with a relatively simple structure, as an illustration of a fuel, although references to other fuels also occur. The mass- and energy values should be seen as illustrations, although also here we add results from research that is based on tests with actual boilers.

Starting of with the global chemical reaction which mainly produces carbon dioxide and water vapour (paragraph 2.2), this chapter looks at the

Mass balance (paragraph 2.3), including

- Stoichiometric volume balance (the 'ideal' theoretical volume balance)
- Air factor/ lambda (excess air)
- Humidity of combustion air
- Influence of CO, NO_x, C_xH_y, SO2 and dust (PM) emissions (fraction of incomplete combustion)
- Conversion of volume to mass balance

Subsequently, we are discussing the energy balance of the boiler, looking at the energy parameters of the combustion process, such as the flame temperature, combustion heat, latent heat of condensation, heat loss through excess combustion air and finally the energy loss concerned with incomplete combustion (paragraph 2.4). The approach is basic (secondary school) and pragmatic (focussed on boilers), largely by-passing the many tools that exist at academic research level to numerically model and predict the combustion process.

Paragraphs 2.5 to 2.8 deal with the energy losses in the main boiler components: the burner (paragraph 2.5), the primary heat exchanger (paragraph 2.6), secondary and tertiary 'condensing' heat exchangers (paragraph 2.7) and finally the energy penalties involved in storage components (paragraph 2.8). The most extensive report is on the efficiency of the primary heat exchanger in paragraph 2.6, where we will be looking at flue gas losses, generator losses and start-stop ('cycling') losses both in on-mode and off-mode.

Paragraph 2.9 gives a brief estimate of losses in auxiliary components such as pump, fan and controls (to be expanded in other parts of the study). Finally, paragraph 2.10 gives a summary of the energy losses in the previous paragraphs for three characteristic boiler types ('Standard', 'Low Temperature' and 'Condensing' boiler) with two types of temperature controls ('Weather-controlled' and 'On-off room thermostat').

2.2 Global chemical reaction

In gas- and oil fired boilers the combustion is the ***stationary***, ***rapid, medium to high-temperature oxidisation***[1] of a hydrocarbon with the oxygen in air. With gas- and oil-fired boilers the combustion products of an ideal combustion process are always carbon dioxide (CO_2) and water vapour (H_2O) [2]. For instance, in the case of methane (CH4), which is the main component of natural gas in Europe, the global chemical reaction can be summarized as:

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

The equation for e.g. heating oil is different but follows the same principle, but with the hydrocarbon being more complex also the equations become more complex. Still, the outcome is again (mainly) CO_2 and H_2O.

2.3 Mass balance

2.3.1 Stoichiometric volume balance

Using *Avogadro's Law* [3] and assuming that air is made of ca. 1 part of oxygen (O_2) and 4 parts of nitrogen (N_2) we can derive the theoretical volume of air that is needed for the reaction and the volumes of carbon dioxide and water vapour produced.

$$\mathbf{1\ vol.}CH_4 + \mathbf{2\ vol.}O_2 + \mathbf{8\ vol.}N_2 \text{ ®} \rightarrow \mathbf{1\ vol.}CO_2 + \mathbf{2\ vol.}H_2O + \mathbf{8\ vol.}N_2$$

$$9{,}1\%\ CH4 + 90{,}9\%\ \text{air} \text{ ®} \rightarrow 9{,}1\%\ CO_2 + 18{,}2\%\ H_2O + 72{,}7\%\ N_2$$

The above is known as ***stoichiometric*** combustion, i.e. assuming a perfect mixing of fuel and air at perfectly controlled pressure and temperature.

2.3.2 Air factor/ lambda

In reality, the stoichiometric volume balance is theoretical. Manufacturers build in a safety factor, called ***air factor*** or ***lambda (λ),*** to make sure that there is always enough air/oxygen to guarantee a complete combustion. The air factor is actually intended to compensate for

- inhomogeneous mixing of air/fuel (oil-fired 'blue burner' 5%, good 'yellow burner' 10%, less good burners 15%). In general the particle size of the fuel (with atomising oil burners this is the size of the droplets) is a very important factor for the air factor [4].
- fluctuations in atmospheric pressure of the incoming air (around 6%)
- fluctuations in relative humidity of air (from 0,1 to 3,5%)
- fluctuations in fuel supply (between 5 and 10%, depending on maintenance, varying gas grid pressure, etc.)

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[1] ***'Stationary'*** as opposed to non-stationary combustion in motors.. ***'Rapid'*** as opposed to slow, low-temperature oxidisation processes in biochemistry (rotting, etc.) and medicine (glucose in muscle power, etc.). ***'High-temperature'*** is also referred to as 'flame-combustion' (>1500 K). ***'Medium-temperature'*** is referred to as 'flameless' combustion (700-1500 K). Medium-low temperature combustion (400-1000 K reaction temperature) is e.g. 'catalytic combustion'. The chemical oxidisation in a fuel cell is classified as 'catalytic combustion of hydrogen'.

[2] Note that the quantity of water vapour depends on the fuel with its specific combustion reaction. For instance solid fuel combustion does not produce water.

[3] "Equal quantities of gases and vapours at the same pressure and temperature have the same number of molecules, i.e. $N_A = 6{,}022137 \cdot 10^{-23}$ per mol."

[4] In fact, the preparation of the fuel, especially the heating oil, is a discipline in itself whereby the viscosity and other physical properties of the oil are a limiting factor in themselves in decreasing droplet size when atomising the oil before combustion. Also the preheating of the oil up to 60°C is a factor. For more details see www.iwo.de

- fluctuations in fuel quality/ combustion value (e.g. in the Netherlands the Wobbe-index[5] can vary between 40,4 and 44,6 MJ/m³, requiring 8% extra air. In the EU these fluctuations are expected to increase with Russian gas imports)
- wind influence on chimney (up to 20% for atmospheric burners, 5% for premix burners with deflectors/ draught limiters)

For instance, an air factor of 1,2 means that 20% extra air is added with respect of the stoichiometrically needed volume. Another way of describing the air factor is the oxygen content (O_2) of the flue gases. For instance an air factor ***λ=1,2*** for natural gas equals around **3% O_2** in the flue gases.

So, with an air factor of ***λ=1,2*** there is some 16,6% (0,2/1,2) extra air that goes into the process and the mass balance of the combustion of methane changes as follows:

83,4% * (9,1% CH_4 + 90,9% air) + 16,6% air →

83,4% * (9,1% CO_2 + 18,2%H_2O +72,7% N_2) + 16,6% air

or, substituting 'air' with 20% oxygen and 80% nitrogen in the result:

7,59% CH_4 + 92,41% air → 7,59% CO_2 + 15,18% H_2O + 73,93% N_2 + **3,3% O_2**

So the 16,6% air in the flue gases equals an oxygen content of ca. 3,3% (20% oxygen in air). Normalizing this volume balance to the fuel input, we can say that for the combustion of 1 m³ of methane 12,17 m³ of air is used, resulting in 13,17 m³ of flue gases with the composition as mentioned above: 1 m³ carbon dioxide, 2 m³ water, 9,73 m³ nitrogen and 0,43 m³ oxygen.

To convert these results from methane to natural gas, we must consider that natural gas contains only some 95% of methane and therefore the oxygen content of the flue gases drops to close to **3% O_2.**

2.3.3 Humidity of combustion air

Not only the combustion reaction produces water vapour as one of the outputs, but also a —relatively small— fraction of the water vapour in the flue gases comes from the humidity of the combustion air input. The EN standard prescribes a relative humidity (RH) of 70% and ambient temperature (20°C) for the air input. The look-up table 2-1 shows that the maximum water content (100% RH) of air at 20°C is 2,4 volume%. At 70% RH this is 1,68 vol.%. At 12,17 m³ of air that goes into the combustion, this results in 0,2 m³ of water vapour or around 0,16 litres of water that needs to be added.

Table 2-1. Max. volume % Water of air (1013 mbar) and saturation pressure (psat) at various temperatures (Temp.) ***[source: Farago, 2004]***

Temp.	- 20	-10	0	10	20	30	40	50	60	70	80	90	100
%water	0,16	0,31	0,61	1,2	2,4	4,2	7,4	12,3	19,1	31,2	47,4	70,1	100
psat	155	308	611	1227	2367	4242	7375	12334	19919	31161	47359	70108	101325

2.3.4 Influence of emissions

The balance is also incomplete because it does not contain the emissions of unburned fuel (CH4) and pollutants: carbon monoxide (CO_2), nitrogen oxides (NO_x) and total organic compounds (TOC).

Pfeiffer 2001 of the University of Stuttgart [6] has done extensive tests of emissions of oil- and gas-fired boilers, looking not only at the stationary boiler operation —as is done in

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[5] Natural gas is a mixture of gases. In the EU it is mostly it is methane, but there are also smaller fractions of propane, butane, etc.. The Wobbe-index is a measure of the combustion value of the mixture.

EN standard tests— but especially during cycling (D. *Taktbetrieb*). For the latter he used the boiler loads as described in DIN 4702-8 and calculated the emissions for around 14000 start/stop cycles per year [7].

As the tables below show, the emissions during cycling were much higher –on an annual basis—than during stationary operation, despite the fact that mainly above-standard pre-mix burners were tested In terms of environmental impact –which will be elaborated at a later stage—these are significant numbers.

In terms of actual mass, the numbers are small. In our calculation of the methane combustion we will use a value of 100-120 mg/MJ: CO 24, CH_4 26, NO_x 25-30 mg/MJ + TOC 23 mg C/MJ (say 30 mg hydrocarbons). At 39,8 MJ/m³ for methane this comes down to a total 4-5 gram. This mass does not come on top of the emissions, but replaces a minute part of the other combustion products.

Table 2-2. Emissions gas fired boilers (source Pfeiffer, 1)

Gas fired appliance	Ref.	CO [mg/MJ]		CH_4 [mg/MJ]		TOC [mgC/MJ]	
		Steady state	*Cycling**	*Steady state*	*Cycling**	*Steady state*	*Cycling**
Boiler with premix burner	H1-G1	2,2	32	0,42	19	0,59	16
Premix condensing, flat burner	G2	0,43	21	0,49	36	0,68	31
Premix condensing, flat burner	G3	3,9	10	2,6	33	2,0	28
Instantaneous boiler, flat burner	G7	14	16	0,89	16	0,99	14
Instantaneous boiler, flat burner	G8	6,5	15	0,45	23	0,99	19
Average		***5***	***19***	***0,97***	***25,4***	***1,05***	***21,6***

* Cycling operation based on relative boiler load acc. DIN 4702 / Part 8

Table 2-3. Emissions oil fired boilers (source: Pfeiffer, 1)

Oil fired boiler	Ref.	CO [mg/MJ]		CH_4 [mg/MJ]		TOC [mgC/MJ]	
		Steady state	*Cycling**	*Steady state*	*Cycling**	*Steady state*	*Cycling**
Boiler 1 with jet burner 1	H1-B1	< 0,33	2,3	< 0,40	0,49	< 0,56	1,5
Boiler 1 with jet burner 2	H1-B2	< 0,35	1,9	< 0,43	0,48	< 0,60	1,0
Boiler 1 with jet burner 3	H1-B3	< 0,34	3,7	< 0,41	0,45	< 0,58	1,2
Boiler 1 with jet burner 4	H1-B4	0,34	2,4	< 0,41	0,44	< 0,58	1,6
Boiler 2 with jet burner 5	H2-B5	1,2	43	< 0,42	1,5	< 0,59	17
Boiler 3 with jet burner 6	H3-B6	4,0	7,3	< 0,40	2,0	< 0,56	6,9
Boiler 3 with jet burner 3	H3-B3	5,4	7,8	< 0,41	0,61	< 0,57	1,9
Boiler 3 with jet burner 7	H3-B7	4,3	3,3	< 0,38	0,74	< 0,53	2,4
Average		*2*	*9*	*0,4*	*0,84*	*0,57*	*4,18*

* Cycling operation based on relative boiler load acc. DIN 4702 / Part 8

[6] Dipl.-Ing. Frank Pfeiffer ; Bestimmung des Emissionen klimarelevanter und flüchtiger organischer Spurengase aus Öl- und Gasfeuerungen kleiner Leistung;; Fakultät Energietechnik der Universität Stuttgart; 2001

[7] For a regular boiler this is fairly close to the German average (other sources like Farago mention 16000 cycles). For instantaneous combi-boilers, with on average 50-60 draw-offs per day, the amount of cycles can triple (e.g. 40000 per year).

The tables are typical of the source (Pfeiffer). More information on emissions from oil- and gas fired appliances can be found in Annex D, where Eurofuel reports on the results of the GEMIS 4.3 software package

Please note, that the values measured by Pfeiffer on commercially available boilers in 2001 were already much lower than the ones mentioned in the EN standards.

Having said that, the above tables also do not take into account a number of emissions in practice. In the paragraph 1.4.5 on energy contained in lost fuel this will be discussed in more detail. In short:

- The measurements were done in a laboratory and did not take into account real-life fluctuations in combustion air (pressure, temperature, enthalpy), fuel supply and quality, flue gas duct pressure (wind), etc.. In analogy with the air factor we add an extra 25% for all emissions
- The measurements were done with DIN 4702-8 conditions (39% load →14000 cycles/year) for regular boilers. Correcting for the lower load factor in practice (9%) and the fact that most boilers deliver hot sanitary water (40 000 cycles) this gives a factor 2,8.
- Gas leakage at boiler level was not taken into account. Following prEN 13836:2005 this adds another 0,1% of methane emissions.

All in all, we estimate that around 10-11 g of fuel is lost per m³ of methane input, or around 1,5 weight %.

2.3.5 Converting volume into mass balance

To convert the volume balance into a mass balance, we can use the atomic mass of the elements involved (O=16, N=14, C=12, H=2), also knowing that the mol-volume at ambient conditions is ca. 22 litres [8]. For instance, 22 litres of CH_4 would then weigh 20 (=atomic weight) grams or 0,909 g/l. = 0,909 kg/m³. Table 2-3 gives the conversion from volume to mass balance of the methane combustion.

Table 2-4. Conversion from volume to mass balance of 1 m³ methane combustion

	volume	ato.mass	calc. density	mass
Input	*m³*	*#*	*kg/m³*	*kg*
CH_4	1,00	16	0,73	0,73
air 12,17 m³			*(1,31)*	
- O_2, 21%	2,56	32	1,45	3,72
- N_2, 79%	9,61	28	1,27	12,24
H_2O in air*	0,20	18	0,82	0,16
Total	13,37			16,84
Output				
CO_2	1,00	44	2,00	2,00
H_2O combustion*	2,00	18	0,82	1,64
H_2O in air*	0,20	18	0,82	0,16
N_2	9,74	28	1,27	12,39
O_2	0,44	32	1,45	0,63
$CO/CH_4/TO_x/NO_x$				0,01
Total	13,37			16,84

*= water vapour, not liquid —> density < 1

As mentioned, this mass balance is for 100% methane and not exact for natural gas.

[8] From Avogadro: In the reference situation of 0 °C and 1013 mbar the mol-volume is 22,41 litres and the kilomol volume around 22,41 m³. At ambient conditions the mol-volume is ca. 22 litres. Furthermore, it is assumed that the ultimate flue gas temperature and pressure equals the temperature and pressure conditions of the fuel and air inputs.

2.4 Energy balance combustion

2.4.1 Introduction

During the combustion the chemical energy of the fuel reacting with the oxygen is transformed into three types of heating energy:

- Radiation energy of the flame/burner
- Convection energy of the combustion products (temperature of the flue gases) and
- Latent heat of the water vapour (the heat released when the vapour condenses into liquid)

Furthermore, the combustion process has to carry the ballast of the excess air, due to the air factor., and at the most parts of the emissions –the ones containing carbon [9] — count as lost fuel.

All in all, the general mass balance for the inputs in the methane combustion in our previous example looks like this:

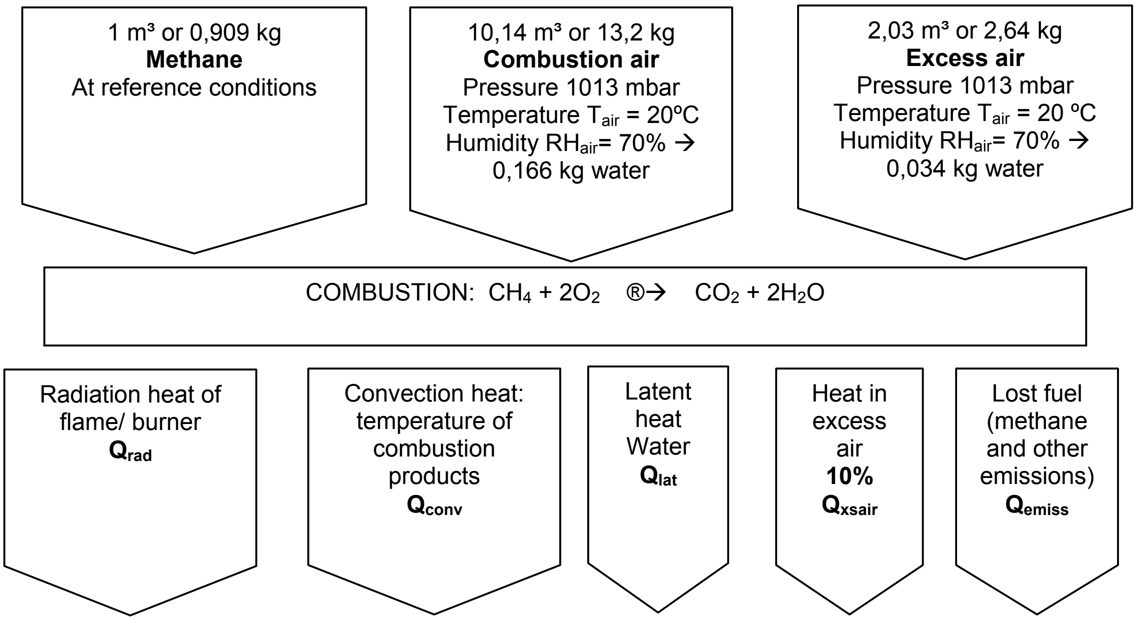


Figure 2-1. Mass balance methane combustion

The total heat heat released by the combustion process is the ***combustion heat***, also known as combustion energy or enthalpy, symbol ΔH. The unit is MJ (megajoules, 10^6) or kWh of heating energy, often expressed as the ***Gross Calorific Value GCV*** or the ***upper heating value uhv*** (D. *Brennwert*) of the fuel. The equation for methane (at 273 K, 1013 mbar [10]) is

$$\Delta H_{methane} = Q_{flame} + Q_{latent} + Q_{xsair} + Q_{fuel\text{-}loss} = 39{,}8\ MJ/m^3$$

If we leave out the latent heat contained in the water vapour, i.e. the heat released when the water condenses, we find a value known as the ***Net Calorific Value***, the ***'dry gas' combustion value*** or the ***lower heating value ulv*** (D. *Heizwert*). In the

[9] For oil this also includes sulphur.

[10] Note that for gases the temperature is an important parameter, e.g. at 25°C the GCV of methane is 36,3 MJ/m³.

global combustion reaction the ΔH has a negative connotation in the right-hand side of the equation, indicating that the reaction is ***exothermic*** (produces heat, as opposed to an ***endothermic*** reaction which consumes heat).

The table below gives the enthalpies for some fuels:

Table 2-5. Energy levels of different fuels at 273 K and 1013 mbar

	Gross calorific value Hs [MJ/m³]	Net calorific value Hi [MJ/m³]	Hs/Hi	Hs – Hi [MJ/m³]	Volume of condensate (theoretical) [kg/m³]
Town gas	19,73	17,53	1,13	2,20	0,89
Natural gas LL	35,21	31,79	1,11	3,42	1,53
Natural gas E	41,25	37,26	1,11	3,99	1,63
Propane	100,87	92,88	1,09	7,99	3,37
Fuel oil (Figurerelate to 1 ltr)	38,45	36,29	1,06	2,16	0,88

In the rest of the paragraph we will explore to see what is the share of Q_{flame} , Q_{latent} , Q_{xsair}, $Q_{fuel\text{-}loss}$ and what temperature levels are associated with these heat energy outputs.

Note that the energy balance of the combustion process is only the first step of the total energy balance, but we will deal with the heat transfer in the burner, heat exchanger(s), etc. in the following paragraphs.

2.4.2 Combustion heat Q_{rad} + Q_{conv}

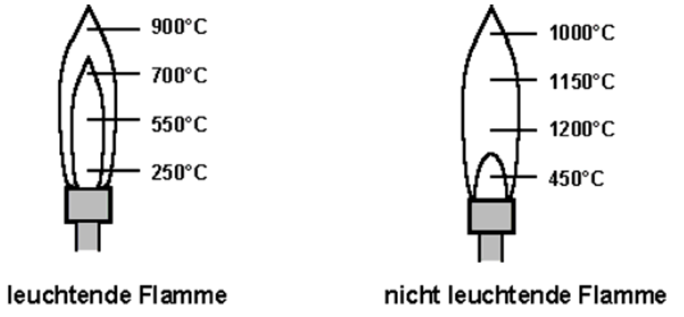
Flames

Starting point of a high-temperature combustion is the flame. In a 'normal' flame, e.g. of a candle, there are three zones:

- A ***fuel-preparation*** zone where the gaseous fuel is heated up to a temperature – the ***'ignition temperature'***—starting the dissociation process (breaking up the hydrocarbon molecules in smaller fractions) leading up the combustion chain reactions. When the gas reaches the ignition temperature (around 300-500°C) it attracts the minimum amount of air necessary from its surroundings (e.g. air factor of 0,5) and starts combustion. In the case of a liquid fuel (oil) this process is preceded by a step where the oil is atomised into droplets, which are then vaporized.
- A ***'rich combustion'*** zone where the flame is above the ignition temperature and minimal air factor but has too little oxygen/air with respect of the stoichiometric combustion (0,5 < air factor < 1). In this zone very small soot particles are formed and burnt, emitting a yellow light. Rich combustion is also usually accompanied by higher emissions of CO.
- A ***'lean combustion'*** zone (air factor > 1) with a blue flame colour. At a certain temperature the flame temperature attracts so much air/oxygen that the air factor becomes too high (e.g. higher than 2) and the flame has reached its visible boundaries. Lean combustion is also usually accompanied by higher emissions of NO_x.

Such a flame is known as a ***'diffusion flame'***, where the air input to the combustion process is dependent on the flame-temperature and the mixing of air/fuel takes place concurrently with the combustion. This flame is typical of candles, matches, etc. but also to a large extend of ***partial pre-mix burners*** (a.k.a. *'atmospheric burners'*, type B_{11}) in boilers, where the primary air flow is regulated (pre-mixed) through a venturi with the fuel flow and secondary air completes the job during combustion. In contrast, in ***pre-mix burners*** the air input to the combustion process is independent of the flame-temperature and a combustion fan gives an exact dosage of air to the mixture. The fuel/air is fully pre-mixed before entering the burner and produces a flame with a very different temperature distribution profile (see picture) but also a more favourable emission profile.

Figure 2-2.
Temperature distribution in a diffusion-flame (left) and a pre-mix flame (right) of a Bunsen-burner [Farago, 2004]



Flame temperature

Calculating the temperature of the flame is not an easy task. A first theoretical value called the ***calorific flame temperature*** can be calculated from the enthalpy of the fuel under the simple assumption that all energy is converted into hot combustion products. The temperature increase (above ambient) of the combustion products ΔT comes from the ***enthalpy*** of the fuel ΔH, the ***mass*** of the combustion products ***m*** and their ***specific heat cp***:

$$\Delta H_{methane} = m * cp * \Delta T$$

The reaction temperature $T_{reaction}$ is then defined as $T_{reaction} = T_a + \Delta T$, where T_a is the start temperature of the combustion products (usually ambient, i.e. 20°C).

The enthalpy of the fuel is known (see paragraph 2.4.1: 39, 8 MJ/m³), the mass of the combustion products is taken from the mass balance in the previous paragraph 2.3 and the specific heat is a look-up materials property (see table below).

Table 2-6. Density and specific heat of some substances

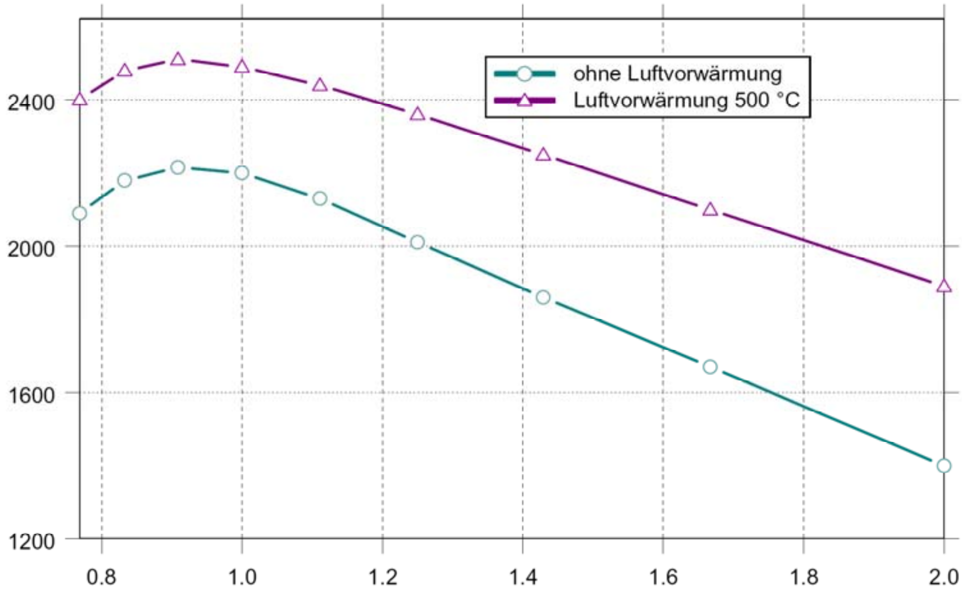
Substance (properties at 293K and 1013 mbar)	formula	density ρ kg/m³	specific heat c_p kJ/(kgK)
water	H_2O	1	4,18
air	79% N_2, 21%O_2	1,29	1
oxygen	O_2	1,43	1,4
nitrogen	N_2	1,25	1,25
methane	CH_4	0,72	2,21
propane	C_3H_6	2,02	1,53
(iso-) butane	C_4H_8	2,67	1,61
carbon monoxide	CO	1,25	1,05
carbon dioxide	CO_2	1,98	0,82
sulphur dioxide	SO_2	2,93	0,64
acetylene	C_2H_2	1,18	1,67

Table 2-7. Calculating calorific flame temperature for 1 m³ methane at air factor = 1,2

Output	**mass** *kg*	**spec. heat** *kJ/(kgK)*	**mass*spec. Heat** *kJ/K*	**Temperature increase at fuel enthalpy in K** *Hs= 39,8 MJ*
CO_2	2,00	0,82	1,64	
H_2O combustion*	1,64	4,18	6,84	
H_2O in air*	0,16	4,18	0,68	
N_2	12,39	1,04	12,89	
O_2	0,63	1,40	0,89	
CO/CH4/TOx/Nox	pm		pm	
Total	16,84		22,94	1735

This calorific flame temperature is in practice never reached, because of dissociation effects (incomplete combustion), especially at air factor=1 (stoichiometric combustion). Another value that takes into account the dissociation process is the ***adiabatic flame temperature***. Especially for gases the adiabatic flame temperature is fairly close to the calorific flame temperature at practical air factor values. The adiabatic flame temperature is independent of the size of the flame and the dimensions of the combustion chamber and the power output of the burner.

Figure 2-3.
Adiabatic flame temperature in K =°C +273 of methane-air mix at various air factors 0,8 to 2.0. The highest temperature is reached at air factor=0,9. *[source Farago, DLR]* Please note that temperatures are give in Kelvin (K =°C +273). The adiabatic temperature at air factor =1,2 is around 2000K or 1730°C, which is close that what we calculated earlier.



2.4.3 Latent condensation heat Q_{latent}

As mentioned in the introduction, the ***latent condensation heat*** is the heat contained in the water vapour from combustion when condensing. Numerically it is the difference between ***Gross and Net Calorific Values (GCV and NCV)*** of the fuel.

In the case of our example of methane combustion around 1,8 kg of water vapour is produced per m³ of methane (see mass balance: 1,64 kg from combustion, 0,16 kg from humidity in the incoming air at air factor 1,2). The specific latent condensation heat of water is 2,27 MJ/kg, so per m³ of methane 4,09 MJ of condensation heat is available. Compared to the GCV of methane of 39,8 MJ/ m³, this is 10,2%. Compared to the NCV it adds an extra 11%. As natural gas consists mostly of methane, the same numbers apply roughly to natural gas.

For other fuels, the stoichiometric combustion equations are different and therefore the water vapour and the maximum amount of latent heat is different. For oil-fired boilers this is around 6% and for propane it is 8-9%.

In theory, the latent condensation heat can be fully recovered, if somewhere in the boiler before the flue gases go up the chimney or flue duct the flue gases are cooled to ambient temperature (<20°C).

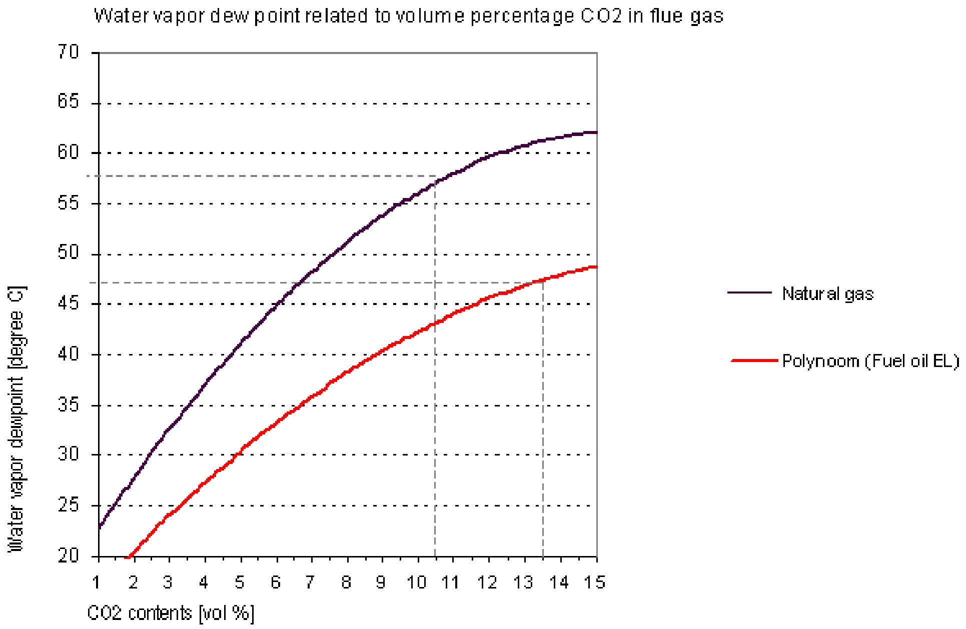
In a ***non-condensing boiler*** the flue gases –after flowing through the heat exchanger— leave the boiler at a temperature of somewhere between 200 and 300°C still in the form of water vapour. Somewhere in the atmosphere the water vapour will condense against the cooler outside air, but in principle all the latent heat of the condensation process is lost for the boiler and the CH-system.

To establish where this point of total-loss is, we can use the EN standards that define that if the flue gas temperature stays above 160°C there is no risk of condensing. This is a technical level, taking into account extreme circumstances.

The EN standards speak of a dedicated '***condensing boiler***' at flue gas temperatures of lower than 80°C. 'Condensing' relates to the fact that the water vapour in the flue gas comes into contact with a cold surface of the heat exchanger and than turns into liquid, releasing the latent heat of condensation. This condensation of air with a 100% ***relative humidity (RH)*** takes place at a temperature level that is known as the ***'dew point'***. This dew point also depends on other parameters, but in general one can say that condensing starts at a surface temperature of the heat exchanger (=boiler temperature) of just below 57°C for gas and 46°C for oil. At a boiler return temperature of 30°C some 70-80% of all latent heat is recovered. At 35°C boiler return temperature around 50% is recovered.

The graph below gives the water vapour dew point at (near) stoichiometric combustion. At higher lambdas, the dew point will even be lower. Natural gas starts condensing at 57° C and oil at 47° C. At lamdbas of 1,25 the CO_2 content decreases and with it also the dew point to approximately 53° C for gas and 44° C for oil.

Figure 2-4.
Dew point water vapor for gas- and oil flue gasses



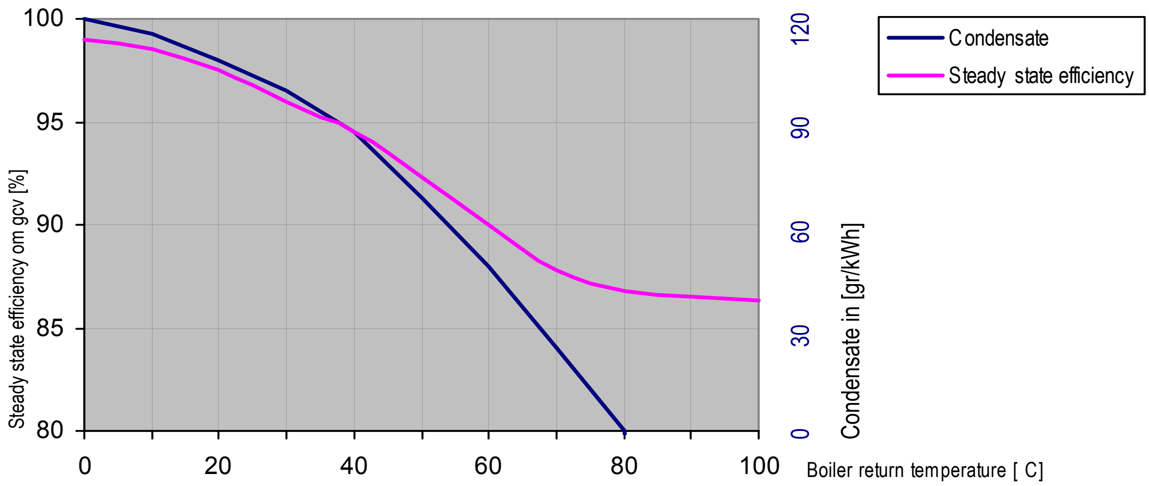


Figure 2-5. Steady state efficiency and amount of condensate related to return temperature of gas fired boiler.

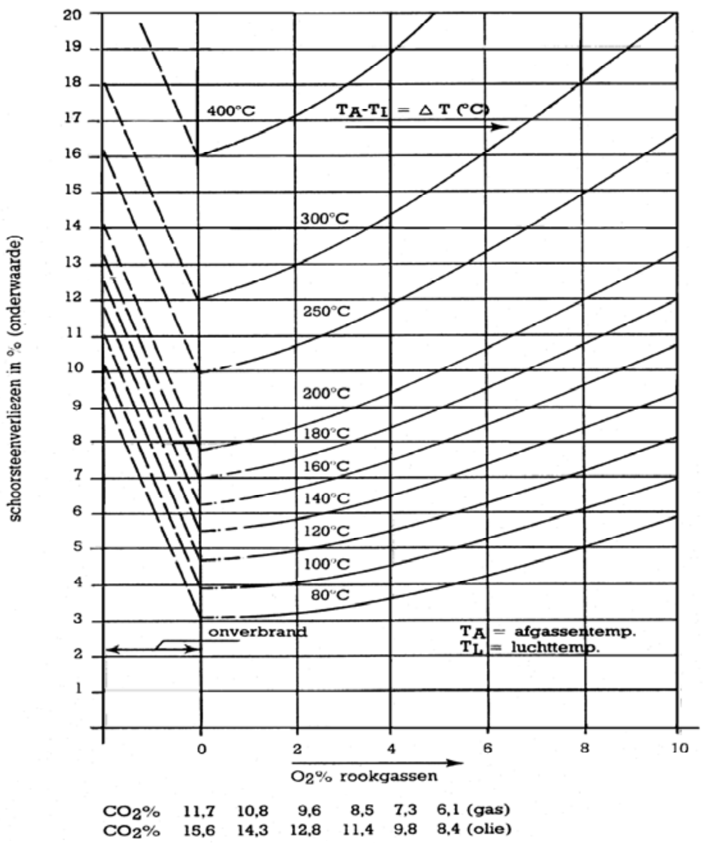
2.4.4 *Heat loss in excess combustion air Q_{xsair}*

To complete the picture of the energy balance of the combustion process we include the excess air that is the consequence of the air factor.

Obviously, the extra air into the combustion process comes at a penalty. For instance, in a 10 kW gas-fired boiler with an air factor ***λ=1,3*** this means that an extra 3 m³ is heated from ambient temperature to e.g. 1000°C combustion temperature. With respect of the stoichiometric process this initially costs some 9% extra[11], of which of course in the heat exchanger a large part is recuperated. But still, 'losses' in the order of magnitude of 2% remain. A rule-of-thumb is that *every 1% O_2 extra results in 0,5% efficiency loss*. This is of course only true when measuring flue gas exit temperatures, but it gives an order of magnitude for the partitioning.

All in all, as described in the EN standards, an air factor of 1,2-1,25 is standard practice for higher power outputs of gas- or oil fired premix burners. For lower outputs (<10 kW) or not-premix burners it can be 1,3 or higher (up to 1,5-1,6).

Figure 2-6.
Flue heat losses due to the air factor, expressed in % O2, showing that every 1% O2 leads to 0,5% efficiency loss. Source: Stooktechnologie, 2005]



2.4.5 *Fuel loss $Q_{fuel\text{-}loss}$*

The research by Pfeiffer, as mentioned in the mass balance, allows us to quantify the energy lost because of incomplete combustion. In principle, we can say that all carbon (C) that ends up in the emissions comes from the methane and quantifies the fuel lost. This leaves out the NO_x emissions, but we are still left with 24 mg/MJ CO (ato.mass 28), 16 mg/MJ CH4 (ato. mass 16) and 12 mg carbon/MJ TOC (carbon ato. mass is 12). Calculating these numbers on a mass basis this means that the equivalent of ca. 3 g of methane is lost per m³ methane of carbon-containing emissions. At a density of 0,73 kg/ m³ this means that some 0,4% of fuel energy is lost.

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[11] Air at 1 kJ/K.m³ for a 3 m³ with a temperature increase of 1000 K → 3000 kJ = 3 MJ= 0,9 kWh/h = 0,9 kW → 9% of 10 kW.

Obviously, this was measured in a laboratory, which means that the fluctuations in combustion air (atmospheric pressure, temperature, etc.), fuel (pressure, wobbe-index), etc. were not taken into account. Following an analogy with the air factor, we can assume that in real-life the emissions are some 25% higher, i.e. 0,5%.

Furthermore, it has to be considered that Pfeiffer did his measurements at DIN 4702-8 conditions, which means on average a heat load of 39%. In reality, as a study of Wolfenbüttel pointed out, the heat load in the heating season is more in the area of 9%. At a modulation ratio of 30% this still means that the average number of on-off cycles in reality is higher than the 14000 cycles assumed by Pfeiffer. No statistics on average cycling behaviour are available, but anecdotal evidence suggests numbers in the range of 16000 - 20000 cycles. This then leads to an annual loss of 0,65% for a regular boiler at say 18000 cycles/year.

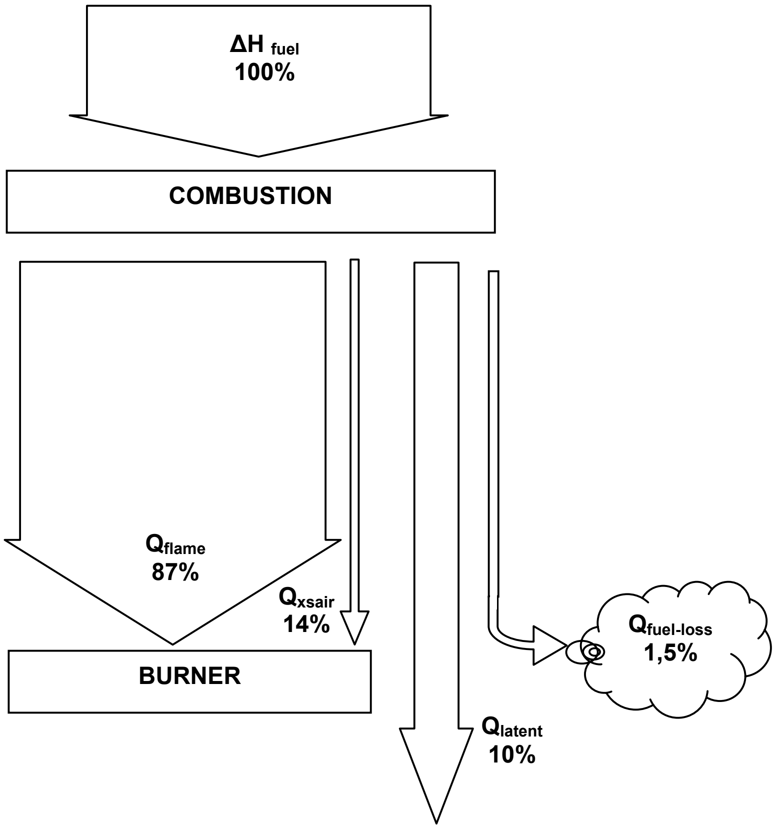
Pfeiffer tested regular boilers, i.e. without the sanitary hot water function. In case of an instantaneous boiler that switches at every draw-off, the number of cycles is much higher, e.g. in the range of 40-50 000 cycles. The corresponding fuel-loss in that situation is almost triple, say 1,5%. According to the BED Market study 2006 by BRG Consult around 90% of the gas-fired boilers are operated with a sanitary hot water function, either as a combi or with an external cylinder, and the vast majority of these are instantaneous. For the average EU domestic boiler a value of 1,4% fuel losses is therefore deemed realistic.

Pfeiffer did not take into account gas leakage. No statistics on the subject are known, but the prEN 13836 specifies that a boiler satisfies the requirements if the leakage is of the gas valve is less than 0,06 dm^3/h (upstream gas pressure 150 mbar) and 0,14 dm^3/h for the whole boiler. One might argue that these are maximum values; on the other hand these are laboratory measurements where no inaccuracies in installation practice should occur. Per annum (8760 hours) this equals some 0,5 to 1 m^3 per annum. At an average consumption of 1000 m^3 /year this adds another 0,1% energy loss.

All in all, we estimate for the Base Case, i.e. the average EU boiler, a figure of 1,5% of energy in fuel losses (combi boiler, largely instantaneous, 40000 cycles/a).

In summary, the heat balance for the combustion process of a gas-fired boiler methane with air factor 1,2 looks like this. Please note that the latent heat includes not only the water vapour from combustion, but also the potential condensation heat of the water from the incoming air.

Figure 2-7.
Energy balance of combustion process methane

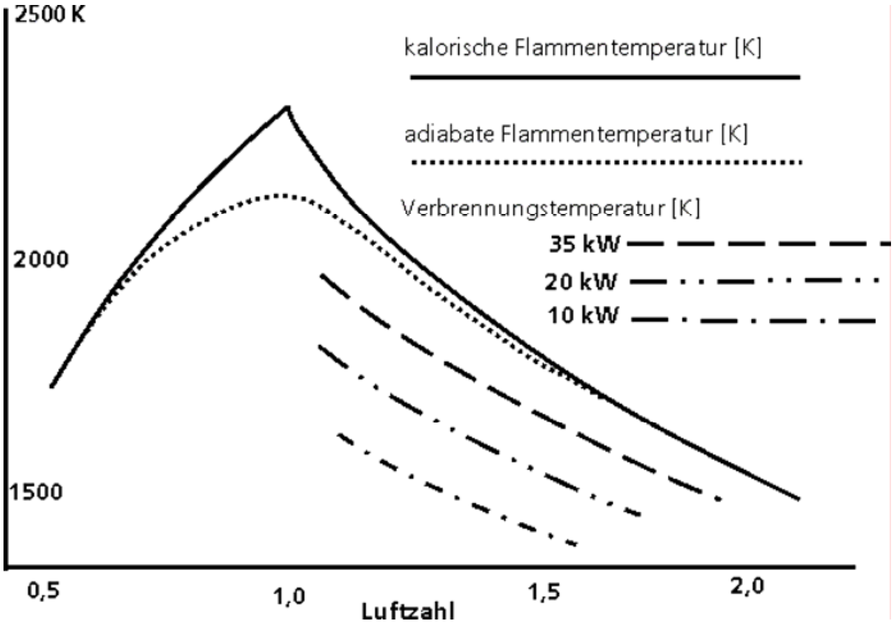


2.5 Energy balance burner

Many authors do not distinguish between the energy balance of combustion and the burner, because in terms of actual measurements it is very difficult to measure the flame temperature without some sort of burner. Yet, in explaining the heat balance of the whole process it is functional, because in the interaction between the flame and the burner construction there is much more going on than meets the eye.

For starters, when you measure the temperature of the combustion products at the burner, the so-called ***combustion temperature*** (D. *Verbrennungstemperatur*), there always seem to be 100-200°C missing compared to the adiabatic flame temperature. The graph below gives an illustration of the above in an actual combustion chamber and burner operated at 10, 20 and 35 kW.

Figure 2-8.
Flame and combustion temperatures in an oil-fired boiler (*D. Blaubrenner*) at 70°C boiler temperature, according to various definitions.
[source: Farago, Brennstoffkunde, DLR 2004]



If we assume the power output of the burner as a measure of the flame size, the picture shows that at a smaller flame size (10 kW on a 35 kW burner), the combustion temperature, i.e. the temperature of the combustion products, is significantly lower than at nominal power/ flame size. Between 35 and 10 kW power the temperature difference is some 350 K. Assuming this is proportional to the temperature difference with the ambient (ca. 1700°C) this means that at 10 kW (30% load) the share of radiation energy has increased by 20% with respect of 35 kW (100% load). On average, every 10% decrease in load has yielded around 2,5-3% more radiation share. It may seem contra-intuitive that a smaller flame gives off relatively more radiation heat, but the keyword here is 'relative', because in fact the size of the burner bed and the combustion chamber do not change. In other words, one could also say that with a larger burner plate (compared to its nominal capacity in W/cm²) the radiation share increases (and the convection share, i.e. the temperature of the combustion gases, decreases) [12].

Of course there is a limit to decreasing the burner load, which has to do with air and flame velocity, flame stability, laminar and turbulent flame fronts, etc.. We will not go into that complex matter[13], but stick to the more profane thermodynamics.

On the next page there are several illustrations of research concerning temperature levels inside a burner, showing that there is more to be considered.

The university of Eindhoven has done experiments of a 'flame in a box', which amongst others give a detailed insight into the temperature fields of a pre-mix burner. The picture shows temperatures near a nozzle of a conventional nozzle, showing that the flame temperature at the burner nozzle is around 600-800 K (300-500°C). From this we assume that the temperature of a conventional burner plate, made of perforated thin refractory steel plate (surface around 240 cm² for 24 kW burner → weight ca. 80-100 g.) is on average around 400°C. The flame temperature itself rises to around the adiabatic flame temperature of 2000 K (1730°C).

Dietzinger [2006] at the university of Stuttgart has done several experiments on the propagation of the temperatures of a methane/air mixture in a porous ceramic burner,

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[12] Electro-magnetic waves in the visible light spectrum, but also in the UV (ultra-violet) and IR (infra-red) spectrum. In fact, the radiation in the UV-spectrum of the flame is the basis for optical flame-ignition control sensors.

[13] Dietzinger 2006 gives a good overview of the latest insights in flame modelling techniques and numerical tools available.

showing the propagation of the temperature at the Z-axis of the burner. In the area between the hole plate ('flame barrier') and burner bed the temperature rises to the ignition temperature (550-600°C) and then —at the bottom of the 20 mm thick burner bed— jumps to a temperature of around 1600°C. Inside the burner bed of this 'flameless burner' the temperature then decreases to around 1100-1200°C before leaving the burner. Already at a height of 5 mm above the burner bed the temperatures have dropped to below 1000°C and laboratory measurements of the flue gases may lead to believe that this is a low temperature burner, whereas in reality the high temperatures are there, but inside the burner. In fact, in this case the average temperature of the burner bed is 1300°C.

The results from Eindhoven and Stuttgart represent two extremes in pre-mix burners. Somewhere in between we find ceramic surface burners, where in fact the flames 'sit' halfway inside the burner nozzles. There, the burner plate reaches temperatures up to 1000°C and the temperature of the combustion products is around 1100°C.

The table below gives an estimate of temperature levels between burner bed, flame and combustion products.

Table 2-8. Estimated temperatures and loads for pre-mix burners (at air factor 1,2, no preheat air)

Pre-mix burner type	Burner plate temperature [°C]	Combustion products temperature at 10 mm [°C]	Radiation share [%]	Max. load [W/cm²]	Surface for 20 kW [cm²]
Steel plate	400	1300	5%	100	200
Radiation burner (ceramic/steel)	900	1100	20-25%	300-400	70
Porous ceramic burner	1200	900-1000	25-30%	300 (>1000, experimental)	70

The table also gives typical burner loads in terms of watts burner output per surface area, showing that the radiation burners can be much more compact for the same output power.

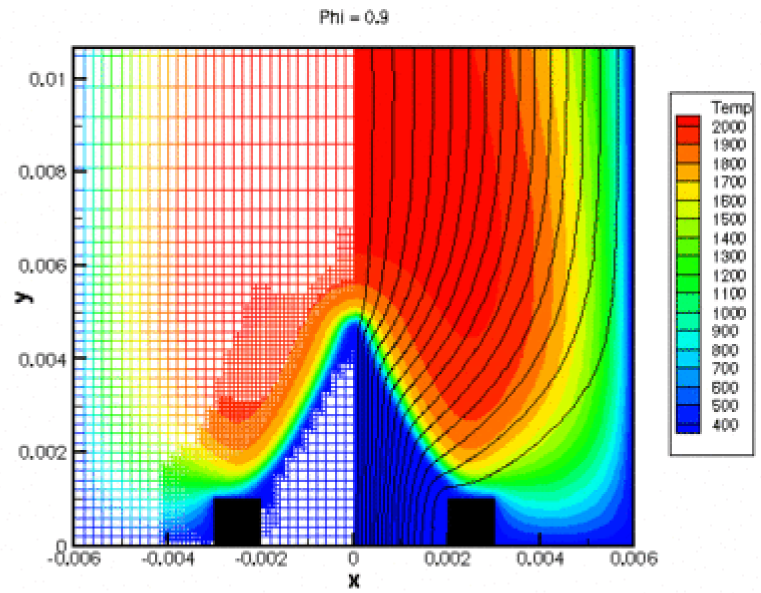
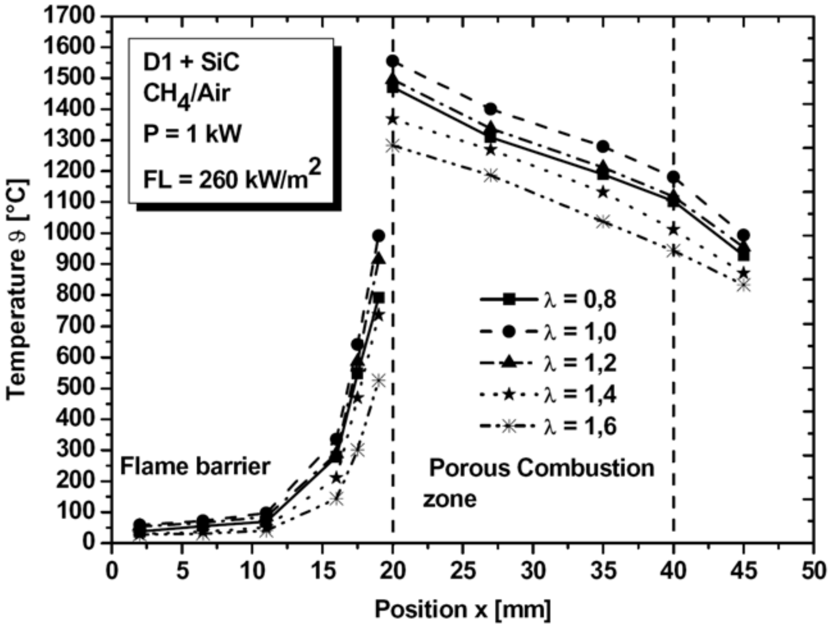


Figure 2-9.
Numerical result of a 2D temperature field of a flame in a box
[source: TU Eindhoven, faculty Mechanical Engineering, 2006]

Figure 2-10.
Ceramic porous burner: Propagation of temperature with a methane/air mix. The graphs show an experiment whereby the temperature is measured in the flame barrier and throughout the thickness of a 20 mm porous ceramic burner. Note that the initial temperature after ignition is close to the calculated adiabatic flame temperature and that the combustion products –while giving off their heat to the burner— cool down to a level <1000°C already 10 mm after the burner surface. [Dietzinger, 2006]



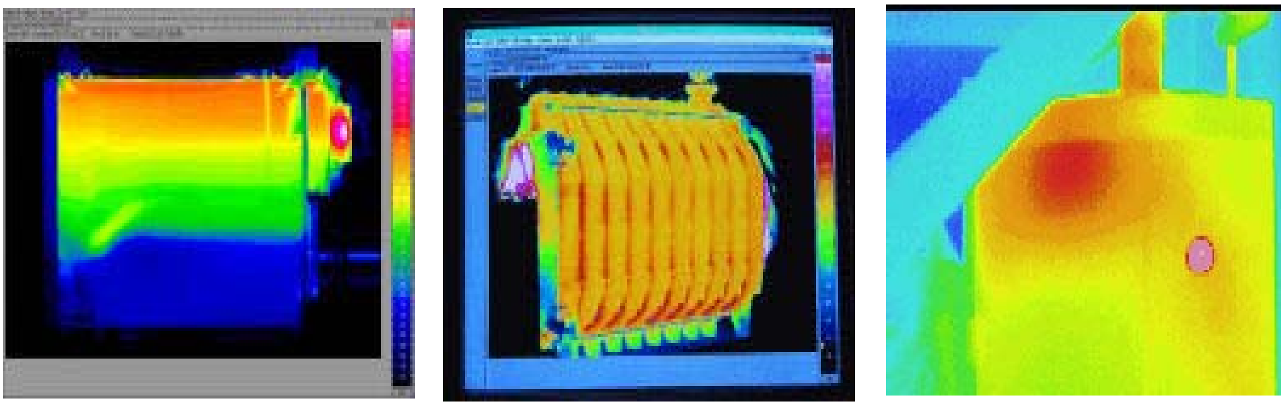


Figure 2-11.
Thermographic pictures of boilers. Left: Steel boiler with jet burner (www.trm.at) Middle: Cast iron boiler (www.trm.at). Right Detail of boiler, burner in red, flue duct in orange.

The wall thickness of a refractory steel plate burner plate is ca. 0,5-0,7 mm, weighing 80-100 g for 20 kW power (+burner frame of 200-250 g). Ceramic radiation burners are 3,5 mm thick and the commercially available porous ceramic burners are now 15 mm thick. For non-stationary (cycling) operation this is relevant, because burner plates cool down in a matter of 3-10 seconds, which means that at every start-up this mass has to be heated.

Calculation example
Assume 350 grams of steel burner+frame with a specific heat of 0,46 kJ/(kgK), to be heated to an average temperature difference dT=300 K. This is 48,3 kJ per cycle. At 40 000 cycles per year, this represents 1932 MJ or 536 kWh. On a total energy consumption of e.g. a combi boiler of 14 000 kWh/year this is around 4%. This energy is not lost. Most of the cooling down will take place during the after-purge at the end of each cycle, where the combustion air will then give off its heat to the heat exchanger and boiler water. If this is then 'useful energy' and not lead to a room temperature overshoot will depend on whether the boiler controls anticipate this extra energy input.

During stationary operation, i.e. during the combustion process, there is also heat transfer.

Steel plate burners are usually fixed to the heat exchanger boiler, which means that a large part of the heat is transferred usefully to the combustion chamber and heat exchanger on the side of the burner. Another part of the heat will be transferred to the space between heat exchanger body and the surrounding casing, where in modern boilers it is picked up to a large extend by the combustion air fan, i.e. it preheats the incoming combustion air. For another part, it will be a major contributor to the heating of the casing, i.e. radiation losses of the boiler to the ambient. The picture at the bottom of the previous page shows thermo-graphic pictures of the boiler casing, showing clearly the 'hot spot' of the burner location.

The following equations summarize the above:

$$Q_b = Q_{b\_conv} + Q_{b\_rad} + Q_{b\_cond}$$

with

$$Q_{b\_conv} = Q_{b\_conv\_combust} + Q_{b\_conv\_case}$$

$$Q_{b\_rad} = Q_{b\_rad\_combust} + Q_{b\_rad\_case}$$

$$Q_{b\_cond} = Q_{b\_cond\_exch} + Q_{b\_cond\_case\_air}$$

where

Q_b = heat out burner

Q_{b_conv} = convection heat burner (combustion temperature* mass combustion products)

Q_{b_rad} = radiation heat burner/ flame

Q_{b_cond} = conduction heat of burner to surroundings

and

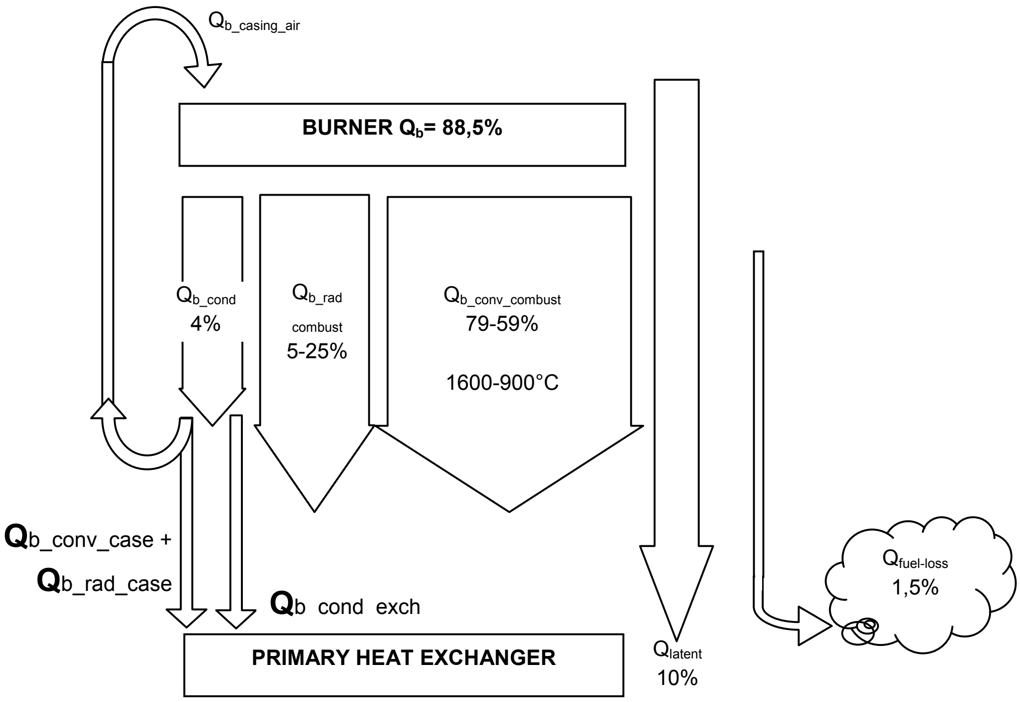
$Q_{b\_conv\_combust}$, $Q_{b\_conv\_case}$ is convection heat transfer to combustion and casing;

$Q_{b\_rad\_combust}$, $Q_{b\_rad\_case}$ is radiation heat transfer to combustion and casing;

$Q_{b\_cond\_exch}$, $Q_{b\_cond\_case\_air}$ is conduction heat to heat exchanger and to the air between casing and heat exchanger.

The picture below gives a Shankey-diagram of the flows. Percentages relate to Q_b = 100%.

Figure 2.12.
Energy balance of gas burner



2.6 Heat balance primary heat exchanger

2.6.1 Introduction

In the primary heat exchanger –and in case of non-condensing boilers the only heat exchanger— the radiation heat and convection heat coming from the burner is transmitted to the boiler water. The boiler water returning from the CH-circuit ('boiler return temperature') has a temperature somewhere between 25 and 70°C. It is heated by somewhere in the range of 5 to 20°C before it leaves the boiler.

In the heat exchanger/ combustion chamber there are the parts that can be 'seen' by the burner and that are subject to the radiation heat. All parts of the heat exchanger are subject to the convection, i.e. the hot flue gases.

Radiation and convection heat transfer are very much linked, but in a publication of the *Verbundnetz Gas AG* [14] an attempt was made at some simplified radiation modelling in an industrial burner, starting from the general Stefan-Bolzman formula:

$$Q_{rad} = A * \varepsilon_{res} * \sigma_s * (T_g^4 - T_w^4)$$

where

Q_{rad} : the radiation heat energy

A : the surface of radiation heat transfer in m²,

ε_{res} : the resulting emission-factor

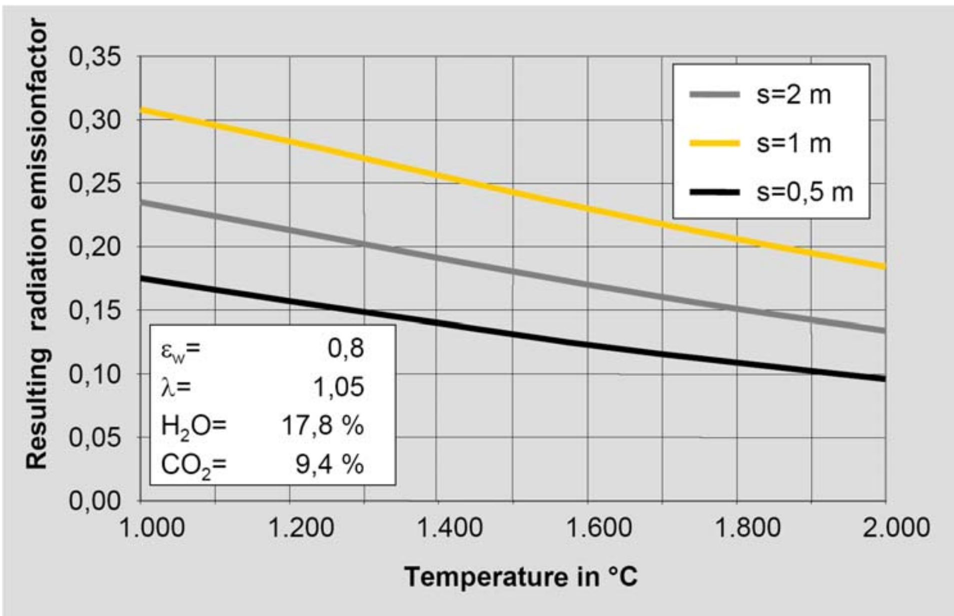
σ_s : the constant of Stefan-Bolzmann: **5,67** * 10^{-8} W/m²K⁴,

T_g,- T_w : temperatures of the gas and the wall in K

The graph below gives an example of the resulting emission factor for an industrial burner/ combustion chamber. It shows that for this burner the maximum radiation is achieved at a height of the combustion chamber of 1 m. At a height of 0,5 m the ε_{res} is almost 50% lower and at 2 m the ε_{res} is around 25% lower. This shows that the dimensions of the combustion chamber are important in maximizing the radiation fraction.

Furthermore, the graph shows that the radiation emission factor increases at a lower temperature from 0,31 at 1000°C to 0,18 at 2000°C.

Figure 2-13
Resulting emission-factor for a combustion chamber, as depending on temperature and the height of the combustion chamber (s= height or 'layer thickness).



[14] Erdgas-Report 1/03, *Industrielle Gasbrenner*, Verbundnet Gas AG

The *Erdgas Report 1/03* mentions a value of ε = 0,2 to 0,3 for normal burners and ε = 0,6 for radiation burners.

The convection heat transfer is depending linearly on the temperature difference. A simplified equation for the convection heat tranfer is given by the same source:

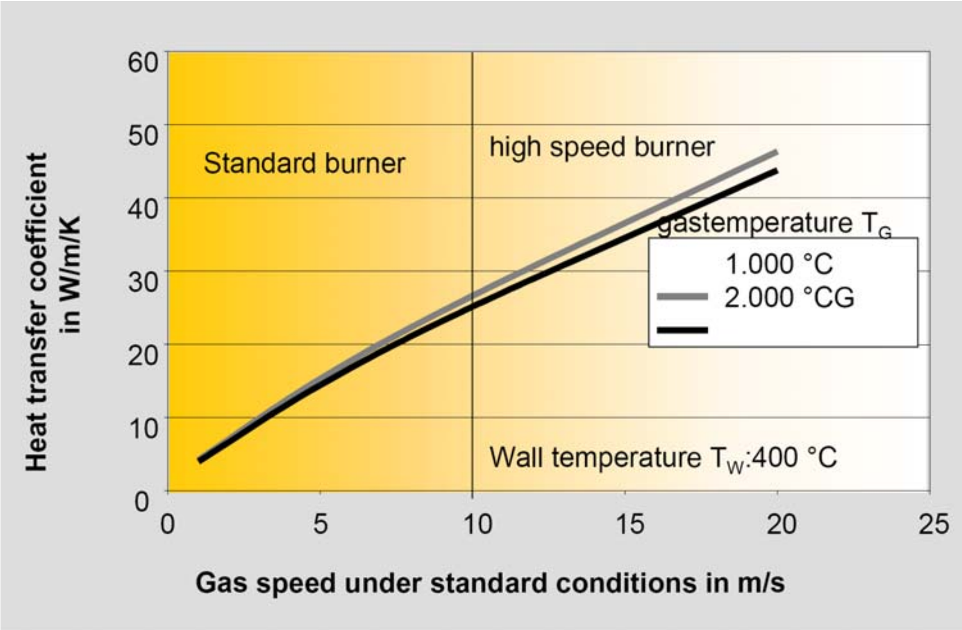
$$Q_{conv} = A * \alpha * (T_g - T_w)$$

where

A	= heat transmission surface
α	= heat transmission coefficient in W/m/K
T_g, T_W	= temperatures of the flue gas and the wall.

The convective heat transmission coefficient depends on the velocity of the flue gas, as shown in the graph below.

Figure 2-14.
Convective heat transfer coefficient
(Erdgas report 1/03, 2003)



Combining the two graphs it is clear that at lower heat output of a modulating boiler (at constant air factor) the convection heat transfer decreases, whereas –given the lower burner load— at the same time the radiation heat transfer increases.

We will not use the formulas for radiation and convection losses directly in describing the heat balance, but they may be useful in describing some of the phenomena in Task 6 (design options).

For the heat balance we will follow the elements of the Boiler Cycle method, distinguishing between energy transfer during 'burner-on' and 'burner off' mode, as well as some additional findings regarding start-stop losses. The important issues are:

'Burner on' operation:

- Flue gas losses
- Radiation, convection and conduction losses through the generator envelope

'Burner off' operation:

- Standing losses (radiation, convection (incl. flue gas) and conduction)

'Start-stop' losses:

- Pre-purge losses
- After purge losses/gains
- Efficiency losses caused by cycling (German: Takten)

2.6.2 Flue gas losses in on-mode

The *primary* heat exchanger is designed to capture the radiation heat from the burner and —after that-to best transfer the heat from the flue gases to the boiler water, but without condensation of the flue gases.

The important parameters are the heat exchanger surface (A), the temperature difference between the flue gas and the boiler water (dT) and convection coefficients that are typical of a configuration (k' and k"). Most heat exchangers are ***counter flow***, i.e. with the hottest flue gases hitting the hottest boiler water (just before exiting) and the coldest flue gases hitting the coldest part of the heat exchanger, i.e. just where the colder return boiler water enters the heat exchanger.

Traditional standard boilers have heat-exchange surfaces that allow flue gas temperatures to cool down to around 250°C; combined with the traditional boiler feed water temperature control (constant at 80 or 90°C) these products are altogether not very energy efficient. With this standard boiler type, weather controlled systems can only be realized by adding mixing valves to lower the feed temperatures.

A mixing valve, mixes the return boiler water (35°C) with the heated feed water coming from the boiler (e.g. 80°C). The result is, that the return water becomes 60°C before it goes into the heat exchanger, i.e. above the dew point, and the feed water becomes ca. 55°C just before it goes into CH-circuit. Effectively the heat exchanger is working at a 90/70 or 60/80°C regime, whereas from the point of view of the CH-circuit the boiler appears to be working at e.g. a 35/55°C regime.

A special case in this context, is the **low-temperature ('LT') boiler**, which in fact is a further development of the traditional standard boiler with constant feed temperature (90 – 70°C regime during entire heating season).

The low temperature boiler was developed in the beginning of the 1980s to reduce flue-gas losses and standing losses and to facilitate lower feed- and return water temperatures without the permanent need of mixing valves,. Goal was to optimize boiler efficiency without achieving condensation in boiler or chimney, despite the fact that the return temperature of the boiler water —typically 35°C— is well below the dew-point (57°C for gas, 46°C for oil).

There are several options to achieve this:

1. Using a **double or triple-walled heat exchanger**, whereby the primary finned tubes are enclosed by a second (and possibly third) tube that prevents the flue gases from direct contact with the primary tube and this secondary tube. The surface of this secondary tube will have an overall temperatures of above dew-point because of its lower heat transfer coefficient. In other words, heat transfer in the critical parts of the heat exchanger is deliberately decreased to prevent condensation of the flue gasses.
2. Using a **U-turn** heat exchanger and flue gas trajectory, whereby at the end the flue gases pick up enough parasitic heat from the beginning of the trajectory to keep them above dew-point. To avoid condensing at start-up, the pump action is delayed until the first part of the heat exchanger is heated up (Weisshaupt)

The general idea is to either reduce heat transfer in critical zones of the HE, or the re-heat the combustion gasses after its last heat transfer to the boiler water. Because condensation could occur in certain circumstances (e.g. start-up), the HE-materials must be resistant to condensate. Traditional untreated cast iron can not be applied for LT-heat exchangers.

Undoubtedly the original idea was that these more efficient LT-appliances could simply replace the old standard boilers without the need for adjusting the chimney. Today it is often advised to seriously consider the renovation of the chimney when a LT-boiler is installed. If adjustment of the chimney proofs to be necessary, a condensing boiler with plastic flue-ducts could be the more economic option.

At nominal load and in real-life, the LT boiler might have 1-2% lower flue gas losses than an atmospheric boiler because it does not have the temperature limitations.

Nominal standby losses in real-life are not significantly lower –depending on the design— because inside the boiler body the temperature level is comparable to that of atmospheric boilers. Nominal standby losses ('burner off') according to the EN test standard are of course lower, because the test standard measures them at a 10°C lower temperature. The main advantages of a low-temperature boiler over an atmospheric boiler are therefore not so much at nominal load, but at part load. In part load, a weather-controlled LT boiler can modulate with low flue gas temperatures and boiler temperatures, whereas an atmospheric boiler could not accommodate this type of control. For this reason the LT boiler efficiency is some 4-6% better than that of a standard boiler at 30% part load.

Another advantage of low-temperature boilers is perhaps the fact that the flue gases are hot enough to help in preheating the air/fuel through a consecutive gas/air heat exchanger (see tertiary heat exchanger), which is especially advantageous for oil-fired boilers (helps in vaporizing).

In any case, after passing through the primary heat exchanger, the flue gas temperatures are in the range of 130-140 (e.g. 'low temperature' and 'condensing boiler') to 180°C (modern atmospheric boiler, older models up to 250°C). Compared to the original 1730°C and the ambient temperature this means that around 6-11% of flue gas energy (excl. latent heat) exits the primary heat exchanger. In case of an atmospheric boiler or low-temperature boiler this is also known as the final flue gas losses.

A relevant concept in this context is the **combustion efficiency η_f** (D. *feuerungstechnischer Wirkungsgrad*), which measures the temperature of the flue gasses and then calculates the sensible heat loss, i.e. without taking into account the latent heat

$$\eta_f = 100\% - Q_f / H_i \quad (in\ \%)$$

where

Q_f = sensible heat of flue gases [kW] (product of mass flow, specific heat cp and ΔT of the combustion products).

H_i = Heat flow of combustion related to the lower combustion value NCV= (methane 35,89 MJ/m³). As a rule of thumb: 10°C decrease in flue gas temperature represents approximately 0,5% decrease in flue gas losses.

In countries like Germany there is specific legislation regarding the flue gas losses, saying that they should be not higher than 11%, when compared to the net calorific value (NCV).

The Boiler Cycling method gives the following default values for the flue gas losses:

Table 2-9. Default flue gas losses of the boiler on-mode as a percentage of nominal power under test conditions (P'ch,on) at typical boiler test temperatures (prEN 15316-4-1, table C1)

Description	θgn,test [°C]	P'ch,on [%]
Atmospheric boiler	70	12
Force draught gas boiler	70	10
Oil boiler	70	11
Condensing boiler (acc. BED)	50	6

Please note that the losses of condensation heat are not included here. Those will be discussed in the paragraph on the secondary heat exchanger.

As the average EU-boiler is moving towards an efficiency on NVC of 90% (82% on GCV), we can take this as a reference for boilers with only primary heat exchangers, meaning flue gas losses of around 6 - 7% (at an average boiler temperature of 50°C and flue gas temperatures of around 150°C).

(To calculate the losses in specific real life situations, corrections on test figures will be necessary to compensate for the differences between the test- and the actual boiler water temperature and cycling behaviour.)

2.6.3 Losses through the generator envelope in on-mode

During operation the heat exchanger will transmit heat directly to the casing and the air between heat exchanger and casing. In case of a type C boiler (closed system) and an open combustion air fan (D. *'Luftumspült'*), the heated air from the heat exchanger that ends up in the envelope will be picked up by the fan and the heat is recovered.

The heat that is transmitted to the envelope itself (mounting frame and casing) is not recovered for the heat transfer to the hydronic system. This heat is mainly lost through radiation and to a smaller extend through convection round the envelope and through conduction (e.g. through wall).

Heat losses through the boilers envelope in on-mode can be determined as the difference between the combustion efficiency and the net efficiency of the boiler and can be indicated as a percentage of the input power.

These heat losses through the boiler envelope in burner on mode depend on:

- combustion temperatures (type of burner)
- heat-exchanger/burner configuration
- boiler water temperature
- insulation, material and finishing of boiler envelope

The *Boiler Cycling method* gives default values for these 'envelope-losses' at test conditions with the formula [15]:

$$P'gn,env = A + B * \log Pn$$

[15] Pn is the nominal boiler power in kW. Note that for Pn=20 kW, log Pn= 1,3. At 30 kW, log Pn=1,5

A and B are appliance specific parameters, but the following default values are given:

Table 2-10. Value of parameters A and B (heat loss through envelope parameters) [prEN 15316-4-1, table C3]

Generator insulation type	A [-]	B [-]
Well insulated, high efficiency new generator	1,72	0,44
Well insulated and maintained	3,45	0,88
Old generator with average insulation	6,9	1,76
Old generator, poor insulation	8,36	2,2
No insulation	10,35	2,64

Whether the envelope losses are considered as 'recoverable' will depend on the position of the boiler. For instance, the *Boiler Cycling method* considers 90% of the radiation losses as useful if a type C (closed system) boiler is in the heated space. See table below.

Table 2-11. Default values of factor kgn,env (reduction factor for recovery of heat losses of envelope) [prEN 15316-4-1, table C4]

Generator type and location	$k_{gn,env}$ [-]
Generator installed within the heated space	0,1
Atmospheric generator installed within the heated space	0,2
Generator installed within a boiler room	0,7
Generator installed outdoors	1

Based on this *Boiler Cycling method* approach and also on the values mentioned in DIN 4702-1, default values for the envelope losses (under test conditions) in the on-mode can be varying from around 2% for the well insulated new appliances to over 14% for old not insulated generators. To calculate the losses in specific real life situations, corrections will be necessary to compensate for the differences between the test- and the actual boiler room and boiler water temperature. Please not that the room temperature for type B boilers will generally be lower because of the mandatory ventilation provisions.

The average figure for a new well insulated condensing boiler (at average water temperature of 70°C) is estimated at 2% and for the average new boiler at 4% (if no envelope losses are recovered). Half of this was already attributed to the burner, which is much smaller than the heat exchanger, but also much warmer. The other half we will attribute to the heat exchanger.

For an atmospheric standard boiler with poor insulation these envelope losses are around 10%.

2.6.4 Standing losses in off-mode

When the burner is switched off, the heat generator still loses heat through radiation, convection and conduction. The convection through the *chimney* attributes largely to these standing losses (most boilers have no flue-valve installed). The other part consist of the radiation, convection and conduction losses of the *boiler envelope*. These standing losses through the boiler envelope and chimney in burner off-mode depend on:

- average boiler water temperature
- average water flow
- use of a flue valve
- insulation, material and finishing of boiler envelope
- operating time of the pump (continuously running or switched off after each burning cycle)
- use of pilot flame (not very common any more)

If the pump is not continuously running, but switched off after each burning cycle we have additional parameters that influence the standing losses:

- heat capacity of the generator
- operating time of the pump after burner switch off
- boiler operating periods over the day

Pump continuously running

The standing losses with a primary pump continuously running are measured in the EN 303 standards by using an electric heater in the CH-boiler loop to keep the temperature at a pre-set level (30°C ± 50 above ambient) and are expressed in [kW]. For installations with the pump continuously running, this test figure can be used to calculate the total standing losses in real life, by correcting for the actual average boiler water temperature and actual boiler room temperature.

The *'Case specific boiler efficiency method'* of the prEN 15316-4-1:2005 proposes the following formula for correction (formula nr. 8):

$$\Phi_{gn,l,Po,corr} = \Phi_{ge,l,Po}\ [\ (\ T_{gn.w}\ -\ T_{i,gn}\ )\ /\ 30\ ]^{\ 1,25} \qquad [W]$$

In which:

$\Phi_{ge,l,Po}$ = standby losses according EN 303

$T_{gn.w}$ = actual average boiler water temperature

$T_{i,gn}$ = actual boiler room temperature

The method also gives the following default values for $\Phi_{ge,l,Po}$ in annex B, Table B1.2 in case the certified test figures for standing losses are not available:

Default stand-by heat losses can be calculated with:

$$\Phi_{gn,l,Po}\ =\ \Phi_{Pn}\ *\ (\ E\ +\ F\ \bullet\ log\ \Phi_{Pn}\ ) \qquad [W]\ \ (formula\ B3)$$

With values of E and F given in the following table.

Table 2-12. Parameters for calculation of stand-by heat losses. [prEN 15316-4-1, table B2]

Generator type	E	F
Standard boiler	25	-8
LT boiler	17,5	-5,5
Condensing boiler	17,5	-5,5

A 24 kW condensing or LT-boiler would have default standing losses of 238 watts; a standard boiler would have 335 watts. To calculate the total real life standing losses per year we would need to correct for the actual average boiler temperature and actual boiler room temperature and then multiply this figure with the operating time of the pump (while burner is off). If average boiler- and room- temperature are identical to test conditions (no corrections necessary) and the additional operating time of the pump is 2/3 of the heating period of 5200 hours, the yearly default standing losses for a condensing 24 kW boiler are:

238 [W] x 2/3 x 5200 [h] x 3600 [s] = 2970 [MJ] or 825 [kWh] (partly recoverable when the boiler is installed in a heated space).

The *'Boiler cycling method'* calculates the *chimney* losses separately from the *envelope* losses in the burner-off mode. If they are not declared by the manufacturer, default values can be used according to annex C table C.6. The default values mentioned in this table are expressed as % of the nominal boiler load. For a 24 kW boiler with premix burner the default value is 0,2% of 24 = 48 watts. A Wall mounted gas fired boiler (24 kW) with fan and wall flue gas exhaust would have 0,4% of 24 = 96 watts.

Atmospheric boilers with long chimneys (>10 m) could go as high as *1,6% x 24* = 384 watts.

According to the '*Boiler cycling method*', the standing losses through the boiler *envelope* in burner off-mode are the same as in boiler on-mode. As explained in the previous paragraph, the average figure for envelope losses for a new well insulated condensing boiler according to this method is estimated at 2% (480 watts for a 24 kW boiler at 70°C). At an average boiler water temperature of 50°C and a boiler room temperature of 20°C, this 2% can be corrected with the factor *[(50–20) /(70–20)] = 30/50 = 0,6*. Envelope losses will in this case be 1,2% of 24 kW or 288 watts. If we assume the boiler-off period 2/3 of the total heating period of 5200 hours, the yearly default envelope losses for a 24 kW condensing boiler would be: 2/3 x 5200 [h] x 3600 [s] x 288 [W] = 3594 [MJ] or 998 [kWh] (partly recoverable, depending on location of boiler)

A more hands-on approach for the assessment of the standing losses through boiler envelope would be to calculate the radiation and convection losses on the bases of rule of thumb formulas or to compare it with known data from comparable appliances.

Standing losses of electric storage heaters (kept at 60°C) with a volume that is comparable to that of a 24 kW wall hung or standing condensing boiler range from 65 [W] for the best appliance to 123 [W] for the worst appliance (source: Save water heaters, Task 2. Technical Analysis). Of course boilers are not continuously kept at 60°C and the insulation quality differs a lot (water heaters are generally a lot better insulated than boilers), but the figures give some indication on the order of magnitude.

A rule of thumb formula for the calculation of radiation losses is:

$$q_{rad} = A_{env} * \varepsilon_{env} * \sigma (T_{env.}^{4} - T_{blr.}^{4})$$

in which

$A_{env.}$	= Surface of the envelope (appr. 2 m^2 for a wall-hung condensing boiler)
ε_{env}	= Emission factor enevelope (between 0,1 en 0,9, depending on material and finishing)
σ	= Radiation constant of Stefan-Boltzmann (5,67 * 10-8 [W/m^2K^4])
T_{env}	= Average temperature of boiler enevelope (in degrees Kelvin)
$T_{blr.}$	= Average temperature boiler room (in degrees Kelvin)

With a boiler room temperature of around 15°C, a surface temperature of the envelope of 30°C, and an emission factor of 0,9 (white painted steel plate) the radiation according to this formula would be: 158 watts. A 40°C surface temperature of the envelope would give radiation losses of 277 watts.

For heat dissipation through natural convection of the boiler envelope the following rule of thumb formula can be used (formula of Nusselt):

$$q_{conv} = 2,6 * A_{env} * (T_{env.} - T_{blr.})^{1,25}$$

Using the same temperature values and a convecting surface of 1,5 m^2, the calculated convection losses of a boiler envelope are approximately 115 watts. A 40°C surface temperature would result in 218 watts of convection losses.

If the conduction losses (e.g. through the wall) are neglected, the calculated total envelope losses (boiler room temperature = 15°C and surface temperature = 30°C) would add up to 273 watts.

Pump switches off 10 minutes after each burning cycle

In principle standing losses will be lower in case the pump switches off (some minutes) after each burning cycle[16], because the appliance is not kept continuously at a certain boiler water temperature, but is allowed to cool down.

In this case the heat capacity of the generator determines how much energy (heat) can be stored, and with that also how much heat can be lost. Depending on the mass of the appliance (mainly heat exchanger and water content) a boiler can easily contain 2 to over 4 MJ of heat (40 resp. 80 kg).

The operating time of the pump after burner switch-off achieves that some of the stored heat in the appliance (heat exchanger) is transferred to the hydronic system.

The number of operating periods per day and the time between operating periods indirectly determine the number of complete cool-downs of the appliance.

During an operating period the radiation and convection losses depend on the average appliance temperature.

The use of a flue valve (valve that switches off the flue duct after each cycle) and the use of insulation for the generators envelope will reduce the radiation and convection losses.

Example

If we assume that a 50 kg generator will operate for 2 hours during the morning, 1 hour in the afternoon and 6 hours in the evening, the total number of hours with a more or less constant radiation convection loss is 9; the number of total appliance cool-downs is 3.

The annual energy loss due to 3 complete cool-downs per day during the heating season can roughly be calculated with the following formula:

$$Q_{rad\&conv;\,p.sw} = a * d_h * c_{av} * m * \Delta T_{appl;avg}$$

In which:

$Q_{rad\&conv;\,p.sw}$: Energy losses through radiation & convection of boiler during pump off –period in [J]

a : Average number of complete cool downs per day (3)

d_h : Number of heating days per year (220 dagen)

c_{av} : Average specific heat of generator (800 [J/(kgK)])

m : mass of appliance (50 [kg])

$\Delta T_{appl;avg}$: average temperature difference between start and end of cooldown period (40 [ºC])

Filling in average values gives:

$Q_{rad\&conv;\,p.sw} = 3 * 220 * 800 * 50 * 40 = 1056$ [MJ]

The *Boiler cycling method* gives a correction on the envelope losses and the chimney losses in burner off mode, for situations in which the pump is switched off.

This correction factor can be calculated, depending on the load factor FC (which is the quotient of the burner-on time and total boiler stand-by time) and an exponent m, that depends on the type of boiler.

For a wall mounted boiler exponent m = 0,5; for a steel boiler m = 0,4 and for a cast-iron boiler m= 0,3 (see Annex C table C.5 of prEN 15316-4-1).

The correction factor for a wall hung boiler that operates (= burner on) 1/3 of the total time is $0{,}33^{0{,}5} = 0{,}57$. If the envelope losses are 1% of nominal power when the pump is continuously running, in a situation were the pump is switched off, the losses are *1 x 0,57* = 0,57% of nominal power. For a 24 kW boiler this is 137 [W].

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[16] The Danish Technology Institute DTI reports that usually 2 to 3 minutes after run is a normal value

If we assume the boiler-off period 2/3 of the total heating period of 5200 hours, the yearly default envelope losses would be: *2/3 x 5200 [h] x 3600 [s] x 137 [W]* = 1707 [MJ] or 474 [kWh] (partly recoverable, depending on location of boiler).

Summarizing the above we can say that standing losses for new condensing and LT boilers may vary from 100 watt for a good insulated appliance with flue valve, to around 300 watts for a poor insulated boiler without flue valve. Standard atmospheric boilers may have 1,5 to 3 times these losses.

If we assume a boiler running time of 1/3, the boiler stand by time is 2/3 resulting in annual standing losses ranging from 1248 [MJ] (347 kWh) to 3745 [MJ] (1040 kWh). For an average house with a nominal heat load in 2005 of 7250 kWh (see stock-model report Save Heating Systems) this represents around 3,4 to max. 10% of the annual energy consumption.

Standing losses will increase as the overall standby period is longer. Losses also increase with higher boiler water temperatures. If the pump switches off after each burning cycle, the losses can be reduced with 50% or more, mainly depending on heat capacity of the boiler.

Data from real life measurements can be taken from the *Wolfenbüttel study*. In their final report[17] the *Fachhochschule Braunschweig Wolfenbüttel* mentions that the average standing-losses of the 60 condensing boilers that were monitored correspond with a fraction of 0,468% of the input power of the boiler. In other words, a boiler of 24 kW would have 112 W standing losses as an average.

2.6.5 Start-stop losses

The graph below describes the energy profile during start-up and cool-down. It shows that –depending on the burner load and the heat capacity of the boiler— it takes some time before the boiler system has reached a steady state situation. During this start-up time, as mentioned earlier, there are the most emissions of fuel and other emissions causing a fuel loss that –with 40 000 cycles/year—result in some 1,5% of fuel loss [18]. During this time the boiler appliance is heated-up until thermal equilibrium is reached, and from that moment on the steady state efficiency (acc. EN 303) applies.

The graph also shows the so-called purge losses, which come from fan action during 'burner-off', which we will discuss hereafter.

Another issue that needs to be addressed here is the fact that energy is lost when the boiler starts cycling. This cycling occurs when the supplied heat is higher than the boiler water can dissipate and the boiler is switched off by the boiler thermostat shortly after burner start. Steady-state efficiencies (acc. to NEN 303) are not achieved in those situations. Losses that are related to this phenomena will be further explained.

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[17] Fachhochschule Braunschweig Wolfenbüttel, Felduntersuchung: Betriebsverhalten von Heizungsanlagen mit Gas-Brennwertkesseln, April 2004

[18] Please note that "fuel loss" does not equal methane (CH4) emissions and also note that 40.000 cycles per year is a maximum and not an average. To calculate "fuel loss" we took into account the mass balance of all emissions of carbon-compounds (CO, CH4, TOC) as found by Pfeiffer in par. 2.3.4. Marcogaz protests strongly against this value and claims that CH4 emissions from a Ruhrgas/CGB study shows values that are a factor 10 lower. We see no contradiction here, especially if the CGB tests were performed at steady state efficiency or with (Danish) boilers with a high primary store.

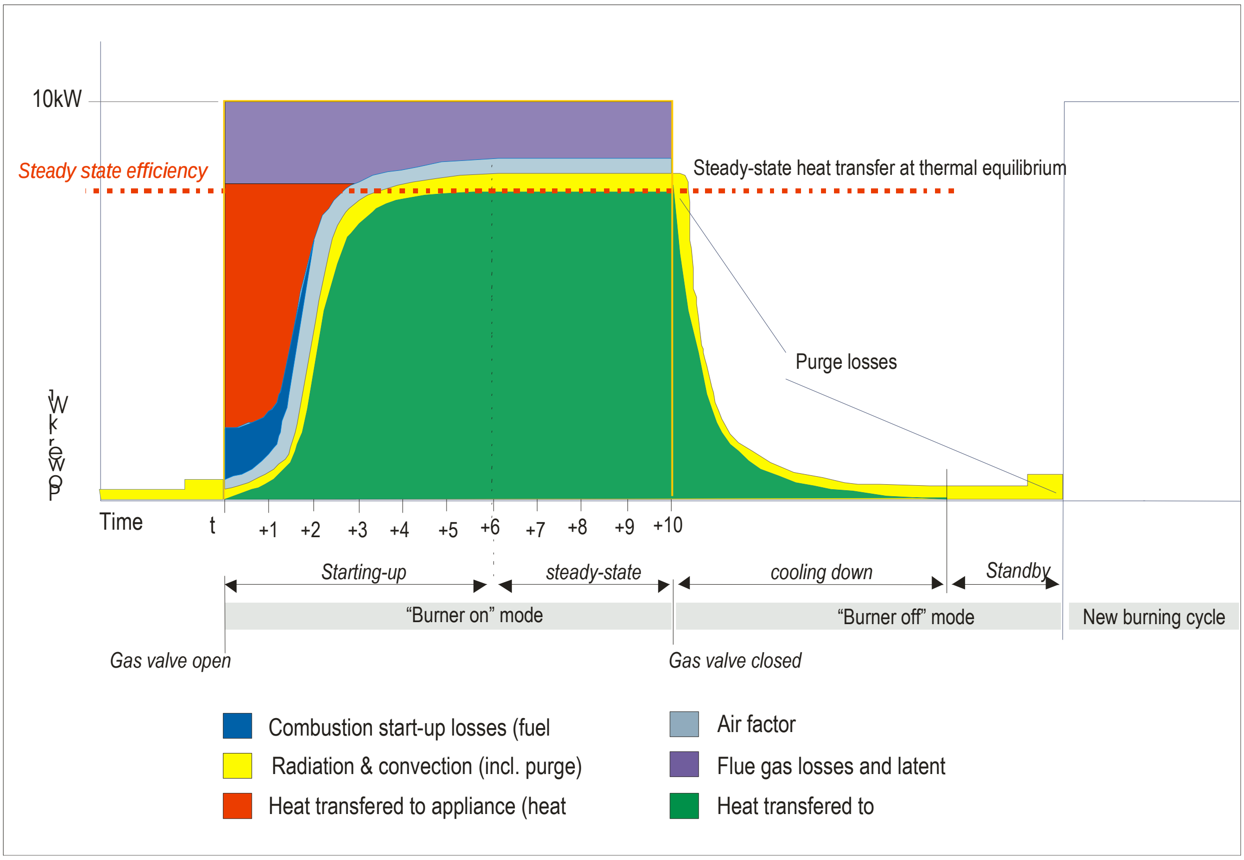


Figure 2-15. Burning cycle and energy losses of boiler

Pre-purge

For safety reasons the combustion chambers of type C boilers need to be purged before each burning cycle. This pre-purging implicates that cold (ambient) air is blown through the combustion chamber and heat exchanger and because of that heat is lost.

According to prEN 13836 a pre-purge period of 30 seconds with an airflow that corresponds to nominal boiler load would comply.

With the following formula a rough calculation can be made off the energy losses related to these purge cycles:

$$Q_{loss;\ purge} = t_{purge;bf} * \varphi_{fan} * \rho_{air} * c_{air} * \Delta T_{air;avg}$$

In which:

- $Q_{loss;\ purge}$:Energy losses per burning cycle caused by pre-and after purge of appliance
- $t_{purge;bf}$: pre-purge time in [s]
- φ_{fan} : air flow in [m³/s]
- ρ_{air} : density of air [kg/m³]
- c_{air} : specific heat of air [J/(kgK)]
- $\Delta T_{air;avg}$: average temperature difference of the purge air [s] before and after passing the appliance

If we assume a pre-purge time of 30 seconds, an after purge time of 10 seconds, an air flow of 24 m³/h (6,7 liters per second, e.g. for a 20 kW boiler) and an average temperature difference of the purge air of 30 °C we can make an indicative calculation:

$$Q_{loss;\ purge} = 30 * 0{,}0067 * 1{,}2 * 1000 * 30 = 7{,}2 \text{ [kJ]}$$

A generator with 40000 starts would loose 288 MJ-year (80 kWh). For an average dwelling with a primary energy consumption for space-heating of around 10500 kWh per year, these burner start losses would represent less than 1%.

After purge

At the end of a cycle the EN standards also prescribe an after-purge of around 10 seconds. The reason for this after-purge is safety, e.g. removing fuel from the combustion chamber. However, up to a certain degree where the flue gases are warmer than the boiler water, the after purge is also beneficial to transfer the residual heat of the burner and heat exchanger body to the boiler water. With the burner it was already calculated that this contributed up to 4%. Also with the heat exchanger, typically containing 3-5 litres of hot flue gases and with a heat exchanger surface considerably warmer than the boiler water at the time of shutting down the burner, there may be an extra gain from the after-purge. All in all, we will not consider the heat transfer of 10 s. after-purge as losses, provided of course —as with the residual burner heat— that the extra contribution of the after purge is taken into account in the boiler control.

Cycling losses

A boiler starts cycling when the energy input is too high for the heat output realized by boiler water flow. These situations especially occur when the water content of the boiler is small, the minimal load of the boiler is too high and the heat demand from installation side is low.

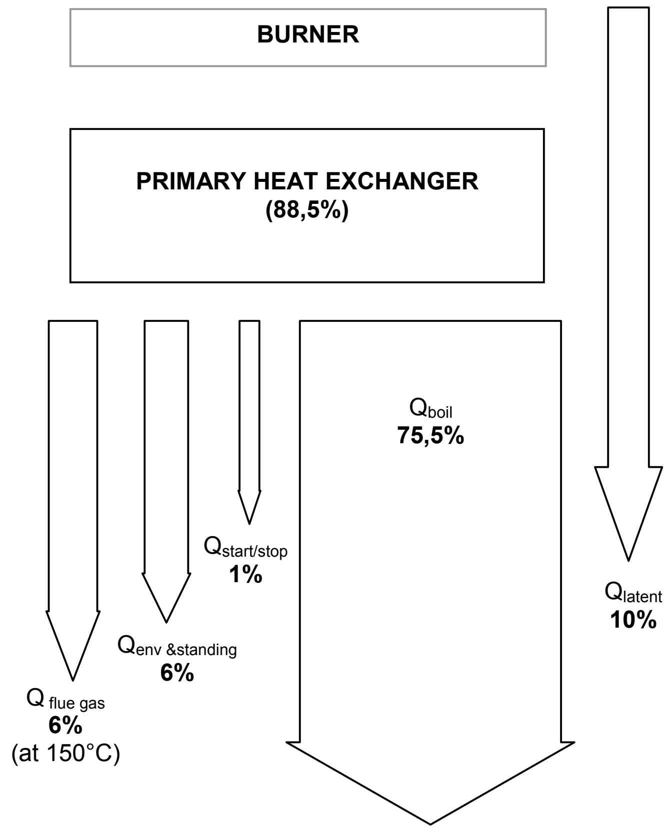
In other words, the average 24 kW wall hung boiler with a turn down ratio of 1 : 3 and a standard pump, connected to an installation with 6 – 8 radiators with thermostatic valves.

When several thermostatic radiator valves close, the heat demand decreases and the resistance in the heating system increases, causing a higher flow speed of the boiler water. As a result the boiler return temperature will be higher. With overflow valves installed in the water circuit, this effect will be even higher. The consequence of this all is that with condensing boilers, efficiencies at part load will be lower than measured according to test standards (EN 303), because higher return temperatures increase the flue gas losses and reduce the amount of condensation. Depending on return temperature increase, part load efficiencies may drop from 1 to 5%

2.6.6 Primary heat exchanger: Flow diagram

The picture represents an energy flow diagram of the primary heat exchanger. The diagram does not make a distinction between 'burner off' or 'burner on' energy transfer, but sums the flows on an average annual basis.

Figure 2-16.
Simplified energy balance of primary heat exchanger



2.7 Heat balance secondary and tertiary heat exchanger

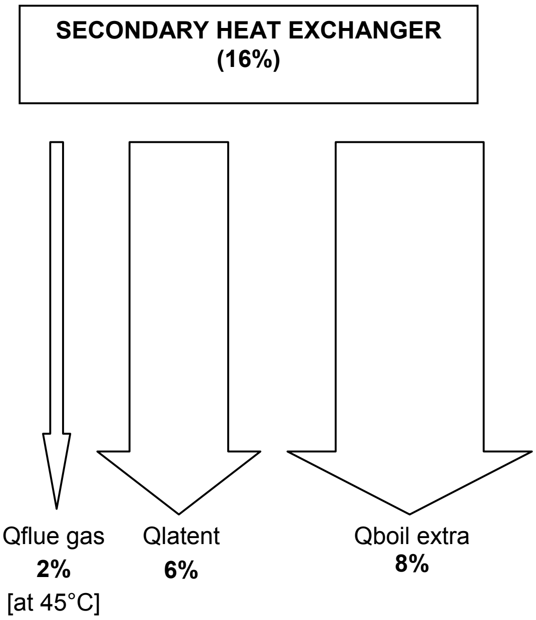
2.7.1 Secondary heat exchanger

In the case of condensing boilers there is also a secondary heat exchanger. In reality, in the case of integrated boilers with a flat burner this is little more than an extension of the primary heat exchangers. In the case of a cylindrical burner and a spiral-tube heat exchanger (round or oval) this may really be a secondary spiral. Or in the case of a jet burner with a plate heat exchanger it may be a second plate heat exchanger. In most cases this secondary heat exchanger is a flue-gas / boiler-water heat exchanger; in some cases (some oil boilers) this can also be a flue-gas / combustions-air heat exchanger in which case it is always a separate (plate) heat exchanger.

In any case, the function of the secondary heat exchanger is to further cool the flue gases to a temperature level where most of the latent heat can be recuperated, alongside of course the remaining sensible heat in the flue gases. The EN standard and the BED foresee that this happens at a boiler return temperature of 30°C, resulting also in flue gases of the same temperature level. If that happens, some 90% of the remaining flue gas losses and of the latent heat can be recovered. In reality, the average return boiler temperature with weather controlled systems is closer to 45°C because installers have a tendency adjust heat curves to prevent complaints, which means that in practice only 50% is recuperated.

The energy flow diagram of the secondary heat exchanger, neglecting losses to the casing, will look like the picture below.

Figure 2-17.
Simplified energy balance of primary heat exchanger



With lower average boiler water temperatures (around 40°C) and longer operating periods, the standing losses in off-mode will also decrease. An additional 2% can be gained compared to the values mentioned in the energy flow diagram of the primary heat exchanger

In case the secondary heat exchanger is a gas / water HE the amount of latent- and flue gas heat that can be regained strongly depends on the return water temperature. If the installation and the control systems do not facilitate low return water temperatures this energy can not fully be regained.

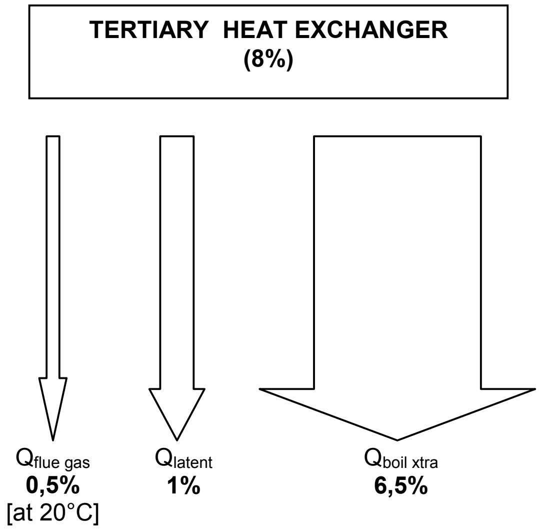
Please note that if the heat exchanging process stops here, the boiler efficiency on GCV is *75,5* + *8* + *2* = 85,5%, which is in line with the results from the Wolfenbüttel study for condensing boilers.

2.7.2 Tertiary heat exchanger

The tertiary heat exchanger is a flue gas to combustion air heat exchanger. This heat exchange can take place in the concentric flue/air duct or in a separate plate heat exchanger. For oil boilers this pre-heating is also functional at higher flue gas temperatures to preheat the incoming air in order to consequently promote the oil vaporisation process. In gas-fired boilers the heat exchange already takes place (to a small extend) through the concentric flue/air tubes, but until now a dedicated counter flow (or cross flow) flue-to-air heat exchanger was not used.

In any case, an effective counter flow tertiary heat exchanger (η=80-90%) allows to recuperate also the last bit of latent heat and sensible flue gas losses. We will discuss this further in Task 6 with the design options. For now, we will just present the flow-diagram.

Figure 2-18.
Simplified energy balance of tertairy heat exchanger



With this tertiary heat exchanger and an additional reduction of the standing losses with 1%, the total real life efficiency for an average house with a heat load of 7250 kWh in this case would be around 93% on GCV. Losses would be:

- fuel losses: 1,5%
- flue gas losses: 0,5%
- latent heat: 1%
- start/stop losses: 1%
- envelope and standing losses: 3%

Whether these envelope & standing losses should be counted as irrecoverable or not, depends on whether the boiler is in the heated space. If the boiler has a closed flue/air system (Type C), has the right dimensions and answers noise requirements, this type of credit could be appropriate. Furthermore, standing losses in off-mode can be further reduced by prolonging the operation periods and switching off the pump after burner cycles. At the same time the fuel losses are reduced to <0,5% and start/stop will be lower (< 0,5%) losses because of the fewer burning cycles.

In any case, even without giving the credit for casing losses to the heated space, the total boiler heating efficiency on GCV could be as high as 96-97% (105-106% on NCV). This is of course without taking into account the auxiliary electrical energy for pump, fan, controls, etc..

2.8 Heat balance with storage facilities

In the previous section we have assumed that the boiler can always follow the heat demand, when it is needed and at the capacity that is needed. In practice, this is not always possible and there is a mismatch between the supply and the demand side. This can lead to unnecessary cycling, causing unnecessary wear of components, additional noise and additional cycling losses (see paragraph 2.6.5).

For a boiler that is only used for space heating, the solution could be to use a primary buffer. With proper appliance insulation, cycling losses can be reduced without increasing the envelope and standing losses too much. This solution will function adequately with weather controlled systems, but will give complaints when an on/off room thermostat is used. Response times are to long.

Combi boilers are designed for direct hot water delivery. Starting up the burner for each individual hot water draw is not an option, because it takes to long before hot water is delivered (purge times etc.)

and because this would cause an additional 14000 cycles per year.

To solve these problems a buffer is used, usually in the form of a storage vessel for sanitary hot water, CH- water or both. In fact, the primary and/or secondary heat exchanger may already be such a storage vessel. In the Task 1 report most of the currently known configurations with a storage facility are listed and we will not repeat this here.

However, some practical standing losses are given, to show the penalty of using storage vessels:

- 4 litre combi store: 15-20 W (insulation 30mm, 80-240 kWh/year, 1-2,5% efficiency loss).
- 80 litre at 65°C (>100 mm insulation): 55-60 W (500 kWh/year = 5% efficiency loss/ year).
- 150 litre (120mm insulation): 65-70W (600 kWh/year, 6% efficiency loss)
- 350 litre solar (110mm insulation): 100 W, 870 kWh/year (8-9% efficiency loss).

Please note that these values are already much lower (ca. factor 3) than the maximum values suggested by e.g. EN 303-6.

2.9 Auxiliary energy

Boilers use electrical components for their operation. Practically all premix modulating boilers use an electronic control unit, a pump, a fan and electrically powered gas valves.

Oil boilers use in addition to the above, electricity for preheating the oil and an oil pump for pressurizing or atomizing the fuel.

Gas and oil igniters also use electricity but only for a short period (10 – 35 seconds). The electricity consumption related to this will be neglected.

Table 2-13 gives an overview of the typical power consumption for the various components from the Boiler Savelec study. Please note that these values may be subject to change later in the underlying, e.g. following the preparatory study on the CH circulators.

Table 2-13. Auxiliary energy consumption (electrical) [Source: Boiler Savelec Study, WP3]

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

The average electricity consumption for gas boilers over the year (space heating purposes only) may vary from approximately 150 kWh_{el} for boilers in which the pump switches off after each burning cycle or with highly efficient modulating pumps, to around 450 kWh_{el}/yr for boilers with a single speed pump continuously running.

For oil boilers the yearly electricity consumption is estimated around a factor 2 higher.

For the average new gas-fired boiler we estimate a consumption of 250 kWh_{el}/year. Using a conversion factor of 2,5 from electricity to primary energy, this comes down to around 625 $kWh_{primary}$/year. Compared to an average heat consumption of 7250 kWh primary energy, this adds around 8 to 9%. Or rather, to find the total primary efficiency of the boiler we have to multiply the strict heating efficiency by around 0,92-0,93.

A part of this electrical energy is transformed to thermal energy and to a certain extend transferred to the hydronic system.

The *Boiler Cycling Method* gives default values for the average electricity consumption and for the fraction of energy that is regained. According to this method, the average electricity consumption for a modulating 24 kW gas boiler is 48 watts, resulting in 250 $kWh_{electric}$ per annum, of which 80% (200 $kWh_{electric}$) can be recovered.

2.10 Total energy balance

Based on the previous paragraphs, the table below and the figures on the following pages give an illustration of the total energy balance for some characteristic boiler types in an average house.

Table 2-14.
Indication of characteristic generator heat losses for 4 boiler types and 2 control types (space heating only) [VHK estimate]

For heat load average house: 7250 kWh.	**Fuel Input on GCV**	Additional Electric Energy [kWh el.]	Conversion factor	**Add. Electr. Energy [MJ pr as % van total]**	includes Latent Heat	**Flue gas losses in "burner-on mode"**	**Rad./conv./cond. losses "burner-on mode"**	**Rad./conv./cond. losses "burner-off mode"**	**Start/stop losses (purge- and cycling losses)**	**Unrecovered latent heat**	**Fuel loss**	**Electric heat transferred to hydronic system**	Ann. Eff. Generator on GCV of fuel inpout	Ann. Eff. Generator on GCV of fuel & electric input
	%	kWh	_	%	%	1	2	3	4	5	6	7	%	%
	energy input					energy losses						energy gain	efficiency	
GAS BOILERS														
Weather controlled [5]														
Standard boiler, 24 Kw [1]	100	200	2,5	4,9	10	10,0	8,0	10,0	0,5	10,0	1,5	1,58	**62%**	**59%**
LT boiler, 24 Kw [2]	100	250	2,5	6,2	10	7,0	3,0	6,0	0,5	10,0	1,0	1,98	**74%**	**70%**
Condensing boiler 24 kW [3]	100	250	2,5	6,2	10	2,0	2,0	3,0	5,0	5,0	1,0	1,98	**84%**	**79%**
Condensing boiler 10 Kw [4]	100	150	2,5	3,7	10	2,0	2,0	1,5	0,5	4,0	0,5	1,19	**91%**	**87%**
On/off thermostat [6]														
Standard boiler, 24 Kw [1]	100	100	2,5	2,5	10	10,0	8,0	6,0	0,5	10,0	1,5	0,79	**65%**	**63%**
LT boiler, 24 Kw [2]	100	150	2,5	3,7	10	7,0	4,0	4,0	0,5	10,0	1,0	1,19	**75%**	**72%**
Condensing boiler 24 kW [3]	100	150	2,5	3,7	10	3,0	3,0	2,0	5,0	9,0	1,0	1,19	**78%**	**75%**
Condensing boiler 10 Kw [4]	100	100	2,5	2,5	10	3,0	2,0	1,0	2,0	8,0	0,5	0,79	**84%**	**82%**

[1] Standard atmospheric boiler (24 kW, high water content, fixed feed temp, mixing valve)
[2] Low Temperature boiler (24 kW, premix, high water content, single speed pump)
[3] Condensing boiler (24 kW, premix, mod 1:3, single speed pump)
[4] Cond. boiler (10 kW, premix, mod 1:10, VS-pump, flue valve, excellent insulation)
[5] Weather controlled heating system (av. T = 40°C)
[6] Room thermostat (on/off) controlled heating system

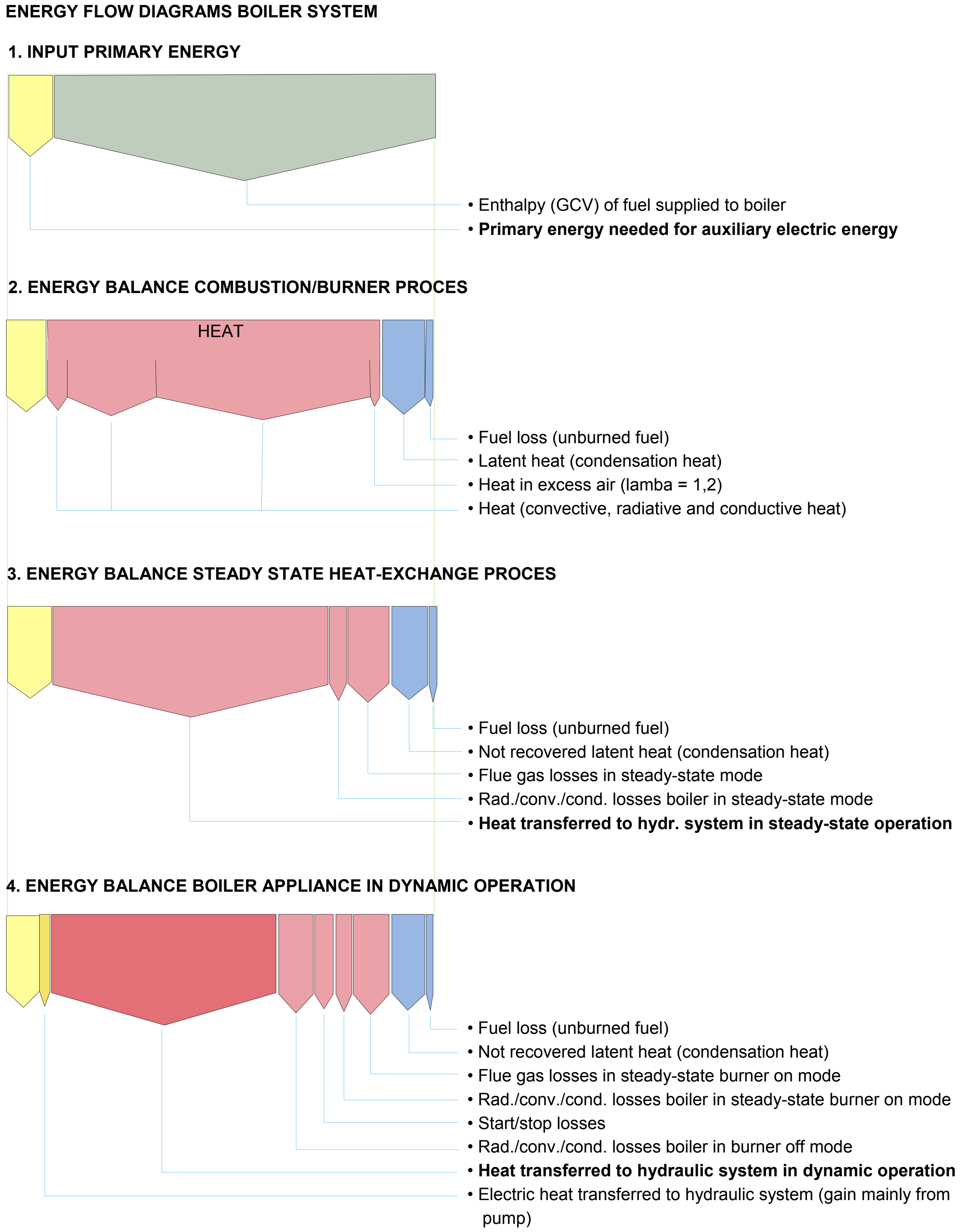


Figure 2-19. Energy flow diagrams of boiler systems

3 EMISSIONS

3.1 Introduction

Emissions of air pollutants from the combustion process in gas- and oil-fired CH boilers are carbon dioxide (CO_2), nitrogen oxides (NO_x), carbon monoxide (CO) and methane (CH4). In oil-fired boilers you have these emissions plus sulphur oxides (SO_x), Volatile Organic Compounds (C_xH_y) and “soot” (Particulate Matter, PM).

In the MEEUP methodology study (VHK 2005) ‘default values’ for the emissions per GJ of heat output were presented for a number of heat generators. An extract (excluding water and waste) is given in the table below.

Table 3-1. Use phase: Energy and emissions per GJ heat out CH boiler, (excl. Electricity for fossil fuel based heating) [VHK 2006, based on Öko-institut GEWIS database]

	HEATING	Energy	Emissions: To Air						To Water	
		primary	GWP	AP	VOC	POP	PAH & HM	PM	HM	EUP
nr.		MJ	kg	g	mg	i-Teq	mg	g	mg	g
66	Electric, η 96%, per GJ	3045	132,9	784	1147	20	180	17	6	0,1
68	Gas, η 86%, atmospheric	1163	64,3	19	846	0	0	0	0	0
69	Gas, η 90%, atmosph.	1111	61,4	18	809	0	0	0	0	0
70	Gas, η 101%, condens.	990	54,7	16	721	0	0	0	0	0
71	Gas, η 103%, condens.	971	53,7	16	706	0	0	0	0	0
72	Oil, η 85%, atmosph.	1176	87,8	110	1519	0	0	2	0	0
73	Oil, η 95%, condens.	1053	78,5	98	1360	0	0	2	0	0
78	Extra for fossil fuel extraction & transport: Gas +7% (row 68-73), Oil +10% (row 72-73), for Wood pellets and logs add 5% of row 72									

Please note that all efficiency values are given in NCV, , whereas we now propose GCV. On GCV the efficiency values are around 10-11% lower for gas and 6% for oil.

The tables are from GEMIS 4.2. More recent information on emissions from oil- and gas fired appliances can be found in Annex D where Eurofuel reports on the results of the GEMIS 4.3 software package. Emission values for GWP and AP from version 4.3 are considerably more favourable for oil than in the GEMIS version 4.2.

VHK has taken this into account in its final recommendations (Task 7), but for the underlying study we used the MEEuP values, as of contract.

Data for fossil-fuel fired boilers were taken from GEMIS 4.2 for fossil fuel powered 10 kW Central heating (CH) boilers in GJ heat produced at the boiler exit in the form of hot CH water. They do not include the auxiliary electricity consumption for pump, fan and controls. The table below gives some details for the specific operating conditions.

Table 3-2. Boiler operating conditions (GEWIS 4.2)

	Gas CH				Oil CH	
Row nr.	**68**	**69**	**70**	**71**	**72**	**73**
% O_2	3%				3%	
% CO_2 in flue	9,96%				13,11%	
Nm³/h flue	11,7	11,2	10,0	9,8	12,2	10,9

In this chapter we will not discuss the emissions from electricity production because their composition cannot be influenced by an (electric) boiler designer. The ‘only’ problem he or she has to face is to use the electric kWh as efficiently as possible.

First we will look at the environmental impacts of oil- and gasfired boilers following the MEEUP methodology and expanding on that. Subsequently, we will look at the emissions from the angle of their origin and some basic design measures. Next the focus is on two most interesting groups from the design point of view: The non-CO_2 hydrocarbon emissions (CO, CH_4, C_xH_y, soot) and especially the nitrogen oxides (NO_x). Finally, it is examined where the contrast and the similarities between energy-efficient and environmentally-friendly design of boilers lies.

3.2 Environmental impact

When looking at the combustion emissions from the angle of their relative environmental impact. there are a number of categories.

Global Warming Potential (GWP). These include CO_2, CO and CH_4 emissions. Legal basis is the Kyoto protocol[19] and the weighting factors for the GWP-100 are prescribed by the Intergovernmental Panel on Climate Change (IPCC). The unit of GWP-100 is CO_2-equivalent (CO_2=1). Carbon monoxide has –per weight unit— a CO_2-equivalent of 1,57. Methane (CH_4) has a significantly higher GWP at CH_4=21.

Acidification Potential (AP). These include SO_x and NO_x emissions. The policy framework for regulating acidification consists of several European Community directives and the so-called Gothenburg Protocol[20]. This protocol considers SO2 to be 50% more harmful in terms of acidifaction than NO_x (weighting factor 1 versus 0,7 respectively. This relationship is also reflected in the emission limit values of the 1999/30/EC daughter directive of the Ambient Air Quality Directive (AAQD)[21]. The AAQD is an interesting framework directive, because the collection of –so far— 4 daughter directives show the relative importance that the legislator gives to very different types of emissions, which are all assessed in a similar (grid-based) method.

From this comparison (see table 3) it is clear that the legislator thinks **NO_x some 50 times more harmful than CO-emissions** from the viewpoint of ambient air quality. This is very significant, because up till now the boiler sector has mostly treated the emission limits for CO as equivalent to NO_x (see Task 1 report). This is not in line with EU environmental policy. If the sector —and the governments in Member States— have treated CO equally stringent this must be due to other reasons, e.g. historical safety reasons when boilers were not room sealed and CO-poisoning was a real danger with open (not room-sealed) units.

Table 3-3. Target/Limit values in EC Ambient Air Quality directives (VHK, MEEUP, 2006)

Pollutant	Target/ limit values* in ng/m³	EC Air Quality directive
Benzo(a)pyrene (as a measure for polycyclic aromatics PAHs)	**1**	2004/107/EC
Cadmium (Cd)	**5**	2004/107/EC
Arsenic (As)	**6**	2004/107/EC
Nickel (Ni)	**20**	2004/107/EC
Lead (Pb)	**500**	1999/30/EC

[19] Council Decision 2002/358/CE of 25 April 2002 concerning the approval on behalf of the European Community of the Kyoto Protocol to the United Nations Framework Convention on Climate Change (UNFCC) and agreed upon by the Conference of the Parties at its third session.

[20] The United Nations Economic Commission for Europe (UNECE) Convention on Long-Range Transboundary Air Pollution (CLRTAP).

[21] Another piece of EU legislation that is relevant is the National Emissions Ceiling Directive (NECD, 2001).

Particulate Matter (PM10)**	**50 000**	1999/30/EC
Sulphur dioxide (SO_2)***	**125 000**	1999/30/EC
Nitrogen dioxide (NO_2)***	**200 000**	1999/30/EC
Ground-level ozone****	**120 000**	2002/3/EC
Benzene (aromatic HC, C_6H_6)	**5 000**	2000/69/EC
Carbon monoxide (CO)	**10 000 000**	2000/69/EC

sources:

DIRECTIVE 2004/107/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 15 December 2004 relating to arsenic, cadmium, mercury, nickel and poly-cyclic aromatic hydrocarbons in ambient air

DIRECTIVE 1999/30/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 22 April 1999 relating to limit values for sulphur dioxide, nitrogen dioxide and oxides of nitrogen, particulate matter and lead in ambient air

DIRECTIVE 2000/69/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 16 November 2000 relating to limit values for benzene and carbon monoxide in ambient air

DIRECTIVE 2002/3/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 12 February 2002
relating to ozone in ambient air

notes:

* For directive 2004/107/EC these are "target values" for the total content in the PM10 fraction averaged over a calendar year. For directive 1999/30/EC these are 24-h "limit values" for human health.

** Particulate Matter is a separate impact category/ indicator in our methodology

*** SO_2 and NO_2 are included in the separate category of acidifying agents with more or less the same relative weighting factor (1 vs. 0,7 for eco-toxicity, 1 vs. 0,62 here)

**** Ground-level ozone is not a direct anthropogenic emission but the result of a photochemical reaction (see text)

Volatile Organic Compounds (VOC). These include the C_xH_y emissions from oil-fired boilers. Strictly also methane (CH_4) is part of VOCs, but because the effect on the environment is different it is excluded. For this reason VOCs are often called NMVOCs (non-methane VOCs).

VOCs appear in Directive 2002/3/EC of 12 Feb. 2002 due to their role in (ground level) ozone and in Directive 1999/13/EC dealing with organic solvents. Furthermore, the European IMPEL network is monitoring fugitive NMVOCs, amongst others from combustion processes. There are no weighting factors mentioned and the MEEUP study proposes to simply make an inventory on a weight basis.

Formation of VOCs in commercial and industrial boilers primarily result from poor or incomplete combustion due to improper burner set-up and adjustment. To control VOC emissions from commercial and industrial boilers, no auxiliary equipment is needed; properly maintaining the burner/boiler package will keep VOC emissions at a minimum. Proper maintenance includes keeping the air/fuel ratio at the manufacturer's specified setting, having the proper air and fuel pressures at the burner, and maintaining the atomizing air pressure on oil burners at the correct levels. An improperly maintained boiler/burner package can result in VOC levels over 100 times the normal levels. Furthermore, as VOC emissions mainly occur at start-up and the end of a burning cycle, a very important measure is a reduction of the number of cycles.

Heavy Metals (Toxicity). Although not a Heavy Metal, the MEEUP classifies CO as a toxic agent, albeit –as an outdoor emission—with a very low weighting factor. Carbon monoxide is a pollutant that is readily absorbed in the body and can impair the oxygen-carrying capacity of the hemoglobin. Impairment of the body's hemoglobin results in less oxygen to the brain, heart, and tissues. Even short-term over exposure to carbon monoxide can be critical, or fatal, to people with heart and lung diseases. It may also cause headaches and dizziness in healthy people.

Particulate Matter (PM). This refers to 'soot' from oil-fired boilers. Emission limit values are mentioned in Directive 1999/30/EC, which indicate that the European legislator takes PM 10-emissions very serious indeed (see table 4). In fact, the emission limits on a weight basis are 4 times more stringent than the ones for NO_x.

PM emissions are primarily dependent on the grade of fuel fired in the boiler. Generally, PM levels from natural gas are significantly lower than those of oils. Distillate oils result in much lower particulate emissions than residual oils.

When burning heavy oils, particulate levels mainly depend on four fuel constituents: sulfur, ash, carbon residue, and asphalenes. These constituents exist in fuel oils, particularly residual oils, and have a major effect on particulate emissions. By knowing the fuel constituent levels, the particulate emissions for the oil can be estimated.

Methods of particulate control vary for different types and sizes of boilers. For utility boilers, electrostatic precipitators, scrubbers, and baghouses are commonly utilized. For industrial and commercial boilers, the most effective method is to utilize clean fuels. The emission levels of particulate matter can be lowered by switching from a residual to a distillate oil or by switching from a distillate oil to a natural gas. Additionally, through proper burner set-up, adjustment and maintenance, particulate emissions can be minimized, but not to the extent accomplished by switching fuels.

The above refers to emissions to air. To complete the picture it must be mentioned that in some regions of the EU there are strict regulations regarding the emissions to water, which –when using heating oil with a higher sulphur content—can apply to affluent of condensate to the sewer.

3.3 Emissions grouped by origin

Taking the angle of their origin, the emissions from gas-and oil-fired boilers can be split into four groups:

Unavoidable products from the combustion reaction. As already explained in the previous chapter water vapour and carbon dioxide (CO_2) are the main combustion products from the reaction between a hydrocarbon and oxygen. The CO_2 production is completely linked with a) the specific fuel and b) the energy efficiency of combustion. Regarding the fuel the CO_2 emissions per MJ gas are 20-30% lower[22] than with oil. Regarding the efficiency, it depends very much on the design. At best the oil-fired boilers in the top-end of the market can keep up (but not surpass) the best gas-fired boilers.

Pollutants that are unavoidable because they are already contained in the fuel. This is the case with SO_x production from sulphur. In principle, without end-of-pipe measures, the sulphur emissions are independent of the design of the combustion process. If we use heavy fuel oil with 3% sulphur, this amount will also result from the combustion process. If we use low-sulphur (<50 ppm) gas heating oil the corresponding lower amount will result. The only design-measure that a boiler designer can take is to make sure that the boiler (also) works with low-sulphur oil, but it is the user —or the regulations on the sulphur content of heating oil in a particular country— that will determine the outcome.

Emissions that are a consequence of incomplete combustion. Basically, these are all other carbon-containing compounds, besides CO_2: Carbon monoxide (CO), Methane (CH_4), hydrocarbons (C_xH_y) and soot (PM). The carbon in these compounds comes from the fuel and is an indicator of how much fuel was subject to incomplete combustion. The most well known cause of this is the lack of sufficient air/oxygen. But there may be other causes, such as the temperature of the fuel is too low to permit oxidation (combustion) to occur. It can occur as a result of flame impingement (flame in contact with metal) because parts of the flame are cooled—quenched—below the burn temperature of the fuel. For instance, on a gas range burner, flame impingement always

[22] Eurogas mentions a figure of 24%, citing the International Gas Union. The MEEUP table shows even higher differences (>30%) for comparable boilers.

occurs when a pot is on a burner. As the pot becomes hotter, the carbon monoxide production decreases because the flame is not cooled as much by the impingement. This makes measurement of carbon monoxide difficult; as impingement surfaces change temperature, the carbon monoxide emissions change. Quenching of a flame can also occur if air blows across a flame rapidly enough to cool it to below its burn temperature. A rule of thumb is that –in order to keep the CO-emissions low—the combustion temperature should be well above 900°C. Finally, the most obvious cause of non-CO_2 carbon emissions is during start- and stop of combustion, i.e. when unburned fuel remains in the combustion chamber. This causes of course a considerable amount of unburned fuel emissions (CH_4 or C_xH_y), but also gives peaks in CO-emissions as the circumstances at start-up (cold heat exchanger) are so favourable for CO-formation. As mentioned in chapter 2, 80-90% of the non-CO_2 carbon emissions occur not during steady-state but during start-up and stop.

Emissions that do not involve the fuel, but are chemical reactions between air molecules triggered by the specific combustion conditions. This relates to emissions of nitrogen oxides (NO_x), NO and NO_2, from the reaction between the oxygen and nitrogen molecules in the air. This occurs only when there is enough air around (excess air, e.g. air factor > 1,4), when the temperature is high enough (above 1200°C) and when there is enough time for the reaction to take place at this high temperature (the so-called 'residence time' should be long enough).

Basically the above is about all there is to tell about the amount of CO_2 and SO_x emissions (point 1 and 2). Once the fuel is chosen[23], the amount of SO_x and CO_2 emissions follow directly from the fuel input per functional unit.

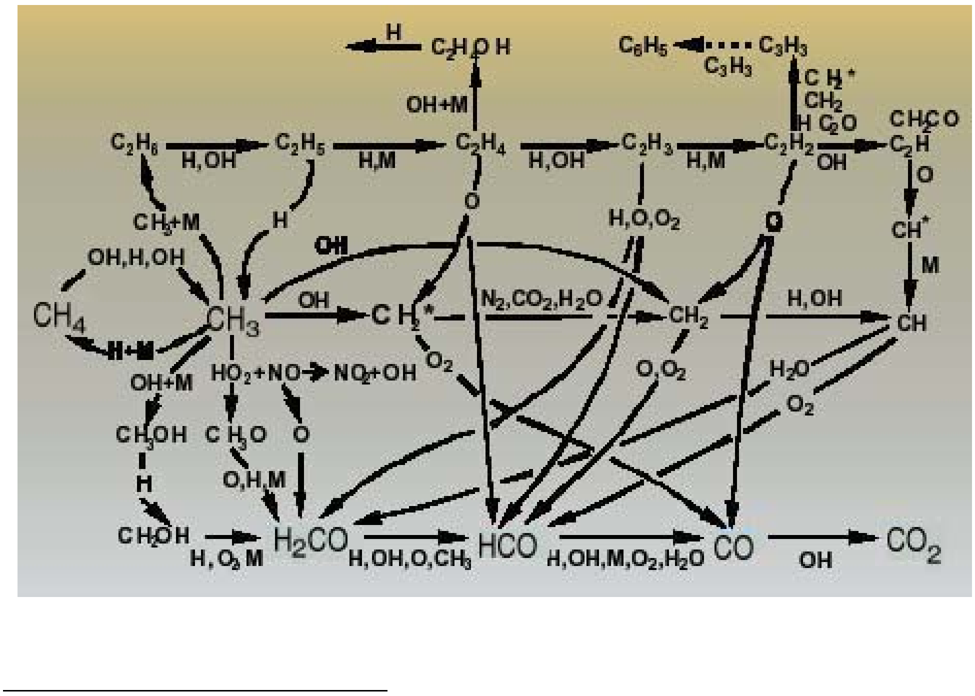
We will now expand on the points 3 and 4 mentioned above.

3.4 Low non-CO_2 Carbon Emission

3.4.1 Formation

The ***global*** chemical reaction just shows the **results** of what in reality is a complex series of simultaneous and consecutive chemical reactions. The picture below gives an impression of that complexity during methane combustion, whereby the molecules are first dissociated into smaller fractions before entering the chain reactions and finally the end-stage.

Figure 3-1.
Possible reactions during methane combustion (for natural gas) (Farago)



[23] And a minute amount is subtracted for unburned fuel (<1,5%, see Chapter 2)

The picture shows the steps in the methane combustion. From left to right it presents the oxidisation steps whereby hydrogen (H) is split off. From top to bottom it represents the oxidisation through taking up oxygen and splitting larger hydrocarbon molecules into smaller parts. In a 'rich combustion' (air factor < 1) the reactions predominantly follow the steps in the upper lines. In a 'lean combustion' (air factor >1) the reactions predominantly follow the steps in the lower lines of the picture. In the lower line the intermediate products are formaldehyde (H_2CO) and aldehyde (HCO) before arriving at carbon monoxide (CO) and finally carbon dioxide (CO_2). In the upper line acetylene (C_2H_2) is the most important intermediate product.

From this it will also be clear that in case of imperfect combustion CO is a combustion by-product. In case of rich combustion there will be a high C_2H_2 – concentration, which increases the tendency for soot-formation. In case of lean combustion there is a concentration of H_2CO, which reduces the formation of soot, but favours the formation of aldehyde.

3.5 Low NO_x technology

3.5.1 Introduction

In the discussions of nitrogen oxide combustion products and their impact to environment, the major nitrogen oxide species of concern are nitric oxide (NO) and nitrogen dioxide (NO2).

Under high temperature combustion conditions, the formation of NO is favoured and consequently, less than 10% of the NO_x in typical exhaust is in the form of NO2 (Pereira and Amiridis, 1995). However, a higher percentage of NO_x in the form of NO2 has been experienced in domestic applications. NO when it cools down in the atmosphere combines with oxygen in air to form NO2 (Eqn. 1).

$$2NO + O_2 \leftrightarrows 2NO_2 \qquad \textit{[Eqn. 1]}$$

In warm, sunny days the NO2 breaks down into NO and a nascent oxygen atom (Eqn. 2) which can combine with a molecule of oxygen to form ozone (Eqn. 3). The ozone reacts with NO to yield back NO2 almost as fast as it is formed.

$$NO_2 \leftrightarrows NO + O \qquad \textit{[Eqn. 2]}$$

$$O + O_2 \leftrightarrows O_3 \qquad \textit{[Eqn. 3]}$$

When volatile organic compounds (VOC) exist in the air, they combine with the NO in the present of sunlight to change it back to NO2. Less NO is then available to remove the nascent oxygen, and hence ozone accumulates, resulting in photochemical smog.

The term low NO_x technology used in the industry has a broad range in terms of the NO_x emission level achieved. In some instances, an emission of 70 - 80 ppm at 0% O_2 on dry basis is regarded as "low". In other instances, it may be down to 10 - 15 ppm or less. In the EU the threshold level of <40 ppm (70 mg/kWh) seems the most appropriate, being used in the German Blue Angel labelling scheme and the Dutch 'Low- NO_x' label and it is the lowest class limit (class 5) in the European Standard prEN 267.

Conversions:
Europe: 1 ppm (at 3% O_2) = 1,83 mg/kWh = 0,508 mg/MJ = 0,508 ng/J.
US: 100 ppm (at 3% O_2) = 0,118 lb/MMBtU (1 lb= 0,4535 kg; 1 Btu= 1,0546 kJ) = 183 mg/kWh.
ppm (at 3% O_2) = (21-3)/(21 – O_2 actual) ppm actual.
1 ppm (at 3% O_2) = 18/21= 0,857 ppm (at 0% O2).

This section is based on a study for the Australian government by environmental consultant Bob Joynt and combustion engineer Stephen Wu, which gives a good overview of the subject, also mentioning technology not strictly from a European angle.

3.5.2 *Formation of NO_x*

NO_x formed during combustion is the predominant source of NO_x to atmosphere. The source may be mobile or stationary, cars or boilers. NO_x consists of NO and NO2. For the convenience of discussion on a theoretical basis, only NO is discussed in this section.

NO can be categorised into the following:

- Thermal NO
- Fuel NO
- Prompt NO

For gas combustion burners such as Bunsen burners and flat flame burners which have a high flame temperature (> 1550°C), the NO formed is predominantly thermal NO, with a small fraction as prompt NO.

Thermal NO

Thermal NO is found mainly in the high-temperature post-flame zone. It is formed by the oxidation of molecular nitrogen in combustion air and fuel gases by the extended Zeldovich mechanism:

$$O+N_2 \leftrightarrows NO + N \qquad \text{[Eqn. 4]}$$

$$N+O_2 \leftrightarrows NO + O \qquad \text{[Eqn. 5]}$$

$$N+OH \leftrightarrows NO + H \qquad \text{[Eqn. 6]}$$

where the nascent oxygen atom in Eqn. 4 is formed (with a large activation energy) from the H_2-O_2 radical pool or possibly from the dissociation of O_2 (Glassman, 1996).

The hydroxyl (OH) radical in Eqn. 6 may come from the following reaction, which obtains the hydrogen atom from the dissociation of hydrocarbon fuel:

$$H+O_2 \leftrightarrows OH + O \qquad \text{[Eqn. 7]}$$

Eqn. 4 is rate-determining. To reduce thermal NO formation, O (nascent oxygen atom) must be reduced. The formation of O, and hence thermal NO, is more dependent on the combustion temperature and less dependent on the oxygen concentration. It increases with temperature. For combustion systems like those obtained on Bunsen and flat flame burners, the temperature, and hence the mixture ratio, is the prime parameter in determining the quantities of thermal NO formed.

Fuel NO

Fuel NO is formed by the oxidation of nitrogen chemically bound in fuel. In the production of natural gas and liquid petroleum gas, combustible gaseous nitrogen compounds such as ammonia and amines have been removed to insignificant levels and little or no fuel NO would be formed.

Prompt NO

Prompt NO is most frequently observed in fuel-rich flames and at low temperatures, and its formation is found to be relatively independent of temperature. There are three possible sources of prompt NO (Glassman, 1996):

Non-equilibrium nascent oxygen (O) and hydroxyl (OH) radical concentrations in the reaction zone and burnt gas, which accelerate the rate of thermal NO (Zeldovich) mechanism.

A reaction sequence initiated by reactions of hydrocarbon radicals, present in and near the reaction zone, with molecular nitrogen (Fenimore prompt NO mechanism):

$$CH + N_2 \leftrightarrows HCN + N \qquad [Eqn. 8]$$

$$C_2 + N_2 \leftrightarrows 2\,CN \qquad [Eqn. 9]$$

The nascent N atom can yield NO by reactions such as Eqn. 5 and Eqn. 6, and CN can form NO with a nascent oxygen atom or oxygen molecule.

Reaction of nascent oxygen (O) with molecular nitrogen to form nitrous oxide (N_2O) via the three-body recombination reaction (Eqn. 10) and the subsequent reaction (Eqn. 11) to form NO:

$$O + N_2 + M \rightarrow N_2O + M \qquad [Eqn. 10]$$

$$N_2O + O \rightarrow NO + O_2 \qquad [Eqn. 11]$$

The non-equilibrium O and OH concentration mechanism is more important for non-pre-mixed flames, stirred reactors for lean conditions, or low pressure premixed flames.

The Fenimore prompt NO mechanism is dominant in fuel-rich pre-mixed hydrocarbon combustion.

The nitrous oxide mechanism becomes more important when the fuel-air ratio decreases, when the burnt gas temperature decreases, or when the pressure increases.

At common combustion temperatures, increase in aeration can reduce prompt NO formation.

Formation of NO_2

Despite the favoured formation of NO dictated by thermodynamics and reaction kinetics, high concentrations of NO_2 have been experienced in domestic applications, e.g., Glassman (1996) cited that high concentrations of NO_2 was reported in the exhaust of range-top burners.

It was observed that NO_2 was formed by HO_2 and NO in the low-temperature regime of visible flames (Eqn. 12) and suggested that the conversion of NO_2 to NO and oxygen in the near-post-flame zone (as given by Eqn. 11) was quenched.

$$NO + HO_2 \rightarrow \; \varsubsetneq O_2 + OH \qquad [Eqn. 12]$$

3.6 Principles of Primary Control of NO_x Emissions

NO_x control may be:

- Primary - to reduce NO_x formation.
- Secondary - to remove NO_x formed.

There are three basic principles of primary NO_x control to reduce NO_x formation:

- Reduction of high combustion/flame temperature since more NO_x will be formed at higher temperatures under thermodynamic equilibrium conditions.
- Reduction of residence time at high combustion temperature to resist the NO_x formation approaches thermodynamic equilibrium concentration.
- Reduction of oxygen concentration and hence the nascent oxygen concentration in the high temperature zone.

It is possible to quench the NO_x reactions, obtain the chemical heat release and prevent NO_x formation (non-equilibrium Zeldovich mechanism) but in practice efficiency often suffers if quenching is done by adding a non-reacting mass such as water or steam to the system.

Any acceptable NO_x control technology should reduce NO_x emissions, at the same time maintain or decrease CO and formaldehyde emissions, and maintain or increase thermal efficiency.

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

- Modification of fuel/air delivery-burner system.
- Modification of gas burner.

3.6.1 Modification of Fuel/Air Delivery-Burner System

The strategies to modify fuel/air delivery-burner systems can be summarized as follows:

- Increasing the primary pre-mixed air from ~ 50% to more than 100%
- Low excess air (LEA) firing
- Flue gas recirculation (FGR). Recirculating combustion exhaust gases into primary combustion air.
- Staging combustion into more than one discrete step, with heat extracted between steps.
- Delaying, distributing, or dispersing fuel/air mixing within the combustion chamber.
- Humidifying fuel gas, combustion air, or the flame.

Increasing the Primary Premixed Air

This measure applies to an atmospheric (partial pre-mix) burner, which uses both primary ('pre-mix') and secondary air. NO_x emissions from blue flames could be reduced from ~ 100 ppm to < 70 ppm (oxygen (O_2) free) by increasing the primary air from ~ 50% to ~150% of the stoichiometric air required.

Effectively any excess air above 100% stoichiometric dilutes the combustion exhaust and brings down the combustion temperature from a maximum of ~ 1900°C to ~ 1200°C, causing less NO_x to be formed.

Lower combustion temperature would result in longer combustion time at high temperature because of slower burning rate. This would encourage NO_x formation, but this effect was observed to be secondary and a net decrease in NO_x emission would result.

Means to increase the primary air flow to ~ 50% excess are a very large venturi, a fan and higher gas- or air-line pressure. In the EU boilers, the use of fans in a *full pre-mix* burner is the most common measure.

In Japan (Tokyo Gas, Rinnai) and US (Burnham, Gas Research Institute) one would find new designs of aspiration such as alternating burner ports fire with primary air < 100% in one port and up to ~ 85% excess air in the adjacent ports to achieve ~ 70 ppm. Also there are new burner design to accelerate the velocity of the burning pre-mixture and shorten the residence time besides reducing combustion temperature, with a hemispherical bluff body re-stabilises the flame.

Burners designed for excess primary aeration would have deeper ports and thicker walls than the usual stamped metal burners. Secondary aeration would not be required and could be eliminated by closed combustion chamber or baffles.

Low Excess Air (LEA) Firing

As a safety factor to assure complete combustion, boilers are fired with excess air. One of the factors influencing NO_x formation in a boiler is the excess air levels. High excess air levels (>45%) may result in increased NO_x formation because the excess nitrogen and oxygen in the combustion air entering the flame will combine to form thermal NO_x. Low excess air firing involves limiting the amount of excess air that is entering the combustion process in order to limit the amount of extra nitrogen and oxygen that enters the flame. Limiting the amount of excess air entering a flame is accomplished through burner design and can be optimized through the use of oxygen trim controls.

Low excess air firing can be used on most boilers and generally results in overall NO_x reductions of 5-10% when firing natural gas.

Recirculating Combustion Exhaust Gases

Recirculation of flue gases could be achieved by:

- Buoyancy
- Aspiration
- Fan

The cooled combustion exhaust gases (mainly molecular nitrogen and oxygen, carbon dioxide and water vapour) are mixed with air entering the burner. The recirculated gases dilute the primary air and lowers the oxygen concentration of the air mixture from ~ 21% by volume to ~ 18%. Consequently the flame temperature is lowered. Research on larger scale applications has demonstrated that NO_x could be reduced by ~ 75% when the primary air contains ~ 30% recirculated flue gas.

Ducting of the exhaust gases to the fuel/air delivery system would be required. The combustion chamber and heat exchanger of the appliance may become larger to accommodate the higher total gas flow rate and lower flame temperature to maintain baseline thermal efficiency. The burner may have to be upgraded to light and stabilise the fuel-air-exhaust mixture which is more difficult to ignite and slower in combustion, although the warm mixture (if the exhaust gases are mixed at a few hundred degrees C) would alleviate this to some extent. Another concern is that lower flame temperature and oxygen concentration would favour CO formation.

Raghavan and Reuther (1994) pointed out that recirculation of combustion exhaust gases had been used at industrial scale to reduce NO_x emission but not in domestic application, which is still true. Because of the high NO_x reduction potential, they felt that domestic application of this strategy should be explored further. Recirculation often requires a fan driven system that may have to work at elevated temperatures and this would increase the cost of the appliance and its operation.

US industrial boiler manufacturer Cleaver Brooks identifies flue gas recirculation (FGR) as the most effective and popular technology for industrial boilers. And, in many applications, it does not require any additional reduction equipment to comply with regulations.

Flue gas recirculation technology can be classified into two types; external or induced.

- **External flue gas recirculation** utilizes an external fan to recirculate the flue gases back into the flame. External piping routes the exhaust gases from the stack to the burner. A valve controls the recirculation rate, based on boiler input.
- **Induced flue gas recirculation** utilizes the combustion air fan to recirculate the flue gases back into the flame. A portion of the flue gases are routed by duct work or internally to the combustion air fan, where they are premixed with the combustion air and introduced into the flame through the burner. New designs of induced FGR that utilize an integral FGR design are becoming popular among boiler owners and operators because of their uncomplicated design and reliability.

Up to a re-circulation ratio of 1, this can be done with conventional flames. Above this ratio of 1, the temperature of the burner/ combustion chamber have to be involved in the process to keep the temperature level above ignition temperature. Between a ratio of 1 to 3,5 it is not possible to realize the combustion process, but at the re-circulation ratios of 3,5 and higher there is a flameless combustion reaction in a large surface. This flameless combustion process is known as ***FLOX*** (Flameless Oxidisation). The temperature and re-circulation rates are shown in the picture below (see also Chapter on Burners).

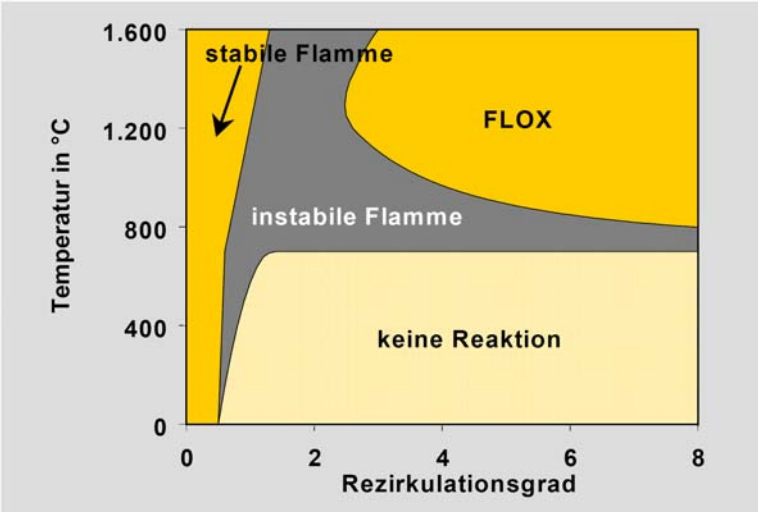


Figure 3-2.
Recirculation-rate of FLOX burners

The FLOX technology has been used in industrial burners, but now –through a new collaboration between DLR and WS Wärmeprozesstechnik– will be further developed for gas turbines. [24]

FLOX technology can be combined with the staged combustion (see below and Chapter on burners).

Staging Combustion

Staged combustion can be conducted in two stages, the first is the fuel-rich combustion with < 100% primary aeration and the second is fuel-lean, with inter-stage cooling such as radiant heat loss from a radiant burner, or heat exchange with air or water. In principle, more stages can be used but the design, manufacture and operation will be more complicated and more expensive.

Staging can be achieved by modifying the gas burner or the combustion chamber, or both. The flame temperature at the two stages is lower than the dual flame combustion using the same overall (primary plus secondary) aeration. In a combined approach for a fan-assisted space heater prototype with a radiant burner, a reduction of NO_x emission by ~75% was reported (Raghavan and Reuther, 1994).

Design and manufacture of staged combustion gas appliances are more complicated and expensive. Many of the components such as channels, flame holder, ignition system, combustion chamber and heat exchanger may have to be increased in number or in physical size. This will increase the manufacturing cost of the appliance.

In principle, staged combustion can be performed with stable flame without fan assistance, but the problem of increased CO emission and decreased thermal efficiency must be addressed together with NO_x reduction.

In the US staged combustion techniques are applied in residential low NO_x burners. Reportedly the US Gas Research Institute (GRI) co-developed boilers and furnaces with staged combustion and internal flue gas recirculation with US manufacturers Burnham, Empire Comfort and Trane, reaching a low NO_x level of 25-29 ppm at 3% O2 and CO was found to be less than 50 ppm air-free.

In Europe the use of staged combustion is primarily limited to industrial and commercial boilers.

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[24] Press release, Deutsches Zentrum für Luft und Raumfahrt, *Neuer Brenner verspricht Stickoxidarme Verbrennung –*

DLR und WS Wärmeprozesstechnik schließen Vermarktungsvertrag, 10. November 2005.

Delaying Combustion

Different from staged combustion, delaying combustion allows the combustion process to occur continuously rather than at discrete stages, over lower temperatures, to retard NO_x formation. This is achieved by dispersion, with slower heat release, over larger volumes and time.

Raghavan and Reuther (1994) cited from the literature four examples of burner design to delay combustion, with one suitable for air heaters and the other for water heaters. They recognised that although this approach was effective to lower NO_x emissions (by up to ~75%) and amenable to a variety of atmospheric or powered burners, the development had been limited, which could be related to higher CO emissions and lower efficiency. The fuel/air delivery might need to be pressurised, the burner, combustion chamber and heat exchanger might need enlargement, and the ignition system might require improvement.

In the EU no examples of delayed-combustion technology were found, probably due to the drawbacks mentioned.

Humidifying the Fuel Gas, Combustion Air or Flame

Humidification can be conducted by:

- Spraying water to the combustion air.
- Spraying water to the combustion chamber.
- Spraying steam to the combustion air or fuel gas.
- Spraying steam to the combustion chamber.

Steam dilutes the combustion exhaust in the same way as recirculated combustion exhaust gases. The effect of water is two fold: water evaporates by absorbing a large quantity of heat (latent heat of evaporation) from the combustion system and the steam evolved dilutes the combustion exhaust gases. Both result in cooling the combustion system.

The spraying rate of water to combustion air is restricted by the ambient humidity conditions and the efficiency of water atomisation. The spraying rate of water to the combustion chamber and the spraying rate of steam would depend on flame stability.

The investigation of the humidification for domestic appliances was limited even though the NO_x reduction could be up to ~ 50 - 60% (Raghavan and Reuther, 1994). It has not been attractive probably because the efficiency of the system will decrease with humidification, unless steam in the exhaust gases is condensed and the heat extracted is recoverable. Condensation would complicate the combustion system, create corrosion problem and increase the equipment cost.

Humidification has been used in commercial scale continuous gas turbine operation but not in domestic situations. The loss of efficiency in gas turbine application is traded off with the increase in power output by the higher mass flow through the gas turbine.

Under normal operating conditions, water/steam injection can result in a 3-10% boiler efficiency loss (Cleaver- Brooks).

3.6.2 Modification of Gas Burner

Raghavan and Reuther (1994) identified the major modifications of gas burners as follows:

- Flame Inserts.
- Blue-flame burner redesign.
- Blue-flame burner replacement.

Flame Inserts

A simple means to reduce flame temperature is to insert a foreign object, such as a solid rod or porous screen, into a blue flame and allow the object to radiate red hot. As part of the heat liberated is transferred by radiation, the flame temperature is reduced and hence the NO_x emissions are reduced. The inserts could be made of refractory metals or ceramics.

Raghavan and Reuther (1994) cited five different flame inserts patented for atmospheric burners:

- A ring shaped solid insert for range or water heater burners.
- A rod shaped solid insert for furnace burners.
- A porous screen insert.
- A solid channel insert for furnaces.
- Small solid fin inserts integral with the burner but not in the flame.
- A perforated radiant insert for fan-assisted power burner was also illustrated.

From the literature search, Raghavan and Reuther indicated that most flame inserts could achieve a ~60% reduction but the CO emissions would typically increase, since the combustion conditions remain the same except at a lower temperature which favours CO formation. Adjusting the position of insert or using secondary-air baffles may alleviate CO formation. Thermal efficiency could be an issue, but it may be overcome depending on the application and design.

Compared to other NO_x control techniques, Raghavan and Reuther believed that flame inserts had the least impact on gas appliance component design. However, because of the change in heat transfer and flame shape, heat exchangers, particularly those used in space heaters, might require re-design.

Flame inserts are typical of reducing NO_x in atmospheric burners. In the US, new designs are developed by DSL Technologies and Lennox.

Blue-Flame Burner Redesign

Blue-flame burners could be redesigned either by changing the burner's thermal mass, port loading, or port design to achieve reduced NO_x emissions.

Thermal Mass

Cast iron burners are more "thermal active" than the traditional stamped steel and aluminium burners, and are found to emit less (~30%) NO_x and CO. This is achieved by dissipating more heat via their high thermal mass (and structure).

Cast iron atmospheric or power burners have been applied to ranges and water heaters to lower NO_x (down to < 70 ppm at 0% O_2 dry basis). Thermal efficiency was reported to increase slightly (Raghavan and Reuther, 1994).

Port Loading

NO_x emissions depend on port loading — the heat released per port area per time. It was reported that NO_x emissions from atmospheric blue flames could be reduced by half if the port loading was reduced by one third. Reducing port loading is achieved by increasing burner size if the same heat input rate is maintained. Thermal efficiency may increase or remain the same, but CO emissions could increase and flashback may occur.

Port Design

Port spacing determines the extent of flame aeration and interaction, which affect NO_x formation. If the heat dissipated by the ports is increased and the secondary aeration of flames is improved, NO_x emissions can be reduced.

Raghavan and Reuther (1994) described the Worgas hyperstoichiometric burner as an example. The Worgas burner uses a venturi-burner system with unique port spacing and 80 - 160% stoichiometric air requirement. The burner is larger than the traditional Bunsen type blue flame burner. It has improved secondary-air entrainment, yielding violet flames with low and uniform temperature distribution. The butterfly-wing flame shape has the aerodynamics designed to bring combustion products back to the flame.

Laboratory results indicated that the Worgas burners could achieve 40 ppm NO_x at 3% O_2, dry basis, which is equivalent to 45 ppm at 0% O_2, dry basis. Thermal efficiency is claimed to be high, and the technology can be used in boilers, instantaneous water heaters, storage water heaters, and room/air heaters.

Blue-Flame Burner Replacement

Blue flame burners have been suggested to be replaced with "flameless" burners which adopt radiant combustion, catalytic combustion, or pulse combustion.

Radiant Combustion

Radiant combustion occurs near or within burners which are either porous or ported, and may be fan-assisted. The burners can have different shapes to suit different heat exchangers. In operation, the burners glow in a red-orange colour (> 680°C).

Similar to flame inserts, radiant burners restrict NO_x formation by lowering the combustion temperature, but in a better and more complete manner. NO_x emission < 25 ppm and CO emission < 50 ppm O_2-free have been reported (Raghavan and Reuther, 1994). Facilitated with high excess aeration and reduced port loading, radiant burners could achieve < 10 ppm NO_x O_2-free. In combination with staged combustion, NO_x emissions < 10 ppm O_2-free was experienced. With proper location of heat exchangers, higher thermal efficiency can be obtained.

Radiant burners are normally larger than blue-flame burners. Modification of other components is often required. Pressurisation of the fuel/air delivery system and filtering may be required depending upon burner port size. Usually the combustion chamber is reduced but the ignition system would require upgrading. The heat exchanger would have to be relocated closer to the burner.

In the US Alzeta Corp[25] and Global Environmental Solutions are manuafacturers. In Australia Bowin[26] has developed a patented technology in this respect.

Pre-mix radiation burners are the state-of-the-art in the EU. For instance burner-manufacturer Bekaert in Belgium produces metal fibre burners for premixed gas surface combustion, developed by Acotech[27]. They can be operated in either radiant combustion mode or blue flame surface combustion mode. In the former mode NO_x emission < 10 ppm at 0% O_2 dry basis is claimed to be achieved. In the latter mode, it is claimed that low NO_x levels (30 ppm NO_x) are achieved at 30% excess air. CO emission is claimed to be < 10 ppm. Other advantages such as homogeneous combustion with high modulation rate, high efficiency, low pressure drop, resistance to thermal shock and flashback safety are also claimed. Major boiler manufacturers such as Vaillant,

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[25] **Alzeta:** Pre-mix radiant burner with a trade name as Pyrocore/Duratherm from alumina-silica fibres fibres which are formed into either cylinders or flat plates with high porosity. This technology has been used by Alzeta's OEM partner, Nuovi Sistemi Termotecnici in Italy on domestic boilers and instantaneous water heaters

[26] Bowin mfg. Pty. Ltd (Australia) Bowin has been manufacturing a number of ultra-low NO_x flued and flueless natural aerated and powered domestic flue heaters using Bowin's patented surface combustion technology. The technology is also applicable to domestic water heaters and cooking appliances (John Joyce, personal communication).

The Bowin low NO_x technology is a hybrid of staged-premixed-radiant combustion technology with a major surface combustion preceded by a minor radiant combustion. In the Bowin burner, air and fuel gas are premixed at a ratio greater than or equal to the stoichiometric combustion requirement.

Combustion is maintained at or adjacent to a combustion surface formed from one or more layers of conductive heat resistant material such as nickel based steel mesh with uniform porosity of 20 - 60% (Australian Patent Document Number: AU-B-64743/90). The porosity provides a flow rate of air-fuel mixture that results in a combustion temperature of 600 - 900°C and radiant heat transfer that maintains the combustion temperature.

Low NO_x (£ 2 ng/J or ~ 4 ppm at 0% O_2 on dry basis) and CO emissions have been achieved (as measured by The Australian Gas and Light Company (AGL)). Further reduction in NO_x emission could be achieved by using baffles, barriers walls or enclosed combustion chamber to restrict or prevent cold secondary air contacting the flame before combustion is completed (Australian Patent Document No.: AU-B-16047/92).

Currently Bowin is collaborating with an Australian water heater manufacturer to develop a prototype low NO_x water heater using Bowin's technology.

[27] A joint Shall/Bekaert company www.acotech.com

Viessmann and Buderus in Germany, Remeha in Holland, and Ecoflam and Baltur in Italy have reportedly been using this technology (see Chapter on burners, also for other radiation burner solutions).

Catalytic Combustion

Catalytic combustion may be fully catalytic (or simply catalytic), or partial which is also known as catalytically stabilised (Ro and Scholten, 1997).

In catalytic combustion, a catalyst such as palladium or platinum is used to reduce the activation energy of combustion and allow the fuel gas to be oxidised by air at a low temperature of 500 - 1000°C. The reaction temperature is maintained low by effective removal of heat liberated from oxidation to the heating medium. Because the reaction temperature is low, Ro and Scholten stated that NO_x levels < 5 ppm could be achieved.

In catalytically stabilised combustion, part of the fuel gas is oxidised by catalytic combustion, and the remaining gas is oxidised by homogenous (blue flame) combustion after or during catalytic combustion. Providing heat is removed from the catalytic system, the product gases from catalytic combustion dilute the exhaust gases from the homogenous combustion and lower the overall combustion temperature, and hence NO_x emission, in a way similar to flue gas recirculation.

Ro and Scholten compared the performance of boilers using catalytic combustion and catalytically stabilised combustion. They concluded that catalytically stabilised combustion had a higher reliability because it could be operated as a conventional radiant burner even if the catalyst was poisoned and totally de-activated, and the security and control system required for temperature/combustion control would be more easily developed. Catalytic combustion on the other hand, emitted less NO_x and CO, and its method of catalyst coating was easier.

In the review performed by Raghavan and Reuther three years earlier than Ro and Scholten, a catalytic burner used in a gas-fired appliance was cited. The burner surface was a matrix of ceramic fibres interspersed with chrome (catalyst) fibres. NO_x emission < 15 ppm and CO emission < 10 ppm O_2-free were reported.

Catalytic converters similar to those used in automobiles were also cited by Raghavan and Reuther. The converter completed catalytically the combustion of the products from an earlier fuel-rich combustion with more cool air at a temperature < 540°C. NO_x emission from this two-staged combustion was lower than that from a second stage combustion which was non-catalytic but conducted at a higher temperature.

Raghavan and Reuther suggested that the requirements of fan-assistance to overcome the problem of low temperatures and low heat fluxes, larger heat-exchange areas, and smaller combustion chamber volumes might be the main draw backs of wide application of catalytic combustion to gas appliances.

Pulse Combustion

In this mode, combustion occurs intermittently and the combustion gases experience high temperatures for very short time only. Heat transfer from gases to heat exchange surfaces is fast due to high turbulence, which maintains a lower temperature and hence lower NO_x emissions.

NO_x levels of < 50 ppm were reported, and the technology had been commercialised in residential heating appliances (Raghavan and Reuther, 1994).

The noise level of pulse combustion systems would be high, and this could limit the application of pulse combustion in domestic situations.

Pulse combustion is used by US manufacturers such as Lennox and Empire Comfort Systems. In Europe it is mostly used in industrial applications.

3.6.3 Primary NO_x Control Technology Status

Ro and Scholten (1997) summarised the NO_x emissions achieved by various types of burners. The results are reproduced in Figure 3-3:

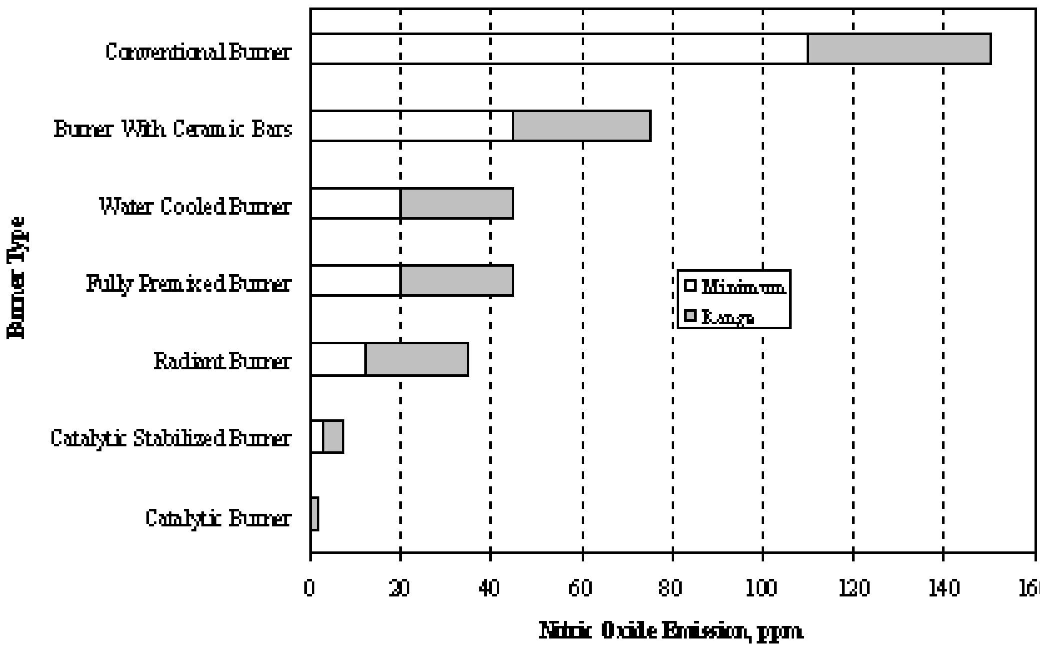

Figure 3-3. Nitric oxide emission levels of various burner types.

On the basis of the above and the summary of Raghavan and Reuther (1994) the status of different primary NO_x control technologies around the year 2000 is reproduced in Table 3-4.

Table 3-4. Comparison of primary NO_x control strategies for residential gas appliances*. (source: Joynt, B, Wu, S., 2000)

Primary NO_x Control Technology	Likely Lowest NO_x (ppm, O_2-free)*	Likely Change in CO Emissions*	Likely Change in Thermal Efficiency*	Technology Status for Domestic Application*
Premixed, High Excess Air	~ 20	Decrease	Decrease	Current
Flue-Gas Recirculation	~ 25	Increase	Decrease	Not Commercialised
Staged Combustion	~ 25	Increase	Decrease	Current
Delayed Combustion	~ 25	Increase	Decrease	Not Commercialised
Humidified Combustion	~ 25	Increase	Decrease	Not Commercialised
Flame Inserts	~ 40	Increase	Decrease	Current
Thermally Active Burner	~ 65	Decrease	Increase	Current
Port-Loading Reduction	~ 50	Increase	Increase	Current
Port Redesign	~ 45	Decrease	Increase	Current
Radiant Combustion	~ 4 - 10	Decrease	Increase	Current
Catalytic Combustion	~ 5	Decrease	Decrease	Not Commercialised
Pulse Combustion	~ 20	Increase	Increase	Current

3.6.4 Secondary Control of NO_x Emission

NO_x can be removed from combustion exhaust gasses in three approaches:

- Selective catalytic reduction (SCR).
- Selective non-catalytic reduction (SNCR).
- Hybrid SNCR/SCR.

These technologies are expensive because consumable reagents and additional NO_x removal systems are introduced. Moreover, the additives such as ammonia if not consumed in the process will escape to the atmosphere which would lead to NO_x. Until now, applications of secondary control are mostly to power generation and other industrial combustion processes.

In the context of the underlying study they are not considered viable and will not be further discussed.

3.7 Low emissions vs boiler performance/efficiency ?

What effect does NO_x control technology ultimately have on a boiler's performance? Certain NO_x controls can worsen boiler performance while other controls can appreciably improve performance. Aspects of the boiler performance that could be affected include turndown, capacity, efficiency, excess air, and CO emissions.

Failure to take into account all of the boiler operating parameters can lead to increased operating and maintenance costs, loss of efficiency, elevated CO levels, and shortening of the boiler's life.

The following section discusses each of the operating parameters of a boiler and how they are related to NO_x control technologies.

Turndown

Choosing a low NO_x technology that sacrifices turndown can have many adverse effects on the boiler. When selecting NO_x controls, the boiler should have a turndown capability of at least 4:1 or more, in order to reduce operating costs and the number of on/off cycles. A boiler utilizing a standard burner with a 4:1 turndown can cycle as frequently as 12 times per hour or 288 times a day because the boiler must begin to cycle at inputs below 25% capacity.

With each cycle, pre- and post-purge air flow removes heat from the boiler and sends it out the stack. The energy loss can be reduced by using a high turndown burner (10:1), which keeps the boiler on even at low firing rates.

Every time the boiler cycles off, before it comes back on, it must go through a specific start-up sequence for safety assurance. It takes between one to two minutes to get the boiler back on line. If there is a sudden load demand, the response cannot be accelerated. Keeping the boiler on line assures a quick response to load changes.

Frequent cycling also deteriorates the boiler components. The need for maintenance increases, the chance of component failure increases, and boiler downtime increases. So, when selecting NO_x control, always consider the burners turndown capability.

Capacity

When selecting the best NO_x control, capacity and turndown should be considered together because some NO_x control technologies require boiler derating in order to achieve guaranteed NO_x reductions. For example, flame shaping (primarily enlarging the flame to produce a lower flame temperature — thus lower NO_x levels) can require boiler derating, because the shaped flame could impinge on the furnace walls at higher firing rates.

However, the boiler's capacity requirement is typically determined by the maximum load in the hot water system. Therefore, the boiler may be oversized for the typical load conditions that may occur. If the boiler is oversized, its ability to handle minimum loads without cycling is limited. Therefore, when selecting the most appropriate NO_x control, capacity and turndown should be considered together for proper boiler selection and to meet overall system load requirements.

Efficiency

Some low NO_x controls reduce emissions by lowering flame temperature. Reducing the flame temperature decreases the radiant heat transfer from the flame and could lower boiler efficiency. The efficiency loss due to the lower flame temperatures can be partially offset by utilizing external components, such as an economizer. Or, the offset technique can be inherent in the NO_x design.

One technology that offsets the efficiency loss due to lower flame temperatures in a firetube boiler is flue gas recirculation. Although the loss of radiant heat transfer could result in an efficiency loss, the recirculated flue gases increase the mass flow through the boiler — thus the convective heat transfer in the tube passes increases.

The increase in convective heat transfer compensates for losses in radiant heat transfer, with no net efficiency loss. When considering NO_x control technology, it is not necessary to sacrifice efficiency for NO_x reductions.

Excess Air

A boiler's excess air supply provides for safe operation above stoichiometric conditions. A typical burner is usually set up with 10-20% excess air (2-4% O2). NO_x controls that require higher excess air levels can result in fuel being used to heat the air rather than transferring it to usable energy. Thus, increased stack losses and reduced boiler efficiency occur. NO_x controls that require reduced excess air levels can result in an oxygen deficient flame and increased levels of carbon monoxide or unburned hydrocarbons. It is best to select a NO_x control technology that has little effect on excess air.

Carbon Monoxide (CO) Emissions

High flame temperatures and intimate air/fuel mixing are essential for low CO emissions. Some NO_x control technologies used on industrial and commercial boilers reduce NO_x levels by lowering flame temperatures by modifying air/fuel mixing patterns. The lower flame temperature and decreased mixing intensity can result in higher CO levels.

An induced flue gas recirculation package can lower NO_x levels by reducing flame temperature without increasing CO levels. CO levels remain constant or are lowered because the flue gas is introduced into the flame in early stages of combustion and the air fuel mixing is intensified. Intensified mixing offsets the decrease in flame temperature and results in CO levels that are lower than achieved without FGR. But, the level of CO depends on the burner design. Not all flue gas recirculation applications result in lower CO levels.

Conclusion

There is no contradiction between eco-design for low emissions and eco-design for energy efficiency and good performance. In fact, the most effective design measures, such as pre-mix burners, radiation burners (lower temperature), reduction of the number of cycles (e.g. through deep modulation), etc. are equally effective in lowering emissions as in increasing the energy efficiency. There are some exceptions and limitations, e.g. where there is a trade-off with CO and NO_x emissions in the combustion temperature, but overall if the designer recognizes these boundary conditions and deals with them appropriately they are not problematic. Overall, there is a great deal of synergy, where intelligent design measures contribute not only to one environmental aspect, but to the whole spectrum of environmental, energy and resources impacts.

3.8 References

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4 BURNERS

4.1 Introduction

Following the more general chapters on energy balance and emissions, this chapter gives a more hands-on overview of the current EU burners sold for gas- and oil fired CH boilers and water heaters. It discusses the trends and the main types and characteristics.

Current burner production is in the hands of both the boiler manufacturers and specialised burner-OEMs. Boiler manufacturers like Weishaupt, Viesmann, Buderus, etc. are mostly manufacturers of jet burners for floor-standing gas and oil boilers. Specialised burner-producers like Bekaert (Belgium), Worgas (Italy), etc. are mainly producing burners for wall-hung gas boilers.

4.2 Trends

Over the last two decades there has been a development from the traditional atmospheric burners towards Low-NO_x pre-mix burners, typically with lower combustion temperatures. This trend was fuelled by the ‘technology push’ of new high-temperature materials becoming available (e.g. ceramics, metal fibres) and the ‘demand pull’ of better energy efficiency (in part load and during cycling), higher heating comfort and lower (NO_x) emissions.

At the moment, this trend seems to have slowed down for a number of reasons.

- In the beginning the new materials had some problems regarding fragility, a too short product life, etc.. Currently this reputation is undeserved[28] when the burners are applied properly. But it is never easy to remedy first impressions.
- Secondly, pushed by the competition and new insights burner-manufacturers found that they could meet large part —at least a sufficient part— of the legislative emission-requirements with traditional materials like perforated refractory steel[29] plate or (half) cylindrical burners.
- Thirdly, although the in the 1990s the legislators in some countries like Germany and Austria were very active in setting maximum emission limit values for boilers, there have been no updates since and few countries have followed, despite m,easures such as the EU NEC Directive. Furthermore, as already indicated in the Task 1 report, the CEN has hardly updated their emission measurement methods – which were originally meant only for safety—for a practice of environmental impact. For instance, the EN standards measure at stationary (full load) conditions, whereas in practice 80 (oil) to 95% of emissions of CO, CH_4, C_xH_y occur during cycling (start/stop).
- Fourthly, regarding a possible contribution of the burner in improving the energy efficiency boiler manufacturers have found that they could achieve this also in another, albeit more economical way at the level of the heat exchanger, e.g. recuperating latent heat of condensation.

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[28] Manufacturers have solved these problems and e.g. ceramic burners are successfully being used in –mostly larger—burners

[29] Temperature resistant, low oxidisation e.g. compare stainless steel.

All in all, this has made the burner into somewhat of a low-interest standard component, where pre-dominantly the most economical pre-mix perforated steel plate version is applied throughout. Prices are in the order of € 8-10, which is hardly more than the price of an atmospheric burner. For integrated boilers the plate or (half-) cylindrical versions are used the most.

Yet, as has been argued in e.g. the chapter on emissions, for Eco-design the burner may be far more than a low-interest product.

4.3 Types

For CH Boilers there are two types of burners:

- Surface burners
- Jet burners

They can be fan-assisted (pre-mix) or not.

4.3.1 Surface burners

A surface burner is a flat or (half)-cylindrical perforated plate or woven-fibre of metal or ceramic material. Each hole in the plate ('burner port') serves as a flameholder. The geometry of the holes, together with the flow and pressure of the fuel and combustion air (or their mixture), determines the shape and the size of each individual flame. Depending on the position of the flame we can distinguish

- the flame hovers over the burner bed (***'free flame'***),
- the flame sits at the burner surface, i.e. at burner nozzle exit (***'radiation burner'***) or
- the combustion takes place inside the burner nozzles (***'flameless burner', e.g.)***

All these three options —and their intermediate variations— result in a different heat transmission of the flame to the burner bed and thereby a different temperature of the resulting combustion products and a different share of the radiation energy (from flame + burner) and convection energy. E.g. For gas-fired burners some typical values are:

- the free flame burners: around 5% radiation share and flue temperatures of 1300-1500°C
- metallic pre-mix burners: around 5-15% radiation share and flue temperatures of 1200-1300°C
- ceramic surface burners: some 20-25% radiation and flue temperatures of 1000-1100°C and
- flameless burners: 30-35% radiation and flue temperature of the combustion products leaving the burner bed below 1000°C [30].

The maximum burner load of these burners varies between <100 W/cm² for the conventional pre-mix burners, up to 300-400 W/m² for ceramic surface and flameless burners. Experiments with ceramic burners have even shown burner loads up to 1300 W/cm².

Effectively what is happening with the transition of the traditional free flame burner to the flameless burner, is that the flame is cooled by the burner surface. Or, to put it the other way around, the burner is heated. The figures on the following pages show many variations of these surface burners.

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[30] Although inside the burner the flue temperatures may be much higher.

Figure 4-1. Selected metallic surface burners.
Atmospheric burners, steel plate [*top row*]:
[top row left]: round, conventional [*left*],
[top row mid]: oval suitable for full pre-mix without fan, lower NO_x
[top row right]: cylindrical, optimised for use with gas-fired storage water heaters.
Pre-mix burners, steel plate & metal fibre [*mid row*]:
[mid row left]: round, refractory steel pre-mix burner, modulation range 1:10, emissions akin to Gaskeur SV/Blue Angel level
[mid row mid]: flat pre-mix burner using metal fibre media, modulation range >1:10, emissions below Gaskeur SV/ Blue Angel (i.e. < 40 mg NO_x /kWh), burner bed dimensions: 70 x 237, 80 x 355 or 90 x 237 mm (or custom made)
[mid row right]: cylindrical pre-mix metal fibre burner, e.g. diameters 63/67, height <400mm.
Pre-mix burner, knitted metal fibre [*bottom row*]:
Compact pre-mix burner, knitted metal fibre welded on foot, optimised for standardisation, low NO_x, CO, noise (no resonance).
[source: http://www.bekaert.com/ncdheating/Home.htm]

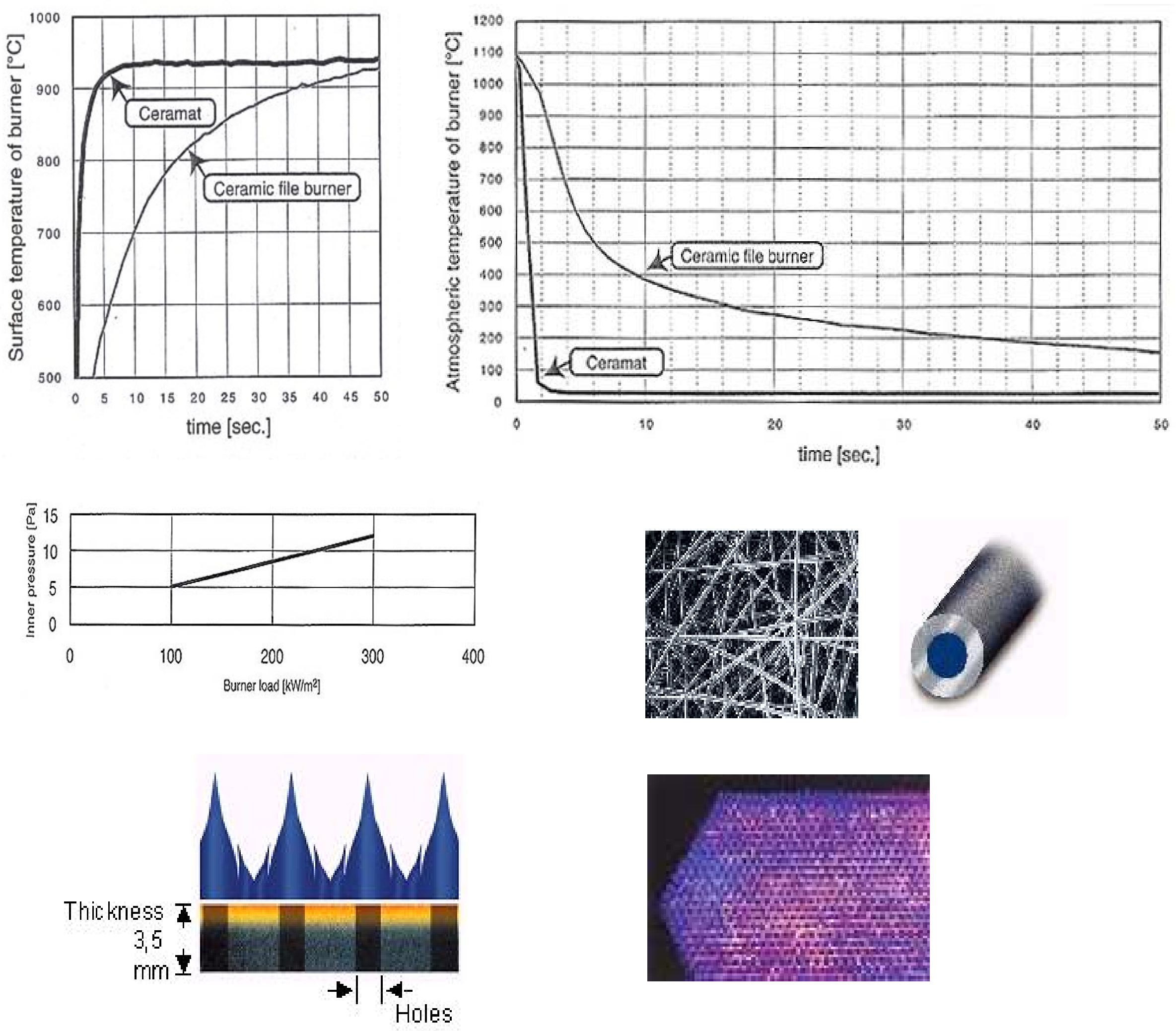


Figure 4-2.
Ceramic radiation burner. Standard size is 250 x 250 mm. Thickness 3,2 +/- 0,5 mm. Standard perforation is 1,4/2,8 or 1,5/3,5 (other sizes and perforation on request. Operating range: Min. 10 W/cm², Max. 400 W/cm². Radiation range: 10 - 75 W/cm². Modulation range 1:35. Maximum surface temperature 1000 ° C. **Pictures** refer to short heat-up time 2-3 s (*top-left)*, quick cool-down 2-3 s(*top right*), low pressure drop +/- 10 Pa (*mid left)*, ceramic fibre material covered with SiC through CVD/PVD-process (*mid right*), small burner plate height of 3,5 mm with holes 1,5 mm → permeability 95% (*bottom left*), front view of burner in action (*bottom-right*) (http://www.schott.com/gasburnersystems/english/)

Figure 4-3.
Radiation burner Viessmann 'off' (left) and in operation (right). Emissions in boilers NO_x: <15 mg/kWh, CO: <15 mg/kWh

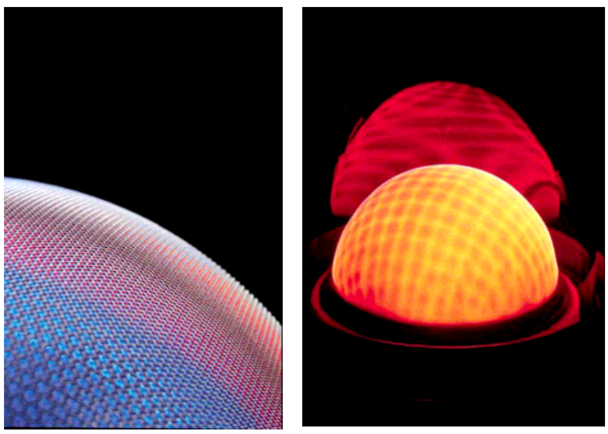


Figure 4-4a.
Porous ceramic burner (SiC). Maximum load: 300 W/cm². Modulation range: 1:20. Thickness 15 mm. Low CO: <20 mg/kWh and low NO_x: <20 mg/kWh, also during burner cycling operations (on/off). Picture left: www.poreos.com

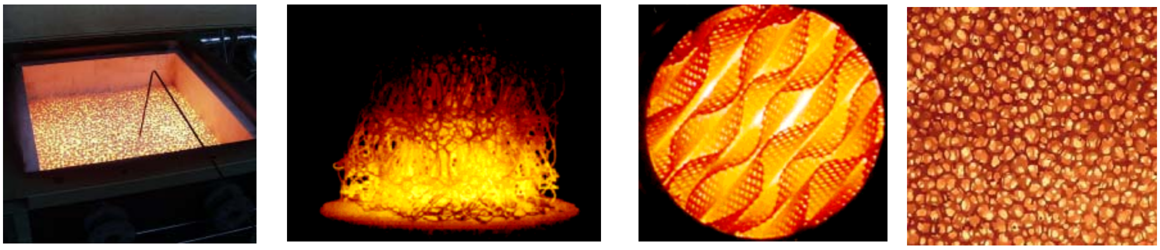


Figure 4-4b.
Ceramic 'flameless' burners. From Left to Right: Ball burner (D. Kugelschüttung), knitted ceramics, mixing/woven burner, ceramic foam [source: Dietzinger, 2006]

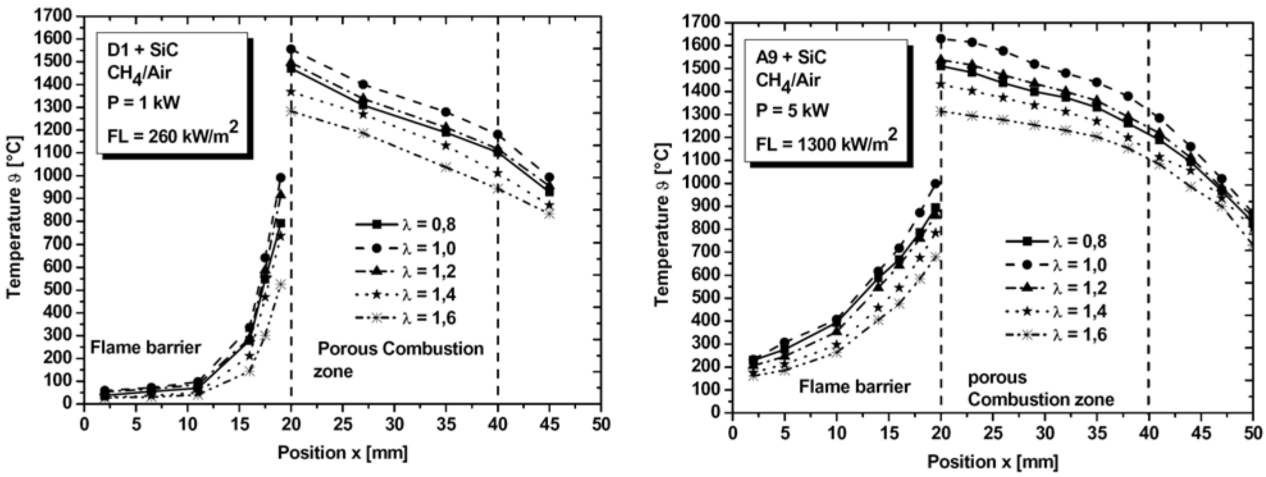


Figure 4-5.
Ceramic porous burner: Propagation of temperature with a methane/air mix. The graphs show an experiment whereby the temperature is measured in the flame barrier and throughout the thickness of a 20 mm porous ceramic burner. Note that the initial temperature after ignition is close to the calculated adiabatic flame temperature and that the combustion products –while giving off their heat to the burner—cool down to a level <1000°C already 10 mm after the burner surface. Left= 1 kW; Right= 5 kW with the same burner but different flame barrie s [source Dietzinger 2006]

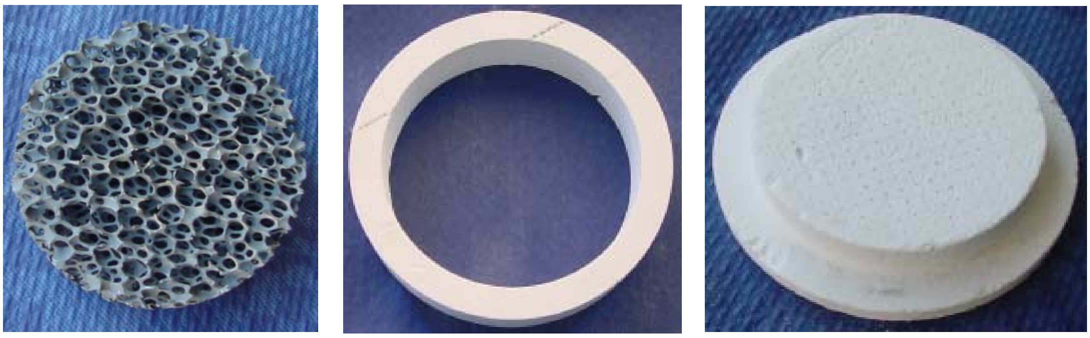


Figure 4-6.
Components of an experimental porous ceramic burner. From left to right: Burner bed in silicium carbide (SiC) foam 10 ppi produced by fa. Erbicol, Al2O3 fibre-based insulation ring, Al2O3 fibre-based hole plate for ceramic burner.
[source: **Diezinger, Stefan**, *Mehrstofffähige Brenner auf Basis der Porenbrennertechnik für den Einsatz in Brennstoffzellensystemen,* dissertation Technical Faculty of the University of Erlangen Stuttgart, 2006 (http://www.opus.ub.uni-erlangen.de)

Figure 4-2 shows a thin ceramic radiation burner from the German manufacturer Schott. In this design the flames typically sit on top of the burner bed (radiation burner). The graphs show that this particular ceramic fibre burner has a quick heat-up (<5-10s) and cool-down (<2s) compared to competing burners.

Another radiation burner, made of a semi-spherical mesh of stainless steel, is shown in Figure 4-3 (production Viessmann).

Figure 4-4 relates to several types of ceramic 'flameless burners' and in particular a burner made of porous ceramic foam, developed by the University of Erlangen and marketed by the firm Poreos in Germany. The burner can be very compact (high heat load per surface area) and have low NO_x and CO emissions (< 20 mg/kWh = 12-13 ppm at 3% O2). However, because of its price and long heat up time it is probably more suited for industrial applications than for residential boilers.

An interesting feature of the thick porous ceramic burner is the fact that the temperature curve through its 20 mm section can be studied. Figure 4-5 shows two examples of such a temperature curve, showing that —although the measured 'combustion temperature' at the burner exit may be as low as 1000°C, in reality inside this flameless burner much higher temperatures of around 1500°C are reached. Figure 4-6 shows the components of the porous ceramic burner.

4.3.2 Jet burners

In principle, a jet burner is nothing more than a nozzle for the fuel/air mix. In case of an atmospheric burner the nozzle and preceding induction trajectory creates a venturi effect through which the fuel sucks in a part of the combustion air (the primary air), after which the rest of the air (secondary air) is sucked in by the flame itself. In case of a full-premix burner, the fuel and all the combustion air are already fully mixed in the right proportion before they are being conducted through the nozzle. A pre-mix jet burner usually requires a fan, which —together with the gas valve, ignition and combustion controls— sits in a self-contained unit, which is then often referred to as 'jet burner'.

Figure 4-7 gives an illustration of a jet burner. This particular jet burner is oil-fired, which means that apart from the combustion head, the fan, ignition and combustion control it also contains an oil pump and atomizer to induce the oil droplets into the air stream. Details of the oil nozzle are given in Figure 4-9.

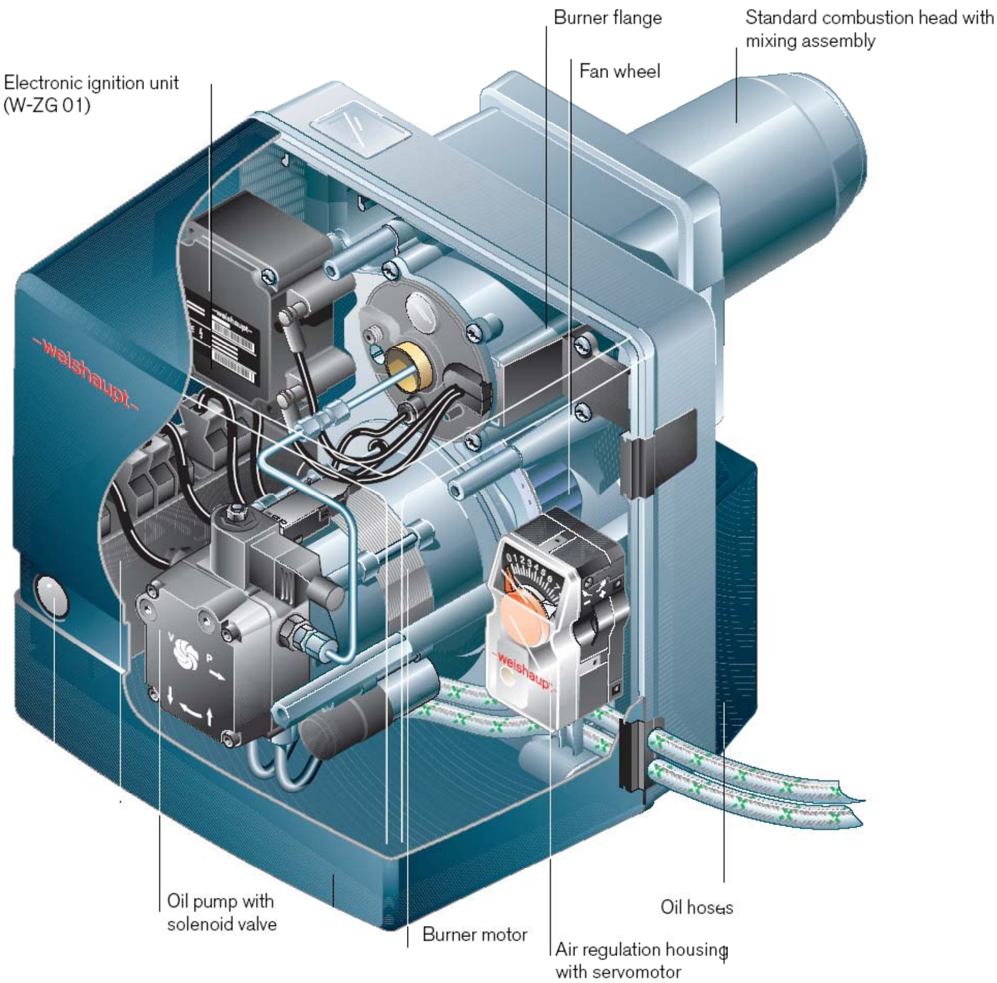
As mentioned, jet burners are fully self-contained and can be mounted on any heat exchanger body with EN standardised attachment for the burner flange (see Figure 4-

7). The units cost around € 800 to € 1200,- for the 15-30 kW range and around € 1500,- or more for 100 kW (prices Germany, incl. VAT 16% [31]).

Figure 4-7.
Jet burner assembly
1 = heat exchanger body with control unit
2 = jet burner
3 = indirect cylinder (sanitary hot water)
4-6 = options

(www.viessmann.com)

Figure 4-8.
Weishaupt oil-fired jet burner WL5. Capacity: 16,5-50 kW. Dimensions excl. combustion head 292 x 286 x 308 mm (www.weishaupt.de)



[31] www.heizungsfachshop.de

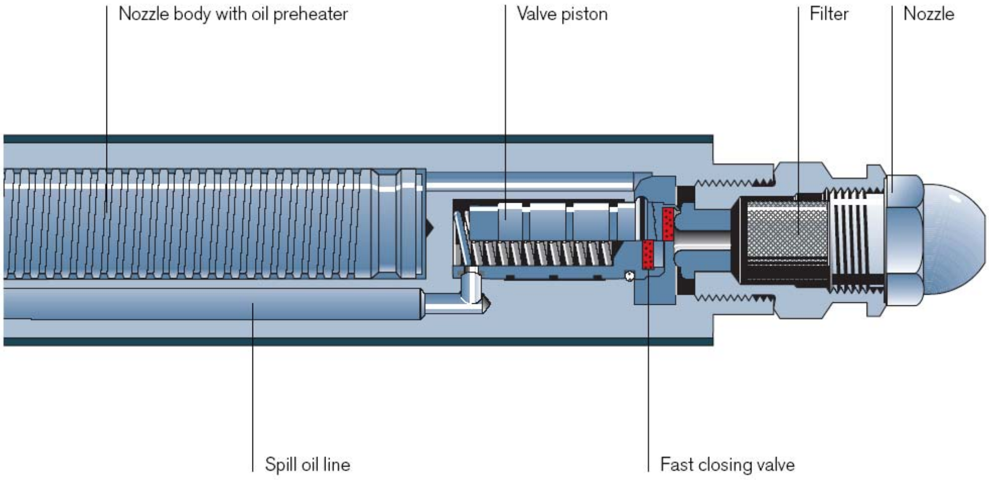


Figure 4-9.
Detail of oil nozzle of Weishaupt oil-fired jet burner WL5. Showing nozzle body with oil preheater, spill oil line, valve piston, fast closing valve, filter and nozzle.

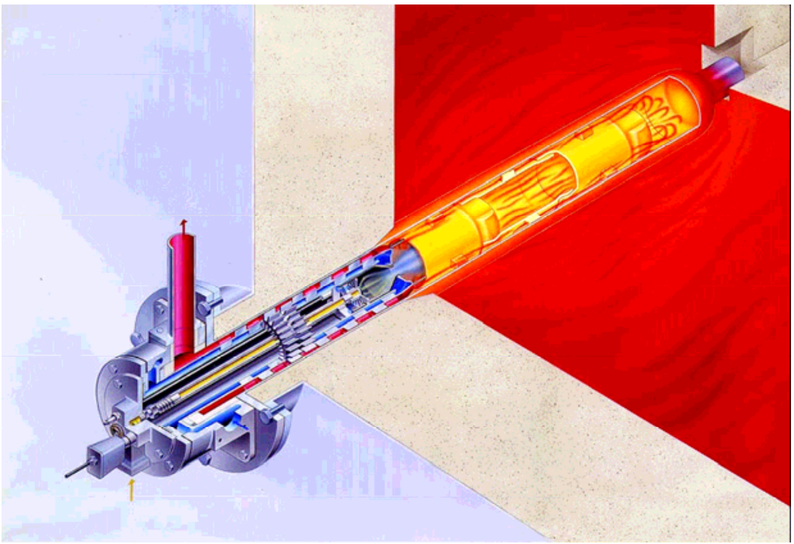
Market trends in the field of jet burners for CH boilers (Weishaupt, Buderus, Riello) seem to go on reliability, design, electronic controls, etc..

In the field of higher energy efficiency and low-emissions most innovations seem to be in the field of industrial burners in Europe. Main developments are in the use of combustion air, i.e. not only through the air factor but also by preheating the incoming air in any number of ways or by mixing the incoming air with the combustion air. These techniques are known as

- recuperator burners (preheating incoming air at burner level),
- regenerator burners (using a heat storage medium and intermittent operation to exchange heat between flue gases and incoming air),
- FLOX burners (high re-circulation rates of flue gases with flameless oxidisation)
- Multi-stage combustion (below-stochiometric pre-combustion)

Especially in the field industrial jet-burners there are new developments regarding the realisation of the cooler flame through recuperator or regenerator techniques. With ***recuperator***-burners the cooler flame and the energy saving is achieved by using/ pre-heating the incoming air with the combustion products.

Figure 4-10.
Recuperation burner whereby there is a heat exchange between incoming air and the outgoing flue gases, allowing the air to be preheated up to 1000°C. The reaction temperature is around 1400°C.

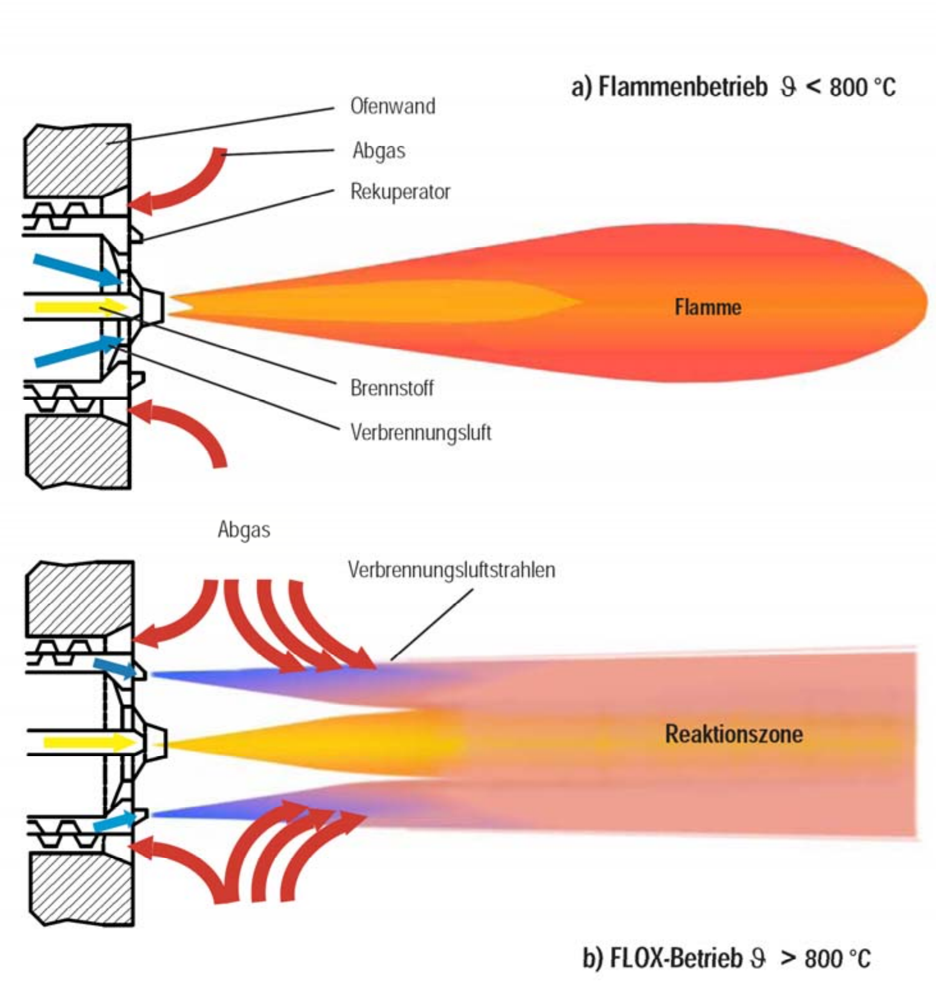


With ***regenerator*** techniques the waste heat recovery is achieved through an intermediate heat storage medium that is intermittently cooled by the air and heated by the flue gases. This technique requires high valve activity and a very integrated construction, which is probably beyond the scope of most domestic burners/boilers.

Another technique is the ***re-circulation of flue gases*** into the combustion process. This has been explained in the chapter on emissions and entails either recirculation rates of 1 with conventional flame technology, or recirculation rates of 3 and higher with the flameless oxidation (FLOX) technology.

The flue gas re-circulation technique, subdivided between internal and external re-circulation, can also be combined with other technologies that reduce NO_x emissions. One such technique is the ***staged combustion*** (D. *Gestufte Verbrennung*), whereby the fuel-air mixture is first combusted at below-stochiometric conditions (air factor < 1) in a pre-combustion chamber and then brought into the main combustion chamber where secondary air is added. This also leads to a reduced flame temperature and lower NO_x. This effect can be vastly increased by a combination with the FLOX operation, where the high flue-gas re-circulation is achieved in two ways: firstly by the impuls of the gas jet and secondly by a delay in mixing the combustion air with the fuel. [32] This is shown in the Figure 4-10.[33]

Figure 4-11.
Principle of a FLOX-recuperator burner, using a conventional flame-mode during start-up and a flameless oxidisation at normal mode [source Erdgasbericht 01/3.]



[32] Note that in a FLOX there is no flame and therefore the conventional UV or ionisation flame sensors cannot be used. Instead the temperature in the combustion chamber is used as a parameter.

[33] Please note that developments in this field are not concluded; especially in the field of emissions of Particulate Matter (PM) with oil-fired FLOX-burners some problems have been reported.[Ökozentrum Langenbruck]

5 HEAT EXCHANGERS

5.1 Introduction

For boiler manufacturers the developments in heat exchangers over the last decades can be characterized firstly by the optimization of the standard cast iron (primary) heat exchangers for floor standing LT-boilers, by improving the heat exchange performance (up until the dew point of flue gas) and by improving the material resistance for corrosion. Manufacturers however, not only improved upon the floor standing standard boiler – and by doing this they prolonged the life of the cast-iron HE—, most of them also developed new light weight heat exchangers and started to apply other materials than cast-iron (being not the best process/material-combination for light weight heat exchangers). Main reason for this was the clear market trend towards wall hung modulating boilers with efficiencies up until dew point (< 90% GCV). This represented another reason for re-evaluating and redesigning the cast-iron heat exchanger. Light-weight materials and heat-exchangers became the preference, and the application of the cast-iron heat-exchanger (non corrosive alloys) remained in the segment of floor standing LT-boilers plus the shrinking market for standard boilers.

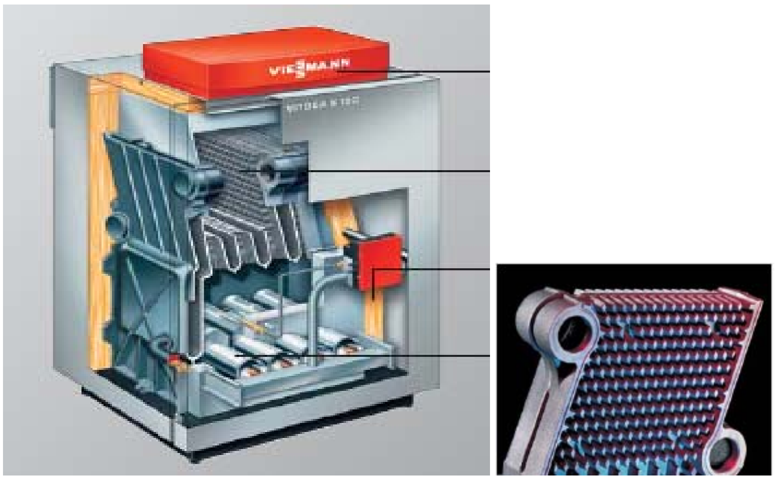


Figure 5-1
Vitogas 100 kW22. Floor standing atmospheric gas fired LT-boiler with cast-iron heat exchanger; Viessmann. Part load eff. 85% (GCV), Net weight: 119 kg. Water content: 9,7 l.

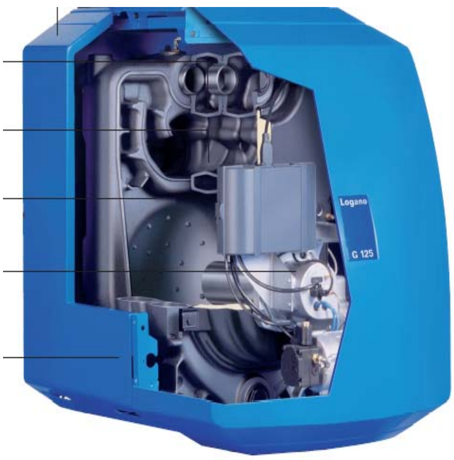


Figure 5-2.
Buderus Logano G125 21kW.
Floor standing oil fired LT boiler with cast iron heat exchanger. Part load eff.: 90% (GCV),
Net weight: 175 kg. Water content 33 l.

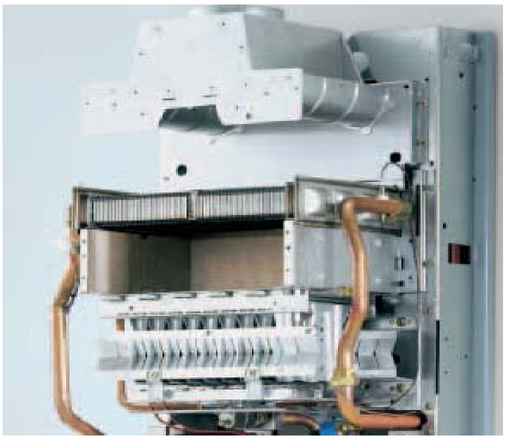


Figure 5-3.
Vaillant atmoTEC VC194 (22kW). Wall-hung atmospheric modulating gas boiler, with light-weight finned plate heat exchanger. Efficiency According DIN 4702 part 8: 85% (GCV) Net weight: 37 kg

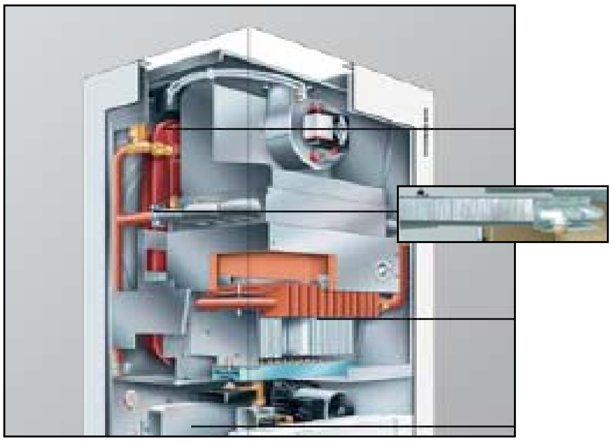


Figure 5-4.
Viessmann Vitopend 200 (24 kW). Wall-hung premix modulating fan assisted gas boiler, with light-weight finned tube heat exchanger. Part load eff.: 85% (GCV). Net weight: 48 kg. Water content: 0,55 l.

A second important trend that characterizes the last two decades is the integration (or combination) of the sanitary hot water heat exchanger with the CH- heat exchanger. A lot of different approaches were used, varying from instantaneous appliances with a sanitary HE within a CH-HE, to combinations of both were the sanitary HE is no more than a small plate HE or tube HE in a small storage tank, to solutions were large storage tanks are used, either for CH of sanitary hot water.

The third element that is typical for the HE- development trends, is the optimization op the primary heat exchanger beyond the dew point of the flue gasses (condensing boilers). Secondary heat exchangers were integrated (gas boilers) or added (floor standing oil boilers) to the primary heat exchangers, and again non corrosive light weight materials were preferred.

More companies started to outsource the development and production of these condensing heat exchangers, mainly because –coming from cast iron primary heat exchangers— the knowledge and hands-on experience needed for the design and manufacturing of these new type of integrated light-weight and condensing heat exchangers were not always available within the company.

Boiler manufacturers without the historical burden of a foundry obviously took the lead here, because they could fully concentrate on the condensing boiler only.

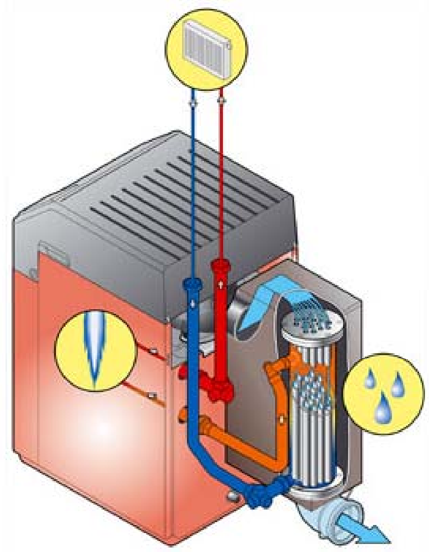


Figure 5-5.
Weishaupt Thermo Unit, WTU 25 GB (25 kW). Floorstanding condensing oil boiler with external secondary (ceramic) heat exchanger. Part load eff.: 96% (GCV). Net weight: 268 kg. Water content: 40 l.

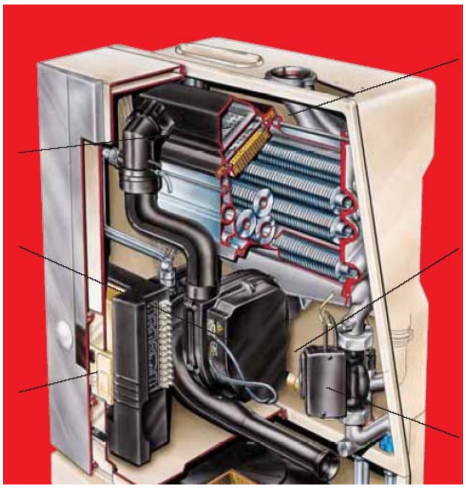


Figure 5-6.
Nefit Ecomline Excellent HR 30 (28 kW)
Wall-hung condensing premix gas boiler, with aluminium finned pipe heat exchanger. Part load. Eff. 97% (GCV). Net weight 59 kg. Water content: 2 l.

The last decade can be characterized by a further optimisation of the different he-solutions that were selected by the various boiler manufacturers, meaning:

- further optimisation of steady state efficiency (EN 303)
- reducing maintenance cost (by improving material specs)
- cost-price optimisation by a further integration of functions within the HE-assembly (integration with burner, air-vent, flue ducts, condensate collector and piping)
- cost-price optimisation through improvement of component commonality throughout the product range and through rationalisation of production.

5.2 Typology

In principle a heat exchanger is a thermal device in which heat is exchanged between media. The three basic principles for heat transfer are:

- Direct
  Direct contact between two media (e.g. steam or gas through water).
- Regenerative
  Heat is transferred through an intermediate material that cycles between receiving and transferring heat; (e.g. electric emitters with thermal store or warmtewiel)

- Recuperative
  In a recuperative he the media are always separated with a thin wall through which the heat is transferred, mainly through convection and conduction. The influencing parameters are A (= size of surface), the shape of the surface, thermal conductivity of the material used, speed and flow characteristics of the media, direction of the flow (counter, cross or parallel flow).

For boilers applications the recuperative heat exchanger us predominantly used. However, to illustrate that a direct contact heat exchanger technically also is feasible, the principle of a prototype that achieved a constant thermal efficiency of 96% is shown below.

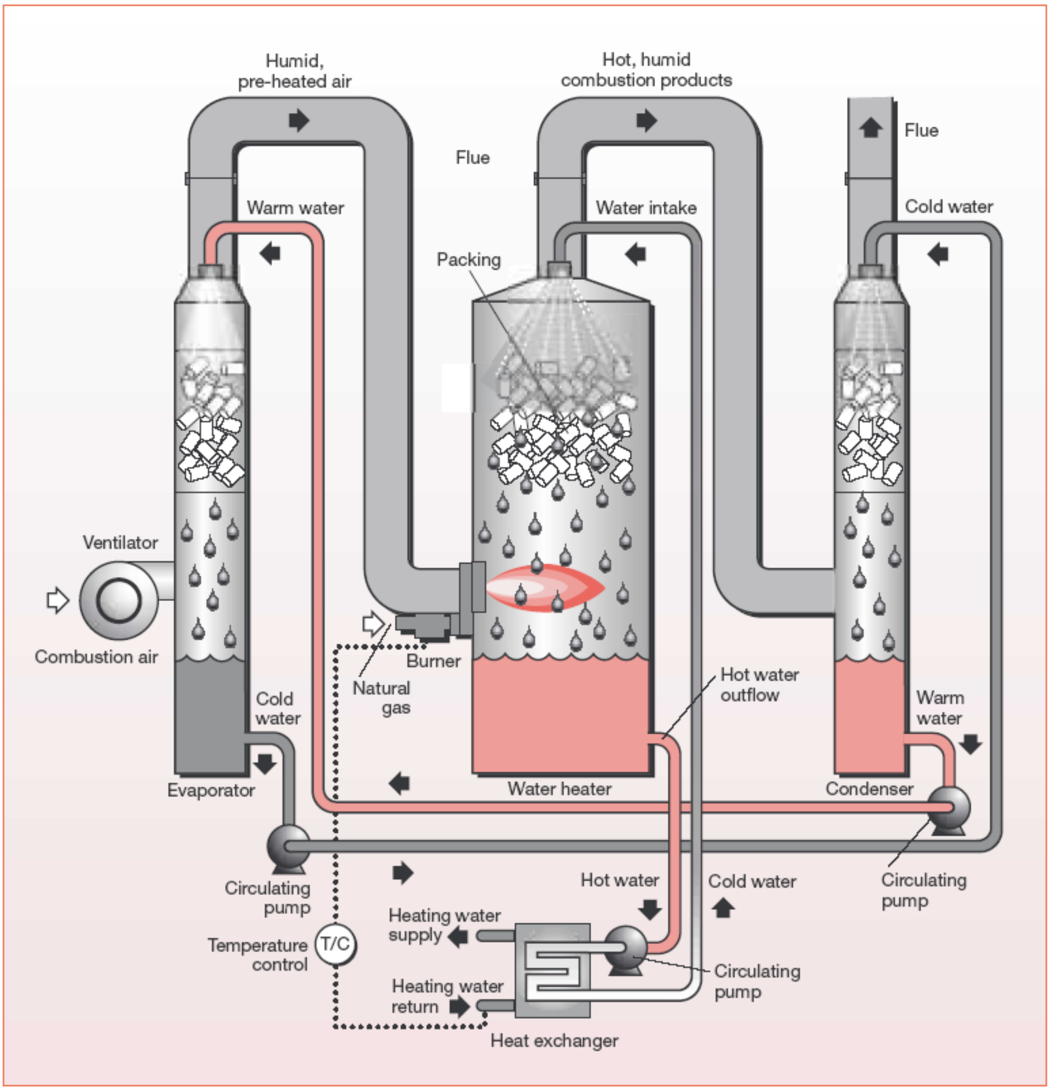


Figure 1: Schematic diagram of the direct contact heat exchanger system.

Figure 5-7. Schematic diagram of the direct contact heat exchanger system. Source: Caddet Energy Efficiency projects, Result 438.

In the following paragraphs the recuperative heat exchanger will be further analysed in terms of its design aspects and application in boilers for primary, secondary and tertiary heat exchangers. In this type of heat exchanger, heat is transferred through a combination of the three mechanisms: conduction, convection and radiation. In heat exchangers for boilers, convection is the most important part in total heat transfer (appr. 60 – 80%, see chapter..). Depending on the type of burner/heat exchanger configuration, the heat transfer through radiation may vary from 5 to approximately 25%.

To give some more detail on the energy transfer processes, the general formulas mentioned in the previous chapter (Basic energy and mass balance) are elaborated on (see box below).

The total heat transfer coefficient “U” of a heat exchanger through *convection* can be expressed with the following formula, (calculates to total heat transfer resistance).

$$1/U = 1/\alpha_h + d/\lambda + 1/\alpha_c + R_f$$

In which:

α_g = heat transfer coefficient on the gas side of the HE [W/(m²K)]
d = wall thickness [m]
λ = thermal conductivity of HE material [W/(mK)]
α_c = heat transfer coefficient on the cold side of the HE [W/(m²K)]
R_f = heat transfer resistance caused by corrosion & pollution [W/(m²K)]

The total heat transferred through convection can be calculated with:

$$Q_{conv} = U * A * (T_g - T_c) \quad [W]$$

The total heat transfer through radiation of the burner towards the HE can be expressed with the formula:

$$Q_{rad} = \psi_{b\text{-}he} * A * \varepsilon_{res} * \sigma_s * (T_g^4 - T_w^4)$$

In wich:

Q_{rad} = the radiation heat energy [W]
$\psi_{b\text{-}he}$ = exchange factor between burner surface and HE-surface [-]
A = the surface of radiating part (burner in this case) [m²]
ε_{res} = the resulting emission-factor [-]
σ_s = the constant of Stefan-Bolzmann: **5,67**. 10^{-8} [W/ (m²K⁴)]
$T_{g,}$- T_w = temperatures of the gas and the wall in [K]

Indications for the exchange factor $\psi_{b\text{-}he}$ can be calculated with the absorption factor method of Gebhart (not further explained here), and largely depends on whether both surfaces can “see” each other and on the emission factors of both materials.

The design parameters for optimising the heat-exchange process are:

- thermal conductivity (λ) of the material used
- wall thicknesses
- surface area (the bigger the better)
- flow characteristics (on both sides of the heat exchanger)
- burner/he – configuration

For the overall boiler design however, other design aspects need to be integrated here, amongst which:

- HE- weight
- HE- size
- Reaction time HE on changing heat loads
- Corrosion / foul-up / maintenance

Following paragraphs give a further explanation on these design parameters.

5.3 HE – material

Apart from cast-iron, the other materials that are predominantly used for *primary and secondary* heat exchangers in boilers and combis, are aluminium alloys, (stainless) steel and copper alloys.

The thermal conductivity λ varies quite a lot: stainless steel 27 [W/mK], cast iron 60 [W/mK], aluminium 237 [W/mK] and copper 390 [W/mK]. The advantage of stainless steel over cast iron is, that wall thickness can be reduced to far below 1 mm, while with cast-iron approximately 2,5 mm is the minimum. Since heat transfer also depends on the wall thickness and total surface, steel is the better material when size, weight and cost need to be optimized. For this reason steel can also compete with cast aluminium. Another advantage of (stainless) steel is its resistance to corrosion and thermal cycling. Copper has the best thermal conductivity and can be produced – like aluminium and steel – in thin plates or strips. Copper is also commonly used for sanitary (hot water) application (including heat-exchangers); main drawback is the price per kg (approximately 3 to 4 times higher than stainless steel).

For floor standing boilers the materials are cast iron or steel or a combination of both, in most cases combined with yet-burners. For the smaller and lighter wall-hung boilers aluminium, steel (finned tubes) and copper are mostly commonly used.

For *tertiary* heat exchangers (flue-gas/combustion-air he) plastics can be used because temperatures of flue gasses are below 90°C. Since with plastic the wall-thickness of the material between the two flows can be reduced to below 0,3 mm. the heat transfer between flows becomes less dependent on the thermal conductivity of the material itself. Plastics (e.g. PP) are then a good option, because they have a very good chemical resistance.

HE-manufacturer Giannoni SAS integrated a tertiary heat exchanger in its condensing HE-design, and uses only plastic for the casing. The tertiary HE itself is made off metal strip.

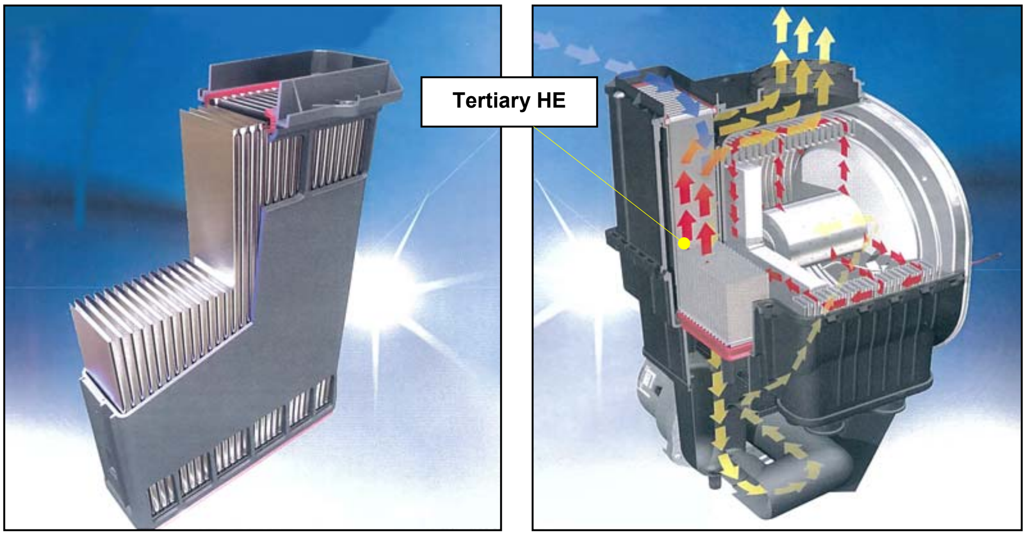


Figure 5-8.
Heat Exchanger from Giannoni, with a stainless steel primary- and secondary heat exchanger and an integrated tertiary heat exchanger. Giannoni S.A.S. / Aeropole Centre, 29600 Morlaix France / www.giannoni.fr

The air to air heat exchanger is positioned between the combustion air intake and the flue gas outlet. It provides continuous condensing operation, regardless of the water temperature regime used. The company claims that fume temperatures are always lower than 55°C, and that it also reduces plume production at the outlet.

The German company Götz Heizsysteme GmbH, uses a thermoformed plastic heat exchanger for its floor standing oil or gas boiler, carrying the name “ProCondens”.

With this plastic tertiary HE the combustion air is preheated to approximately 60°C and the flue gasses are cooled down to around 40 – 50°C. As a result flue gasses will condensate also with higher boiler water return temperatures.

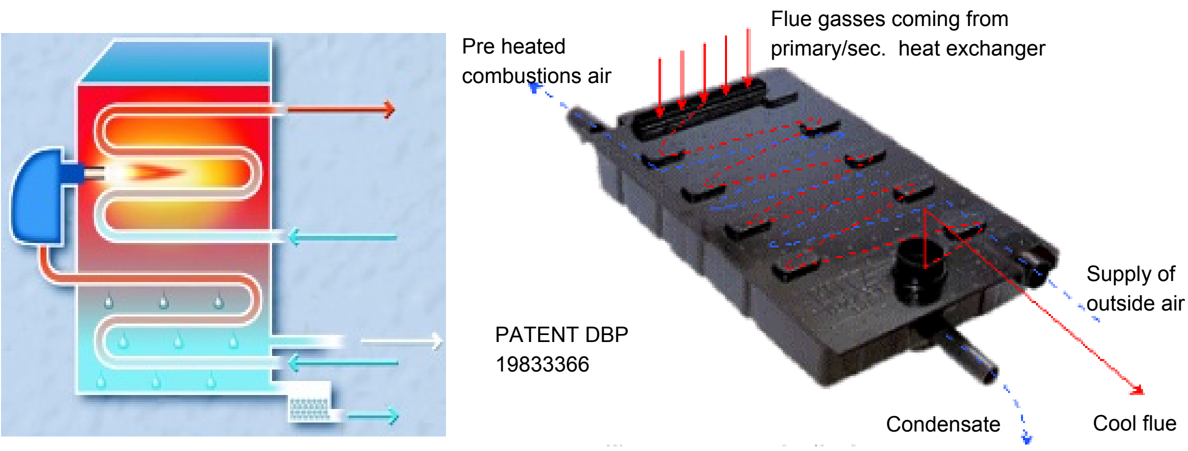


Figure 5-9. ProCondens, from Götz Heizsysteme GmbH, www.procondens.de

5.4 HE – shape and design

Apart from the material (λ) that is used, the shape and surface of the HE plays an important role in the optimization of the heat transfer through convection and radiation.

Shape and overall design however strongly depends on the basic (semi finished) material that is used. This can be tubes, plates or the raw material being casted in the requested shape.

5.4.1 Heat exchangers based on tubes (finned and un-finned)

The traditional type of heat exchanger based on un-finned tube is the "Shell & Tube" heat exchanger, in which a tube- or pipe-arrangement is placed within a shell that contains the connections for the two flows. Flue gasses are guided through the space between the tubes, while boiler water flows through the pipes. In the industry, this still is the most applied type of heat exchanger. But also for small scale heat generators for residential applications, all kinds of variations on this type of HE are commonly used.

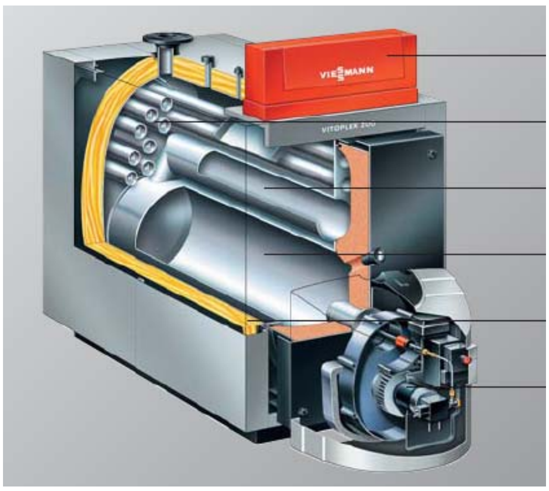


Figure 5-10.
Viessmann Vitoplex 200 / 90kW.
Example of a shell & tube heat exchanger in a floor standing boiler gas fired LT-boiler.
Part load eff. 85% (GCV). Net weight 345 kg. Water content 180 l.
Source: Viessmann

For smaller light weight appliances other configurations are used, where the tube-part consists of one long spiral formed tube, that is placed in a casing (the shell) to which also the burner is attached.

This configuration is not only used in the traditional kitchen water heater (geiser), but also in the latest high efficient condensing boilers.

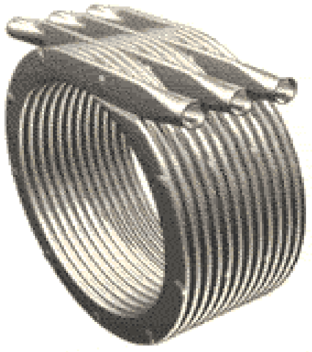
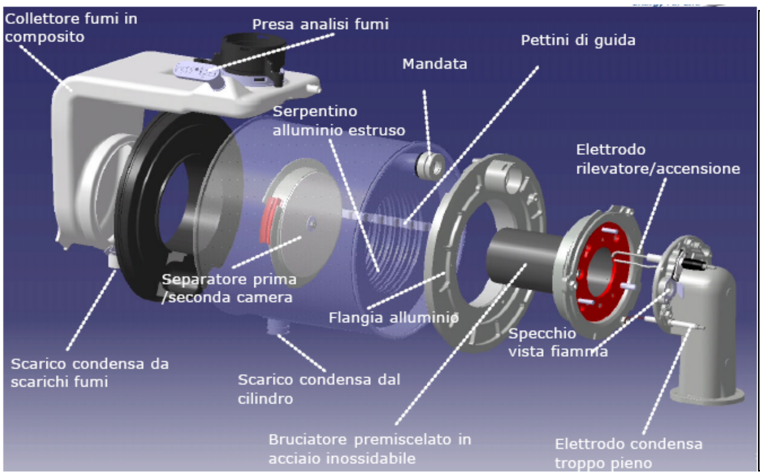


Figure 5-11.
Spiral flat-tube heat exchanger
Made of 0,8 mm stainless steel, consisting of 3 hydroformed coils of each appr. 8 kW (source: Giannoni).

Figure 5-12.
The casing (or shell) of the spiral flat tube exchanger. Casing holds the 4 connections for the two media (flue gas and water) and mounting plate for the concentric burner (source: Giannoni).

The secondary heat exchanger is an extra coil, that is divided from the primary coil and burner by a well insulated deflector disc. The cold return water enters the HE in the last segment of this secondary coil and exits the HE at the first segment of the primary coils surrounding the burner. Because of its configuration and material, a relatively large part of the radiation heat is transferred to the boiler water, which can reduce the amount of material needed compared to a heat exchanger with mainly convective heat transfer.

This type of heat exchanger is being applied by several condensing boiler manufacturers, to name a few: Remeha (Avanta en Aquanta), Vaillant ecoTEC, Viessmann Vitodens.



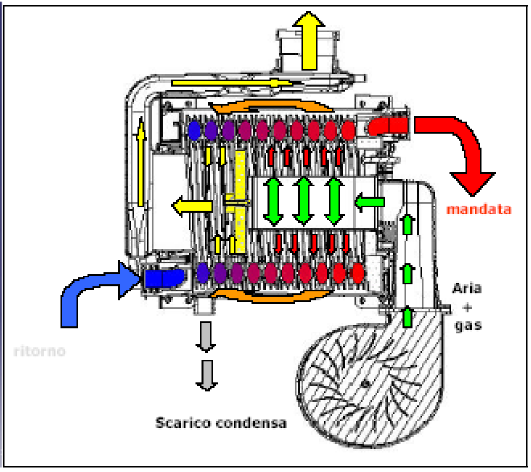


Figure 5-13. Spiral tube Heat Exchanger from Riello Group, Italy.
Spiral tube is made of an aluminium extrusion profile.

Finned tube heat exchangers are probably the most commonly applied for light-weight wall-hung boilers and combis. The fins are added to the tube/pipe to increase the heat transfer through convection

on the gas-side (outside) of the tubes. To improve the heat transfer on the inside of the tube (water-side) groves can be applied.

Several techniques are used to manufacture the finned pipes, like brazing, welding (high frequency /resistance), rotary extrusion (in case of aluminium) etc.

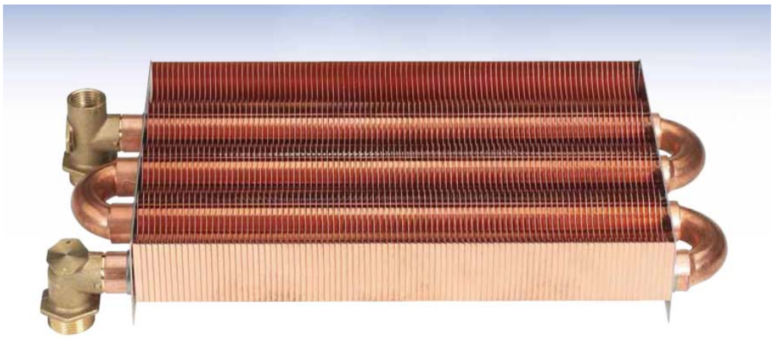


Figure 5-14.
Most commonly applied finned tube heat exchanger for wall-hung non condensing boilers, combis and water heaters. Fins or plates are brazed unto the copper pipes, the surface is hardened by shot blasting and painted with a silicone-aluminium mixture. This subassembly is placed within a casing (shell) on top of the burner (source: Fugas Italy).

Other options to apply fins are illustrated below.

Figure 5-15.
Solid fin, in the form of a metal strip: The fin is helically wound around the specified pipe/tube and continuously fillet welded to the tube using the M.I.G. weld process (source: Tex-Fin, USA).

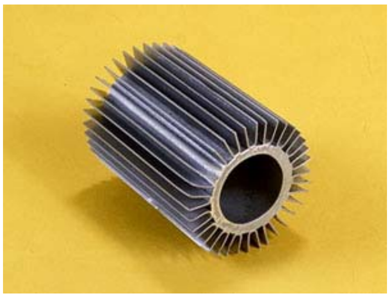
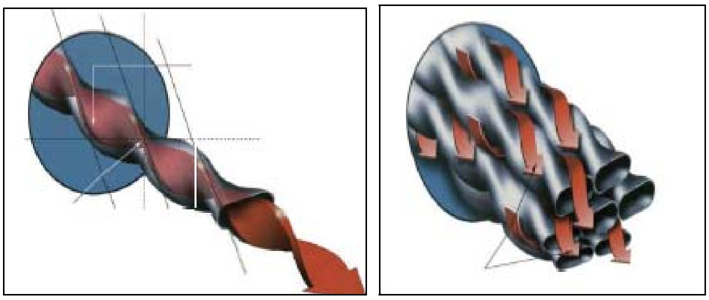


Figure 5-16.
Longitudinal fin: Fin in the form of a U-shaped fin channel, is resistance welded along the tube's longitudinal axis (source: Tex-Fin, USA).

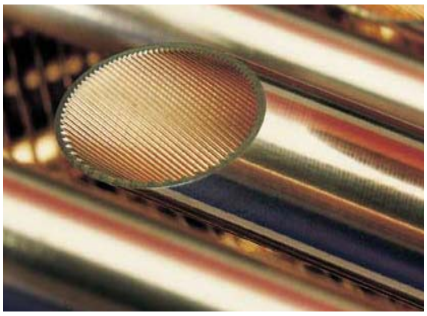
Figure 5-17.
Serrated Fin: A metal strip that has been serrated or cut and then helically wound around the specified tube. The fin is welded to the tubular base using a high frequency weld process (source: Tex-Fin, USA).

Figure 5-18
Extruded fin: This finned surface is formed as a thick walled aluminium tube is put through cold rotary extrusion, forming fins that are much longer in diameter than the original tube. This process hardens the aluminium so the fins are very strong, resulting in good heat transfer efficiency and high durability. It can be applied to single aluminium tubes (mono aluminium) or with the addition of a liner tube within the original aluminium tube (bi-metal) (source: UniFin, Canada)

Figure 5-19
Twisted Tubes^{TM}: It is also possible to twist the whole tube, causing a turbulent flow both inside and outside the tube. According to the manufacturer heat transfer increases with roughly 40% (source: Brown Fintubes).



To improve the heat transfer coefficient on the inside of a tube the following options are available.



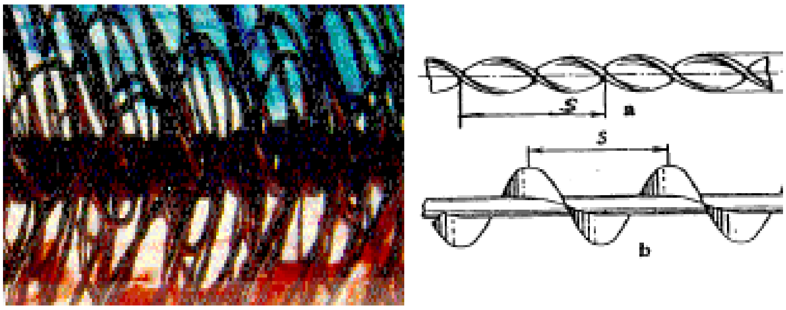
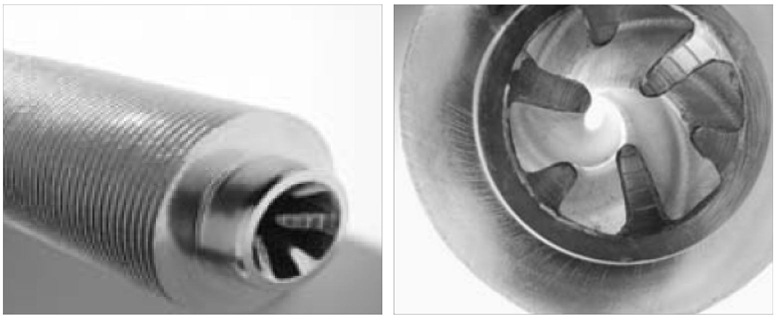


Figure 5-20
Inner grooves: Inner grooved cooper tubes achieve a high energy transfer coefficient inside the tube at low pressure drop.

Figure 5-21.
Inserts: Another way to improve heat transfer on the inside of the tube is to add inserts into the tube that influence the flow characteristics of the boiler water. The left picture is an illustration of an internal wire matrix and the right picture is an example of a “twisted tape” insert.

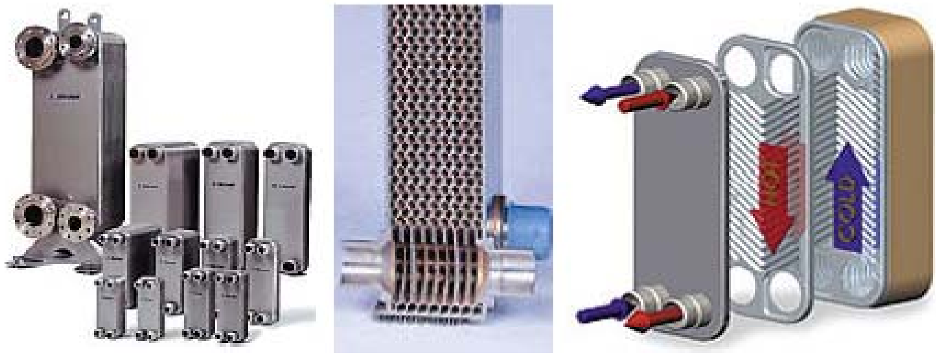
Figure 5-22
Alu finned pipes: Nefit Latest aluminium heat exchanger of Nefit with twisted flow technology (twisted ribs on the inside of the tubes).
In this version of the HE the aluminium ribs are coated to minimize oxidation and pollution of the HE (source: Nefit BV)



5.4.2 Heat exchangers based on plates

A plate heat exchanger consists of several rectangular plates (with a flow pattern pressed into them), that are mounted on top of each other. Between two plates a compartment is created through which the flows are guided. Each plate contains four opening (one in each corner) to allow the flows to enter and leave a compartment. Each medium only flows through half of the total number of compartments, each time skipping one compartment. As a result the two media always flow next to each other, with a heat-transferring plate in between.

Figure 5-23
Pictures of soldered plate heat exchangers (source: ECN, Overzicht commercieel verkrijgbare warmtewisselaars, juli 2001).



These types of compact plate heat exchangers are mainly used for liquid media or media with similar heat transfer coefficients (α). In the boiler industry this type is widely applied for sanitary hot water production, where heat from CH-water is transferred to the sanitary water.

For this purpose, plate heat exchangers are considered the most compact and cost-efficient solution.

This type of compact plate HE is not suited for the heat-exchange of flue-gas to water.

A plate heat-exchanger that is more suited for boiler application is the spiral plate heat exchanger (SHE).

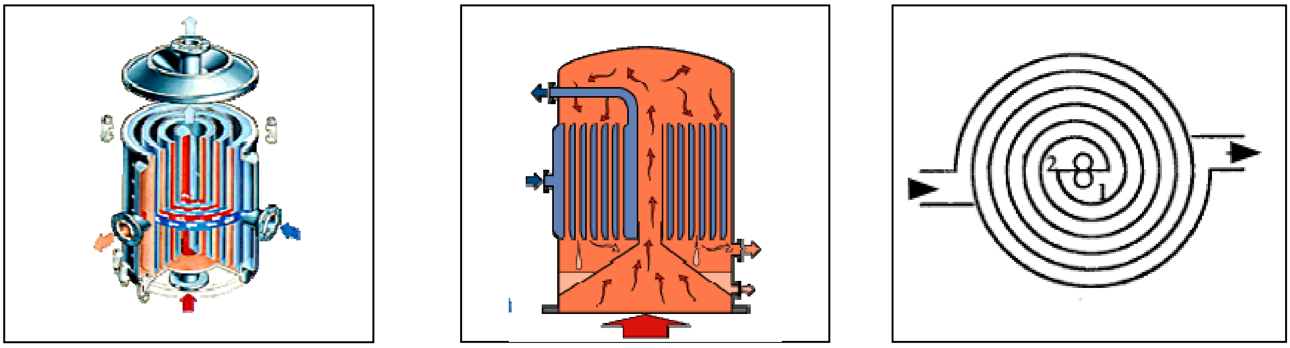


Figure 5-24
Principle of spiral plate heat exchangers (source: ECN, Overzicht commercieel verkrijgbare warmtewisselaars, juli 2001).

Also for plate heat exchangers it is possible to add fins to the surface and achieve higher heat transfer coefficients. Plates and corrugated fins (deformed plates) are then collated together in a sandwich construction.

5.4.3 Heat exchangers based on integrated castings (aluminium, cast-iron)

Various boiler manufacturers use their own integrated casted heat exchanger. Some manufacturers still use cast iron as base material (floor standing boilers) but the many companies already changed to aluminium alloys. The advantage of this approach is that casing and heat-exchanger and all necessary connections can be integrated into the castings, reducing the number of components and assembly times.

Another advantage of this integrated approach is that heat exchanger design can be further optimised for radiative heat transfer, by creating a configuration where the burner surface is fully surrounded by the water containing heat exchanger- surface.

A few design- and engineering companies are specialised in this field.

Figure 5-25.
Integrated aluminium HE by Aluheat.
The company Aluheat (taken over by Bekaert may 2006) designed a new family of condensing heat exchangers. This new product line is available for all boiler manufacturers.
Characteristics
+ available in capacity of 28 kW, 36 kW and 46 kW
+ monobloc casting, so no internal weldings or couplings
+ low water content
+ low hydraulic resistance
+ small compact design
+ fire chamber water cooled, so no ceramic insulation required
+ water channels in full serial water flow
+ smoothened heat transfer through optimised flue and water geometry
+ aluminium; good anti corrosion properties, high heat conductivity, low weight
(source: Aluheat)

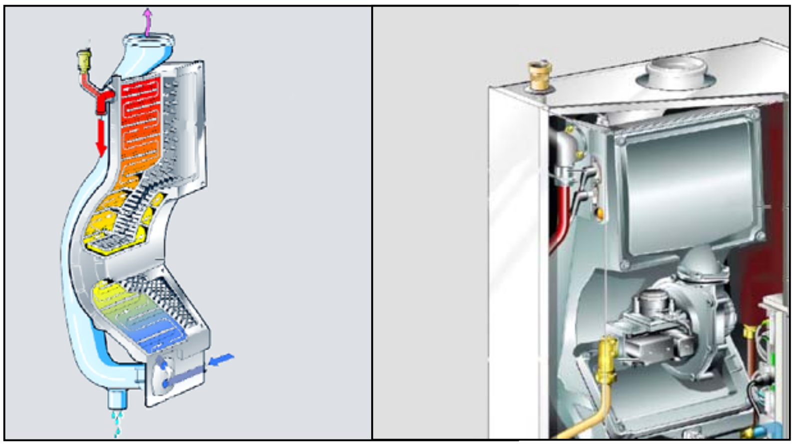


Figure 5-26.
Weishaupt Thermo Condens: Weishaupt uses an integrated al/si-casting as heat exchanger in their new Thermo Condens wallhung boilers. The heat transfer coefficient is optimised for each temperature zone through a dedicated surface design per zone. The half-moon shaped channel prolongs the surface and optimises the heat transfer from the condensate. Combined with the flat radiative burner part-load efficiencies (40/30°C) of 100% (GCV) are reported (EN303) (source: Weishaupt)

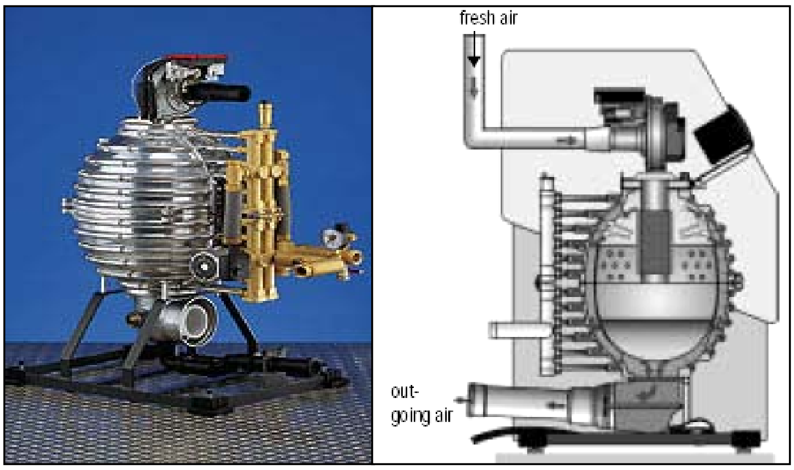


Figure 5-27.
Rotex A1 BG Gas condensing boiler: This floorstanding boiler used a ball-shaped aluminium die-casted heat exchanger in combination with a premix burner. Part load eff. 99% (GCV) Net weight: 74 kg (incl. 49 kg for boiler chassis) (source: Rotex GmbH)

5.4.4 *Other heat exchangers types.*

Concentric tube-in-tube heat-exchangers

A special configuration of the “Shell & Tube” principle in the concentric tube-in-tube HE. In the boiler industry this type of HE is used for instantaneous combi appliances. This HE can both be used as a separate secondary HE for hot water production, or it can be used as the combined primary HE in the burning-chamber, heating both flows (CH- and sanitary water) at the same time.

Figure 5-28.

6 BOILER CONTROLS

6.1 Introduction boiler control &components

Over the years boiler controls have evolved from simple electro-mechanical devices that only control and secure the combustion process (atmospheric burners with a thermocouple controlled gas supply and max. thermostat), to microprocessor controlled systems with solid state sensors, that facilitate premix full modulating type C boilers and secure safe boiler operation, optimize boiler efficiency and maximize heating comfort.

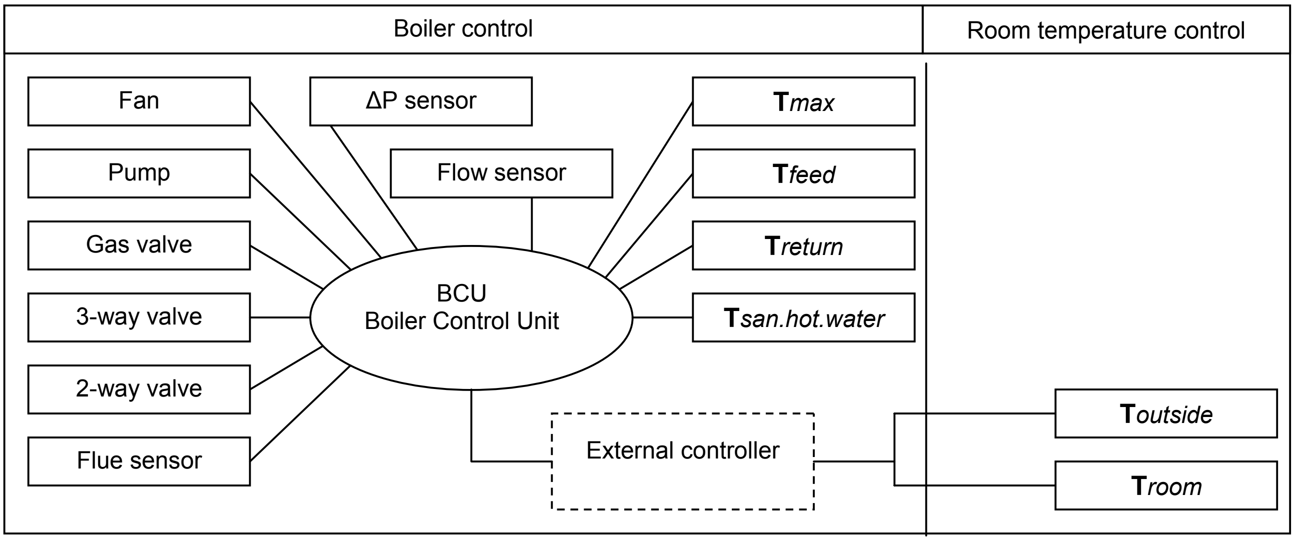


Figure 6-1. Schematic representation of components involved in boiler control and room temperature control systems

A room thermostat or a weather dependent controller gives the information on the requested room- or feed water- temperature and the actual room- or feed water-temperature. On the basis of this information the BCU switches the boiler on/off or high/low (depending on type of boiler). For this task the BCU controls the fan, the pump and the gas valve. With temperature sensors, ΔP and/or flow sensors the process is monitored.

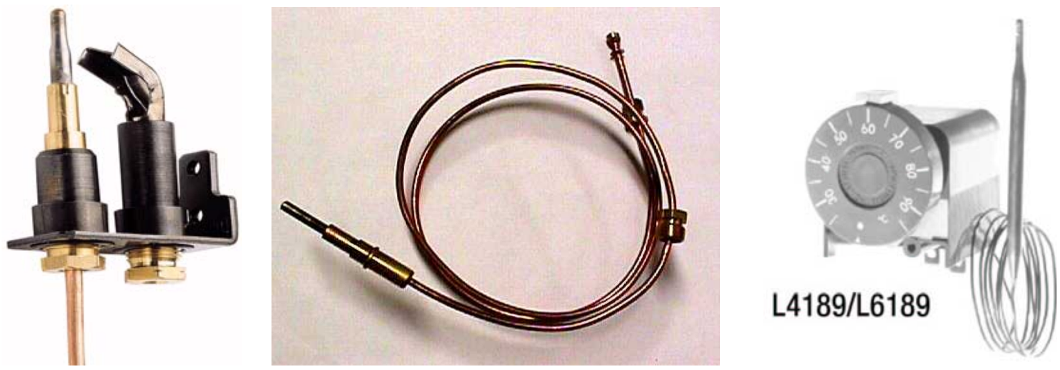


Figure 6-2. Traditional boiler control equipment (Pilot burner, thermocouple and water temperature thermostat)

With state of the art boiler control systems, several parameters are measured in order to operate and control more sophisticated boiler functions, such as automatic ignition, fuel/air-ratio control, power input control, feed temperature control, 3-way valve control (combi-boilers).

For these purposes a low voltage system is applied for the operation and communication of sensors, microprocessor and actuators.

Typical <u>sensors</u> that are used in this context are:

Figure 6-3.
Surface mount NTC

<u>Temperature sensors</u>

Sensors that measure the temperature of fluids, air or gasses. These sensor elements are usually thermistors (NTC) but other types are also possible. Sensors are available as surface mount sensor and as direct immersion sensor, in various shapes and with various NTC characteristics.

T-sensors are used in safety thermostat, and for measuring boiler feed- and return temperature, and in combis the sanitary water temperature.

Figure 6-4.
DHW Water flow sensor

<u>Waterflow sensors</u>

Flow sensors are primarily designed to measure the hot water flow rate in domestic water heater appliances. Most of them are mechanical and use a turbine that indicates the flow with its rotation speed. The turbine supports a magnet which rotates in front of a Hall effect sensor, which results in an electronic frequency that is directly proportional to the water flow through the sensor.

Figure 6-5.
Pressure sensor switch

<u>Pressure sensor switch</u>

Pressure sensing switch is sensor/actuator –combination used in gas combustion appliances in which the combustion air is fully dependant on the fan (as in premix burners). Therefore, the air flow needs to be closely monitored.

Outdoor temperature sensors are used for weather controlled systems, and *Room thermostats* are applied for room temperature controlled systems (see chapter X.3 for further explanation).

Besides these sensors, some actuator components (*Fan*, *Pump*) also communicate information (about their operating status) to the boiler control unit. Typical <u>actuators</u> are – as already mentioned- the *Fan*, *Gas-valve* and the *Pump*. Sometimes a *Water valve* (2- way valve for on-off zone-control or 3- way valve for combi appliances) or a *Flue valve* (valve that opens the flue duct before combustions and closes it after shutting down the burner and purging the appliance) are also used. Not all of the actuators necessarily are within the boiler casing and in that case will be part of the CH-installation instead of the boiler appliance.

The boiler control unit is the key device that processes the data input from the various sensors and actuators and subsequently operates the boiler components on the basis of the control algorithms that are loaded into the unit. However, not all algorithms to control the boiler and actuators are necessarily within the boiler control unit. Signals to start the burner may also come from external control devices such as weather dependent controllers or room thermostats. In that case, the boiler control unit only facilitates a safe boiler operation.

Depending on boiler and type of installation various boiler control systems can be used. Some important trends in boiler control are discussed in the next three paragraphs.

6.2 Power input control

Minimum heat demands for an average house can be lower than 1 kW. An average heat load of 1 – 2 kW is very common for the average house (Felduntersuchung Wolfenbüttel). To be able to deliver these lower heat outputs, a boiler must either be able to reduce its power input (burner load) or use some kind of primary store, or a combination of both. If this is not the case, the boiler will switch on and off every couple of minutes, which not only increases energy-losses and emissions, but also will reduce comfort-levels and the lifetime expectancy of the appliance because of increased thermal cycling of boiler components.

The use of a primary store generally will increase the standing losses, so lowering the burner input seems the preferred option.

Controlling the burner-output between the range of 100% back to 30% of the nominal load, is already quite common for gas condensing boilers. For boiler with a maximum output of 8 – 11 kW this could already be a sufficient turndown ratio. However, the most common type of boiler is a 24 kW combi boiler. The 24 kW boiler output is selected for the sanitary hot water production. As a result, the minimal power input (30% of 24 kW) is around 8 kW and the boiler will still cycle on and off during most of its operating hours.

Several companies are already developing techniques to further reduce the modulation range, preferably up to 10% of nominal load. Since most boilers that are either fan assisted or fully premix, the modulation control not only affects the gas valve, but also the fan.

6.2.1 Pneumatic ratio-control

Most commonly applied technique for modulation is the pneumatic ratio control. With this type of control the BCU (boiler control unit) sets the rotation speed of the fan, based on the requested feed temperature or heat demand. The air flow resulting from this fan speed, causes a specific air pressure that is sensed by a *control membrane* or *venturi* of the pneumatic ratio control unit. Based on this pressure-difference the gas valve opening is adjusted. These control techniques compensates for weather conditions like changes in temperature of barometric pressure.

Pneumatic ratio-control systems operate without problem to modulation ranges up until 1 : 4. At increased control ranges, the resolution of the measured pressure differences becomes too small and the control principle becomes unstable.

Figure 6-6.
Increasing pressure difference in pneumatic ratio control devices.

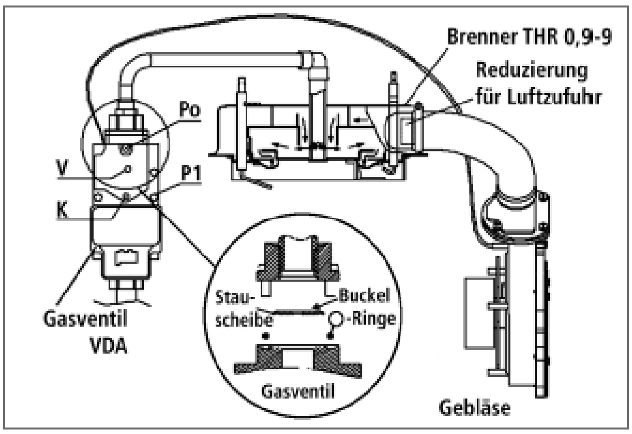


Figure 6-7.
Thision, condensing boiler from Elcotherm, modulation range 9,5 - 0,9kW.

One way of solving this is the use of an extra diaphragm which increases the available air pressure at the pneumatic ratio control unit. With a similar device that increases the pressure over the gas valve in the same proportion, the pneumatic ratio control unit can function again, but now at higher resolutions.

6.2.2 Integrated mixing & control valve

Another technique that is being developed is the IMS control, "Integriertes Misch- und Stellventil". This development project (by Kromschröder, Ruhrgas and Remeha) also aims at improving the modulation range to 1 : 10. The IMS is a system is a combined mix- and control unit, using two valves that are both controlled by a motor. The motor adjusts the position of both valves. The position of the gas valve is based on the requested heat load, the position of the air valve is derived from that.

Figure 6-8
Principle of the IMS-control

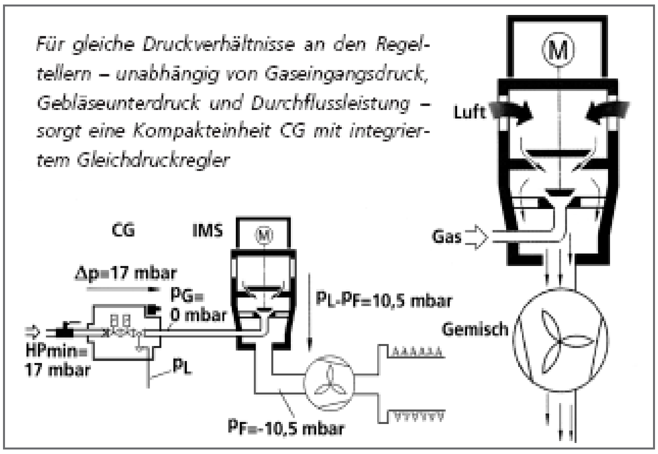
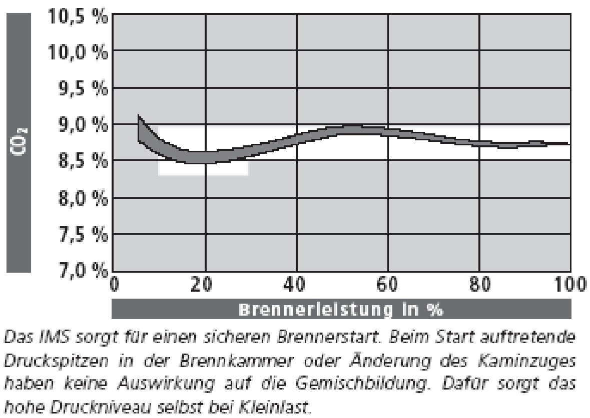


Figure 6-9
CO2-percentages at varying power inputs



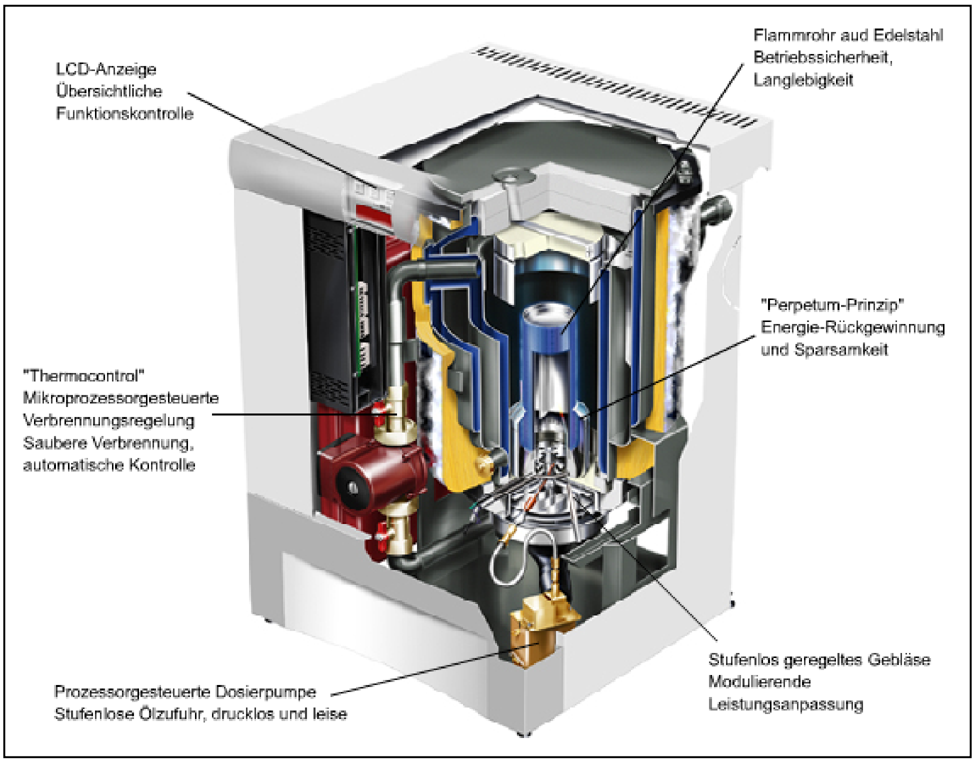
A relatively high flow speed of the combustion air insures good system stability also in the lower regions of modulation. The fan is positioned between the IMS and the burner and if the burner is switched off, the IMS closes, which decreases convection losses over the boiler in the off-mode.

6.2.3 Turndown ratio oil boilers

For oil boilers it is technically more difficult to vary the power input, simply because the mixing of oil (druplets) with air is more complex. Being already a standard technical product feature for gas boilers, modulating the power input is certainly not a standard feature for oil boilers. Common products today are oil boilers with a single or two stage burner. Controlling feed temperatures can only be done by switching the burner on and off between the given control-band (hysteresis) of the boiler control system. As a result the cycling behaviour is amplified and as a result, the emissions increase and generator efficiency decreases.

The best product now on the market is probably the modulating oil boiler "Mira" from the Austrian company Windhager Zentrall Heizung with a turn down ratio of 1 : 2,4.

Figure 6-10
Mira, modulating oil boiler from Windhager.
Modulation range
Mira 142: 14 to 5,7 kW
Mira 112: 10,9 to 4,7 kW
Water content: 12 l.
Netto weight: 119 kg.



Within the EU, there is a Project called BIOFLAM, which has been set up to investigate new combustion technology for liquid fuels. The main aim of the Project was the development of modulating oil burners for outputs between 2 and 20 kW, where primarily liquid biofuels can be used. In addition, the reduction of pollutant emission, such as NOX and CO (up to 50 %) should be achieved.

Conventional liquid-fuel boilers atomise the fuel and burn it in diffusion flames. The Bioflam project has developed a new ceramic burner for premixed liquid fuel, which makes innovative use of a process known as 'cool flame vaporisation', and considerably improves combustion. The vaporiser component, which converts the liquid fuel into a premixed mixture in the gas phase, was manufactured by EST-Aachen. LSTM-Erlangen, meanwhile, optimised the design and operation of a new porous-medium burner using high-temperature ceramics developed by PTC-Novazzano, which allows a much more controlled combustion environment. The designs of the vaporiser and porous burner were supported through computational simulations performed by NTUA-Athens and IST-Lisbon, respectively. The new combustion system will also have an impact in other areas of energy research. By linking with other projects in the Sixth Framework Programme, for instance, Bioflam will seek to adapt the cool-flame and porous-burner technology for fuel cells.

With a boiler specially designed and made to suit the BIOFLAM Burner, Hoval have been able to maximise the energy efficiency of the Boiler/Burner Unit.

At the CER 2005 in Brussels, Hoval exhibited their BIOFLAM Unit, consisting of a Boiler and BIOFLAM Burner, which reached the target set at the start of the project. The innovative combustion process takes place in 2 stages. Initially, fuel is vaporized at a temperature between 400° C and 500° C. In this „cool flame vaporization“ phase, fuel is vaporized completely without any droplets. This is a key requirement for combustion with optimum efficiency and minimum emission. The vaporized fuel is then burned in a temperature regulated porous burner at 1200 - 1400° C. The combustion gases are cooled down to 30° C through a condensing section. On extreme cooling of the combustion gases, the superheated steam condenses. And so all of the energy within the fuel can be used.

The Project BIOFLAM has been part financed by the EU. The Project-consortium has been coordinated by OMV AG (Vienna). Also in the consortium are RWTH University Aachen, University of Erlangen-Nürnberg, Hovalwerk AG (Vaduz), Instituto Superior Tecnico Lisbon, PTC SA (Novazzano CH), CSEM SA (Neuchâtel CH) and the European Boiler and Burner Institute (EHI).

The first field test facilities started operation in winter 2004/05 in Vorarlberg and Vienna in Austria.

6.2.4 Multi-stage boiler installations (cascade)

For larger buildings and multi family homes the similar problem exists with controlling the power input. In these applications the difference between minimal, average and maximal heat load will even be higher than for single famlily homes. An insufficient modulation range of a single high capacity boiler would have an even worse effect on emissions and efficiency due to cycling; let alone a single or two stage high capcity oil boilers.

An installation with several smaller modulating boilers (cascade installations) could help overcome this problem. As the heat load of the building increasing or decreases or more more of the individual smaller boilers can be switched on or off. Such an installation can severally reduce cycling and standing losses.

Although from an energy perspecitive this approach is sensible, the economic aspects of such cascade installations (higher installation costs because of more complex hydraulic and flue gas systems) are often considered more important.

More technologies are being developed for controlling the fuel/air-ratio. Main driver however is not the burner modulation, but the changing enthalpy of the fuels, due to the use of different gas-qualities.

Related techniques are discussed in the next paragraph.

6.3 Fuel/air ratio control

The liberalization of the EU-gas markets forces distribution companies to allow gases from different suppliers into their networks. Already today gasses of different suppliers and qualities (including tests with bio-gas and hydrogen) are mixed and supplied into the network. As a result the enthalpy or Wobbe index of the supplied gas may change, causing a shift in the air factor λ (fuel/air-ratio). Varying gas qualities will cause *higher emissions* and a *lower boiler efficiency*, unless control systems are used that measure either the quality of the fuel or the quality of the combustion and adjust fuel/air-ratio likewise.

Figure 6.10 gives an impression of the magnitude of the air ratio shift due to gas quality variations.

A design point of air ratio $\lambda = 1{,}3$ for methane was taken as a reference and the resulting air ratios of some other gases used in Germany are compared. The figure shows air ratio shifts of 1,2 to 1,61 (adjusted for different gas densities).

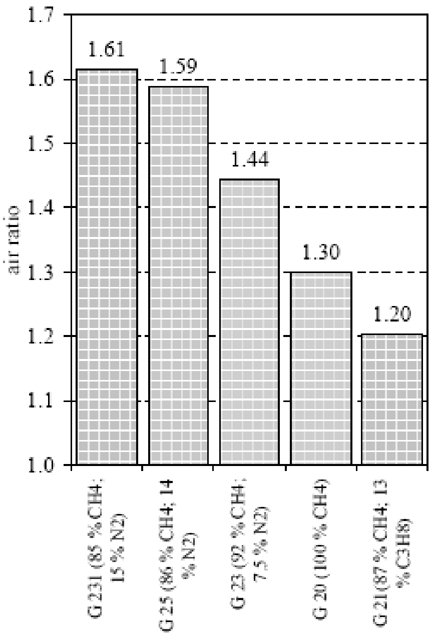


Figure 6-11.
Air ratio shift duet to gas quality variations, adjusted for the also varying density of the supplied gas.

As a result of these shifts in air factor, emissions will show large variations, flames might blow off, thermo-acoustic resonance could occur and efficiencies may drop considerately. Especially for condensing boilers, the efficiency drop is important because not only the flue gas losses increase (higher exhaust flow volumes) but also the dew-point is lowered due to a shift of the partial pressure of the water vapour.

Figure 6-12. Principles and parameters for fuel/air-ratio control

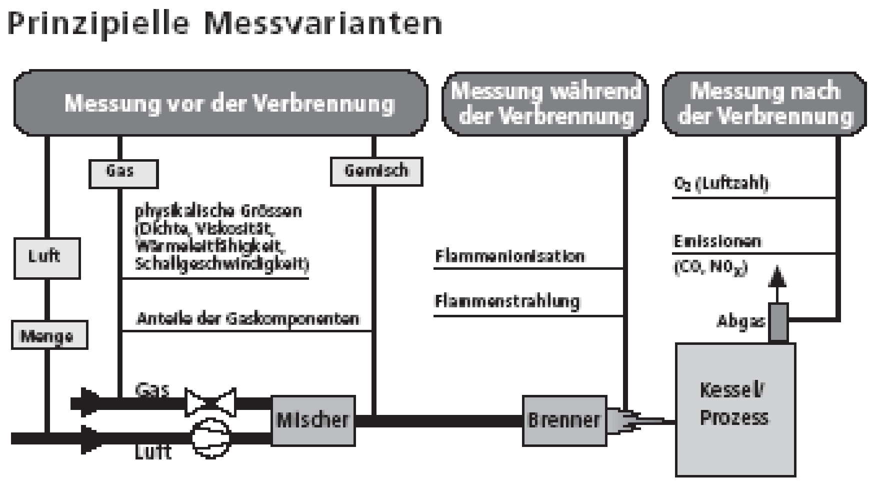


Figure 6.11 summarizes the parameters that can be used to measure and control the fuel/air-ratio.

Parameters that can be measured *before* combustion:

- specific mass
- viscosity
- thermal conductivity
- sonic speed
- substance

Parameters that can be measured *during* combustion:

- flame ionization
- flame radiation
- temperature

Parameters than can be measured *after* combustion:

- oxygen
- CO
- NOx

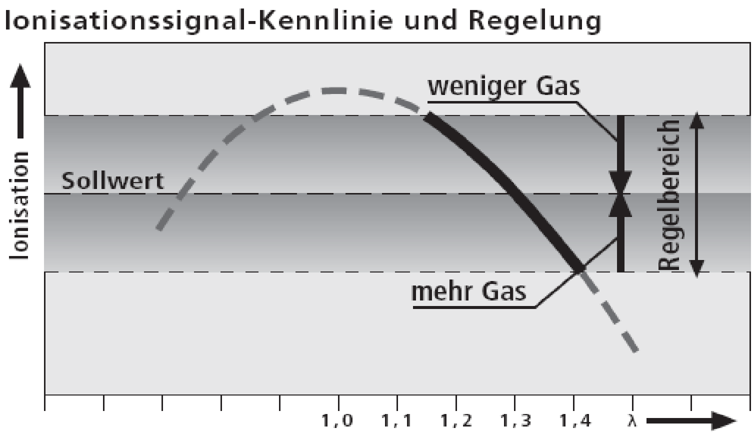
A lot of research has been done over the years to design, built and test the various options. Some of these R&D activities have actually evolved in solutions that are applied today in state of the art boilers.

6.3.1 *Measurement of flame ionization*

This technology is based on the measurement of the ionization voltage over flame and gas mixture. This ionization is already used for flame-control reasons (in case no ionization signal is measured, there is no flame and the gas valve is closed). With additional electronic circuitry the intensity of the ionization signal can be measured. And because the flame temperature (ionization voltage) is directly related to the air factor, the ionization signal is a indication for the quality of combustion.

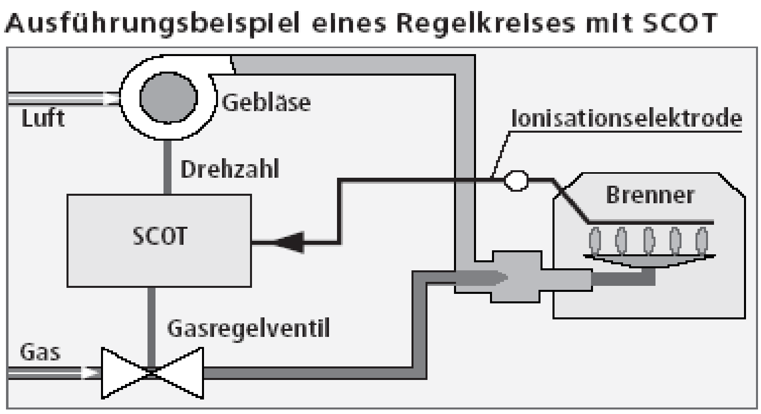
For surface burners with laminar flames the relation between ionization signal and air factor is unambiguous and similar to a parabolic curve (see Figure 6.11). The maximum ionization signal is always measured at air-factor λ= 1. This point is used for the automatic calibration of the combustion control system.

Figure 6-13.
Typical curve for the ionization signal. Source: VSG



Next step for a fully functional combustion control system is to use this ionization signal for an active control of the gas valve and the fan. Weishaupt uses this type of active combustion control in their wall hung gas condensing boilers called Weishaupt – Thermocondens.

Figure 6-14.
Schematic representation of combustion control system using the ionization signal.(In Germany this technology is called SCOT, meaning System Control Technology).



Viessmann uses this technology in the VITODENS boilers, and they gave it the name "Lambda Pro Control".

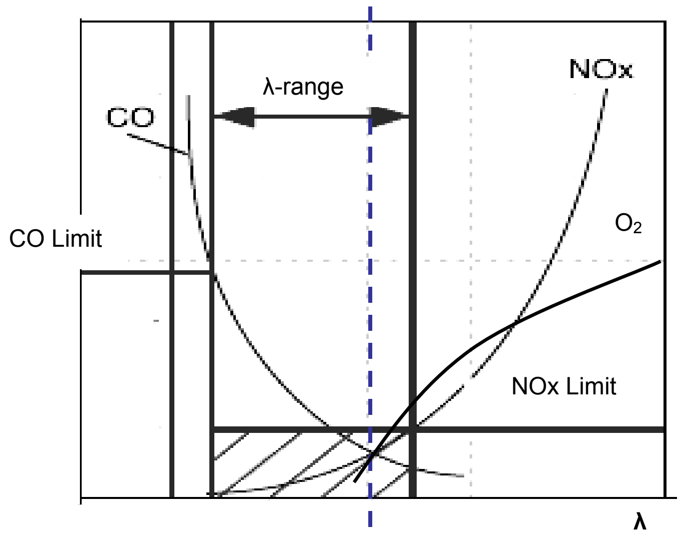
Buderus uses the ionization signal for controlling the gas supply in their atmospheric gas fired LT-boilers.

6.3.2 Measurement of O2

Oxygen sensors are already in use for some time now in cars and gas motors. They control the air- factor within the limits set by the catalytic reformer. The amount of oxygen in flue gasses can directly be related to the combustion quality and the air factor and O2 analysis therefore could offer proper feed back related to combustion control. However there are certain drawbacks. Boilers are usually operated at slightly negative pressure. Any leaks cause air to be drawn in and as a result the O2 readings in the stack will be higher than those actually found in combustion zone. Also, stratification of stack gases can make O2 sampling at a single point inaccurate.

Several boiler companies have tried and are trying to apply these sensors for combustion control in residential boilers as well, but so far didn't succeed in getting the technology beyond prototype stage. Sensor stability and price remain as the prohibitive thresholds.

Figure 6-14.
Relation between air-factor λ and CO, excess O2 and NOx



For a car the sensor would need an operational lifetime expectancy of approximately 4000 hours. For a boiler one would need 30 to 40.000 hours.

6.3.3 Measurement of CO

The other flue gas component than can be measured for combustion control purposes is CO.

CO is a product of incomplete combustion which will combine with oxygen to form CO2 if sufficient O2 is available. Ideally, if combustion is complete, the level off CO will drop to zero. Since complete air/fuel mixing is not possible, the practical level of CO for control purposes is usually < 160 ppm.

Using CO to trim combustion control systems offers an advantages over O2/trim: the CO control point remains constant for all types of fuels.

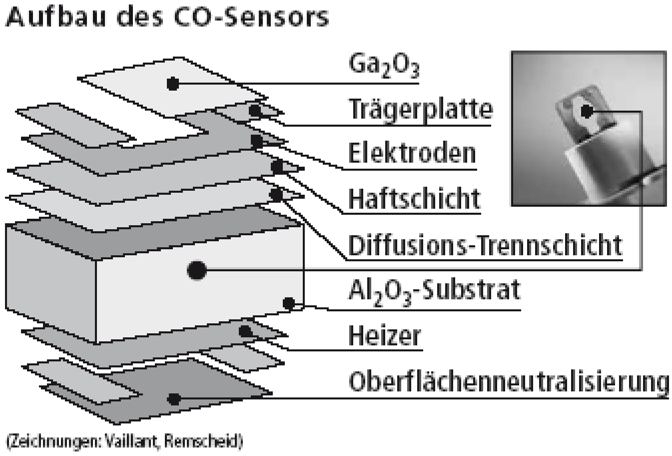


Figure 6-16
Construction of CO-sensor used by Vaillant.

Vaillant GmbH developed together with Steinel Solutions AG a CO-sensor based on a Ga_2O_3 sensor platform.

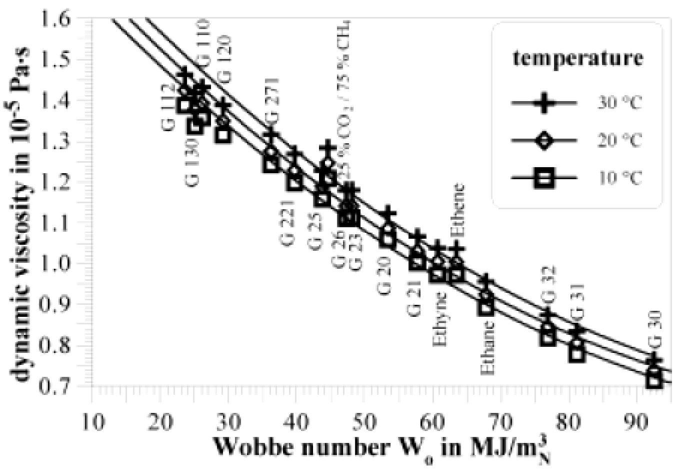
Based on the information from the CO-sensor the also new gas valve / safety valve assembly is operated with a step controller, which again influences the fan rotation speed.

The CO-sensor can also be used to detect wear to components like the fan or pollution of the burner, resulting in more efficient maintenance schemes.

6.3.4 Measurement of viscosity

Gas quality can be expressed by the Wobbe number. The Wobbe number correlates with the dynamic gas viscosity, according to the figures given in Figure 6.15.

Figure 6-17.
Relation between dynamic viscosity and Wobbe number.



The correlation between the Wobbe number and the viscosity is well known since a long period of time but so far not used for combustion control purposes. Only due to developments in micro technology a compact sensor design became possible. The Institute of Fluid Mechanics of the University of Erlangen – Nuremberg, Germany, developed a prototype of a viscosity sensor a tested the principle on a test rig.

The principle could work but additional development on the sensor part (based on capillary viscosimetry) is needed.

6.4 Pump- and feed temperature control

For comfort- and efficiency reasons, it is sensible to optimize the supply of thermal energy to the heat emitting system. The amount of thermal energy that is supplied by the boiler is determined by the combination of the *feed temperature* [°C] and the flow [l/s]. Optimizing here means several things.

Optimizing for comfort:

- Minimize temperature variations related to room temperature set point.
- Re-heat a habitable room quickly after increasing temperature setpoint.
- Minimize noise related to boiler water flow

Optimizing for efficiency and fuel costs:

- Minimize feed temperatures (improves boiler efficiency and minimizes transmission, conduction and ventilation losses of heat emitters and pipe work)
- Only heat rooms when people are present
- Minimize temperature variations related to room temperature set point (the bigger the variations, the higher the *average* room temperature must be to acquire a similar comfort level)
- Only use pump when thermal energy is requested

Ideally the transmission and ventilation losses of the house are continuously compensated with the equivalent supply of thermal energy from the heating system. This should preferably be achieved with minimal feed and return temperatures, because in that way radiation and convection losses of all heating system components can be minimised, start/stop-losses of the generator can be minimised and steady-state generator efficiency can be optimised. To give a rule of thumb for the energy losses that are caused by higher return temperatures than strictly necessary one could say that for condensing boilers, every 10°C increase causes an additional loss of 2 to 2,5% on the annual efficiency.

General goal for thermal energy supply therefore should be: minimise feed temperatures and optimise temperature difference between supply and return.

6.4.1 Flow control

In general pumps are comprised within the boiler casing and considered as an OEM component, that is simply switched on and off by the boiler control system. This is the case with practically all wall hung boilers (appr. 80% of yearly boiler sales). The energy consumption for circulators used in this context, varies between 25 and 90 W, depending on the type of pump, the resistance in the heating system and the control options of the pump.

The save project “Promotion of Energy Efficient Circulator Pumps” already indicated that the use of variable speed pumps can lead to substantial saving on the electricity consumption of circulator pumps in heating systems. The use of variable speed pumps ensures that, when the resistance in the heating system increases because radiator valves are closed, the rotation speed is reduced to keep the pressure difference in the system more or less at a constant value. As a result the power consumption is reduced and the noise-levels are lowered. The questions that remain are: will a pressure controlled variable speed pump also influence the steady state efficiency of the boiler and can a pump that is fully controlled by the boiler control unit (BCU) achieve additional savings?

The parameters that determine the feed and return temperatures of a heating system are:

- boiler power input [kW]
- flow [m³/h]
- heat output through emitters [kW]

The heat output of the emitters itself depends again on the feed temperature and the flow speed through the emitter. Together the feed temperature and flow speed determine the average emitter temperature, which can directly related to the output of the emitter. The average radiator temperature influences the convective and radiative heat performance of the emitter (the higher the feed temperature the better its performance); this relation is non lineair.

To give an example: A standard panel radiator radiator (n = 1,3) that emits 1000 watts at a temperature regime of 75/60°C and 20°C ambient temperature, only emits 150 watts at a temperature regime of 40/25°C. The relation between heat output and temperature regime will vary for different type of emitters (depending on radiation share in total heat-output, also referred as exponent *n* (see also NEN-EN 442)).

The flow itself directly corresponds to the temperature difference between feed and return temperature of the emitter (the higher the flow, the lower the ΔT).

Consequences of un uncontrolled pump on return temperatures

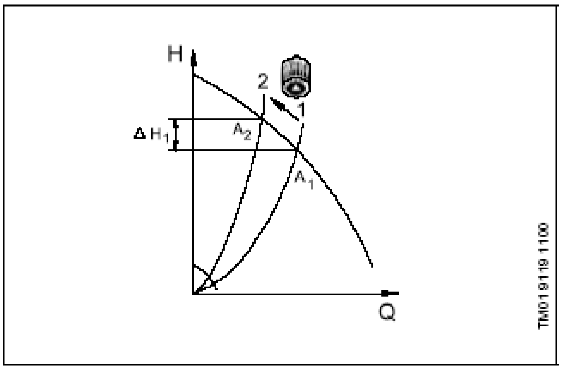


Figure 6-18
Uncontrolled single speed pump. Illustration of the changing head and flow when radiator valves are closed.

When single speed pumps are used (as is the case in practically all boilers that are already installed, but also in most of the new sales)), the rotation speed of the impeller is kept at a more or less constant level. When some of the radiator valves in the heating systems are closed, the flow resistance in the overall system increases which results in a reduced flow and an increased pressure difference over the boiler. The increased pressure difference however cause an increased flow over the fewer radiators that remain open. This could lead to noise in the system, but it definitely will lead to increased return temperatures and therefore lower generator steady-state efficiencies (compared to test conditions).

Consequences of pressure controlled pump on return temperature

When proportional pressure controlled pumps are used (variable speed pumps with pressure control) in installations in which some of the radiator valves are closed (pressure difference in system increases), the rotation speed of the pump impeller will automatically decrease, until approximately the same level of differential pressure over the heating system is achieved. As a result the total flow over the boiler is reduced, but the flow over the few radiators with the valve still fully open, remains approximately the same. The return temperatures will not be influenced too much and will remain approximately the same in comparison to a situation were all the radiator valves are fully open. Provided the overall system is correctly designed and the working field of the pump covers the systems need, the steady state efficiencies of the generator in such a system should be inline with test figures.

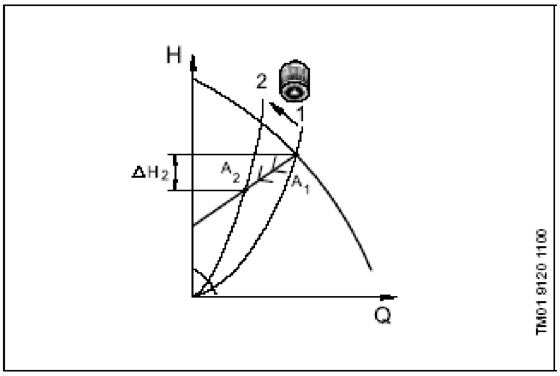


Figure 6-19.
Proportional pressure controlled pump. Illustration of the changing head and flow when radiator valves are closed.

This means that – in comparison to boilers with an uncontrolled pump - a boiler with a pressure controlled pump not only saves energy because of reduced electricity consumption of the pump, but also because of improved generator efficiency.

Additional saving options of a fully controlled pump

Following naturally from the preceding line of argument, a fully controlled pump can continuously optimise the flow in a heating system, resulting in lower average return temperatures and therefore an increased steady state efficiency of the generator.

One could for instance apply control strategies that hold the ΔT at a constant level, while minimising the feed temperature. One could also experiment with an increased temperature difference. A temperature difference of 20°C with an average of 40°C (regime 50/30) will achieve higher efficiencies than a regime of 45/35. The average temperature remains the same here, meaning that the distribution losses will not change too much.

6.4.2 Feed temperature control

As explained in the preceding pages, additional energy can be saved by minimising feed temperatures. As long as comfort limits are not exceeded and complaints like "heating up takes too long" or "it remains too cold in the room" can be avoided, minimising feed temperatures will help improving the annual efficiency of the heating system.

Technically speaking this optimization is quite feasible. A boiler with a sufficient turn down ratio and adequate pump control is technically able to supply every thermal energy demand that is needed. No problem there. The problem however is twofold:

1. Every heated room may need a different "optimal" feed temperature. Generally the emitters/radiators are connected to a combined supply- and a combined return pipe. Optimization of feed (and return) temperatures can only be done as an average.
2. To determine what the average optimal supply actually is, information on the actual and the requested temperature of each heated room would be necessary, including information on the type of heat exchanger (heat output at different temerpature regimes). This information is unfortunately not available.

The only information that is available, is the supply temperature and often also the return temperature, which combined gives an *indication* of the actual heat demand in the whole building or house.

This can only be an indication, because data on the actual flow is missing, meaning that the precise amount of heat that is transferred can not be calculated. E.g. an uncontrolled pump could cause a rise in the return temperature, which could wrongfully be explained as a drop in energy demand.

But even if detailed information on the flow would be available, we would still be looking to the average heating demand of the house. One room might need higher feed temperatures (large difference between temperature set point and actual room temperature), while for other rooms a lower feed temperature would be sufficient. An average value (calculated on the basis of flow and return temperature) could lead to comfort problems. Additional data per room is need to optimise the system both form comfort and efficiency reasons.

Today's practice is, that feed temperatures are determined by the *room temperature control systems* that are used. These *room temperature control systems* do not supply sufficient information to optimise the feed temperature as an average. As a result, higher feed temperatures are supplied during a large part of the heating season, which leads to a higher energy consumption than strictly necessary.

The feed temperature control systems that are predominantly used, are either systems that determine boiler feed temperatures on the basis of secondary information (a heat curve that related outside temperature to a feed temperature, return temperature, temperature in a single reference room) or systems that only control the maximum feed temperature (for safety reasons). Flows (pumps) are not controlled, or at best they control themselves on the basis of the varying flow resistance in the system.

Developments in the field of *feed temperature control optimisation* mainly depend on the developments in *room temperature control systems*, because these systems provide the information on the basis of which the feed temperature can be optimised and controlled.

In chapter 7 these *room temperature control systems* are further discussed.

6.5 Summary

Boiler controls are essential in achieving low emissions and high efficiency levels.

Power input control (modulation) of the boiler to a large extend determines the on and off cycling of the boiler. If the power input is higher than the heat demand, the boiler switches on and off in its attempt to deliver the requested amount of heat or (depending on the room temperature control system) requested feed temperature. This results in higher emissions and lower generator efficiencies, but its also influences the lifetime expectancy of the product. In most cases the minimal power input of the boiler is considerably higher than the minimal and average heat demand of the dwelling. The improvement potential here is considerable.

Combined with pump control, power input control facilitates the full control of feed and return temperatures, which is directly related to the steady-state boiler efficiencies. Yet most of the boilers on the market today do not have these product features. As a result, installations with low temperature emitter systems do not fully profit from their energy saving potential.

Fuel/air-ratio control already is an important feature in achieving a viable power input control system. But with the liberalization of the EU-gas markets, the inserting of biogas, hydrogen and other gas qualities into the distribution system, this feature will become even more critical. Changes in gas qualities cause a shift in the air factor and if boilers can not anticipate these changes in Wobbe number, problems could occur like increased emissions, reduced efficiencies, thermo acoustic resonance and the blowing off of flames. Especially for condensing boilers a change in air factor is important because it also influences the dewpoint and with it, the generator efficiency

Optimizing feedtemperatures – or better formulated – minimizing feed temperatures, is an important tool in the further optimization of the generator efficiency. To achieve this, the boiler needs additional information from room temperature control systems.

7 ROOM TEMPERATURE CONTROL

7.1 Introduction

Room temperature control systems are defined here as the aggregate of sensors, actuators and software algorithms on the basis of which a certain room temperature is achieved. They control the on- and off switching of the boiler and often also the boiler feed temperature for the purpose of achieving a predefined temperature in a heated room.

Most commonly used today are the Room thermostat controlled system - a system that continuously controls the temperature in one reference room (e.g. living room)- and the Weather controlled systems, which are systems that control the boiler feed temperature on the basis of the outside temperature.

In both systems Thermostatic radiator valves (TRV) can be used to maximise the heat output of an emitter on the basis of the actual room temperature.

In the last couples of years these three control techniques are more or less blending together in the strive for the ultimate room temperature control system. The most recently introduced room temperature control systems are capable of determining the heat demand of each individual room with multiple room temperature sensors.

Table 7-1. Description of room temperature control systems that are predominantly used (1 & 2) and a recently introduced room temperature control system (3).

	1. Single Room Thermostat controlled systems	**2. Weather Controlled systems**	**3. Multiple Room Temperatures controlled system**
Control principle	• Room Thermostat is placed in reference room (e.g. living room) • Boiler and pump is switched on and off by the room thermostat, based on ΔT in the reference room. • Depending on type of room thermostat, boiler is either switched between 0 and max power or values in between (modulating thermostat)	• Outdoor sensor measures T-outside and relates value with a certain boiler feed temp. (heating curve) • Boiler is switched on and off by the feed thermostat, based on ΔT in feed temperature (hysteresis). Pump is continuously running. • Depending on type of weather controller, boiler is either switched between 0 and max power or values in between (modulating thermostat)	• T-sensors in each room measure ΔT between set point and actual value. • A central control unit acquires the ΔTs of each individual room and calculates the optimal feed temperature. Boiler is switched on when feed temp. is lower than required. Pump is switched on when there is a heat demand. • Depending on type of central controller, boiler is either switched between 0 and max power or values in between (modulating thermostat).
Operation	Select temperature set point in reference room, through operating interface on room thermostat	Select heat curve (through operating interface of controller) and operate TRVs for individual room temperature adjustment.	Select room temperature set point on each individual room temperature controller (or through operating interface of central controller (if possible))
Temperature control in each individual room	• Good T-control in reference room • Inferior T control in other rooms (TRVs can maximize room T)	Relatively good T-control in all rooms (depending on size radiator and hydraulic balancing of the overall system)	Optimal T-control in all rooms.
Options for clock- controlled room temperature	Clock program controlled temperature in reference room	• Clock-program controlled boiler water temperature • Additional clock program controlled TRVs with room T-setpoints	Clock program for each individual room temperature controller

In the following paragraphs the principles, developments and pros and cons of these systems and their influence on the energy consumption, are further explained.

7.2 Single Room Thermostat controlled systems

7.2.1 *Principle*

Room thermostat controlled systems generally are systems that actively control the room temperature in one specific room, preferable the living room. The room thermostat determines when the boiler is switched on and when the boiler is switched off again. Normally the pump is switched off 6 to 10 minutes after the burner is switched off, in order to facilitate the transport of the heat that is accumulated in the boiler appliance. The temperatures in the other rooms are in this case a derivative, meaning that only when there is a heat demand in the reference room, thermal energy is also supplied to all the other rooms as well. The temperatures that will be achieved in those other rooms depend on:

- nominal power of emitters in reference room related to its heat load (influences cycling behaviour heat generator);
- external heat supplied to reference room (solar radiation, open fire, appliances, people);
- nominal power of emitters in "the other" rooms;
- and the position of the radiator or thermostatic valves of the radiators in "the other" rooms.

In other words, the temperature in the "other rooms" is not controlled and can at best only be maximised with thermostatic radiator valves.

Room thermostat controlled systems with the traditional on/off control, generally do not control the boiler feed temperature in a sophisticated way. If the room thermostat generates a heat demand, the boiler is switched on and will – in some cases after a slow start – operate at full power. The burner is switched off again either by the room thermostat (room temperature set point is reached) or by the safety thermostat of the boiler (e.g. 90°C feed temperature).

Example

With an input of 24 kW and a water flow of 1000 l/h the boiler increases the feed temperature with approximately 20°C. If only the two radiators in the living room are supplied (thermostatic valves in other rooms are closed) approximately 100 to 150 ltr of water is recycled and the cycle will take around 6 minutes. Because only 1 to 5 kW of heat can be transmitted into the living room (depending on feed temperature), the return temperature will only be 2 to 5°C lower than the feed water temperature, making the feed water temperature rise with 15 – 18 degrees with each 6 minute cycle.

When starting at 20°C, the feed water temperature will have risen to around 80°C after 15 – 20 minutes. If the room thermostat doesn't switch of the burner, the safety thermostat in the boiler will do so during the next 6 minute cycle. The pump will be on for another 6 – 10 minutes. If the room thermostat is still asking for heat, the burner will start again when the feed temperature has cooled down to a certain value (depends on hysteresis). Because boiler water temperature is already quite high, the burner switches off again within the first consecutive 6 minute cycle.

Obviously this operating mode is not optimal from efficiency perspective. Feed and return temperatures are too high, resulting in lower generator efficiencies. Assuming that the average return temperature in the example above is approximately 50°C, the additional generator losses due to the type room temperature control (compared to a situation with an average feed temperature of 30°C) is around 5%. In addition, variations and drifts in room temperatures will cause heat losses of around 7%.

7.2.2 *Developments*

Coming from the mechanical on/off room thermostat, these products have improved considerately over that last 10 years. Most of the these products now have a clock controlled options for a night setback. A room thermostat nowadays can have the following technical features:

- mechanical/electronic
- powered by batteries of through wiring with BCU
- with or without option night setback
- with or without clock program
- with or without optimiser
- on/off or modulating

Modulating room thermostats are gradually becoming the standard product in some countries, resulting in more intelligent control of the boiler power input, based on the difference between actual temperature and set point temperature. This results in lower average feed- and return temperatures - which increases generator efficiency - and in lower temperature variations which increases the control efficiency.

Danfoss is performing system tests with modulating condensing boilers and three different types of room thermostats: mechanical on/off, electronic on/off and modulating. The tests are performed in a controlled laboratory test house. This is a first real life test that generates figures indicating tha actual savings of modulating thermostats. Test results will be inserted in later drafts of this Task 4 report, provided approval is acquired from Danfoss[34]

Optimiser functions are introduced, that prolong the re-heating period after the night setback, thus facilitating a lower feed- and return temperature over a longer period. To a certain point the increased generator efficiency that is achieved in this way, outweighs the additional transmission and ventilation losses that are caused by prolonging the heating period. And finally sophisticated clock-programmes offer an option for programming more heating- and setback periods per day.

7.2.3 Pros and Cons

Looking at the technical features of current state-of-the-art room thermostats (electronic, modulating, clock program and optimiser) this type of control system is fairly optimised for comfort- and efficiency purposes. Main draw back of this room temperature controlled system is the fact that only the temperature in one specific room is actively controlled. The temperature in the other rooms is not controlled. This either leads to energy loss (rooms are unnecessary heated) or to comfort problems (rooms are too cold). Energy losses related to this can be minimised by:

- a strict discipline of the consumer, turning on and off the radiator valve when he enters or leaves the room;
- the use of thermostatic valves, maximising the room temperature to the selected value;
- the use of thermostatic valves with clock, with which setback periods can be programmed.

The comfort problems (rooms are too cold) can not be solved.

Advantage of this system is the fact that the pump is not continuously running, resulting in lower standing losses of the boiler, lower distribution losses and lower electricity consumption.

Room thermostat controlled systems are predominantly used in countries like the UK, Ireland, Netherlands, Belgium, France, Italy, Portugal, Spain and Greece.

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[34] Source: Danfoss Randall Limited, Mr. Martin O'Hara and Mr. Philip Smith.

7.2.4 Product examples

Apart from the boiler manufacturers who develop and sell their own room thermostat systems, there are several other companies that develop, produce and supply these products, amongst which: Honeywell, Danfoss, Stuhl, Theben, Seitron, Oventrop, watts, Heimeier, Resol, Techem, etc.

Figure 7-1. Honeywell Round T87F, with anticipation unit based on mercury switch (now prohibited).

Traditional mechanical on/off room thermostat.
Honeywell Round T87F — Technical features:

- fully manual control (no automatic night setback or other clock program)
- no optimiser
- proportional switching band 1,5 – 2°C

A large percentage of dwellings with room thermostat controlled systems are still using this type of room thermostats.

Figure 7-2. Honeywell Round Modulation T87M, with "OpenTherm" boiler communication protocol.

Honeywell Round Modulation T87M Electronic modulating room thermostat — Technical features:

- fully manual control (no automatic night setback or other clock program)
- no optimiser
- feedback boiler status (burner on or off)
- resolution: 0,5°C
- powered through wiring with boiler

OpenTherm®
Communication System

Figure 7-3.
Danfoss TP7000

Electronic on/off room thermostat with programmable clock.
Danfoss TP7000 (with or without RF) — Technical features:

- Clock-controlled with options for manual override
- Max. of 7 different day-programs of 6 periods per day
- With optimiser function (duration of optimisation period can be selected)
- Powered through batteries or 230V. 50 hZ (cell battery)
- Measurement accuracy: ± 1%

Figure 7-4.
Nefit Moduline 300

Electronic modulating room thermostat with programmable clock. Technical features:

- Clock-controlled with options for manual override
- Total of 42 switching points over 7 days
- Info button for extra display information (e.g. graph of clock-program)
- With optimiser function(duration of optimisation period can be selected)
- Powered through wiring with boiler
- Resolution: 0,5°C
- Automatic switch from summer- to wintertime
- Energy consumption 0,3 watts

Figure 7-5.
Honeywell Chronotherm IV Modulation

Electronic modulating room thermostat with programmable clock.

Honeywell Chronotherm IV Modulation — Technical features:

- Clock controlled with options for manual override
- Maximum of 7 different day-programs of 6 periods per day
- With optimiser function (duration of optimisation period can be selected)
- Powered through batteries or 230V. 50 hZ (cell battery)
- Measurement accuracy: ± 1%
- Party- and holiday function (programmed setback periods)
- Semi automatic switch from summer- to winter time.
- Resolution: 0,5°C
- Memory chip remembers all settings after power interruption

OpenTherm®
Communication System

7.3 Weather controlled system

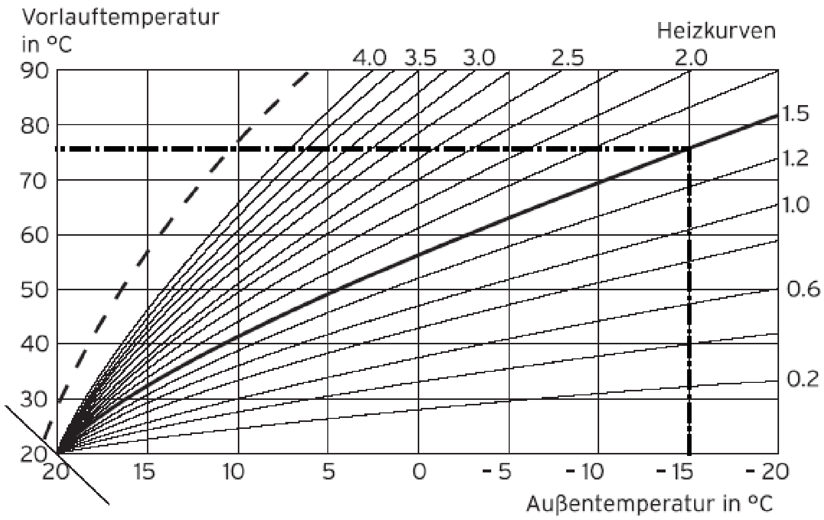
7.3.1 Principle

Weather controlled systems are systems that control the boiler feed temperature on the basis of the outside temperature. The feed water temperature is continuously kept at a level determined by the heating curve that is selected. This heating curve relates the outside temperature to a specific feed temperature.

Pumps are continuously running in weather controlled systems.

The basic idea behind this type of control systems is that with decreasing outside temperatures the transmission and ventilation losses of a house will increase and more thermal energy is needed to compensate for these losses. A feed temperature that increases with dropping outside temperatures is an elegant way to compensate for these losses.

Figure 7-6
Heating curves for weather controlled systems.



Weather controlled systems are predominantly used in Austria, Denmark, Germany, Finland, Sweden and Luxemburg.

The components that are needed to facilitate this type of room temperature control system are:

1. Outside temperature sensor
2. Thermostatic valves on emitters in each room
3. A controller that calculates the preferred feed temperature and interacts with the boiler to achieve this feed temperature.

The controller with its human interface can either be a separate device, or it is integrated in the boiler and the BCU (boiler control unit).

With this pre-controlled feed temperature the emitter will be able to roughly heat the room to the requested temperature levels. The thermostatic radiator valve (TRV) will fine tune the heat output of the emitter to the requested room temperature values and avoid large room temperature overshoot.

The requested varying feed temperatures can be realised in several ways:

- with a mixing valve
- with a proportionally on/off controlled burner
- with a modulating burner

Obviously the best way to fully benefit from the potential energy savings these feed temperature controlled systems can achieve, are fully modulating condensing boilers.

7.3.2 *Developments*

Developments in the field of weather controlled systems focus on:

- User friendly operation
- Feed temperatures based on a combination of both outside temperature and reference room temperature
- Clock-controlled thermostatic valves
- Centralised operation and control of individual rooms (through wireless communication or EIB[35]-communication between a centralised control& operation-unit and T-sensors emitter-actuators)

7.3.3 *Pros and Cons*

Advantages of weather controlled systems are:

- The temperature in each individual room can be controlled separately
- Feed temperatures are lowered, with rising outdoor temperatures which will result in higher system efficiencies (savings are optimised when fully modulating condensing boiler are used)
- Temperature variations in individual rooms can be reduced, which results in higher control efficiencies.

Draw-backs are:

- Pumps are continuously running and boiler temperature continuously heated to the requested levels, also when there is no heat demand.
- To prevent comfort problems (heating up the room takes too long, room temperature remains too low) heat curves are generally set too high, which reduces the potential savings
- When thermostatic valves are not strictly operated (opened when entering the room and when leaving) the rooms will unnecessary be heated up.
- More sensitive to variations in room temperature (compared to room thermostat controlled systems)

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[35] European Installation Bus

7.3.4 *Product examples*

Figure 7-7.
Vaillant calorMATIC 630

System with weather dependent modulating feed temperature control and clock program for setback temperatures.
Vaillant calorMATIC 630 — Features:

- additional control options for hot water storage tank, 2nd. modulating boiler and two heating circuits.
- Party- and holiday function (programmed setback periods)
- Adjustable settings for many parameters (duration for optimiser function, sensor corrections, language, heating curve, maximum feed temperatures, etc.)
- Power consumption: 4 watts

Figure 7-8
Danfoss ECL Comfort 300

System with weather dependent on-off feed temperature control and clock program for setback temperatures.
Danfoss ECL Comfort 300 — Features:

- additional control options for 3 additional components (extra boilers, pumps or valves)
- 6 ports for T-sensors
- With additional PCBs functions can be determined
- Clock program for 2 temperature levels (comfort and setback)
- Switching times can be optimized on the basis of existing indoor and outdoor temperatures.
- Controller can communicate with other ECL 300 controllers (two wires)

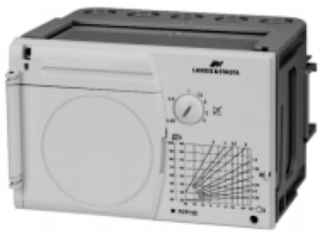


Figure 7-9.
RVP102 controller of Siemens system VILLAGYR

System with weather dependent on-off feed temperature control and clock program for setback temperatures.
Complete VILLAGYR set consists of:

- Controller RVP102 (see picture left)
- Operation interface QAW70 (see picture below)
- Outside temperature sensor QAC32
- Feed water thermostat QAD22

Heating curve is selected on the RVP102 – unit.

Figure 7-10.
Human Interface QAW70 of Siemens system VILLAGYR

The operation of the VILLAGYR control system is conducted through the Human Interface QAW70. Main function is controlling the feed water temperatures by switching the boiler on and off.
Technical features:

- Option for a) feed temperature is outside temperature controlled or b) feed temperature is based on combination of outside- and room temperature
- In the combined operating mode b) the effect of room temperature on feed temperature can be adjusted/selected (default 20%)
- Optimiser function
- Options for manual override of clock program
- Power consumption: 7 watts
- Feed temperature switching hysteresis adjustable from 1 – 20 K with a default of 6 K
- Switching intervals: minimum of 2 minutes

Figure 7-11.
Buderus Logamatic ERC

Electronic modulating room thermostat with clock program and optional module for weather dependent modulating feed temperature control.
Buderus Logamatic ERC — Technical features:

- Option for a) feed temperature is outside temperature controlled or b) feed temperature is based on combination of outside- and room temperature or c) only room temperature control (no feed temperature control)
- Clock-controlled with options for manual override
- Total of 42 switching points over 7 days
- With optimiser function (duration of optimisation period can be selected)
- Powered through wiring with boiler
- Resolution: 0,5°C
- Automatic switch from summer- to wintertime
- Clock program for sanitary hot water boiler

7.4 Thermostatic valves

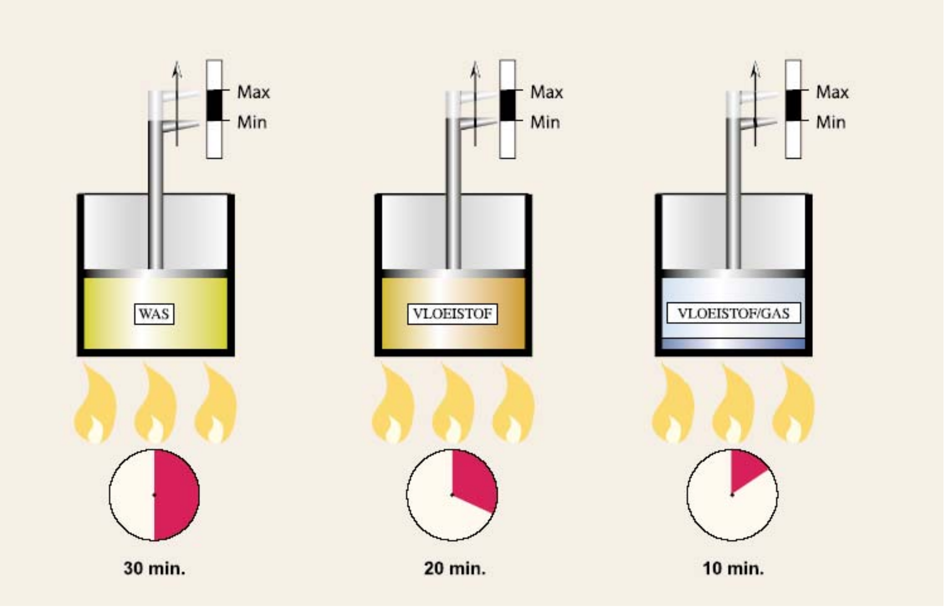
7.4.1 Principle

A **Thermostatic Radiator Valve**, (TRV), is a self regulating valve fitted to the radiator or emitter system. The TRV controls the temperature of a room by regulating the flow of hot water to the emitter.

TRVs are always used in combination with one of the three Room Temperature Control systems mentioned in paragraph 1.3.1. (It should be noted that TRVs should not be used in the same room with an electronic room thermostat. The operation of the two devices are not compatible and would interfere with each other so that effective temperature control would be lost).

TRVs consist of two parts, a valve that opens or closes to control the hot water flow, and an actuator that controls the opening of the valve. The actuator adjusts the valve opening based on the temperature in the room via a mechanical linkage or pin connected to the valve. The actuator is usually some kind of piston, containing wax, a fluid or a gas. The piston expands or contracts as the temperature of the room – and with it the temperature of the medium inside the piston - rises or falls. The piston is preset by a screw mechanism that positions the piston a set distance from the connecting pin.

Figure 7-12
Reaction speed of the TRV may vary and generally depends on the medium that is used in the actuator head. The reaction speed may vary from half an hour to approximately 10 minutes.



As the temperature of the room increases the piston expands, pushing the pin down and restricting the flow of hot water. As the room cools the plug contracts, the pin is let out a little and more heat is admitted into the room. This process, completely self contained and without complex electronic circuitry, keeps the room temperature at a desired constant level.

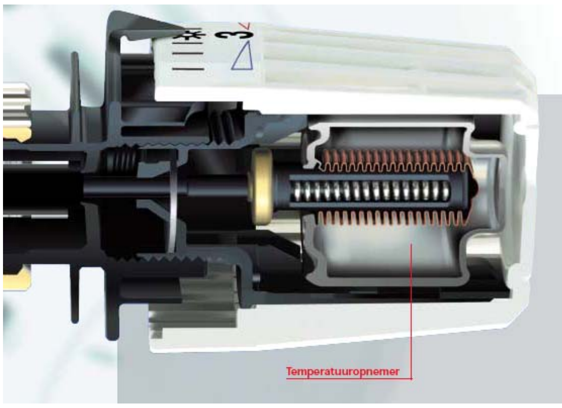


Figure 7-13
Section of Heimeier Thermostatic Valve

Depending on their reaction speed and p-band, TRV temperature control are subject to swings in temperature inside the heated space due to external conditions, e.g. ambient temperature and/or wind velocity and direction, and since their setting is not readily adjusted, once the initial setting is made it is rarely changed.

TRVs are selected on the basis of the maximal allowable flow through the radiator in question. Calculations on the requested maximal flow at certain pressure differences are necessary to determine the Kv-value of a TRV. The Kv-value gives the flow in m^3/h at a Δp of 1 bar. Flows at different Δps can be easily calculated with the formula $Kv = Q / \sqrt{\Delta p}$.

Generally TRVs can be pre-adjusted, meaning that the maximum opening of the valve can be pre-adjusted at the given stroke of the actuator part. In that case the Kv-value is presented in combination with the selected position of the pre-adjusted valve. The position of the pre-adjustable valve are often related to as positions N, 7, 6, 5 etc. to 1.

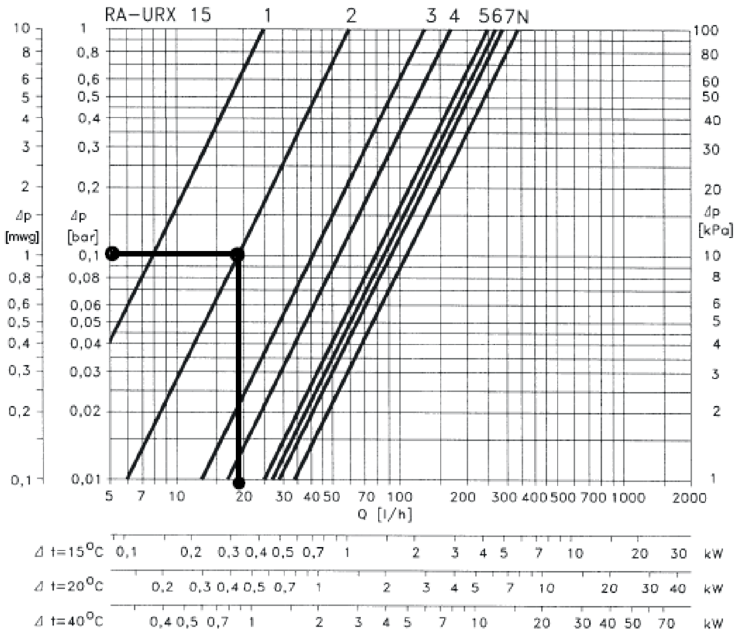
Figure 7-14
Pre-adjustable valve part of a RTV

Position N means (according to EN 215) that the valve is fully closed (or opened) when there is a temperature difference of 2 degrees between setpoint of TRV and the temperature measured by the T-sensor of the RTV. In other words, Position N corresponds to a proportional band (P-band) of 2 K. Position 1 corresponds to a P-band of 0,5 K.

Kv2 = flow in $[m^3/h]$ at a ΔP of 1 bar when the valve part is pre-adjusted at a P-band of 2 K (position N).

Kv 0,5 = flow in $[m^3/h]$ at a ΔP of 1 bar when the valve part is pre-adjusted at P-band of 0,5 K (position 1).

The Kvs-value gives this value at maximum opening of the valve-part of the TRV.



Capaciteit met RA-URX, RLV-X en RAX thermostatisch element bij een P-band tussen 0.5 K en 2 K.

Figure 7-15
Capacity/Δp graph of a RTV at P-bands between 0,5 K and 2 K (respectively position 1 to N).

The following values can be deducted from the graph:

- Kv2 = 340 l/h
- Kv0,5 = 25 l/h

The P-band of the TRV influences the temperature variations in a room. The bigger the P-band, the higher temperature variations may become. This may result in increased heat losses through the building envelope. Also the ability to utilise internal gains (from people, equipment, solar radiation) depends on the type of heat emission- and control system (p-band).

The heat loss due to this phenomena can be calculate in different ways. The draft version of prEN 15316-2-1[36] provides examples of values for these control losses in the informative annexes B.2 and C.

For dwellings with an average heat load of 40 kWh/m²/a and radiators with thermostatic valves as emitters, the losses may vary from 1 to 7,4 % of the heat theoretically required, depending on the type of room temperature control system that is used (see table below)

Table 7-2 from prEN 15316-2-1: Energy performance factors for heat emission control according to DIN 4701-10

System	Room Control	Energy performance factor $e_{c,em}$ Demand in [kWh/m²a]						Additional losses [kWh/m²a]
		40	50	60	70	80	90	
Radiators on outside wall	Thermostatic valve							
	- with p-band of 2 K	1,08	1,07	1,06	1,05	1,04	1,04	3,3
	- with p band of 1 K	1,03	1,02	1,02	1,02	1,01	1,01	1,1
	Electronic							
	- clock + PI-temp. control	1,02	1,01	1,01	1,01	1,01	1,01	0,7
	- clock + optimizer + PI-temp. control	1,01	1,01	1,01	1,01	1,0	1,0	0,4

36 Method for calculation of system energy requirements and system efficiencies – Part 2-1: Space heating emission systems (October 2005).

If TRVs are used for room temperature control, it is important that they are selected with an optimal P-band for the required application. This will not always be possible (product range) and will most probably not always be calculated and optimised.

7.4.2 Developments

Developments in the field of TRVs focus on:

- motorized valves with improved T control (electronic, PI)
- programmable TRVs
- RF-controlled RTVs
- Intelligent programming (self learning)

7.4.3 Pros and Cons

Advantages of TRVs are:

- they save energy compared to traditional radiator valves
- control room temperature without additional power consumption
- long life time

Draw-backs are:

- could lead to relative high temperature variations in the room when p-band is high.

7.4.4 Product examples

TRVs with fixed capacity

RA-FN recht RA-FN haaks RA-FN haaks (UK) RA-G recht RA-G haaks

Figure 7-16. TRVs with fixed capacity (Danfoss)

TRVs with adjustable capacity settings

RA-N, Straight Angle Left mounted angle UK-angle

Type	Code nr.	Design	Connections		Pre-setting								
			In-let	Out-let	k_v-max. 1) (m³/h at Δp = 1 bar)								k_{vs}
			R_p	R	1	2	3	4	5	6	7	N	N
RA-N 10	013G0011 013G0012 013G0151 013G0231 013G0232	Angle Straight UK Angle R Angle L	3/8	3/8	0.04	0.08	0.12	0.19	0.25	0.33	0.38	0.56	0.65
RA-N 15	013G0013 013G0014 013G0153 013G0233 013G0234	Angle Straight UK Angle R Angle L	1/2	1/2	0.04	0.08	0.12	0.20	0.30	0.40	0.51	0.73	0.90
RA-N 20	013G0015 013G0016	Angle Straight	3/4	3/4	0.10	0.15	0.17	0.26	0.35	0.46	0.73	1.04	1.40
	013G0155	UK	3/4	3/4	0,16	0,20	0,25	0,35	0,47	0,60	0,73	0,80	1,00
RA-N 25	013G0037 013G0038	Angle Straight	1	1	0.10	0.15	0.17	0.26	0.35	0.46	0.73	1.04	1.40

Figure 7-17. TRVs with adjustable capacity settings (Danfoss) including Kv-data at different pre-settings (table below)

TRVs with remote Operation

Heimeier Fluid filled TRV — Technical features:

- Frost protection
- Max. temperature T sensor: 50°C.
- Hysterese 0,4 K.
- Wassertemperatureinfluss 0,3 K
- Running time: 26 Min.

Figure 7-18.
TRVs with remote Operation

TRVs with remote sensor

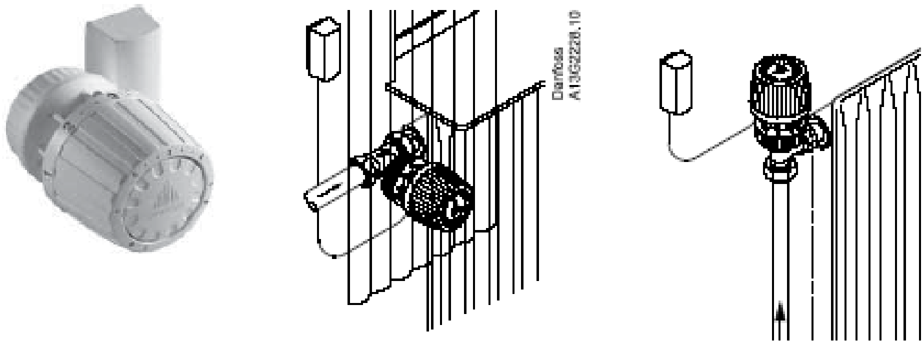


Figure 7-19.
TRVs with remote sensor
(manufacturer: Danfoss)

If a TRV with integrated T-sensor is placed in a position where it is not possible to measure an adequate reference temperature (e.g. behind curtains), a TRV can be used with a remote sensor.

TRVs with motorized actuator

Honeywell motorized valve M5410C — Technical features:

- Suitable for *ON/OFF* control without feedback
- Short runtime (push/pull) ca. 1.8 mm/s and 0.4 mm/s
- Electronic switch-off in the end position
- Input voltage 24 Vac, +20...-20%; 50 Hz
- Power cons.<8 W during operation, <0.5 W in end position
- Input signal <10 mA
- Stroke 2.5 mm and 6.5 mm
- Stem force 100 N (minimum)
- Protection standard IP 54

Figure 7-20.
Motorized actuators 1:
Honeywell motorized valve
M5410C

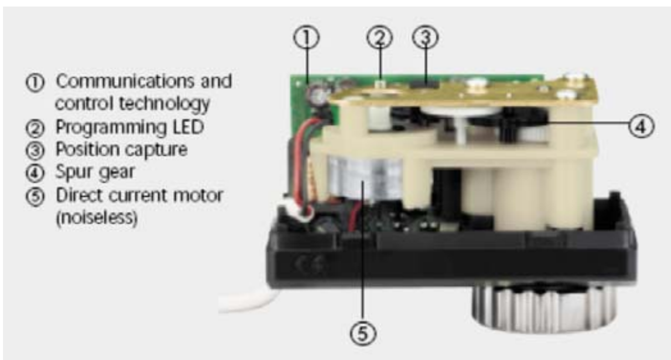


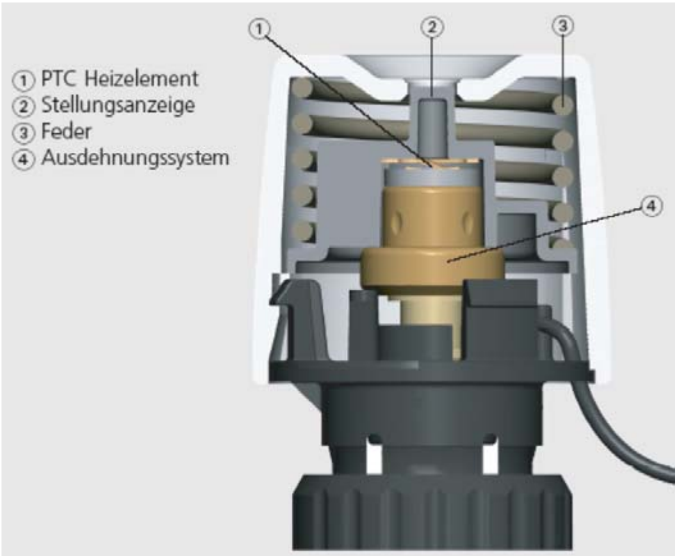
Figure 7-21.
Heimeier motorized actuator 2 for connection to EIB (European Installation Bus)

The EMO EIB *proportional* actuator is designed for connection to the European Installation Bus (EIB). The connection is made directly, and a separate bus coupling is not necessary. Programming the phsysical address without direct contact is made possible with the aid of the programming magnet. A red programming LED acts as a status indicator.
Technical features:

- Voltage supply: from EIB Bus (SELV according to IEC 364-4-41)
- System voltage: 24 V DC (+6V / –4V)
- Power consumption: typ. 10 mA (= 240 mW; ¡ approx. 2 BA modules)
- Participants per EIB-Linie: max. 64 (depending on the nature of the voltage supply and participants)
- Communications: 3 x 8 bit format for set value, actual value and status 1 x 1 bit format for compulsory position (open window recognition)
- Valve stroke: min. 1.0 mm; max. 4.0 mm
- Running time: 25 s/mm

TRVs with thermal actuator

Figure 7-22.
Thermal actuators: Heimeier EmoTec



Thermal actuators use a small heating element to heat the medium (wax, fluid of fluid/gas) in the actuator head; in this manner the TRV can externally be controlled. These thermal actuators are deliverable in a 24V or 230V version.
Product: Heimeier EmoTec — Technical features:

- Powerless in closed position
- Protection standard IP 43
- Power consumption: 3W (continuous), 9W (during operation) for the 24V version; for 230 V version resp.: 3W and 90W
- Frequency: 0 - 60 Hz
- Running time: approximately 3 min.

MT010 small linear actuators are used with Honeywell room-temperature controllers for time-controlled modulating regulation of heating and cooling systems. A microprocessor based positioner guarantees accurate control. The MT010 is designed for applications where space is limited. Suitable valves are the 2-way V5822 and V5832 series of small linear valves with 2.5 mm stroke as well as the Honeywell TRV series with 2.5 - 3 mm stroke.
Product: Thermoelectric actuator MT010 from Honeywell — Technical features:

- Power supply 24 V, 50/60 Hz (-10% +20%)
- modulating control (0 – 10V DC)
- Maximum stroke: 3,5 mm
- Running time: 30 s /mm
- Power consumption: 1,5 W
- Protection standard IP 40

Figure 7-23.
Honeywell thermoelectric actuator MT010

TRVs with clock program

Figure 7-24.
Programmable TRVs:
Honeywell Radiatronic HR40

Programmable TRVs have their own programmer that facilitates the selection of a clock program for each individual radiator (room). Positions of the TRV are altered with a battery powered sprocket wheel or similar device.

Product: Honeywell Radiatronic HR40 — Technical features:

- Week program with two selectable comfort- and setback periods per day
- Comfort- and setback temperatures adjustable
- PI-control
- Automatic switch from summer to winter time
- Automatic adjustment to stroke of valve-part
- Options for temporary manual override (big knob)
- 2 x 1,5 V alkaline batteries (type AA)
- Lifetime expectancy approximately 2 years
- Optional: power supply and remote control
- Valve is automatically closed when window is opened

Figure 7-25.
Danfoss RA-PLUS

Product: Danfoss RA-PLUS — Technical features:

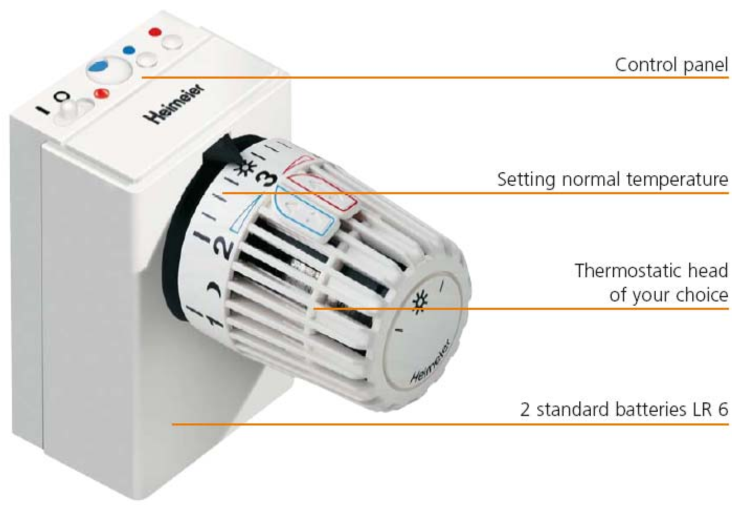
- Two programs with each three selectable comfort- and setback periods per day
- Comfort- and setback temperatures adjustable
- 2 x 1,5 V alkaline batteries (type AA)
- Lifetime expectan cy approximately 4 years

Self learning motorized TRV

For a lot of people it is not that easy to set a weekly clock program. To overcome this problem Heimeier introduced a TRV-device with self learning capabilities. The operation is easy: first select one of the two options for the program period you prefer: “day-program”or “week-program”. To program the unit for the selected period, just consequently operate the blue button when night setback temperature is required (room remains empty) and the red button when comfort-levels are required (room is occupied). After one week (or one day) the unit has learned its clock program.

Figure 7-26a.
Product: E Pro from Heimeier.
Technical features:

- Week program with 4 selectable comfort- and setback periods per day
- Proportional control
- Options for temporary manual override
- 2 x 1,5 V alkaline batteries (type AA)
- Lifetime expectancy up to 5 years
- Valve is automatically closed when window is opened
- Protection standard IP 20



Decentralised Heating Pumps (radiator pumps)

Pump manufacturer WILO is field testing an innovative system whereby the flow through the radiator is regulated directly by a high-efficiency radiator-specific pump. WILO claims that a network of these pumps gives distinct advantages in terms of electricity use (pumps use very little electricity ca. 1-1,5 W+ no system pump is needed), hydraulics (perfect balance, 0% bypass losses) and controls (e.g. after night-setback). No cost-estimate is available yet.

Figure 7-26b.
Decentralised Heating Pump

7.5 Combined room- and feed temperature controlled systems

7.5.1 Principle

Over the years (starting in the 90-ties) the weather controlled system and the single room thermostat controlled system have been mingling. Both systems added technical features from the other system in order to improve on their functionality and today several weather controlled systems have incorporated the option of switching to a room controlled system. On the other hand, several room controlled systems offer accessories that facilitate the expansion of a single room thermostat controlled system to a weather controlled system, while maintaining the original control option.

Mean reason for this probably is the fact both control systems inherently have shortcoming that can partly be solved by “the other” system.

The problem with single room thermostat controlled systems is the fact that the other rooms are only heated when there is a heat demand in the reference (living) room. A frequently cold bedroom or study is the consequence.

Drawback of the weather controlled systems is the fact that the temperature in the rooms (e.g. in the living room) is not always adequately be controlled. The parameters involved to achieve a good controlled room temperature are too many:

- Position of the outdoor sensor
- Selection of heat curve
- Size of radiators in the room
- Hydraulic balance of the system
- Type of TRV

The combination of both control systems result in the following control options:

1. Fully weather dependent feed temperature control
2. Fully room thermostat controlled systems.

3. Weather dependent feed temperature control with room temperature compensation

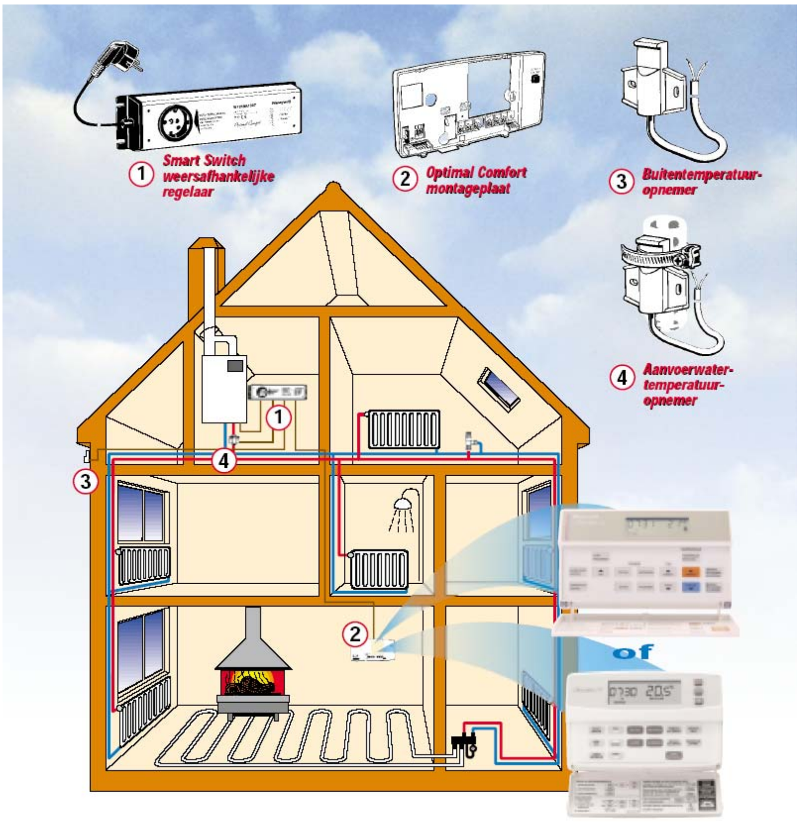
The last option is the gain of combining both systems and can help solving the shortcomings of the two separate systems. The extend to which the ΔT (difference between set point and actual room temperature) corrects upon the weather controlled feed temperature can - in some of these combined systems - be adjusted, for instance from 1 to 100%.

7.5.2 Product examples

Honeywell Optimal Comfort System.

Add-on system for room thermostats Chronotherm III and Chronotherm IV

Figure 7-27.



The system consists of:

1. Smart Switch weather dependent controller
2. Mounting plate (with additional switch) for the room thermostat Chronoterm II of IV
3. Outdoor temperature sensor
4. Feed water temperature sensor

7.6 Multiple room temperatures controlled systems

7.6.1 Principle

Multiple room temperature controlled systems are relatively new systems on the market. The principle is that the heat demand in each individual room is measured and communicated with a central controller. The central controller determines the overall heat demand and optimizes the feed temperature to this specific and continuously varying heat demand. The result is that feed temperatures can continuously be optimised (= minimised) which leads to higher system efficiencies and comfort levels.

The prevent the installation of multiple cables, running from each room to a central control unit and to the boiler, these systems generally are RF-based and facilitate wireless communication.

Combined with thermal actuators on the TRVs of each radiator/emitter, the central controller can be used to program the temperature levels for each individual room over each day of the week.

7.6.2 Product examples

Viessmann Vitohome 200

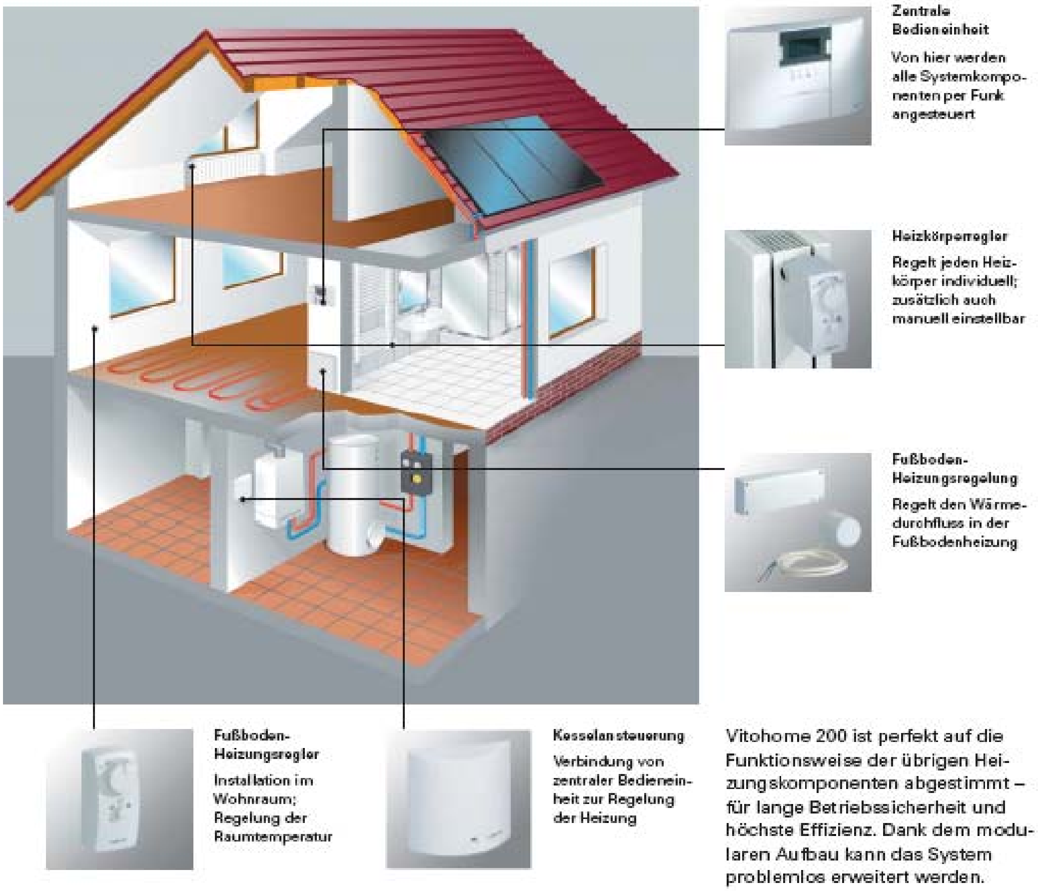


Figure 7-28.
Vitohome 200, Viessmann

Technical features

Figure 7-29.
Central programming unit Vitohome 200

From here, all system components are controlled by radio waves.

- Wireless radio range: 12 m
- Number of wireless radiator control units: max. 30
- Power supply: 230 V – 50 Hz - 2 pin euro-plug
- Power consumption: typical 1,5 W; max. 15 W.
- Radio frequency: 868 MHz
- Protection: IP 32 to EN 60529

Figure 7-30.
Wireless radiator control unit

Wireless radiator control unit

- Controls each radiator individually; also suitable for manual adjustment.
- Operating modes: automatic (clock), frost protection, constant heating
- Power supply: 3 V Lithium battery (3 pieces)
- Lifetime expectancy batteries: 5 years
- Connection M30 x 1,5
- Protection: IP 32 to EN 60529
- Closes valve when window is opened (sudden big ΔT-change)

Figure 7-31.
Wireless room temperature sensor.

Wireless room temperature sensor

Can be used if a wireless radiator controller is located in an unfavourable position, or in combination with under floor heating systems.

- Operating modes: automatic (clock), frost protection, constant heating
- Power supply: 3 V Lithium battery (1 piece)
- Lifetime expectancy batteries: 5 years
- Connection M30 x 1,5
- Protection: IP 32 to EN 60529

Figure7-32.
Under floor heating control unit

Under floor heating control unit

A maximum of 6 control circuits can be connected to this control unit; a maximum of 10 servo motors can be controlled.

- Power supply: 230 V – 50 Hz - 2 pin euro-plug
- Power consumption: typical 8 W; max. 32 W.
- Output voltage: 24 V
- Protection: IP 32 to EN 60529

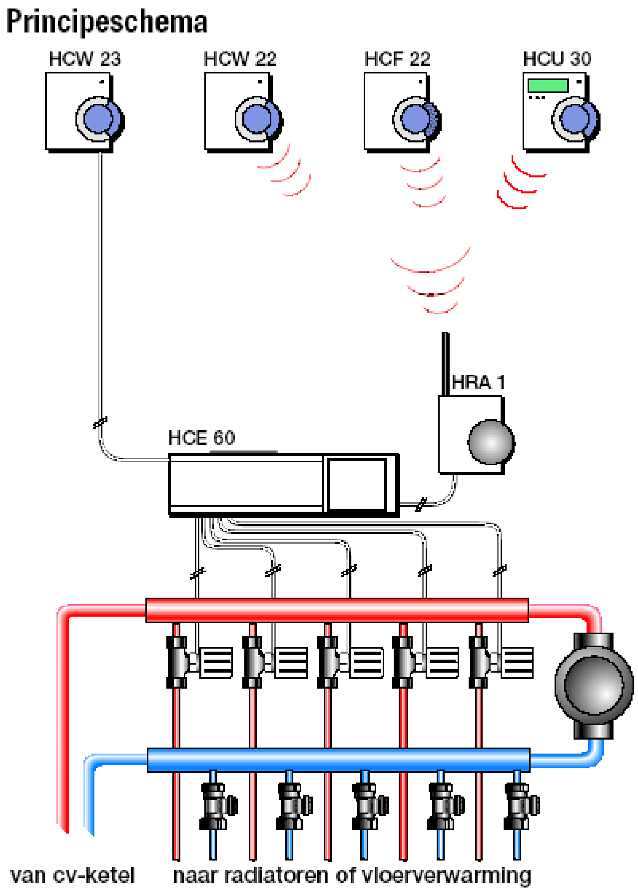
Figure 7-33.
Wireless boiler control unit

Wireless boiler control unit

This unit provides a demand-dependent boiler water or flow temperature. The heat demand of the Vitohome 200 is received via a radio signal and transmitted via the KM BUS.

- Rated voltage 24 V ; Power consumption from the KM BUS max. 6 mA
- Operating temperature up to 55°C
- Distance from the Vitohome 200 up to 12 m
- Protection IP 32 to EN 60529

Honeywell zone- and under floor heating control system



HCU30
Wireless clock controlled room temperature sensor (for room temperature clock programs, temporary adjustments and wireless communication on actual temperature)

HCF22
Wireless room temperature sensor (for room temperature adjustments and wireless communication on actual temperature)

HCE60 Controller
Can control 5 different heating groups (expandable to 8 groups). The controller operates the thermal actuators on the distribution unit on the basis of the information received from the wireless room temperature sensors.

Figure 7-34. Honeywell zone- and under floor heating control system

This system does not use the information of the individual heat load from each room to optimise the boiler feed temperature. It just controls the TRVs on the basis of the difference between temperature set point and actual temperature.

7.7 Occupancy control

7.7.1 Principle

In an ideal situation, habitable rooms are only heated when people are present. When they have left the room, the predefined setback temperatures are pursued.

With fully manual controlled systems (e.g. rooms with thermostats, TRVs) this can only be achieved with a strict discipline of the users, turning the thermostat low when he/she leaves the room. Obviously, in practice this will not be the case and energy is lost.

A better option is the use of clock programs that enable the user to preset the periods in which people are present or not. Setting the correct clock program however isn't the favourite occupation of the user, and often default programs are used that at least ensure that the room is at the requested comfort levels when people are present. Clock programs are seldom updated when the schedules of the inhabitants change. As a result more energy is used than strictly necessary.

From energy perspective, the best option would be a system that facilitates an occupancy controlled heating per room. Self learning algorithms can help optimising the on and off switching times for comfort- and efficiency purposes. The energy losses due to unnecessary room heating can in this way be minimised.

(No documentation was found on reported savings due this these type of occupancy sensors, but it is obvious that these savings are substantial compared to non occupancy controlled systems).

Table 7-3. Overview of presence detection sensors

Type of sensors	Information grade / resolution	Information: Movement detection	Number of occupants	Person identification	Person localisation	Physical activity	Price	Main problems
Passive IR	low	+	-	-	-	+/-	low	low resolution
Light barriers	low	+/-	+	-	-	-	low	low resolution
Microwave detectors	low	+	-	-	-	-	medium	
Ultrasonic (simple)	low	+	-	-	-	-	low	low resolution
Ultrasonic (intelligent)	relatively low	+	+/-	-	+/-	+/-	medium	low price / information grade relation
Shock sensors	high	+	-	-	-	+/-	medium	relatively complex installation
IR camera	very high	+	+	-	+	+	very high	very high price
360° PIR	very high	+	+	-	+	+	medium	mechanical noise
Transponder	low	-	-	+	-	-	medium	low information grade, short range

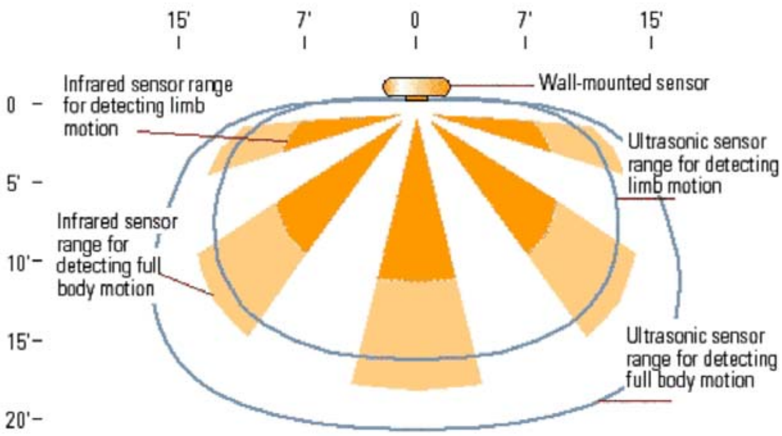
Occupancy sensors, also called motion sensors, react to disturbance in a space. They are designed to turn lights or other equipment on when the space is occupied and off (after a specified time delay) when the space is vacated. The most common load controlled is lighting; however, more recently occupancy sensors can be designed to control heating ventilating and air conditioning (HVAC), and office equipment (monitors, printers, fans, etc.). The two most common sensor types are passive infrared, which require a direct line of sight to the movement of infrared (heat) sources, and ultrasonic, which detect any movement, human or otherwise (for example, curtains).

Passive infrared (PIR) — PIR sensors, the most commonly used type, are able to "see" heat emitted by occupants. Triggering occurs when a change in infrared levels is detected, as when a warm object moves in or out of view of one of the sensor's "eyes." PIR sensors are quite resistant to false triggering. They are best used within a 15-foot range for two reasons: first, there are potential "dead" spots between their wedge-shaped sensory patterns that get wider with distance (**Figure 1**); and, second, being passive, they do not send out any signal. Instead, PIR sensors depend on the intensity of the heat output of the moving part of the subject.

Ultrasonic (US) — Ultrasonic sensors emit a high-frequency (more than 20,000 cycles per second) sound above human and animal audibility ranges and listen for a change in frequency of the reflected sound. Because they emit a signal instead of receiving it, they are able to cover larger areas than PIR sensors and are more sensitive. US sensors are prone to false triggering and can be set off by air movement, such as that produced by a person running by a door or the on-off cycling of an HVAC system.

Microwave (MW) — Microwave and audible sound sensors are less common. Audible sound sensors, which listen for noise made by people or machines, are best applied in an industrial facility or warehouse. Microwave sensors are similar to ultrasonics, in that they emit a signal and measure a change in frequency when that signal is reflected.

Figure 7.35.
Sensor coverage diagram



Ultrasonic sensors can detect motion at any point within the contour lines. Infrared sensors see only in the wedge-shaped zones, and they don't generally see as far as ultrasonic units. The ranges are representative; actual sensors may be more or less sensitive.

Hybrid or dual-technology sensors incorporate features of both PIR and US sensors—or of other sensor types, such as microwave—in one sensor. The most common combination of sensor types is that of PIR and ultrasonic sensors, to take advantage of the PIR units' resistance to false triggering and the sensitivity of ultrasonics.

7.7.2 Product examples

Smart Systems, People Sensing Automatic Setback Thermostat

Figure 7.36.
Computerised thermostat SS5000 (left), communicating with wireless occupancy sensor SS2000 (right)

Technical features SS5000 thermostat:

- Operating voltages: 10 – 30 VAC (can also use 7 to 12 VDC
- Time settings for comfort temperature recovery: 2 to 99 minutes
- Radio frequency: 49,85 MHz
- Accuracy: ± 1 F
- Relays: 5

Technical features SS2000 Occupancy Sensor:

- Power supply: 4 AA batteries
- Lifetime expectancy: 3 to 4 years
- Resistant to sun, TV, moving curtains
- Uses digital signal processing to recognize people (incl. sleeping people)
- Dual element pyro-electric infrared sensor
- Detection range: 30 feet

The SS5000 uses a wireless radio network to communicate with occupancy sensor and in this way determines whether or not a room or space is occupied. When people are present the SS5000 maintains comfort levels; when vacant the unit automatically reduces the temperature set point in the room. The unit constantly performs calculations to ensure that the comfort temperature level is achieved within the specified timeframe upon occupants return. The microprocessor inside also records detailed occupancy and heating usage data.

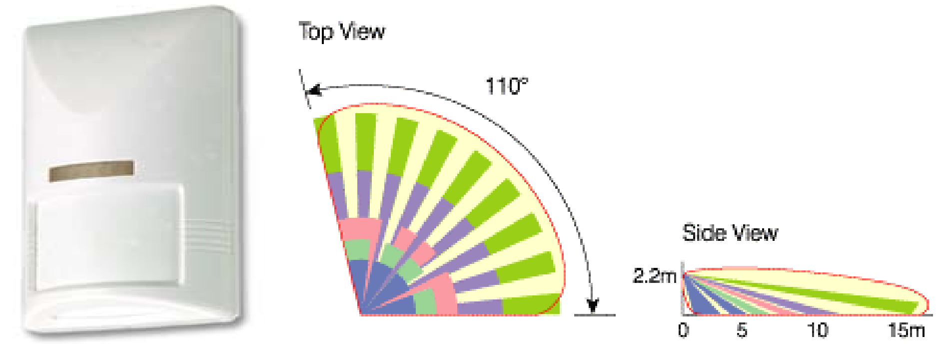
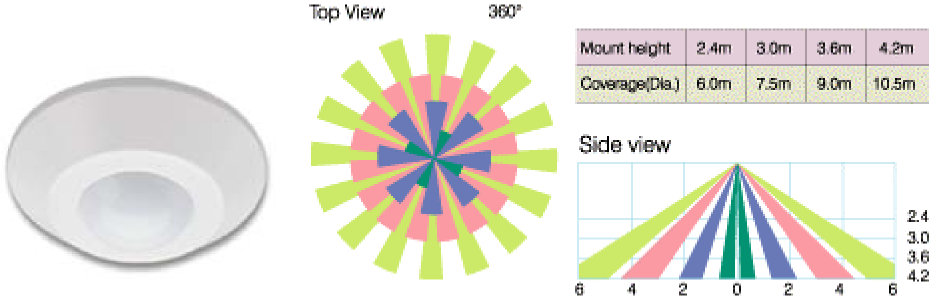


Figure 7-37.
IR-Tec OS-550DT Dual Technology Occupancy Sensor.
Technical features:

- Combined PIR and Microwave sensors in one
- Programmable ON and OFF delays
- Power supply: 18 – 26 VAC/DC
- Current: 20 mA@24 VDC

Detection range: 110 o; 15 m (at 25°C)



Mount height	2.4m	3.0m	3.6m	4.2m
Coverage(Dia.)	6.0m	7.5m	9.0m	10.5m

Figure 7-38.
IR-Tec OS-362D Duplex Occupancy Sensor.
Technical features

- Omni-directional infrared sensor
- Selectable ON & OFF delays
- Power supply: 24 ± 2VAC/DC
- Current: Standby 8 mA; Active 20 mA@24 VDC

7.8 Summary

Room Temperature Control Systems have a large influence on the energy consumption for space heating. They relate to three different parameters that directly influence the energy consumption, which are:

1. Feed water temperature
2. Room temperature variations
3. Unnecessary Heating of unoccupied zones

Ad 1. Feed water temperature

Room Temperature Control Systems to a large degree determine the boiler feed temperature. On/off room thermostat controlled systems for instance allow the boiler feed temperature to rise to maximum values, while modulating room thermostats anticipate the amount of thermal energy that is needed which will reduce the feed temperatures like wise. Weather controlled systems related the feed temperature to an outside temperature which will result in a feed temperature varying over the heating season from e.g. 80 to 40 degrees.

Because the feed temperature influences to the return temperature, and the return temperature influences the generators efficiency, we can conclude that the Room Temperature Control System influences the real life efficiency of the generator.

Generally speaking, Room Temperature Control Systems do not optimise (= minimise) the feed temperature

and higher feed temperatures are applied, causing a 2-2,5 % efficiency loss per 10°C feed temperature increase (for condensing boilers and provided ΔT is kept constant).

Ad 2. Room temperature variations

Further more, a non-ideal room temperature control causes temperature variations and drifts around the prefixed set point temperature, due to the physical characteristics of the control system, sensor locations and characteristics of the heating system itself. This may result in increased or decreased heat losses through the building envelope. Also the ability to utilise internal gains (from people, equipment, solar radiation) depends on the type of heat emission- and control system. The heat loss due to this phenomena can be calculate in different ways. The draft version of prEN 15316-2-1 provides examples of values for these control losses in the informative annexes B.2 and C. For dwellings with an average heat load of 40 kWh/m²/a and radiators with thermostatic valves as emitters, the losses may vary from 1 to 7 % of the heat theoretically required, depending on the type of room temperature control system that is used. The 7% related to a system with an on/off room temperature control with hysteresis and no feed temperature control. The 1% corresponds with an electronic room temperature controller with a proportional band of 1°C with additional feed temperature control on the basis of the outside temperature (Appendix B2 of prEN 15316-2-1).

Ad 3. Heating unoccupied rooms

The heating of unoccupied zones beyond setback temperature levels are generally considered a waste of energy. Room Temperature Control Systems define to which degree the temperature in individual rooms can be controlled and related to the actual accupancy. The following classification can be used:

1. no or limited control
2. manual controlled
3. clock program controlled
4. occupancy sensor controlled

Obviously the last classification represents the current state of the art and will result in minimal energy losses due to the heating of unoccupied zones. The other classes will cause increased losses with a diminishing degree of control. Clearly these energy losses can be substantial. To give some examples:

Example 1

An average house (living room, 3 bedrooms and a bathroom) with a on/off room thermostat (in the living room) controlled system with a correctly set clock program and traditional radiator thermostats on all radiators may use 20 – 40% more energy when the radiators valves in bath- and bedrooms are not operated (remain open) and these rooms remain unoccupied until bedtimes (4 smaller radiators are heated together with the radiators in the livingroom). RTVs will give some improvement, because temperature overshoot are avoided (estimated losses 10 – 30%). With a weather controlled system and RTVs in every room these losses related to "heating of unoccupied zones" would be similar, but in this case savings can be made on the generator efficiency because of the average lower feed temperatures (see ad 1)

Example 2

If this house uses a weather controlled system with clock programs on each TRV, the losses related to "heating of unoccupied zones" would reduce considerately and can in this case be related to degree in which the various clock programs correspond to the real life occupancy. Losses of 5 – 10% are estimated.

Example 3.

The optimal situation is an occupancy sensor controlled thermostat in each room. In this case losses can only be related to the comfort levels that are required (selected ON/OFF –delays).

8 DISTRIBUTION SYSTEM

8.1 Introduction

The "Distribution System" is defined as the agglomerate of pipe work, fittings, valves, applied insulation, etc. that is necessary to distribute the thermal energy to the heat emitters.

The energy losses caused by this "heating system component" are caused by:

1. heat losses in the distribution system
2. losses due to unnecessary hydraulic power consumption

The nature and relevance of these losses will be further discussed in the following two paragraphs.

8.2 Heat losses

The distribution system emits heat; the amount of heat that is emitted depends on:

θ_m : the average of the feed and return temperature [°C]

θ_a : the ambient temperature [°C]

t_H : the time during which the distribution system is heated [s]

L : the size (total length) of the distribution system [m]

U' : the thermal conductivity per length for every pipe material used [W/mK]

The heat that is emitted from the distribution system to the surrounding air or building construction, can partly be considered as useful, because some of the heat is actually transferred to occupied spaces that need heating. How to value the heat emissions from the distribution system is rather arbitrary.

The prEN 15316-2-3 "Method for calculation of system energy requirements and system efficiencies – Part 2 – 3: Heat distribution systems" uses the following approach:

- the heat that is transmitted by those parts of the distribution system that are located in heated rooms can be considered useful (= $Q_{D,h}$)
- the heat that is transmitted by the parts of the distribution system that are located in unheated rooms can be considered as losses (= $Q_{D,u}$)

The total amount of heat that is transmitted by the distribution system (Q_D) is – according to prEN 15316-2-3 — the sum of these two values. In formula this is:

$$Q_D = Q_{D,h} + Q_{D,u}$$

Of course it can be debated whether the heat that is transmitted to heated rooms when the room is not occupied can be considered as useful, when it contributes to a room temperature above setback levels.

The total amount of heat that is emitted by distribution system can be calculated with the following formula:

$$Q_D = \sum_i U_i' * (\theta_m - \theta_{a,i}) * L_i * t_H$$

In which *"i"* is the index for pipes with the identical specifications.

Because not only pipes emit heat, but also all other additional components used in the distribution system (supports, valves, distribution units, etc.) a multiplier is used that increases the actual total pipe length, an as such incorporates the losses for these other components.

Clearly the *U'*- value and the average boiler temperature θ_m play an important role in minimising these distribution emission losses.

Table A.2 of prEN 15316-2-3:2005 gives values for the total heat emitted by the distribution system per year for an average house with an average distribution system. The table gives values for different design temperatures and different surfaces of heated areas. The U'-values are 0,2 W/mK for the pipes in unheated parts of the house and 0,255 W/mK for pipe work in the heated rooms.

Table 8-1. Table A.2 from prEN 15316-2-3:2005

	Annual heat transmitted by distribution system at different design temperatures in kWh/yr (heating period: 5000 hours)			
Heated surface A [m²]	90/70°C	70/55°C	55/45°C	35/28°C
100	3508	2546	1861	834
150	4827	3488	2534	1102

For an average house with an annual heat load of 7250 kWh and an energy consumption of 10110 kWh per year, these values for the heat transmitted by the distribution system would represent 8 to 35% for a 100 m³ heated surface at design temperatures of respectively 35/28°C and 90/70°C.

If we assume that approximately 1/3 of these values can actually be related to heat losses (1/3 of pipe length situated in unheated areas), we are talking about a loss of 278 kWh/a and 1170 kWh/a (respectively 2,7 to 11,5 % of the yearly energy consumption of 10110 kWh).

The U-values used in this example of the average house are already quite good. If we were to make the same calculation for a not insulated distribution system (U = 1 W/mK), the values would be approximately 5 times higher, meaning that distribution losses could rise to 1390 kWh/a at 35/28°C and 5850 kWh/a at 90/70°C.

No documented studies were found, where the percentages of houses with insulated distribution system are mentioned. If these percentages are similar to the figures related to floor and roof insulation in for instance the Dutch market we are talking about 40 – 60% of the houses.

8.3 Hydraulic losses

Electric energy is required to circulate the hot water through the heating system. The amount of electric energy needed per year varies from approximately 100 kWh for smaller well designed systems with VSP –pumps to over 450 kWh for the bigger systems with single speed pump.

The amount of electric energy that is needed, depends on the pump, the way it is controlled and the hydraulic resistance in the heating system.

According to the prEN 15316-2-3:2005, the electrical energy needed can be calculated with the following formula:

$$W_{d,e} = W_{d,\,hydr} \cdot e_{d,\,e}$$

In which

$W_{d,\,e}$	= the required amount of electric energy	[kWh/year]
$W_{d,hydr}$	= the required amount of hydraulic energy	[kWh/year]
$e_{d,\,e}$	= multiplier fort he type of pump operation	[.]

The hydraulic energy that needs to be delivered by the pump is determined by the hydraulic system design point at average the load of the distribution system (β_D) and by the duration of the heating period

$$W_{d,hydr} = (P_{hydr} / 1000) * \beta_D * t_H * f_V * f_{Sch} * f_A * f_{Ab} \qquad \text{[kWh/year]}$$

In which:

P_{hydr}	= hydraulic energy at system design point	[W]
β_D	= average load of the distribution system	[.]
t_H	= duration of heating period	[h/year]
f_V	= correction factor for feed temperature control	[.]
f_{Sch}	= correction factor for type of hydraulic circuit	[.]
f_A	= correction factor for sizing of radiators	[.]
f_{Ab}	= correction factor for hydraulic balancing of heating system	[.]

The multiplier $e_{d,\,e}$ for the type of pump operation (part load efficiency, control efficiency etc.) can be determined with:

$$e_{d,e} = f_\eta * f_{TL} * f_{Ausl} * f_R$$

In which:

f_η	= correction factor related to pump efficiency	[.]
f_{TL}	= correction factor for part load	[.]
f_{Ausl}	= correction factor related to the actual pump operating point	[.]
f_R	= correction factor for type of pump control	[.]

The hydraulic losses that are referred to in this chapter, relate to electric energy consumption of the pump that could have easily been avoided. Minimizing power consumption for heat distribution, primarily is a task for the person that designs the heating system but in most cases also for the manufacturer of the boiler. After all, most heating systems are designed and installed as single home heating systems and the pump in the boiler often is the only pump that is used. Since boilers are mass produced product, the pump that is selected by the manufacturer must be able to cover a large range of distribution systems with varying hydraulic resistance. The pump selected will therefore generally be to big and will most of the time operate far below its best efficiency point.

Without going in to too much detail, the steps that can be optimised to minimise the electricity consumption of the pump, are mentioned below:

1. Minimize hydraulic resistance (P_{hydr}) in the system (in other words: optimise layout of piping).
2. Select a pump with a BEP (Best Efficiency Point) as close as possible to the actual operating point (f_{Ausl} , f_{TL} and f_η).
3. Select a pump with a high electrical efficiency (f_η)
4. Use a variable speed pump that is controlled either by the ΔP and/or ΔT (f_R)
5. Select the correct operating range for the pump (switch position nr. 1,2 or 3)
6. Be sure that the system is hydraulically balanced (f_{Ab})

An optimal hydraulic design of the distribution system and a correct selection of the pump can reduce electricity consumption for the pump considerably.

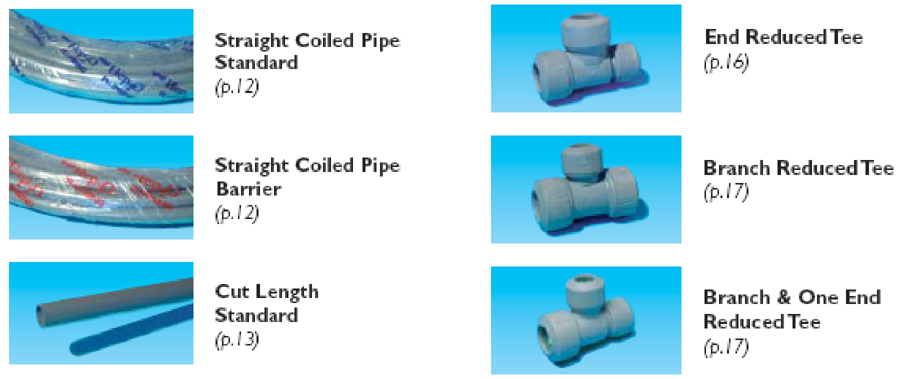
Comparing an uncontrolled single speed circulator with a ΔP-controlled variable speed pump already can reduce the power consumption with more than 40%. With a state of the art permanent magnet motor this ΔP-controlled variable speed pump can even save up to 70%.

Assuming the average house with an average annual energy consumption for heating of 10110 kWh, the average pump covers approximately 5% of this figure (primary energy use is approximately 500 kWh/yr).

Savings of 40 – 70% on the pumps power consumption relates to approximately 2 to 3,5% of the overall annual consumption for heating.

8.4 Piping components and materials

Apart from the traditional metal piping, new plastic products are on the market and suitable for application in heating and hot water systems. Because the pipes are flexible, bends can easily be made without additional tooling or (hydraulic resistance increasing) components. Furthermore U-values are generally better for plastic piping than for metal plumbing components and materials. Although the specific heat of the plastic tubing is higher than for metal pipes, this will reduce the heat losses in the distribution system



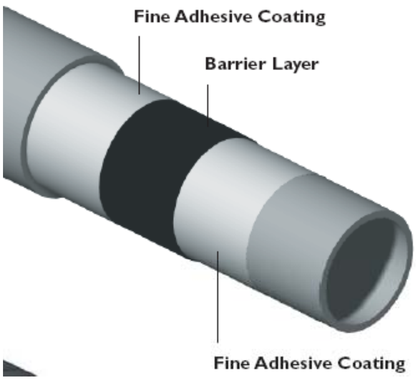


Figure 8-1.
Hep2O polybutene piping components and materials.
Technical data polybutene components:

- nominal diameters pipe: 10, 15, 22, 28 mm
- related minimum bend radii: respectively 80, 120, 176, 224 mm
- lifetime expectancy > 50 years
- designed for operating conditions between 12 bar/20°C and 6 bar/90°C
- manually bendable
- thermal conductivity: 0,22 W/mK
- exposure to sunlight must be avoided
- no transmission of noise through piping

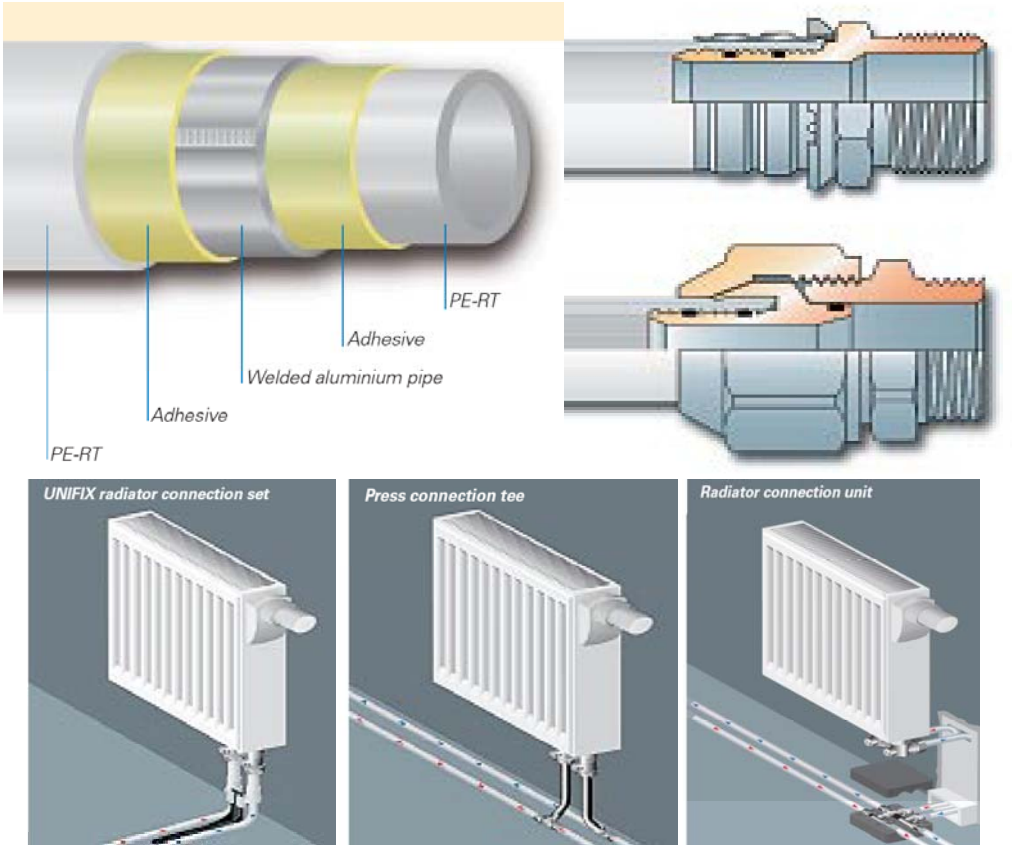


Figure 8-2.
Unipipe plumbing components.
Technical data unipipe components:

- nominal diameters pipe: 14, 16, 18, 20, 25, 32, 40, 50, 63 mm.
- easy to bend manually
- 100% diffusion tight
- Max. constant pressure: 10 mbar
- Max. constant temperature: 95°C
- thermal conductivity: 0,40 W/mK
- corrosion and sunlight resistant

Eco profiles

The plastics technology department of the Technical University Berlin has conducted an environmental analysis on drinking water installation systems, utilising their self-developed standardised comparison method VENOB (German: vergleichende normierende Bewertung).

The work was supported by the Kunststoffrohrverband (KRV) and various pipe producers. The environmental analysis is based on scientific fact. It summarises and compares the energy consumption and the emissions in air, water and soil during the individual stages from raw material production to the installation in the building.

The TU Berlin looked at 6 different raw materials for pipes used in drinking water installations according DIN 1988 Part 3, for a multiple dwelling with 16 apartments with central warm water distribution (pressure of supply: 4 bar). Figures below show the outcome of this study.

Materials investigated:

- Metals: galvanised steel, copper
- Plastics: PE-X, PB-1, PP-R, PVC-C

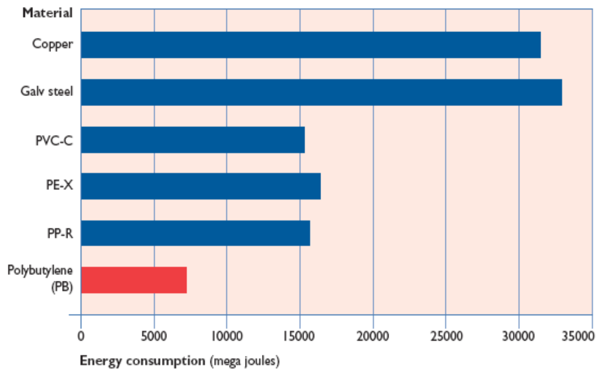


Figure 8-3.
Energy Equivalent Value of a complete piping system for a multiple dwelling with 16 units.

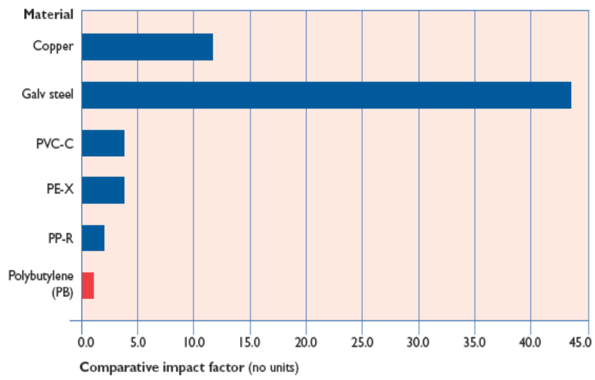


Figure 8-4.
Standardised Comparison (VENOB) of various pipe materials.
Emissions in Soil.

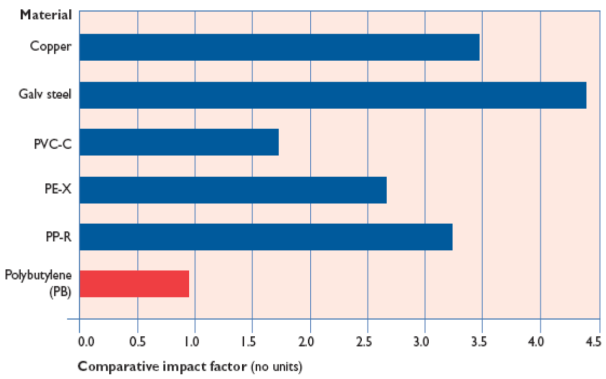


Figure 8-5.
Standardised Comparison (VENOB) of various pipe materials.
Emissions in Water.

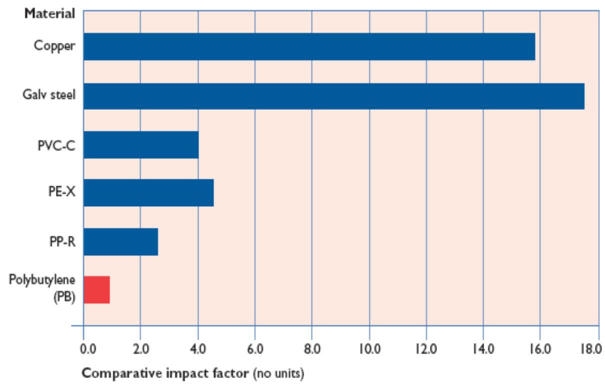


Figure 8-6.
Standardised Comparison (VENOB) of various pipe materials.
Emissions in Air.

8.5 Variable Speed Pumps[37]

High-efficiency pumps have been available predominantly for large buildings until now, but a few years ago pump manufacturers introduced high-efficiency pumps for the domestic marketplace.

The technology of this new generation of pumps reduces the energy consumption by up to 70 per cent compared with conventional single speed pumps. These pumps are equipped with EMC motor technology

(Electronic Commutated Motor) and, in addition, the hydraulics have often been optimised. Due to the 'permanent magnet technology' used in the new pump, the starting torque is three times higher than that of any previous small heating pump which reduces the risk of blockages. All high-efficiency pumps are equipped with thermal insulating shell as standard, which minimises the heat loss through the pump housing.

[37] Grundfos notes that the shown pumps does not fit to the prices VHK is using in the LCC calculation. But most of the described functions of these stand-alone pumps could be created as well with a pump installed inside a boiler, if the boiler control forms a functional unit with the controlled pump.

Figure 8-7.
Wilo Stratos Eco series
Technical features:

- ΔP controlled rotation speed (ΔP manually adjustable)
- Automatic reduction flow speed during T-setback
- Permanent magnet rotor
- Statos-Eco series facilitates heads from 3 to 5 m at speeds from 1400 – 3500 rpm
- Power consumption: from 5,8 to 59 watts
- Weight: 2,5 kg
- Material housing: cast-iron
- Protection: IP44

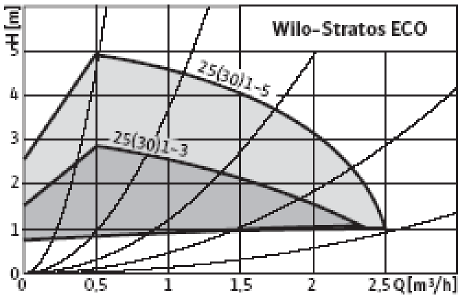


Figure 8-8.
Grundfos Magna — Technical features:

- Operating modes: ΔP controlled rotation speed (with or without auto adapt) or constant pressure.
- Automatic reduction flow speed during T-setback
- Permanent magnet rotor
- Magma 25-60 series facilitates heads from 1 to 5 m at speeds from 1400 – 3800 rpm
- Power consumption: from 10 to 90 watts
- Weight: 5,4 kg
- Material housing: cast-iron
- Protection: IP44 na 25-60

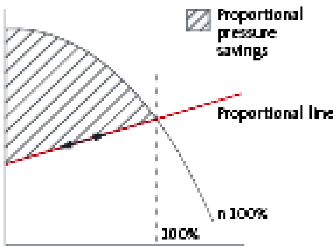


Figure 8-9.
Proportional-pressure control

In units employing the proportional-pressure control principle, the differential pressure across the pump is automatically adjusted to match the flow. When the flow falls, so does the pressure required. This results in a correspondingly reduced load on the motor – and reduced energy consumption. And things get even better when you add the AUTO*ADAPT* function (see next graph).

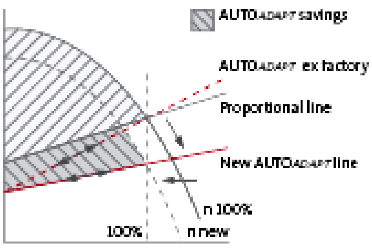


Figure 8-10.
Proportional-pressure control with AUTO*ADAPT*

At start-up, a MAGNA pump operates with a lower differential pressure than other comparable units (shown by the AUTO*ADAPT* factory setting line). As the flow increases, the pump pressure follows the line for the AUTO*ADAPT* factory setting until the pump operates on the maximum curve, continuing downwards until it reaches the required flow.

When the flow is subsequently reduced, the AUTO*ADAPT* function ensures that the operating profile does not simply return to the original curve – it sets a new, lower pump speed (n new) that results in greater energy savings!

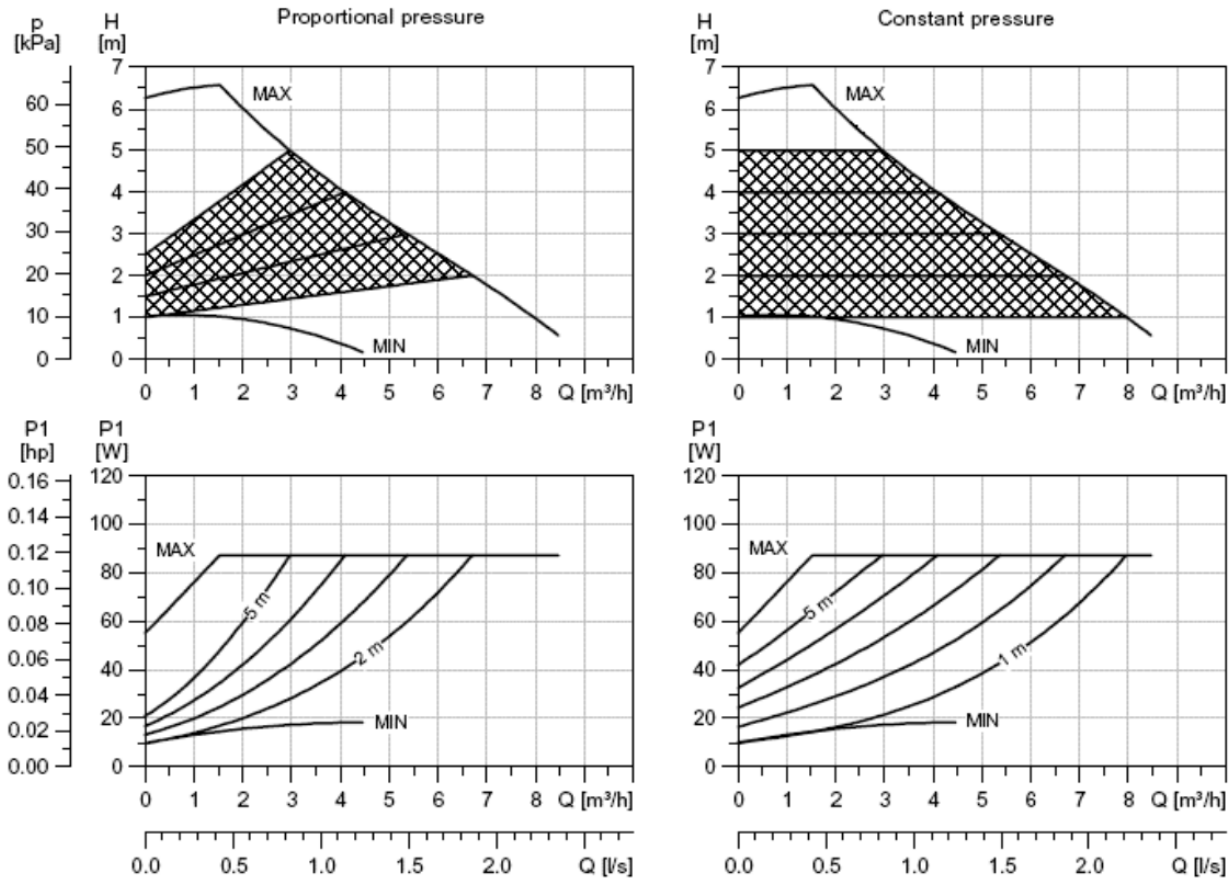


Figure 8-11. Grundfos Magna 25-60

8.6 Summary

Feed and return water temperatures have a high impact on the energy losses that occur in heat distribution systems. For a well designed and insulated distribution system in an average house these losses will correspond with 0 tot 12% of the annual energy consumption for heating, depending on lay-out of the distribution system (situated in heated zones or not) and on the temperature regime. For not insulated distribution systems, these losses will be even higher.

Minimal losses (below 3%) are achieved at a temperature regimes with an average of 32°C (average = (T_{feed}+ T_{return}) / 2) and no or minimal piping in unheated zones. Maximal losses (12% and higher) are achieved when average temperatures are above 80°C and approximately 1/3 of the distribution system is situated in unheated zones.

For this reason the generator, with its controls, its pump and the selected room temperature control system will have a large influence on the annual losses of the distribution system that are partly located outside the heated zones. In other words, the generator (with its peripherals) to a large extend determine the heat distribution efficiency.

9 EMITTER SYSTEM

9.1 Introduction

The "Emitter System" is defined as the agglomerate of heat emitting devices that is used for the purpose of obtaining and maintaining a comfortable temperature in a room or building, either by radiation, convection or a combination of both.

Types of hydronic heat emitters most commonly used are:

- Radiator
- Convectors
- Surface heating (floor- and wall heating)

Emitters do not have an efficiency figure themselves. The just emit heat and the heat that is not emitted goes back into the distribution system. But *the way* heat is transmitted into a room, can influences the heat emission efficiency. That are two parameters that are generally linked to the efficiency of the emitter system, which are:

1. Non-uniform temperature distribution
2. Position of emitter and related additional heat transmission

In the following two paragraphs these parameters will be further discussed. In the second part of this chapter the emitter types, their technical features and their efficiency related features will be further discussed.

9.2 Non uniform temperature distribution

Calculation of the monthly or annual energy use according to EN 832 is based on the assumption that air temperature and mean radiant temperature are equal. In real life practice this will not be the case. For systems with a significant part of radiant heating and spaces with large cold surfaces, the mean radiant temperature may differ significantly from the air temperature. To ensure adequate comfort levels with low mean radiant temperature, mean air temperatures levels must be higher. Higher mean air temperatures will result in higher transmission losses and higher ventilation losses. In addition, convective systems will cause vertical temperature gradient (especially in high ceiling spaces) which lead to even higher mean air temperatures for a similar comfort level at 1 m above the floor.

A method for calculation of the additional heat loss is given in EN 12831.

It appears difficult to define general input and output data for these detailed methods and the product characteristics. CEN/TC 228/WG4 is dealing with this issue and will propose a practical approach. For the time being, the tabled values for energy losses due to non-uniform temperature distribution – as presented in the prEN 15316-2-1[38] - will be used as a reference.

The energy loss can be caused by:

- a temperature stratification, resulting in an increased internal temperature under the ceiling and upper parts of the room;
- an increased internal temperature and heat transfer coefficient near windows;

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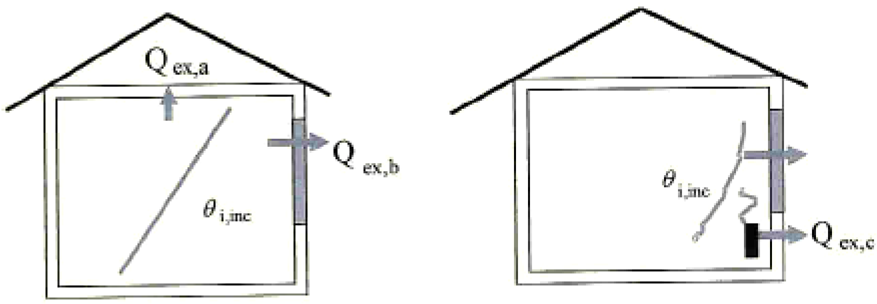
[38] Draft prEN 15316-2-1; Heating systens in buildings – Method for calculation of system energy requirements and system efficiencies – Part 2-1 : Space heating emission systems.

- convection and radiation from the heating system through other outside surfaces.

Figure 9-1.
Effects due to non-uniform temperature distribution



The heat loss due to a non-uniform temperature distribution is calculated using the general equation for transmission heat loss, taking into account the increased internal temperature, $\theta_{i,inc}$, and the increased heat transfer coefficient, which is included in the U-value, U_{inc}, of the surface area exposed:

$$Q_{em,emb} = \sum * A \cdot U_{inc} * (\theta_{i,inc} - \theta_e) * t \qquad [J]$$

where:

A	= area of the ceiling, outside wall behind heat emitter or window in square metres (m²);
U_{inc}	= calculated from the insulation of the surface and the surface coefficient in watts per square metres per Kelvin (W/m²·K). This is influenced by the convective air flow from the heat emitter, reflective material behind the heat emitter, etc.;
$\theta_{i,inc}$	= locally increased internal temperature in degrees Celsius (°C) which is a function of the heating system and the surface temperature or the supply air temperature of the heating system;
θ_e	= outdoor temperature in degrees Celsius (°C);
t	= time in hours (h).

The prEN 15316-2-1 contains the following approach for the tabled efficiency value for non-uniform temperature distribution (stratification).

If the efficiency, η_{em}, of the heat emission system is given, the additional heat loss due to the heat emission system, $Q_{em,str}$, can be calculated as:

$$Q_{em,str} = (1 - \eta_{em}) / \eta_{em} * Q_h \qquad [J]$$

Annex A and annex B of the prEN 15316-2-1 provide the following examples of efficiency values for heat emission systems.

Table 9-1. Heat emission efficiency for spaces with normal ceiling heights (< 4m); from informative annex A prEN 15316-2-1.

	Annual average heat demand in W/m²			
Heat emission system	< 20	20 - 40	40 - 60	> 80
Radiator under window	0,97	0,96	0,93	0,90
Radiator internal wall	0,94	0,94	0,93	0,93
Convector under window	0,93	0,93	0,89	0,86
Floor heating	1,00	1,00	1,00	1,05*
Ceiling heating	0,96	0,96	0,96	1,01*
Warm air heating	0,91	0,90	0,85	0,83

** The heat emission efficiency is > 1 because of the lower air temperature and thus less heat loss due to ventilation*

Another approach is the one used in EN ISO 13790, in which the heat loss due to non-uniform temperature distribution is calculated using an equivalent increased internal temperature difference. The equivalent increase in internal temperature may be used to calculate the corresponding increase in heat loss in two different ways:

- by multiplying the calculated building heat demand, Q_h, with a factor based on the ratio between the equivalent increase in internal temperature, $\Delta\theta_i$, and the average temperature difference for the heating season between the indoor and outdoor temperature for the space:

$$Q_{em,str} = Q_h \cdot (1 + \Delta\theta_i / (\theta_i - \theta_{e,avg})) \qquad [J]$$

- by recalculation of the building heat energy requirements, according to EN 832 or EN ISO 13790, using the equivalent increased internal temperature.

Annex D provides examples of values for the equivalent increase in internal temperature, $\Delta\theta_i$, for different types of heat emitters.

Table 9-2. Spatial temperature variation by type of emitter and the corresponding spatial variation class; from informative annex D prEN 15316-2-1, based on the French RT2000

Class of spatial variation	Heat emitter	Spatial variation for ceiling height < 4m (K)	Supplementary spatial variation for ceiling height > 4m (K/m)
A	Floor heating	0	0
B	Cassettes, tubes and radiate ceilings	0,5	0,2
C	Other emitters	0,5	0,4

On the basis of this theoretical approach we may roughly conclude that the energy losses due to non-uniform temperature distribution may vary between 1 and 10%, depending on type of emitter and type of building.

9.3 Position of emitter

Embedded surface heating devices may cause extra heat losses, due to additional transmission to the outside

This applies to floor heating, ceiling heating and wall heating systems and similar. This is only considered as a loss when one side of the building part containing the embedded heating device is facing the outside, the ground, an unheated space or a space belonging to another building unit. If embedded heat emitters with different characteristics (e.g. insulation) are used in the heating installation, it is necessary to take this into account by separate calculations.

Note
If the increased temperature in the building element has been taken into account in the calculations according to EN 832 or EN ISO 13790, this shall not be done again. For a slab on ground, it is for large buildings important to use the equivalent U_e value according to EN 13870 or EN 12831.

The heat loss due to additional transmission to the outside is calculated as follows (see figure below):

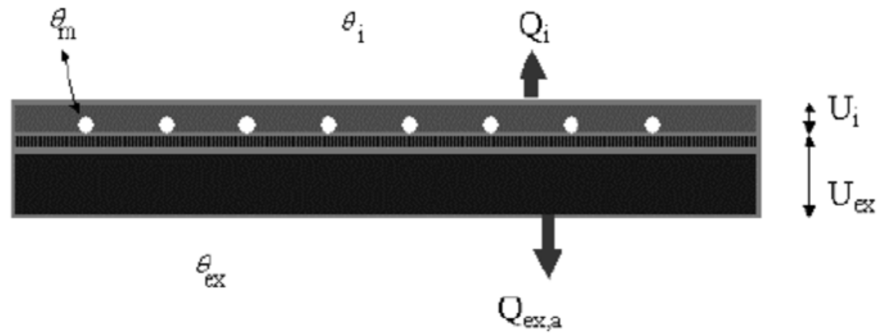


Figure 9-2.
Heat transfer of embedded surface heat emitter

Necessary room heat emission:

$$Q_i = A * U_i * (\theta_m - \theta_i) * t \qquad [J]$$

Heat loss to the other side:

$$Q_{ex,a} = A * U_{ex} * (\theta_m - \theta_{ex}) * t \qquad [J]$$

Combining these equations gives:

$$Qex,a = [(Uex / Ui) * Qi + A * Uex * (\theta i - \theta ex)] * t \qquad [J]$$

where:

A	= surface area with embedded heating device in square metres (m²);
U_{ex}	= heat transfer coefficient between the level of the heating medium and the outside, ground, neighbouring unit, or unheated space in watts per square metres per degrees Celsius (W/m²*°C);
U_i	= heat transfer coefficient between the level of the medium and the heated space in watts per square metres per degrees Celsius (W/m²*°C);
θ_m	= average temperature at the level of the heating medium in degrees Celsius (°C);
θ_{ex}	= external temperature, ground temperature, temperature in neighbouring unit or temperature in unheated space in degrees Celsius (°C);
θ_i	= internal temperature in degrees Celsius (°C);
t	= time in hours (h).

Heat transfer to the ground can be calculated according to ISO EN 13370.

9.4 Radiators

9.4.1 Introduction

Technically speaking, radiators are no more than flat metal (steel, cast iron or aluminium) containers that are filled with warm/hot water. These flat containers can be shaped and arranged in numerous ways to finally become a radiator. The water content of radiators may v ary from a couple of liters for a small radiator to over a hundred litres for the real big ones (average is approximately 20 ltr. per kW output). Radiators emit heat through a combination of radiation and convection. Convector panels can be added to improve the heat emission through convection. Undoubtedly they are the most frequently applied type of heat emitters in the market. The following types of radiators can be distinguished (see also pictures further on in this paragraph):

- panel radiators
- panel convectors
- column radiators
- design radiator

9.4.2 *Heat output*

The output of the radiators is selected on the basis of heat loss calculations per room. Heat standardized output of radiators is measured under certified conditions according to test methods as described in EN 442. According to this method the heat output is measured with a supply temperature of 75°C, a return temperature of 65°C and an ambient temperature of 20°C.

In practice however radiators are not always operated at these test conditions and additional correction factors are necessary to calculate the heat output under these different conditions.

At least two important correction factors need to be incorporated when selecting the radiator for a specific situation:

φ_o	= correction factor compensating for different temperatures
φ_w	= correction factor compensating for different radiator arrangements

In the standardized position (Φ_w = 1), the radiator has a distance from the wall of at least 0,05 m and a distance to the floor of at least 0,11 m. and no further heat emitting obstructions. The use of window sills, front plates, grills, cupboards or whatever, will reduce the heat output of the radiator. This needs to be incorporated when the radiator is selected.

If other temperature regimes than the standardized regimes are applied, the heat output may change considerably. The correction factor Φ_o that compensates for this can be calculated with the following "radiator formula":

$$\varphi_o = T_n * (\theta_2 - \theta_3) \,/\, \left((\theta_3 - \theta_1)^{1-n} - (\theta_2 - \theta_1)^{1-n} \right) \qquad [-]$$

With

$$T_n = ((\theta_{3s} - \theta_{1s})^{1-n} - (\theta_{2s} - \theta_{1s})^{1-n}) \,/\, (\theta_{2s} - \theta_{3s}) \qquad [-]$$

In which

θ_{2s}	= standardized feed temperature (75°C)	[°C]
θ_{3s}	= standardized return temperature (65°C)	[°C]
θ_{1s}	= standardized ambient temperature (20°C)	[°C]
n	= radiator constant (to be taken from product documentation manufacturer)	
θ_2	= supply temperature	[°C]
θ_3	= return temperature	[°C]
θ_1	= ambient air temperature	[°C]

The exponent "*n*" generally has a value between 1,2 and 1,4 and corresponds with the share of heat emitted through convection and radiation. The higher the radiation share, the lower the exponent "n".

The radiation share "s" depends on the type of radiator. The following table is drawn from EN 442.

Table 9-3. Share of radiation in total heat output of radiators (according to EN 442)		
Type of radiator		**Share of radiation "s"**
Single plate panel radiator	P	0,5
Double plate panel radiator	PP	0,35
Triple plate panel radiator	PPP	0,2
Single panel convector	PC	0,35
Double panel convector	PCP or PCCP	0,20
Triple panel convector with single convector between plates	PCPP	0,15
Triple panel convector with 2 convector between plates	PCPC	0,10

To select the correct radiator on the basis of standardized output data the following formula is used:

$$\Phi_s = \Phi * \varphi_w / \varphi_o$$

In which:

Φ_s = corrected standardized output figure for radiator

Φ = calculated heat loss of the room

φ_o = correction factor compensating for different temperatures

φ_w = correction factor compensating for different radiator arrangements

Example

DATA

Calculated heat load room Φ : 1000 watts
Supply temperature θ_2 : 45°C
Return temperature θ_3 : 30°C
Ambient temperature θ_1 : 20°C
n-factor radiator : 1,3 (data provided by radiator manufacturer)
φ_w : 1 (standardized mounting of radiator)

CALCULATION

$\varphi_o = T_n * (15) / (10^{-0,3} - 25^{-0,3}) = T_n * 15 / 0,1204 = T_n * 124,5$
$Tn = (45^{-0,3} - 55^{-0,3}) / 10 = 0,00186$
$\varphi_o = 0,232$
$\Phi_s = 1000 * 1 / 0,232 = 4300$ [W]

Summarizing

If we want to heat a room that has a heat load of 1000 watts with a design temperature regime of 45/30°C, we need to select a radiator with a standardized output according to EN 442 of 4300 watts.

9.4.3 *Pros and Cons*

Radiators are available in all sizes and heat outputs and in numerous designs and colours. Standardized heat outputs may range from 200 watts to over 8 kW, depending on size, number of plates and convectors.

The advantages of radiators are:

- can meet practically all output requirements
- can be applied for high and low temperature heating systems, although for 40/25°C regimes the standardized heat-output need to be approximately 4 times bigger!
- can be added and altered afterwards (renovation, refurbishment)
- they facilitate a swift heating-up of the room
- can react fairly adequately on variations in heat demand (depends also on room temperature control system).

- they can help solving comfort problems due to cold air draughts

Drawbacks:

- they need free space against a (preferably outside) wall
- no furniture or curtains to be placed directly against or in front of the radiator (hinders heat emission)
- they are relatively big and heavy

9.4.4 Developments

Developments in radiators mainly focuses on new designs and on cost price reduction through changes in and relocations of the production facilities (are moving to eastern Europe and the far East). The little R&D work that is being done focuses on improving heat emission at lower temperature regimes.

An example of this type of development is the Therm X2 radiator from Kermi. Multi panel radiators usually are simultaneously fed with supply water; because of the heat supply is divided over all panels, leaving less heat for the panel that also responsible for the heat radiation, namely the front panel. Kermi changed the parallel hydraulic lay-out of its traditional radiators into a parallel hydraulic flow for the new Therm X2 radiators. As a result the first (and radiating) panel is supplied with heat, before the other panels are fed. This causes a higher average temperature of the front panel which increases the radiative output of the radiator with approximately 10%. The manufacturer indicates that heating up periods can be reduces and return temperatures as an average are lower, resulting in higher generation efficiencies.

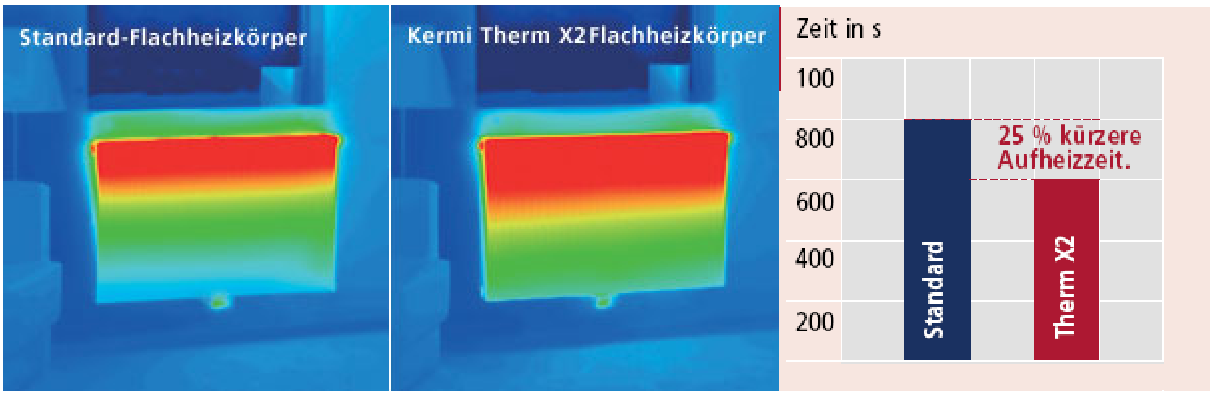


Figure 9-3. Kermi: Therm X2 - Energie sparen – Behaglichkeit gewinnen

9.4.5 Product examples

Figure 9-4.
Kermi Panel radiators

Figure 9-5.
Brugman Panel Convectors

Figure 9-6.
Myson Column Convectors

Figure 9-7.
Stelrad column radiators

Figure 9-8.
Vasco Design Radiators

9.5 Convectors

9.5.1 Introduction

Convectors differ from radiators because they are not composed of a number of water containing panels, but generally consist of a number of tubes to which thin plates or fins are connected. For this reason the share of radiation is minimal and convectors are predominantly optimised for heat transfer through convection. They heat the surrounding air with their fin coils, and the warm air circulates through the space by convection. Convectors typically consist of long horizontal heating elements that usually have sheet metal guards. The surface temperature of the coils is too low for strong radiation of heat, and the guards block most radiation that does occur. The heating coils may be enclosed at the bottom of a tall enclosure. In this case, the enclosure acts as a chimney, increasing heat output somewhat.

The water content is considerable smaller (one to maximal a couple of litres per convector) and because of this, a heating system with convector will react faster than a system with radiators.

The following type of convectors can be distinguished:

Natural convectors and *fanned* convectors in the following applications

- free standing convectors
- wall convectors
- skirting board convectors
- in-floor convectors
- kick-space convectors

9.5.2 Heat-output

For natural convectors, the way the requested heat output is calculated is similar to the way it is done for radiators, meaning that the radiator formula can be used, with which the correction factor φo can be calculated. If additional output related installation features are applicable (windowsill blocking the flow, distance from the floor, lowered into a sink in the floor, etc.) an additional correction factor φ_W needs to be used as well. The final heat output that needs to be selected can be determined with:

$$\Phi s = \Phi \bullet \varphi w / \varphi o$$ *(see paragraph 9.4.2)*

Convectors (natural and fanned types) can be used for high and low temperature heating systems, however the decrease in heat output is considerable (decrease of 80% if we change from 75/65°C to 45/30°C regime). In other words, at a 45/30°C temperature regime, the standardized heat ouput (EN 442) needs to be 5x bigger. Because heating elements themselves (finned tubes) are relatively small, this could be feasible in various situations.

9.5.3 Pros and cons

Convectors are available in all sizes and heat outputs and in numerous designs and colours. Standardized heat outputs may range from 200 watts to over 8 kW, depending on the number of finned tubes and their lengths.

The advantages of convectors are:

- can meet practically all output requirements
- can be applied for high and low temperature heating systems, although for 40/25°C T-regimes the standardized heat-output need to be approximately 10 times bigger!
- can be added and altered afterwards (renovation, refurbishment)
- they facilitate a swift heating-up of the room
- can react quickly on variations in heat demand (depends also on room temperature control system).
- they can help solving comfort problems due to cold air draughts
- they are relatively small and light-weight products

Drawbacks:

- they heat and move the air which is considered somewhat less comfortable that heat from radiation.
- they increase the number of dust particles in the air and are themselves dust-collectors; unfortunately they are difficult to clean because of their numerous fins.
- they need –although limited- some free space against a (preferably outside) wall.

- no furniture to be placed directly over or above a convector and no curtain can be placed before and over a convector.
- can overreact when feed water temperature is uncontrolled and room temperature control systems are insufficient.

9.5.4 Developments

Like for radiators, the developments in convectors mainly focuses on new designs and on cost price reduction through changes in and relocations of the production facilities (are moving to eastern Europe and the Far East). For natural convectors some R&D work is being done to improve the heat output at lower temperature regimes, either by varying the shape and size of the fins (thus improving the thermal-buoyancy-effect) or by adding some kind of forced convection with fans.

An example of this type of development is the “Dynamic Boost Effect” from JAGA.

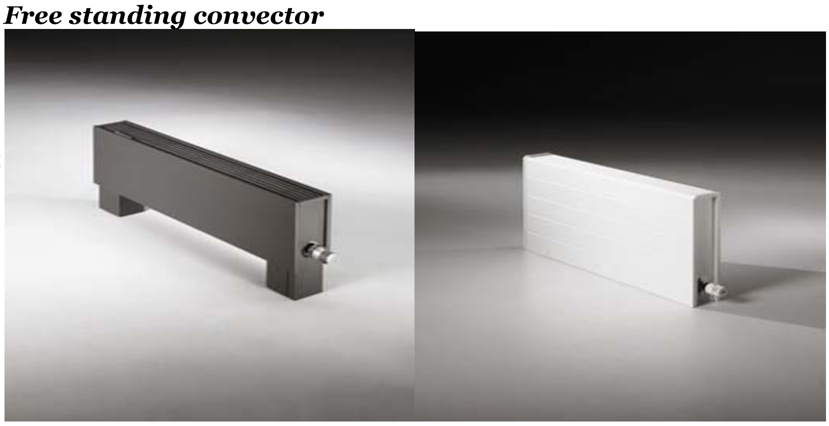
Figure 9-9.
Convector from JAGA with Dynamic Boost Effect (DBE). The DBE-systeem uses a 12V-24W power supply for fans and controls.

For fanned convectors, the R&D work focuses on improving controls and reduction of the power consumption for the fans.

9.5.5 Product examples

Free standing convector

Figure 9-10.
Free standing natural convectors from JAGA. Available in heat outputs ranging from 262 to 7746 watts; water contents ranging from 0,25 to 7,61 litres.



Wall convectors

Figure 9-11.
Fanned wall convector from Smith´s Environmental Products Ltd. Available in heat outputs ranging from 1350 to 8000 watts; water contents ranging from 0,3 to 1,04 litrs; power consumption for the fans ranging from 20 to 60 watts.

Figure 9-12.
Natural wall convector "Linea" from JAGA. Available in standardized heat outputs ranging from 284 to 5342 watts; water contents ranging from 0,3 to 3,5 litrs.

Figure 9-13.
Coanda Natural Convectors from Hurlcon (Australia). Available in standardized heat outputs ranging from 560 to 5680 watts.

Skirting board convectors

Skirting Board Convectors take the place of a traditional skirting board and deliver an even distribution of warmth throughout the whole room. Generally applied in homes and buildings with glas facades.

Figure 9-13.
Skirting board convector "Plint"from Acova. Available in standardized heat outputs ranging from 369 to 3000 watts per meter length. Water contents ranging from 1,2 to 13 litre per meter length.

In-floor convectors

These in-floor units are available for situations where panels are not able to be used. Such as when there is no wall space available or in front of large glassed areas. Also suitable to be fitted into concrete slabs. Available in a variety of lengths and widths to cater for various situations.

Figure 9-14.
Infloor convector

Kick space convectors

Kick space convectors will fit into small spaces which are normally wasted such as under stair wells, kitchen cupboards, bathroom cupboards, wall cavities, etc.

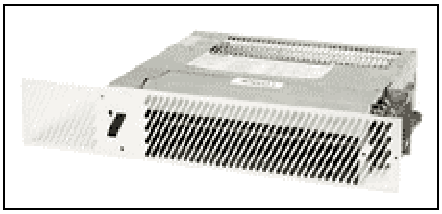


Figure 9-15.
Kick space convector

9.6 Surface heating

9.6.1 Introduction

The name **Surface Heating** is generally applied for systems that use radiation as the main principle for heat transfer. With surface heating systems relatively large building surfaces are heated, resulting in large radiating surfaces. Only a small part of the heat is transmitted through convection. Most common system used is the floor heating system, but wall heating and ceiling heating systems are also being applied.

Surface heating systems consists of a pattern of flexible tubes filled with warm water, that is invisibly applied in or under the floor or wall. Surface heating systems can be applied in two different ways:

- wet application in a cement screed or plaster
- dry application under a finishing floor or wall panels

Feed water temperatures of surface heating systems are relatively low (below 55°C) and are restricted by the maximum allowable floor temperatures. The following values for floor surface temperatures are applicable.

Table 9-4. Optimal and maximal average floor temperatures for various rooms (acc. ISSO manual).

Application	$\theta_{av.opt}$ [°C]	$\theta_{av.max}$ [°C]
Room with constant moving people	22	25
Room with people performing standing activities	23	27
Livingroom, study, office, school, church	25	29
Bathrooms and swimming pools	27	31
Floor partitions next to cold walls, windows and doors	29	34

The heat conductivity of the floor combined with the centre-distance between the heating tubes, determine the heat output of the floor per square meter and with it, the difference between (floor) surface- and ambient temperature.

A floor heating system with a constant surface temperature and uniform ambient conditions will have a heat output of roughly 10 [W/m²] per degree temperature difference between floor- and ambient temperature. Example: if the floor surface temperature is 3°C higher than the ambient temperature, the heat output of the floor surface will roughly be 30 W/m².

The difference between feed- and return temperature is generally kept at approximately 5 K. At lower temperature differences more water needs to be circulated (increasing the pump's power consumption); at higher temperature differences the differences in heat output can become to big, leading to comfort problems.

Table 9-5. Properties of wet and dry floor heating systems

	Wet system	Dry system
Thermal mass heating floor [kg/m²]	170 - 240	30 – 90
Heating-up period [h]	Several hours	< 2 hours
Max. heat outpout at tube spacing of 0,1 m [W/m²]	100	100
Max. heat outpout at tube spacing of 0,3 m [W/m²]	90	80
Difference between feed and return temperature [K]	5	5
Thickness of floor heating system [mm]	80 - 110	20 - 50

Table 9-6. Properties of wet and dry wall heating systems

	Wet system	Dry system
Thermal mass heating floor [kg/m²]	20 - 90	5 - 15
Heating-up period [h]	< 2	< 1 hours
Max. heat outpout at tube spacing of 0,1 m [W/m²]	150	140
Difference between feed and return temperature [K]	5	5
Thickness of floor heating system [mm]	10 - 40	2 - 5

The "wet" systems are applied in newly built dwellings and utility buildings. The "dry" systems can be used in new built dwellings, but also in renovation projects and in houses with wooden floor.

Temperature distribution

Rooms with a floor heating system produce other temperature distribution patterns than rooms heated with a radiator or convector system. Because of the convection principle the air temperature with radiator and convector systems is at its highest near the ceiling (23-24°C) and at its lowest near the floor (approximately 17°C). Floor heating systems demonstrate an opposite temperature distribution with floor temperatures of around 24℃ and ceiling temperatures of approximately 16 – 17°C.

A high vertical air temperature difference between head and ankles may cause discomfort (upward temperature increase). According to ISO Standard 7730, a temperature difference of 2°C between head and ankles causes a predicted percentage of dissatisfied people (PPD) of 2%. With a temperature difference of 4°C this percentage increases to 15%. People are less sensitive for decreasing temperature (cool head – warm feet).

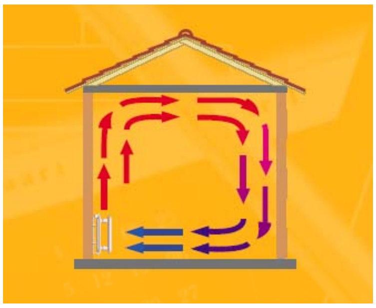


Figure 9-16.
Heat distribution wirh convection

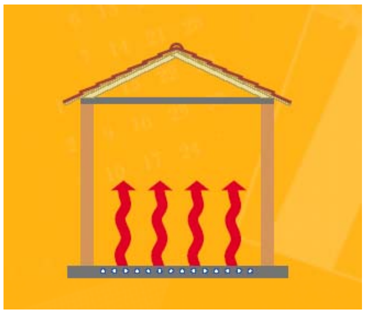


Figure 9-17.
Heat distribution with radiation systems

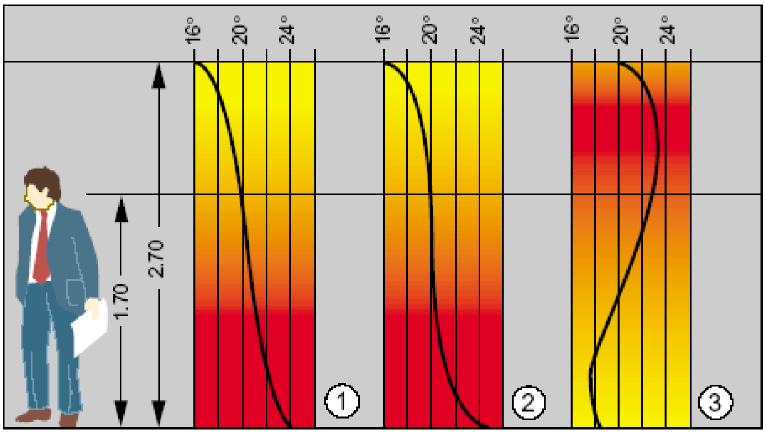


Figure 9-18.
Vertical temperature distribution: 1. Ideal situation 2. With Floor Heating and 3. With Radiator Heating (Source WTH).

Energy consumption

When floor heating systems are used, a lower average air temperature will suffice to achieve similar (or better) thermal comfort levels. As a result energy use can be lower with surface heating systems.

As explained in paragraph 9.2 there will be no losses due to non-uniform temperature distribution (increased transmission heat losses) provided that the surface heating systems are well insulated from the building construction parts that are in direct contact with the outside or with unheated rooms. According to EN 1264 correctly insulated surface heating systems have a heat transmission to the opposite (cold) side that equals 10% of the heat emission on the warm side.

Finally, because surface heating systems operate at low feed- and return temperatures, the generator (boiler) efficiency will be higher, especially for condensing boilers.

Application of surface heating

Surface heating systems can be applied in practically all houses and buildings, either as main source of heating or as a supplementary heating system. Surface heating can be the main source of heating when insulation- and airtightness levels are high enough and no cold draught or radiation occurs. If this is not the case, the application of surface heating can best be limited to supplementary heating, combined with radiators or convectors near the windows and outside walls. If surface heating is used as a supplementary system, its main purpose is to ensure a comfortable floor temperature and the system is designed to complement the radiator or convector based heating system.

Control of surface heating systems

Depending on the feed temperatures coming from the generator, the hydraulic system from the generator is connected either with or without a mixing valve to the hydraulic system of the surface heating. If feed temperatures higher than 55°C may occur, a mixing valve is necessary.

A disadvantage of surface heating systems is their slow response time and long warming up periods. The warming-up period primarily is related to the amount of thermally active building mass. In cases where the actual floor and wall heating systems are well insulated from the rest of the building mass, and the heating package itself is relatively light, the warming up periods can be reduced to less than one hour. Also, for high-insulated buildings the energy gain from a night setback is small, due to the fairly limited reduction at night of the building structure temperature. Therefore, surface heating systems preferably use minor night setback temperatures (2 -3°C) and warming up becomes less of an issue.

Another control aspect concerns the ineratia of surface heating systems with incoming solar heat or sudden changes is internal heat loads, resulting in unwanted temperature changes. This effect in practice however is less spectacular than presumed, due to the so called "self-regulating" effect. Due to the small differences between the surface temperatures and inside air (only a couple of degrees), a sudden increase in heat load (e.g. solar input in the room) instantly results in a considerable reduction of the ΔT. Because the ΔT over the surface is the main driving force for the heat emission, a declined ΔT immediately decreases the amount of heat that is emitted in that specific room. In other words, the amount of heat that is emitted is immediately self-regulated by the ΔT over the heating surface.

9.6.2 Heat output

To calculate and optimize the heat-output of a floor heating system, the following parameters are used:

R_b : thermal resistance of the surface heating floor [(m²K)/W]

d_c : tube spacing

$\Delta\theta_h$: Mean Effective Temperature Difference (M.E.D.)
M.E.D. = Flow + Return water temp. / 2 - Room Temp.

First determine what the heat load (under design condition) of the room in question is. This results in a specific heat output figure per square meter, e.g. 50 W/m². With the aid of a calculation diagram (not incorporated in this report) and the given parameters R_b and d_c the Mean Effective Temperature Difference $\Delta\theta_h$ can be selected, with wich the requested heat output can be achieved.

Example: With an R_b of 0,01 (floor tiles) and a d_c of 20 cm, a $\Delta\theta_h$ of 12°C would be necessary for a heat output under design conditions of 50 W/m². The design feed temperature would than have to be around 35°C.

Depending on the three parameters mentioned, the heat output of surface heating systems may vary from 0 to 100 watts/m² for floor heating systems and from 0 – 150 watts/m² for wall heating systems.

9.6.3 Pros and Cons

Apart from the advantages that already mentioned in the previous part of this paragraph, such as:

- a more comfortable thermal environment
- the ability the improve emitter efficiency and generator efficiency

there are a couple of other positive aspects about this emitter system that are worth mentioning:

- cleaner indoor air due to reduced number of dust particles
- higher relative humidity in winter periods due to reduced air temperatures
- no space consuming heat emitters; fully free to position interior componets
- no maintenance

Because there is no or minimal convective air movement, no dust is transported nor collected (e.g. in convector fins or radiator convectors) and the maintenance activities related to this, can be avoided. The reduction of floating dust particals in the indoor air is especially beneficient for people with allergies and pulmonary diseases.

Drawbacks:

- Surface heating systems are more sensitive to cold draughts (ventilation grids and unsufficient air-tightness of outside walls can already cause comfort problems)
- Depending on type of system, warming-up and response times can be long
- Not always possible in existing building/houses
- Random drilling or nailing in floors or walls is no longer possible.

9.6.4 Developments

Developments mainly focus on

- improvement of installation activities (systems that are quick and easy to apply and install)
- design and engineering of dry systems for floor and wall heating that can easy be applied in existing houses
- improvement of temperature control

9.6.5 Product examples

Wet floor heating systems

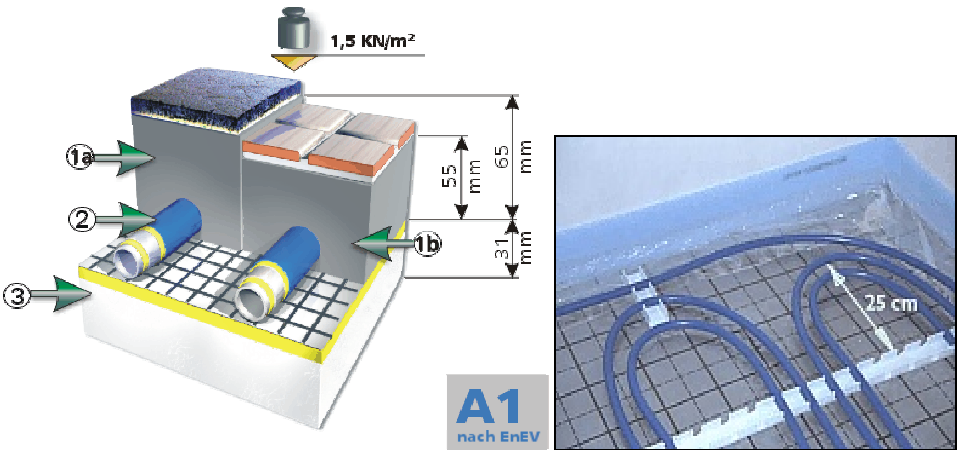


Figure 9-19. Standard wet floor heating system for construction floors between heated zones (package to be added to the construction floor):
1a : 65 mm cement screed
1b : 55 mm concrete mortar
2 : heating tube
3 : Insulation 31 mm (5 mm PUR plus 26/28 mm PST)
Thermal resistance insulation package R [m²K/W] : 0,75
Thermal conductivity [W/m²K] : 1,09

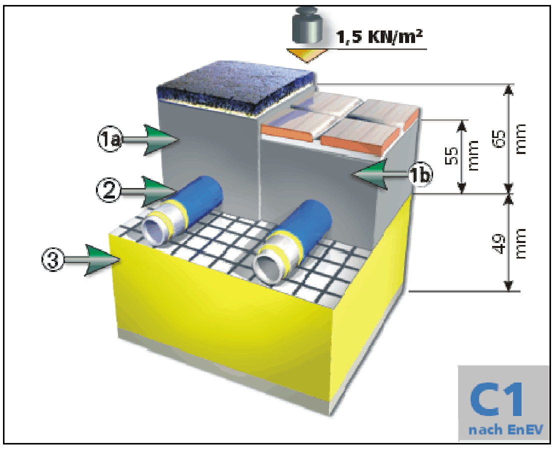


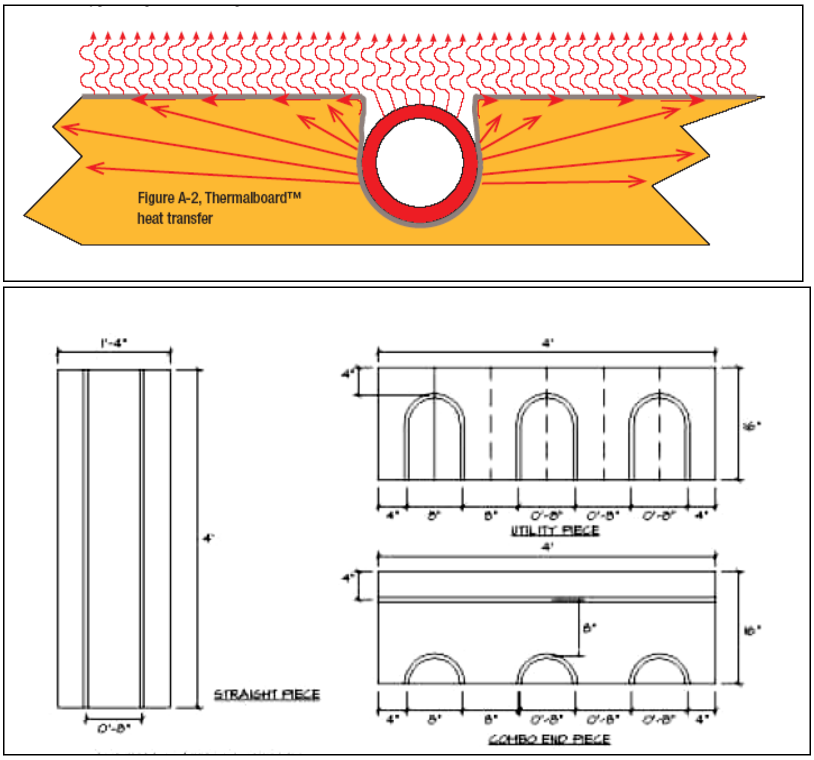
Figure 9-20. Standard wet floor heating system for construction floors that are in contact with the outside or unheated zones (package to be added to the construction floor):
1a : 65 mm cement screed
1b : 55 mm concrete mortar
2 : heating tube
3 : Insulation 49 mm (44 mm PUR plus 5 mm PE noise damping material)
Thermal resistance insulation package R [m²K/W] : 2,0
Thermal conductivity [W/m²K] : 0,46

Dry floor heating systems

Figure 9-21. Thermalboard™ from "Warmzone Radiant Heating Ltd" is a thin (16 mm) dry floor heating system that provides an attractive alternative to wet concrete systems:

- Good response time to heat up/cool down
- Easy layout and installation
- Lightweight – 5 times lighter than concrete
- Even distribution of heat.

Thermalboard™ comes in 3 different board configurations. These are "straight", "utility" and "combo end piece". They are assembled to make a channel for the pipe. Thermalboards cut easily with a circular saw.



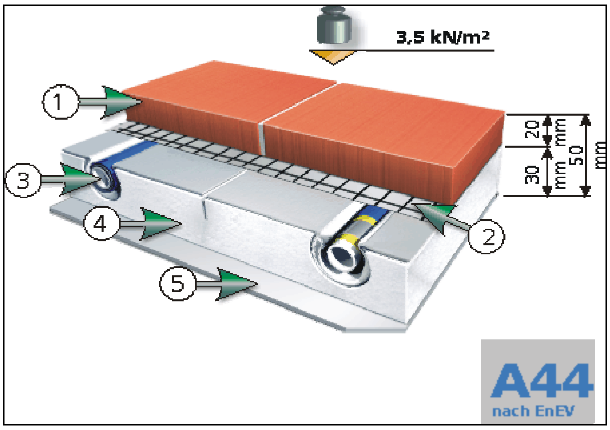


Figure 9-22.
Dry floor heating system for construction floors between heated zones (to be added to construction floor):
1 : 20 mm CREATON Estrich tiles
2 : Film
3 : Heating tube
4 : System element IDEAL 30 mm PS30
5 : Damp proof film
Thermal resistance system element R [m²K/W]: 0,86
Thermal conductivity [W/m²K]: 0,97

Wet wall heating systems

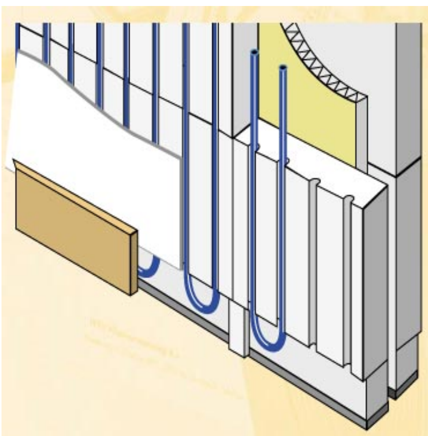


Figure 9-23. Wet wall heating system with prefab lime-sandstone bricks.
Heating tubes are placed in the grooves and covered with plaster; insulation is applied in the cavity.

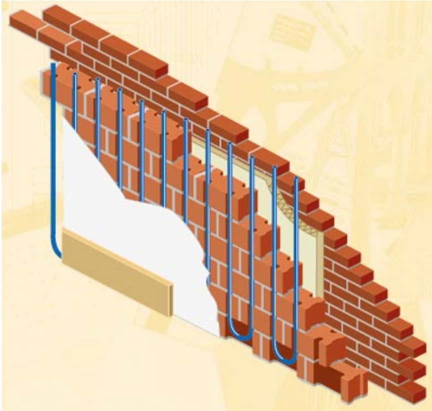
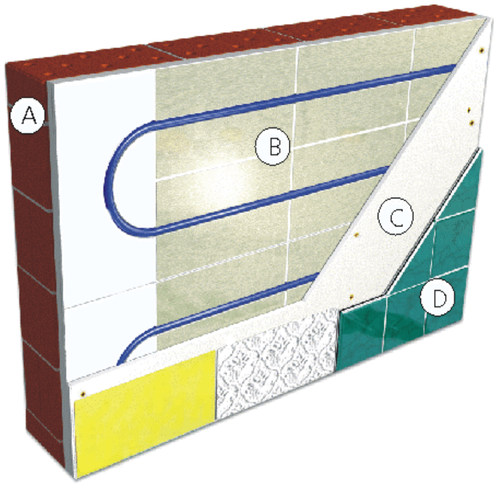


Figure 9-24.
Wet wall heating system with prefab ceramic bricks
Heating tubes are placed in the grooves and covered with plaster; insulation is applied in the cavity.

Dry wall heating systems

Figure 9-25.
Jupiter Wall heating systems
A : Wall
B : Thermal insulation
C : Dry cover plate (plasterboard) 12,5mm
D : Wall covering



Dry ceiling heating system

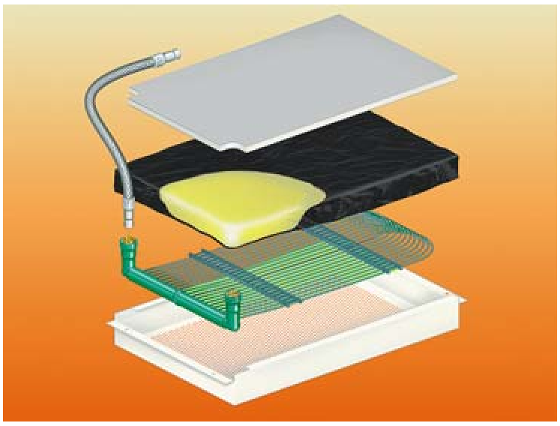


Figure 9-26.
Karo-systems. Ceiling panels suited for heating and cooling.
Cailing panels with a sandwich construction consisting of perforated steel plate/casing with a polypropylene capillary ducting net, a noise damping foam plate and an end plate.
The capillary heating net facilitates an even temperature distribution because of its minimal tube spacing. Typical heat output: 85 W/m²

9.7 Summary

Emitters play an important role in the overall efficiency of a heating system. Emitters that use convection as their main heat transfer principle (e.g. convectors and – to a lesser extend- radiators) will have additional losses due to non uniform temperature distribution in the room. This causes additional transmission losses due to higher air temperatures near ceilings, windows and outside walls (up to 16% additional losses).

Provided they are well insulated from the building construction, surface heating systems will have higher emitter efficiencies, because these systems are not burdened with the additional transmission losses due to locally higher air temperatures. Important precondition however is, that these surface heating systems are applied in very well insulated and air-tight dwellings and buildings. If this is not the case, comfort problems will occur.

In addition to their influence on the heat transmission efficiency, heat emitters also influence the generator efficiency. The type and size of emitter that is selected determines the feed temperature that is necessary to achive the requested heat output. The bigger the emitter, the bigger their standardized heat output (acc. EN 442), the lower feed temperatures can be at which the requested heat output is delivered. In other words, oversizing heat emitters is a key approach in facilitating lower feed temperatures. As explained in previous chapters, the generator efficiency for condensing boilers improves with 2 to 2,5% for each 10°C decrease in return temperature. A emitter system with a design temperature of 45°C instead of 85°C will improve the generator efficiency with 8 – 10%.

Together these two efficiency related emitter parameter may increase the energy consumption of the generator with values up to 27%

10 HEAT PUMP BOILERS

10.1 Introduction

"Heat Pump" is a collective term for a wide range of products using a similar technical principle for pumping heat from one temperature level to another temperature level. Heat pumps have been on the market for over one and a half century[39] and are widely applied in air conditioning and refrigeration appliances. But also for space heating and water heating purposes heat pumps are considered a favourable technology with a relatively long history. After all, heat pump technology facilitates the transfer of low temperature heat - that in principle is inexhaustible and available everywhere around us (air, ground, water) - to higher temperature levels that can be used for space heating and hot water.

In Europe, a sustainable market has only been established in some countries like Sweden, Norway, Germany and Switzerland.

This chapter describes the various types of heat pump boilers available on the market, their application and installation properties, their seasonal efficiencies and environmental impacts. Finally it discusses the technical development that can be expected within the next couple of years.

10.2 Typology

Heat flows naturally from a higher to a lower temperature. Heat Pumps however, are able to force the heat flow in the other direction, using a relatively small amount of high quality drive energy (electricity, fuel, or waste heat). The key feature of Heat Pumps lies in the fact that the amount of energy that is required to operate the heat pump process is smaller than the amount of thermal energy that is delivered by the heat pump. This relates to primary energy of course A current field test [40] shows that electric Outside-Air HPs not using low temperature heating achieve a SPF of 2,3 which is –in primary energy at a factor 2,5--below the performance of some conventional boiler technology in terms of primary energy. Because of this heat pumps are considered an important technology for the reduction of emissions and energy consumption.

Irrespective of their type, heat pumps can be described as appliances that vaporize a process medium that as a result absorbs heat from the direct environment. In a next step the process medium is compressed using auxiliary energy, and raised to a higher temperature level. Residential Heat Pump Boilers acquire their heat from the environment (e.g. ground, water, air) at relatively low temperature levels, and raise the heat to higher temperature levels, suitable for space heating.

There are several ways to energize this thermodynamic heat pump cycle, but most common are the *vapour compression cycle* and the *absorption cycle*.

Theoretically heat pumping can be achieved by several thermodynamic cycles and processes, including Stirling and Vuilleumier cycles, single-phase cycles (e.g. with air, CO_2 or noble gases), solid-vapour sorption systems, hybrid systems (notably combining

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[39] The first known heat pump applied for heating purposes was designed and installed in 1855 by an Austrian engineer named Peter Ritter von Rittinger, but for several reasons the *heat pump boiler* didn't really reached world wide public recognition.

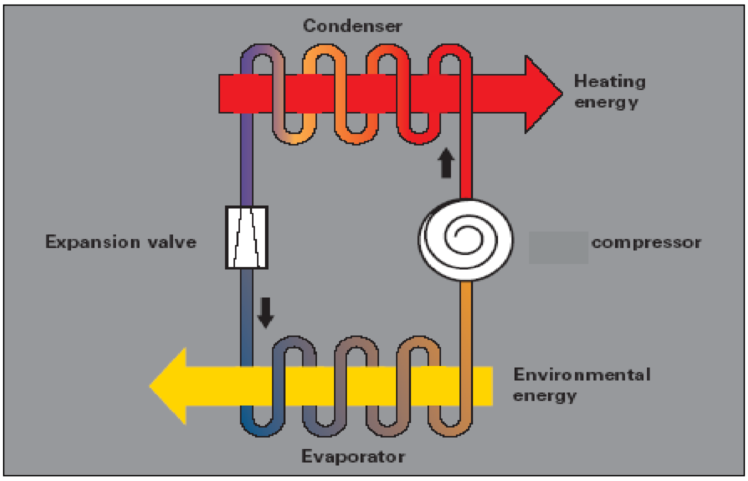
[40] https://www.badenova.de/web/media/dokumente/unternehmensbereiche/pro/innovationsfonds/zwischenberichte/2006-08-02.pdf, available in German only.

the vapour compression and absorption cycle), electromagnetic and acoustic processes. Some of these could become significant in the future. [41]

10.2.1 Vapour Compression Cycle

In a heat pump using the vapour compression cycle, the main components are the compressor, the expansion valve and two heat exchangers, referred to as evaporator and condensor. A graphic representation of this heat pump principle is given in Figure 10-1. The components are connected to form a closed circuit and a volatile liquid – known as the working fluid or refrigerant – circulates through the four components.

Figure 10-1.
Heat Pump principle with vapour compression cycle .
(Illustration by Viessmann).



In the evaporator the temperature of the liquid working fluid is lower than the temperature of the heat source (ground, water, air), causing heat to flow from the heat source to the liquid which enables the working fluid to evaporate. The vapour form the evaporator is compressed to a higher pressure and temperature. The hot vapour then enters the condenser, where it condenses and gives off useful heat. Finally, the high pressure working fluid is expanded in the expansion valve to the evaporator pressure and temperature. The working fluid is returned to its original state and once again enters the evaporator. The compressor can be driven by an electric motor or a combustion engine.

10.2.2 Absorption cycle°

Absorption heat pumps are thermally driven, which means that heat rather than mechanical energy is supplied to drive the cycle. Absorption heat pumps for space heating (or cooling) are often gas-fired, while industrial installations are usually driven by high-pressure steam or waste heat.

Absorption systems utilise the ability of liquids or salts to absorb the vapour of the working fluid. The most common working pairs for absorption systems are:

- Water (working fluid) and lithium bromide (absorbent)
- Ammonia (working fluid) and water (absorbent)

In absorption systems, compression of the working fluid is achieved thermally in a solution circuit which consists of an absorber, a solution pump, a generator and expansion valve as shown in figure 1-2.

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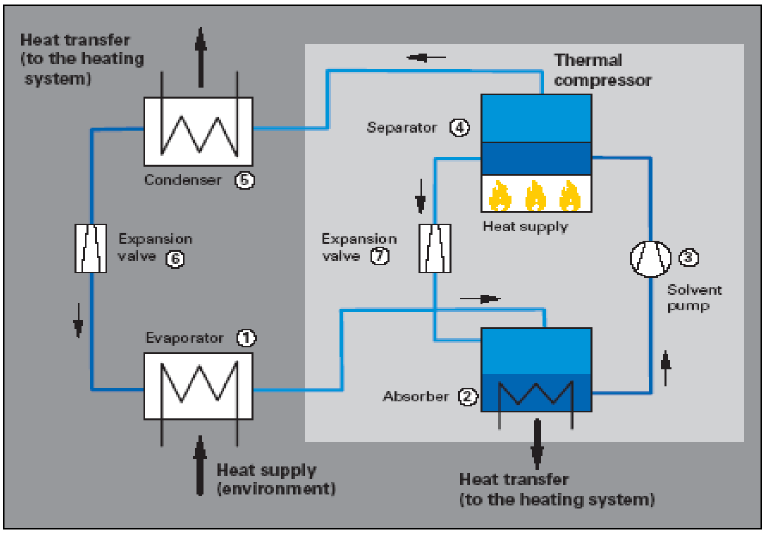
[41] Source: The IEA Heat Pump Programme's Information Centre / www.heatpumpcentre.org

Low-pressure vapour from the evaporator is absorbed in the absorbent. This process generates heat. The solution is pumped to high pressure and then enters the thermal compressor, where the working fluid is boiled off with an external heat supply at high temperature. The working fluid (vapour) is condensed in the condenser while the absorbent is returned to the absorber via the expansion valve.

Heat is extracted from the heat source in the evaporator. Useful heat is given off at medium temperature in the absorber and in the condenser. In the generator high-temperature heat is supplied to run the process. A small amount of electricity may be needed to operate the solution pump, or when a bubble pump is used, no electricity is needed at all.

Figure 10-2.
Principle of absorption cycle.
(Illustration by Viessmann)



10.2.3 Heat Sources and Heat Sinks

Several sources can be used from which low temperature heat can be extracted. These sources are:

- Air – Ambient air or exhaust air (e.g. from mechanical ventilation systems)
- Ground – Use of soil (horizontal collector) or ground (vertical probe collector)
- Water – Ground water or surface water
- Waste heat – subject to availability, volume and temperature level of waste heat
- Solar collector – use of solar collector as source for the heat pump
- Or combinations of these sources.

The Heat Sink is the system to which the high temperature heat from the heat pump system is transferred to. Per definition the Heat Pump Boiler transfers its heat to the hydronic central heating system and depending on the type of heating system the temperature regime will vary. Similar to gas boilers, heat pump boilers can also be used for sanitary hot water and combinations of both (combis). The following categorisation of heat sinks are given:

- Low temperature space heating systems (feed temperature < 35°C); surface heating (floor and wall heating) systems.
- Medium temperature space heating systems (< feed temperature < 55 oC); oversized radiators and/or convectors).
- High temperature space heating systems (traditionally sized radiators and/or convectors).

Or combinations of sinks, which for instance could be the case with heat pump combis.

As a general rule that universally applies to all heat pumps one can say that "the lower the temperature difference heating water and energy source, the higher the coefficient of performance and consequently the higher the efficiency. In other words, strive for low temperature space heating systems and high temperature heat pump sources.

10.2.4 Technical Product Configurations

For space heating purposes, the heat pump technology can be applied in different ways. The distinctive parameters for heat pump configurations are: heat source, thermodynamic principle, heat sink, the use of additional energy sources, use of buffer and finally the products functionality.

Table 10-1. Matrix, illustrating the configurational options for heat pump application in heat pump boiler systems (compiled by VHK)

<table>
<tr><th rowspan="2">Heat source</th><th rowspan="2">Thermodynamic Principle</th><th rowspan="2">Temperature Level Heat Sink</th><th colspan="2">Additional Energy Source</th><th rowspan="2">Use of Buffer</th><th rowspan="2">Functionality</th></tr>
<tr><th>Source</th><th>Type of use</th></tr>
<tr><td>Ambient Air</td><td rowspan="3">Vapour compression</td><td rowspan="3">< 35°C</td><td rowspan="2">No additional source</td><td rowspan="4">Parallel</td><td rowspan="4">No buffer</td><td rowspan="2">Space heating only</td></tr>
<tr><td>Exhaust Air</td></tr>
<tr><td>Ground water</td><td rowspan="3">Add. electric element</td><td rowspan="3">Space heating and Cooling</td></tr>
<tr><td>Surface water</td><td rowspan="3">Absorption</td><td rowspan="3">< 55°C</td></tr>
<tr><td>Soil (horiz)</td><td rowspan="5">Alternating</td><td rowspan="5">With buffer</td></tr>
<tr><td>Ground (vert.)</td><td rowspan="2">Gas heater</td><td rowspan="2">Space heating & san. hot water</td></tr>
<tr><td>Solar collector</td><td rowspan="3">Other</td><td rowspan="3">< 80°C</td></tr>
<tr><td>Waste heat</td><td rowspan="2">Oil heater</td><td rowspan="2">Sanitary hot water only</td></tr>
<tr><td>Combinations</td></tr>
</table>

A large number of combinations can be made and each market segment will have its own characteristics that will lead to market specific configurations, designed for optimal technical and economic performance for each specific segment.

Only a couple of specific heat pump configurations have been developed so far and they are mainly focussed on new built market segments. It is expected that new developments will lead to other product configurations that have better application possibilities in the existing markets.

10.2.5 Current Heat Pump Products

Most common applied heat pumps for space heating (with hydronic central heating system) and for DHW in Europe are the following products:[42]

- Ground source closed loop brine/water heat pump
- Exhaust air/water heat pump

Because installation costs are considerably lower and performance of the latest appliances is increasing, the air-source heat pump is expected to gain market share.

- Ambient air/water heat pump

For the more southern member states where space cooling in the summer season is more important than heating, there is a growing interest for reversible ambient air heat pumps.

- ambient air/air heat pump

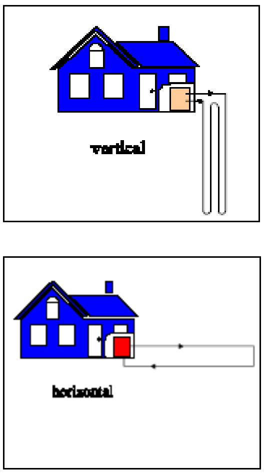
<u>Ground source closed loop brine/water heat pump</u>

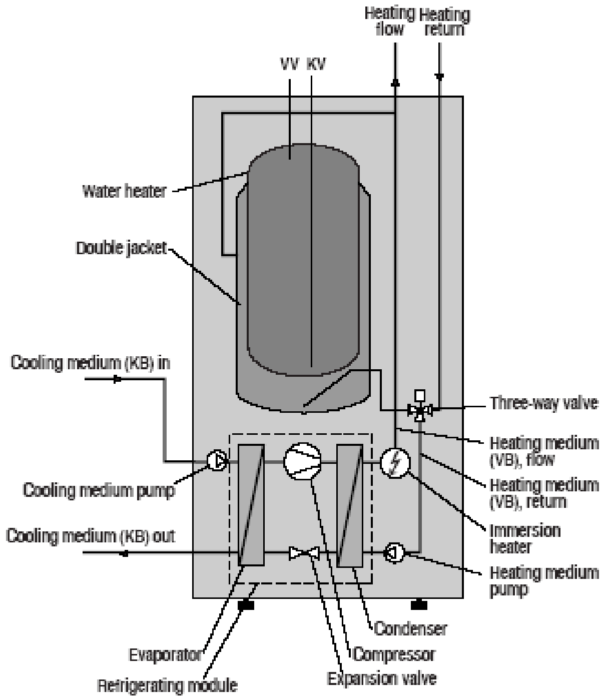
[42] European Heat Pump Association, Heat Pump Statistics 2005.

The most common type of ground-source heat pump boiler today is the vapour compression heat pump with a heat source that extracts heat from the ground (vertical source) or borehole (ground water). The temperature levels of the heating system (heat sink) is either 55/45°C for existing houses or 35/28°C for new houses with surface heating (floor heating) systems. The typical heat pump boiler is designed to cover 50 to 60% of the maximal required power for space heating purposes and the remaining part is covered by a supplementary heat source, which typically is the electric heater (cheapest and simplest solution), but gas and oil burners/boilers are also used. The supplementary heat source (electric or fuel) is used in parallel mode, meaning that thermal heat is added when higher feed temperatures (than can be supplied by the heat pump) are necessary. Because most heat pumps predominantly are on/off controlled systems, they generally use a buffer to facilitate a continuous operation of the heat pump. Ground source heat pump boilers are either used for space heating purposes only, or for both space heating and sanitary hot water heating.

The installed costs for a typical 8 kW system would vary between 10.000 euro and 16.000 euro (consumer prices, VAT included, heat distribution system excluded). [43] A product like this can cover 80 to 95% off the annual energy consumption for space heating.

Figure 10-3. Typical configuration of a ground source heatpump with a double wall storage tank (outer tank for heating, inner tank for sanitairy hot water). (Illustration by Nibe)





Exhaust air/water heat pump

The most common type of air-source heat pump boiler today is the vapour compression heat pump with a heat source that extracts heat from the ventilation exhaust air. Obviously this system needs te be combined with a mechanical ventilation extraction system. The capacity of an exhaust-air heat pump is limited by the exhaust air flow and can therefore **not** be sized to cover 50 to 60% of the maximal required power for heating the house. A supplementary heat source must always be available and in most cases this is an electric heating element, used in parallel mode.

The exhaust air/water heat pump is either used as a water heater or as a combi and is built around a large storage vessel.

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[43] Factsheet on Ground Source Heat Pumps from the Energy Saving Trust, 21 Dartmouth Street, London, December 2005.

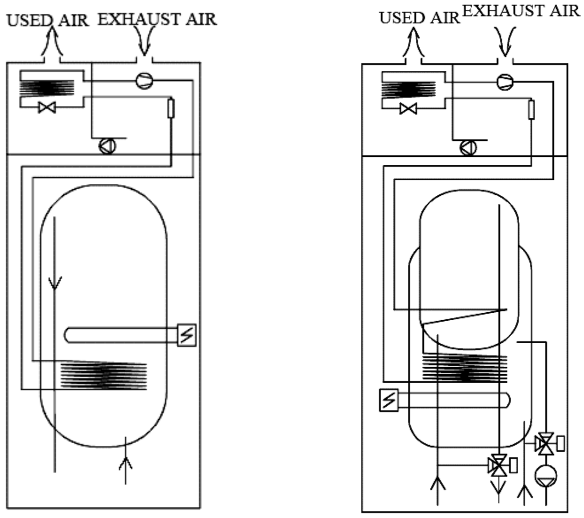

Figure 10-4.
Typical configuration of an exhaust air heat pump for hot water only (left) and for hot water and space heating (right).

Average costs for a completely installed exhaust air heat pump with a 225 litre storage tank (sanitary hot water only and excluding costs for the ventilation ducting system in the house) varies from 2000 to 3500 euro for a heat pump water heater. For a combi-appliance the total costs could rise to around 6000 Euro (consumer prices, VAT included).[44] A product like this can only cover a smaller part of the heat necessary for space heating and/or DHW. If we assume a continuous exhaust average airflow of 150 m^3/h and an average temperature difference of 10 0C, the heat input from the exhaust air is 420 watts. Assuming that 50% of the 385 [W] electrical input for an average heat pump compressor is also transferred to heat, we have a total heat input of around 600 watts. For a whole heating season (5000 hours) the heat production would amount to approximately 3000 kWh, and 5200 kWh for a whole year.

The exhaust airflow rate of a 150 m^3/h corresponds to the average mechanical ventilation rate that is more than sufficient to keep indoor air quality below 1200 ppm CO_2 in average 87 m^2 dwelling (0,7 ACH[45]). Exhaust air/water heat pumps can deliver more output when the ventilation rate is increased. However, increasing the ventilation rate would also increase the amount cold air coming in to the house and with it the energy consumption for space heating. What we gain for the air source heat pump, we lose in the heat load of the house.

Sizing of an exhaust-air HP depends on the heat load of the building. It is possible to cover almost 100 % of required power when heat energy demand is consequently reduced (as in low-energy house).

<u>Ambient air/water heat pump</u>

Heat pumps that use the outside air as a heat source are considerably less expensive and easier to install than ground source systems. Main drawbacks of air-source heat pumps however are the lower COPs that are achieved which increases the need for additional back-up power (during heating season the COP is the lowest because outside temperatures are low) and the noise related to the high airflow that is required (1500 to 4000 m^3/h). Further more air-source heat pumps generally need a defrost cycle. If the outdoor temperature falls to near or below freezing, the moisture in the air passing over the outside coil will condense and freeze on to it. This frost built-up decreases the efficiency of the heat exchanger and at some point the frost must be removed with a defrost cycle. This defrosting further reduces the seasonal performance of the air-source heat pump.

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[44] Based on NIBE pricelist, april 1st 2006 (list prices are product only prices, excluding VAT and excluding installation costs)

[45] ACH = Air Changes per Hour

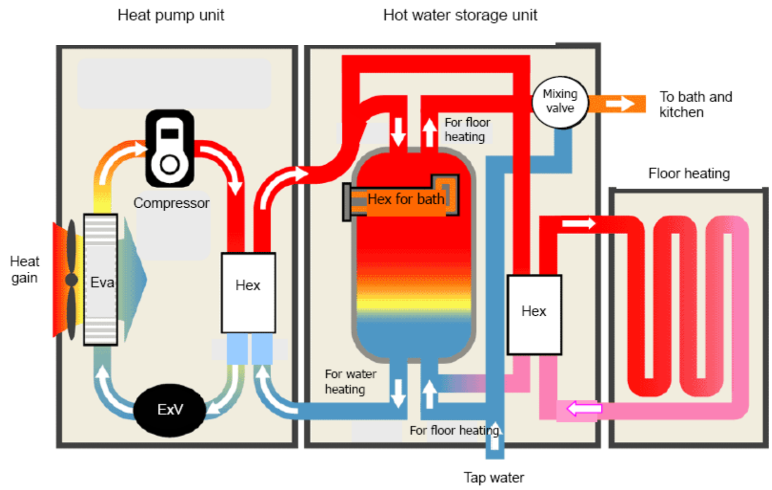

Figure 10-5.
Air-source heat pump for combined hot water heating and floor heating (Toshiba Electric Applainces Co.)

Ambient air-source to water heat pumps are used for space heating only (in which case a storage tank is not indispensable) or for combinations of space heating and domestic hot water, in which case the storage tank is a prerequisite. Prices for completely installed heat pump systems (heat distribution system excluded) vary from 3000 Euro (space heating only, no storage tank) to maximal 10.000 euro for combi systems with storage tank.

Figure 10-..
Ambient air-source heat pump from IVT, connected to storage tank inside the house, Sweden (IVT was acquired by BBT Thermotechnik in November 2004)

Reversible Ambient air/air heat pump

The ambient air to air heat pumps are the most applied heat pump products on the market, because they are the key product in air conditioning systems (cooling and de-humidification). In that sense they are an optimised and mass produced product. Regions with buildings that predominantly need space cooling, and only a little bit of space heating are perfectly served with a reversible air to air heat pump that performs both the cooling and heating function. And although COPs in the heating mode drop due to decreasing outside temperatures and defrost cycles, these air to air heat pumps gain also market share in middle and northern Europe, undoubtedly due to their lower first time costs (2000 to 3000 Euro for an installed air-tot-air heat pump (compact unit) in 2004, costs for heat distribution system excluded[46]).

[46] Source: Rad & Rön, no. 10 December 2004 (in Swedish) and SP Swedish National Testing and Research Institute.

The market for air-to-air reversible heat pumps has increased rapidly in e.g. Norway and Sweden over the last couple of years. As a response, the Swedish Consumer Agency and the Norwegian magazine “Forbrugerrapporten” ordered a test on eight air-to-air heat pumps which were available on both markets. The performance of the heat pumps in heating mode was tested in accordance with standards EN 14511 and CEN/TS 14825.

At – 15°C (outside air) the COP for the heat pumps ranged from 1,4 to 2,3. However, the tests also showed that there were quite considerable variations in performance between different brands. It also showed the importance of testing the heat pumps at more than one operating point. Tuning of variable speed control and defrost strategy are of the utmost importance since they can make or break the performance of heat pumps in real life and at lower temperatures [7].

Note: The air-to-air heat pump will not further be discussed in this chapter, because – being an air based heating system - it does not fall within the scope of the Ecodesign Study.

10.3 System Design and Application

Heat pump boilers can not be selected and installed like oil or gas boilers because they require a system design approach that goes beyond the design practice of the traditional heating systems. Correct system design and a high quality realization and installation are important constraints for achieving the required seasonal performance.

Het pump boiler systems today are primarily used in countries or regions where fuels (gas, oil) are not abundantly available and by people that are willing to invest considerably more than for condensing boiler systems. As stated before, installed costs for an 8 kW ground-source system vary between 10.000 and 16.000 euro (VAT included, heat distribution system excluded), compared to two-and-a-half to three thousand euro for an installed 24 kW condensing combi boiler.

Key driver for the application of heat pump systems are their low energy consumption and operating costs and the related low CO_2 emissions.

This paragraph elaborates on system design aspects and the application of heat pump boilers in market segments like new-built, renovation and retrofit. The last sub-paragraph illustrates the market development in three key HP-boiler countries: Sweden, Germany and Switzerland.

10.3.1 System design and installation: critical factors

Designing and installing a (ground-source) heat pump boiler system is a complex process that requires specific knowledge of heat pump technology and their applications, knowledge and skills related to installation critical factor and not in the least the knowledge and skills to perform a situation specific assessment. There are various parameters that need to be evaluated in order to design and install a heat pump boiler system with the right technical and economic features; to name a few:

- The Load Side System (=Heating System). The heating system needs to be designed and optimised for low temperature heating (preferably LT surface heating, with design temperature regime of 35/28°C). For existing houses the options for a reduced temperature regime (55/45°C) need to be evaluated (asses - and if feasible adjust - heat emitters and thermal insulation of the building). The hydraulic design and the control system design of the heating system are critical and need to be optimised for heat pump systems.
- The Source Side System. The source system (ground-source-heat exchanger) needs to be selected (type of heat exchanger) and dimensioned in the correct way. Successful application of the heat pump to a large extend hinges on the competent design and installation of this component. Contrary to popular belief, the efficiency of a ground source heat exchanger is not identical to the nominal COP-rating of the selected heat pump equipment; rather it depends on the quality of the design and installation of the ground source. If the ground source is too big, the heat pump

switches on and off and reduces its seasonal performance. On the other hand, if it is too small the output will either not be enough to comfortably heat the building, or the back-up system will be operative for too long a period which also reduces the seasonal performance. A competent ground source system design also considers thermal storage effects of the ground and its thermal response to long-term heat extraction. Adequately sized ground-source heat exchangers ensure that source side temperatures stay within the acceptable operating range.

- The Heat Pump Boiler System. The heat generator part of the system also needs to be designed and optimised from a technical and economic perspective. The parameters are: availability and costs for alternative heat sources (fuels, electric power), the requested heat output and design temperatures of the heating system, the choice for monovalent or bivalent systems and the related balance point, the functional options (space heating only or combi), the use of buffers and finally the control systems.

Clearly the design and the economic and technical optimisation of a ground source heat pump boiler system is a difficult task and can not be performed by the average heating installer. Further market penetration of heat pump boiler systems would definitely require extensive vocational training and additional support from e.g. national experts.

In this context, the Annex 32 of the IEA Heat Pump Centra is investigating different configurations of integrated multifunctional heat pump systems for space heating, DHW[47] production, space cooling, ventilation for the application in low and ultra-low energy houses. The objective is to find the best system configurations to minimise the overall energy use for the different building needs under the boundary conditions of thermal comfort and costs. The activities started on the 1st of January 2006. End date: 31 December 2008. The objective is to evaluate, assess and enhance integrated heat pumps systems for low energy houses with regard to overall energy use and costs under the boundary conditions of thermal comfort. The final goal (Task 4 of the annex) is to draw-up Design Guidelines for integrated heat pump systems for low energy houses including its controls.

10.3.2 Application in new-built, renovation and retrofit

New-built

Since heat pump boiler systems preferably require a dedicated low temperature heat distribution system and minimal heat loads, the new built market is the evident segment for these kind of systems. Low temperature floor- and wall heating systems can be applied, and because of the optimised insulation and air tightness of the building the heating system only requires a couple of kW on the real cold days. The new built segment is therefore probably one of the few market segment for which a monovalent heat pump boiler systems can be considered (i.e. without auxiliary energy source).

Apart from the load side of the system, the new-built segment is also advantageous for the installation of the ground source heat exchanger. The necessary ground works and drilling of boreholes can better be performed during building stage of the house than afterwards. If the property around the new-to-built house is big enough, a horizontal ground source heat exchanger could be applied. Each square meter of garden could deliver as an average 10 – 35 watts, which would imply that a maximal heat load of 6 kW would need 200 - 600 m² of soil. Since most dwellings do not have this surface available, vertical ground-source heat exchangers are often the only solution.

If more houses are being built simultaneously, a collective ground-source heat pump system will decrease the average first time investment costs per dwelling.

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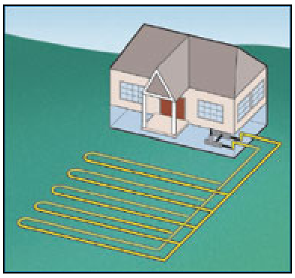
[47] Domestic Hot Water

Figure 10-5.
Installation of ground source heat exchangers. Soil (surface) heat exchanger (left and middle) and vertical heat exchanger (right).



<u>Renovation</u>

The renovation market can be an interesting segment for heat pump boilers, provided the renovation project budget permits the financial investments needed. Depending on the budget, air-source or ground-source heat pumps may be selected and the heat distribution system can be made suitable for low temperature heating. In all cases, a feasibility study is necessary to determine the technical and economical options and to evaluate (predict) the (non-renewable) primary energy savings and CO_2 reductions that can be achieved in real life situation. This feasibility study is a crucial and difficult step, not only from a design perspective (how to optimise these predominantly bivalent heat pump systems), but also from an economic point of view. First time costs need to be evaluated against operating costs, but how does one accurately predict operating costs when there are so many operating parameters and case specific parameters involved .

<u>Retrofit</u>

The retrofit market is a very difficult market for heat pump boilers. Replacing a gas or oil boiler or combi with a heat pump is for several reasons either simply not possible or difficult and expensive to realise. In most cases the heating system is designed for higher temperature levels and an the feasibility of operating at lower temperatures (preferably < 55°C) needs to be evaluated. In all situations one would probably need a bivalent system, meaning that a heat pump system is added to an existing gas or oil boiler/combi. Only when dedicated space-heating boilers are used (no combi) and the temperature level of the heating system can be reduced, a boiler could theoretically be replaced by a bivalent heat pump (with electric back-up heater). For achieving an optimal COP the ground source heat pump boiler would be the preferred product, but for practical and costs reasons an ambient air source heat pump is probably more obvious. Important question that remains: does this situation specific retrofit bivalent heat pump system with its controls and operating conditions in real life reduces emission levels and energy consumption?

10.3.3 Experience HP countries

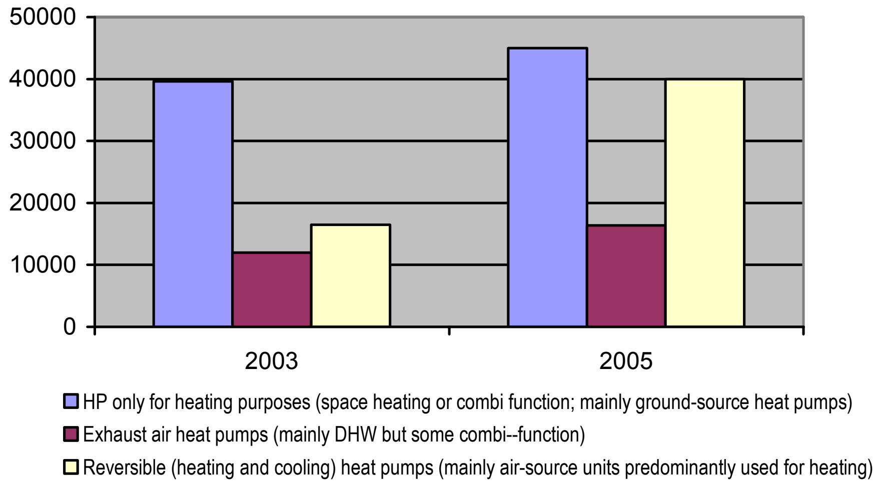
The European market for heat pump boilers according to BRGC (see task 2 report) is still relative small (> 119.000 sales in 2004) but definitely growing. According to EHPA Heat Pump Statistics that include the countries Norway and Switzerland, and include reversible heat pumps (predominantly used for space heating) and exhaust-air heat pumps (primarily DHW but also contributing to the CH-system), the total sales figure for 2005 was 235.000 units. Most important heat pump boiler countries are Sweden, Germany and Switzerland. Sweden of course because most of the dwellings are electrically heated, in which case heat pumps are practically always more efficient. A quick look at the development of their heat pump markets can be instructive for the understanding the technical features behind successful heat pump configurations.

Sweden

The Swedish heat pump market has become quite strong. Nearly 102.000 units were sold in 2005 and there is no sign of market decline. From the data collected in task 2 and 3 we learn that 1,5 million households (34% of the Swedish households) use electricity for space heating (1 million hydronic, 0,5 million dry electric systems). Energy costs per household (electricity, gas & other fuel) as average account to 1746 euro per annum in 2005, which is 23% of total expenditure for housing. Considering that heat pumps can achieve savings of 50 – 70% compared to direct electric heating, it is no wonder that heat pump boilers have acquired an important market share in this country. Because around 25.000 new houses are built per year, the biggest markets must be the renovation and replacement segment.

The total boiler market in Sweden in 2004 was around 81.000 units, of which 60.000 (**74%**) where heat pump boilers. In one year time (2004 to 2005) the heat pump boiler market grew with more than 60% to over a hundred thousand units in 2005, with the biggest growth for the ambient air-source reversible heat pump, and the biggest market share for the ground-source heat pump (see figure 1-6).

Figure 10-6.
Heat pump boiler market in Sweden (source EHPA Heat Pump Statistics)



The break-through of ground-source heat pumps for single family homes is often attributed to the market transformation programme organised by NUTEK (Swedish Agency for Economic and Regional Growth). One of the main barriers for market diffusion that had been identified by NUTEK was the poor reputation of heat pumps, caused by malfunctioning equipment in units installed. Malfunctioning of HP-installation was also an important cause for strongly changing market shares for different HP-solutions. Where the exhaust-air to water heat pumps maintained a more or less stable market share, the ambient air-to-water heat pumps saw their market share going down from almost 40% in 1984 to 2% in 1990 and around 9% in 2004. The ground-source heat pump eventually acquired the biggest market share (from 9% in 1990 to around 45% in 2005), mainly through a strong support from the Swedish government, the Swedish heat pump association and the electricity utilities. Important still on-going activities in Sweden are education of installers and labelling of heat pumps (P-mark (quality label) and Swan (Nordic Eco Label)).

Germany

In Germany the total boiler market in 2004 was around 810.000 units, of which 1,5% (around 13.000 units) were heat pump boilers. In 2005 heat pump boiler market increased with 44% to 18.217 units.

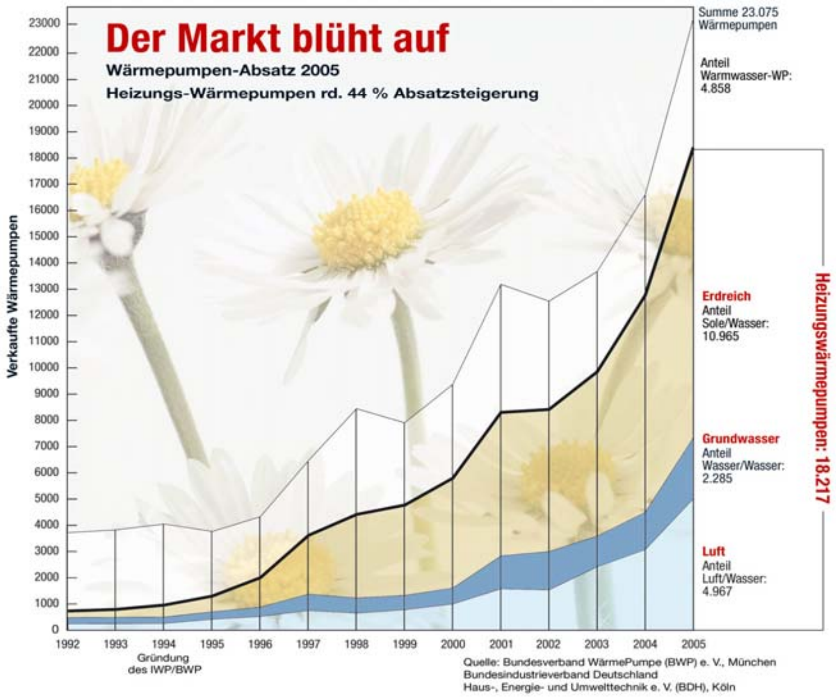


Figure 10-7.
Heat pump boiler market in Germany (Source: BWP)

Around 60% of the heat pump boilers sold were ground-source heat pump and 33% air-water heat pumps[48].

Main barriers in the German heat pump market are [49] :

- Knowledge of the heat pump technology; many craftsmen still don't have the competence to design and install heat pump boiler systems (although there are some good trained and experienced specialists).
- The investments costs for heat pump systems (including source and installation) are high in comparison with conventional technologies, and when buying a heating system, price is still the most important consideration.
- Problems during procedures to obtain permission for using ground source systems (e.g. water rights, permission to drill); proceedings are often tedious and misjudged during the planning stage
- Many HP inquiries of potential clients are diverted to convectional heating systems, because many heating designers/engineers/installers are not interested in and/or not capable of installing a heat pump system.

Supporting activities still going on in Germany:

- Bundesverband WärmePumpe (BWP) coordinates and plans well defined promotion activities in cooperation with heat pump producers, utilities and the heating engineers, electricians, sanitary- and refrigerant engineers.
- Campaigns that promote the reputation (damaged by malfunctioning HP-installations) of heat pump boilers.
- Dissemination of neutral; information- and advisory material as well as technical descriptions with planning and design guidelines.
- Congresses, special conferences and action weeks, public relations.
- Subsedies for heat pump boiler systems (measures differ per province)

The Fraunhofer-Institut für Solare Energiesysteme ISE (Germany) is going to start a four years field test for heat pumps (starting 2nd half 2006). ISE is planning to measure

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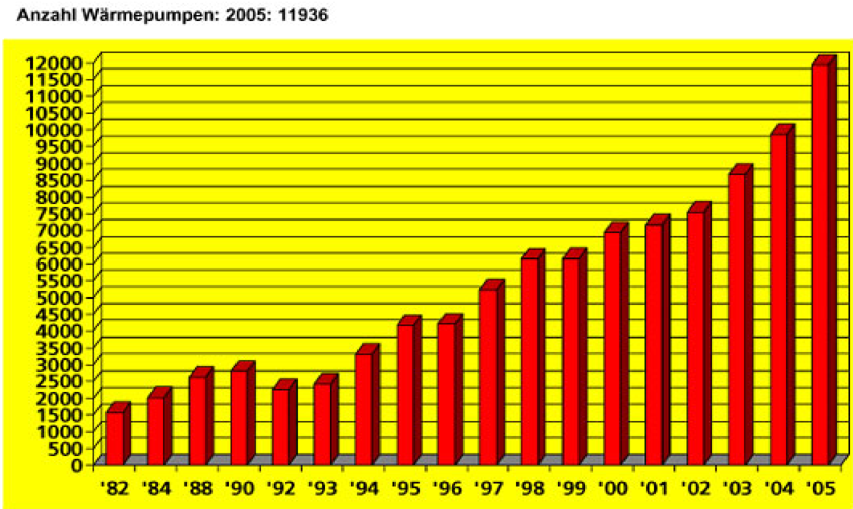
[48] Source: Bundesverband Wärmepumpe (BWP) e.V.; München.

[49] EHPA: Heat Pumps – Technology and Environmental Impact, July 2005.

140 heat pumps installed in one-family houses. In cooperation with seven heat pump manufacturers and two utilities the researchers will investigate how efficient electric heat pumps can meet the heat requirements of low energy houses.

Switzerland

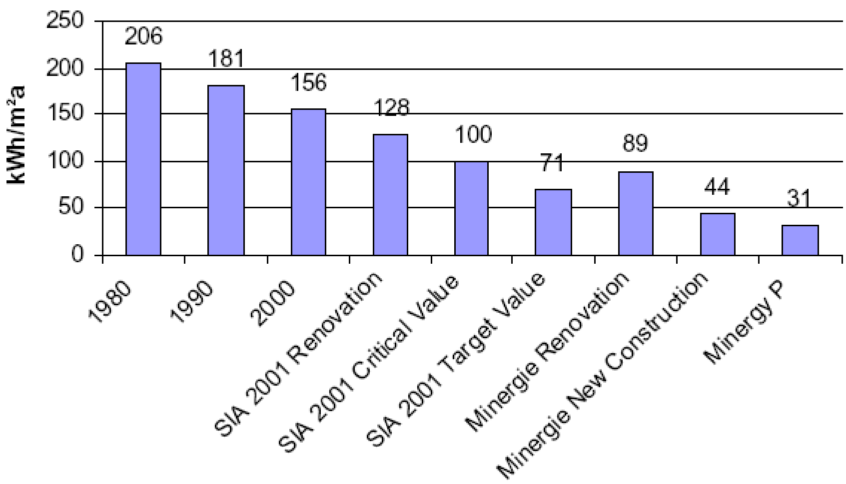
Figure 10-8.
Heat pump boiler market in Switzerland. (Source: fws.ch)



The total boiler market in Switzerland is around xx.000 units per year of which almost 12.000 units are heat pump boilers. Approximately 40% consists of ground-source/water heat pumps and 57% are air/water heat pumps; the remaining 3% are water/water heat pumps. In 2005 approximately 72% of all new one-family houses a heat pump boiler was installed (which amounts to around 10.000 units). The remaining 2000 units were installed in renovation projects. The prices for a ground-source heat pump boiler systems went down from 40.000 CHF in 1982 (25.000 euro at an exchange rate of 0,627) to 19.000 CHK (12.000 euro) in 2005.

Main cause for this high penetration in the new built sector are the continuously developing building standards, and their professional promotion.

Figure 10-9.
Development of building standards in Switzerland (Source FWS, 2003).



Also in Switzerland the main barriers in the beginning were lack of knowledge and experience with heat pump technology. There were only a few manufacturers and the installers lacked the required know-how to integrate the heat pump with the heating system in the right way. This fact brought a lot of problems in the field of after sales service.

An important reason behind the current success certainly is the fact that all relevant institutions and persons (manufacturers, Swiss Federal Office for Energy, Power

Utilities and Installer Associations) have joined forces. SFOE developed a strategic program to encourage the use of heat pumps for heating, built on three components:

- The setting up of a Swiss Association for Promotion of Heat Pumps (FWS) with the following tasks: 1. realisation and promotion of a HP quality label; 2. Coordination of training activities; 3. Deployment of information- and marketing activities.
- Improving quality and performance of heat pumps
- Financial incentives for customers installing heat pumps

Stakeholders now try to ensure quality by regularly checking and testing heat pump boiler systems, by using product quality labels, by educational programs, by certifying installers and drilling companies, and finally by appointing so called "heat pump doctors", a contact point for heat pump users with problems.

Because heat pump boilers are now considered standard for the new built sector, focus is now diverted to heat pump boilers for the replacement market.

10.4 Energy Balance and Seasonal Efficiency

Several terms are used to indicate the energy efficiency of a heat pump. This paragraph tries to shed some light upon this jumble of terms and will prepare for the framework that will be necessary to compare system efficiency of heat pumps boilers with the system efficiency of gas- and oil boilers. The prEN 15316-4-2 "Heating systems in buildings – Method for calculation of system requirements and system efficiencies" – Part 4-2: Space heating generation systems, heat pump systems" will be the key document for this framework.

10.4.1 Terminology

The following terms are being used to indicate the energy efficiency of heat pumps.

COP

The COP (Coefficient Of Performance) of a heat pump is defined as the ratio of the heat delivered by the heat pump and the energy supplied to the compressor, under steady state conditions and at a given set of temperature conditions for the heat source and the heat sink.

With electric heat pumps the COP is defined by the ratio between heat output and the electric input to the compressor. Normally additional electricity used for e.g. transportation of fluids, for controls or additional features like defrosting cycles, are not included in the COP-figure. If COP-figures are stated, there should also be a reference to the temperature conditions to which the measurement relates. Unfortunately in product documentation this is not always the case. Often COP-figures are not mentioned; instead the electric input and the related heat output is mentioned separately, in which case the COP can easily be calculated.

Example: Nibe mentions in their brochure on ground-source heat pumps the following performance data.

Technical data Fighter 1110:

- Electric input (B0/W35)* 1,0 kW.
- Heat capacity (B0/W35)* 4,8 kW.

* In accordance with EN 255 for heat source entry at 0°C / hot water flow at 35°C. The electric input for the circulation pumps is not included.

The COP of the product in this example is 4,8/1 = 4,8.

PER

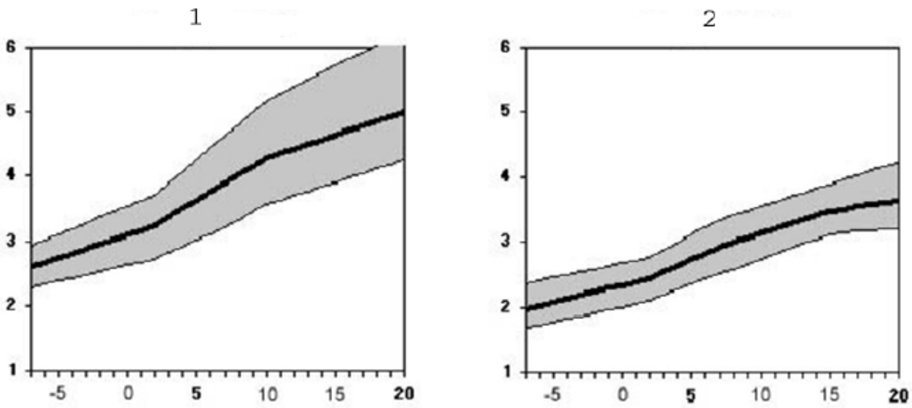
PER stands for Primary Energy Ratio, and is in fact the COP-figure but then converted to primary energy, related to the gross calorific value (GCV) or higher heating value (HHV) of the fuel supplied. As such, the PER can be compared with the steady-state efficiency figures boiler efficiency test standards.

If, for instance, the heat pump from the previous example is powered by electricity that is produced with an overall efficiency (including distribution losses) of 38%, the PER of this product is 4,8 * 0,38 = 1,82.

As is the case for boilers, the PER only gives the efficiency at a certain test condition (temperature regime for source and sink), which doesn't necessarily reflect the actual real life operating conditions of the heat pump boiler.

To give an example of typical COP- (and PER) values for different type of heat pumps at various temperature regimes, the following graphs from the Swiss national test centre WPZ at Winterthur-Töss are presented.

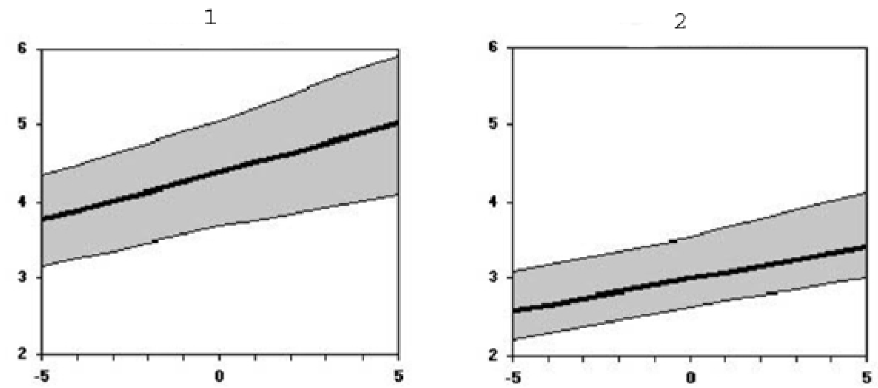
Figure 10-10.
COP values of air-to-water electrical heat pumps versus source temperatures.
Black line represents the average values; the grey area is a scatter band of values measured.
(Source: Test centre WPZ at Winterthur-Töss).



Graph left: T-supply = 35°C ; Graph right: T-supply = 50°C.
Vertical axis: COP values
Horizontal axis: Air temperatures

The COP (PER) of an electric air-source/water heat pump can vary from 1,7 (0,65) at source temperature of -7°C and sink temperature of 50°C, to 6 (2,28) at a source temperature of 18°C and a sink temperature of 35°C. A difference of a factor 3,5!

Figure 10-11.
COP values of brine-to-water electrical heat pumps versus source temperatures.
Black line represents the average values; the grey area is a scatter band of values measured.
(Source: Test centre WPZ at Winterthur-Töss).



Graph left: T-supply = 35°C ; Graph right: T-supply = 50°C.
Vertical axis: COP values
Horizontal axis: Air temperatures

The COP (PER) of an electric brine to water heat pump can vary from 2,2 (0,84) at source temperature of -5°C and sink temperature of 50°C, to almost 6 (2,28) at a source temperature of 5°C and a sink temperature of 35°C. A difference of a factor 2,7!

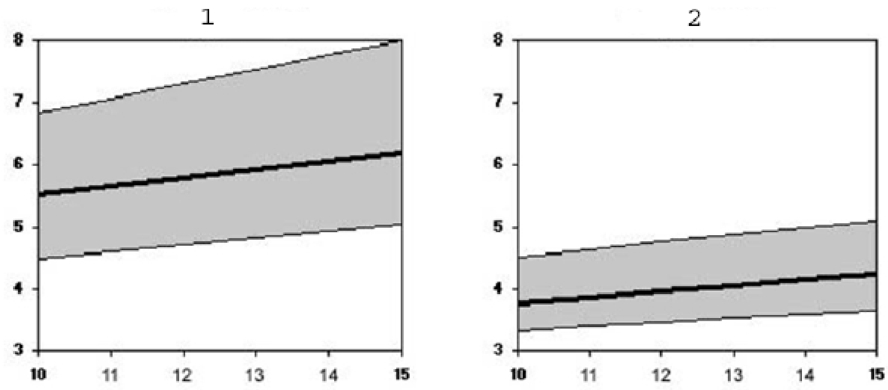


Figure 10-12.
COP values of water-to-water electrical heat pumps versus source temperatures.
Black line represents the average values; the grey area is a scatter band of values measured.
(Source: Test centre WPZ at Winterthur-Töss).

Graph left: T-supply = 35°C ; Graph right: T-supply = 50°C.
Vertical axis: COP values
Horizontal axis: Air temperatures

The COP (PER) of an electric water to water heat pump can vary from 3,3 (1,25) at source temperature of 10°C and sink temperature of 50°C, to 8 (3,04) at a source temperature of 15°C and a sink temperature of 35°C. A difference of a factor 2,4!

EER

The EER (energy efficiency ratio) is equivalent to the COP with the exception that the heat from the source is expressed in Btu/h instead of watts of kW. The EER is in fact the COP-figure multiplied by 3,413 (conversion factor from kW to Bth/h).

The term EER is a commonly used (in USA) for rating air conditioners (rating standardized by ARI (Air-conditioning and Refrigeration Institute)). This EER-figure reports the steady-state-efficiency at 95°F outdoor and 80°F indoor temperature. However, to acquire a better indication for the *seasonal performance* of air conditioners, the US Department Of Energy developed the SEER rating.

SEER[50]

The SEER is the Seasonal Energy Performance Ratio for a air-source heat pump in cooling mode. For single speed equipment, the SEER rating can simply be calculated, starting with the EER-figure that is acquired under test condition "B", which relates to temperature conditions of 82°F outdoor and 80°F indoor (respectively 27,7°C and 26,6°C). This EER test result is then lowered by a degradation coefficient (CD) to account for cycling losses, which depend on fan time delay and refrigerant control strategies (median value around 0,10). A default value for CD of 0,25 may be used in stead of testing, but this option is rarely exercised because of its relatively high impact on estimated performance. Manufacturers prefer separate testing to establish the SEER because by using a timed indoor fan off delay and control of post-cycle refrigerant migration, they can achieve lower CD values (≤ 0,05).

HSPF[11]

HSPF stands for Heating Seasonal Performance Factor and is determined in an analogous fashion to SEER, but in heating mode. It is the total useful heating output of the heat pump during in normal seasonal use, divided by the total electrical power input. Here test data is obtained at three different test conditions: 47°F, 35°F and 17°F (respectively 8,3°C, 1,6°C and -8,3°C). The performance is interpolated at these points with the results applied against appropriate weather bin data to obtain seasonal performance. The labelled HSPF on heat pumps is that for US climate zone IV. Similar to cooling a part load degradation value (CD) is used to evaluate the impact of system

[50] Fairy, P., D.S. Parker, B.Wilcox and M. Lombardi, "Climate Impacts on Heating Seasonal Performance Factor (HSPF) and Seasonal Energy Efficiency Ratio (SEER) for Air Source Heat Pumps", AHRAE Transactions, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc., Atlanta, GA, June 2004.

cycling under less than design load conditions. However, CD-values tend to be somewhat higher for heating although there is less empirical data (0,15 to 0,20). An assessment of a typical heat pump found that a 90% runtime at 10°C outside resulted in an 11% drop in capacity and a 7% lower COP than steady state operation. At 20% runtime fraction heat output was degraded by 35% with COP reduced by 30%.

Researchers Fairy, Parker, Wilcox and Lombardi[11] found, that there are several aspects within the procedure for determining the HSPF, that artificially benefit the HSPF rated performance, compared with the actual situation in the field:

- Share of back-up heating; within the test protocol, the heat capacity of the HP-system is arbitrarily reduced by 23% relative to the actual design heat load. Furthermore the test procedure temperature setpoint is 65°F (18,3°C), instead of the more common indoor temperature levels of 68°F (20°C). Both aspects will reduce the frequency with the back-up heater is used.
- Whereas the defrost operation is typically based on the compressor runtime in real systems, the ARI[51] procedure assumes there is no defrost operation below 17°F (-8,3°C) where defrost operation will, in fact, be most often triggered due to the extended compressor operation.
- Although the vast majority of air-source heat pumps operate auxiliary strip heat (electric resistance heating) during the defrost cycle to prevent “cold blow”, the ARI procedure specifically requires that strip resistance heaters be prevented from operating during the frost accumulation test. This is important since such operation of strip heat during the 3 – 10 minute defrost cycles satisfies part of the house heating load at a lower efficiency that is not reflected in the ARI procedure.
- To achieve better performance in the most severe climate, the ARI procedure computes a smaller building load for the colder Climate Zone V than it does in the more moderate climate zones (I - IV and VI). This results in much lower use of strip heat than would otherwise be encountered in the coldest locations. Conversely, the ARI procedure intrinsically assumes that homes in the mildest locations (Zone I) are just as well insulated as those in Zone IV.
- Finally, beyond standard test conditions, while lower than nominal indoor unit coil air flow will actually increase latent heat removal in cooling mode, there is no such compensation in heating mode. All reductions to system heating capacity due to low coil airflow are a loss to system operating efficiency, generally resulting in increased strip resistance backup.

Main topic of the study performed by Fairy a.o.[11] however was not to determine how much the standardized HSPF deviates from real life seasonal performance, but how much one may expect performance to vary with climate conditions, as deviation from the single published HSPF-value based on performance in region IV. The analysis shows that the seasonal heating performance changes substantially with climate conditions. The climate related HSPF was found to be as much as 40% better (in warmest climates) or 50% worse (in coldest regions) than the published values on energy labels. An air source heat pump with a standard HPSF of 8,5 (annual COP = 2,5) can have an actual heating seasonal performance factor of 4,25 (annual COP = 1,25) in the extreme cold regions.

For Europe these deviations will be smaller, due to smaller differences in heating degree days and average temperatures. It is estimated that the seasonal performance in colder climates can vary to maximum of 35% (compared to the 50% for the USA) from the standardized average HSPF-value, which will also be another figure that the US- value.

With ground-source heat pumps these deviations will be smaller because of smaller variations in heat-source temperatures.

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[51] Air-conditioning and Refrigeration Institute, USA

The existing minimum standards of residential air-source heat pumps in the US are given in the table below.

Figure 10-13.
Minimum standards for HSPF (Heating Seasonal Performance Factor) for air-source heat pumps in US.[52]

	Minimum standards in effect since 1992	**Minimum standards in effect since January 2006**
Split systems	6,8 HSPF (2,0 an.COP)	7,4 HSPF (2,2 an.COP)
Through the wall split systems		7,1 HSPF (2,1 an.COP)
Through the wall single package	6,6 HSPF (1,9 an.COP)	7,0 HSPF (2,05 an.COP)

SPF

The SPF is similar (not identical) to the HSPF-value used in the USA. SPF stands for Seasonal Performance Factor and can be defined as the ratio between the year heat output and the year energy input of the heat pump boiler system.

In Europe the prEN 15316-4-2 standard[53] is being developed for the calculation of this Seasonal Performance Factor for heat pump systems. Objective of this new standard is, to evaluate the seasonal performance (SPF) of "*heat pump systems"* rather than the seasonal efficiency of the heat pump as component. After all, current innovative heat pump systems practically always use back-up heat generators working in parallel or alternating mode. They are also often combined with storage tanks for domestic hot water production, and the heat sources (air-source, ground source, water source) and temperature levels of heat sinks can largely influence the annual performance.

Key input for the prEN standard was the FHBB method for calculating SPF, developed in annex 28 of IEA Heat Pump Program.

10.4.2 Seasonal Performance Factor (IEA HPP Annex 28)

Within the Heat Pump Program of the IEA (International Energy Agency) an Annex was formed (Annex 28) with the objective to develop test- and calculation methods for combined operating heat pumps systems. In a first step, existing standards and calculation methods were evaluated, with a specific focus on calculation methods that evaluate the annual frequency of the ambient temperature as basis to weigh corrected COP values at various operating points (a similar method is pursued in Annex 28). Five of these methods were found and evaluated, by comparing their calculation procedures and the parameters involved. To compare these methods, the SPF was calculated for a standard commercial air to water heat pump projected in a well defined building.

Depending on the method used, SPF values vary between 2,8 and 3,9 (see graph in next figure). Results clearly show that an unambiguous approach is necessary in order to be able to compare energy efficiency of heat pumps.

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[52] Tomlinson, J.J., Oak Ridge National Laboratory, "US standards, testing and labelling procedures for residential heat pumps and air conditioners", IEA Heat Pump Centre Newsletter, Volume 21 – No. 1/2003.

[53] prEN 15316-4-2, Heating systems in buildings – Method for calculation of system energy requirements and system efficiencies – Part 4-2: Space heating generation systems, heat pump systems; October 2005

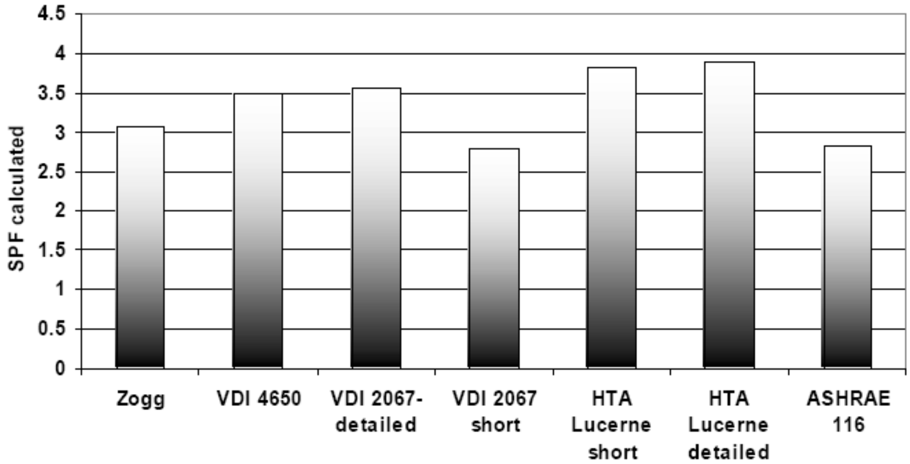
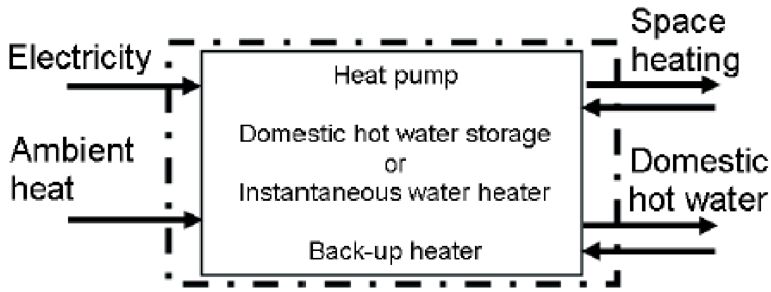


Figure 10-14.
Comparison calculation methods for SPF (climate data of Stuttgart).[54]

Based on a survey of state-of-the-art systems on the market and existing standards and calculation methods, the participants in IEA HPP Annex 28 agreed on extending existing standards to include combined operation of heat pump systems. The objective of the new test standard is to provide enough data on components characteristics to enable seasonal performance to be calculated. The following system boundaries were defined.

Figure 10-15.
System boundaries for new testing procedure.
Source: IEA Heat Pump Newsletter No.3 / 2005



When only the values at the system boundaries are measured, different internal system configurations can be taken into account (including heat pump, hot water storage and any back-up heater).

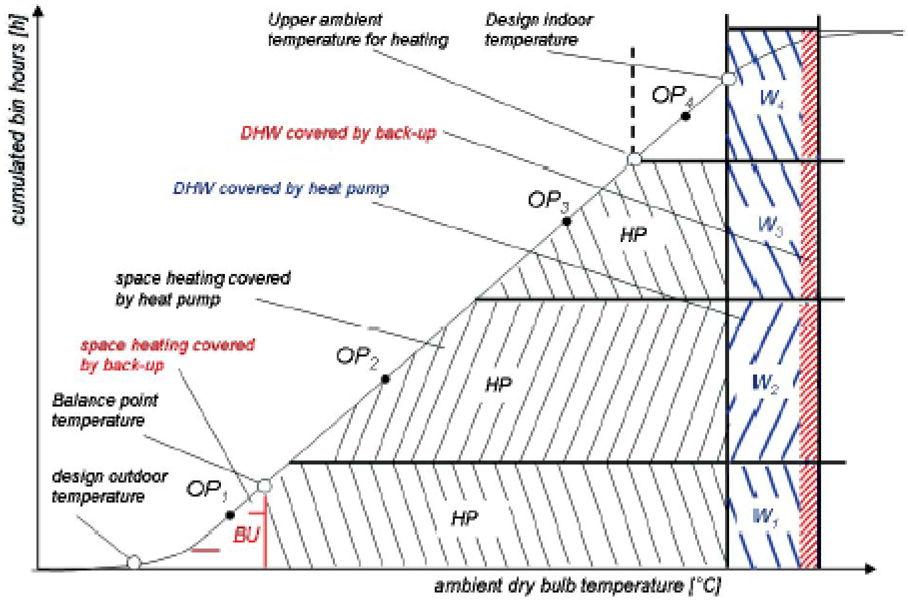
For calculation of the seasonal performance factor based on standard testing, the participants agreed to build on the bin methodology which is already included in different national calculation standards for heat pumps. Calculation is performed by temperature ranges (bins), related to the duration of the outdoor air temperature. The performance factors at the centre of the bins, which are based on the test results and additional system energies, e.g. storage losses or auxiliary energy that are not considered during testing, are weighted with the respective energy amount which is represented by the ratio of the bin area to the total area. Subsequently, the single bins are summed up to the seasonal performance factor.

Figure 1-16 shows the principle of the bin methodology. For combined operation, this bin-method is extended to a third operation mode. The fraction of each operation mode is evaluated by the running time which is calculated by the energy requirements divided by the respective output capacity in each bin. The three seasonal performance factors

[54] Wemhöner C., prof. Dr. Th. Afjei, University of Applied Sciences Basel, Institute of Energy, "Seasonal performance calculation for residential heat pumps with combined heating and hot water production (FHBB Method)", October 2003.

are combined to an overall seasonal performance factor by weighting them with the energy fraction in each operation mode.

Figure 10-16.
Principle of the bin calculation
Source: IEA Heat Pump Newsletter No.3 / 2005



The final result of Annex 28 is a calculation method (FHBB Method) that is well suited for the determination of the SPF of heat pumps for combined heating and domestic hot water production (also applicable to single heating or DHW heat pumps). The FHBB method is based on the wide spread bin method, which considers the following physical effects:

- Meteorological influences of ambient dry bulb temperature of the site
- Impact of sink and source temperature on the COP and output capacity
- Losses due to cyclic operation
- Storage losses of both heating and hot water storages integrated in the system
- Additional auxiliary energies supplied to source and sink temperatures
- Mono-energetic operation for three operating modes (partly parallel, parallel, alternate)

To consider simultaneous operation, an approach of energetic weighting with the respective energy demand fraction in single and combined mode has been integrated. The energy fraction of combined and single operation are estimated by the respective energy requirements and the output capacities of the heat pump.

An imperfection in the FHBB-method (that will become more important in the near future) is the fact the impact of control systems are not considered. Preliminary simulations show for instance, that an enhanced control system, which takes into account the operating state of the heating system, increases the simultaneous operation from about 30% to 60% compared to a simple two point hysteresis control that does not consider the state of the heating system. Moreover, dynamic testing of the heat pump to characterize cyclic operation does not exist in European standardisation.

In Switzerland a test procedure has been developed[55], on the basis of which an approach for cyclic operation has been integrated in the FHBB method.

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[55] E. Shafai, D. Zogg, M. Ehrbar, L. Wirth: "Dynamischer Wärmepumpentest, Phase 1 – 3, Etappe 3, Modellansatz für die prüftechnische Charakterisierung der Minderwärmeproduktion, SFOE research program heat pumping technologies, 2001)

The existing European standards do not deliver sufficient information for a proper assessment of simultaneous operating systems. For this reason a test procedure based on the existing European standards has been developed in the Swiss national test centre WPZ Töss. This method however is time consuming and therefore expensive, because it involves the measurement at several operating points, while a test cycle according EN 255-3 for one point already lasts about 4 days.

As far as possible, analytical approaches have been chosen. For the impact of cyclic operation and the impact of controls on simultaneous operation, simulations have been used to derive appropriate correction factors.

Example

The FHBB-method is applied to two different brine-to-water heat pump boiler systems:

- An alternate working heat pump combi system with a Novelan Siemens SIC 9M brine-to water heat pump. The system has a parallel domestic hot water storage with a hot water priority control (i.e. in case of hot water demand, the supply of the heating is interrupted until the hot water demand is covered).
- A simultaneous operating combi system that consists of two heat pumps mounted in a cascade. The lower one (KWT Swissline 40 NHB) has a bigger compress and is used for space heating. The upper one (KWT SC 15G) has a smaller compressor and is used for the domestic hot water.

Figure 10-17.
System characteristics of the alternate and simultaneous heat pump system.

	Alternating HP System	**Simultaneous HP system**
	HEAT PUMP DATA	**SPACE HEATING HP DATA**
Manufacturer /type	Novelan Siemens SIC 9M	KWT Swissline 40 NHB
Output capacity at B0/W35	9,1 kW	9,5 kW
Electrical power input at B0/W35	2,1 kW	2,1 kW
COP at B0/W35	4,4	4,6
Compressor type	Scroll, fully hermetic	Scroll, fully hermetic
Condensor	Flat plate heat exchanger	Flat plate heat exchanger
Evaporator	Flat plate heat exchanger	Flat plate heat exchanger
Injection valve	Thermostatic	Thermostatic
Refrigerant	R 407 C	R 407 C
Source working fluid	25% ethylene glycole	25% ethylene glycole
		DHW HEAT PUMP DATA
Output capacity at B0/W35		KWT SC 15 G
Electrical poer input at B0/W35		2,2 KW
COP at B0/W35		0,7 KW
Compressor type		3,1
Condensor		Recipr. piston, fully hermetic
Evaporator		H.E. in DHW storage
Injection valve		Flat plate heat exchanger
Refrigerant		Thermostatic
Manufacturer /type		R 134 a
	DHW STORAGE	
Volume	300 litre	200 litre
Heat exchange area	3,5 m²	
Thermal insulation (PUR foam)	50 mm	50 mm

Both heat pump systems were applied in a Retrofit building and a Low-energy building (according to MINERGIE standards). In case of the low-energy building, a monovalent operation is possible. IN case of the retrofit building, a mono-energetic parallel operation is applied for space heating (electric back-up heater). As both heat pumps can deliver heat up to a temperature of 55°C, no electrical back-up heater for the DHW-mode is applied. The parameters for the demand side of the HP-systems are given in the following figure.

Figure 10-18.
Demand-side parameters for the heating and DHW-system in the Low-energy building and the Retrofit building[56]

Parameter	Low-energy building	Retrofit building
Meteorological data	Zurich SMA	Zurich SMA
HEATING SYSTEM		
Design building load (SIA 384/2)	5 kW	8,8 kW
Design outside temperature (SIA 384/2)	-11°C	-11°C
Heating energy demand (SIA 380/1)	10.039 kWh/a	20.158 kWh/a
Interruption of electricity supply	3 hours per day	3 hours per day
Design room temperature	20°C	20°C
Settings of the heating curve	-11/35°C | 15/25,2°C	-11/55°C | 15/28,7°C
DHW SYSTEM		
Cold water temperature	15°C	15°C
Hot water temperature with heat pump	55°C	55°C
Tapped amount of hot water	45 l/pp at 55°C	45 l/pp at 55°C
Number of persons	4	4
Hot water energy demand	3053 kWh/a	3053 kWh/a
Fraction of DHW on energy demand	23,3%	13,2%

SPF-values, calculated according to FHBB method.

Figure 10-19.
SPF-calculations according to FHBB-Method for two different heat pump configurations in two different building types.

	Alternating system with NOVOLAN SIC 9M		Simultaneous system with KWT Swissline	
	Low-energy building	**Retrofit building**	**Low-energy building**	**Retrofit building**
COP values according to EN 255 *(electricity consumption source pump not included)*				
COP acc. EN 255-2 (heating)	B0/W35 =4,35 / B0/W50 =3,0		B0/W35 =4,46 / B0/W50 =2,95	
COP acc. EN 255-3 (DHW)	B0/W35 = 3,02		2,36	
SPF values HP generator *(= Source- and HP system, electricity consumption source pump included)*				
SPF hp; heating	4,54	3,57	4,87	3,68
SPF hp; DHW	2,75	2,75	2,02	2,19
Combined SPF_{hp} for Heating &DHW	**3,83**	**3,40**	**3,50**	**3,22**
SPF values HP generation system *(= Source- and HP system, storage tank, back-up heater, electric input auxiliaries)*				
SPF gen; heating	4,54	3,37	4,87	3,42
SPF gen; DHW	2,11	2,11	1,60	1,73
Combined SPF_{gen} for Heating &DHW	**3,58**	**3,13**	**3,30**	**3,03**
SPF values overall system *(= Source- and HP system, storage tank, back-up heater, electric input auxiliaries, circulation pump)*				
SPF syst; heating *(η on prim.energy*)*	3,76 *(1,5)*	3,14 *(1,26)*	3,87 *(1,55)*	3,16 *(1,26)*
SPF syst; DHW *(η on prim.energy*)*	2,11 *(0,84)*	2,11 *(0,84)*	1,60 *(0,64)*	1,73 *(0,69)*
Combined SPF_{syst} for Heating &DHW	**3,18 *(1,27)***	**2,95 *(1,18)***	**2,95 *(1,18)***	**2,85 *(1,14)***
* conversion factor electricity to primary = 0,4				

Only for the retrofit buildings a small amount of back-up resistance heating is incorporated in the calculations (112 kWh and 204 kWh for respectively the alternating system and the simultaneous system).

Comments on SPF-values of example FHBB calculation method

Please note, that these examples are built around typical monovalent HP-systems, where the electric back-up heater was not at all or only minimally used. In the low energy houses no back-up electricity was used at all (not for heating nor for DHW), and

[56] Wemhöner C., prof. Dr. Th. Afjei, University of Applied Sciences Basel, Institute of Energy, “Seasonal performance calculation for residential heat pumps with combined heating and hot water production (FHBB Method)“, October 2003.

in the retrofit building only very small amounts of electric back-up heating for space heating were incorporated into the calculations. This most probably does not correspond with the average real life situations for space heating and DHW in retrofit applications:

- For DHW the storage tank temperatures are generally (temporarily) kept at 60°C or higher (instead of the 55°C delivered by the heat pump) to prevent legionnaires disease. Further more, the electric back-up element is sometimes also used to speed-up partial reloading of the storage tank, to cope with heavy tapping patterns. These conditions are not taken into account. Calculations are based on the COP_t – value as defined in EN 255-3, which measures the efficiency of one draw, during which half the storage tank is unloaded and adds the average electrical input needed to cover for standing losses. It is therefore expected that in average real practice, the $SPF_{syst\ DHW}$ values will be lower**.** At a storage temperature of 60°C an alternating back-up heater would have an additional input of around 300 kWh/a and the SPF would reduce to 1,74.
- For space heating, the SPF calculations are also based on monovalent operation of the HP-heating system, i.e. In real life, retrofit buildings will not be equipped with a heat pump output that covers the design heat load of the house, but typically 40 to 70% (for bivalent parallel or bivalent alternating operation (acc. DIN 4701))). Main reasons for this are a) minimisation of cyclic operation of heat pump and b) optimisation of the investment/savings-ratio. With a HP-system that covers 60% of the design heat load, approximately 90% of the yearly thermal energy for space heating can be covered with the HP-system. The remaining 10% is produced by the back-up heater. Assuming an alternating electric back-up heater in the exemplary calculations for the retrofit buildings on the previous page, around 2015 kWh would have to be produced by the back-up system. The $SPF_{syst;heating}$ figures would drop from 3,14 to 2,38 for the alternating combi system and from 3,16 to 2,40 for the simultaneous operating combi system. Multiplying with 0,4 (conversion factor electricity to primary energy) the annual performance factor (on primary energy) would be approximately **95%.**
- Another issue that is not addressed in the calculation method is the fact that the heat distribution system doesn't always operate with a constant flow and with a constant ΔT over supply and return of the heating system. As a result the amount of heat that is transferred from the condenser to the heating system will vary and consequently the COP-values will change. Measured COP-values tend to be better with small volume flow rates in the condenser, as the average temperature decreases. The EN 255-2 only defines the supply temperature of the heating system (i.e. the outlet temperature of the condenser), but not the volume flow nor the return temperature. For test conditions, a heat pump manufacturer would opt for a low flow and a low return temperatures because they result in higher COP-values. In real life (as we know from previous chapters in this task 4 report) the volume flows can vary considerably (caused by TRVs and pump), causing increased flows, increased return temperatures, higher average temperatures and as a result, lower COPs.
- The overall SPF-values for the combined operation (heating and DHW) for the retrofit buildings with 10% of the annual heat produced by an electric back-up heater and 60°C DHW storage temperature will be 2,35 (for the alternating system) and 2,28 for the other system. On primary energy: **94%** respectively **91%.**

10.4.3 prEN 15316-4-2

The prEN 15316-4-2 (Methods for calculation of system efficiency of heat pump systems) is largely based on the method developed in Annex 28 of the IEA HP Program (see previous paragraph).

The calculation method applies to electrically driven heat pumps, gas motor driven heat pumps and absorption heat pumps. Similar to the Annex 28 method, the following physical parameters are taken into account:

- type of hp generator (air-to-water, liquid-to-water, air-to-air)
- space heating and domestic hot water requirement
- effects of variation of source and sink temperature on output capacity and COP
- effects of part load operation (cyclic losses)
- auxiliary input needed to operate the generator not considered in standard testing of output capacity and COP.
- System losses due to built-in storage components.

The calculation principle uses the following input data:

- Type of heat pump (air-to-water, liquid-to-water, air-to-air etc.)
- System configuration (integrated DHW-production, operating mode of the back-up system)
- Ambient conditions (meteorological data of the site)

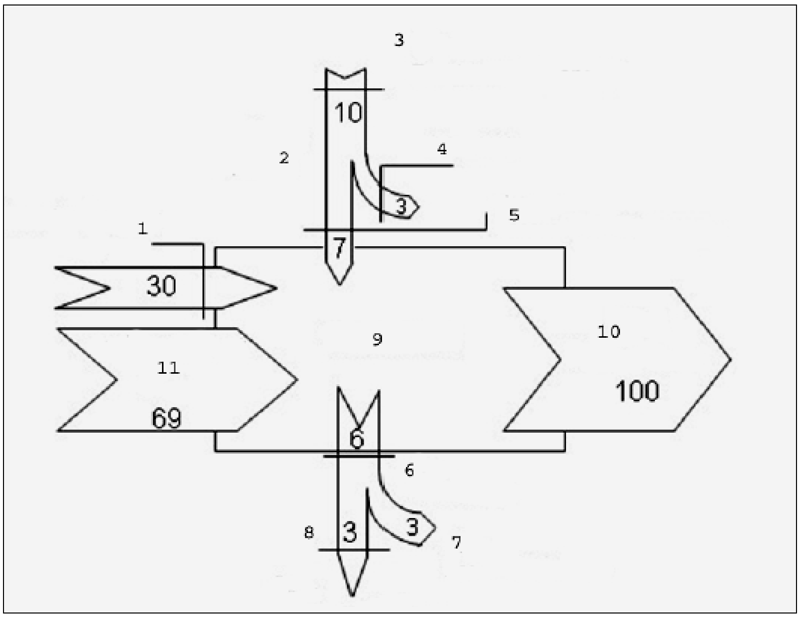
And generates the following output data:

- Required energy as electricity ($E_{in;g}$) or fule ($G_{in;g}$) to meet the space heating and/or domestic hot water requirements
- Total Generator heat loss ($Q_{l;g}$)
- Required auxiliary energy (W_g) to operate the generator
- Total recoverable generator heat losses ($Q_{rl;g;t}$)

The following heat balance can be made for the generator subsystem

Figure 10-20.
Heat balance generator system with electr. driven heat pump.

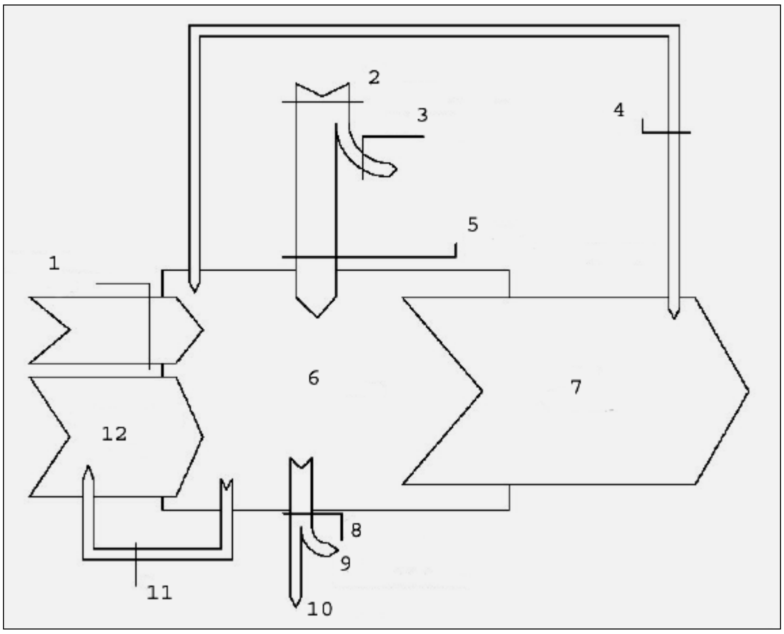
1. $E_{in,g}$: electrical energy
2. auxiliary energy
3. W_g: total aux. energy
4. $(1 - k_{rd,g}) * W_g$
5. $W_g * k_{rd,g}$:recov.aux.energy
6. $Q_{l,g}$: total losses
7. $Q_{rl,g}$: recov. losses
8. $Q_{nh,g}$: unrecov. losses
9. generation
10. $Q_{out,g} = G_{in,distr}$: heat
11. $Q_{in;g}$: ambient heat



Energy balance: $E_{in,g} = Q_{out,g} + Q_{l,g} - Q_{in;g} - k_{rd,g} * W_g$

Figure 10-21.
Heat balance generator system with gas driven heat pump.

1. $E_{in,g}$: fuel energy
2. W_g: total aux.energy
3. $(1 – k_{rd,g}) * W_g$
4. Qmot,g
5. $W_g * k_{rd,g}$: recov.aux.energy
6. generation
7. $Q_{out,g} = G_{in,distr}$: heat
8. $Q_{l,g}$: total losses
9. $Q_{rl,g}$: recov. losses
10. $Q_{nh,g}$: unrecov. losses
11. Qx
12. $Q_{in;g}$: ambient heat



Energy balance: $E_{in,g} = Q_{out,g} + Q_{l,g} - Q_{in;g} - Q_{rd,mot,g} - k_{rd,g} * W_g$

The term $E_{in,g}$ refers to the energy needed to operate the heat pump according to the testing in EN 14511 (formerly EN255). This standard considers partly the auxiliary energy for source and sink pump, namely the fraction to overcome the internal pressure drop in the heat pump evaporator and condenser, as well as the auxiliary energy for the control and defrosting.

Heat losses $Q_{l,g}$ only occur in unitary systems with integrated hot water storage in the form of storage heat losses. Part of the losses are recoverable, and part of these recoverable losses are actually recovered. The recovered losses are determined by the location of the generator and the utilisation factor (gain/loss-ratio, see ISO 13790). Recoverable heat losses are e.g. heat losses through the envelope of a generator that is installed in a heated space.

$K_{rd,g}$ describes the fraction of auxiliary energy, which is recovered as thermal energy, e.g. for pumps where a fraction of the auxiliary energy is directly transferred as thermal energy to the transporting fluid.

W_g is the auxiliary energy that is needed to operate the generation subsystem, e.g. the source pump and the control system of the generator. Because this prEN standard calculates the heat pump output capacity and COP on the basis of results from component testing (according to EN 14511/ EN255), only the auxiliary energy not included in the component test results is considered in W_g.

Combined space heating and DHW operation

For combined operation of the heat pump, the prEN distinguishes two control systems: the alternate and the simultaneous operation.

For alternate operation the heat pump switches from the space heating system to the DHW system in case there is a domestic hot water demand. A domestic hot water storage vessel is placed in parallel with the space heating system; the DHW-operation is given priority, i.e. the space heating operation is interrupted in case of DHW-demand.

In simultaneous operating systems, the heat pump covers the space heating and DHW-requirements at the same time. For simultaneous operation 3 modes can be distinguished:

- Single space heating
- Single domestic hot water operation
- Simultaneous operation.

SPF calculation methods according to prEN 15316-4-2

The prEN describes two SPF-calculation methods, the *simplified method* and the detailed *BIN-method* based on component efficiency data. The two methods differ with respect to the required input data, the operating conditions that are taken into account and the calculation period.

<u>Simplified method</u>

For this method, the considered calculation period is the heating season. The performance is chosen from tabulated values for fixed performance classes of the heat pump, based on test results according to the heat pump testing standard EN 14511 (or EN 255) and on the Bin-method (see below). The operation conditions (climate, design and operation of the heating system, heat source type) are based on typology of implementation characteristics and are not case specific. The method allows a country/region specific approach and requires a country/region specific national annex. This method can only be used if an appropriate national annex is available.

<u>Detailed Bin method</u>

This method is also based on the test results according to heat pump testing standard EN 14511 (or EN 255), but supplementary data are needed to asses the specific operating conditions of each individual installation. Therefore, the heating period is split-up in bins dependent on the outside temperature. The calculation is carried out for the corresponding operating conditions of the heat pump (see also FHBB-method). This method is not limited and can be used with the default values given in the informative Annex B of the prEN, if no other values are available.

Example of the simplified method

Simplified Method Netherlands (normative Annex B in NEN 5128:2004)

<u>Seasonal generation efficiency (η_{gen}) of monovalent heat pump boiler for residential buildings</u> (auxiliary energy excluded)

Default values

	$\Theta_{suppl} \leq 35°C$	$35 < \Theta_{suppl} \leq 45°C$
Soil-to-water	$3,8 * \eta_{el} * C_{bron}$	$3,4 * \eta_{el} * C_{bron}$
Groundwater -to- water	$4,5 * \eta_{el} * C_{bron}$	$4,1 * \eta_{el} * C_{bron}$
Outside air - to- water	$3,7 * \eta_{el}$	$3,3 * \eta_{el}$

Values based on minimal COP-values acquired through EN 14511-2

	$\Theta_{suppl} \leq 35°C$	$35 < \Theta_{suppl} \leq 45°C$
Soil-to-water	$4,4 * \eta_{el} * C_{bron}$	$4,1 * \eta_{el} * C_{bron}$
Groundwater -to- water	$5,0 * \eta_{el} * C_{bron}$	$4,6 * \eta_{el} * C_{bron}$
Outside air - to- water	$3,8 * \eta_{el}$	$3,5 * \eta_{el}$

With:

- η_{el} = value for the overall efficiency for power generation: 0,39
- c_{bron} = correction factor for collective source or regeneration of individual ground source; if not applicable, value = 1,0.

<u>Auxiliary energy heat pump boiler</u>

- uncontrolled circulation pump: $2,2 * A_{g,l}$ ($A_{g,l}$ = surface of habitable heated space)
- variable speed circulation pump: $1,1 * A_{g,i}$

Controls HP-system : $0,88 * A_{g,i}$

<u>Seasonal generation efficiency (η_{gen}) of bi-valent heat pump boiler for residential buildings</u> (auxiliary energy excluded).

When more than one generator is used, the seasonal generation efficiency (η_{gen}) of the combined generator system can be calculated with the following formula:

$$\frac{1}{f_{pref} / \eta_{pref} + (1 - f_{pref}) / \eta_{npref}}$$

With:

f_{pref} = share of heat supply of the preferential generator (to be based on the beta-factor, which is the ratio of the preferential generator to the total energy consumption for space heating, see table below)

η_{pref} = the related seasonal efficiency (see tabled values)

η_{npref} = seasonal efficiency of the secondary heat generator

According to this Dutch Simplified Method, the following formula can be used to determine the beta-factor : $\beta = 1000 * P_{pref}$ [in kW] / $0{,}5 * Q_{out;g}$ [in MJ]

The relation between the beta-factor and the share of heat supplied by the preferential generator is given in the following table:

β	0,05	0,10	0,15	0,20	0,30	0,40	0,50	0,60	0,70	0,80	0,90
f_{pref}	0,15	0,29	0,44	0,59	0,88	0,91	0,92	0,94	0,95	0,97	0,98

How to determine SPF

The SPF of a heat pump boiler system with the requested minimal (EN 14511-2) COP values, according to this Simplified Method is:

$$\frac{Q_{heat\ load}}{Q_{heat\ load} / \eta_{gen} * C_{bron} + (2{,}2 + 0{,}88) * A_{g,l} / \eta_{el}}$$

This simplified method is applied to an average new house and an average existing house (data from task 3 report) for all the three different type of heat sources. For the new house a monovalent system is used, for the existing house a bivalent (mono-energetic) system, with an electric back-up heater that delivers 9% of the total heat.

Calculation example with simplified method

Figure 10-22.
SPF-calculations according to Dutch version of Simplified Method (referred to in prEN15316-4-2) for an average new house and an average existing house

Remark
*Values for the seasonal generation efficiency for the air-to-water heat pump (3,8 * η_{el}) acc. to the Dutch Simplified Method, appear relatively high, given the EN test COP of A7W35 = 3,0*

	Average new house Size : 103 m² Heat load : 4129 kWh HP system : monovalent HP-share : 100% (f_{pref}) Back-up : n.a. T-regime : ≤ 35°C	**Average existing house** Size : 86 m² Heat load : 7483 kWh HP system : bi-valent HP-share : 91% (f_{pref}) Back-up : electric. T-regime : ≤ 45°C
Groundwater-to-water heat pump with min. COP acc. EN 14511-2 W10/W45 = 4,2 / W10W35 = 5,1	140%	124%
Soil-to-water heat pump with min. COP acc. EN 14511-2 B0W45 = 3,4 / B0W35 = 4,0	128%	111%
Outside air-to-water heat pump with min. COP acc. EN 14511-2 A7W45 = 2,9 / A7W35 = 3,0 / A-7/W45 = 2,0	114%	99%

The detailed Bin-method

This detailed "case specific seasonal performance method" is based on the following data:

- Meteorological data of outside temperature of the site in hourly distribution (e.g. test reference year).
- Heating energy requirement according ISO 13790
- Controller setting of the heat emission system (characteristic heat curve)
- Heat pump characteristics for output capacity and COP according to EN 14511 (or former EN 255-2).
- Heat pump characteristics for part load operation according to CEN/TS 14825
- Balance point (or alternatively design heat load acc. to EN 12831)
- System configuration (type and capacity of heat emission system, operating mode of back-up system, installed storage systems and heat loss coefficient)
- Auxiliary components
- Nominal power of source pump, storage loading and/or circulation pump.
- Power consumption of control system.

In case Domestic Hot Water (DHW) is also produced, the following additional data is necessary:

- Domestic hot water energy requirement.
- Set temperature for the energy delivery by the heat pump (e.g. 55°C)
- Parameters of the domestic hot water storage (heat loss coefficient, DHW-design temperatures, controller settings of the storage (T for start of storage reloading)

Calculation is then performed according to the following steps:

1. Determination of energy requirements of the single Bins
2. Determination of back-up energy of the single Bins
3. Correction of steady-state output capacity for actual Bin- source and –sink temperature (inter- and extrapolation between COP values of EN-testing points)
4. Correction of COP for part-load operation.
5. Calculation of the running time of the heat pump in different operating modes.
6. Calculation of auxiliary energy input
7. Calculation of generator heat losses and recoverable generator heat losses
8. Calculation of total energy input to cover the heat requirements

Informative Annex B of the prEN 15316-4-2:2005 contains default values for parametering this case specific heat pump calculation method (Bin-method).

The example that is used in Annex D of the prEN 15316-4-2:2005 to illustrate the Bin-method is almost identical to the exemplary calculations made by Annex 28 of the Heat Pump Program if IEA (the FHBB-method; see previous paragraph).

This Annex D example is summarized in the following table, giving an overview of the Bins that were used for the calculation, the corrected COPs for each Bin, the heat production for each Bin and the electrical consumption (for all components) in each Bin.

Calculation example with detailed Bin method

Figure 10-23.

SPF-calculations according to detailed Bin-method (referred to in prEN15316-4-2)

Source: Informative Annex D of the before mentioned prEN.

HEAT PUMP BOILER SYSTEM DATA										
System configuration	Brine source heat pump for combined operation for space heating and DHW (COP B0W35=4,35)									
Operating mode	Alternating mode for Space-heating/DHW									
Energy input	Bivalent, with parallel operation of electric back-up heater; balance point at -6°C									
Data on space-heating system	Required heating energy for heat distribution system: 20158 kWh/a | Emitter type: radiators; | Upper T-limit for space heating: 15°C | Heating curve & design temperature: 55°C at -11°C and 28,7°C at 15°C | Indoor temperature: 20°C |									
Data on DHW-system	Required energy: 3434 kWh | Hot water temperature: 60°C | Max. DHW-temperature provided by heat pump: 55°C | Storage volume: 300 ltr. | Storage heat losses: 2,4 W/K | Storage loading control: switch on at 50°C and switch off at 60°C |									
Bin number	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**Sum**	**Total**
Averega Bin temperature ($T_{outside}$)	-9	-2	3	8	12	18	24	30		
Space heating function									*SPF*	***3,15***
COP corrected for actual source/sink temp.	2,64	3,13	3,58	4,07	4,48					
Heat requirement [kWh]	1077	7075	7088	3210	1597	-	-	-	20046	**20158**
Electr.cons. heat pump [kWh]	408	2259	1981	788	357	-	-	-	5793	
Electr.cons. back-up [kWh]	112	-	-	-	-	-	-	-	112	**6393**
Aux.electr.cons (pump &controls)									448	
DHW function									*SPF*	***1,83***
COP corrected for actual source/sink temp.	3,09	3,21	3,31	3,41	3,50	3,64	3,69	3,69		
DHW heat output [kWh]	57	480	633	480	641	579	162	22	3053	**3434**
Electr.cons. heat pump [kWh]	18	150	191	141	183	159	44	6	892	
Electr.cons. back-up [kWh]	7	60	79	60	80	72	20	3	382	**1875**
Electr.cons. for storage losses [kWh]	9	74	96	72	95	84	23	3	456	
Aux.electr.cons (pump &controls)									145	
Combined operation									*Overall seasonal performance SPF*	***2,85***

10.4.4 Assessment of SPF -calculation methods

The calculation methods for determining the efficiencies of heat pump boiler systems as proposed in the prEN 15316-4-2 are comprehensive and incorporate a lot off aspects that are not covered in the EN 244 & EN 14511. As such this prEN represents a huge step forward concerning the energetic evaluation of heat pump boiler systems.

There are however some issues that are not fully covered by this prEN which may lead to differences in real life SPF for heat pump systems with an identical calculated SPF acc. prEN, operating under the same conditions.

<u>a. Real life system return temperature</u>

The system return temperature of a heat pump boiler system is varying and dependent on the amount of thermal energy that is actually transferred to the rooms (or buffers), on the type of pump used in the heat distribution system, and on the type of room temperature control system. This phenomenon is comparable to the situation as described for condensing boilers, where the varying return temperatures also influence the generator efficiency.

The COP-figure that is used as basis for the estimated COPs for each Bin however is determined on the basis of EN 14511, which doesn't incorporate the phenomenon of varying return temperatures. The EN 14511 part 2 describes the test conditions under which COP should be measured and for application rating conditions only specifies the source inlet temperature and the heat-sink outlet temperature (which is the system feed temperature). By selecting a favourable flow for both the source and sink system the average temperature over the condensor and evaporator can respectively be decreased and increased, resulting in favourable COP-figures (ΔT is maximised to 10°C). The prEN 15316-4-2 proposes a correction factor (table B1 in Annex B) to compensate for a difference in test- ΔT and system design ΔT. This correction factor will indeed compensate for favourable COP figures, but will not compensate for the differences between design- and real life system parameters, which can be considerable as we know from current practice in the condensing boiler market (Wolff, D., et al, April 2004). As a result the heat pump systems that in real life perform better on the issue of "*return temperature"*, are not discriminated with the calculation method of the prEN 15316-4-2.

b. Part load operation

Because the heat load of the house varies throughout the heating season, the requested heat output from the heat pump boiler system will also vary. The heat pump has to operate at part load to deliver a reduced output. A heat pump with fixed compressor realizes part-load operation through cycling between on and off status. Heat pumps with variable capacity, achieve part load operation by stepwise or continuously (through inverter) controlling the output of the heat pump. In all cases part-load operation reduces the COP. For on/off controlled systems the COP reduction will be bigger than for variable capacity heat pump. The prEN 15316-4-2 proposes to test part load efficiency according to the test procedure described in CEN/TS 14825 at 25%, 50% and 75% of the maximum output but at least at 50%. This TS 14825 prescribes for on/off controlled heat pumps a cycle of 30 minutes ON and 30 minutes OFF. If these figures are not available the prEN 15316-4-2 proposes to use default correction factors for part load operation for each Bin, based on the water content of the heat distribution system and on the load factor. These correction factors vary from 0,891 if the heat pump runs only 10% of calculated bin hours (load factor = 10%) to 0,999 of the heat pump runs 99% of the Bin-heating time (load factor is 99%) for radiator and floor heating systems (heating system capacities above 20 l/kW). For inverter controlled heat pumps the correction factors are similar for heating system capacities above 15 l/kW.

The cycling that occurs in real life operation however, not only depends on the water content of the heat emitters, but on the overall hydraulic design (including use of buffers) and type and accuracy of the control systems. These parameters determine how often a heat pump switches on and off and how long the runtime of the cycle is. The actual number- and duration of heat pump cycles in each Bin determine the part load efficiency in real life. When following the TS 14825 and the calculation method of the prEN 15316-4-2, the influence of intelligent control systems and hydraulic design upon the heat pump generator efficiency is not valued. As such heat pump systems that in real life perform better on the issue of "*part-load operation"* are not discriminated. Even if cycling of the heat pump is limited by means of an anti-cycling software, there can be considerable differences in real life and calculated part load operation.

c. Use of back-up system

There are several reasons for adding a (electric) back-up heater to a heat pump system. System reliability and the guarantee for sufficient heat output certainly aren't the last two reasons. Obviously, the way in which the back-up heater is controlled (when is it used at what power rating) is essential for assessing the SPF of the system. The control strategy for the back-up system can improve or ruin the SPF. Unfortunately the way the back-up heater is controlled in real life operation is not assessed within the SPF-calculation method. When the COPs are measured according EN14511 or EN255 the back-up heating system is not taken into account (electric heating element is often an accessory provided by option). Only in the calculation method of the prEN 15316-4-2 the share of back-up heating is estimated (calculated) on the basis of the operating mode of the back-up system, which can be *alternate*, *parallel* or *partly parallel*. For alternate operation all of the heat generated in the Bin(s) below the system balance point is considered to be produced by the back-up heater. In parallel operation the heat pump is considered to produce whatever is can produce at maximum load (at relative low source temperatures), and the additional heat that is necessary is produced by the back-up heater. In partly parallel mode the heat pump is operating parallel if the ambient dry bulb temperature is below balance point and above the cut-out temperature; it operates in alternate mode (back-up heater produces all requested heat) below the cut-out temperature.

Although this approach is quite logical from a theoretical and practical point of view, it does not assess the real life operation of the back-up system and its influence on the SPF. Two heat pump boiler can have identical SPF-values according to the prEN 15316-4-2 calculation method, and show huge differences in real life SPF because of a different control strategy for the electric back-up heater. Control strategies can for instance be designed to optimise the heating comfort (minimize consumer complaints) by switching on the electric back-up heaters when:

- a quick heat-up of the house is requested
- the emitter system unfortunately proves to be a little bit to small and the heat curve is adjusted
- the heat source is not delivering the calculated amount of heat (heat source is designed too small, thermal conductivity of soil is other than assumed, cooling of soil during heating season, polluted air-source heat exchanger, air-source heat pumps have reduced COPs when more heating is needed).
- DHW-use deviates from test use (in case of heat pump combi systems)

Another remark that can be made to the calculation methods for back-up heater energy use is the fact that in parallel mode the calculation method assumes system return temperatures that facilitate a large share of heat production by the heat pump, while in fact real life return temperatures are generally higher, especially when back-up heaters are used to increase the system feed temperature. This will reduce the share of the heat pump and improve the share of heat production by the back-up heater.

For the application of mono-energetic heat pump boiler systems it is essential to correctly value the input of the back-up system, especially in retrofit applications since this can determine the difference between systems with a real life SPF above or below 2,5 (above or below 100% on primary energy). Several studies in the USA (Fairy, P., e.a., June 2004) found that real live HSPF can show remarkable differences with the calculated HSPF, that to a large extend can be attributed to the use of a back-up heater .

d. Room temperature control system

The type of room temperature control system that is used, influences the system return temperature and the cycling behaviour of the heat pump and therefore has an influence on the SPF (see also item a. and b.). In the proposed prEN 15316-4-2 calculation methods the influence of room temperature control systems are not assessed.

e. Hydraulic balancing

For heat pump boiler systems a proper hydraulic balancing of the heat distribution system is even more critical than for condensing boiler systems. The calculation methods of the prEN 15316-4-2 assume a perfect hydraulically balanced system and does not incorporate current real life installation practice on this topic (see also item a. and b.)

f. Standing losses

Standing losses (radiation and convection losses of the generating appliances) are not clearly covered in the prEN 15316-4-2. According to this prEN 15316-4-2 heat pump appliances without storage are considered to have no standing losses.

If a storage is integrated in the heat pump appliance, the losses of the storage can be calculated based on the average storage and average ambient temperature. The losses of an external heating buffer of DHW storage is not covered in the prEN 15316-4-2 and needs to be valued separately.

According to EN 14511 however, the heating capacity of the heat pump (which determines the COP) is determined with the direct method at the water *or* brine heat exchanger, meaning that the volume flow of the heat transfer medium is measured together with the in- and outlet temperatures. The heating capacity can then be determined with the formula:

$$P_H = q * \rho * c_p * \Delta T$$

where

P_H	is the heat capacity [W]
q	is the volume flow rate [m^3/s]
ρ	is the density [kg/m^3]
c_p	is the specific heat at constant pressure [J/kg.K]
ΔT	is the difference between inlet and outlet temperatures [K]

Theoretically these measurements can be performed both at the source side (e.g. brine heat exchanger) and on the sink side (CH-water heat exchanger). It looks like the EN 14511 allows both options. If measurements are done at the source side, the standing losses during operation of the heat pump appliance are not included. If these measurements are done at the sink side, these standing losses are incorporated. Furthermore the test conditions for heat pumps installation indoors, prescribe an ambient temperature between 15 and 30°C for water-to-water and air-to-water heat pumps. At an ambient temperature of 15°C the standing losses of a heat pump boiler appliance (possibly including storage) can be considerable (several hundreds of kWh/a) and actually influence the COP. But at ambient temperatures of 30°C the standing losses of this same appliance are reduced to almost 0 kWh/a, leading to higher COP-values. Finally the standing losses that occur in off-mode (heat pump is off, but warm CH-water still circulates through the generator system) is not valued at all.

To assess the influence of real life installation practice and the other heating system components upon the SPF of heat pump boiler systems, further detailed monitoring of real life installations is necessary.

Fraunhofer-Institut für Solare Energiesysteme ISE (Germany) is going to start (probably this year) a four years field test for heat pumps. ISE is planning to measure 140 heat pumps installed in one-family houses. In cooperation with seven heat pump manufacturers and two utilities the researchers will investigate how efficient electric heat pumps can meet the heat requirements of low energy houses.

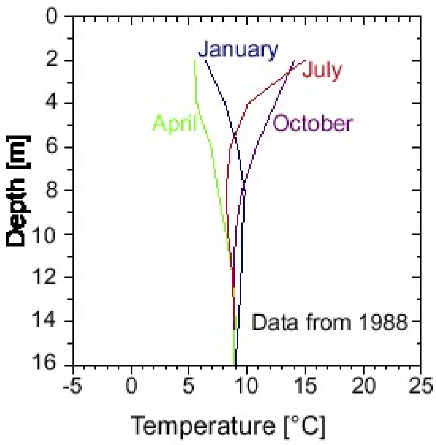
10.5 Technical Analysis Heat Pumps

10.5.1 Ground- and Water Source Heat Pumps

Introduction

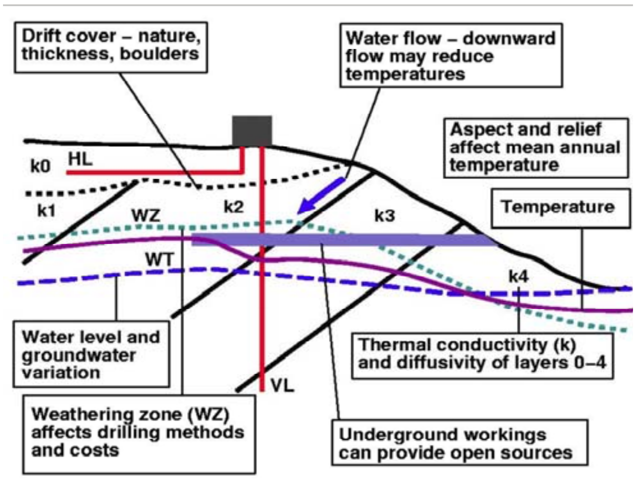
The underground in the first approximate 100 m is well suited for supply and storage of thermal energy. The climatic temperature change over the seasons is reduced to a steady temperature as 10 – 20 m depth and with further depth temperatures are increasing according the geothermal gradient (average of 3°C for each 100 m of depth)

Figure 10-24.
Underground temperature from a borehole south of Wetzlar, not influenced by the heat pump operation.



Ground Source Heat Pumps (a.k.a. Geothermal Heat Pumps) can be used to extract this thermal energy from the ground and transfer is to the requested temperature level. The feasibility of a site or location to function as potential geothermal source depends on a variety of aspects, to be described in a geo-report. Based on such information the preferred type of heat collector can be decided and whether the system can be an open loop (no separate heat transferring circuit) or closed loop (with a heat transfer circuit filled with brine or water with anti-freeze). The figure below indicates aspects to be covered by a geo-report

Figure 10-25.
Geo-report aspects



Ground source heat exchangers can be qualified by their shape and the type of heat carrier. The following collector parameters are distinguished:

- Open or closed systems
- Horizontal or vertical collectors

Closed systems

In closed systems, heat exchangers are located in the underground (either in a horizontal or vertical fashion) and a heat carrier medium (e.g. Brine[57] or a mixture of water and antifreeze) is circulated within the heat exchangers, transporting heat from the ground to the heat pump (or vice versa). The heat carrier is separated from the rock/soil and groundwater by the wall of the heat exchanger, making it a closed system. The thermal conductivity is an important parameter for the determination of size and type of ground source heat exchanger.

For vertical closed systems for instance, the design of borehole heat exchangers for small, individual applications can be done with tables, empirical values and guidelines (existing in Germany and Switzerland). A popular parameter to calculate the required length of borehole heat ex-changers is the specific heat extraction, expressed in Watt per meter borehole length. Typical values range between 40-70 W/m, dependent upon geology (thermal conductivity), annual hours of heat pump operation, number of neighbouring boreholes, etc. With the known capacity of the heat pump evaporator, the required length can easily be calculated:

$$Length\,[m] = \frac{HP\ evaporator\ \ capacity\ \ \ [W]}{specific\ heat\ extraction\ \ rate\ \ [W/m]}$$

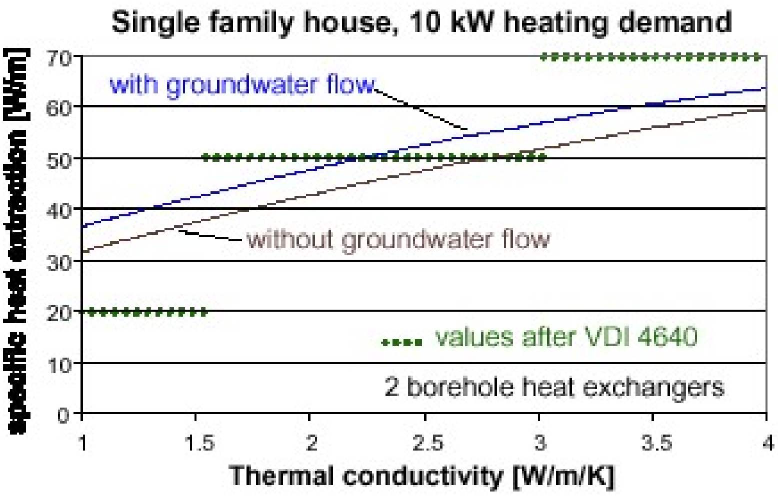
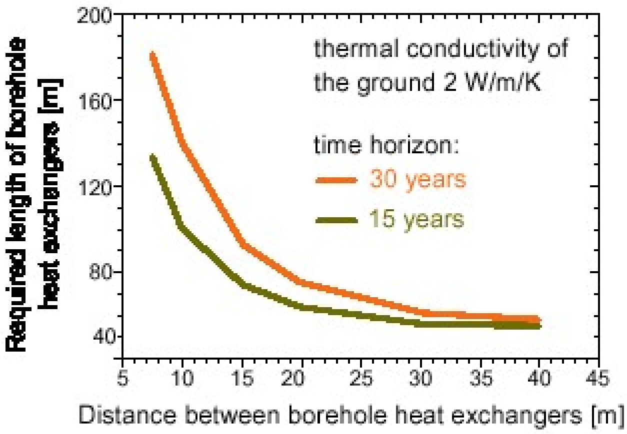


Figure 10-26.
Example of specific heat extraction values for a small ground source heat pump without DHW (heat pump run time 1800 h/a)
Source: Sanner, B., Description of ground source types for the heat pump.

For larger borehole heat exchangers (systems with heating and cooling, systems with more than 2000 h/a of heat pump operation) calculations have to be made to determine the required number and length of borehole heat exchangers. Computer programs are developed to facilitate these calculations and for real difficult cases, simulations with numerical computer models can be done. In case a large number of smaller ground source heat exchangers need to be realized, longer (deeper) heat exchangers may be necessary because of geothermal cooling. In other words, there is a relation between the length and the distance of the borehole heat exchangers.

[57] Brine is water saturated or nearly saturated with salt. Brine is a common fluid used in the transport of heat from place to place. It is used because the addition of salt to water reduces the freezing temperature of the solution and a relatively great efficiency in the transport can be obtained for the low cost of the material.

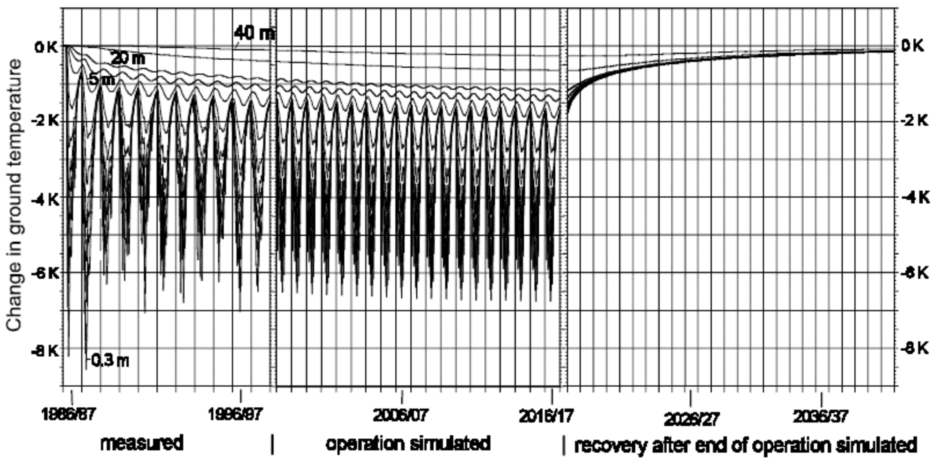
Figure 10-27.
Required borehole length in a field of 60 houses (7 kW heat load each) with two borehole heat exchangers for each house (no groundwater flow, no artificial recharge
Source: Sanner, B.,



The heat source for thermal recovery is solar heat (in the upper ground layers) and geothermal heat flux (in the lower parts), with some influence from flowing ground water or percolating water. The influence of groundwater however is in most cases not very big. The thermal conductivity of the ground is the main parameter.

During operation of the ground source heat pump, the temperature in the surrounding ground is decreased to generate a thermal gradient from the natural ground temperature to the heat carrier fluid inside the heat exchanger. The thermal conductivity determines the radius of the influenced zone and the temperature drop. Measurements and simulations can visualise the temperature development during operation as well as the thermal recovery after operation.

Figure 10-28.
Changes in ground temperature at various distances from the borehole heat exchanger (105 m deep), over many seasons, measured and simulated for a system in Elgg ZH, Switzerland (incl. simulated thermal recovery after end of operation).
Source: Sanner, B.,

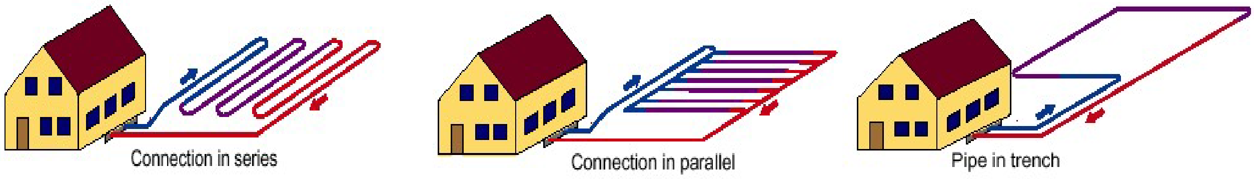


During operation (heating season) ground temperatures drop with 6 – 8°C at a distance of 0,3 m from the heat exchanger. Natural regeneration will bring this temperature drop back to 1 to 2°C. After 30 years of operation the temperature will have permanently dropped with approx. 2 degrees at 5 m distance from HE. It will take 30 years after the heat pump stopped operating, to reach the original temperature levels.

Collector types for closed systems

Horizontal soil collectors

Figure 10-29.
Different types of horizontal heat exchangers



The system easiest to install is the horizontal ground heat exchanger made of plastic pipes (polyethylene or polypropylene), also known as horizontal loop or ground heat collector. Depending on the shape of the horizontal collector the required surface (area) is determined. In Western and central Europe the pipes are laid in a relatively dense pattern, connected either in series or parallel, to reduce the size of the area that would be necessary. The double use of the heat exchanger (both extracting and adding heat) allows for more compact ground collectors.

To save surface area with ground heat collectors, some special ground heat exchangers have been developed. Exploiting a smaller area at the same volume, these collectors are best suited for heat pump systems that are applied for both heating and cooling, and systems that apply frequent regeneration (e.g. heat from exhaust ventilation air). For these compact horizontal heat exchangers active regeneration of the ground temperature is important.

Figure 10-30.
Different types of compact horizontal heat exchangers

The thermal recharge for all horizontal systems is mainly provided by the solar radiation to the earth's surface. It is important not to cover the surface above the ground heat collector, or to operate it as a heat store, if it has to be located e.g. under a building.

IVT has developed a new type of collector for soil heat which needs a plot of only 40m² to supply soil heat to a normal-sized house. The technology has been carefully tested and patented. There are many areas where the bedrock is deep and many plots are too small for conventional ground coils. The IVT compact collector allows more house owners to use soil heat! Because the collector is designed in the form of a coil, the maximum output can be extracted from a very small area. The collector is a part of a whole unit. It should always be combined with exhaust air heat, either from an IVT 495 TWIN or IVT Greenline heat pump with an exhaust air recovery unit. This means that the collector recharges using surplus heat when the house does not need to be heated, which significantly increases its efficiency.

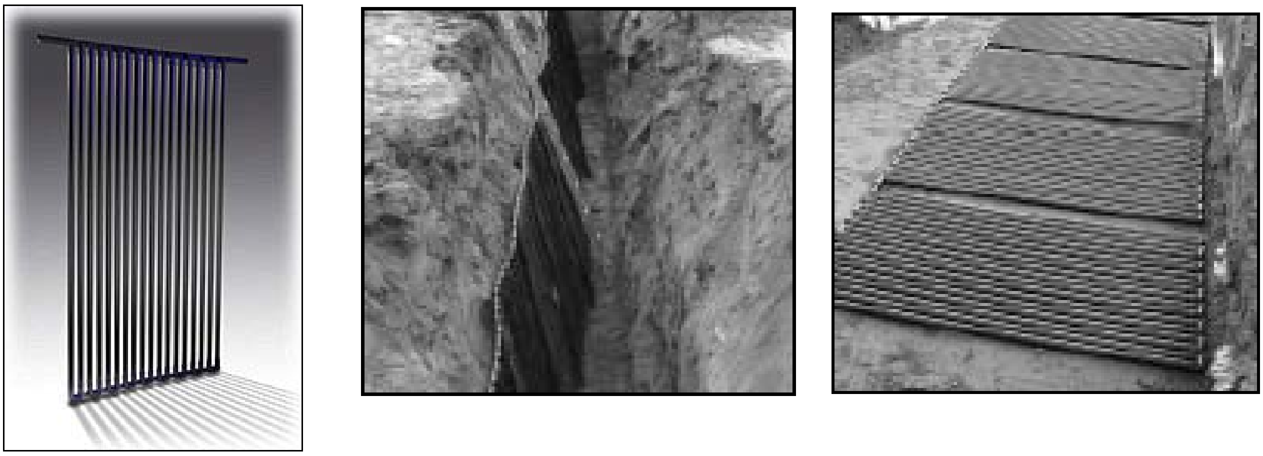


Figure 10-31
IVT compact collector

The compact collector is easy to install. It is two metres high and is buried one metre below ground level (vertically or horizontally). The sections can easily be joined at the top. The number of sections used depends on the heating requirements of the house. The collector is recharged with heat when the house does not need heating. The combination of the heat pump, exhaust air recovery unit and compact collector provides inexpensive heating and hot water and a healthy indoor environment.

Figure 10-32.
IVT compact collector

Direct expansion systems

A variation of the horizontal ground source heat pump is the direct expansion system. In this system, the working medium of the heat pump (refrigerant) is circulating directly through the ground heat collector pipes (in other words, the heat pump evaporator is extended into the ground). The advantage of this technology is the omission of one heat exchange process, and thus a possibility for better system efficiency. In France and Austria, direct expansion also has been coupled to direct condensation in the floor heating system. Direct expansion requires good knowledge of the refrigeration cycle, and is restricted to smaller units (Sanner, B., 2001).

Vertical ground collectors

As can be seen from measurements, the temperature below a certain depth („neutral zone", at ca. 15-20 m depth) remains constant over the year. This fact, and the need to install sufficient heat exchange capacity under a confined surface area, favours vertical ground heat exchangers (borehole heat exchangers). In a standard borehole heat exchanger, plastic pipes (polyethylene or polypropylene) are installed in boreholes, and the remaining room in the hole is filled (grouted) with a material that can be pumped. In Sweden, boreholes in hard, crystalline rock usually are kept open, and the groundwater serves for heat exchange between the pipes and the rock. If more than one borehole heat exchanger is required, the pipes should be connected in such a way that equal distribution of flow in the different channels is secured. Manifolds can be in or at the building, or the pipes can be connected in trenches in the field.

Several types of borehole heat exchangers have been used and tested. The basic concepts are:

- U-pipes, consisting of a pair of straight pipes, connected by a 180° turn at the bottom. One, two or even three of such U-pipes are installed in one hole. The advantage of U-pipe is low cost of the pipe material. This is probably why double U-pipe heat exchangers are the most frequently used borehole heat exchangers in Europe.
- Coaxial or concentric pipes, either constructed in a very simple way with two straight pipes of different diameter, or in more complex configurations (extrusion).

Figure 10-33.
Cross-sections of different types of borehole heat exchangers

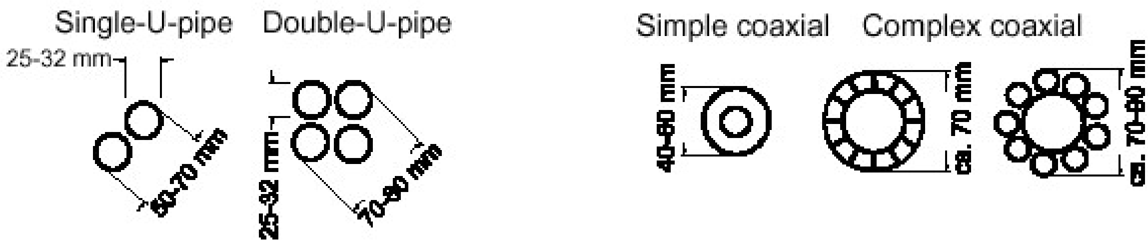
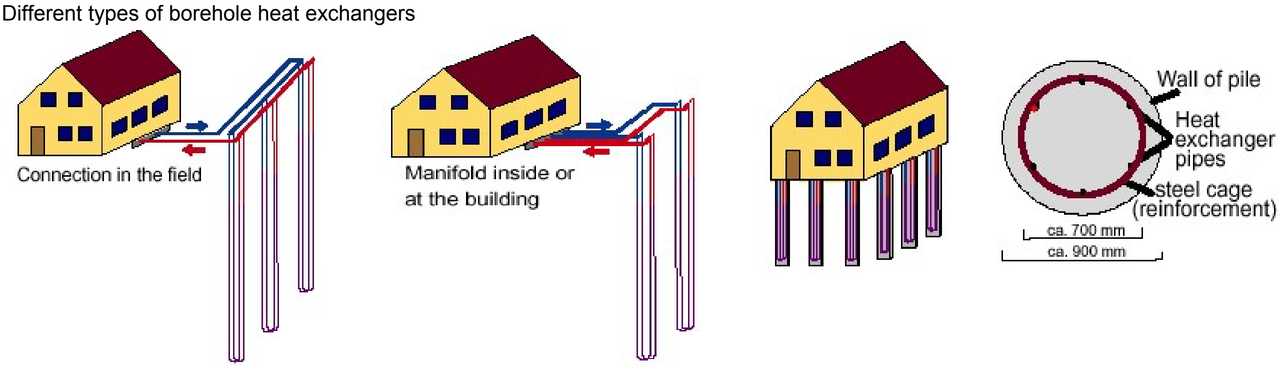


Figure 10-34.
Different types of borehole heat exchangers

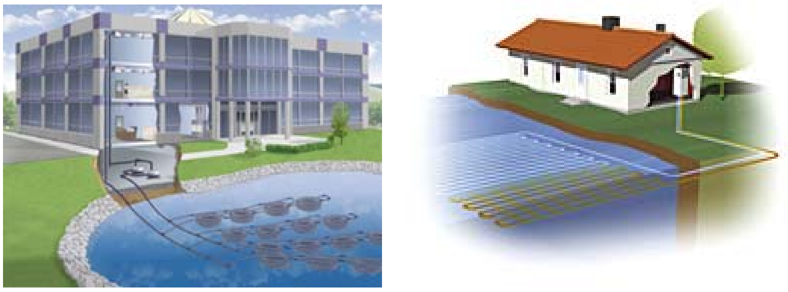


The borehole filling and the heat exchanger walls account for a further drop in temperature, which can be indicated as the thermal resistance of the borehole. Values for this parameter usually are on the order of 0,1 K/(W/m); for a heat extraction of 40 W/m, this means a temperature loss of 4 K inside the bore-hole. Thermally enhanced grouting (filling) materials have been developed to reduce these losses. A special case of vertical closed systems are „energy piles", i.e. foundation piles equipped with heat exchanger pipes. All kind of piles can be used (pre-fabricated or cats on site), and diameters may vary from 40 cm to over 1 m (figure 10-32).

<u>Surface water collectors</u>

Lakes, rivers and other large water courses are a perfect source for the extraction of heat. Water has a high thermal capacity and remains relatively warm throughout the year and for that reason contribute to the achievement of higher COPs. The simplest way to realize a surface water collector is to anchor the collector to the bed of the lake, pond or river. The only digging that needs to be done is dig a trench from the water to the house for the connecting hose.

Figure 10-35.
Surface water collector



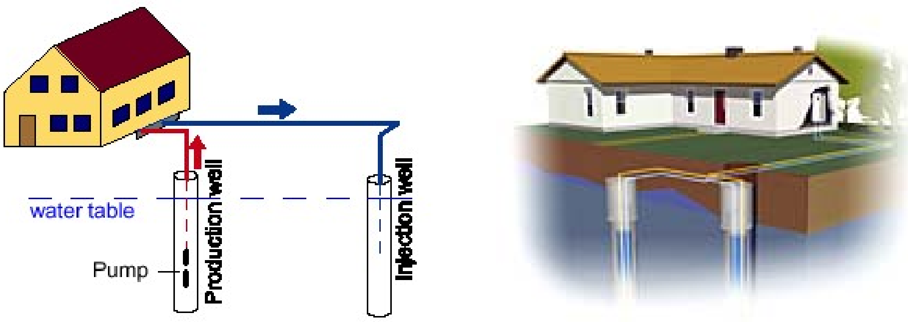
Open systems

In open systems the groundwater is used as a heat carrier and is brought directly to the heat pump. There is no barrier between rock/soil, ground water and the heat pump evaporator. With open systems a powerful heat source can be exploited at comparably low cost. Open systems in general are confined to sites with suitable aquifers and the main requirements are:

- Sufficient permeability, to allow the production of the desired amount of groundwater with little drop in ground water level.
- Good ground water chemistry (low iron content to avoid problems with scaling, clogging and corrosion)

Open systems tend to be used for larger installation.

Figure 10-36.
Ground water heat pump (doublette)



This type is characterised by the fact that the main heat carrier (ground water) flows freely in the underground. Water is pumped from a production well and after it has transferred its heat it is returned again and injected at a lower temperature in the injection well. Main technical part of open systems is the realization of the ground water wells that extract and inject water from and to the water bearing layers in the underground (aquifers).

In a standing column well, water is pumped from the bottom of the well to the heat pump system and, after leaving the return hose system, the water percolates through gravel in the annulus of the well. Standing column wells need a certain depth to provide enough power without freezing of the water, and thus most plants have boreholes several hundred meter deep. Examples are known from Europe (Switzerland and Germany) and from USA. With the expensive borehole, the technology is not suited to small installations.

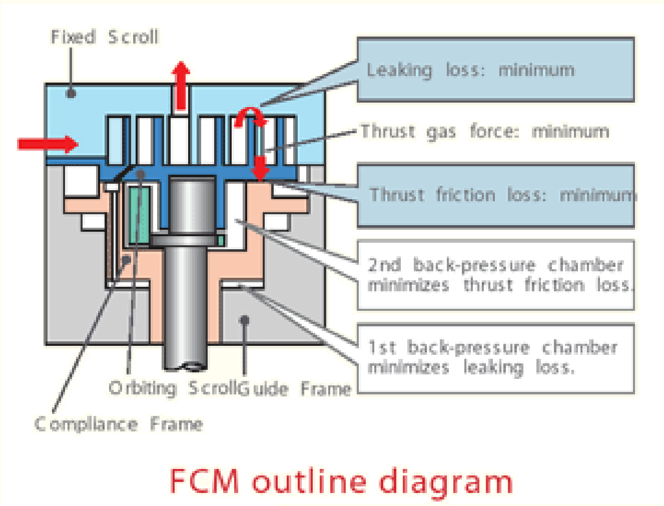
Products and components

IVT Greenline HT Plus

The Greenline HT Plus is IVT's top-of-the-range ground source heat pump. It uses the scroll compressor from Mitsubishi Electric, that is specially designed for heat pump operation. It has dual pressure chambers and uses the FCM technology (Frame Compliance Mechanism technology) for the better performance and reliability. The Greenline series can produce water at a temperature of 65 degrees without using the electric auxiliary heater. This means that higher system feed temperatures can be used and that all the heat for the DHW-tank can be suppl;ied by the heat pump unit. This higher system supply temperature means that the IVT Greenline HT Plus could be a solution for replacement projects.

In their aim for additional energy saving and improving the reliability of the compressor, Mitsubishi Electric developed the Frame Compliance Mechanism. It enhances the compressor efficiency and justifies the thrust force to the suitable level thus reducing the excessive energy and weariness. FCM can minimize gas leakage in scroll compression chamber, keep refrigerating capacity and reduce power loss by self-adjustment system of Orbiting Scroll position to pressure load and accuracy of Fixed Scroll profile. It is a big feature that FCM has not only a moveable Orbiting Scroll but also a moveable Frame unlike other manufacturer's one which is known so far.

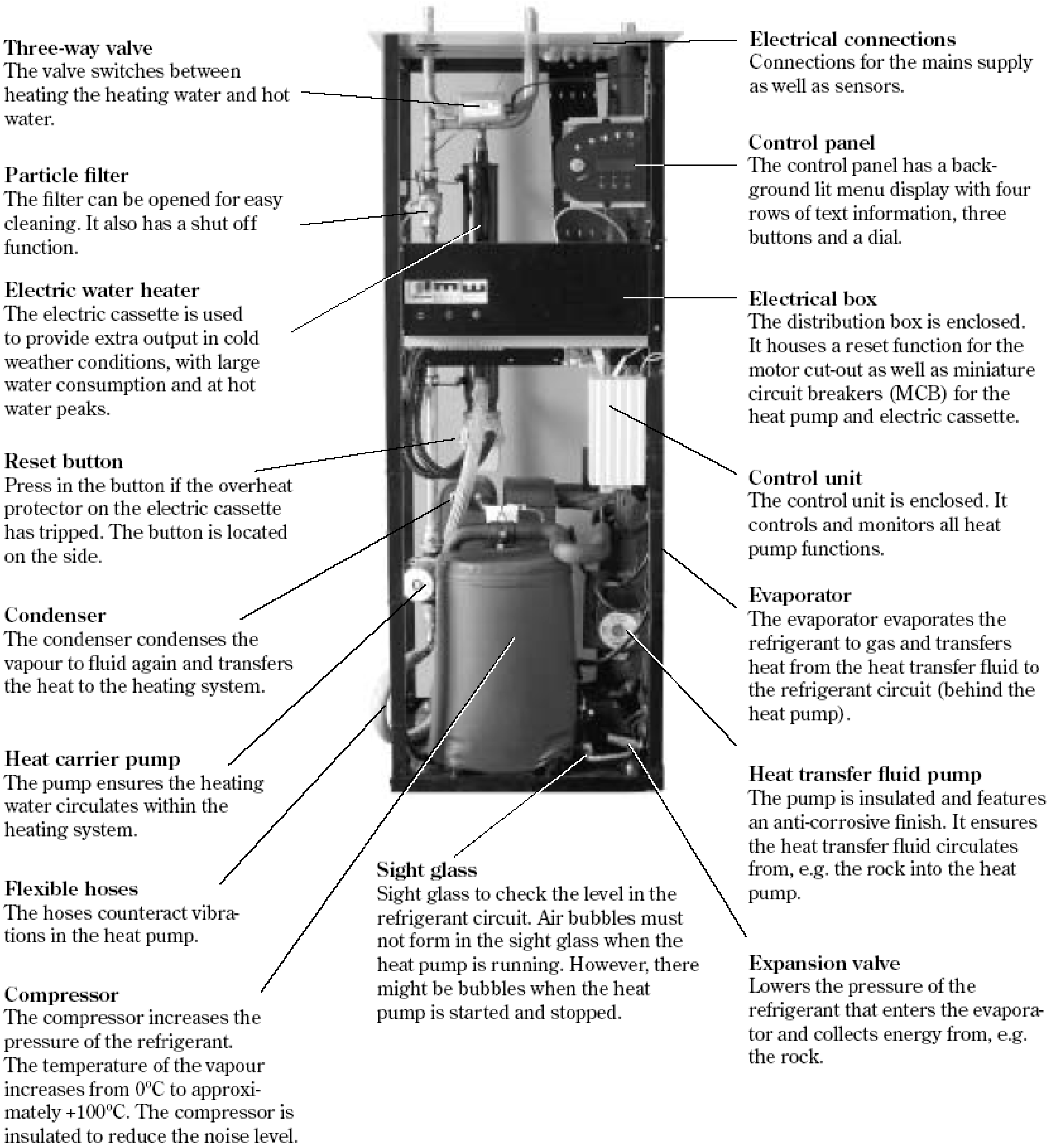
Figure 10-36. Frame Compliance Mechanism (FCM) by Mitsubishi Electric.



Advantage of the Frame Compliance Mechanism:

- High efficiency: resulting from the higher performance of the mechanism in term of adjusting the proper thrust force and eliminating energy losses.
- High reliability & durability: resulting from less friction force and automatic lubrication. The automatic lubrication creates from a different pressure inside the compressor that allows the lubrication oil to flow from high pressure chamber to the lower pressure one, without the necessity of oil stirrer (oil distribution equipment).
- Low noise & vibration: resulting from the frame compliance technology in adapting the pressure against the scroll internal chamber to the optimize level, decreasing clashes of internal parts, therefore, delivering a more quieter scroll compressor.
- Technology of the future inverter: with the proven Frame Compliance Mechanism that enabling the automatic lubrication, the advanced scroll is capable at running at various speed ranging from the very low to high revolution without any problem. It is a good solution for inverter and Multi Refrigerant System air conditioners.

Figure 10-36.
IVT Greenline HT Plus E, components of the heat pump



The IVT Greenline "E"- series are heat pumps without an integrated DHW-tank but with the necessary controls and valves for use in combination with an optional external storage cylinder. The Greenline "C"-series are heat pump combi products with an integrated double shelled hot water cylinder, that holds 165 litres of domestic hot water and another 60 litres of hot water storage for space heating purposes.

For the controls the weather dependent control system is used, with an option for correction on the basis of the temperature in a reference room. To secure a minimum flow over the heat exchanger of the heat pump (0,14 m/s) a pump with a fixed rootation speed is integrated in the products. Valves must be fully open and if there is a risk of TRVs closing and minimum flow can not be guaranteed, the heat pump should be connected, using a bypass and an external main pump.

Figure 10-37. Technical data Greenline HT Plus (according to IVR documentation

Model		6C/E	7C/E	9C/E	11C/E	14E	17E
Emitted/Supplied output at 0/35°C*	kW	5,9 /1,3	7,3 /1,6	9,1 /2	10,7 /2,2	14,4 /3,1	16,7 /3,7
COP at 0/35°C*	-	*4,54*	*4,56*	*4,55*	*4,86*	*4,64*	*4,51*
Emitted/Supplied output at 0/50°C*	kW	5,4 /1,7	6,9 /2,1	8,4 /2,6	10,1 /3,0	13,9 /4,2	16,2 /4,9
COP at 0/50°C*	-	*3,18*	*3,28*	*3,23*	*3,37*	*3,31*	*3,31*
Minimum flow heating medium	l/s	0,14	0,18	0,22	0,26	0,35	0,40
Nominal flow heating medium	l/s	0,20	0,25	0,31	0,37	0,50	0,57
Permitted ext. pressure drop heating medium at nominal flow	kPa	36	36	34	33	54	51
Highest outgoing heating medium temp	°C	65					
Electrical connection		400V, N3-Phase					
Additional electric beack-up heater	kW	3,0 / 6,0 / 9,0					
Recommended fuse size at 6 kW back-up heater	AT	16	16	20	20	20	25
Recommended fuse size at 9 kW back-up heater	AT	20	20	25	25	25	32
Compressor		Scroll					
Refrigerant R407C	kg	1,35	1,4	1,5	1,9	2,2	2,3
Sound level	dB	?	?	?	?	?	?
Weight (empty E / Empty C)	kg	146/346	152/353	155/365	170/388	190	195

** According to EN 255*

Figure 10-38.
Consumer product price, VAT included, installation & collector excluded, according to pricelist Konsumentverket 2006

Greenline E-type (single)	€	6126	6292	6402	6623		
Greenline C-type (combi)	€	6568	6679	6789	7230		
1 SEK = € 0,110394							

The costs for installation (drilling and digging excluded) are estimated at an additional 1100 to 2200 euro. Costs for drilling or digging largely depend on the site and the size of the collector. Costs are estimated between 1500 (simple digging works) and 4000 euro (difficult drilling in bedrock).

Carrier AB Sweden offers an identical product in Scandinavia under the name Carrier NewHeat 60

Stiebel Eltron WPC

The WPC is a heat pump with an integrated 175l. DHW-storage tank, suited for brine/water systems. Included in the product are the brine circulation pump, the circulation pump for the heating system and a 3-way valve. The WPC contains an integrated weather control system that controls the system return temperature on the basis of the outside temperature. Optional is the correction on system feed- or return temperature on the basis of the actual temperature in a reference room. The maximal achievable system feed temperature (heating and DHW) is 60°C. In case higher temperatures are requested an electric back-up heater or gas boiler needs to be installed.

The operating mode for the circulator pump for the heating system is two fold: either it is continuously switched on, or it is only switched on during heat pump operation; in off-mode the circulator pump is switched on and off (operating time is 5 minutes) in a cycle that is related to the outdoor temperature. With an outdoor temperature of 10°C switches on 2 times per hour for a five minute operation. When it is -10°C the pump switches on 6 times per hour.

Figure 10-38. Technical data Stiebel Eltron WPC (according to Stiebel Eltron documentation)

Model		**WPC 5**	**WPC7**	**WPC10**	**WPC13**
Emitted/Supplied output at 0/35°C*	kW	5,8 /1,35	7,8 /1,78	9,9 /2,2	13,4 /3,05
COP at 0/35°C*	-	*4,3*	*4,38*	*4,5*	*4,39*
Emitted/Supplied output at 0/50°C*	kW	5,5 /1,96	7,3 /2,51	9,5 /3,16	12,7/4,23
COP at 0/50°C*	-	*2,80*	*2,90*	*3,0*	*3,0*
Minimum flow heating medium	l/s	0,14	0,19	0,25	0,33
Permitted ext. pressure drop heating medium at nominal flow	kPa	52	4	38	2
Highest outgoing heating medium temp	°C	60			
Electrical connection		3/PE~ 400/50			
Additional electric back-up heater (max. power)	kW	8,8			
Starting current	A	<30	<30	<30	<30
Recommended fuse size	A	16	16	16	16
Input power (compressor and heat source pump) min/max	kW	2,0 / 2,9	2,3 / 3,7	2,9 / 4,5	3,5 / 5,9
Compressor		?			
Refrigerant R407C	kg	1,35	1,4	1,5	1,9
Sound levels (at 1 m)	dBA	35	36	40	42
Weight (empty/full)	kg	275/437	285/447	295/457	305/467

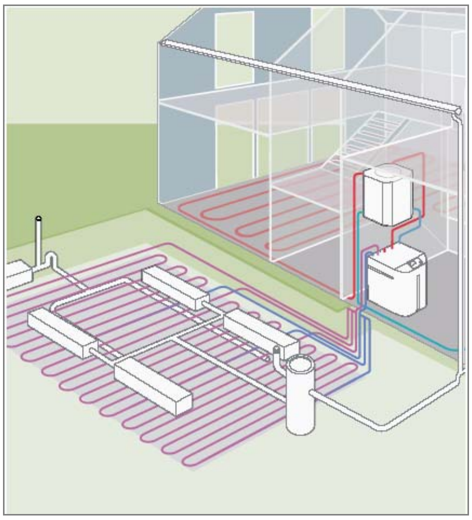
** According to EN 255*

Figure 10-39.
Consumer product price, VAT included, installation & collector excluded, according to pricelist Konsumentverket 2006

Stiebel Eltron WPC (combi)	€	6338	6756	7354	?
1 SEK = € 0,110394					

The Stiebel-Eltron AquaGeo-Collectors are an option when there is not enough surface outside for an adequately sized horizontal ground collector. Prerequisite for an AquaGeo-Collector is a rainwater drainage system with an interim storage. The amount of energy coming from the soil collector can be increased by applying a water tight sheet under the collector and humidify the soil above this sheet artificially with the rainwater. The moisture increases the heat capacity of the soil and as a result the collector can remain smaller. If there is a surplus of rainwater it must be drained through the overflow of the interim storage.

Figure 10-40.
AquaGeo Collector (source: Stiebel Eltron)



Viessmann Vitocal 350 / 343

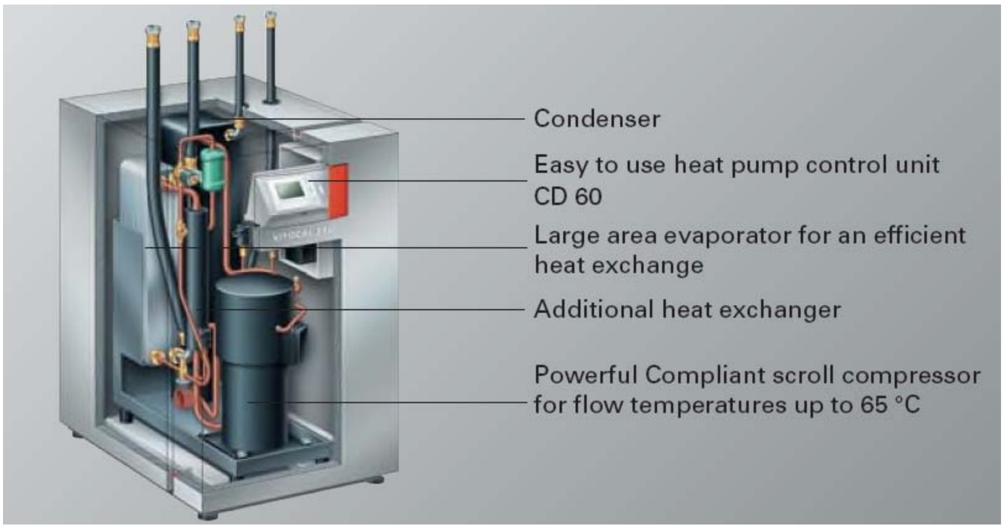
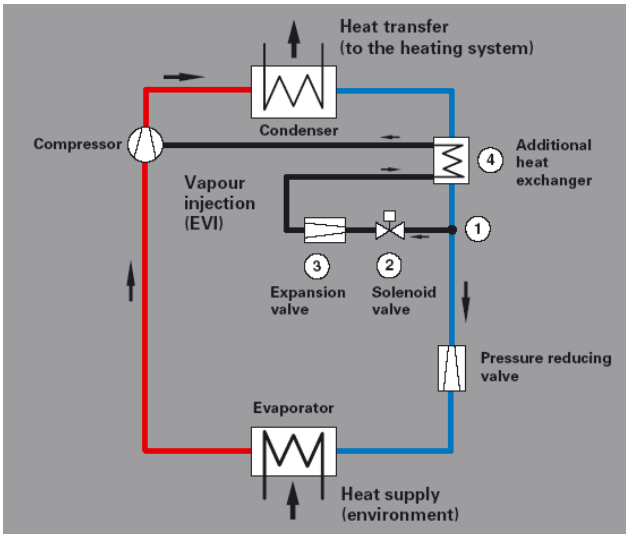
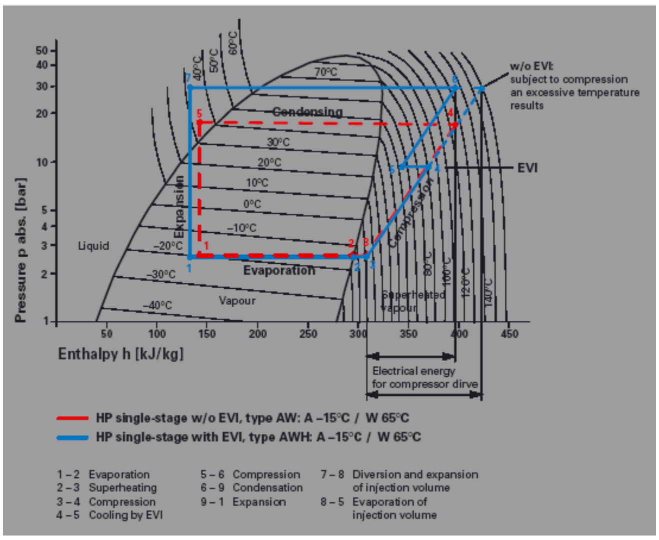


Figure 10-41.
Vitocal 350
(source: Viessmann)

The Viessmann Vitocal 350 is an electric compressor driven heat pump also capable of delivering system feed temperatures up to 65°C and capable of delivering cooling on hot summer days. The high system feed temperatures are achieved through a modified single stage refrigeration circuit also referred to as the “Enhanced Vapour Injection” (EVI) cycle. A small amount of refrigerant - upon demand- is diverted via a solenoid valve. This liquid refrigerant, which is under high pressure, is depressurized in the expansion valve to the injection pressure and vaporised in an additional heat exchanger. The vaporised refrigerant is transferred from the additional heat exchanger to the compressor, where it is directly injected into the compression process. Only when there is a high heat demand (large temperature lift requested) the EVI-cycle comes into play.

Figure 10-41.
Enhanced Vapour Injection cycle (source: Viessmann)





The cooling allows the refrigerant to be compressed, to a greater extent, without exceeding the permissible temperature level inside the compressor. This achieves higher temperatures at the entry point into the vapour region. At the same time, the injected refrigerant increases the flow rate, which leads to a higher heat transfer to the heating system (this process also is particularly effective with air/water heat pumps of this design, enabling a flow temperature of 65°C being achieved even at -15°C). The electrical energy required by the compressor drive is lower for the EVI-process than for a theoretically comparable compressor without vapour injection.

Viessmann Vitocal 343

The Vitocal 343 is a heating tower, containing a brine/water heat pump, a 250 l. DHW storage cylinder, circulation pumps for the brine, heating and optional solar heating circuit, plus all hydraulic connections and a control unit. The heat pump achieves system feed temperatures up to 60°C. For higher feed or DHW temperatures an integrated multi stage electric heater rod can boost the water temperature to 70°C.

The menu-guided CD 70 control unit offers all functions for a weather-compensated heating operation and the integral solar heating circuit. With special accessories the Vitocal 343 can also offer a cooling function.

Figure 10-42. Technical data Vitocal 350 and 343 (according to Viessmann documentation)

		Vitocal 350 b/w		Vitocal 343	
Model		**110**	**113**	**6,1**	**7,7**
Emitted/Supplied output at B0/W35°C*	kW	11,0 /2,55	16,2 /3,75	6,1 /1,42	7,7/1,79
COP at 0/35°C*	-	*4,31*	*4,32*	*4,3*	*4,3*
Emitted/Supplied output at B2/W65°C*	kW	13,2 /5,10	17,7 /7,10	5,9 /2,1	7,5 /2,8
COP at 2/65°C for Vitocal 350 | COP at B2/W55 for Vitocal 343*	-	*2,59*	*2,49*	*2,8*	*2,7*
Minimum flow heating medium	l/s	0,29	0,37	0,22	0,22
Permitted ext. pressure drop heating medium at nominal flow	kPa	?	?	32	32
Highest outgoing heating medium temp	°C	65		60	
Electrical connection		3/N/PE~ 400/50			
Additional electric back-up heater (max. power)	kW	-		2/4/6	
Starting current (with limiter)	A	23	26	25	14
Recommended fuse size	A	3x20	3x20	3x16	3x16
Input power circulator collector step1/2/3 | circulator heating system step 1/2/3	W			62/92/132 | 45/75/89	
Compressor		Scroll with injection		Scroll	
Refrigerant	kg	R407C 2,9	R407C 3,2	R410A 1,0	R410A 1,2
Sound levels (at 1 m)	dBA	?	?	?	?
Weight (empty)	kg	145	165	270	280

** According to EN 255*

Figure 10-43.
Consumer product price, VAT included, installation & collector excluded, according to PreisRoboter 2006 (www.preisroboter.de)

Vitocal	€	7434 to 9950	6990 - 7450

On the same website borehole drillings are offered for around € 3500 and horizontal collectors for approximately € 1500.

10.5.2 Air Source Heat Pumps

Air as heat source

Outside ambient air

Air is an omnipresent and everywhere available heat source. This clearly is an advantage for the application of air source heat pumps. Main drawbacks of this heat source however are twofold:

- Air has a low heat capacity (1000 [J/m³K] versus 4180 * 10³ [J/m³K] for water)
- Air temperature drops when more heating is needed (heating season).

As a result large flows of air are needed and the achievable COPs and SFPs (for heating purposes) are lower than for ground source heat pumps. The energy performance of air source heat exchangers very much depend on outside temperatures during heating season. Current technical developments are focussed on improving the COP of air/water heat pumps are lower air temperatures.

Having said that, air-source heat pumps nevertheless have the capability to achieve a seasonal performance based (on primary energy) that can outrank those of condensing boilers, provided they are correctly applied (milder climates), designed to work in cooperation with low temperature heating systems and operate with minimal or no usage of an electric back-up heater.

Exhaust air

Exhaust air heat pumps that are applied in the residential sector use the ventilation exhaust air as heat source. Residential exhaust air heat pumps can therefore only be used in dwellings that use a ventilation system based on central mechanical extraction. The temperature of the used indoor air throughout the year generally is around 20 – 22°C, which is advantages for the COP of the heat pump. The amount of air however is limited by the indoor air quality level. In an average house of a 100 m² an average ventilation rate of 150 m³/h (ach = 0,5) is more than enough the secure the requested indoor air quality levels (< 1000 ppm CO_2). The energy content of this air at a ΔT of 10°C is approximately 420 watts. During a whole year this amounts up to approximately 3700 kWh which can be used for DHW and space heating purposes.

Products and components

Heatking air source heat pump

The BWarm air source heat pump can be delivered in a 6 and 8 kW version and incorporates a fan, an air/refrigerant heat exchanger (evaporator), a compressor, a condenser, control unit and circulator pump. It can be mounted directly on e.g. an outside wall and connected to the central heating pipe work and it is sold as a plug and play appliance. Minimum ambient temperatures: -15°C.

Figure 10-44.
Air-source heat pumps BWarm8000 and BWarm6000 from Heat King Ltd UK

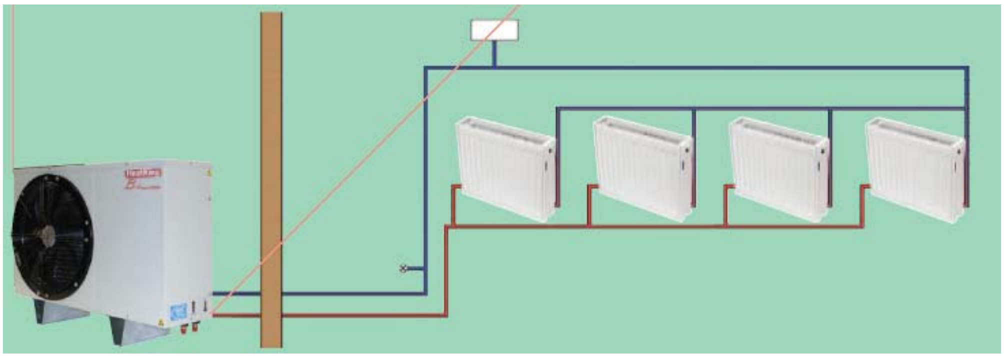


Figure 10-45.
Graphic presentation of the ASHP cycle (source: Heatking Ltd.)

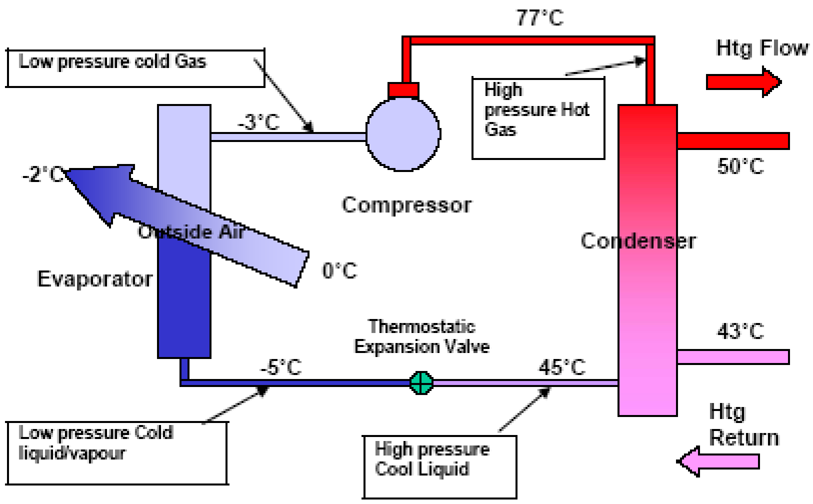


Figure 10-46.
Performance curves BWarm 6000 at 35°C sink temperature (Source: http://www.heatking.co.uk/WhyBwarm.htm)

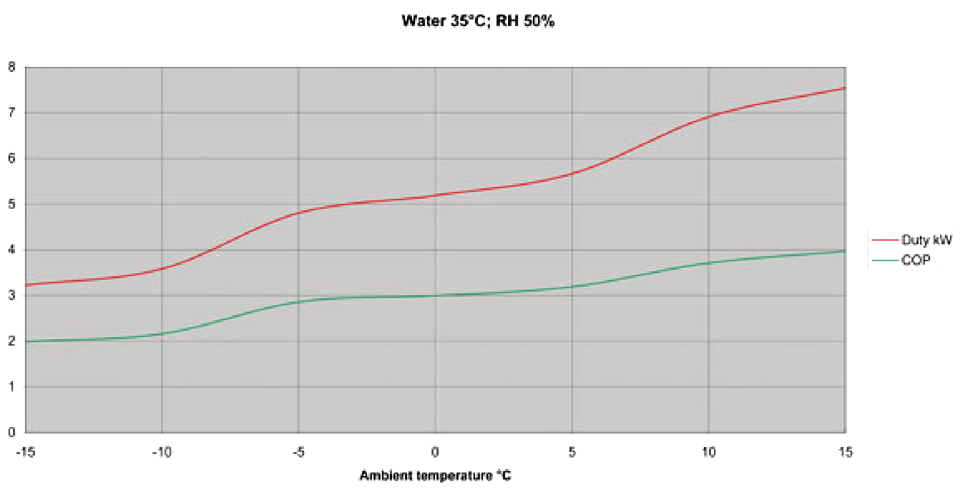
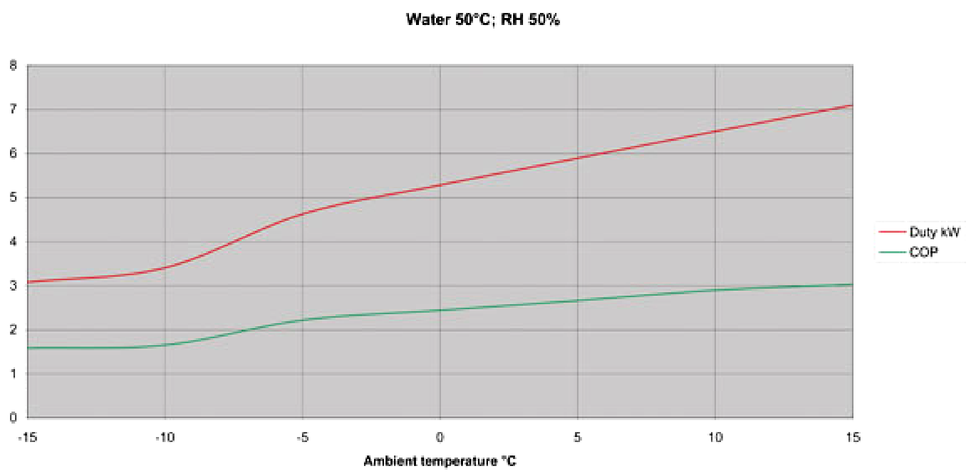


Figure 10-47.
Performance curves BWarm 6000 at 50°C sink temperature (Source: http://www.heatking.co.uk/WhyBwarm.htm)

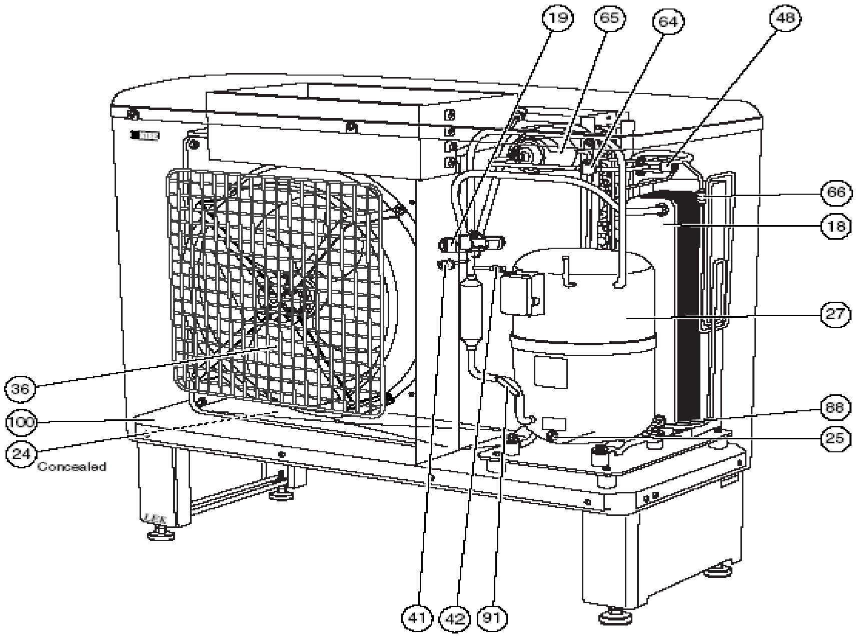


At an outside air temperature of -5°C and a 50°C system feed temperature the COP will be around 2,2 with a heat output of 4,7 kW. Correcting for pump energy (60 W) and cycling losses and provided no electric back-up energy is used, the COP will drop to around 2 (80% on primary energy). At an outside air temperature of 5°C and a 35°C system feed temperature the COP will be around 3,2 with a heat output of 5,6 kW. Correcting for pump energy (60 W) and cycling losses and provided no electric back-up energy is used, the COP will drop to around 3 (120% on primary energy).

Nibe Fighter 2005

Fighter 2005 is a air/water heat pump specifically developed for the climate of Northern Europe. Fighter 2005 is equipped with automatic two-stage capacity control for the ventilator and an electronic control for the heat pump operation. The product has been specifically developed for operation with radiator heating systems and can be controlled either on the basis of a the system return temperature (which is preset by a weather dependent control system) or on the basis of a room thermostat. In combination with the SMO 10 control unit the Fighter 2005 can be used in combination with a back-up heater and DHW-storage tank.

Figure 10-48.
Nibe Fighter 2005



18. Condenser
19. 4-way valve
24. Drip tray heater
25. Compressor heater
27. Compressor
36. Fan
41. High pressure pressostat
42. Service connection (high pressure)
48. Expansion valve
64. Sight glass
65. Drying filter
66. Non-return valve
88. Temperature sensor, fluid pipe
91. Temperature sensor, discharge

Technical Data FIGHTER 2005

	Single Phase FIGHTER 2005-8	Single Phase FIGHTER 2005-11	FIGHTER 2005-8	FIGHTER 2005-10	FIGHTER 2005-14
Heat capacity/receptivity* at A2/W35 °C ** (kW)	6,9/1,9	9,3/2,6	6,3/1,8	8,3/2,3	11,9/3,5
Heat capacity/receptivity* at A7/W35 °C ** (kW)	7,5/1,9	10,6/2,6	7,9/1,8	10,5/2,4	13,8/3,7
Heat capacity/receptivity* at A-7/W45 °C ** (kW)	5,3/2,2	6,9/2,8	3,8/1,7	5,8/2,2	7,8/3,2
Heat capacity/receptivity* at A0/W45 °C ** (kW)	6,3/2,2	8,5/3,0	5,6/1,9	7,4/2,5	10,5/3,7
Heat capacity/receptivity* at A7/W45 °C ** (kW)	7,7/2,3	10,5/3,1	7,4/2,1	9,7/2,7	13,8/4,1
Heat capacity/receptivity* at A-7/W50 °C ** (kW)	5,1/2,3	6,9/3,1	3,5/1,7	5,2/2,2	7,2/3,2
Heat capacity/receptivity* at A2/W50 °C ** (kW)	6,8/2,5	9,1/3,3	5,9/2,1	7,4/2,6	10,9/3,9
Heat capacity/receptivity* at A7/W50 °C ** (kW)	7,5/2,4	10,4/3,4	7,0/2,2	9,3/2,8	13,3/4,3
Heat capacity/receptivity* at A15/W50 °C ** (kW)	9,4/2,5	12,6/3,5	9,9/2,0	11,7/3,1	16,3/4,7
Start current (A)	24	33	24	33	26
Motor protection setting (A)	15	21,5	15	21,5	11
Soft start relays	are standard				
Operating voltage	230V + N + PE 50 Hz		3x400V + N + PE 50 Hz		
Compressor	Scroll compressor		Piston compressor		
Nominal heating circuit flow (l/s)	0,17	0,24	0,17	0,24	0,33
Internal pressure drop at nominal heating circuit flow (kPa)	1,1	2,0	1,1	2,0	2,4
Air flow (m3/h)	1320/1750	1320/1750	1320/1750	1320/1750	2250/3050
Nominal ventilator output (W)	155/185	155/185	155/185	155/185	175/190
Protection class	IP 24				
Maximum exit temperature to the heating circuit (°C)	58				
Refrigerant amount (R407C) (kg)	2,1	2,1	2,1	2,1	2,4

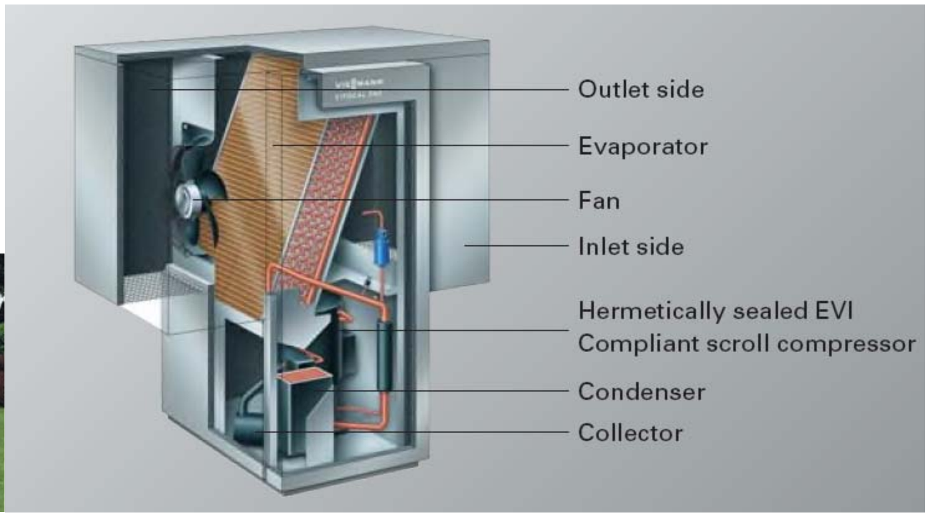
Viessmann Vitocal 350 Typ AWI/AWO

The additional vapour injection in the compression process (EVI cycle) enables flow temperatures up to 65°C. That makes it an alternative for boilers in older heating systems with radiators. The heat pump can be installed outside (AWO-type) - saving considerable space inside the house — and inside the house (AWI-type). When the heat pump is installed outside, heat needs to be “transported” to the interior of the house. For this, specially matched and thermally insulated pipe systems for routing underground are offered.

Subject to system version, the higher flow temperature enables DHW temperatures up to 58°C inside the DHW cylinder. This allows the Vitocal 350 to deliver particularly high DHW convenience.

Figure 10-49.
Vitocal 350 AWO

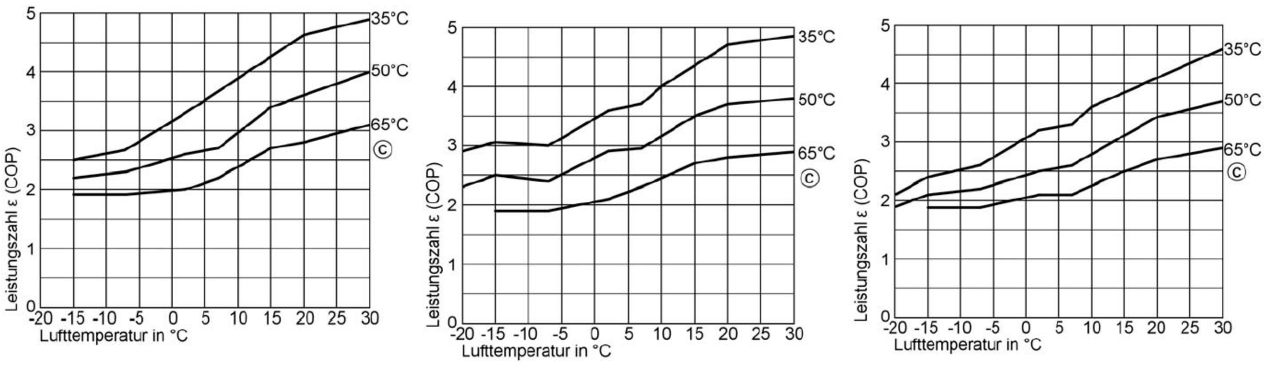
Table: Performance data measures according to EN 255 at A2/W35°C.



Technische Angaben

Vitocal 350	Typ	AWI 110	AWI 114	AWI 120	AWO 110	AWO 114	AWO 120
Leistungsdaten*1							
Nenn-Wärmeleistung	kW	10,6	14,8	18,5	10,6	14,8	18,5
Kälteleistung	kW	7,4	10,7	12,7	7,4	10,7	12,7
Elektr. Leistungsaufnahme	kW	3,2	4,1	5,8	3,2	4,1	5,8
Leistungszahl ε (COP)		3,3	3,6	3,2	3,3	3,6	3,2
Wärmegewinnung							
Ventilatorleistung	W	250	350	480	250	350	480
Luftmenge	m³/h	3500	4000	4500	3500	4000	4500
max. zul. Druckverlust Zu- und Abluftkanal	Pa	30					
Lufttemperatur min.	°C	–20					
Lufttemperatur max.	°C	35					
Abtauleistung	kW	3,3	4,2	6,2	3,3	4,2	6,2
Anteil Abtauzeit/Laufzeit	%	7 bis 17					
Heizwasser (sekundär)							
Inhalt	Liter	3,3	3,8	4,0	3,3	3,8	4,0
min. Durchsatz *2	Liter/h	1150	1200	1800	1150	1200	1800
Durchflusswiderstand	mbar	20	17	30	20	17	30
max. Vorlauftemperatur	°C (A−20)	55					
	°C (A−5)	65					
Elektrische Werte							
Wärmepumpe							
Nennspannung		3/N/PE ~ 400 V/50 Hz					
Nennstrom (max.)	A	10,0	14,0	18,3	10,0	14,0	18,3
Anlaufstrom*3	A	23,0	26,0	30,0	23,0	26,0	30,0
Anlaufstrom (bei blockiertem Rotor)	A	64,0	70,5	99,0	64,0	70,5	99,0
Absicherung*4	A	3 x 20	3 x 20	3 x 25	3 x 20	3 x 20	3 x 25
Absicherung Ventilator		T 6,3 A H					
Schutzart		IP 21			IP 24		
Nennspannung Steuerstromkreis		230 V~ 50 Hz					
Absicherung Steuerstromkreis		T 6,3 A H					
Kältekreis							
Arbeitsmittel		R 407 C					
Füllmenge	kg	4,0	4,3	7,2	4,0	4,3	7,2
Verdichter	Typ	Scroll Vollhermetik mit Einspritzung					

Figure 10-49.
Performance Vitocal 350 AWO/AWI, for types 110, 114 and 120 (from left to right.



Consumer product price for the Vitocal 350 AWO 110 (VAT included, installation & collector excluded, DHW tank excluded) according to pricelist Konsumentverket 2006 is € 8610.

10.6 Technical developments

10.6.1 Heat pump controls[58]

Employing an effective control strategy can considerably increase the seasonal performance of a heat pump system. Several approaches can be used to implement an enhanced control scheme, including capacity control techniques, improved expansion valve control and optimised secondary side control (control of secondary fans and pumps on source- and sink side of the heat pump sytem).

Capacity control

Most heat pump systems are optimised to operate under design loading condition, Due to changes in ambient conditions the actual load is predominantly lower than the peak load. Therefore, some means of system capacity regulation must be employed. Forms of capacity control are for instance compressor cycling, hot gas bypass, cylinder unloading, suction gas throttling, and variable speed compressor.

Compressor cycling is the simplest and most common method of capacity control. The compressor cycles on and off in response to changes in the load. During periods of high loading the compressor operates longer and during periods of low loading the converse occurs. The compressor is cycled on by a control loop when temperature in a conditioned space (of system return temperature) decreases a presser value.

Hot gas bypass is a technique by which the capacity is varied, by artificially loading the compressor as the system load reduces. The hot gas bypass valve allows some of the high pressure hot gas from the discharge line to circumvent the condenser and expansion valve and directly enter the evaporator. This reduces the capacity of the evaporator, without reducing the compressor power consumption.

Capacity control in a system employing multiple cylinders can be achieved through cylinder unloading. This is implemented practically by holding open the suction valve of one or more of the cylinders during compression, or by diverting some of the gas from

[58] Deliverable D5 from project "Sustainable Heat and Energy Research for Heat Pump Applications" called: Control Algorithm Suite for HC, NH3 and CO2 components, University College Dublin, June 2005.

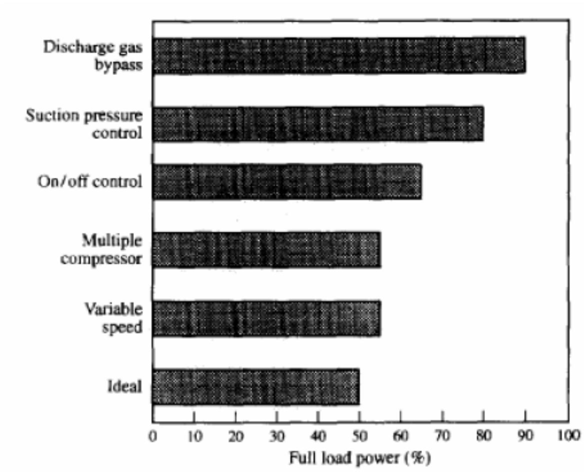
one or more of the cylinders back into the suction line. The cylinders are arranged in parallel, so the more cylinders that are unloaded the lower the mass flow rate of refrigerant through the evaporator, the lower the capacity. Cylinder unloading can also be used at start-up to reduce starting current and power requirements.

Suction Gas Throttling (throttling the gas at the inlet of the compressor) can control the capacity of a vapour compression system. The reduction in pressure of the suction gas causes a reduction in mass flow rate of the refrigerant through the system, and hence reduces the system capacity.

With variable speed control, the capacity of the system is matched to the load by regulating the speed of the compressor motor. The modulation of the mass flow rate through the system is then used to control the system capacity. There are a number of different ways of realising variable speed compression to control the capacity of the refrigeration system. They can be subdivided into two categories. In the first category the systems utilise fixed speed motors, and a speed control device is inserted between the motors and the loads. The second group comprises systems that directly couple the load to a variable speed motor. In heat pump applications either stepwise or infinitely variable control of the motor speed are possible. Stepwise speed control is achieved by using multi-pole electric motors. Step less speed control is achieved by electronic variable speed drives (VSD's), consisting of a rectifier and an inverter. VSD's can broadly be classified into three categories: six step voltage inverter (VSI), six step current inverter (CSI) and pulse width modulated source inverter (PWM). The PWM inverter is the most commonly used in the industry.

A lot of the work carried out on recent years has concentrated on variable speed systems. IN practice variable speed capacity control is the only technique that produced higher COP's at reduced capacity

Figure 10-50.
Comparison of capacity control techniques at half load (Qereshi and Tassou, 1995)

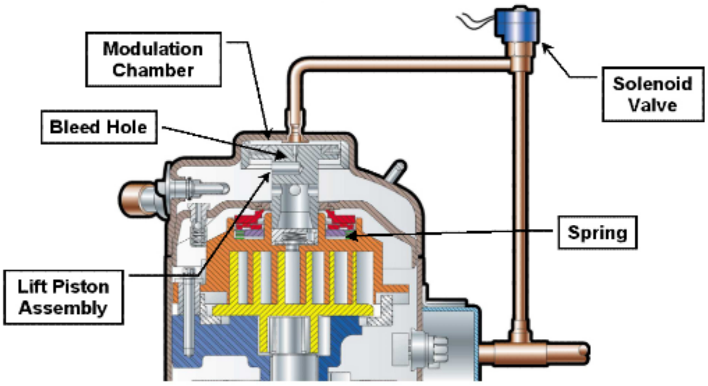


<u>Example of a PWM expansion control</u>

The Copeland digital scroll compressor is a fine example of a pulse width modulated expansion valve. The beauty of digital scroll compressor technology is its inherent simplicity. The standard scroll has a unique feature called axial compliance. This allows the fixed scroll to move in the axial direction, by very small amounts, to ensure that the fixed and orbiting scrolls are always loaded together with the optimal force. This optimal force holding the 2 scrolls together at all operating conditions ensure the high efficiency of scrolls. The Digital Scroll operation from Copeland (Emerson) builds on this principle.

The Digital Scroll operates in two stages - the "loaded state", when the solenoid valve is normally closed and "unloaded state", when the solenoid valve is open. During the loaded state the compressor operates like a standard scroll and delivers full capacity and mass flow. However, during the unloaded state, there is no capacity and no mass flow through the compressor. At this stage, let us introduce the concept of a cycle time. A cycle time consists of a "Loaded State" time and "Unloaded State" time. The duration of these 2-time segments determine the capacity modulation of the compressor. Example: In a 20 seconds cycle time, if the loaded state time is 10 seconds and the unloaded state time is 10 seconds, the compressor modulation is (10 seconds x 100% + 10 seconds x 0%)/ 20 = 50% (Figure 3). If for the same cycle time, the loaded state time is 15 seconds and the unloaded state time is 5 seconds, the compressor modulation is 75%. The capacity is a time averaged summation of the loaded state and unloaded state. By varying the loaded state time and unloaded state time, any capacity (10%-100%) can be delivered by the compressor.

Figure 10-51.
Source: Digital Scroll from Emerson Technologies)



Digital Scroll Compressor

A piston is fixed to the top scroll to ensure that when the piston moves up, the top scroll also moves up. There is a modulation chamber at the top of the piston that is connected to the discharge pressure through a bleed hole of diameter 0.6 mm. An external solenoid valve connects the modulation chamber with the suction side pressure. When the solenoid valve is in the normally closed position, the pressure on either side of the piston is discharge pressure and a spring force ensures that the two scrolls are loaded together. When the solenoid valve is energized, the discharge gas in the modulation chamber is relieved to the low pressure. This causes the piston to move up and consequently the top scroll also moves up. This action separates the scrolls and results in no mass flow through the scrolls. De-energizing the external solenoid valve again loads the compressor fully and the compression is resumed. It should be noted that the movement of the top scroll is very small - 1.0 mm and consequently the amount of high-pressure gas that is bled from the high side to the low side is very little.

10.6.2 Other Heat Pump Principles

Gas absorption heat pumps (GAHP).

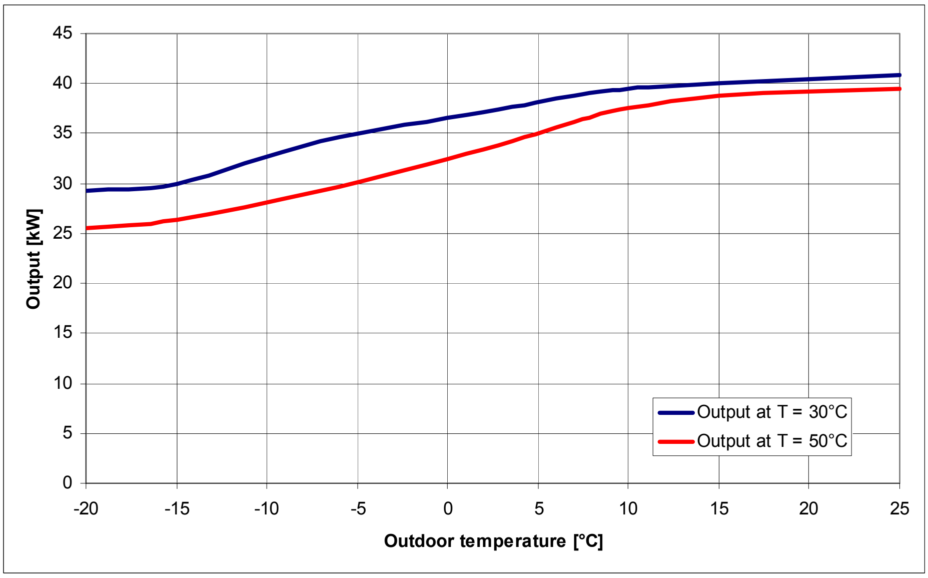
Small sorption heat pumps for individual retrofit applications are still in their infancy, but they could become an important heat pump technology, because the gas fired absorption principle has some important advantages over the electric compressions heat pump principle. Like compression heat pumps they can be used in combination with the different type of heat sources namely air, water and ground.

For *air* source GAHP's the mean advantages of gas fires absorption principle is the fact that the unit can operate at outside air temperatures of -20°C and still have a PER of around 1 (ad sink temperatures of 60°C). At these air temperatures electric heat pumps are already switched off or at best achieve a COP of almost 2 (PER ≤ 0,8). A second advantage is the fact that in defrost mode, the unit still delivers 50% of its nominal heating capacity to the heating system and that there is no need for additional electric energy input.

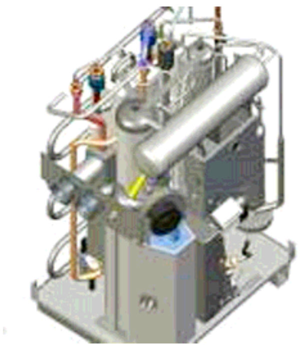
Figure 10-52.
Robur GAHP type A

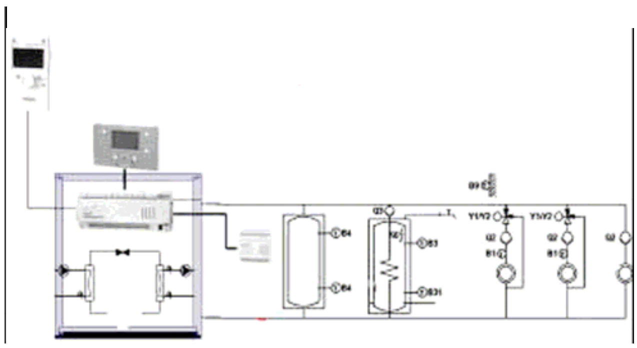
The efficiency (or GUE = Gas Utilization Efficiency) of GAHP's is measured according to EN 12309 "Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW ".

The graph applies to current generation of "heating only" products available on the European market, including both the air-water and the ground-water heating product. These are produced in larger quantities (several 1000s/year) for the Italian and international "light commcercial"(35 kW upwards) market at end-user prices of around 8-12.000 Euro/unit.



The picture shows the future generation of GAHP (>35 kW output), to be introduced in March 2008. It will feature a significantly higher efficiency, employing condensing technology and a "system approach", whereby the package already contains all the approporiatre pumps and controls.





An important advantage for ground source GAHP's is the fact that the size of the ground source collector can be approximately 50% smaller that for electric compression heat pumps, which is an important economic advantage.

Finally, an GAHP doesn't require a special or additional power connection terminal, with the corresponding fuses etc. nor does it suffer from high start-up currents that could influence the power grid when larger numbers of heat pumps are switched on.

11 SYSTEM ANALYSIS

Introduction

Modelling the heating system and specifically the CH-boiler is relevant for the future assessment of boiler efficiency for several tasks of the preparatory study:

- the definition of the basecase (Task 5),
- the evaluation of design options (Task 6) and
- the sensitivity analysis (Task 7/8)

The aim is to provide the mathematical background for an evaluation of boiler features, which –as in many EPB standards—comes on top of the outcome of the existing test standards (EN 303, etc.). The most important task will be not only to generate numerical values the efficiency effect of single design options/ features, but especially to evaluate the combined effect of several options.

The ECOBOILER model builds on

- Task 1, where –amongst others- the deficiencies of current test standards are discussed, which underline the need for the modelling.
- Task 2, where amongst others the technical market segmentation gives us insights in the penetration of boiler features. And above all
- Task 3, where statistical data are retrieved on EU-building characteristics and where a mathematical model of the Heat Balance of the residential dwellings was developed that is in line with EN, prEN and national energy performance of buildings standards.

The integrated ECOBOILER model stems from two separate models: The Heat Balance of the dwelling/building (Task 3) and a Boiler Model (Task 4).

The Heat Balance relates not only to the climate and building --following EN 832, SAP, DIN V 4801-6, RT 2005 etc.—but also to the heating emitters, CH-distribution network and (self-contained) controllers, following the EN 15136 series and several national standards. The Heat Balance assumes an 'ideal boiler' (100% efficient, no inertia, etc.), a radiator network [59], controllers for radiators and sensors and distribution losses resulting from a system temperature that is 'unavoidable' with a certain control strategy.

The addition of the Boiler Model will make the boiler less 'ideal' i.e. more realistic and will allow a 'system approach', not only looking at the efficiency of heat exchanger and burner in idealized circumstances, but also including controls and hybrid heating systems.

Development in stages

The development of the system analysis for central heating (combi-boilers) took several steps:

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[59] Radiators are present in >80% of the dwellings where new boilers will be placed (see Task 2). Floor heating and convector systems are not included here.

First the model of the dwelling (in Task 3) and the model of the central heating system (in Task 4) were developed as separate entities. A trace of this exercise can be found in the Annexes of the Task 3 report –for the dwelling model—and in the current Task 4 report for the boiler model (see Annex XX). This separate development took place from Nov. 2006- Jan. 2007.

Subsequently the two separate models were integrated in the ECOBOILER Integrated Model (or simply "ECOBOILER model"), enabling a system approach (not just the heat generator). The focus of the first version, presented in March 2007 to the expert meeting, was on the space heating function.

In the period May-June, the water heating function (combi) was added, consistent with the ECOHOTWATER model of the preparatory study for Lot 2 but with its own peculiarities regarding the use of wast. Multiple generators (including heat pump and solar) were accommodated.

In parallel, the ECOBOILER model was being reviewed by the experts from the boiler industry, who also called in the expertise of the iTG Dresden to review the suitability of the model for labeling purposes.[60]

Annex A contains the documentation of the Excel file containing the ECOBOILER model. It should be read alongside the Excel file and handles the references and equations in that file row by row. The Excel file containing the model is issued as a separate deliverable electronically (file on CD-ROM).

Annex B gives a summary of the review by iTG and boiler experts, including the first comments by VHK.

Annex C describes the separate Boiler Model, which preceded the ECOBOILER integrated approach.

Important Notice:

The Excel-file, this documentation and description of the model is appropriate for the purpose of this preparatory study, i.e. a review by boiler experts who have been involved in the development process. However, it is also clear that if and when the ECOBOILER model is used for other purposes, i.e. labelling and/or integration with harmonised EPBD standards (see Task 7 report), that a complete re-structuring and clean-up of the Excel file, the documentation and an appropriate discussion of the equations is absolutely necessary. This is most definitely outside the budgetary and time scope of the underlying preparatory study.

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[60] Institute für Technische Gebäudeausrüsting Dresden (iTG), prof. Oschatz. Et al.. "Analysis of the Kemna model as a basis for the labeling of heat generators, Dresden, May 21,2007. Note that the iTG was one of the parties working on the EnEV and the latest German EPB standards DIN 18599

ANNEX A: ECOBOILER DOCUMENTATION

This annex contains the documentation of the Excel file containing the ECOBOILER model. It should be read alongside the Excel file and handles the references and equations in that file row by row.

Important Notice:

The Excel-file, this documentation and description of the model is appropriate for the purpose of this preparatory study, i.e. a review by boiler experts who have been involved in the development process. However, it is also clear that if and when the ECOBOILER model is used for other purposes, i.e. labelling and/or integration with harmonised EPBD standards (see Task 7 report), that a complete overhaul and clean-up of the Excel file, the documentation of that file and an appropriate discussion of the equations is necessary.

Documentation

Ecoboiler Integrated Model–

Draft V.3 + addendum for distribution losses (June 2007)

SECTIONS:

- USER INTERFACE [ROWS 1-110]
- INTERMEDIATE INPUT PARAMETERS [ROWS 111-148]

- NET HEAT LOAD [ROWS 149-172]
- THERMAL MASS [ROWS 173-216]
- INTERNAL HEAT TRANSFER [ROWS 217-265]
- TEMPERATURE FLUCTUATIONS [ROWS 286-353; preparatory calculations in ROWS 266-285]
- TEMPERATURE STRATIFICATION [ROWS 353-407]
- DISTRIBUTION LOSSES [ROWS 408-486]
- STANDBY HEAT LOSSES [ROWS 487-522]
- START-STOP LOSSES [ROWS 523-535]
- BOILER EFFICIENCY [ROWS 536-609]
- ELECTRICITY USE [ROWS 610-632]

- SUMMARY OUTPUT [ROWS 633-657]
- ADDENDUM: ON-OFF THERMOSTAT [ROWS 658-757]

USER INTERFACE [ROWS 1-110]

The user interface is optimised for showing/printing main inputs and outputs on one single page.

Most inputs (in blue font) are loaded as data subsets from the LOOKUP worksheet. They can be edited, but editing should preferably take place in the LOOKUP sheet. The data-subsets can be loaded by selecting one option from the drop-down lists in the yellow cells. Only a few data (in black font) can be edited directly on the MAIN worksheet.

INTERMEDIATE INPUT PARAMETERS [ROWS 111-148]

In a user-friendly version of the model, rows 111-148 can be hidden. They contain some more elaborate inputs like outdoor temperature and solar irradiation data. Also intermediate parameters are defined, mainly for the convenience of programming the model.

NET HEAT LOAD [ROWS 149-172]

ROWS 151-153 calculate the “lowest possible temperature”. This is usually the set-temperature (Tset, per zone and dayperiod), but there are a few dayperiods in April-Sept. where this is not the case, i.e. where outdoor temperature + gains are higher than Tset.

Tlowest =
*MAX {Tset ; Tout + fi_1*F*(Qgain_a+qsol*sgf) /(V*((ah*(qv*(1-qrec)+qinf))+(AV*U)))}*

Consequently, the heat load in degree hours is given, using the weighted average of Tlowest for the 3 temperature zones [ROW 155]

Degreehours =
hd_a(fi_1*(Tlowest_zone1-Tout) + fi_2*(Tlowest_zone2-Tout)+fi_3*(Tlowest_zone3-Tout))*

The number of degreehours is not used in the rest of the calculation, but is a characteristic value of the system.

Finally there is the heating energy demand per zone and day-period that follows from Tset [ROWS 163-166]:

QH =
*MAX {0 ; hd_a*0,001*fi_1*(V*(Tset-Tout)*((ah*(qv*(1-qrec)+qinf))+(AV*U))-F*(Qgain_a+qsol*sgf)) }*

From QH the heating power demand per zone and day-period is derived [ROWS 169-172]:

PH = QH / hd_a

THERMAL MASS [ROWS 173-216]

FULL NIGHT SETBACK [ROWS 173-193]

Calculation of the end-temperature –with the start temperature Tsetprev from the previous period-- after a cool down during the entire period (e.g. 8h in the night) [CELL J175]

Tcooled =
IF (QH > 0; Tout + tmfc(Tsetprev-Tout) * EXP (-hd_e / (0,001*(Tsetprev - Tout) * fi_1*tm*V/PH)); Tlowest)*

Calculation of the reheat time (treheat in h) to bring the temperature from Tcooled to the temperature level of the next period Tsetnext [CELL F175]:

treheat =
*MAX(0;((Tsetnext-Tcooled)*0,001*tm*V*fi_1+0,0232*Pradnom1*(PH/Pradnom1)*50) / (Pradnom1*Preheat-PH))*

From the above the average cool down and reheat speed (in K/h) are calculated [e.g. CELL J180 and F180]:

vcooldown = (Tsetprev – Tcooled) / hd_e

vreheat = (Tsetnext – Tcooled) / treheat

The cooldown time was calculated for the full period (e.g. 8 hours during the night). But with the condition that the indoor temperature has to be back to the required level (Tsetnext) the reheat time has to be subtracted from the cooldown time.

First the reheat time is corrected proportionally to the ratio between vcooldown and vreheat [e.g. CELL F185]

*treheat_corr = IF(vreheat+vcooldwon>0; treheat *vreheat /(vreheat+vcooldown);0)*

Then the new end-temperature is determined [e.g. CELL J185]

Tcooled_corr=IF(AND(J163>0;J158<I158);

Tout+tmfc(Tsetprev-Tout)*EXP(-(hd_e-treheat_corr)/((Tsetprev-Tout)*0,001*tm*V*fi_1/PH +0,0232*Pradnom1*(PH /Pradnom1)*50));Tlowest)*

Tcooled_corr is the lowest temperature that occurs during the period. But for the heat balance we need the average temperatures per period [e.g. CELL J190].

Tavg_setbackperiod= Tsetprev – 0,5(Tsetprev – Tcooled_corr)*

After this is done for all setback-periods we find a new indoor-temperature matrix in ROWS 190-193 for a “full setback” strategy.

REDUCED SETBACK [ROWS 194-204]

A reduced setback means that the Tcooled_corr temperature is not allowed to drop below a minimum temperature Tmin. Should this happen then the boiler will enter into a steady-state (cycling) behaviour to keep the temperature above Tmin.

Reheat time calculation for a temperature difference of (Tset – Tmin) if [CELL F196]: =IF (Tcooled_corr>Tmin; treheat; MAX(0;((Tset-Tmin)*0,001*tm*V*fi_1+0,0232*Pradnom1 * (PH/Pradnom1)*50) /(Pradnom1*Preheat-PH)))

The cool down time is calculated using the cool down speed. Then [e.g. CELL J196] both the new reheat and cool down time are subtracted from the length of the day period (e.g. hd_e=8h):

Steady state time = IF(Tcooled_corr <= Tmin; hd_e – treheat - (Tsetprev-Tmin) /vcooldown;0)

Similarly to the “full setback” the average temperatures per setback-periods are now calculated for the “reduced setback” [ROWS 201-203].

In the ROWS 207-209 either the “full setback” or the “reduced setback” is selected, depending on the setting of CELL F26.

For this new set of indoor temperatures the heating energy demand QH is calculated in ROWS 212-214 and the difference with the ROWS 163-166 is calculated and listed as the energy penalty of the thermal mass (transferred to the OUTPUT table).

INTERNAL HEAT TRANSFER [ROWS 217-265]

Heat transfer between zones 1 and 2 in kW [ROW 219]:

Qint1=0,001(Tzone2-Tzone1)*(Aij_1*F*b*Uij + fi_1*V*qinfi*ah)*

Heat transfer between zones 1 and 3 in kW [ROW 220]:

Qint2=0,001(Tzone3-Tzone2)*(Aij_2*F*b*Uij + fi_3*V*qinfi*ah)*

Heat transfer between zones 2 and 3 in kW [ROW 221]:

Qint3=0,001(Tzone2-Tzone3)*(Aij_3*F*b*Uij + fi_3*V*qinfi*ah)*

Heat balance per zone [ROWS 224-226]:

- Zone 1: dQH_1 =Qint1-Qint2
- Zone 2: dQH_2 = -Qint3-Qint1
- Zone 3: dQH_3= Qint2+ Qint3

Indoor temperature adjustment [ROWS 229-231]:

*dT1 = dQH_1/ (0,001*V*fi_1*(ah*(qv*(1-qrec)+qinf)+(AV*U)))*

Etc.

This adjustment leads to a new indoor temperature matrix [ROWS 234-236]. Usually, this matrix will show a temperature increase for Zone 2 (for most CELLS) and a temperature drop for Zones 1 and 3. If the temperature drop is below Tset, then extra heating action will be required to keep Tset to the required level.

The corrected matrix, with extra heating in Zones 1 and 3 ["multi-zone"], is in [ROWS 239-241].

In case of a room thermostat in Zone 1, the correction will only apply to Zone 1 ["single zone"]. The corrected "single zone" matrix is shown in [ROWS 244-246].

Correction of Zone 1 also leads to a temperature increase in Zone 3 (assumed proportional), but this may not always be enough to reach Tset. Therefore, once the choice between "multi-zone" and "single-zone" has been established in [ROWS 250-252] (depends on the type of controller), there is a check for discomfort (expressed in % degree hours, ROWS 255-257).

TEMPERATURE FLUCTUATIONS [ROWS 286-353]

After the section on internal heat transfer, the approach of the 3 zones has served its most important purpose and we now switch to the heating energy and power of the whole house. Only in the first ROWS we also keep in the Zone 1 separately, so in the future others could expand on that (e.g. for room thermoststats)

As a first step we calculate QH and PH [ROW 262 and 265 respectively], using the equations mentioned earlier, but now with indoor temperatures that take into account the internal heat transfer.

We then calculate the radiator temperature (Tsys) that would fit the required power levels:

*Tsysavg = 20+50*POWER(PH/Pradnomsum;1/radc)*

In ROWS 271 and 273 we have to make some preliminary calculations for the weather control, where the distribution temperature (D. Vorlauftemperatur) is not based on the normal Tset-regime, but on a fixed value TW = 25°C (TWN= at night = 21°C).

PHW=MAX(0; 0,001(V*(TW-Tout)*((ah*(qv*(1-qrec)+qinf))+(AV*U))-F*(Qgain_a+qsol*sgf)))*

---

*Tsys_weather = TW+Cpar+50*Cgrad*POWER(F271/Pradnomsum; 1/radc) –CL;*

Please note that CL < 5°C → always gives a full reduction.

The following equations in ROWS 275-284 result in the total volume (in l.) of the CH-circuit in “on” and “off” mode:

Water circulating			
water in radiators			20*Pradnom
Lenght circ. pipes: **L1** = 28 + 0,05 * F; d=22 mm			L1*0,38
Length radiator pipes: **L2** = 0,515 * F ; d = 14 mm			L2*0,115
water in boiler			massw
TOTAL “on”			Sum of the above
water in bypass	2		2
TOTAL "off" (boiler+bypass)			massw+2
Specific heat CH water circ. in kWh/K "on"	wh=0,00116	kWh/kg.K	Total “on” * wh
Specific heat CH water circ. in kWh/K "off"	bh=0,000139	kWh/kg.K	=massb*0,7*bh+massw*wh

TFLUCT CALCULATIONS

The Tfluct is calculated for the 4 main controller types:

- Room modulating thermostat (“mod”)
- Weather control (“weather”)
- Fixed boiler temperature (“nominal”)
- Time proportional (“tprop)

The 5tht controller type (“room on/off”) will be derived later from “room mod.” for distribution and standby heat losses and “fixed”. (see Addendum)

The first step is to calculate the boiler power output that would ideally (with a large enough turndown) exist. The equation first describes the situation where the minimum boiler output Pbmin8060 is more than the required output. Otherwise the boiler output meets the requirement PH, with a maximum of the boiler output Pb8060.

*Pb = IF (AND (PH<Pbmin8060;PH>0); Pbmin8060 + ((dptc-Tsysavg-dTrad_mod) / (dptc-30)) * (Pbmin5030-Pbmin8060); MIN(PH;Pb8060))*

For the weather control the equation is the same, but with Tsysavg→ Tsysavg_weather, PH → PHW and dTrad_mod → dTrad_weather.

For “nominal” control Pb=Pb8060 (boiler maximum output).

The output of the time proportional boiler “tprop” is the same as “mod” Pb_tprop = Pb_mod.

The temperature increase of the system is calculated from the given tband

- dTrad_mod=20*tband*(Tsysavg-20)/50)   (default: 10*(Tsysavg-20)/50))   [CELL 295]
- dTrad_weather=2*tband/((TsysW-20)/6) (default:10/((F273-20)/6) [CELL 296]
- dTrad_nom= 2*tband/((Y32-20)/6)   (default: 20/(50/6)) [CELL 297]

dTrad_tprop=Tsysavg-(20+(Tsysavg-20)*EXP(-tcooldown/3)) [CELL 306; this parameter can only be calculated after tcooldown is established]

Cool down time during steady state:

- tcooldown_mod=-1,5*LN((Tsysavg-dTrad_mod-20)/(Tsysavg+dTrad_mod-20))   [ROW 301]
- tcooldown_weather=-1,5*LN((TsysW-dTrad_weather-20)/(TsysW+dTrad_weather-20)) [ROW 302]

- tcooldown_nom =-1,5*LN((Tbmax-dTrad_nom-20)/(Tbmax+dTrad_nom-20))
  *[ROW 303; default -1,5*LN(40/60))]*
- tcooldown_tprop = (1-(PH/ Pb_tprop))*(1/fcyc) *[ROW 304]*
- theatup_mod=(PH_mod/(Pb_mod – PH_mod))*tcooldown_mod *[ROW 308]*

etc. [ROWS 309-311]

The second copy of tcooldown and theatup (identical to the above) is placed in ROWS 341-349

- Tfluctvalve_mod = 0,25*valveband*(1+valvedelay/theatup_mod)
  *(default: 2*0,25*valveband) [ROWS 315-318]*
- Tfluctcntrl_mod =0,25*xtband*(1+xtdelay/ theatup_mod)
  *(default 2*0,25*xtband) [ROWS 320-323]*

The total Tfluct is calculated in ROWS 225-228:

- Tfluct total _mod = 0,5* Tfluctvalve_mod +0,5* Tfluctcntrl_mod
- Tfluct total _weather = Tfluctvalve_weather
- Tfluct total _nom = Tfluctvalve_nom
- Tfluct total _tprop = 0,5* Tfluctvalve_tprop +0,5* Tfluctcntrl_tprop

The new heating demand QH4 can now be calculated [ROWS 330-333]

*QH4= hd_a*0,001*(V*(Tprev+**Tfluct_total**-Tout)*((ah*(qv*(1-qrec)+qinf))+(AV*U))-F*(Qgain_a+qsol*sgf)))*

This concludes the calculations of Tfluct. But for future reference a copy of tcooldown and theatup is placed in ROWS 341-349 to allow for possible corrections of the cycle times if needed in the future.

TEMPERATURE STRATIFICATION (ROWS 354-407)

The effect of Tfluct on average system temperature is calculated, first by establishing the increment and then by summing the increment to the average Tsysavg.

The increment follows from the radiator formula (ROWS 356-359)

dTsys=
*IF(Tsys>0; MAX(22; 20+50*POWER(QH4/(hd_a*Pradnomsum);1/radc))-Tsys;0)*

The new Sysavg –calculated in ROWS 362-365-- follows from summing the increment to the previously calculated system temperature in ROW 268:

Tsysavg = Tsysavg_prev + dSys

Of course we could have calculated the new Sysavg in one step, but this gives us an extra check.

For the calculation of Tstrat we do not (only) need the average system temperature Tsysavg, but the system feed temperature Tsysfeed, i.e. the temperature of the boiler leaving the boiler. Amongst others this Tsysfeed temperature, at a given boiler output Pb, depends on the pump flow rate. For this the model gives two options:

A variable speed circulator that regulates its flow rate as a function of a pre-determined boiler temperature increase. As a default we calculate the flow rate that follows from a boiler temperature increase ΔTb = 10 K. The equations in ROWS 369-372 follow the following format:

*fvar=MAX(fmin;Pb/(ΔTb *wh))*

fmin	= minimum pump flow rate in l/h (default 100 l/h)
ΔTb_set	= desired boiler temperature increase (default 10 K for mod/weather/tprop ; default 20 K for fixed)
wh	= specific heat of water in kWh/kg.K (constant 0,00116 kWh/kg.K)
Pb	= boiler output (from ROWS 289-292)

For a fixed speed pump the flow rate is given (default f = 1000 l/h, in CELL C374) and boiler temperature increase ΔTb can be calculated [ROWS 375-378]

*ΔTb =IF(fixflow>fvar;Pb/(fixflow*wh);fvar)*

The condition fixflow>fvar is only critical (within the range of our model) for a fixed controller with a boiler operating at nominal output, where we have to make sure that the boiler temperature remains below boiling point. (Tb<100°C).

In ROWS 382-385 the Tsysfeed is calculated on the basis of the pump selected:

Tsysfeed=IF(pumpselected=fixflow ; MIN(Tsysavg+20;Tsysavg+ ΔTb_fix); Tsysavg + ΔTb_set)

In other words: If a fixed speed pump is used Tsysfeed follows from the addition of the average system temperature and the boiler temperature increase ΔTb_fix that follows from the flow rate. This latter ΔTb_fix can never be more than 20 K. If the a variable speed pump is used the boiler temperature increase ΔTb_set is a given and should be added to the average system temperature Tsysavg.

With the above we can now calculate Tstrat, using the general formula

ΔTstrat =
0,5{ 0,225+0,1*Qt/(Qv+Qt) + 0,5*Qv/(Qv+Qt) } * MAX { 1,5; 1,5+(Tsysfeed – 20)*0,05 }*

Qt/(Qv+Qt) and Qv/(Qv+Qt) give the relative shares of transmission resp. ventilation losses in the total and have to be calculated only once. This is done in CELLS D390 and D391.

*Qt/(Qv+Qt) =(AV*U)/(ah*(qv*(1-qrec)+qinf)+AV*U) [CELL D390]*

Qv/(Qv+Qt) = ah(qv*(1-qrec)+qinf)/(ah*(qv*(1-qrec)+qinf)+AV*U)[CELL D391]*

With the outcomes of the above the new heating demand QH5 can be calculated [ROWS 395-398]:

*QH5 = hd_a*0,001 * (V * (Tprev+**Tfluct_total**+**Tstrat**-Tout) * ((ah*(qv*(1-qrec)+qinf)) + (AV*U)) – F * (Qgain_a+qsol*sgf)))*

And also the required heating power PH5 [ROWS 400-403]:

PH5= QH5/ hd_a

Please note that in any real-life system –also with a floor heating system-- a minimum amount of stratification losses is unavoidable. For a radiator system these unavoidable ("ideal") stratification losses are calculated in ROWS 405 and 407 as a reference for the system efficiency.

*Δtstrat @ ideal = 0,5 * (0,225+0,1 * Qt/(Qv+Qt) + 0,5* Qv/(Qv+Qt)) * MAX(1,5; 1,5 + (Tsysavg_ideal-20) * 0,05)*

Where Tsysavg_ideal is the Tsysavg <u>without</u> the effect of Tfluct (in an ideal system there is no/little cycling).

The "ideal" stratification losses now become:

*QH5 @ ideal = hd_a * 0,001 * (V * (Tprev+ **Tstrat@ideal**-Tout) * ((ah * (qv * (1-qrec) + qinf)) + (AV * U)) - F * (Qgain_a + qsol * sgf)))*

DISTRIBUTION LOSSES [ROWS 408-486]

Preparatory calculations:

In ROWS 410-415 the increment of the system temperature, taking into account **both** the fluctuation losses Tfluct and the stratification losses Tstrat, is calculated for the 4 controller options plus the "ideal" option. The increment follows from the radiator formula (equation as in ROWS 356-359).

Consequently, the **calculated** system return temperature **Tsysreturn** is given in ROWS 418-421 for the burner **"on" mode**, by adding the increment to the average system temperature as calculated in ROW 268.

This requires some **explanation**, because here –for the first time—we abandon the traditional notion that the system return temperature directly depends on the system feed temperature and the boiler power. This traditional notion assumes that the system water makes a loop and therefore any change in the system feed temperature almost immediately results in a change in the system return temperature. This traditional notion assumes a system where the radiators and the boiler output are dimensioned for the exact heat demand and where the radiators are perfect "mixers". In a perfect mixer the return temperature is calculated from volumes and temperatures of the incoming water and the water in the radiators.

However, this is only valid for an extremely cold winter (the design situation) and/or a system that is hydraulically unbalanced (e.g. only one radiator in the total system with heat demand; see later). In average weather conditions and a balanced system this is not what happens: Both the radiator and the boiler have to work at a load which is only around 10% of their capacity. This means that the radiators have –relative to the load—a huge water content and the thermostatic valve (or the room thermostat) shuts off long before all the radiator water has been replaced by incoming boiler water. Furthermore, most radiators are not perfect mixers but show a considerable degree of stratification. This means that mostly the incoming water "pushes" the water sitting in the radiator out during the "on" mode. In other words, this system return water still has the old radiator temperature. After the burner switches off the radiator water will become fully mixed during the "cool down" period. Partly this mixing will be due to the pump action (with thermostat systems) and in partly this will take place through convection and conduction inside the radiator.

In ROWS 424-427 the average system temperature during the "burner off" **Tsysreturn_off** is calculated. This is the temperature of the water in the distribution pipes when all valves are closed and/or the thermostat indicates that there is no heat demand.

For room thermostat systems ("mod" and "tprop" in ROWS 424 and 427) this system temperature in off mode is set equal to the average system temperature. This may seem a bit high, but it has to be considered that there are several unknown factors.

For systems that work with a boiler temperature controller (e.g. "weather" and "fixed" in ROWS 425 and 426) the distribution loop is usually kept at a constant temperature (*D. Vorlauftemperature*), dictated by the outdoor sensor and/or manual settings.

Tsysreturn_off_weather = Tsys_W *(defined in ROW 273)*

Tsysreturn_off_fixed = Tbmax *(defined as an input of the controller)*

From the above Tsysreturn_off and the Calculated Tsysreturn_on we now calculate the **Real Tsysreturn_on** in ROWS 430-434. This Real Tsysreturn_on depends on a single parameter **bypmix:**

Real Tsysreturn_on =
*bypmix * Tsysreturn_off + (1 – bypmix) * Calculated Tsysreturn_on*

Please note that for room thermostat systems ("mod" and "tprop") Tsysreturn_off roughly equals Calculated Tsysreturn_on, so there is hardly any effect for bypmix.

But for weather controlled systems and fixed systems the bypmix parameter takes into account two effects:

- Hydraulical unbalance in the system, and
- The effect of a bypass in the hydraulic system.

To understand the effects involved the figure below depicts the principle of a system with a boiler, a single radiator and piping with a bypass. The radiator valve is half open and in such a way that 44% of the flow passes through the radiator. The other 56% flows through the bypass and meets the return water from the radiator. The system return water is thus a mix of the water from the radiator (34°C) and the water from the bypass (54°C), resulting in 45.2°C. This is significantly higher than the situation witout the bypass, but it is widely believed that a bypass is needed for the situation that all radiator valves are closed and the system has to monitor heat demand.

In other words, the bypass –especially when not properly installed— increases the return temperature and thereby decreases the boiler efficiency.

A similar effect comes from a hydraulically unbalanced system, where often only 1 radiator out of a total system of 7 or 8 radiators is causing the heat demand. In this case the "active" water content of the system is small and here –contrary to the balanced system described earlier-- the system feed temperature is really determining the system return temperature. The latter will be much higher than in a balanced system.

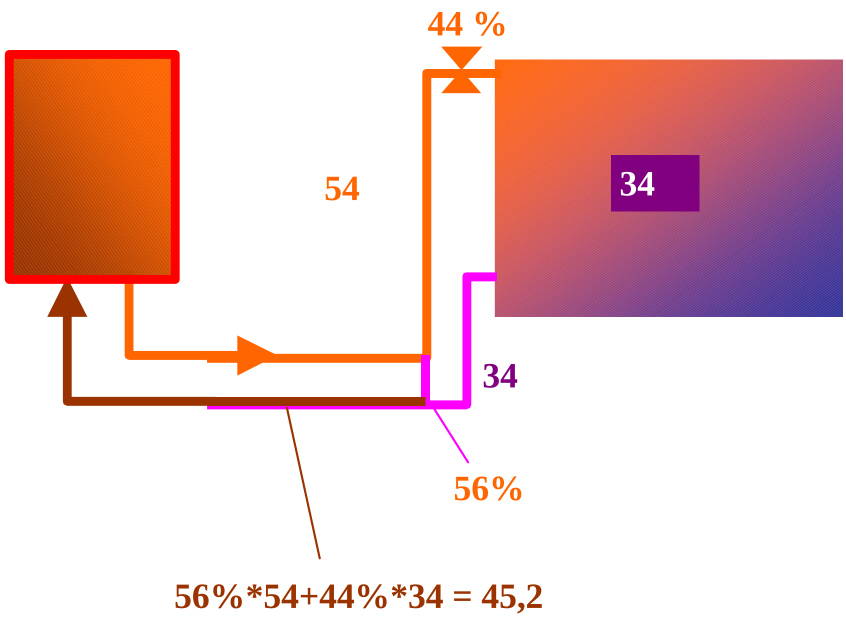

Figure A-1.

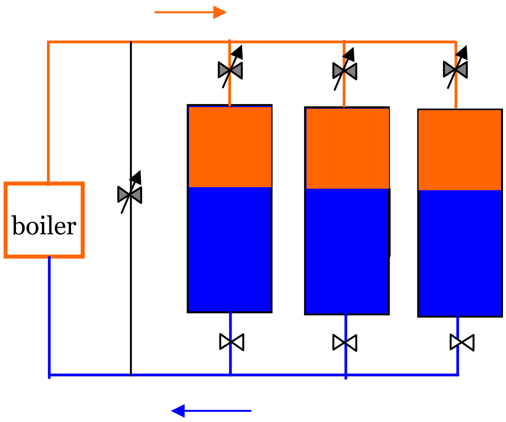

Figure A-2.
Hydraulically balanced system of 1 boiler + 3 radiators with TRVs and footvalves. The hot CH water is distributed equally over the radiators and pushes a part of the resident (colder) radiator water out. The TRVs close (= the heat demand is met) before the incoming hot CH water reaches the return loop.

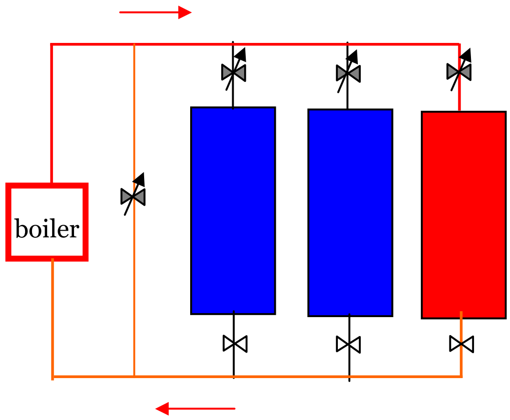

Figure A-3.
Hydraulically unbalanced system of 1 boiler + 3 radiators with TRVs and footvalves. The heat demand is caused by only one radiator (room) in the network. The water content of the whole loop is much smaller and the hot CH water quickly pushes the resident radiator water out and then reaches the return loop. Result: Higher return temperature. The heating may be stopped either by the TRV or the boiler thermostat. Depending on the setting, also the bypass may have a significant impact on the system return temperature.

In ROWS 436-439 the average boiler feed temperature in "on" mode is calculated. In fact the increase in radiator temperature (ROWS 411-414) is added to the previously calculated system feed temperature (ROWS 382-385).

In ROW 442 the system temperature during reheating is calculated. The default value is Tsysavg= 70°C, based on the formula Tsysavg = 20 + 50*Preheat

In ROW 443 the system temperature during cooldown is calculated. The default value is Tsysavg= 30°C, based on the assumption that the radiator temperature cools down to around 25°C within one or two hours (during setback) and therefore the overall average is around 30°C. A more exact calculation would be possible, but the amount of error in this simplified approach is assumed to be small.

In ROWS 444-447 the system temperature during the steady state of a "reduced setback" setting is calculated as an average of the "normal" return and feed temperature.

Tsysavgss = 0,5(Tsysfeed + Tsysreturn)*

Now we know the system temperatures for each of the 5 states: ON, OFF, REHEAT (during setback), COOL DOWN (during setback) and STEADY STATE (during setback). What is missing for the calculation of the distribution losses is the number of hours for each of these states. This is calculated in ROWS 457-472, but in preparation of this we first assess the number of cycles Ncyc in ROWS 451-455.

The **number of cycles** depends on the average cycle time. This is the sum of heat-up time in ROW 341-344 and cool down time in ROW 346-349. Formula:

Ncyc_a = hd_a/ (heat-up time per cycle+ cool down time per cycle)

Where Ncyc_a is the number of cycles in period a and hd_a is the number of hours in day period a.

The results are summed in ROWS 451-455 and columns

- D (total cycles),
- C (total cycles with full night setback, i.e. D minus night cycles + 1) and
- B (total cycles with reduced night setback, i.e. C + cycles during the night in steady state)

The number of cycles during the reduced night setback is shown in ROW 455 and is derived from the thermal mass calculations in ROW 196.

The number of "ON" hours (ROWS 457-461) and the number of "OFF"hours (ROWS 464-467) are simple multiplications of the number of cycles with the heat-up time per cycle (ROWS 341-344) and the cool down time per cycle (ROWS 346-349) respectively.

This is done for day-periods a to d (morn, midday, eve, late).

For the night (period e) we copy the REHEAT period from F185 or F196, depending on whether it is a full setback or a reduced setback mode. These values can be found in ROW 470.

Likewise we copy for the STEADY STATE hours in ROW 471 the value from J196 in case of a reduced setback. If the system uses a full nigh setback the value is 0 (zero).

The COOL DOWN hours in ROW 472 follow from subtracting the STEADY STATE and REHEAT hours from the 8 hour day period.

Now that we know both the system temperatures and the hours of each system-state, we can finally establish the average distribution temperature Tdistr in ROWS 475-478.

Formula for day period a (similar for b to d):

*Tdistr= (Tsysavg_on * hrs_on + Tsysavg_off * hrs_off) / hd_a*

Formula for day period e (night):

Tdistr =
*(Tsys_reheat * hrs_reheat + Tsys_cooldown * hrs_cooldown + Tsys_steadystate * hrs_steadystate) / hd_e*

Tdistr is one of the most important inputs for the calculation of the distribution losses in ROWS 481-486.

Qdistr = 0,001 hrs * F*Lpipe*hlf*Upipe*(Tsys - Ta),*

Where

- 0,001 is the conversion factor from W to kW
- hrs is the number of hours that the system is running (default heating season is 5000 h/ year)
- F is the heated floor area in m²
- Lpipe is the specific pipe length in meters pipe per m² heated floor area (from lookup table below)
- Hlf is the heat loss factor (default 0,15 for heated space, 1 for unheated space; actually varies between 0,3 and 0,37 see table below)
- Upipe is the specific heat loss per meter pipe in W/K.m (see look-up table below)
- Tsys is the average system temperature in°C (determined by the model)
- Ta is the ambient temperature (default 20°C)

The table below will be used in the ECOBOILER model to find the values for the various size classes (in LOOKUP worksheet, cells D805:F813, furthermore the Lpipe parameter will be added on the MAIN worksheet at and Qdistr will change in MAIN rows 481:484).

Table A-1. ECOBOILER Look-up Table

Size	Lpipe	hlf	Upipe excl. hlf if hlf=0,7
	m/m²		W/K.m
XXS	1	0,30	0,22
XS	1	0,30	0,22
S	1	0,30	0,44
M	1	0,30	0,44
L	1	0,30	0,44
XL	0,75	0,37	0,27
XXL	0,75	0,37	0,50
3XL	0,55	0,37	0,66
4XL	0,55	0,37	0,66

Please note, that in ROWS 481-484 the distribution losses are given for each controller type.

In ROW 486 we give the "unavoidable" distribution losses **Tdistr@ideal,** i.e. the minimum achievable with the given hlf and Upipe factor. The distribution temperature Tdistr is determined in ROWS 268 (determined after the internal heat transfer) and 415 (increment due to stratification).

STANDBY HEAT LOSSES (ROWS 487-523)

The standby losses of the boiler at an average boiler temperature of 50°C follow from tests according to EN 303 etc. standards. This value Pbstby must be corrected for the actual boiler temperature, for which we use the formula given in prEN 15136:

*Pbstby_ini = Pbstby * POWER((Tsys_off -10-inheatedspace*10) / 20;1,25)*

Tsys_off is given in ROWS 424-427.

DTrad is given in ROWS 296-298 (mod/weather/nominal) and 306 (tprop).

Inheatedspace is a correction factor for the fraction of boilers that is placed in the heated floor area

This is calculated in CELL B514 with the equation depending on the volume of the boiler envelope volumeb:

Inheatedspace= MAX(0;1-(0,25+volumeb))

The expression "-10-inheatedspace*10" is the temperature in the boiler room. The average temperature of the unheated boiler room is assumed to be 10°C in the heating season. If however the boiler is placed in the heated floor area of the dwelling (inheatedspace=100%), then the boiler room temperature is 20°C.

For small-medium boilers the value of inheatedspace is 60%. Note that this parameter is double important for the standby losses: First it reduces the standby losses significantly and second –as we will discuss later—a part of the standby heating losses that remains is considered useful.

Pbstby_ini is the standby heat loss at the beginning of the cooldown cycle, based on the known Tsys. During the cooldown period the temperature Tsys decreases. In ROWS 496 to 499 the temperature at the end of the cooldown period (without any intermediate reheat) is calculated. Especially for room thermostat controllers, where by definition there are no radiator valves in the living room (Zone 1), it makes a difference how long the pump is running after the burner is off. This is indicated by the user variable tpmp (default 0,166 h = 10 minutes).

Tsys_cooled=
*20 + (Tsys_off-20) * EXP(-tcooldown / ((TMrad_off * (tpmp/tcooldown) + fi_1*TMrad_on * (1-(tpmp/tcooldown))) * (Tsys_off-20) / Pbstby_ini))*

Tsys_off is given in ROWS 424-427

tcooldown is given in ROWS 341 (mod.) and 344 (tprop).

fi_1 is the share of Zone 1 in the total dwelling (default 0,5)

The thermal mass of the "off" loop TMrad_off –which is the one considered when also the pump is off—is given in ROW 284.

The thermal mass of the "on" loop TMrad_on –which is the one considered when also the pump is running after the burner is off—is given in ROW 283.

Of course tpmp/tcooldown is always $\leq$ 100% and (1-(tpmp/tcooldown) is always $\geq$ 0% , which explains the MIN and MAX functions in the formula that is used in the spreadsheet.

For the boiler thermostat controllers (weather and fixed) the formula is simpler, because the cool down always occurs in the "off" loop.

*Tsys_cooled = 20 +(Tsys_off-20) * EXP (-tcooldown /((TMrad_off*tcooldown) * (Tsys_off-20) / Pbstby_ini)))*

Using the newly calculated Tsys_cooled instead of Tsys_avg we now can calculate the standby heat losses at the end of the cooldown cycle. This is done in ROWS 502-505 , using the same formula as for Pbstby_ini but now with Tsys_cooled instead of Tsys_avg. Note that this calculation only applies to the room thermostat controllers (mod. and tprop); for the boiler thermostat controllers we just copy Pbstby_ini.

For the boiler thermostat controllers (weather and fixed) the calculation of a cool down temperature is a bit theoretical, because when the boiler temperature sinks below the set temperature minus the tband the boiler will reheat the loop. Nevertheless, we have calculated this theoretical temperature, because it helps us to establish whether such an extra boiler cycle to keep the boiler at the desired temperature would occur.

ROWS 507 to 510 indicate these **extra cycles** needed to reheat the small CH loop (boiler + bypass) during off mode. This contributes to assessing the total number of cycles, but is not further used in any calculation. For room thermostat controllers (mod and tprop) this is zero by definition. For boiler thermostat controllers (weather and fixed) we assume that an extra cycle occurs when the Tsys_cooled drops below the set temperature minus the tband. If the calculated cool-down is twice the bandwidth we assume that the reheating occurs twice.

From the above the total standby heat losses QHbstby (= QH7) are calculated in ROWS 515-518. For the day periods a to d we multiply the number of "OFF" hours with the straight average of Pbstby_ini and Pbstby_end. This then leads

*QHbstby = hrs_off * 0,5*(Pbstby_ini + Pbstby_end)*

For the night period (=e) we use the cooldown hours (ROW 471) and half of the steady state hours (ROW 472) instead of hrs_off.

The summation of the heat losses per day period can be found in CELLS B515 – B518, but it is corrected for the useful heat when the boiler is in the heated space. The formula builds on the German DIN 4701-10:

*QHbstby_corr = (1-0,85*inheatedspace)* SUM(QHstby)*

The parameter inheatedspace was explained above.

Finally, although we will probably not use this data in our study we have added in ROW 520 the calculation of the energy consumption of the pilot flame. This is a multiplication of the number of OFF-hours and the power of the pilot flame (Pign).

*QHign = hrs_off * Pign*

Please note that the energy use of the pilot flame during hrs_on is part of the standard tests (see boiler efficiency)

For the “ideal” boiler we assume that at least during the night setback period (6 hours) a certain amount (0,1 kW) of net heat loss will be unavoidable. We have set the “ideal” standby loss at ca. 20 kWh/year.

START STOP LOSSES (ROWS 523-536)

For the most part the extra energy losses caused by cycling –as opposed to continuous operation-- are included in the standby heat losses, distribution losses, etc.. But there are two start-stop items that are not yet addressed: The energy losses during pre-purge and the fuel loss (incomplete combustion).

Pre-purge losses QHpurge are indicated in ROWS 525-528 and depend on the pre-pruge time tpurge (default 30 s), the power involved (we use Qbmin8060 as an indicator), the air factor λ (CELL L48), the specific heat of air ah (0,33 Wh/K.m³), the boiler temperature (Tsysreturn_on with respect of the outdoor temperature Tout), the number of cycles (in ROWS 451 and 508).

*QHpurge =tpurge*Qbmin8060*(0,1+ λ)*0,001*ah*(Tsysreturn_on-Tout)*(Ncyc+Ncyc_extra)*

For the fuel losses per cycle QHfuel we use the data from Pfeiffer (see Task 4 report), which come down to a 1,5% fuel loss factor (flf) at 14.000 cycles per year with a standard (pneumatic) air-fuel mixing control.

The formula used in ROWS 530-533 is

*QHfuel = Ncyc*flf*((Ncyc_tot+Ncyc_extra_tot)/14000)*

For an “ideal” boiler (continuously running), the influence of the above parameters is assumed to be negligible.

BOILER EFFICIENCY [ROWS 536-609]

As a first step we calculate in ROWS 539-542 the energy consumption per day period QH8, including all the losses calculated so far.

*QH8 = QH5 +ΔQH6 +(1-0,85*inheatedspace)* ΔQH7+ QHign+ QHpurge + QHfuel +F530*

- QH5 is the energy consumption including Tfluct and Tstrat, as well as the effect of thermal mass and internal heat transfer
- ΔQH6 is the increment caused by the distribution losses ΔQdistr
- ΔQH7 is the increment caused by the standby heat losses (Δqbstby), taking into account the useful energy when boilers are in the heated space described by the parameter inheatedspace
- QHpurge is the increment caused by purge losses
- QHfuel is the increment caused by fuel losses
- QHign is the increment caused by the pilot flame

From this we can easily derive the average heating power demand PH8 in ROWS 545-548

PH8 = QH8/hd_a

The next ingredient is the average return temperature in on mode at the time when the heat-up starts. This value for **Tsysrini** is copied in ROWS 550-553 from the one calculated for the distribution losses (ROWS 430-433) for reasons of convenience.

Now we can calculate in ROWS 555-558 the actual boiler power output Pb that fits the heating power demand PH8 and that fits the 4 controllers. Furthermore, the actual boiler power output Pb is limited by the turndown ratio at the low end and the maximum heat output at the high end.

Equation for mod/weather/tprop:

$Pb=IF(AND(PH8<Pbmin8060;PH8>0);Pbmin8060+((dptc-Tsysrini)/(dptc-22))*(Pbmin5030-Pbmin8060); MIN(PH8;Pb8060))$

For a fixed boiler we assume Pb= Pb8060

We can now calculate the boiler efficiency. This depends on the input and output energy from the EN test standards (EN 303 etc.) given in CELLS C35, C40-C42 and C44-C45 as a user input. These values are first recalculated into efficiency values in CELLS B561-B564:

$$\eta 8060 = Pb8060/ Qb8060$$

$$\eta 8060min = Pbmin8060/Qbmin8060$$

$$\eta 5030 = Pb5030/Qb8060$$

$$\eta 5030min = Pbmin5030/Qbmin8060$$

These efficiency values are the main ingredients for the efficiency of the boiler during normal operation ηbss (in ROWS 563, 575, 587 and 599) and the efficiency during the heat-up at the end of a setback period ηbhu (in ROWS 568, 580, 592 and 604), which all take η8060 as a basis.

But there are also two additional components Csysr and Cload per efficiency value.

Csysr depends on the one hand on the

a. the actual boiler power output Pb calculated in ROWS 555-558,
b. the return temperature Tsysrini from ROWS 550-553 and

and on the other hand it depends on the given η8060, η8060min, η5030, η5030min and the corrected dewpoint dptc in CELL L49.

The dewpoint is typically around 52°C for gas-fired and ca. 41°C for oil-fired boilers at an air factor of 1,3. It is an important parameter, because where the return temperature is above the dewpoint the efficiency improvement is relatively small, only caused by a reduction of the flue gas losses. Following e.g. the RT2000 we will attribute above the dew-point a maximum of 1% per 10K.

Below the dew-point, the efficiency gain also comes from the recovery of the latent heat (condensation) and there we will attribute the rest of the efficiency differences between a 50/30 and 80/60 regime.

At nominal (maximum) power and Tsysrini<dptc:

$$Csysmax = (60-dptc)*0,001 + ((\eta 5030 - \eta 8060) - (60-dptc)*0,001) * (dptc-Tsysrini)/(dptc - 30)$$

At minimal power and Tsysrini<dptc:

$$Csysmin = (60-dptc)*0,001 + ((\eta 5030min - \eta 8060min) - (60-dptc)*0,001) * (dptc - Tsysrini)/(dptc - 30)$$

The efficiency at actual power is given by

$$\eta bss = \eta 8060 + Csysmax + Pb/P8060 * (Csysmin - Csysmax)$$

and at heat-up:

$$\eta bhu = \eta 8060 + Csysmax + \eta 8060*Preheat/P8060 * (Csysmin - Csysmax)$$

Naturally the return temperatures at heat-up will be different (62°C default at 100%= 70°C – 8°C)

$$Tsysrhu= 20+Preheat*(62-20)$$

Where Preheat can of course not be lower than the turndown ratio.

ELECTRICITY USE [ROWS 610 –632]

Manufacturers supply the electricity use at maximum heating power elmaxon, at minimum heating power elminon, at zero load elstby and give the electricity consumption of the pump.

In order to use the electricity use we need to assess the actual heating power (above the minimum turndown). This is done in ROWS 612, 617, 622, 627 for the 4 controller types, using the formula:

$$Xtrapower = (Pb-Pbmin8060) /(Pb8060-Pbmin8060)$$

Consequently, the electricity consumption in on mode is calculated in ROWS 513, 618, 623 and 628 with the equation

$$elon\_tot = (elminon+Xtrapower*(elmaxon-elminon)) * hrs\_on * primenergy$$

Where primenergy (default 2,5) is the conversion factor from electricity to primary energy. hrs_on can be found in ROWS 458-461.

The electricity consumption in off mode

$$elstby\_tot = hrs\_off * elstby * primenergy$$

hrs_off can be found in ROWS 464-467.

The electricity consumption per day period of the pump in off mode can be found using the

$$elpmp\_tot = MIN (hd\_a ; Ncyc* MIN(tpmp; tcooldown))*elpmp*primenergy$$

This expression takes as a basis either the post-burn pump time (tpmp) or the cooldown time in a cycle (tcooldown) --whichever is smaller—and then multiplies this with the number of cycles Ncyc. The outcome can never be more than the total number of hours in a dayperiod (for instance hd_a).

The resulting pump time is then multiplied with the specific energy consumption of the pump and the primary energy conversion factor.

When summing the annual electricity use of the pomp in off mode, also the option for smart pump management can be taken into account with the so-called pump setback parameter pmpsb (CELL Q39). In CELLS C215, C220, C225 and C230 we find

$$elpmp\_year = 30{,}5 * (SUM (elpmp\_tot) – 9 * MIN(tpmp;pmpsb) *elpmp * primenergy)$$

SUMMARY [ROWS 633-659]

Sums the output totals and prepares some values for the indoor temperature graph

ADDENDUM [ROWS 658-757]

At the later stage it was decided to also develop an extra trail for the 5tht controller option: the on/off room thermostat.

For space heating

$$Qout = (1{,}029*Y - 0{,}065*X - 0{,}245*Y*Y + 0{,}0018 * X*X + 0{,}0215*POWER(Y;3)) * Qload$$

with

$$X = Asol*(a\_1+ (5+0{,}5*Asol))*0{,}8*(100 - Tout) * POWER(75*Asol/Vsol; 0{,}25) / (Qload * 1000)$$

where a_1 is 3,5/15/1,8 for glazed/unglazed/vacutube collectors respectively

$$Y = Asol * IAM *0{,}8*0{,}8 * qsol * hd\_a / (Qload*1000)$$

where IAM is 0,94/ 1/ 0,97 for glazed/unglazed/vacutube collectors respectively

*Qout = (1,029*Y - 0,065*X - 0,245*Y*Y + 0,0018 * X*X + 0,0215*POWER(Y;3)) * Qload*

Parameters are: Asol, type (glazed/unglazed/vacutube), Vsol (storage tank volume), the share fhwof Asol and Vsol for space heating vs. DHW heating and max. share fhh in space heating vs. the boiler

Conditions: Qout>0 Qout<Qload (maximum output Qout=Qload)

1 -El. brine/ water (0/50)	3,0	Output correct=1+(50-Tsnk)*0,0047 - (0-Tsource)*0,028
2 -El. water/ water (10/50)	3,8	Output correct=1+(50-Tsnk)*0,0047 - (10-Tsource)*0,013
3 -El. outside air/ water 7/50	2,8	Output correct=1+(50-Tsnk)*0,00266 - (7-Tsource)*0,0237
4 -Gas eng. water/water	2,8	Power corr=((50-Tsink)/15)*(0,73+(5+Tsource)*0,022) + (1-(50-Tsink)/15)*(0,65+(Tsource+5)*0,0235)
5- Gas absorpt NH3/H2O	1,1	Output correct=1,07 (at Tsink=40 and Tsource=10)
6 -Gas absorpt H2O/LiBr	1,1	1 (at 12°C source and 34°C sink)

1 -El. brine/ water (0/50)	COPcorr=((50-Tsink)/15)*(1,29+0,02*(Tsource+5)) + (1-(50-Tsink)/15)*(0,88+0,012*(Tsource+5))
2 -El. water/ water (10/50)	COPcorr=((50-Tsink)/15)*(1,51+0,038*(Tsource-10)) + (1-(50-Tsink)/15)*(1+0,032*(Tsource-10))
3 -El. outside air/ water 7/50	COPcorr=((50-Tsink)/15)*(0,91+(7+Tsource)*0,0304) + (1-(50-Tsink)/15)*(0,7+(7+Tsource)*0,0222)
4 -Gas eng. water/water	COP corr=((50-Tsink)/15)*(0,73+(5+Tsource)*0,022) + (1-(50-Tsink)/15)*(0,65+(Tsource+5)*0,0235)
5- Gas absorpt NH3/H2O	COPcorr=1,07 (at Tsink=40 and Tsource=10)
6 -Gas absorpt H2O/LiBr	1 (at 12°C source and 34°C sink)

1 -El. brine/ water (0/50)	
2 -El. water/ water (10/50)	
3 -El. outside air/ water 7/50	if load>=50% then y=0,891+1,35*LN(load-0,47) else y=0,891
4 -Gas eng. water/water	if load>=50% then y=0,891+1,35*LN(load-0,47) else y=0,891
5- Gas absorpt NH3/H2O	load_corr=MIN(1;0,131*LN(load) + 1,03)
6 -Gas absorpt H2O/LiBr	

DIN 4701-10

If collector area is at least 1,8 times the area needed for DHW, then a coverage of 0,10 (10%) can be used for the space heating contribution. For the final efficiency calculation also the pump and tank losses have to be taken into account.

*Aux energy q = Ppsol * t / 1000 * An * alpha*

Of t=1750 h/a

Solarpump P=30+0,05*A= 35 W (1750*35=61 kW)

1,2*(50-Ta)/45 * 300 d/a * qbs

qbs (in kWh/d) = 0,4 + 0,2*POWER(V;0,4)

at 200 litre → 2 kWh/d → 600 kWh/a

NL NEN 5128: Als behoefte/zon = >1,6 → jaarrendement = 40% d.w.z. bij 2200 kWh → 1375 kWh zonopvallend → 2,5 m². Kleinste waarde is 24% bij <0,38

The original plan was to use the prEN defaults, but this would never lead to any contribution of solar to space heating. Especially the default Uc value was very high (collector loop losses) and the Annex A example uses values which are a factor 50 lower (0,2 W/K.m² instead of 9 W/K.m²).

ADDENDUM (7.6.2007)

CH distribution losses in the ECOBOILER model

Following the suggestion of iTG to include vertical piping and make the values more realistic also for existing buildings, we propose to use the findings of the IWU – Darmstadt, which were used also as an input into the German energy certification of buildings ("EnergiePass") and the DIN 18599. IWU presents data not only for new buildings (according to EnEV), but also for existing buildings ("Bestand"). This would allow us to work with lookup values for relevant parameters of the distribution losses, but of course the system temperature Tsys (i.e. the part that can be influenced by the CH-boiler system) should be maintained as a variable in all this.

[„Entwicklung eines vereinfachten, statistisch abgesicherten Verfahrens zur Erhebung von Gebäudedaten für die Erstellung des Energieprofils von Gebäuden" Kurztitel: „Kurzverfahren Energieprofil", IWU-Darmstadt, 2005. http://www.iwu.de/datei/iwu-kurzverfahren_energieprofil-endbericht.pdf]

The table below shows the piping lengths for existing and new buildings. As recommended by IWU, the case with vertical piping in the unheated space ("aussenliegende") was taken as the default. For existing buildings the horizontal piping length is detailed per number of floors.

Table A-2. CH-piping lengths in existing and new buildings Germany (source: IWU Darmstadt)

	Existing buildings (vertical piping in unheated space)							New buildings (vertical piping in unheated space)*			
heated floor area	horizontal pipes no. of floors				vertical pipes	radiator pipes	total	horizontal piping	vertical pipes	radiator pipes	total
	1	2	4	8							
80 m²	35	12			11	17	75	34	8	55	97
120 m²	57	23			17	25	122	36	11	83	130
160 m²	79	35	12		22	33	181	39	15	110	164
240 m²	124	57	23		33	50	287	44	23	165	232
400 m²		102	46	18	56	84	306	54	38	275	367
600 m²		157	74	32	84	126	473	66	56	413	535
800 m²		213	102	46	112	167	640	79	75	550	704
1200 m²			157	74	167	251	649	104	113	825	1042
2000 m²			269	130	279	419	1097	154	188	1375	1717
4000 m²			548	269	558	837	2212	279	375	2750	3404
8000 m²			1106	548	1116	1674	4444	529	750	5500	6779

From the table it can be seen that the ENEV (for new buildings) is indeed generous with piping lengths and we propose to use the piping lengths for existing buildings. For houses with individual boilers (80-150 m²), which are in the size classes XXS-L, it is reasonable to assume 1 m pipe/ m² heated floor area. For small collective boilers (XL-XXL) in houses of around 500 m² the value is around 0,75 m/m². For the larger buildings we can assume a value of 0,55 m/m².

If we assume the overall default heat loss factor hlf of 0,6 this leaves us with the question what is a sensible U-value for the piping. For this we look at the following table from the same source:

Table A-3. Central Heating distribution losses (IWU-Darmstadt 2005)

		no. of floors		
		1	2 to 5	6+
Central heating 1950's-70's	kWh/m².a	27,9	23,9	19,3
Central heating 1950's-70's + renovated	kWh/m².a	21,5	19,5	17,2
Central heating 1980's-90's	kWh/m².a	17,5	15,5	13,3
Central heating newly built (EnEV)	kWh/m².a	9,4	7,0	5,0
Extra per dwelling (collective systems)	kWh/dwelling.a	5,4	5,4	5,4

From this table we have derived the following values for the annual distribution losses for the ECOBOILER model size classes, using rounded values.

Table A-4. Central Heating distribution losses for typical classes (at Tsys = 50°C and Tambient= 20°C)

		no. of floors			size
		1	2 to 5	6+	
Individual system average (75% exist/25% new)	kWh/m².a	19,1	16,5	13,7	XS
Individual system average existing	kWh/m².a	22,3	19,6	16,6	S-M-L
Individual system average new	kWh/m².a	9,4	7,0	5,0	XXS-XS
Collective system 8 apt. (ca. 500 m²)	kWh/m².a	19,1			
	kWh/dwellings.a	43,2			
Collective system 8 apt. existing	kWh/m².a	**22,3**			XXL
	kWh/dwellings.a	**43,2**			
Collective system 8 apt. new	kWh/m².a	**9,4**			XL
	kWh/dwellings.a	**43,2**			
Collective system 32 apt. avg (ca. 2300 m²)	kWh/m².a	19,1			3XL
	kWh/dwellings.a	172,8			
Block heating 4 * 32 apt. avg. (ca. 9200 m²)	kWh/m².a	19,1			4XL
	kWh/dwellings.a	691,2			

And the final step is to derive appropriate values for our equation. For this it is necessary to know that overall IWU calculates with a default system temperature Tsys of 50°C and an ambient temperature in the heated space of Ta=20°C. The heat loss factor (hlf) in the table below is first derived from what we know is the EnEV U-pipe value for new builts (class XXS and XS), i.e. between 0,2 and 0,25 W/m.K which then results in hlf=0,3. The hlf for the collective heating is estimated with the assumption that the piping distributing to each dwelling will also be in the unheated space, which then results in hlf=0,37. If we know the hlf, we can then calculate the values for Upipe in the last column.

Table A-5.Parameters for ECOBOILER per size class

Size	Heated floor area	Qdistr per year per m².a	fixed per a	Qdistr	Lpipe m/m²	Qdistr/Lpipe	Pdistr/Lpipe	Upipe incl. hlf	hlf	Upipe excl. hlf if hlf=0,7
	m²	kWh/m².a	kWh.a (coll.)	kWh/a	pipe m/m²	kWh/m pipe.a	W/m pipe	W/K.m		W/K.m
XXS	78	10	0	780	1	10	2,0	0,07	0,30	0,22
XS	101	10	0	1010	1	10	2,0	0,07	0,30	0,22
S	66	20	0	1320	1	20	4,0	0,13	0,30	0,44
M	86	20	0	1720	1	20	4,0	0,13	0,30	0,44
L	106	20	0	2120	1	20	4,0	0,13	0,30	0,44
XL	624	10	40	6280	0,75	15	3,0	0,10	0,37	0,27
XXL	528	20	40	10600	0,75	28	5,6	0,19	0,37	0,50
3XL	2304	20	160	46240	0,55	36,5	7,3	0,24	0,37	0,66
4XL	9216	20	640	184960	0,55	36,5	7,3	0,24	0,37	0,66

We can summarize the above in the following equation for the ECOBOILER model:

$$Qdistr = 0{,}001 * hrs * F * Lpipe * hlf * Upipe * (Tsys - Ta)$$

Where

- 0,001 is the conversion factor from W to kW
- hrs is the number of hours that the system is running (default heating season is 5000 h/ year)
- F is the heated floor area in m²
- Lpipe is the specific pipe length in meters pipe per m² heated floor area (from lookup table below)
- Hlf is the heat loss factor (default 0,15 for heated space, 1 for unheated space; actually varies between 0,3 and 0,37 see table below)
- Upipe is the specific heat loss per meter pipe in W/K.m (see look-up table below)
- Tsys is the average system temperature in°C (determined by the model)
- Ta is the ambient temperature (default 20°C)

The table below will be used in the ECOBOILER model to find the values for the various size classes (in LOOKUP worksheet, cells D805:F813, furthermore the Lpipe parameter will be added on the MAIN worksheet at and Qdistr will change in MAIN rows 481:484).

Table A-6. ECOBOILER Look-up Table

Size	**Lpipe**	**hlf**	**Upipe excl. hlf if hlf=0,7**
	m/m²		**W/K.m**
XXS	1	0,30	0,22
XS	1	0,30	0,22
S	1	0,30	0,44
M	1	0,30	0,44
L	1	0,30	0,44
XL	0,75	0,37	0,27
XXL	0,75	0,37	0,50
3XL	0,55	0,37	0,66
4XL	0,55	0,37	0,66

ANNEX B: MODEL REVIEWS

Review iTG / EHI

This annex contains the summary of the intermediate review by iTG (21.5.2007) and boiler experts, including the first reaction by VHK

Report:

ITG Dresden,

Institut für Technische Gebäudeausrüstung Dresden

Forschung und Anwendung GmbH

Prof. Richter - Prof. Bolsius - Dr. Felsmann - Dr. Hartmann - Prof. Oschatz - Dr. Werdin

Analysis of the Kemna model as a basis for the labelling of heat generators

Client: EHI

Dresden at May 21, 2007

First comments VHK/ RK

1 Summary

The „ECOBOILER INTEGRATED MODEL" made by VHK to benchmark heating systems is evaluated by ITG Dresden as follows:

Basic estimation

• The methodology of a heating system approach is treated as reasonable since in reality operation of single components (for instance boiler) depend on application conditions of other components and boundary conditions of the whole technical system. This is big difference compared to the performance of components at a test rig. That is why the evaluation of single components needs to take into account the system operation as well.

• The simplified calculation method based on Excel is also seen as reasonable, because

o Main effects can be described in a sufficient matter,

o Calculation procedure is comprehensible and can be arranged relatively clear

o It is not necessary to make time-consuming experiments

o Transient computer simulations are more complex, more irreproducible and more error-prone (although more detailed and more accurate results will be produced).

Agreed, these were exactly our considerations. Thank you for this confirmation.

• **The present draft V5 of the Ecoboiler Model should neither be used for a concluding evaluation of single components or heating systems nor for the labelling of products**, because[61]

o The model offers a number of essential vacancies, inaccuracies and errors (see list of evaluation details)
See below

o The provided benchmark called „CH system efficiency" seems to be unsuitable See below

o A matching with evaluation algorithms of EPBD standards is missing
The documentation contains the references to EPB standards that were used and we will extend/ improve this documentation. We do not have time/ budget to make an extensive report, showing all the (parts of) standards that were not used (and why not).

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[61] Note 1: iTG refers to a version of the model of 21 May, 2007 and not the versions 5a or 5b (2 June) that have been drafted afterwards and published and which have corrected –in as much as was appropriate—most of the "inaccuracies and errors".

Note 2: In the written questions/ comments of the EcoBoiler Workshop 11 Sept. 2007 several times this sentence has been misrepresented e.g. "we support the view of ITG Dresden that such a model should not be used for labelling due to the large number of " inaccuracies and errors". This is misleading, because in the "Basic estimation" iTG largely supports "**such a model**" and their criticism is directed towards "The present draft".

o Not enough time is available for an extensive test as a basis for an objective evaluation. *That is true, but the underlying standards from which the formulas are taken were for the most part already tested. The new parts, i.e. where the standards showed gaps, relate e.g. to controllers, where we did however received confirmation from Danfoss/ Honeywell/ Siemens experts that our results were in line with their research.*

Evaluation details

A) Evaluation parameters

• The definition of an ideal system (needed for the calculation of „CH system efficiency“) is difficult to understand because in fact the system is not ideal in the calculation process but combined with some kind of arbitrary energy losses. As an example the efficiency of an ideal boiler is handled as 99.9% instead of 100%.

The differences between energy requirement for an ideal system and energy demand of the building (Tset or net heat load, respectively) are quite small. The user selection of an ideal system (idealised boiler efficiency but real distribution system) sometime leads to CH system efficiency >100%.

The " ideal system" was used to account for the fact that even the " ideal" CH-boiler system cannot avoid all distribution and stratification losses, because it will also depend on emitters and piping found. The maximum net efficiency for the best boiler (without renewables) can therefore never be more than 88-90%. Optically this makes it not such an attractive number to put on a label., because consumers will always wonder where is the 100% efficient CH-boiler system (which doesn't exist, at least not without renewables)

However, iTG is right that it does make it scientifically more confusing to use this ideal system as a yardstick and not the net heat load of the dwelling. Also it makes it more difficult to feed the efficiency number into the EPBD.

Following also talks with some industry experts, we therefore propose the following:

- *use the net heat load efficiency as proposed by iTG*
- *not to use a numerical scale alongside the " A+++ to G" classification on the label*
- *if we need a number on the label, then it should be clearly be marked that it is not an efficiency number (so without " %") , but instead an index (dimensionless).*
- *The class limit between and " A" and " A+" will not be at 100% net efficiency, but at 88-90% (to be determined in detail, but should be equal to what is the maximum achievable without renewables)*

The evaluation parameter „CH system efficiency“ (Central Heating System Efficiency) highlighted in the user interface NDX is calculated from the relation between Total energy demand of an idealised heating system and the energy demand of a real system including energy losses. This kind of evaluation parameter is uncommon and not qualified for an efficiency analysis because significant physical effects are reflected in an inaccurate way. As an example CH system efficiency is nearly constant if a clearly oversized boiler is used whereas system losses increase. The reason for that effect is that the ideal system is evaluated with an oversized boiler as well and relation, between idealised and real system is not changed. Similar effect can be observed when an auto-timer control will be used which leads to significant energy savings but the evaluation parameter „CH system efficiency“ does not change.

Correct: CH system efficiency should have been fixed.. But if we use the net efficiency, this problem does not arise.

• A more suitable parameter for the evaluation of system efficiency is the relation between annual building heating energy demand (net heat load) and total energy delivered to the system (total energy). This relation is called net efficiency in the Kemna model.

Agreed (see above)

A similar value can also be calculated for a combined heating and domestic hot water system where the energy demand should be independent from the control of the heating system to avoid some noises in the result.

Not agreed. In political discussions, i.e. with people who are not experts, it is widely believed that night- & day-setbacks will result in huge savings. In reality, the savings are relatively modest (but a bit more than " noise"), due to the effect of thermal mass, etc. But the model should show this, otherwise the discussion will continue.

B) Vacancies

• In case of evaluation of systems, a building can be chosen between one-familyhouse and apartment. The thermal building insulation is separated into new and existing buildings. Meaningful parameters vary in closer limits (values of middle Europe)

o heated floor area between 67 m² and 125 m²

o net head load between 2.536 kWh/a (apartment new) and 11.710 kWh/a (house existing)

o heating power (P rad nom) between 2,6 kW and 7,6 kW

For that reason, the model can be used to evaluate heating systems covering one apartment and in one-family-houses. Central heating systems in larger buildings (with two and more individual units) or more powerful components may not be modelled neither from the point of building heating demand nor from the point of specific technical systems (separator with some heating circuits).

This was discussed during the expert meeting. We will expand the range to cover the whole range of boilers (up to XXL, 3XL and 4XL)

• Principally, fan burners are considered. The wide spread atmospheric burners may not be modelled.

True. The reason why we did not model atmospheric boilers in detail (we just gave some rough indication in the efficiencies) is that our Base Case (=avg. EU-25) is already a low-end LT burner. This is the starting point for projections and design options in our reports and we don't have to model downward.

Still, if the Commission decides that the model should be universally applied to labels, then it would be appropriate to also tackle this issue. We will see what we can do (if we have the time)

• The options of boiler temperature control are constricted. Combinations of two room thermostats or the coupled boiler control consisting of a room thermostat and an outdoor temperature control is a widely used technical solution, that may not be modelled.

Incorrect. The combination of a weather controlled Boiler Thermostat and a Room Thermostat can be modeled by changing the parameter " CL" (CELL Y33 of the MAIN worksheet) after you have chosen the " weather ctrl. BT". This is the " RT compensation range" and lets you choose the boiler temperature range (usually 5-10°C) that can be influenced by a Room Thermostat. For instance, if the weather control says that the Boiler Temperature should be 50°C, the CL=5°C and the Room Thermostat says it could be 42°C, then the final Boiler Temperature will be 45°C. If the CL = 10°C, then the Boiler Temperature will become 42°C.

Actually, the addition of a room thermostat to a weather control is most useful if the installer has applied a large " safety" factor (parameters Cpar and Cgrad) for the system. If the system is set correctly (i.e. using a correct Cpar and Cgrad) then the saving effect of adding a room thermostat is negligible (and only extra costs). Therefore we have not made the addition of the room thermostat an explicit design option, but instead have given the option to correctly set Cpar and Cgrad.

Innovative control systems (bus systems, networks of room information, decentralised pumps, etc.) can generally not be evaluated.

The option "motor + CPU" implies that there are local sensors and that the signal of these sensors will be fed to the boiler CPU where there is some intelligence to handle this signal correctly and control the motor-valves (no TRV!) accordingly. As it is modeled today, the saving effect of this option varies according to the rest of the *system. If you already have very good boiler temperature control, then the relative saving will be small with respect of e.g. " motor + PID-loop" , because in fact local intelligence and sensors will do already a very good job. If the boiler system is generally not of the highest quality, then it can have a more significant effect.*

Decentralised pumps cannot be evaluated as such, but the methodology prescribes that if a CH-boiler system will be offered for CE-marking (or labeling) and it does not have a pump, a " reference pump" will be assumed (e.g. fixed speed 90 W),

The model assumes fixed design temperatures of 80/60°C. The design temperatures at all have a great impact on the energy efficiency of the technical system (boiler, distribution losses, performance of control, etc.). This strong impact is completely neglected, while relatively small effects are considered in detail (e.g. start-stop-losses).

I am a bit at a loss where we have explicitly said that we are using a fixed design temperature of 80/60°C. So perhaps it is intended implicitly? We calculate the nominal radiator capacity on the basis of the power output required at –10°C outdoor temperature (see e.g. Cell F144, MAIN worksheet). According to EN 442, a 65/75°C regime (avg. 70°C) is used to indicate nominal radiator capacity. Is this what is intended?

Of course the "design temperature" of a CH-system is important. It directly affects the radiator capacity (size) and thereby the possibility to realize a low temperature regime. And in a full EPB application of this model it could/ should be a third factor (apart from the safety factor and the design outdoor temperature) to determine the radiator capacity. But it is not a factor that can be influenced by the CH-boiler system: The emitters and the piping are assumed as a given and we have chosen with each class (XXS-XS-etc.) an emitter and piping system that we think is an average of what a boiler in that size-class can expect (70-80% existing buildings, 15-20% new). Should this model in the future play a role in the overall EPB of a dwelling, then it can easily be corrected for radiator capacities that are higher or lower than what we have assumed. For instance in class XXS the efficiency becomes 1-2% better if you double the radiator capacity or ca. 2% worse if you half the radiator capacity. But this sophistication is not functional for labeling of heat generators, because –as mentioned—radiators and piping are not part of the boiler system.

• Generally, the model assumes radiators for the heating emission to the space. The wide spread floor heating system (with its great influence on the thermal and hydraulic system behaviour) may not be represented.

Mostly the same answer as above. The emitter system is not part of the boiler-system and we had to choose one (1) reference emitter + piping system, which in this case is a radiator system. We did not choose floor heating systems as the reference, because if you take into account the existing buildings, most boilers will encounter radiator systems (>70-80% of the market). By the way, the correction for a floor heating system can easily be incorporated if ever the model is used in a full EPB package (e.g. we have the equations for the stratification effects, extra pump energy, etc.)

• A circulating system for domestic hot water systems can not be calculated within this model.

True. Hot water circulation systems are not included, because for our current work this is not needed: We only have to model from the BaseCase upwards. In a full EPB application this should be added.

• It should be taken care that all user defined inputs are placed within one single worksheet and marked consistently. A documentation of the program inputs and main calculation procedures is necessarily required.

ITG is absolutely right. Before the model is used for labeling there should be a complete overhaul of the Excel file to make it more linear and clean-up the patches of corrective programming.

C) Errors

• The correlation between CH power class (in worksheet NDX) and dwelling type (in worksheet main) is not correct. The choice CH power class XL is leading to an error, no calculation can be done.

True. To correct.

• The length of heating pipes is calculated according to the German standard DIN V 4701-10. The length of the vertical pipes (s region in DIN V 4701-10) is neglected.

This might be suitable for apartments but is incorrect for a house.

True. On the other hand, also according to the VDI-manual and several experts the DIN 4701-10 is very generous with its piping lengths plus it is based on a typical German house, whereas we have to define an average EU-situation where the boiler is not always in the cellar (but in the attic, cellar, bathroom, etc.) and the piping is not always incorporated in the wall but often in the heated space (in sight).

So, all in all, we think to have found a good compromise for an EU-average piping situation (at least up to class L or XL), which of course could be parameterized once the model is used for a full EPB application.

We do think that iTG has a good point regarding the new boiler sizes of at least XXL, 3XL and 4XL, which are usually collective or commercial boilers with extensive vertical piping. For those we will adapt the model to include the vertical piping.

• The calculated value „combi aggreg. efficiency“ is for CH relating to the total energy of the ideal system and for HW relating to the load (probably because there is no ideal system for HW so far defined). Consequently an incorrect mixed value for „combi aggreg. efficiency“ is calculated.

True. Will be corrected (see earlier remarks)

• For the „combustion air intake” there is a choice of three alternatives:„room sealed”, „open” and „none”. Parameters „open” and „none” deliver exactly same outputs. There seems to be no influences on the calculation.

We will look into that and correct

• If a system without „auto timer control“ is evaluated, the field “electronic optimiser” is invisible, nevertheless the input parameter electronic optimiser yes/no is still used and influences the outputs of the calculation. That makes little sense, because there is no electronic optimiser.

We will look into that and see where this small bug comes from. For clarification:

An “ electronic optimizer” is defined as in the UK: It is a piece of logic that calculates when the boiler should be switched on to reheat the house after a night-setback. But not only that: It calculates this time as a function of the boiler capacity at which the heat generator has the best efficiency (usually the 30% part load). In our model we always assume that the temperature should be back at the old level at a certain time, otherwise we would be comparing different performances (not fair). So in fact, the only thing that the “ electronic optimizer” does is to set the reheat power (MAIN cell Y14): 100% of nominal capacity if there is no optimizer; 30% of nominal capacity if there is an optimizer.

• A comparison of variants XS and S of „Hot water Power Class“ shows some disagreements:

The total energy of the variant S is lower than for XS although higher system heat losses. Distribution and storage heat losses are overcompensated by reduced generator losses (standby, generator). This is physically seen not very reasonable.

We will look into that.

D) Inaccuracies

• The boiler efficiency calculated with the Kemna model for a condensing boiler in an average German new built one-family-house is 83.3% (all numbers related to the gross calorific value). In a wide field test an average boiler efficiency of 86.6% was observed. The calculated boiler efficiency is 3.3% lower then in reality. For low temperature boilers the difference is 0.5% only.

Thank you. We think this is a very good result. We know of lttle models that come so close to real-life.

• A unique primary energy factor of 2.5 is used to calculate the overall energy balance taking into account both heat and electric energy. It is not reasonable to over-simplify the conversion into primary energy due to the fact that the primary energy factor is very different for several European countries. In this context it has

to be mentioned that some inputs of the Kemna model (i.e. building characteristics, climate conditions etc) depend on specific countries. The primary energy factor for fossil fuels is neglected.

The ratio 2,5/ 1 for electricity/fossil fuels is now used throughout the EU, including in the French building regulations (RT2005 uses Cep=2,58). There is a long explanation for this, which has something to do with the marginal costs and carbon emissions of building new power plants, rather than the average environmental costs, but I will spare you that. It is sufficient to say that it is a generally accepted value and the Commission would not appreciate if we would change that, especially not in a pan-EU legislation.

• The weather data underlayed for individual countries apply (as described within the Excel sheet) to the country capital and are not necessarily representative of the country as a whole. This might cause some differences in CH efficiency calculations.

True. But we just need the weather data for a sensitivity analysis. In a full EPB application this can be easily be extended to more or other cities.

• The used phrase „Hot Water Power Class" suggests an engine-power class but only represents an energy related characteristic value that have to be selected.

We are open for suggestions. In our day-to-day work we use " size class" a lot, which is clear (but not scientifically correct). Other alternatives: " Hot Water Load" or " Hot Water Energy" etc..?

• The design temperature of radiators is estimated depending on the morning temperature of a mean day in January. This temperature value is not necessarily the coldest annual temperature (sometimes there are colder days in February). It is recommended do define design temperatures for each country within the Lookup worksheet.

Changing the equation in MAIN!CELLD140 is quicker for us to find the lowest value.

So instead of 1,25(F124-10) we will use 1,25*(MIN(F124:AX124)-10)*

• The selection of "CH Power Class" in the NDX-Worksheet suggests an enginepower class of the heat generator but there is a dwelling type selected only.

See above

• The number of operating cycles of the heat generator is effected by various parameters. The Kemna model only takes into account a few of them as for instance the type of generator control, boiler nominal power, thermal mass of the boiler. The impact of other important parameters as for instance pump speed control, limitation of operating cycles, burner starting power etc is not taken into consideration.

This is a part of the model for which no standards exist, so we have made a model with the minimum amount of equations that will give us a realistic result. In that respect we believe the parameters mentioned by iTG although of course scientifically correct will not have a large impact on the outcome. The burner " soft start" may prolong the cycle time only by seconds (e.g. can be tested by prolonging the purge time), the limitation of the number of cycles in the boiler-CPU is usually an emergency measure which prevents over-cycling (e.g. >12 times/h) and that doesn't occur in the model. The pump speed is still somewhat of an enigma. Everyone expects that the variable pump speed is A Good Thing and I am sure this is true for pumps that are not controlled by the boiler. And even for boiler-integrated pumps there should be hydraulic benefits. But in interaction with the burner we have some question marks…because it is evident that a lower-than-nominal pump speed is increasing the boiler temperature (at the lowest turndown ratio) and this is not a good thing from the exergy point of view. For that reason –although we did model the pump both in fixed and in variable speed—we prefer to use the fixed speed pump.

• The predefined U-value of pipe insulation (0.255 W/mK) is for new buildings according to the German standard DIN V 4701-10 but it is neither typical for average European conditions nor for heating systems in existing buildings.

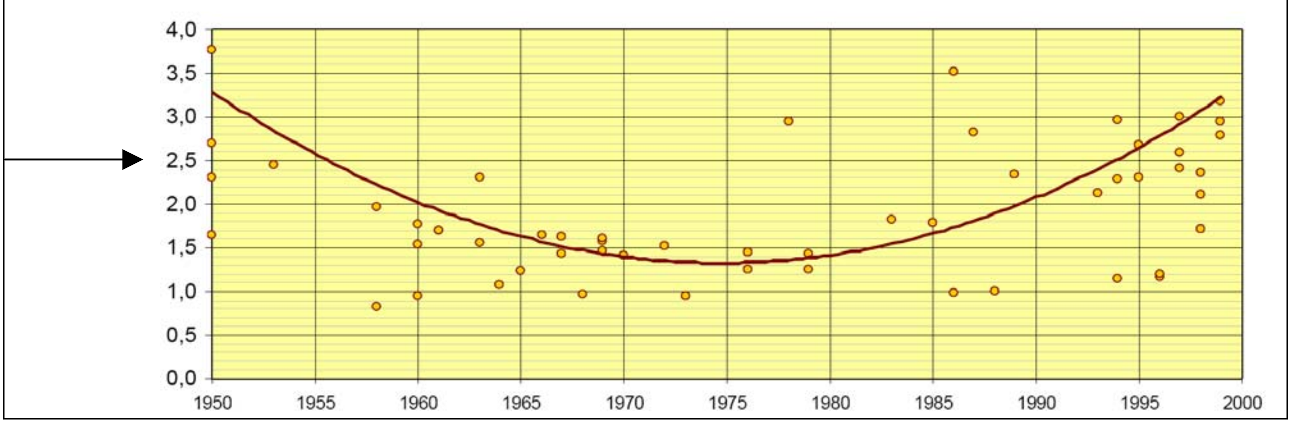
True. But there is very little alternative. We have found no standards or legislation (current or past) that would give us another reference value. Also, we have considered that a part of the distribution losses (i.e. the losses at a minimum required temperature) are beyond the reach of the CH-boiler system designer. And for instance if we had used a high U-pipe value (e.g. 0,5-1) this would have made the distribution losses even more prominent. We are open for suggestions, but we think this to be an acceptable solution for heat generator labeling. Of course, in case of a full EPB application of this model the piping parameters should be treated as variables matching the system of the house.

Normally radiators are oversized compared to the nominal heating power demand. The oversize factor of 2.5 as used by the Kemna model by default seems to be to big and does not represent existing buildings.

The Optimus project found an oversizing factor between 1,5 and 3. Furthermore, in order to compensate for possible LT-dimensioning (45/55 or 28/32 regimes) in new houses we used a slightly higher than average figure.

• Number of days per month is simplifying estimated to 30,5. The use of real number of days per month should be easily possible without any considerably impact on the complexity of the algorithm and clearness of the calculation process but increases accuracy and better fits monthly averaged weather data.

Over-dimensioning of heating system vs. building year of built (OPTIMUS project)



Sorry, but the gain in accuracy is negligible, while the advantages in programming are considerable. For instance, it allows us to calculate annual totals with simple equations instead of –as would be necessary with especially the month of February—lookup tables.

GENERAL COMMENT: I don't see a marked impact on efficiency system values in the case of heat generators having excessive heat input and/or lower modulating range as stated in other documents of preparatory study. On the contrary, if Qb increases (i.e. 100 Kw) also sys eff. Increases.

The fact that sys. eff. increases with increased Qb is definitely an error. We will hope to correct this with the use of the net efficiency. Other than that, we will check again why the correlation is so weak.

Sheet Main / row 36: Selection of open chamber or sealed boiler
If "open chamber" is selected, fan electrical consumption should not be included into energy's calculation. Types B11bs usually do not have it.

As mentioned in our answer to ITG, we have not looked in detail at items that are worse-than-BaseCase. If the model is used for labelling then of course this has to be done.

Sheet Main / row 35: min/fixed Flow

In row 387 it is said: "Conclusion (surprise?): Lower pump speed -> lower el. use but higher Tsysf -> higher radiation losses boiler (in line with Optimus —> 2,5% heating efficiency difference between fixflow and variable speed pump, in favour of fixflow)". Does it means that in general a circulating pump having a fixed speed provides for more energy savings instead of a variable one? It seems also that the presence of high technology room controllers cannot change this aspect.

Strictly speaking, and this is of course a model that looks mainly at what is in the standards, we do not see any big advantage in a variable speed for a boiler-integrated pump. The advantage comes from the fact that usually variable speed pumps are also more efficient pumps. Anyway, for the labelling this will be less important, because we would rely on test results.

Sheet:Main / Row 37: minimal heat input

Have I to mantain 5% of nominal Qb or can I change the value?. According to your knowledge, may I consider 5% as the Best Case in Europe and 30% (ratio 1:5) as base case?

You can change the value. When you play with the model you will find that it is very sensitive to exactly the right value of turndown ratio. We still have to look into that (make it less sensitive). But generally we use 33% (ratio 1: 3) as the base case and 10% as a good design value.

Sheet:Main / row 145: Inheatedspace

Which is the average european value for each type of house/apartment we can select from the calculation?

Currently this parameter depends on the air-intake and volume of the boiler (volumeb), but –as I also announced at the expert meeting—the whole part of the waste heat recovery we still have to bring in line with e.g. the Ecohotwater boiler. So this will change.

Controllers

The system doesn't allow for a combination of weather controller BT + room termostat (on/off or modulating). For what I know, this solution is very common and it would be very useful to evaluate which performance it provides.

See comment on similar question by iTG.

Sheet "Main" cell: X83

Why when space heating is integrated by heat pumps , syst eff falls down?

Probably your " CH-fraction served" is set at 0%, while at the same time 80% of heat pump energy (Ratio CH:DHW) is set to be going to CH. In other words, the model does not automatically give you the right mix of parameters for " CH fraction served" , " Ratio CH: DHW and the heat pump power. This has to be done manually, but I am sure if you play with these 3 parameters you will get a more plausible outcome.

Sheet "Main" cell: X82

Why if space heating is integrated by solar syst eff. Remains fixed?

Because probably your " CH-fraction served" is set at 0%, which means that it is not connected to the CH-system. Try 50% or 100%.

Pittner:

Sheet Indoor Settings

In 3 time profiles there are incorrect data:

4 -Wknd 21/20/24: Check night of “ day zone” and “ night zone”

5 -Wknd 20/19/22: Check all periods of “ night zone

6 -Wknd 19/18/21: Check all periods of “ night zone

will correct

The default calculation mode is "reduced setback", which means that during the low-temperature period the indoor temperature is not allowed to drop under an given value (Where/what is the given value?) Values and zones are given

Sheet Tabelle 1

Comment/change/questions	adress	Bug found in (Draft 4)	Draft 5
Replace "heat loss at burner off" = kW through "% of Qb8060" acc. to EN303	A48:D48		*We can add the % and change the kW values per size-class as a %, but we need the kW value for the equation*
Calculation of SYS-EFFICIENCY seems not correct or not reliable, because relation between overall energie demand and losses. Higher losses mean higher demand, higher demand leads in some cases to a higher efficiency	Z41		not corrected/no change *will be corrected, see also answer to iTG*
Input Pilotflame with default 110 W? Correct calculation (value and steadystate)?? As shown below: max-value = 6588 h/HP * 0,11 kWh = 724,7 kWh/ HP; but results i .e. without pilotflame (steady state!!!) = 722 kWh; standby heat: 154 kWh with pilotflame (110,0 = 28937(!) kWh; standby heat: 154 kWh (?) Why is there an impact only on "steady state" and not "standby"- values?	C49 + output: steady state + standby heat	Unit (input in [W])? Calculation?	not corrected/no change *comma error, will correct and add to “ standby” as appropriate*
Using different Valves/CPU = calculation results not correct!! max. Difference = (6.588 h/hP * 10 W * 2,5)/1000 Wh/kWh = 165 kWh/HP Example: choosing Valve 1-P/Pstandby (10/0 W) = 566 kWh/HP choosing Valve 2-P/Pstandby (10/10 W) = 1031 kWh/HP; Difference: 465 kWh = 456.000 Wh/HP 465.000 Wh/ (10 W * 2,5) = 18600 h/HP (condensing year???:-));	J41 + "output electric"		corrected
Formula: Circulation pump (-) - consumption (=30,5*(SUMME(F620:AX620) - 9*MIN(tpmp;pmpsb)*elpmp*primenergy)); row 620; results (in case of "weather control BT") sometimes in a negative xpump!			not corrected/no change *will check/correct*
Relations between "elc. -off" and pump-power, times & max. values. No difference in "off"-results when choosing "best" or "worst" pump in case of weather controlled Tb.			not corrected/no change *will check/correct*

If choosing "Fixed Tb" -> calculated temperature "bath" in "night" (after Tmass) below outside temperature! (i.e. 1,7°C)	P21	Table: (MAIN AZ259:BE253) wrong place for LOOKUP R188 ! "=MAIN!BE252" is empty (= "0"!)	not corrected/no change *will check/correct*
Explanation in pdf. file of temp setting "wknd" (= simple night-setback 3K) and default-values in "Tset (reduced setback scheme)*" does not corrospond (i.e. choosing "4- wknd 21/20/24" results in "night" 15/15/21 <- 3 K only bath???- choosing 6 - Wknd 19/18/21 results in "night" 15/18,5/18 <- here especially the second value 18,5 is higher then the the setting 18,0?? (look at Sheet settings_indoor_temp)			not corrected/no change *will check/correct*
General: The assumed weather data is not representative for real dynamic behavior as the amplitudes in reality sometimes are bigger and sometimes smaller. The proposed data may be feasible for consumption calculation, but are not so for dynamic behavior, comfort and efficiency evaluation. For the latter purposes, there are not enough days (more than 24, with substaintial variation of solar radiation, are necessary)			*We know that in theory that a space heating system should be able to react when there is a sunny spell on a cloudy day or a cloud on a sunny day. E.g the UK SAP corrects for this, but the effect is very small (or not there in case of floor heating systems), mostly due to the thermal mass. Given that we also use an emitter sytem with considerably water content (LT), I think we can rightfully ignore this influence. Of course, if this model is applied to full EPB with quick-acting emitter-systems, you can make an extension for this. But it will not be easy…. Do you have more data on this effect that would allow better modelling?*
General: Especially implemented control algorithms improving boiler operation cannot be considered yet. A way to acknowledge them has to be found. Examples: - Algorithms that switch off the pump for some times when burner is off are not considered. - Special antipendel algorithms changing cycle rates and feed temperature levels. - Algorithms compensating room sensor time constants (smaller time constants could be assumed or even measured.)			We did take pumps with night-setback into account; they are incorporated in the 2 most efficient types. *The anti-pendel algorhytms we know are emergency measures to stop the boiler from cycling e.g. more than 10-12 time per hour, which doesn't occur in the model. Could you explain?* When using a " modulating thermostat" (or better) we assume the handling of time constants (UK: " room temperature compensator") to be standard
General: If a supplier claims to use one of the considered control mechanisms, this is always handled the same although there might be "better" or "worse" ways to do it (for example with different algorithms in controls)			*True. Although the OEM-experts from Danfoss and Honeywell seemed happy enough with our selection, this is exactly one of the main reasons that a test/emulation procedure needs to be developed, at least up to class L or XL, i.e. to capture future innovations. But this will take time and until such an EN standard is ready, we can use this in the interim.*
Is it correct that for weather controlled systems totally open thermostatic valves are assumed (that is 25°C during day/21°C during night) - or only in heat-up time? In the event of the first case this is not really feasible, and denies objective comparisons.			*No, that is not correct. The model only assumes that the boiler temperature lay-out with a weather-controlled system must be such that a temperature of 25°C (valve fully open) can be realized, but the real room temperature depends on the TRV position*

General: on the one hand, the short deadlines for evaluating of the model are obvious (as mentioned i. e. in the minutes of the last meeting), but on the other hand missing calculations of existing technologies like different controller-solutions are not acceptable, especially if the sheet will be the base-case for comparisation of systems, for the recommandations to the EC and for labeling of systems.			*The control solutions that we have elaborated are the ones in the EPB standards (no more no less) and cover current situation. As mentioned, for the future (Jan. 2013) we see that for individual boilers (up to XL) tests/procedures should be developed. Furthermore, if a very new control has been proven to be superior in efficiency and it cannot wait until the test procedure is there, then the Commission can amend the legislation (e.g. in line with amendments to EPB standards).*
General: How will be work this in future? How should industrie show the energy-saving impact of i.e. current or future (not-in-the-sheet) control solutions and other developments?			See above
General: impact of changing fuel (i. e. oil-boiler has no "valves", but a higher fan-electricity demand a.s. o) uncalculable			not corrected/no change *fuel influences dewpoint + electricity consumption in on-mode from oil pump etc. will show from test results*
Selection of "load pattern" = 0 (CH only) -> caculation of "%" in "output-heat energy" = #NV	Sheet "Main", Cell T59		only draft 5 (AH_EUP_02_07_lot_1_B_Integtd_model_dwell-inst_draft_V5.xls) *will correct throughout*
Changing "heat traps dhwtrap ?" from "YES" to "NO" - after that only "NO" available	Sheet "Main", X 94		only draft 5 *will correct throughout*
Efficieny of hot water heating depending on controller? (i.e. "Fixed BT" = 42%; "weather-control": 33%; "on/off" = 41; "modulating" = 36%; "time-propotional" = 36%); Standby-heat losses for "weather-control" twice as much as in case of "fixed BT"??			only draft 5 *will correct throughout*
Relating to case above: the default-values for "weather-control" changed from Cpar = 1 -> 0; C grad = 1,5 -> 0 Why that, why without any documentation?? In calculation this results to different temperatures a. s. o….-> lower losses a.s.o			only draft 5 *will correct throughout*

ANNEX C: BOILER MODEL NOTES

Boiler Modelling Notes

This annex contains the documentation of the first version of the separate Boiler model. The separate model was later incorporated in the ECOBOILER Integrated Model, where it was then altered, debugged etc.. Nevertheless, although it is not a 100% compliant with that model it provides an insight into how several equations were arrived at.

It is added here as a general background information.

Outlines of the model

As mentioned, the model starts from the work that is being done within CEN/TC 228 and laid down in the prEN 15316 –series. If another approach is used this will be explicitly mentioned and explained. Another goal is to use the figures that are available from official tests as much as possible. Additional data that is necessary to asses the real life annual efficiency can either be obtained through manufacturer declaration (technical datasheets), or – if these are lacking - default values are used.

As in the Task 3 Heat Balance, the Boiler Model uses a monthly approach, but starting from an average month-day with 5 day-periods. This time-step is still sufficiently large to contain the number of data and calculations in a non-numerical method, although it is sufficiently small to go beyond e.g. prEN 15316-4-1 (dealing with efficiency of heating generation systems) and provide a credible assessment of:

- the actual system return temperature
- The actual number of cycles and their duration

The Boiler Model uses an average house that is split-up in three different zones (living &dining, bed, bath) and has five different periods (morn, mid, eve, late, night). See table below for an example of such a data set (temperatures mentioned in this table are exemplary figures) that could also be quite feasible for Excel-type calculations.

Table 11.1

Zones	**Periods**				
	Morn: 7-9	Mid: 9-16	Eve: 16-21	Late: 21-23	Night: 23-7
Zone 1 (living & dining): 50%	21°C	21°C	21°C	21°C	15°C
Zone 2 (bed): 40%	15°C	15°C	20°C	15°C	15°C
Zone 3 (bath): 10%	24°C	21°C	21°C	24°C	21°C
Average indoor temp: 100%	18,9°C	18,6°C	20,6°C	18,9°C	15,6°C

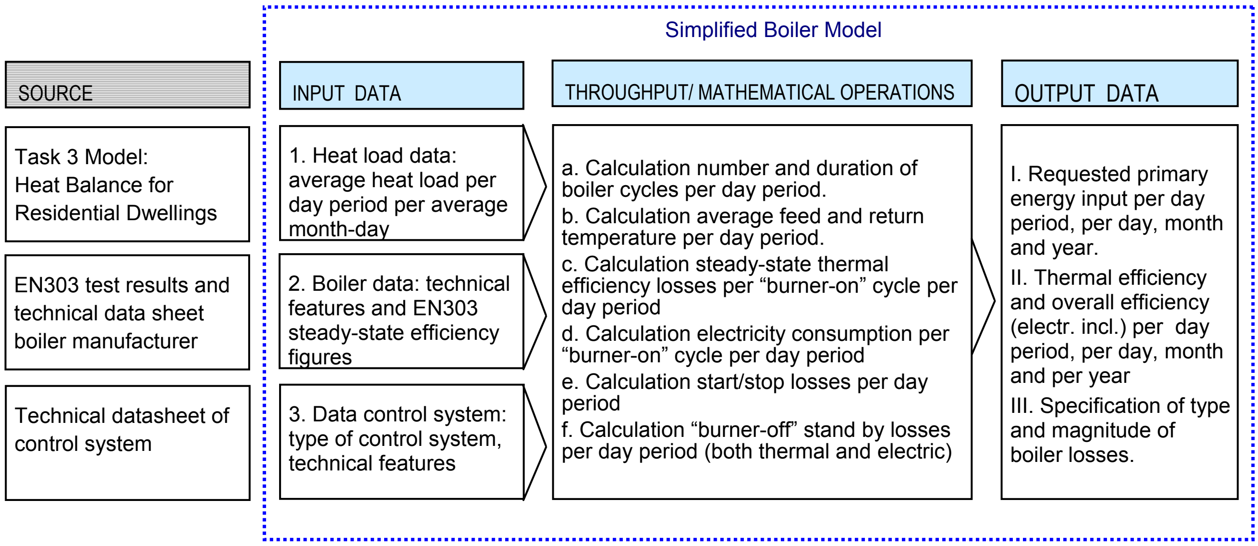
The Heat Balance (Task 3) has look-up tables for each month on the average outside temperature and the global solar irradiance per day period for EU-25 average and each Member State's capital. Also average geometrical and construction data for average dwelling, house and apartment come from statistics (e.g. Boverket). Several indoor temperature regimes can be tested (example above), control, emitter and distribution loss parameters can be defined. The radiator lay-out and capacity per temperature zone are calculated (from a design outdoor and indoor temperature, e.g. –10 and +21°C) also taking into account a safety (oversize) factor. From all this, the Heat Balance calculates the 'unavoidable' losses and creates the inputs for the Boiler Model:

- The initial heating demand (QH) for each day-period in each average month-day,
- The initial average 'system temperature' (Tsys) needed to meet the heat demand with a certain control strategy.

Please note, that the initial average 'system temperature' (Tsys) depends on the control strategy employed (fixed, outdoor sensor, central room-thermostat, multi-zone temperature control, etc.) and is primarily used in the Heat Balance to determine the distribution losses. In the Heat Balance, the Tsys is always assumed to be a continuous temperature, supplied by a boiler with unlimited modulation possibilities. Only in the Boiler Model the restrictions of the boiler come into play, i.e. the fact that modulation is restricted and yields (partially) on/off behaviour with the accompanying losses of heating up and cooling down the thermal mass of the boiler water, piping, etc..

The diagram below (figure 11-2) gives a schematic impression of the Boiler Model.

Figure 11-2. Schematic representation of Simplified Boiler Model



A prototype of such a model is built in Excel and shall be distributed to the members of the expert group for further optimization.

Input data

Data boiler

The basis for the Boiler Model are the results from the EN product test standards (EN 303 etc.), as described in Task 1.

These existing test standards, of which the industry has already indicated that they will not be changed in the short term, distinguish three 'types' of boilers, in accordance with the BED (92/42/EC): 'condensing', 'low-temperature' and 'standard' boilers. And it produces 'full-load' and 'part-load' (30%) energy efficiency values for these types. Especially the values for 'part load' are aggregated values and can also be composed, e.g. through an indirect method. The deficiencies of this procedure have been discussed in Task 1.

But what is often forgotten, is that the test standards also prescribe the test procedure for the basic energy and performance tests:

- Nominal useful output and minimal useful output Pbnom and Pbmin in kW at 80/60°C and preferably also at 50/30°C
- Nominal heat input and minimal heat input Qbnom and Qbmin in kW.

The ratio of the two gives the steady-state boiler efficiency at maximum and minimum boiler load (not necessarily 30% load) at a 80/60°C temperature regime and for most boilers also at the 50/30 oc temperature regime. These steady state efficiency figures include

- envelope losses ('radiation losses'),
- flue gas losses,
- losses due to unrecovered latent heat and
- the useful heat input coming from the electric components (pump, fan) *during burner on phase*.

Not included is e.g. the electricity consumption.

These are mandatory values (with the exception of 50/30°C tests, but these are often given) and can be found for every boiler. The efficiency values found as a ratio provide technically more plausible outcomes for the efficiency and performance than the BED-values and could be a basis for measures.

Example

Parameter	***Symbol***	***Unit***	***Value*** (exemplary figures)
Nominal input	Qbnom	[kW]	24,00
Minimal input	Qbmin	[kW]	9,24
Nominal output at 80/60	Pbnom80/60	[kW]	20,88
Minimal output at 80/60	Pbmin80/60	[kW]	8,00
Nominal output at 50/30	Pbnom50/30	[kW]	22,84
Minimal output at 50/30	Pbmin50/30	[kW]	8,80
These exemplary input and output figures lead to the following steady-state efficiency figures			
Full load steady-state efficiency at 80/60	ηbnom80/60	[%]	20,88/24,00 = 87,0
Min load steady-state efficiency at 80/60	ηbmin80/60	[%]	8,00/9,24 = 86,58
Full load steady-state efficiency at 50/30	ηbnom50/30	[%]	22,84/24,00 = 95,17
Min load steady-state efficiency at 50/30	ηbmin50/30	[%]	8,80/9,24 = 95,24

Technical features boiler appliance

Important drawback of the standardized method for determining part load efficiency is the fact that the influence of controls on start/stop behaviour and the related stand-by losses to a large extend are neglected. Furthermore, because a fixed (and optimised) flow is applied, the influence of varying flows on the return temperature and with it on boiler efficiency, is not taken into account. Finally, the figures that are acquired through this method are relatively high and do not always reflect the physical limitations of the heat balance of a boiler system. In other words, with the Pnom, Pmin and Qnom, Qmin we can solve a part of the problem (i.e. the efficiencies at steady-state boiler operation), but there are other parts where the Boiler Model will have to provide guidance.

The table below gives additional information about the boiler that is necessary to be able calculate:

- the return temperature (and with it the actual steady-state efficiency) per period
- energy losses during the burner-off phase.

Parameter	***Symbol***	***Unit***	***Value*** (exemplary or default value)
Boiler type	Btype	-	condensing | low temperature | regular
Fuel	fuel	-	natural gas | oil
Burner control	Brn;ctrl	-	on/off | high/Low | modulating
Ignition	Ign.	-	pilot flame | electronic
Power in case of pilot flame	Pign	[kW]	0,085 | 0,110 | 0,135
Boiler control band	bband	[°C]	8
Duration pre-purge cycle	tpurge	[s]	30
Lambda at full load	λnom	-	1,3
Lambda at minimal load	λmin	-	1,45
Type of pump	Pumptype	-	fixed speed | variable speed
Power consumption pump	Ppmp	[kW]	Power at nominal flow | power range
Nominal pump flow (at 2,5 kPa)	qnom	[l/h]	1000
Pump operating time after burner switch-off	tpump	[s]	240 | 360 | 480 | 600 | 86400
Reduced pump operation night setback	Pmpred	-	yes | no
If Pmpred=yes, Nbr of pmpcycles/hour	Npmpred	-	1
Boiler weight (empty)	mb	[kg]	68
Water content boiler (CH-only)	Vb	[l]	4
Electricity consumption in standby mode	Pbstby;el	[kW]	0,012
Electricity consumption at Pmin	Pbmin;el	[kW]	0,09
Electricity consumption at Pmax	Pbmax;el	[kW]	0,12
Standby burner-off heat losses acc. EN303	Pbstby;h	[kW]	Measured value | default value acc. prEN 15316-4-1

Average ambient temperature boiler room	Tau	[°C]	15
DHW *	DHW	-	No | Yes
If DHW = yes	DHWtype	-	Instantaneous | internal storage | external storage
If DHW = yes	VDHW	[l]	Water content of the storage or DHW-HE
Keep hot facility DHW	DHWcomf	-	yes | no

* data that is needed for calculating real life efficiency of combis (Eco-Design Lot 2)

Most of this data can be retrieved from the technical datasheet manufacturers provide together with their product documentation. For data that is not provided, default values will be proposed, following the values that are mentioned in the prEN 15316-4-1.

Data room temperature control system

Data related to the room temperature control system are necessary in order to be able to asses the number of boiler cycles, the duration of the boiler cycles, the related boiler temperatures and the remaining burner-off and pump-off periods. In relation to this the average standing-state efficiency can be determined and burner-off standing losses can be calculated as well as the start/stop losses.

The following data are required:

Feature	*Symbol*	*Unit*	*Value* (exemplary or default value)
Type of Room temperature control system	CTRLtype		I : Fixed boiler temperature
			II : Weather dependent control
			III : Room thermostat control
		to be added later	*V : Multiple room thermostats controlled system*
			VI : Individual room temperature control (multizone)
In case of control type I			
System feed temperature	$T_{sys;feed}$	[°C]	70
In case of control type II			
Parallel heat curve correction installer	C_{par}	°C	1
Gradient heat curve correction installer	C_{grad}	-	1,5
Formula for resulting heat curve	Heatcurve	-	$T_{sysf} = T_{set} + C_{par} + 50 (C_{grad} * QH / Pradnom)$ ^1/n + 0,5 * dT_{des}
Room temperature (load) compensation	RTcomp	[°C]	Max. T-range that is influenced by compensator
In case of control type III			
Place of room thermostat	RT*pos*	-	Zone1 | Zone 2 | Zone 3
Type of RT	RTtype	-	Mech. on/off | Electr. on/off | Modulating
Resulting T-swing in indoor temperature			
If mech. on/off RT.: Hysteresis	band	[°C]	± 0,75 | ± 1,0 | ± 1,25 |
Electr.on/off with PI-algorithm: Droop	band	[°C]	| ± 0,4 | ± 0,5 | ± 0,6 | ± 0,7 |
Electr. on/off with TP: Nbr of cycles/hour	Nrt	[°C]	3 | 4 | 5 | 6 |
Modulating RT	band	[°C]	± 0,15 | ± 0,20 | ± 0,25 | ± 0,30 |
Reaction time RT	*t*delay	[s]	300
Optimizer function (multiplier reheat period)	*f*opt	[-]	1 | 1,5 | 2,0 | 2,5 | (value 1 : no optimizer)
Clock program	RTclck	[-]	yes | no

Data emitter system

Data related to the emitter system is necessary for the calculation of the average heating system water temperature.

Feature	*Symbol*	*Unit*	*Value (or default value)*
Total nominal output emitter system	Pradnom	[kW]	18,6
Nominal output emitters zone 1	Pradnom1	[kW]	8,6

Nominal output emitters zone 2	Pradnom²	[kW]	6,0
Nominal output emitters zone 3	Pradnom³	[kW]	4,0
Design system feed temperature	Tsysfdes	[oc]	75
Design delta T	dTsysdes	[°C]	10
Emitter constant	Radc.	-	1,3
Water content of the emitter	Vrad	[l]	$P_{emit;nom}$ * 20 = 372
Water content of circulation pipe circuit	Vdistcirc	[l]	L1 * A1 = (28 +0,05*A) * A_{pipe}
Water content radiator pipes	Vdistrad	[l]	L2 * A2 = (0,515 * A) * A_{pipe}
Bypass	Bypass	-	no

Mathematical operations boiler with weather dependent control

Calculation system feed and return temperature

First step is the calculation of the system feed temperature according to the heat curve that is applied (by the installer).

Formula [1]

$$T_{sysf} = T_{set} + C_{par} + 50\,(C_{grad} * Q_H / P_{radnom})^{1/n} + 0{,}5 * dT_{sysdes}$$

Where

- T_{sysf} is the CH system feed temperature [°C]
- T_{set} is the average indoor set temperature [°C]
- C_{par} is the parallel correction on heat curve made by the installer [°C]
- 50 is the dT between ambient and average radiator temperature (acc. EN442)
- C_{grad} is the gradient correction on the heat curve made by the installer [-]
- Q_H is the gross heat load of the house [kW]
- P_{radnom} is the nominal total capacity of the emitter system [kW]
- n is the emitter constant [-]
- dT_{sysdes} is the design delta T of the for the radiator system [°C]

Please note that values used are averages for a certain day-period (see Excel file).

For weather controlled systems, T_{set} will be 25°C during daytime (and 21°C during night setback) because radiators with the thermostatic valves in position 5 are supposed to achieve indoor air temperatures of at least 25°C.

With this calculated system feed temperature (Tsysf), the actual Tsys is calculated (based on the actual pump flow) and and iterative calculation is made for the control losses (dTfluct) the emitter efficiency losses (dTstrat) and the distribution losses (dTdistr). The related calculation method is explained in the documentation of the Heat Balance Model (Task 3). The results of these iterative calculations give us the final the load of the house (QH), compensated for the specific settings of the heat curve.

Now that we know the Tsysf and the corrected heat load of the house, we can calculate the system return temperature Tsysr that corresponds with the nominal flow in the system.

Formula [2]

$$T_{sysr} = T_{sysf} - Q_H / \rho * c * q_{pmp}$$

Where

- T_{sysr} is the CH-system return temperature [°C]
- T_{sysf} is the CH system feed temperature acc. formula [1] [°C]
- Q_H is the gross heat load of the house [kW]
- ρ is the specific mass of water (=1000) [kg/m³]
- c is the specific heat water (4186) [J/kg.K]
- q_{pmp} is the flow through the system [l/h]

Please note that values used are averages for a certain day-period (see Excel file).

Calculation boiler output

The average boiler load needs to be determined to be able to calculate the duration of the average boiler-on cycle. It is assumed that, given the technical limitation of the boiler concerning burner control, the boiler output that is closest to the heat load of the house will be the average boiler output per period. For the three different types of burner control options the following formulas can be applied.

For modulating burner control

Formula [3]

if $P_{bmin80/60} > QH > 0$ $\quad Pb = Pbmin8060 + (Tdew - Tsysr) / (Tdew-30) * (Pbmin5030 - Pbmin8060)$

if $P_{bmin80/60} < QH \leq P_{bnom}$ $\quad Pb = QH$

For high-low burner control

Formula [4]

if $P_{bmin80/60} > QH > 0$ $\quad Pb = Pbmin8060 + (Tdew - Tsysr) / (Tdew-30) * (Pbmin5030 - Pbmin8060)$

if $P_{bmin80/60} < QH \leq P_{bnom}$ $\quad Pb = Pbnom8060 + (Tdew - Tsysr) / (Tdew-30) * (Pbnom5030 - Pbnom8060)$

For on-off burner control

Formula [5]

$$Pb = Pbnom80/60 + (Tdew - Tsysr) / (Tdew-30) * (Pbnom5030 - Pbnom8060)$$

Where

- P_b is the average instantaneous heat output of the boiler [kW]
- $P_{bmin80/60}$ is the minimal boiler output at 80/60°C[kW]
- $P_{bnom80/60}$ is the maximal boiler output at 80/60°C [kW
- $P_{bmin50/30}$ is the minimal boiler output at 50/30°C [kW]
- $P_{bnom50/30}$ is the maximal boiler output at 50/30°C [kW
- Tdewp is the dewpoint of the flue gasses [°C]

- QH is the gross heat load of the house [kW]

Please note that values used are averages for a certain day-period (see Excel file).

The second part of the formula "(Tdew – Tsysr) /(Tdew-30) * (Pbnom5030 – Pbnom8060)" is in fact a correction on the nominal or minimal heat load at 80/60-regime. This part of the formula compensates for changing boiler outputs at lower system return temperatures, due to the use of latent heat. For non-condensing boilers this part of the formula is negligible.

Example:
A 24 kW modulating condensing gas boiler with a nominal heat output of 20,88 kW at 80/60-regime can be modulated back to 8 kW (also at 80/60-regime). So Pbmin80/60 = 8,00 kW. At a temperature regime of 50/30 the minimum boiler output is 8,80 kW (Pbmin50/30 = 8,80 kW). Formula 3.1a gives the following result when the system return temperature (Tsysr) is e.g. 40°C:
Pb = 8,00 + (52 – 40)/(52-30) * (8,80 – 8,00) = 8,44

Calculation boiler cycle time

The duration of the burner-on cycle is determined by calculating how long the boiler must operate at the previously calculated boiler load ***Pb*** to increase the system feed temperature with a value equal the total boiler control hysteresis (Hb). When this is achieved, the boiler switches off again until the system temperature has dropped with the same value (Hb), due to the continuous heat load of the house. Contrary to the Boiler Cycling method (prEN 15316-4-1) that calculates the yearly total of boiler-on time, we here estimate the average boiler-on time of each individual cycle per period. The following formula is used:

Calculation burner-on cycle

Formula [6]

if Pbo > QH

$$t_{cycon} = ((V_{rad}+V_{distrad}) * f_{circ} + V_{distcirc} + V_{blr}) * bband * \rho * c/((P_b - QH)*3600)$$

if Pbo = QH

$$t_{cycon} = hrs * 3600$$

Where

- t_{cycon} is the duration of the boiler-on cycle [s]
- V_{rad} is the water content of the emitter system [l]
- $V_{distrad}$ is the water content of radiator pipes [l]
- $V_{distçirc}$ is water content of circulation pipes [l]
- *fcirc* is the fraction of the radiator water content that is actually circulated [-]
- *bband* is the total boiler control hysteresis [°C]
- hrs is the duration of the related day period [h]
- QH is the gross heat load of the house [kW]
- ρ is the specific mass of water (=1000) [kg/m³]
- c is the specific heat water (= 4186) [J/kg.K]

In the same way, the duration of the boiler-off cycle is calculated. The thermal energy stored in the water of the CH-system is transmitted with an average rate that equals the heat load.

Calculation burner-off cycle

Formula [7]

if Pb > QH

$$t_{cycoff} = ((V_{rad} + V_{distrad}) * f_{circ} + V_{distcirc} + V_b) * b_{band} * \rho * c / (QH * 3600$$

Otherwise

$$t_{cycoff} = 0$$

Where

- t_{cycoff} is the duration of the boiler-off cycle [s]
- V_{rad} is the water content of the emitter system [l]
- $V_{distrad}$ is the water content of radiator pipes [l]
- $V_{distçirc}$ is water content of circulation pipes [l]
- f_{circ} is the fraction of the water content that is actually circulated [-]
- V_b is the water content of the boiler [l]
- b_{band} is the total boiler control hysteresis [°C]
- QH is the gross heat load of the house [kW]
- ρ is the specific mass of water = 1000 [kg/m³]
- c is the specific heat water (=4186) [J/kg.K]

The total cycle time "t_{cyc}" thus becomes:

Formula [8]

$$t_{cyc} = t_{cycon} + t_{cycoff}$$

Calculation number of cycles

By dividing the duration of the day-period by the boiler cycle time, the total number of cycles per period can be calculated.

Formula [9]

$$N_{cyc} = hrs * 3600 / t_{cyc}$$

Where

- N_{cyc} is the number of cycles per day-period [-]
- hrs is the duration of the related day-period [h]
- t_{cyc} is the duration of the total boiler cycle time [s]

Calculation average steady-state efficiency and related losses

The average steady-state efficiency per burner-on cycle is calculated from the efficiency anchor points that are derived from the test data figures $Q_{blr;nom}$ & $Q_{blr;min}$ and $P_{blr;nom}$ & $P_{blr;min}$ measured at a temperature regime of 80/60 and for condensing boilers often also at a temperature regime of 50/30. The efficiency anchor points for condensing boilers are constructed in the following manner (see table below).

	80/60°C regime	50/30°C regime

$\eta bnom$	$Pbnom80/60$ / $Qnom$	$Pbnom50/30$ / $Qnom$
$\eta bmin$	$Pbmin80/60$ / $Qmin$	$Pbmin50/30$ / $Qmin$

For low temperature boilers the anchor points at the 50/30 temperature regimes will in most cases not be available. However, for LT-boilers these figures are less important than they are for condensing boilers, because the system return temperature does not have a big influence on the steady state efficiency of these boilers.

For standard boilers the situation is similar.

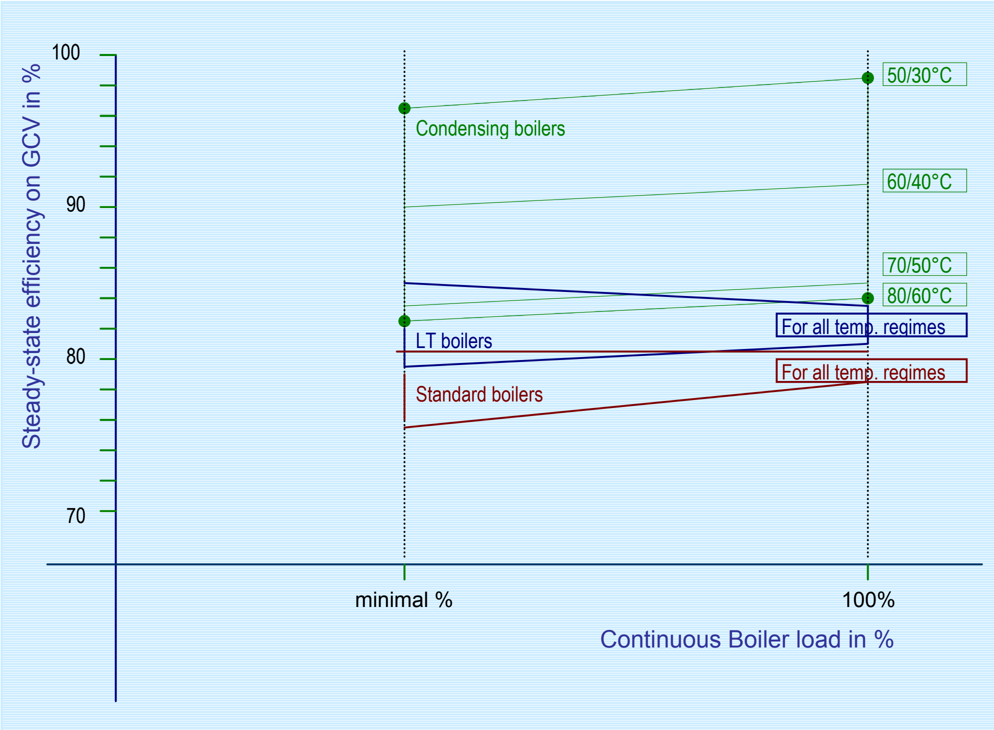
If the boiler cannot modulate nor has a reduced step in heat output, there will only be one efficiency anchor-point: the efficiency figure at full load.

For the various kinds of boilers these steady state efficiency figures will roughly range within the following values (on GCV):

	$\eta bnom80/60$	$\eta bmin$*[62]$80/60$	$\eta bnom50/30$	$\eta bmin$*[1]$50/30$
Condensing	84 – 88 %	83 – 87 %	94 – 98 %	93 – 97 %
Low temperature	80 – 83 %	79 – 85 %	-	-
Standard	78 – 81 %	76 – 79 %	-	-

[62] Only available when the boiler has the capability to operate at a lower load

In graph-form:



The graph illustrates that for condensing boilers the system return temperature is needed to determine the steady-state boiler efficiency. The improvement of the steady-state efficiency is the biggest at return temperatures below 52°C (dewpoint for gas boilers with lambda of 1,3). For LT- and standard boilers the system return temperature is of little influence. The load factor could have a small influence on the steady-state efficiency figures and shall be incorporated in the calculations as well.

Formula [10]

$$\eta_b = (P_{bnom80/60} / Q_{bnom}) + C_{sysr} + C_{load}$$

With, if Tsysr < dewpoint

$$C_{sysr} = (T_{dewp} - T_{sysr}) / (T_{dewp} - 30) * (\eta_{bnom50/30} - \eta_{bnom80/60})$$

Otherwise Csysr = 0

$$C_{load} = (P_{bnom80/60} - P_b)/(P_{bnom80/60} - P_{bmin80/60}) * (\eta_{bmin80/60} - \eta_{bnom80/60})$$

Where

- η_b is the average steady-state efficiency per burner-on cycle [%]
- C_{sysr} is the correction on efficiency for actual system return temp [%]
- C_{load} is the correction on efficiency for actual boiler load [%]
- T_{dewp} is dewpoint of the flue gasses (relates to the air factor) [°C]
- $\eta_{bnom80/60}$ is steady state efficiency at max load and 80/60 T-regime [%]
- $\eta_{bmin80/60}$ is steady state efficiency at min. load and 80/60 T-regime [%]
- $\eta_{bnom50/30}$ is steady state efficiency at max load and 50/30 T-regime [%]

- $\eta_{bnom50/30}$ is steady state efficiency at min load and 50/30 T-regime [%]

Please note that values used are averages for a certain day-period (see Excel file).

The calculation method that is applied here is in principle similar to the method proposed in the "Case Specific Boiler Efficiency Method" from the prEN 15316-4-1. Main difference lies in the fact that the linear interpolation is now only applied for steady-state burner-on situations, meaning that the losses caused by cycling and the losses during burner-off periods are valued separately. Another important difference is the fact that burner-on efficiency is now assessed for the heat load of each individual average day period per month, instead of for an aggregated average heat load per year. As such the method proposed here is expected to be more accurate.

The method proposed in the "Boiler Cycling method" of the same prEN - where the condensate production at 50°C and 30°C system return temperature is necessary to compose the actual condensate production and related efficiency - is not used here, simply because these data are generally not available. In case for a condensing boiler the output data at 50/30°C (min and nom load) are not available, default values could be based on the default figures for condensate production as mentioned in table c.13 of informative Annex C of the prEN 15316-4-1.

Please note that this method for determining the steady-state efficiency per cycle, does not incorporate efficiency losses that are caused by an operating mode in which fan and pump are continuously running, and only the burner is cycling on and off. In such operating modes the fan will cause additional heat losses (cooling down of heat exchanger).

Steady-state efficiency losses

Now that we know the average steady-state efficiency that is achieved during each burner-on cycle, we can calculate the steady-state efficiency losses. These losses include envelope losses, flue-gas losses, unrecovered latent heat but they include also the thermal gains coming from the electric components (fan, pump) during burner-on phase.

The first formula ([11] calculates the total heat output in kWh per period which can serve as a final check on whether the heat load of the house is actually covered by the output of the boiler. The second formula [12] the total calculates the steady-state efficiency losses in kWh per day period.

Formula [11]

$$\mathbf{P}_b = P_b * t_{cycon} * N_{cyc} / 3600$$

Formula [12]

$$\mathbf{Q}_{blosseff} = \mathbf{P}_b / \eta_b - P_b$$

Where

- $\mathbf{P}_b$ is the heat output of the boiler per day-period [kWh]
- P_b is the average boiler load (output) per burner-on cycle in [kW]
- t_{cycon} is the duration of the average burner-on cycle [s]
- N_{cyc} is the number of cycles per day period [-]
- $\mathbf{Q}_{blosseff}$ is the steady state boiler efficiency loss per day-period [kWh]

Calculation start/stop losses

Start/stop losses occur because of pre-purging the boiler and flue-system en because of the fuel losses that are inherent to a start-up cycle (see Task 4 Chapter 2). For gas boilers these losses are calculated separately with the next two formula's.

For oil boilers other formulas need to be used (TO DO)

Calculation of pre-purge losses gas boilers

The total amount of pre-purge air per period is calculated. The heat capacity of this total amount of air is calculated on the basis of the difference between the average system temperature and the outside temperature. After all, the combustion air will always come from outside, either direct or indirect (through ventilation of the boiler room).

Formula [13]

$$Q_{losspurge} = t_{purge} * Q_{bmin} * (0{,}1 + \lambda_{min})/3600 * \rho * C * ((T_{sys} - T_{out}) * N_{cyc}/3600$$

Where

- $Q_{losspurge}$ is the heat loss due to pre-purge cycles day-period in [kWh]
- t_{purge} is duration of the pre-purge cycle [s]
- Q_{bmin} is the minimal boiler input in [kW]
- 0,1 is the relation between kWh fuel input and amount of gas in m^3
- λ_{min} is the air factor at minimal boiler load [-]
- ρ is the specific mass of air = 1,25 [kg/m³]
- C is the specific heat of air = 1000 [J/kgK]
- T_{out} is the related outdoor temperature (combustion air from outside) [°C]

Calculation of fuel losses gas boilers

According to Pfeiffer (Task 4, chapter 2), around 5% of the fuel is lost when the total number of cycles per year is around 14.000.

Formula [14]

$$Q_{lossfuel} = Q_b * 0{,}005 * N_{cycyear} / 14000$$

Where

- $Q_{lossfuel}$ is the fuel losses due to pre-purge cycles day-period in [kWh]
- Q_b is the energy content of the fuel input per day-period [kWh]
- $N_{cycyear}$ is the total number of cycles per year

Calculation burner-off standby heat losses

In standby mode (burner and fan is switched off) the boiler loses heat through envelope (radiation) and chimney. The “Indirect Method” for determining useful part load efficiency of EN 303 describes the test method for determining the standby heat losses of a boiler. For standard boilers the average system temperature Tsys is brought to a value of 30°C above ambient the temperature (which is 20°C), for LT- and condensing boilers the temperature rise is 20°C above ambient. Under these conditions the standby heat losses of the boiler appliance are measured, which in fact means that the quality of the envelope/boiler-insulation and the effect of a possible flue-valve is measured.

The standby heat losses at the actual ambient and system temperature can be calculated through with the following formula [15] that is both mentioned in the EN303 and in the Case Specific Boiler Efficiency method of prEN 15316-4-1.

Formula [15]

$$Q_{blossh} = P_{bstby;h} * ((T_{sys} - T_{au}) / 20)^{1,25} * t_{cycoff} * N_{cyc} / 3600$$

Where

- $Q_{bloss;h}$ is the standby heat loss in burner-off mode per day-period, in [kWh]
- $P_{bstby;h}$ is the standby heat loss acc. to EN303 in [kW]
- T_{au} is the ambient temperature in the boiler room [°C]
- 20 is the temperature rise above ambient on which the test figure is based (must be 30 for standard boilers)

If this test figure for the standby heat losses is not available, the default values that are proposed in the prEN could be used.

<u>According to Case Specific Boiler Efficiency Method:</u>

Pbstby;h = Pnom * (E + F * logPnom)

With:

Boiler type	E	F
Standard boiler	25	-8
LT boiler	17,5	-5,5
Condensing boiler	17,5	-5,5

Calculation energy consumption pilot flame

In case a pilot flame is used, the following simple formula calculates the energy losses per period

Formula [16]

$$Q_{lossign} = P_{bign} * t_{cycoff} * N_{cyc} / 3600$$

Where

- $Q_{lossign}$ is the energy consumption pilot flame per day-period, in [kWh]
- P_{bign} is the load of the pilot flame in [kW]

Calculation electricity consumption during burner-on cyle

The electricity consumption during boiler-on mode, is related to the actual heat output (Pb) of the boiler. Based on the power consumption figures at nominal and minimal boiler load, a linear interpolation is made to the determine the actual power consumption at the actual boiler load per cycle. The electricity consumption is converted to primary energy with a "*primary energy factor"* of 2,5.

Formula [17]

$$Q_{b;el} = (P_{bmin;el} + (P_b - P_{bmin})/(P_{bnom} - P_{bmin}) * (P_{bnom;el} - P_{bmin;el}))*T_{cycon} * N_{cyc} * f_{conv} / 3600$$

Where

- $Q_{b;el}$ is the electricity consumption in burner-on mode per day-period, converted to primary energy [kWh prim]

- $P_{bmin;el}$ is the electricity consumption at minimal boiler load [kW]
- $P_{bnom;el}$ is the electricity consumption at maximal boiler load [kW]
- f_{conv} is the primary energy factor (= 2,5) [-]

Calculation standby electricity consumption

In standby mode (burner is off) the boiler the electricity consumption is related to the standby electricity consumption (e.g. CPU) and the consumption of the pump which may be continuously running (in case of weather dependent controls), or have a reduced operating time (reduced pump operation during night setback, reduced operating time after burner switch-off).

The first formula [18] calculates the standby power consumption, the second formula [19] the power consumption related to the pump and its operating times. The electricity consumption is converted to primary energy with a "*primary energy factor*" of 2,5.

Formula [18]

$$Q_{bstby;el} = P_{bstby;el} * t_{cycoff} * N_{cyc} * f_{conv} / 3600$$

Where

- $Q_{bstby;el}$ is the standby electricity consumption in burner-off mode per day-period, converted to primary energy [kWh prim]
- $P_{bstby;el}$ is the standby electricity consumption of the boiler in [kW]
- f_{conv} is the primary energy factor (= 2,5) [-]

Formula [19]

In case of a fixed speed pump

$$Q_{pmp;el} = P_{pmp} * t_{cycoff} * N_{cyc} * f_{conv} / 3600$$

In case of variable speed pump

$$Q_{pmp;el} = (P_{pmpmin} + (P_b - P_{bmin})/(P_{bnom} - P_{bmin}) * (P_{pmpnom} - P_{pmpmin})) * T_{cycoff} * N_{cyc} * f_{conv} / 3600$$

Where

- $Q_{pump;el}$ is the electricity consumption of the pump in burner-off mode per day-period, converted to primary energy [kWh prim]
- P_{pmp} is the electricity consumption of the fixed speed pump in [kW]
- P_{pmpmin} is the electricity consumption of the var.speed pump at min flow [kW]
- P_{pmpmax} is the electricity consumption of the var.speed pump at max flow [kW]
- f_{conv} is the primary energy factor (= 2,5) [-]

Mathematical operations boiler with single room thermostat

Calculation actual boiler output (Pb)

The first step in modelling a room thermostat is to determine what the boiler output Pb is that will be used for covering the heat demand of a dwelling (or zone). For this, the same formula's as for the weather dependent control can be used, namely: Formulas 3, 4 or 5 , depending on the type of burner control.

For a modulating boiler formula 3 is applicable

Formula [3]

$$Pb = Pbmin8060 + (Tdew - Tsysr)/(Tdew - 30) * (Pbmin5030 - Pbmin8060)$$

Example:

For a Boiler with nominal heat output 20,88 kW (input 24 kW) that can be modulated back to 42% and that has the following testdata: Pbmin8060= 8,00 kW, Pbmin5030= 8,80 kW, Tdew= 52°C (for gas boiler)

The following calculation can be made when the Tsysr = e.g. 45°C:

$$Pb = 8{,}00 + (52 - 45)/(52 - 30) * (8{,}80 - 8{,}00) = 8{,}25\ kW$$

If the heat demand PH (in January) of the average EU-dwelling is 3,5 kW, the value of Pbmin is too high for a modulating operation and the boiler will work in on-off mode.

Calculation percentage 'on' and 'off' time

If the actual heat demand is more than Pbmin, then the burner will operate continuously (modulating) and the heat demand from the Heat Balance (incl. distribution losses) will apply.

If the actual heat demand is below Pbmin, then the burner will operate in on-off mode. The Boiler Cycling method (prEN 15316-4-1) calculates –strictly on the basis of the minimum power and the heat demand—the proportion between the on and off times. In other words:

Formula [20a &b]

$$tcycon\% = PH / Pb$$

$$tcycoff\% = 1 - tcycon\%$$

The Boiler Cycling method proposes no calculation of the (average) length of each cycle (tcyc) and therefore there is no assessment of the average number of cycles per year (Ncyc).

Yet, not only for energy assessment but also on the long term for the emissions, it is important to have these values. In the rest of this section we use a solo boiler (not a combi) and a simple p-band room-temperature controller ('room-thermostat') in a reference room (assume Zone 1 = living room) .

In a <u>first instance</u> the cycling is governed by the room-thermostat, i.e. the on/off-mode' is triggered by the room air-temperature being below/above the set-temperature minus/plus the bandwidth[63]:

Formula [21a &b]

$$Tair < Tsetair - pbandair \quad : \text{boiler is on}$$

$$Tair > Tsetair + pbandair \quad : \text{boiler is off}$$

[63] Ignoring e.g. controls for frost protection or maintenance cycles

The values for Tsetair1 are in the range of 19-23°C (after corrections for dTstrat and dTfluct), whereas for a simple mechanical room-thermostat a pband1 of 2K is not unusual.

With all this it has to be taken into account that the air-temperature is only one of the two components to determine the 'indoor' or 'operative' temperature that is used for the assessment of heating comfort. The other component is the radiation temperature of the walls/floor/etc., which also counts for 50% of the heating comfort. And during normal cycling this radiation temperature is almost constant, i.e. varies around ± 0,2-0,3 K while the room air temperature varies ± 2 K.

In other words, if we use simplified expressions that relate to the 'operative temperature' we should use half of the pband (e.g. ± 1 K instead of ± 2 K) and we will call 'pbandair' with 'band'. The equations [21] above are now replaced by

Formula [22a & b]

$$Tair \quad < \quad Tsetair - 0{,}5 * band \quad : boiler\ is\ on$$

$$Tair \quad > \quad Tsetair + 0{,}5 * band \quad : boiler\ is\ off$$

This will be our first approach, because it does not require the complication of having to work with 'operative', 'air' and 'construction' temperatures, which would complicate matters considerably (see e.g. EN 832 for intermittent operation).

In a second instance the cycling is governed by the boiler-thermostat, i.e. if the 'on' signal from the room-controller persists over a longer period, then the boiler thermostat switches the burner on/off within the boiler control band (bband).

Formula [23a &b]

$$Tsysf \quad < \quad Tbmax - bband \quad : boiler\ is\ on$$

$$Tsysf \quad > \quad Tbmax + bband \quad : boiler\ is\ off$$

Input values for Tbmax are in the range of 60-80°C (say 70°C) whereas a bband (boiler control band) of 8-10K is not unusual (say 10 K)

Calculation boiler cycle time: tcyc

The 'ideal' system temperature Tsysideal (i.e. at modulating mode) for an indoor temperature T is known from the Heat Balance of the dwelling (zone).

Using expressions [22a and 22b] and the Heat Balance we can calculate the system temperatures Tsyslo and Tsyshi for an indoor set temperature of (with Tset = 20°C and *band* = 1):

$$Tlo = Tset - 0{,}5 * band = 20 - 0{,}5 = 19{,}5°C$$

$$Thi = Tset + 0{,}5 * band = 20 + 0{,}5 = 20{,}5$$

With the Heat Balance program we can calculate the related "ideal" system temperatures, that belong to these "high" and "low" indoor temperatures:

Tsysideal = 41,3°C

Tsyslo = 39,5°C

Tsyshi = 43,2°C

The radiator cool down period X from 43,2 to 39,5°C can be calculated as in expression [8], explained in Task 3 chapter 11:

Formula [24] (formula [8] from Task3, chapter 11)

$$tcycoff = 1{,}5 * \ln\left((Tsyslo - T) / (Tsyshi - T) \right)$$

Example:

*tcycoff = 1,5 * ln ((43,2-20) / (39,5-20)) ≈ 0,261 h*

Using expressions [20a &b] we find:

Formula [25]

$$tcycon = PH / Pbmin * tcycoff$$

in this example:

tcycon = (3,5/8,25) 0,261 = 0,11 h*

The total cycle time thus becomes

Formula [26]

$$tcyc = tcycon + tcycoff$$

So, in our example the cycle time becomes 0,37 h (22 minutes). This would be the cycle time if the P-controller reacts immediately to the (change in) temperature. In reality –because of the inertia of the controller, the thermal mass of the air, etc.-- there is a delay time (tdelay). Typically for a mechanical room thermostat (even helped by a mechanical anticipator[64]) this is in the order of 15-20 minutes.

The delay time tdelay is an independent input of the model. Within the model tdelay is first 'translated' into the extra indoor temperature rise or decline dTdelay that it causes. For this we use the cool down expression, knowing that it took a time tcycoff for the room temperature to decline by 2 * 0,5* band = band.

Formula [27]

$$dTdelay = tdelay * band / tcycoff$$

Example

dTdelay = 0,25 * 1 / 0,261 = 0,96°C

This is a simplification. E.g. we took tcycoff as a basis and not tcycon to fit empirical data. In reality the thermodynamics of a controller is very complex and –even with the simplest thermostat—there is always some form of anticipation[65].

The Heat Balance of the dwelling (or zone) can now be calculated for the high and low indoor temperatures corrected for the delay.

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[64] Which is now banned under the ROHS directive because of the mercury content.

[65] PID= Proportional Integral Derivative

Formula [28]

$$Tlo = T - 0{,}5 * pband - dTdelay$$

$$Thi = T + 0{,}5 * pband + dTdelay$$

This again yields new values for Tsyslo and Tsyshi, which can be used in expressions **[24]**, **[25]** and **[26]** to give the final cycle times tcycoff, tcycon and tcyc.

Calculation number of cycles per period

From tcyc and the number of hours per day-period (hrs) it is easy to calculate the total number of cycles (Ncycss) for day-periods where there is a steady state, i.e. where the timer doesn't dictate a night setback or where there is a steady state in a 'reduced setback' mode.

Formula [29]

$$Ncycss = hrss / tcyc$$

Note that the number of hours (hrss) that occur when there is a steady state in a 'reduced setback' mode is known in the mathematical model from earlier calculations.

In case of a 'full setback' period or a 'reduced setback' period without a steady state the number of cycles in the setback period (Ncycsb) is 1 (1 heat-up after 1 cool-down).

Formula [30]

$$Ncycsb = 1$$

These last two expressions, seem easy enough but take some sorting out in the spreadsheet, which we will not go into.

Combining the two formulas [29] and [30], the total number of cycles is:

Formula [31]

$$Ncyc = Ncycss + Ncycsb$$

A typical values for the average EU dwelling with a mechanical thermostat (pband 0,5 K; tdelay=0,33 h) is 7000 cycles/year and an indoor temperature fluctuation of ± 1 K (weighted annual average, max. 1,3 K in January). Cycle-time (tcyc) in January is 0,54 h (0,12 h heat up, 0,42 h cool-down).

For a faster and more accurate electronic thermostat (pband 0,1; tdelay = 0,2h) we find 14.000 cycles/year and an indoor temperature fluctuation of ± 0,6 K (weighted annual average, max. 0,8 K in January). Cycle-time (tcyc) in January is 0,31 h (0,07 h heat up, 0,24 h cool-down).

Note: The indoor temperature fluctuations could be used as a direct input for dTfluct.

Calculation zone 2 and 3 discomfort

With a central room-thermostat in Zone 1, the cycles are dictated by the heat demand in Zone 1. In case this leads to a temperature overshoot in the other temperature zones (bedrooms, bathroom) it could be controlled by a TRV. The modelling of the TRV action could be similar to the approach for Zone 1 but probably with different values for bandwidth and delays

In case, however, this leads to insufficient power to Zones 2 and 3 to achieve the set-temperatures there is a discomfort factor to be taken into account. This discomfort can be expressed as the ratio between the degree hours of discomfort, i.e. below the desired indoor temperature, and the total number of degree hours that the heating system has to supply.

Formula [32]

$$discom\% = deghrsdiscom / deghrstot$$

This then leads to a percentage, which should be added to the energy consumption found. Alternatively we can also asses the difference between heat demand and heat input (better).

Example:

During 1000 hours the indoor temperature is 1,5°C below the desired temperature. The total number of degree hours is 5000 * (18,8 – 8,8) = 50.000 hours.

Discom% = 1500 / 50.000 = 3%

Calculation average system return temperature

The additional temperature fluctuations that are caused by the room thermostat will lead to a corrected heat load of the house. This corrected heat load is calculated and used for determining the final average system feed and return temperature (calculations are performed described in the Heat Balance model). With this final Tsysr (system return temperature) the efficiency can be calculated.

Calculation steady-state efficiency and related losses

With the corrected Tsysr (see previous paragraph) the actual boiler output Pb is recalculated with formula [3], [4] or [5], depending on the type of burner control.

With the corrected system return temperature Tsysr and the corrected boiler output Pb, the efficiency per average cycle is calculated according to formula [10] (similar to the calculations for the weather dependent control). See paragraph 11.4.5

The steady-state efficiency losses are calculated according to formulas [11] and [12].

Calculation start/stop losses

Start/stop losses are calculated according to formulas [13] and [14], see paragraph 11.4.6.

Calculation burner-off standby heat losses

For a boiler that is controlled by a room thermostat, the standby heat losses for the boiler and the distribution system needs to be calculated. Both boiler and distribution system cool down and lose an amount if thermal energy until the next boiler cycle starts. These heat losses are not accounted for in the Heat Balance Model and therefore need to be added. At first energy losses occur during the pump operating time after burner switch-off. After the post operating time of the pump – provided the off cycle continuous – boiler and distribution system continue to lose heat (radiation and convection) while there is no flow in the system.

For both periods (1. before and 2. after pump switch off) and both systems (boiler & distribution system) the heat losses are calculated.

<u>Heat losses boiler during post-operating time pump</u>

For this calculation we need the Tsys at the start and at the end of the pump operating time. Tsys at the start is the Tsys coming from the Heat Balance model. The Tsys at the end of the pump cycle (Tsys;e) is calculated with the following expression that calculates the radiator temperature after a cool down period X. The formula is explained in the description of the Heat Balance model (Task 3, chapter 11):

Formula [33] (formula [6] from Heat Balance model)

$$Tsys;e \quad = \quad Tamb + (Tsysprev - Tah) * e^{-(X/1{,}45)}$$

Where

- X is the cool down time in hours [h] (in this formula X = *ppmp*)

Example:

With Tsysprev = 48°C; X = 500/3600 = 0,139 h; Tah = 20, we get

$Tsys;e = 20 - (48 - 20) * e^{(-0{,}139/1{,}45)} = 45{,}5°C$

For the calculation of the heat losses we can now use formula [15], with Tsys being the average of Tsysprev and Tsys;e.

Tsys = (48 + 45,5) / 2 = 46,75

With this average system temperature over the pump operating time, we can calculate the corrected Pbstby;h and the related heat losses of the boiler with previous formula [15].

Formula [34] (identical to formula [15])

$$Q_{blossh;1} \quad = \quad Pbstby;h * ((Tsys - Tau) / 20)^{1{,}25} * tpump * Ncyc / 3600$$

Example:

With Pbstby;h = 0,22 kW; Tsys = 46,75; Tau = 15; tpump = 500; Ncyc = 4

the heat losses of the boiler during post operating time of the pump for a period with 4 cycles are:

$0,22 * ((46,75 - 15)/20)^{1,25} * 500 * 4 /3600 = 0,218$ [kWh]

<u>Heat losses distribution system during post-operating time pump</u>

In the same time the distribution system will experience heat losses. The amount of heat losses can be calculated in the traditional way with:

Formula [35]

$$Q_{distr;1} = ((L1*U1*((Tsys - Tsys;e)/2 - Tau) + (L1*U1*((Tsys - Tsys;e)/2 - Tah)) *tpmp * Ncyc/(3600*1000)$$

Where

- L1 is the pipe length of the circulation circuit [m]
- U1 is the specific pipe heat loss per m per K for circulation pipes [W/K.m]
- L2 is the pipe length from the circulation circuit to the radiator [m]
- U2 is the specific pipe heat loss per m per K for radiator pipes [W/K.m]

The lengths L1 and L2 of the pipes are related to the heat surface **F** of the dwelling and can be calculated with formula [22] and [23] from the Heat Balance, which are based on the DIN 4701-10.

Formula [36] (

$$L1 = 28 + 0,05 * F$$

$$L2 = 0,515 * F$$

For the U-values the default values from the DIN 4701-10 are taken:

U1 = 0,2 W/mK

U2 = 0,255 W/mk

<u>Heat losses boiler after post-operating time pump</u>

To calculate the heat losses of the boiler after the pump has switched off we use a cool down formula on the analogy of formula [5] of the Heat Balance model (as described in Task 3, chapter 11). With this formula the boiler temperature at the end of the off-cycle is calculated .

Formula [37] (formula [5] from Heat Balance model)

$$Tb = Tau + (Tbprev - Tau) * EXP (-1 * (X / (tcb/Pbstby;h /(Tprev-Tau))))$$

Where

- X is the cool down time in hours [h] (here X = *tcycoff – tpmp*)
- Tb is the boiler temperature [°C]
- *tcb* is the thermal capacity of the boiler [Wh/K]
- *Pbstby;h* is the heat loss of the boiler at *Tprev* and *Tamb* in [kW]

When we know the boiler temperature at the beginning and at the end of the cool down period and we know the thermal capacity of the boiler we can calculate the heat loss of the boiler

Formula [38]

Q*blossh;2*	=	*(Tbprev – Tb) * tcb * Ncyc*

Determining *tcb*

For an estimate of the thermal capacity of a boiler, the boiler weight (mb) and water content (vb) are used. It is estimated that 70% of the boiler weight adopts the system temperature and that the average specific heat of the 70% boiler mass (mainly metal) is around 500 [J/K.kg]. Together with the water content of the boiler and the specific heat of water we have acquired a fair estimate of the thermal capacity of the boiler

Example:

With mb = 68 kg; Vb = 4 ltr

Tcb = (68 * 0,7 * 500 + 4 * 4186) / (3600 * 1000) = 0,0113 kWh/K

Determining *Pbstby;h* at *Tprev*

Pbstby;h at Tprev can be calculated with formula 15, where the Tsys now is the system temperature at the end of the pump operating time.

Formula [39] (similar to formula [15])

Pbstbyh;2	=	$Pbstby;h * ((Tsysprev - Tau) / 20)^{1,25}$

Example:

With Pbstby;h = 0,22 ; Tsysprev = 45,5; Tau = 15°C,

Pbstbyh;2 becomes: $0,22 * ((45,5 - 15)/20)^{1,25} = 0,37$ kW

With tcb and Pbstbyh;2 known we can now calculate the boiler temperature after period X, with X = (*tcycoff – tpmp*) with formula [37]

Example:

Tsysprev = 45,5°C; tcb = 11,3 Wh/K; X = (1500 – 500)/3600 = 0,28 h; Tau = 15

$$Tb = 15 + (45,5 - 15) * e^{-1 * (0,28/(11,33 / 0,37/(45,5 - 15))))} = 35°C$$

The boiler heat losses can now be calculated with formula [38]

Qblossh;2 = (45,5 – 35) * 0,0113 * Ncyc

In a 4-cycle period the heat losses are: 10,5 * 0,0113 * 4 = 0,46 kWh

Heat losses distribution system after post-operating time pump

For the heat losses in the distribution system a similar approach is used, meaning that the temperature after the cool down period is calculated with the general cool down formula. The related heat losses are based on the temperature difference between start en begin of the cool down cycle and the thermal capacity of the circulation circuit. To keep this calculation simple, the small heat losses in the radiator pipes are considered to be usefull.

Formula [40] (adjusted formula [5] from Heat Balance model)

Tcirc	*= Tau + (Tcircprev – Tau) * EXP (-1 * (X / (tcd/Qcirc;h /(Tcircprev-Tau))))*

Where

- X is the cool down time in hours [h] (here X = *tcycoff – tpmp*)
- Tcirc is the temperature of the circulation circuit [°C]
- *tcc* is the thermal capacity of the circulation circuit [Wh/K]
- *Qcirc;h* is the heat loss of the circulation system at *Tcircprev* and *Tamb* in [kW]

For the Qcirc;h, the heat losses of circulation circuit is calculated according to according to formula [41].

Formula [41]

$Q_{circ;h}$	$= ((L1 * U1 * ((Tcircprev - Tau)) / 1000$

tcc can be calculated by assuming a diameter for the circulation pipes. If the default diameter d is set at 22 mm, the water content of the circulation system will be:

Example:

With L1 = 32 m (F = 100 m²); U1 = 0,2 W/mK; Tcircprev = 45,5°C; Tau = 15°C

Qcirc;h becomes: 32 * 0,2 * 30,5 /1000 = 0,195 kW

With d = 22 mm the *tcc* becomes:

$tcc = 32 * 1/4\pi\, d^2 * 1000 * 4186 / (3600 * 1000) = 0{,}014$ kWh/K

The Tcirc at the end of the cool down period can now be calculated with formula [40]

$T_{circ} = 15 + (45{,}5 - 15) * e^{-1 *(0{,}28/(14 / 0{,}195/(45{,}5 - 15))))} = 42°C$

The distribution losses after pump switch off can now be calculated with formula [42]

Formula [42]

$Q_{distr;2}$	$= (Tcircprev - Tcirc) * tcc * Ncyc$

$Q_{distr;2} = (45{,}5 - 42) * 0{,}014 * Ncyc$

In a 4-cycle period the heat losses are: 3,5 * 0,014 * 4 = 0,196 kWh

Calculation electricity consumption during burner-on cycle

Calculations are identical to the ones described 11.4.9 (formula [17]).

Calculation standby electricity consumption

Calculations are identical to the ones described 11.4.10 (formula [18] and [19]).

Mathematical operations boiler with time-proportional controller

The energy loss calculations for the time proportional room thermostat are the same as the calculations in paragraph 11.5 “Boiler with single roomthermostat”. The only difference concerns the calculation of the cycle-on and off time.

Calculation boiler cycle time

A time-proportional controller imposes a fixed (=input parameter) frequency of cycles per hour **fcyc**. Practical values vary between 3 and 6 cycles per hours.[66] The controller determines (from past data) the optimum between tcycon% and tcycoff%.

The advantage of this controller over a traditional mechanical or electronic thermostat is that it largely eliminates the delays.

Because the time-proportional controller ‘knows’ so much more about the system, it is much easier to fit into a mathematical model.

The cycle-time tcyc = 1/fcyc and the on- and off times can be determined with expressions [20a &b].

Formula [20a &b]

$$tcycon\% = PH / Pb$$

$$tcycoff\% = 1 - tcycon\%$$

Example:

fcyc= 4 → tcyc= 1/fcyc = 0,25 h

PH = 3,5 kW; Pbmin = 8,25 kW →

tcycon = tcyc * PH/Pbmin = 0,25 * 3,5/8,25 = 0,10 h

tcycoff = tcyc – tcycon= 0,25 – 0,10 = 0,15h

The fluctuations in indoor temperature can be calculated from this using e.g. expression [2] from the Heat Balance (Task 3, chapter 11). The system average, feed and return temperatures (Tsys, Tsysf, Tsysr) follow from the usual Heat Balance model equations.

[66] In Germany there seems to be a consensus that 4 times is an appropriate default

Output data

The following data are produced as output of the Boiler Model

		Per day-period	Per av. month day	Per month	Per year
Total energy (incl. electr) input boiler	[in kWh and %]				
Energy losses during burner-on steady state operation	[in kWh and %]				
Start/stop losses	[in kWh and %]				
Standby heat losses boiler during off-cycles	[in kWh and %]				
Energy consumption pilot flame	[in kWh and %]				
Electricity consumption during burner-on cycles (conv. to prim)	*[in kWh and %]*				
Standby electricity consumption during off-cycles (conv. to prim)	*[in kWh and %]*				
Electricity consumption pump during off-cycles (conv. to prim)	*[in kWh and %]*				
Energy used for covering heat load house / annual overall efficiency	[in kWh and %]				

List of parameters boiler model

temperatures

T	indoor temperature	°C
Tah	indoor air temperature in heated zone	°C
Tau	indoor air temperature in unheated zone	°C
Tb	temperature boiler appliance (average HE-temperature)	°C
Tdew	Dewpoint of the flue gasses	°C
Tout	outdoor temperature	°C
Tset	indoor set temperature	°C
Tsysf	system feed temperature	°C
Tsysr	system return temperature	°C
Tsys	system temperature = (Tsysf +Tsysr) / 2	°C
Tsysprev	system temperature in previous period	°C
dTfluct	correction for temperature fluctuations by controls	°C
dTstrat	correction for temperature stratification caused by emitters	°C
dTdistr	correction for distribution losses caused by non optimal system temperatures	°C
	times	
hrs	number of hours of a day-period (2, 7, 5, 2 or 8 for morn/mid/eve/late/night)	h
tcool	indoor cool-down time in a setback period	h
treheat	indoor reheat-time in a setback period	h
tdelay	reaction time (delay) of a controller	h
tpurge	pre-purge time	s
tpmp	pump operating time after burner switch-off	s
tcycon	duration of boiler-on cycle	s
tcycoff	duration of boiler-off cycle	s
tcyc	duration of total boiler cycle	s
	energy and power	
QH	heating demand over a period	kWh
Qblosseff	steady-state efficiency losses of the boiler	kWh
Qlosspurge	heat losses due to pre purge-cycles	kWh
Qlossfuel	fuel loss due to pre-purge cycles	kWh
Qblossh	stand-by heat loss boiler in burner-off mode	kWh
Qlossign	energy consumption of pilot flame	kWh
Qb;el	electricity consumption boiler during burner-on mode, converted to primary energy	kWh
Qbstby;el	standby electricity consumption in burner-off mode, converted to primary energy	kWh
Qpmp;el	electricity consumption pump during burner-off mode, converted to primary energy	kWh
Qdistr	heat losses in distribution system	kWh
QT	transmission losses	kWh
QV	ventilation losses	kWh
Ggain	internal gain (people, appliances, etc)	
QG	total solar and internal gains over a priod	kWh
PH	heating power demand	kW
Qbnom	nominal boiler input (acc. EN 303)	kW
Qbmin	minimal boiler input (acc. EN 303)	kW
Pb	actual boiler output	kW
Pbnom80/60	nominal boiler output at 80/60°C	kW
Pbmin80/60	minimal boiler output at 80/60 oc	kW
Pbnom50/30	nominal boiler output at 50/30°C	kW
Pbmin50/30	minimal boiler output at 50/30	kW
Pign	power of pilot flame	kW

Ppmp	power consumption pump	kWe
Pbstby;el	power consumption boiler in standby mode	kWe
Pbmin;el	power consumption boiler at minimal load	kWe
Pbnom;el	power consumption boiler at nominal load	kWe
Pbstby;h	standby burner-off heat losses boiler (acc. EN 303)	kW
Pradnom	nominal radiator capacity (acc. EN 442)	kW
Pradnom;i	nominal radiator capacity in zone i (acc. EN 442)	kW
U1	specific heat loss per meter circulation pipe per degree K (default = 0,200)	W/mK
U2	specific heat loss per meter radiator pipe per degree K (default = 0,255)	W/mk
	geometry and flow	
L1	pipe length circulation circuit	m
L2	length of radiator pipes (from circulation circuit to radiator)	m
F	heated floor area of the building	m²
qpmp	actual flow through the system	l/h
	installation	
fuel	type of fuel used	-
Brn;cntrl	type of burner control (on/off, high/low, modulating)	-
bband	boiler control band	°C
λnom	air factor at nominal boiler load	-
λmin	air factor at minimal boiler load	-
Pmpred	reduced pump operation during night set back (values: yes or no)	-
mb	boiler weight (empty)	kg
vb	water content of boiler	l
CTRLtype	type of room temp control system (values: fixed, WD, RT1, RT2)	-
Cpar	parallel heat curve correction	-
Cgrad	gradient heat curve correction	-
RTcomp	room temperature compensation	°C
band	T=swing caused by room thermostat	°C
Nrt	number of cycles per hour of time proportional room thermostat	-
fopt	optimiser multiplier (multiplier for reheat period)	-
Radc	radiator constant	-
Vrad	water content of radiators	l
Vdistcirc	water content of circulation pipes	l
Vdistrad	water content of radiator pipes	

ANNEX D: REFERENCE MATERIAL

This annex contains reference material for the Subtasks 4 and 5.

It is added here as a general background information.

Contribution of European Heating Industry 2005, Bill of Materials

During the MEEUP study, the European Heating Industry (EHI) presented the following Bills of Material for several boiler typesr. These are given in the tables below.

Table A1: BoM Heating Boilers, Functional Split, avg. EU [i]

		Gas fired wall hung condensing boiler	Gas fired wall hung low temperature boiler	Gas fired floorstanding low temperature boiler	Oil fired low temperature boiler
		20 kW	**20 kW**	**18 KW**	**20 kW**
Functional split	***Funktionsgruppen***				
Safety circuits	Strömungsichering		1360	3140	
Flue System	Abgasführung	940			9070
Open vent	Ausdehnungsgefäss	7690	6890		
Heat exchanger	Wärmetauscher	19970	5590	76490	133500
Burner	Brenner	5070	3770	4380	16680
Insulation	Dammung			1380	2400
Circulation pump	Umwälzpumpe	2740	2740		
Frame	Rahmen	3630	6970		
Gas circuit	Gasstrecke	2470	1440	2560	
Controls	Steuerung	5590	4440	5500	6460
Housing	Gehäuse	17200	6930	21350	18860
Fittings	Heizkreisarmatur	3700	5200	430	
Total	**Summe**	**69000**	**45330**	**115230**	**186970**

Table A2. BoM Heating Boilers, Split by Material, avg. EU [i]

		Gas fired wall hung condensing boiler	Gas fired wall hung low temperature boiler	Gas fired floorstanding low temperature boiler	Oil fired low temperature boiler
		20 kW	20 kW	18 KW	20 kW
Material	***Material***				
Aluminium die cast	Aluminiumdruckguss	10460	1970		4110
Stainless steel	Edelstahlblech	11910	2660	1440	4790
Electronics	Elektronik	880	690	730	1010
Cast Iron	Grauguss	1340	1170	70880	105110
Plastics	Kunststoff	3800	3650	2970	
Plastics ABS	Kunststoff ABS				3210
Copper	Kupfer	2020	4260	740	
Copper	Kupferdrahtlack				2040
Brass	Messing	2250	1650	470	
Insulation	Mineral-Keramik	1910	800	2370	
Insulation mineral wool	Mineralwolle				4500
Steel	Stahl				61480
Steel	Stahlblech	33850	28180	34550	
Others	Rest	570	350	70	550
Total	**Summe**	**68990**	**45380**	**114220**	**186800**

Table A3. BoM Heating Boilers, Details per Component, avg. EU [i]

		Gas fired wall hung condensing boiler	Gas fired wall hung low temperature boiler	Gas fired floorstanding low temperature boiler	Oil fired low temperature boiler
		20 kW	20 kW	18 KW	20 kW
Component group					
Draft diverter	**Strömungssichering**				
Aluminium	Aluminium		130	10	
Stainless steel	Edelstahlblech			60	
Electronics	Elektronik			10	
Plastics	Kunststoff		10	90	
Copper	Kupfer		10	60	
Brass	Messing			30	
Insulation	Mineralfaser			60	
Steel	Stahlblech		1220	2870	
Zinc	Zink			20	
Other	Rest		10		
Total	**Summe**		**1380**	**3210**	
Flue duct	**Abgasführung**				
Aluminium die cast	Aluminiumdruckguss	570			
Stainless steel	Edelstahl	30			
Cast Iron	Grauguss				8600
Plastics	Kunststoff Polypropylen	90			
Brass	Messing	40			
Insulation	Mineral	30			170
Steel	Stahlblech	180			310
Total	**Summe**	**940**			**9080**
Expansion vessel	**Ausdehnungsgefäss**				
Plastics	Kunststoff SBR	740	650		
Steel	Stahlblech	6920	6220		
Other	Rest	30	20		
Total	**Summe**	**7690**	**6890**		
Heat exchanger	**Wärmetauscher**				
Aluminium die cast	Aluminiumdruckguss	8290	1050	130	3900
Stainless steel	Edelstahlblech	11620	2490	1380	1430
Electronics	Elektronik	40			170
Cast iron	Grauguss			70400	97520
High temperature resistant	Hochtemperaturfest				410
Carton	Karton				70
Plastics	Kunststoff	680	290	10	580
Copper	Kupfer	310	2320	10	1170
Brass	Messing	770	210	170	110
Mineral	Mineral				140
Mineral fibre	Mineralfaser			670	
Mineral fibre insulation	Mineralfaserdämmung	700	440	300	1980
Mineral ceramics	Mineral-Keramik	820	210		
Steel	Stahl	1590	2290	7740	40290
Zinc	Zink				40

Other	Rest	250	80	80	30
Total	**Summe**	**25070**	**9380**	**80890**	**147840**
Circulation pump	**Umwälzpumpe**				
Aluminium die cast	Aluminiumdruckguss	230	230		
Stainless steel	Edelstahlblech	170	170		
Electronics	Elektronik	20	20		
Cast iron	Grauguss	1030	1030		
Plastics	Kunststoff	130	130		
Copper	Kupfer	380	380		
Brass	Messing	70	70		
Mineral	Mineral	60	60		
Steel	Stahlblech	640	640		
Other	Rest	30	30		
Total	**Summe**	**2760**	**2760**		
Frame	**Rahmen**				
Aluminium	Aluminium	70			
Stainless steel	Edelstahlblech	30			
Steel	Stahlblech	3520	6920		
Other	Rest	20	50		
Total	**Summe**	**3640**	**6970**		
Gas circuit	**Gasstrecke**				
Aluminium die cast	Aluminiumdruckguss	770	480	550	
Electronics	Elektronik		100		
Cast iron	Grauguss	90		480	
Plastics	Kunststoff PC	90	130	110	
Copper	Kupfer	80	110	120	
Brass	Messing	80	100		
Steel	Stahl	1360	520	1300	
Other	Rest	20	20	10	
Total	**Summe**	**2490**	**1460**	**2570**	
Controls	**Steurerung**				
Aluminium	Aluminium	50	80		
Stainless steel	Edelstahl	70			
Electronics	Elektronik	760	530	740	850
Plastics	Kunststoff	170	2100	2560	2200
Copper	Kupfer	710	640	810	870
Brass	Messing	20	50		20
Mineral filler transformer	Mineral Füllstoff Transformator	300	100	20	60
Steel	Stahl	1930	900	1370	2450
Other	Rest	50	50	30	40
Total	**Summe**	**4060**	**4450**	**5530**	**6490**
Housing	**Gehäuse**				
Plastics	Kunststoff ABS	110	90	170	430
Steel	Stahlblech	16940	6790	21160	18420
Other	Rest	150	50	20	10
Total	**Summe**	**17200**	**6930**	**21350**	**18860**
Heating flow fittings	**Heizkreisarmatur**				
Aluminium	Aluminium	490		50	210
Stainless steel	Edelstahl			10	

Electronics	Elektronik	70	40		
Cast iron	Grauguss	230	140		
Plastics	Kunststoff	280	270	20	
Copper	Kupfer	560	820		
Brass	Messing	1280	1230	280	
Mineral	Mineral			1330	2170
Steel	Stahl	790	2670	120	20
Other	Rest	30	50		10
Total	**Summe**	**3730**	**5220**	**1810**	**2410**

Contribution of the European Heating Industry 2005, Use Phase

During the MEEuP-project EHI has been very active in testing the data above that are in the EuP EcoReport. This has led to some adjustments in the EuP EcoReport. Furthermore the data for 21 boilers were collected and used to make a complete analysis. During meeting with the industry in September 2005, the results of this exercise were presented and are given below. Overall, the industry stated that the results were in line with industry's expectations and that the use of the EcoReport form did not pose any specific problems.

EHI has applied the proposed EuP-methodology to 21 boilers. These boilers cover a wide 15 -215 kW wide range and represent various technologies.

The analysis shows that two parameters are important:

- the total energy consumed during the use-phase
- the acidification during use-phase.

The "total energy "during use-phase depends mainly on the thermal efficiency of the boiler, its average power output and the operating time.

The "acidification" is mainly caused by sulphur dioxide (SO_2) and nitrogen oxides (NO_x) for oil boilers and nitrogen oxides (NO_x) for gas boilers.

The production phase has a minor influence on the life-cycle assessment.

Approach used when applying MEEuP

In order to compare different boiler types and EHI defined a set of fixed operating conditions.

In order to keep the approach simple, two boiler classes have been retained and three climatic zones have been defined.

The analysis applies to only room heating purposes. The hot water production was not considered.

The parameters applied are:

Heat

218 Average Heat Power Output 7,5 kW/ 15 kW (two performance size classes 25 and 45 kW)

219 Number of hours/year 2500/1700/1000 (numbers for three different climatic zones)

220 Type and efficiency at part load 30%

211 Product life in years 15

Electricity

213 On-mode, h/year 750 /510 /300 (2500/1700/1000)

215 Standby-mode, h/year 5090 /3870 /2620 (3340/2680/1920)

217 Off-mode, h/year 2920 /4380 /5840 (corresponding to 4, 6 and 8 months)

Note: For electricity the operating hours listed relate to boilers with "one-stage burners"; in brackets the figures for modulating burners are placed. The results of the calculations, according to boiler type and operating time are summarised in the table and graphs below.

Table A4. Outcomes for 25 kW boiler at 30% load (7,5 kW) over product life

	2500 h/y	1700 h/y	1000 h/y
Total energy in MJ			
Condensing	**1087006**	**739894**	**435883**
Low-temperature	**1228990**	**836440**	**492671**
Standard	**1273442**	**866667**	**510452**
Total acid. in SO2 eq.			
Condensing	**23553**	**16192**	**9677**
Low-temperature	**25836**	**17744**	**10589**
Standard	**24652**	**16940**	**10117**

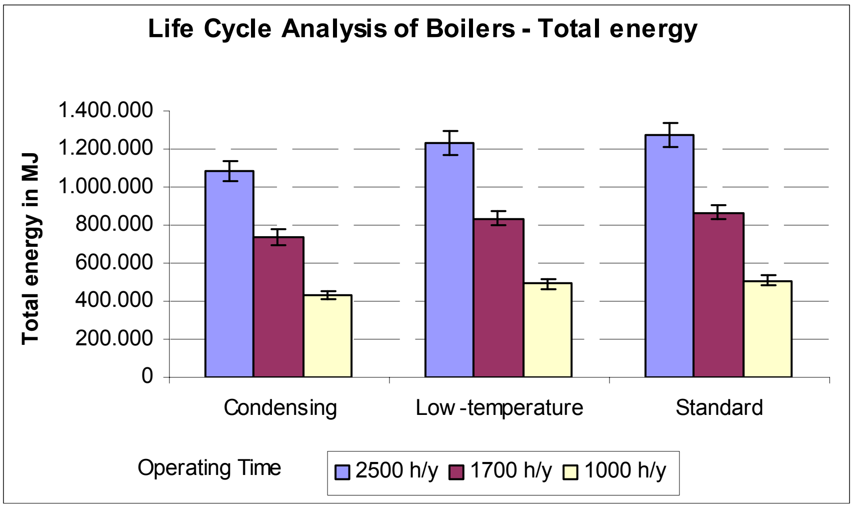


Figure A1. LCA of boilers - Total energy

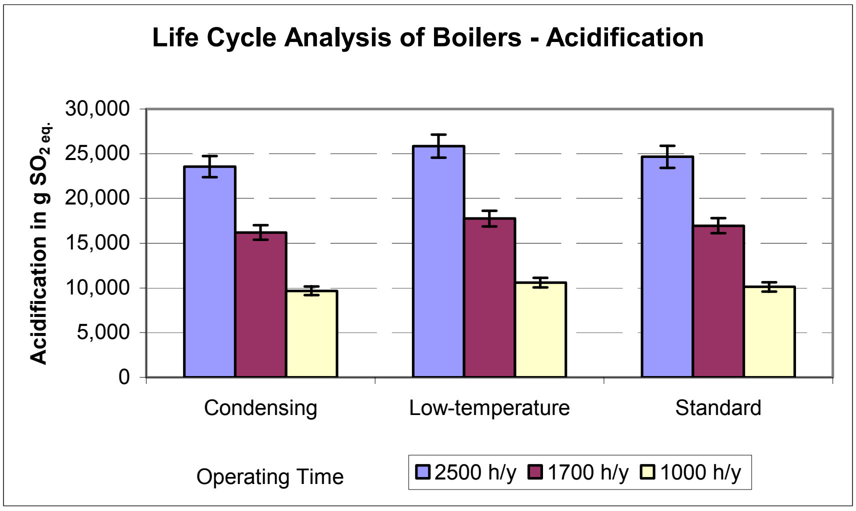


Figure A2. LCA of boilers - Acidification

ECCP 2003

The European Climate Change Programme started in 2000 as a collaboration between DG ENV, DG ENTR and DG TREN of the European Commission on one hand and the stakeholders in the relevant sectors on the other hand to identify and evaluate possible EU policy measures. Starting point for the quantitative analyses in this programme –from the side of the Commission— was the so-called Shared Analysis (1999) performed by the University of Athens for the European Commission, using the *PRIMES* model.[67] This analysis was built upon with a higher level of detail by reports of independent consultants and reports by the stakeholders. Especially in the building sector, responsible for 40% of the CO_2-related greenhouse gas emissions, the programme has tried to identify and clarify which products are responsible. EurACE put forward studies by CALEB Consulting and the Commission has tried to harmonise these results with other inputs, e.g. from the then ongoing SAVE studies in the field of space and water heating (BRE 2002, Novem 2001). The outcome of the quantitative analyses were scenarios for the 1990 and 2010 baseline, as well as a scenario for the 2010 'with measures', presented in Annex I of the second ECCP report 2003. The tables below show the relevant parts for space and water heating (incl. cooling).

Table A5. All Sectors 1990

	Fuel-Related CO_2 emissions in 1990 (in Mt CO_2)			
Sector/function group	***Residential***	***Tertiary***	***Industry***	***Total***
Space heating/cooling, *of which*	**481**	**305**	**76**	**860**
Fossil, *of which*	371	227	57	653
Transmission losses	*190*	*116*	*29*	*334*
-windows	*75*	*46*	*11*	*132*
-walls	*55*	*34*	*8*	*97*
-floors	*30*	*18*	*5*	*53*
-roofs	*30*	*18*	*5*	*53*
Ventilation losses	*70*	*43*	*11*	*123*
Heating system losses	*111*	*68*	*17*	*195*
Electric, *of which*	90	78	19	187
Heating (incl. heatpump)	*73*	*33*	*8*	*114*
Cooling (airconditioners)	*2*	*32*	*8*	*42*
CH pump	*15*	*13*	*3*	*31*
District heating	*20*	*?*	*0*	*20*
Hot water, *of which*	**103**	**35**		**138**
Fossil	67	24		91
Electric	36	11		47

Of course the ECCP data are only a rough indication, but this table suggests that the heating system losses are around 30% of the (theoretically) avoidable heat losses (e.g. 111 Mt on a total of 371 Mt in the residential sector.

[67] Please note that the European Commission continues to use the PRIMES model as the basis for its projections and that coherence between these policy documents at a higher aggregation level and the underlying study at product level is important. The latest publication involving PRIMES is 'European energy and transport: Scenarios for energy efficiency and renewables', European Commission, Aug. 2006.

SAVE Study 2002

SAVE: Heat demand

The SAVE study BRE 2002 gives an estimate of a breakdown of “heating system losses” for an average dwelling 2005. For the 2005 stock, the boiler efficiency is estimated to account roughly for half and the *control inefficiency* plus the *emitter/circuit losses* account for the other half. For newly installed boilers, the share of *control inefficiency* alone is even much higher, accounting for more than 50% of the total. The definition of the control inefficiency relate to unnecessary heating of rooms, temperature overshoot and start-stop losses of the boiler. Especially these latter two points do not show up in the standard boiler efficiency test, but are certainly points that could be influenced by boiler design, i.e. deep modulation and weather-dependent boiler temperature control.

Table A6. Gas- and oil-fired CH-boiler heat demand, New (sales) and Installed (stock) 2005

	NEW (SALES) 2005		INSTALLED (STOCK) 2005	
	CH gas	**CH oil**	**CH gas**	**CH oil**
	kWh/year	kWh/year	kWh/year	kWh/year
Heat load avg. dwelling 2005	7250	7250	7250	7250
Boiler efficiency losses (eff. 87/ 80/ 83,3/ 77,6%)*	1083	1813	1453	2093
Circuit & emitter losses (eff. 96%)	347	378	363	389
Control inefficiency losses (18/ 18/ 20/ 20%)	1563	1699	1632	1752
Effective boiler load (fuel)/dwelling	**10243**	**11139**	**10698**	**11484**
Effective heating system efficiency	70,8%	65,1%	67,8%	63,1%
	mln #	mln #	mln #	mln #
Dwellings served EU 25, in mln. #	**6,954**	**1,709**	**89,792**	**30,250**
Of which, EU-15, in mln. #	6,750	1,594	87,158	28,214
Total fuel energy use EU 25, in TWh/yr	**71**	**19**	**961**	**347**

*= efficiencies in Net Calorific Value

SOURCES:
Kemna, R.B.J., "Task 3.1, VHK Stock Model of Residential Heating Systems", report VHK for BRE, "Study on Heating Systems Labelling/Standards" (EU SAVE II programme), Delft, The Netherlands, Sept. 2001. With contributions from Consult GB (EU), BRE (2nd source UK), VHK (EU/ NL), Energie (IT data), Wuppertal Instuitute (D) and AFECI (B data).

SAVE: Electricity demand

Apart from the fuel consumption there is also the electricity consumption of the boilers is caused by electric fans for combustion (‘pre-mix burner’) or flue gasses, boiler electronics and control, and the circulators.

The electricity consumption of circulators in heating systems is described in the MEEUP Product Cases. The table below gives an overview.

Table A7. Gas- and oil-fired CH-boiler electricity use (excl. CH pump), New/Installed 2005

	NEW (SALES) 2005		INSTALLED (STOCK) 2005	
	CH gas	CH oil	CH gas	CH oil
SINGLE DWELLING BOILER, ca. 22 kW				
7 kW (30% capacity @ eff. 82/ 75%) hours/year	810	827		
22 kW (100% capacity @ eff. 92/ 85%) hours/year	200	204		
Subtotal hours on-mode A (avg. 10 kW)	1010	1031		
Test-time per switch in seconds	30	30	VHK estimate: only 30% with pre-mix burner (=with fan) and electronic control	
No. of switches	12000	12000		
Subtotal hours test-mode B	100	100		
Total hours boiler-fan per year A+B	1110	1131		
Power boiler-fan in W	37,5	37,5		
Total fan electricity use in kWh/yr	41,6	42,4	12,5	12,7
Boiler Control power in W	7,5	7,5	5	5
Hours boiler control/year	8760	8760	8760	8760
Total control electricity use in kWh/yr	65,7	65,7	43,8	43,8
Total electr., excl. pump, kWhe/yr	107	108	56	57
MULTI-DWELLING BOILER, ca. 100 kW				
Power boiler-fan in W	150	150	as above	
Boiler Control power in W	7,5	7,5	5	5
hours fan and control		as above		
Total electr., excl. pump, kWhe/yr	232	235	94	95
EU 25				
No. of single-dwelling boilers, mln. #	3,457	0,677	65,3	19,2
No. of multi-dwelling boilers, mln. #	0,423	0,082	6,1	1,8
Total no. of boilers EU-25, mln. #	3,880	0,759	71,4	21.0
Electricity use single-dwelling boilers, TWh/yr	0,371	0,073	3,675	1.087
Electricity use multi-dwelling boilers, TWh/yr	0,098	0,019	0,576	0,171
Total electr., excl. pump EU-25, TWh/yr	0,469	0,092	4,251	1,258

SOURCES:
Kemna, R.B.J., "Task 3.1, VHK Stock Model of Residential Heating Systems", report VHK for BRE, "Study on Heating Systems Labelling/ Standards" (EU SAVE II programme), Delft, The Netherlands, Sept. 2001. With contributions from Consult GB (EU), BRE (2nd source UK), VHK (EU/ NL), Energie (IT data), Wuppertal Instuitute (D) and AFECI (B data).

Combining the data for circulators with those of fan and controls the MEEUP study is giving a first estimate of the total consumption for the average single dwelling in kWh/year and the average multi-dwelling boiler below (please note: as explained above the data for gas boilers includes a certain percentage stand alone boilers, raising the average circulator consumption from 260 kWh/year to 344 kWh/year).

Table A8. Electricity consumption of boilers

	Sales		Stock	
(kWh/year)	**Gas**	**Oil**	**Gas**	**Oil**
Single dwelling				
Fans	**41,6**	**42,4**	**12,5**	**12,7**
Controls	**65,7**	**65,7**	**43,8**	**43,8**
Circulators	344 *	540	344	540
Total	451	648	400	648
Multi-dwelling				
Fans / controls	**232**	**235**	**94**	**95**
Circulators	947 *	1490	947	1490
Total	1179	1725	1041	1725

* : is based upon stock data, due to unknown ratio boiler control/stand alone in sales

SAVE reference case: Heat load of avg. EU dwelling

The reference case from the SAVE study describes the resulting energy consumption for space heating by oil- and gasboilers if no action is taken. It assumes the ongoing trends of increased efficiency of boilers, increased heating comfort, reduced heat load of housing and growth in number of households. The table relates to all space heating generators (incl. Local heaters, etc.), but for calculations with CH boilers the average heat load of the house (7250 kWh per dwelling in 2005) is an important issue.

Table A9. Reference scenario

YEAR	avg. gross heat load per dwelling	heating system efficiency incl. ODF	avg. EU dwelling energy consumption	EU dwellings	total EU energy consumption for space heating residential
	kWh/hh/yr	%	kWh/hh/yr	million	PWh/yr
1960	10638	58%	18416	97	1,78
1965	10807	57%	18923	103	1,87
1970	10892	55%	19870	109	2.02
1975	10817	52%	20701	116	2,17
1980	10419	54%	19366	122	2,13
1985	9870	56%	17539	130	2.06
1990	8816	63%	14076	138	1,86
1995	7857	69%	11468	146	1,69
2000	7496	70%	10697	152	1,64
2005	7250	72%	10110	155	1,58
2010	6982	73%	9622	158	1,53
2015	6658	74%	8963	161	1,47
2020	6343	76%	8345	164	1,41

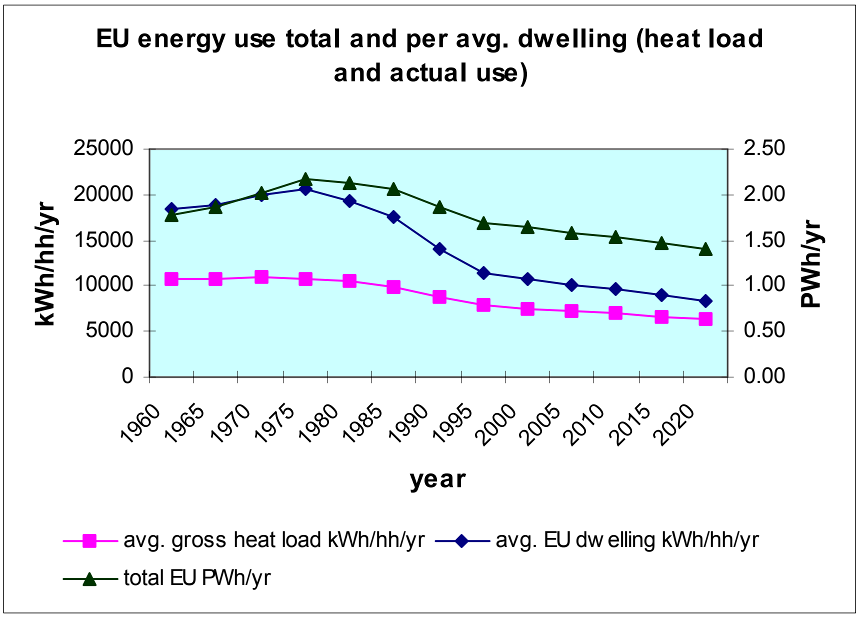


Figure A3. Reference scenario graph [Save study 2002, ECI, Oxford, 2001 for BRE]

SAVE Policy scenarios

The policy scenarios in the SAVE study take into account the potential improvements of heating system efficiency as described in the table below

Table A10. Potential improvements in heating system efficiency [ECI for BRE 2002, SAVE study]

	Existing	Potential improvement
Efficiency of generator (%)	85 (conventional)	up to 111 – condensing up to 140 – heat pump
Efficiency of control (%)	83	<10 improvement
Efficiency of emitter (%)	96	<10 improvement

Using potential improvements in efficiency levels of new equipment it is possible to generate future scenarios of consumption based on these improvements. The two scenarios are a 'more realistic short-term policy' based scenario (Scenario 1), and a longer-term maximum theoretical potential savings (Scenario 2).

Scenario 1 – assumes all heat generators sold by 2005 are 10% more efficient, which is equivalent to half of all EU sales being condensing boilers by this year.

Scenario 2 – This is a theoretical scenario that the 'optimal' level of efficiency can be achieved by 2010. It is unlikely that this level of efficiency could occur without significant intervention by policy makers, at EU and Member State level, but will provide a notional maximum for potential savings for the heating system. Using the input data already described the potential reductions in energy, emissions and costs are given in Table 32.

To reach these levels of efficiency improvements would mean that all new installations would have to be of heat-pump type or a similar level of efficiency.

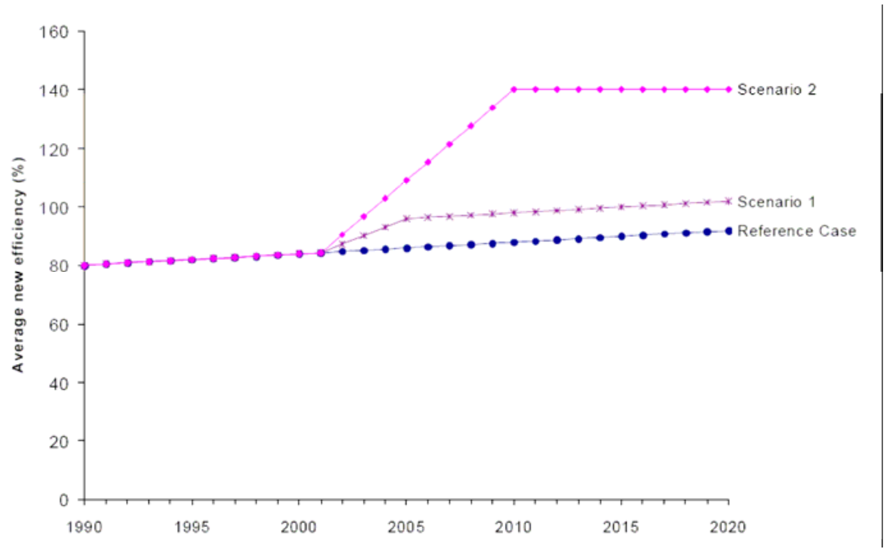


Figure A4. average efficiency of new EU heat generators (% lhv)

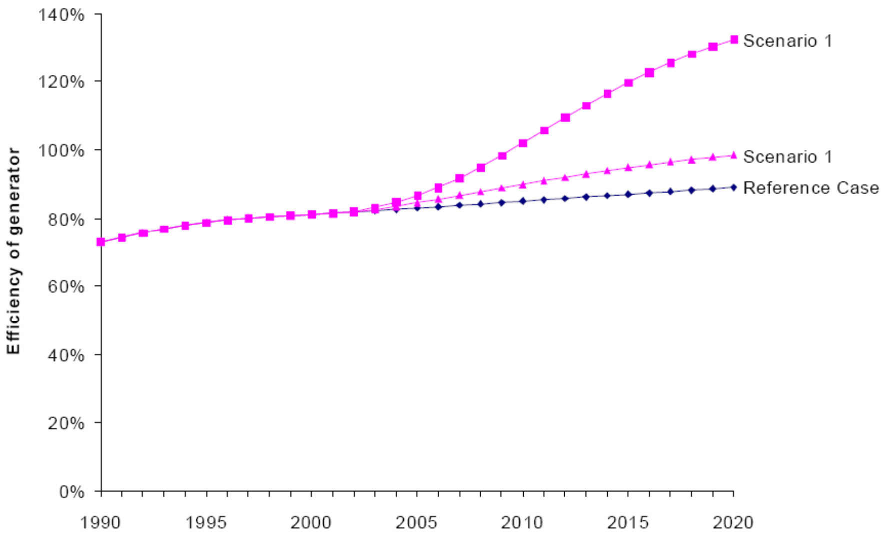


Figure A5. Average efficiency of EU heat generators in stock (heating systems product life 15 years)

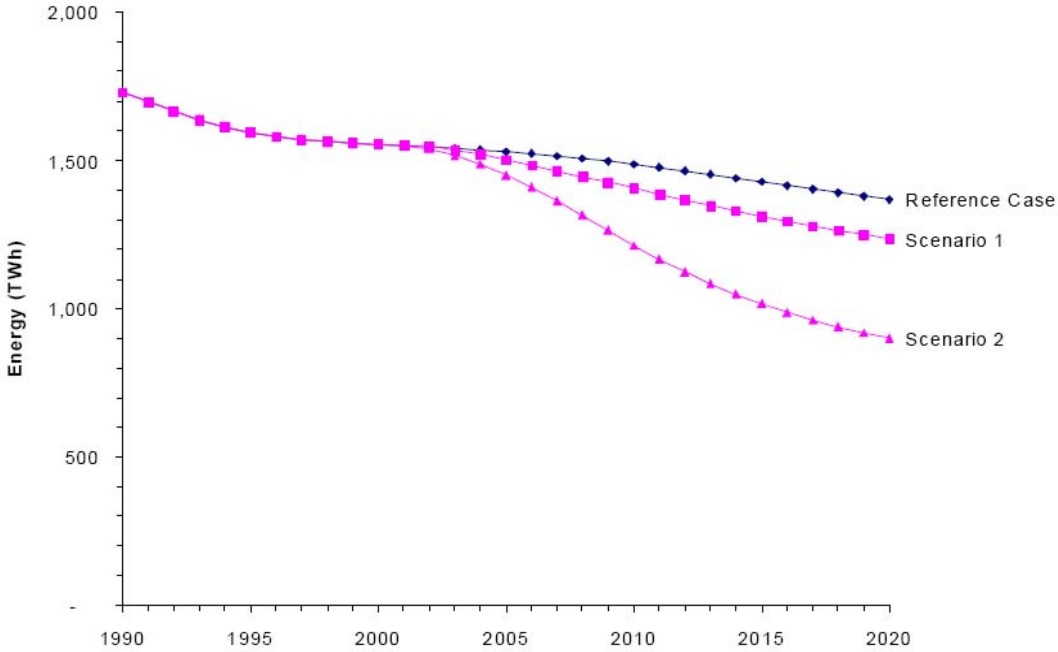


Figure A6. Annual energy consumption by EU space heating, RC and potential reductions

Table A11. Overview of scenario results [ECI for BRE, SAVE study]

	Annual	RC	Scenario 1	RC-Sc1	Scenario 2	RC-Sc2
EU	Energy, 2000 (TWh)	1555	1555	-	1555	-
	Energy, 2010 (TWh)	1487	1407	81	1213	274
	Energy, 2020 (TWh)	1368	1237	131	901	468
	Carbon, 2000 (Mt CO_2)	365	365	-	365	-
	Carbon, 2010 (Mt CO_2)	350	331	19	285	65
	Carbon, 2020 (Mt CO_2)	322	291	31	212	110
Household	Energy, 2000 (kWh)	11652	11652	-	11652	-
	Energy, 2020 (kWh)	8622	7797	825	5672	2947
	Cost, 2000 (euro)	350	350	-	350	-
	Cost, 2020 (euro)	259	234	25	170	88

VHK 1993, Bill of Material

Bill of Materials CH-BOILER

Source: VHK, Milieumaten, 1993 [ii]

Product: Nefit Turbo '90

Type: Wall-hung gas boiler, condensing

Years on market: 1988-1996

Market: NL, D

Table A12. Bill of materials Nefit Turbo

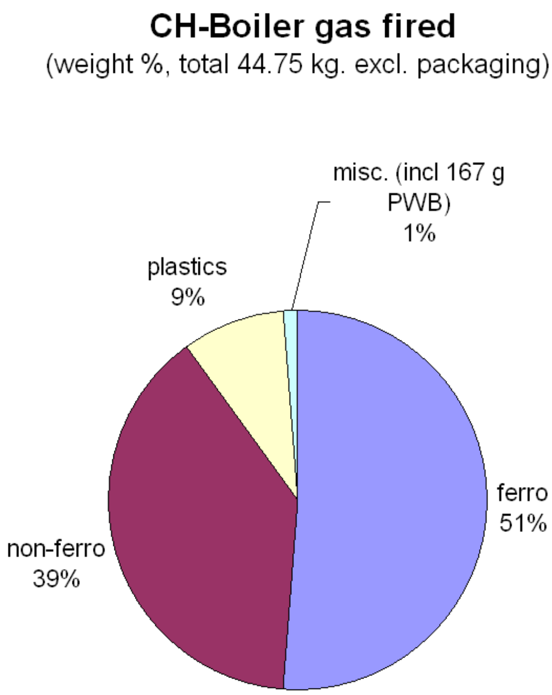
Assembly - material	**weight (g)**	**remarks**	**remarks**
01-input-circuit			
Fe360 I	723		Galvanized 0,04 m²
Synthetic rubber, at plant/RER U	1		
CuZn30 I	37		Brass diecast
PP I	2		
		763	
02-housing			
Fe360 I	9007		Galvanized 0,54 m²
PMMA I	21		
ABS I	1078		
38Si6 I	2		
X5CrNi18 (304) I	11		
		10119	
03-burner unit			
X5CrNi18 (304) I	158		
Fe360 I	407		Galvanized 0,024 m²
Al99 I	157		
G-AlSi12 (230) I	1931		
CuZn30 I	11		
paper gasket, div materials and siliconfoam			
		2664	
04-wall frame			
Fe360 I	5146		Galvanized 0,3 m²
CuZn30 I	4		
PP I	52		
38Si6 I	7		
PA 6 I	3		
05-gas control unit			
CuZn30 I	169		
Fe360 I	277		Galvanized 0,016 m²
Synthetic rubber, at plant/RER U	2		
G-AlSi12 (230) I	650		Diecast
		1097	
06-Condensate receptor			

PVC I	128		
PP I	71		
NBR I	2		
G-AlSi12 (230) I	178		Diecast
G-AlMg5 (314) I	694		Diecast
silicone rubber	0		
		1073	
07-wall profile			
GG15 I	47		
PP GF30 I	104		
Fe360 I	1010		Galvanized 0,06 m²
Cu-E I	202		
ABS I	808		
		2170	
08-panel			
Fe360 I	578		Galvanized 0,035 m²
Cu-E I	19		
ABS I	76		
PA 6 I	2		
		675	
09-return circuit			
09A-CH-Pump based upon % from EPD circulator pumps Grundfos 25-40			
Epoxy resin (liquid) P	119		
G-AlMg5 (314) I	167		
Fe360 I	466		
Cu-E I	287		
G-CuSn12 I	36		
Ceramic (fine)	60		
X5CrNi18 (304) I	191		
Electronics for control units/RER U	167		
GG35 I	848		
EPDM rubber ETH U	24		
Carbon black	24		
		2389	
09B- Other components			
EPDM rubber ETH U	7		
PP I	2		
PVC I	12		
PA 6 I	20		
38Si6 I	1		
GG35 I	2225		
CuZn30 I	130		
Fe360 I	120		
PVC (e) I	50		
Cu-E I	10		
ABS I	40		
		2617	
10-thermostaathouder			
Cu-E I	17		
PVC (e) I	150		

PP GF30 I	92		
Epoxy resin I	75		
Cu-E I	75		
Fe360 I	50		
		459	
11-air circuit to premix-burner			
Fe360 I	635		
G-AlMg5 (314) I	323	Diecast	
PVC I	109		
EPDM rubber ETH U	133		
PP GF30 I	800		
Cu-E I	450		
		2449	
12-heat exchanger			
AlMgSi0,5 (6060) I	6578		Diecast
G-AlSi7Mg (Thixo) I	5246		Diecast
GG35 I	376		
Fe360 I	578		
X5CrNiMo18 (316) I	74		
38Si6 I	16		
CuZn30 I	27		
Synthetic rubber, at plant/RER U	38		
PE (HDPE) I	2		
Glass tube, borosilicate, at plant/DE S	12		
Glass wool, at plant/kg/CH U	116		
		13063	
TOTAL excl. packaging		**39539**	
13-packaging			
Materials/Assemblies			
Paper, recycling, with deinking, at plant/RER S	192		
Corrugated board, recycling fibre, double wall, at plant/RER U	1879		
PP I	25		
PE (LDPE) I	189		
PS (EPS) I	444		
		2729	
TOTAL incl. Packaging		**42268**	

Table A13. Bill of materials Nefit Turbo, grouped

materials	weight (g)	totals
Fe360	18997	
GG15	47	
GG35	3449	
X5CrNi18 (304)	360	
X5CrNiMo18 (316)	74	
		22926
Al99	157	
AlMgSi0,5 (6060)	6578	
G-AlMg5 (314)	1184	
G-AlSi12 (230)	2758	
G-AlSi7Mg (Thixo)	5246	
G-CuSn12	36	
Cu-E	1060	
CuZn30	378	
		17397
ABS	2002	
EPDM rubber	163	
Epoxy resin (liquid)	119	
Epoxy resin	75	
NBR	2	
PA 6	25	
PE (HDPE)	2	
PMMA	21	
PP GF30	996	
PP	127	
PVC (e)	200	
PVC	250	
		3981
Glass tube, borosilicate, at plant/DE	12	
Glass wooll	116	
Carbon black	24	
Ceramic (fine)	60	
PWB (incl. Electronics)	167	
paper gasket, misc. materials and siliconfoam	na	
silicone rubber	na	
38Si6	26	
Synthetic rubber	41	
		446
TOTAL excl. packaging		**44750**



ANNEX E: EMISSIONS ACCORDING TO GEMIS 4.2

Task 4, Chapter 3.1

Table 3-1. Use phase: Energy and emissions per GJ heat out CH boiler, (excl. Electricity for fossil fuel based heating)

[Öko-Institut GEMIS database, Version 4.3]

HEATING		Energy				
		primary				
		heat in - CEU/KEV renewable (incl. life cycle supply chain)	heat in - CEU/KEV non renewable (incl. life cycle supply chain)	heat in (combus-tion)	η_{HI}	heat out (combus-tion)
nr.		MJ_{Hi}	MJ_{Hi}	MJ_{HI}	-	MJ_{Out}
66	Electric, mix	2	2972	1010	0,99	
68	Gas, atmosph.	2	1366	1176	0,85	
71	Gas, condens.	2	1165	1000	1,00	
72	Oil (S=900 mg/kg)	2	1382	1176	0,85	
73	Oil, condens., low S (S=30 mg/kg)	3	1229	1020	0,98	1000
new	el-heatpump-mono-air-mix	37	881	308	3,25	
new	el-heatpump-mono-soil-mix	32	759	256	3,90	
new	Wood-logs	1540	48	1538	0,65	
new	Wood-pellets	1268	237	1250	0,80	

HEATING		Emissions: To Air											
		GWP			AD			VOC		POP	PAH&HM	PM	
		CO2 equivalent (incl. life cycle supply chain)	CO2 (incl. life cycle supply chain)	CO2 from combustion only	SO2 equivalent (incl. life cycle supply chain)	SO2 (incl. life cycle supply chain)	SO2 from combustion only	NMVOC (incl. life cycle supply chain)	NMVOC from combustion only	POP	PAH (incl. life cycle supply chain)	PM (incl. life cycle supply chain)	PM from combustion only
nr.		kg/GJ_{Out}	kg/GJ_{Out}	kg/GJ_{Out}	g/GJ_{Out}	g/GJ_{Out}	g/GJ_{Out}	mg/GJ_{Out}	mg/GJ_{Out}	i-Teq	mg/GJ_{Out}	g/GJ_{Out}	g/GJ_{Out}
66	Electric, mix	264,6	232,3	-	265	163	-	9305	-		0	7,46	-
68	Gas, atmosph.	83,2	74,8	65,7	50	4	0,5	16457	856		0	2,43	0,33
71	Gas, condens.	71,1	63,9	55,8	43	4	0,4	14014	727		0	2,15	0,28
72	Oil (S=900 mg/kg)	104,2	102,7	87,5	149	102	49,6	24173	1520		0	5,52**	0,09**
73	Oil, condens., low S (S=30 mg/kg)	92,6	91,2	76,0	88	49	1,4	20671	880		0	5,03**	0,03**
new	el-heatpump-mono-air-mix	60,1	57,0	-	83	38	0,0	4269	4		0	2,94	-
new	el-heatpump-mono-soil-mix	51,8	49,1	-	72	33	0,0	3726	0		0	5,24	-
new	Wood-logs	8,5	3,3	-	127	56	52,7	434050	430235		0	183,40*	76,92*
new	Wood-pellets	18,2	16,0	-	207	42	26,8	47475	36628		0	98,05**	13,38**

* PM for Wood-logs 50 kg/TJ_{in} based on BUWAL (Bundesamt für Umwelt, Wald und Landschaft), Switzerland, "Handbuch Emissionsfaktoren für stationäre Quellen"

** PM for Wood Pellets 10,7 mg/MJ_{in}, 0,075 mg/MJ_{in} for regular heating oil and 0,032 mg/MJ_{in} for low sulfur heating oil based on Edenhofer, "Erdöl Erdgas Kohle", 5/2007, page 222-226

NEW:
- GEMIS 4.3 not 4.2: changes in data because of updated raw data in GEMIS (GEMIS 4.3 is available since Dec 2005)
- average sulfur content of regular heating oil 900 mg/kg (not 1600 mg/kg) because of new limitation beginning 1.1.2008 according to EU-law RL1999/32/EG from 26.04.1999
- condensing oil boilers using low sulfur heating oil with an average sufur content of 30 mg/kg

ANNEX F: REFERENCES

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