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“Experimental and numerical characterization of heating domestic appliances for energetic efficiency improvement”

Ing. Antongiulio Mauro

Supervisor

Ch.mo Prof. Ciro Aprea

Coordinator

Ch.mo Prof. Vincenzo Sergi

Co-tutor

Ing. Angelo Maiorino

"Start where you are,

Use what you have,

Do what you can."

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Nomenclature

$\big\vert_{MR,H_{eq}}$	calculation for a fixed building and MR	
$\dot{Q}$	heat	(kW)
$\bar{h}$	tank equivalent thermal loss	(W $m^{-2}K^{-1}$)
$\dot{V}$	volumetric flow rate	(m^3 s^{-1})
A	area	(m^2)
alpha	statistical significance level	
bcc	boiler compensation curve	
$\big\vert_{H_{eq}}$	calculation for a fixed building	
C	gas consumption	(kWh)
c	specific heat capacity	($kJkg^{-1}K^{-1}$)
CC	primary energy coefficient of conversion	(-)
cf	boiler cycling frequency	(h^{-1})
CH	central heating	
CO	carbon monoxide	

CO_2	carbon dioxide	
CR	cooling coefficient	($W\ m^{-1}s$)
DD	degree day	(kWh)
e	air excess	(%)
E	electrical energy	(kWh)
ERC	efficiency reduction coefficient	(-)
Erp	Energy related products regulation	
Ex	statistical expected value of random variable	
F	correction terms to seasonal efficiency	(%)
f	generic function	
G20	methane	
G21	Methane family gas limit	
G231	Methane family gas limit	
GCV	gross calorific value	($MJ\ m^{-3}$)
h	occurrence	(h)
H	specific heat loss	($W\ K^{-1}$)
J	boiler jacket area	(m^2)

L	house load	(kWh)
mass	mass quantity	(kg)
MR	maximum modulation ratio	(-)
mr	modulation ratio	(-)
N	total bins number	(-)
NOx	nitrogen oxides	
P	electrical power	(kW)
p	pressure	(kPa)
PLR	partial load factor	(-)
q	normalized losses	(-)
Q	thermal energy	(kWh)
R^2	Coefficient of correlation	
S/V	surface volume ratio	(m^{-1})
SCF	smart control ratio	
SF	specific flow rate	
smart	smart function compliance	
ss	sample size	
T	temperature	(K)

t	time	(s)
Ultramodulation	modulation ratios higher than 10	
V	Volume	(m^3)
var	statistical variance	
Z	statistical z-value	

Gas reference conditions: temperature= 15°C, pressure= 1013.25 mbar

Greek Symbols

Δ	absolute variation	
η	efficiency	
σ	population standard deviation	
δ	relative variation or differential partial variation	
α	convective-radiative heat exchange coefficient	($W\ m^{-2}\ K$)
ρ	density	($kg\ m^{-3}$)

Subscripts

$_{stbyls}$	standby heat thermal losses
$_{\%dry}$	percentage of dry flue
$_{\%wet}$	percentage of wet flue

acc	acceptable
air	air related quantity
b	boiler
bench	test bench related quantity
c	control
C	critical
cold	cold water
d	distribution
DUT	device under test
e	external
eb	electronic board
el	electrical
elec	electrical consumption for hot water production
elr	electrical resistance related quantity
em	emission
EPE	electrical primary energy
eq	equivalent
f	flow

fa	forced air
fan	fan
fl	flue losses
fr	firing
fuel	fuel consumption for hot water production
g	global
H2O	water
i	index
ign	ignition related quantity
in	input
j	jacket
k	discretization index
load	load
min	minimum
na	natural air flow
nc	normal cooling
nom	nominal conditions
OFF	off

ON	on
Op	optimal
out	output
output	water output and losses
p	primary energy
PE	total primary energy
postp	post purge
prep	pre purge
pump	pump
r	return
ref	reference conditions
room	room
s	stationary
seas	seasonal space heating efficiency
tank	tank related quantity
unc	uncertainty
year	year

Introduction

Every year 5 million of boilers for domestic heating are sold in the European Union (EU). Because heating contributes to more than 20% to the whole energy use in the EU (Karsten D. H., 2000), strategies for energy saving are of great interest and have been widely explored.
Currently in the European Union efficiency requirements on boilers are becoming more and more stringent: following the common target of the energy reduction, European directives 2005/32/EC on the eco-design of energy-using products and 2009/125/EC on the energy-related products, oblige the labeling of central heating (CH) boilers in performance categories. Furthermore, EU policy is to phase out appliances with low performance, by no longer giving the right to be sold on the European market. The regulation “Erp” (European Commission, 2013) which enters into force from 26th of September 2015 sets efficiency bans so that low efficiency boilers are going to be excluded from EU market. As a consequence, boilers manufacturers should develop products to meet regulatory requirements and perform a wide experimental mapping covering the whole product range. Manufacturers have to face significant investments to adopt appropriate facilities in terms of technical specifications and testing capability. Robust measurements become necessary to establish correct performance categories and not generate conflicts with surveillance bodies. Additionally, an efficiency oriented product development could be necessary to improve or strengthen the positioning in energetic ranking. At the same time, experimental activities oriented to establish products phase-out becomes necessary.
In the present PhD thesis work, carried out at Ariston Thermo Group in Osimo R&D center, experimental and modeling activities have been performed and reported concerning heating appliances.
Concerning domestic methane supplied gas boilers, a comprehensive analysis of energy fluxes has been carried out, different methods for efficiency estimation have been compared with related measurement uncertainties (ASME, 2013). Existing testing protocol and standards have been considered for efficiency determination and investigation (LABNET, 1998) (CEN , 2013).

The boiler energy balance closing problem has been undertaken through a novel statistical approach.

Subsequently, insulation testing methods have been set-up and compared, in particular an automated test rig has been constructed and compared to thermal camera measurements.

Experimental investigation have been undertaken in order to improve products efficiency with respect to Erp ban, analyzing products limitations.

Concerning boiler efficiency, it has been showed that installing condensing high efficiency boilers rather than standard ones, improve control management and rightsizing boiler capacity (Heselton K. E., 2000.) (Heslton K. E., 1998) produce a great effect on boilers energy consumption and efficiency on field.

Furthermore the boiler modulation is an important parameter in order to rightsize the boiler as well as to minimize fuel consumption (Lazzarin, 2014).

Most of the domestic gas boilers on the market provide space heating below nominal capacity within a range described by the modulation ratio. This parameter expresses the maximum heat output reduction in respect to the nominal one. Currently the typical modulation ratio available on the market is equal to 10 for the top class condensing boilers, which means for a 22kW nominal heat output a minimum output of 2.2 kW.

Experimental evidences underline that boiler efficiency significantly varies from the nominal value when boiler works in partial load (Verma V.K., 2013) or in cycling conditions (Verma V.K., 2013). Yet, high modulation ratios could improve efficiency by reducing cycling frequency, nevertheless would improve boiler operating times and electrical consumption. The real convenience of adopting ultra-high modulation ratio is currently under investigation. (Marcogaz, 2013)

Finally, in the present thesis a stationary bin model has been presented in order to simulate boiler behavior on field for different cities, buildings and plants and investigate the convenience of adopting ultra-high modulation ratios. A stationary bin method base model was performed in order to investigate advantages and limitations for such boilers.

CHAPTER 1: European Directives on boiler efficiency

Considering the replacement of standard boilers with condensing ones it is theoretically possible to reach energy saving of the order of 10-15%; this fact has led to a number of European initiatives to phase out poor efficiency boilers from the market.

Energy labeling for electrical appliances (EU directive 92/75/EC, which is from July 2011 replaced by Directive 2010/30/EU) has been found to be a successful tool to promote energy efficient appliances in the market.

Efficiency requirements in connection with CE-labeling of gas-fired boilers are mentioned in Directive 92/42/EC. This was later modified through Directive 2004/8/EC. Moreover, other more recent European directives are important for CH boiler performance. The energy performance of buildings directive (EPBD) 2002/91/EC resulted in many countries introducing new minimum requirements for boiler efficiency accompanied by regular mandatory inspections of the boilers (Kessen, 2007).

In the past, the European commission had taken initiatives to remove boilers with very poor efficiency from the market with the purpose of reducing energy use and emissions. The European Commission developed a new approach through directive 92/42/EEC, later converted into national legislation (Kessen, 2007). Boilers have to satisfy essential requirements before being introduced into the market and put into service, such as the requirements indicated in Table 1.

Type of boiler	Efficiency requirements for full load ($\dot{Q}_{out\_nom}$)		Efficiency requirements for 30% part load ($0.3\dot{Q}_{out\_nom}$)	
	Mean temperature of the water [°C]	Requirements (in%)	Mean temperature of the water in the boiler [°C]	Requirements (in%)
Standard boiler	70	$\geq 84 + 2\log(\dot{Q}_{out\_nom})$	50	$\geq 80 + 3\log(\dot{Q}_{out\_nom})$
Low temperature boiler	70	$\geq 87.5 + 1.5\log(\dot{Q}_{out\_nom})$	40	$\geq 87.5 + 1.5\log(\dot{Q}_{out\_nom})$
Gas-fired condensing boiler	70	$\geq 91 + \log(\dot{Q}_{out\_nom})$	30	$\geq 97 + \log(\dot{Q}_{out\_nom})$

Table 1. Efficiency requirements of central heating boilers

The $\dot{Q}_{out\_nom}$ is the nominal heat output of the boiler which corresponds to the maximum heating capacity. Notified Bodies have been nominated by member governments and notified by the European Commission to investigate the conformity of CH boilers and water heaters to the requirements of the directive mentioned above; tests could be performed by the notified body or an accredited agency according to ISO 17025. There have been national labeling, boiler database and information systems implemented to promote high efficiency boilers such as BOILSIM in Denmark and SEDBUK in UK.. (Kessen, 2007). The present version of Directive 2005/32/EC on the ecodesign of Energy using Products (EuP) and 2009/125/EC on the energy-related products (Erp) oblige the labeling of central heating boilers: a classification of the CH boilers into different categories is proposed. Commission Regulations (EU) No 811/2013, 812/2013, 813/2013, 814/2013 implement Directive 2009/125/EC clarifying technical aspects of ecodesign and energy labeling processes. In the present work, directive 2009/125/EC and related regulations are generally referred to as “Erp” regulation.
Erp regulation is going to become effective as of the 26th of September 2015.

CHAPTER 2: Boiler energy balance affected by measurement uncertainty

2.1 Boiler space heating efficiency characterization

Considering boiler space heating behavior and neglecting domestic hot water production, for a finite and generic working period, average heat balance on the boiler can be written as follows:

$$\dot{Q}_{in} = \dot{Q}_{out} + \dot{Q}_{j} + \dot{Q}_{fl} + \dot{Q}_{cl} \tag{1}$$

Where $\dot{Q}_{in}$ is the heat input, $\dot{Q}_{j}$ the jacket losses, $\dot{Q}_{fl}$ the flue losses, $\dot{Q}_{out}$ the heat output and $\dot{Q}_{cl}$ the cycling losses.
In case of stationary behavior so that no cycling effects occur, heat output $\dot{Q}_{out}'$ and boiler efficiency η_s can be defined:

$$\dot{Q}_{out}' = \dot{Q}_{in} - (\dot{Q}_{j} + \dot{Q}_{fl}) \tag{2}$$

$$\eta_s = \frac{\dot{Q}_{out}'}{\dot{Q}_{in}} \tag{3}$$

The heat output could be also expressed as heat provided to water:

$$\dot{Q}_{out}' = \rho \dot{V}_{H2O} \bar{c}(T_f - T_r) \tag{4}$$

$$\bar{c} = \left(\frac{c(T_f) - c(T_r)}{2} \right) \tag{5}$$

Where ρ and c are interpolated value of the water density and specific heat, supposed functions of water temperature T only; ρ is dependent on the temperature of the section where the water flow rate $\dot{V}_{H2O}$ is measured. The specific heat $\bar{c}$ is calculated as the arithmetic mean between temperatures T_f and T_r, which are the flow and return temperatures of the water, according to standard nomenclature (CEN , 2013).

The heat input $\dot{Q}_{in}$ is equal to methane chemical power and can be calculated as follows:

$$\dot{Q}_{in} = \frac{T_{ref}}{T_{gas}} \frac{p_{gas}}{p_{ref}} \dot{V}_{gas}\, GCV \tag{6}$$

Where GCV is the higher calorific value per volume in gas reference conditions, $\dot{V}_{gas}$ the gas flow rate, $\frac{T_{ref}}{T_{gas}} \frac{p_{gas}}{p_{ref}}$ is the volume correction ratio related to reference conditions of the calorific value. Losses terms could be expressed as follows:

$$\dot{Q}_{fl} = \dot{V}_{gas} \cdot \left((V_{\%dry} \cdot \bar{c}_{fl\_dry} + V_{\%wet} \cdot \bar{c}_{fl\_wet}) \cdot (T_{fl} - T_{air}) \right) \tag{7}$$

Where $V_{\%dry}$ and $V_{\%wet}$ are respectively the percentages of dry and wet flue gases related to gas flow, $\bar{c}_{fl\_dry}$ and $\bar{c}_{fl\_wet}$ the mean specific heat, T_{fl} the flue discharge temperature and T_{air} the air temperature.

The jacket losses could be expressed as follows:

$$\dot{Q}_j = \iint_J \alpha\left(T_j - T_{air}\right) dJ \cong \sum_k^M J_k \alpha_k \left(T_{k\_j} - T_{k\_air}\right) \tag{8}$$

Where α is the radiative-convective coefficient, T_j is the jacket temperature, T_{air} the air temperature near the boiler, J is the area of the

boiler jacket, M the size of surface discretization and k a discretization index.
Referring to equation 3, by measuring losses terms and heat input it is possible to evaluate boiler efficiency indirectly:

$$\eta_{s\_ind} = 1 - \frac{\dot{Q}_j + \dot{Q}_{fl}}{\dot{Q}_{in}} = \eta_s \qquad (9)$$

So defined indirect stationary efficiency is mathematically equal to efficiency η_s.
The η_s could be considered as a "direct" way to evaluate efficiency and in the present work will be also noted as follows :

$$\eta_{dir} = \eta_s \qquad (10)$$

When considering cycling behavior $\dot{Q}_{cl} \neq 0$, an average long time efficiency can be defined for the boiler η_b, where $\dot{Q}_{out}$ is equal to:

$$\dot{Q}_{out} = \dot{Q}_{in} - \dot{Q}_{fl} - \dot{Q}_j - \dot{Q}_{cl} \qquad (11)$$

Dividing by $\dot{Q}_{in}$:

$$\frac{\dot{Q}_{out}}{\dot{Q}_{in}} = \left(1 - \frac{\dot{Q}_{fl} + \dot{Q}_j}{\dot{Q}_{in}}\right) - \frac{\dot{Q}_{cl}}{\dot{Q}_{in}} \qquad (12)$$

The boiler efficiency can be written as follows:

$$\eta_b = \eta_s - q_{cl} \qquad (13)$$

Where q_{cl} are cycle losses normalized by the heat input. An efficiency reduction coefficient (ERC) could be then defined as the ratio between boiler efficiency and steady state efficiency.

$$\eta_b = \eta_s\left(1 - \frac{q_{cl}}{\eta_s}\right) = \eta_s ERC \qquad (14)$$

Where:

$$ERC = \frac{\eta_b}{\eta_s} \qquad (15)$$

This ratio expresses the boiler efficiency degradation due to on-off cycling behavior.

The ratio between the load $\dot{Q}_{load}$ and the minimum heat output $\dot{Q}_{out\_min}$, determinates cycling frequency (cf_b), this ratio is defined as partial load ratio PLR:

$$PLR = \frac{\dot{Q}_{load}}{\dot{Q}_{out\_min}} \qquad (16)$$

So that the cycling frequency could be considered as a function of PLR only:

$$cf_b = f(PLR) \qquad (17)$$

In particular when the boiler is oversized compared to the load (PLR<1), cf_b increases reducing the ERC. Indeed:

$$\begin{aligned} &\text{When } PLR < 1, ERC < 1 \\ &\text{When } PLR \geq 1, ERC = 1 \end{aligned} \qquad (18)$$

Moreover, the external temperature influences cycling losses because it affects heat losses during off periods (Bonne U., 1985). As a consequence, ERC can be than considered as a function of the external temperature and of the PLR (Heselton K. E., 2000.) (Heslton K. E., 1998) (Landry R. W. et al., 1993):

$$ERC = f(PLR, T_e) \tag{19}$$

Boiler heat output can be described in terms of modulation ratio (mr):

$$mr = \frac{\dot{Q}_{out\_nom}}{\dot{Q}_{out}} \tag{20}$$

So that the minimum heat output $\dot{Q}_{out\_min}$ is provided at maximum modulation ratio:

$$MR = \max(mr) \tag{21}$$

Thus MR is also defined as:

$$MR = \frac{\dot{Q}_{out\_nom}}{\dot{Q}_{out\_min}} \tag{22}$$

An example of boiler cycling behavior is shown in Figure 1: when the load is lower than heat output, boiler starts on-off cycles at maximum modulation (MR) and flow temperature starts oscillating, in present example MR is equal to 4.

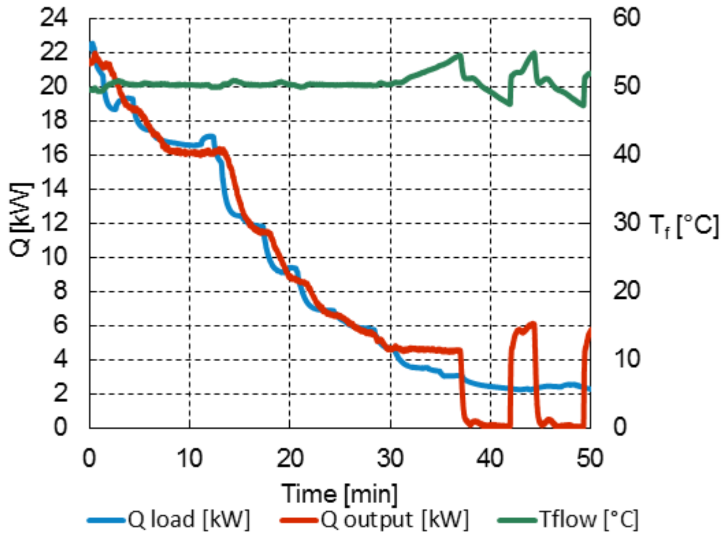


Figure 1. Example of boiler cycling to satisfy a load lower than minimum heat output.

2.2 Efficiency measurement and uncertainty

Boiler instantaneous efficiency η_s has been measured according to standard (CEN , 2013) specs using the following experimental apparatus: boiler is operated by an automatic test bench, allocated in a climate chamber with controlled air temperature set to 20+-1°C, with a chiller that dissipates the heat produced. Three RTD PT100 positioned as shown in Figure 2 are used for each water temperature measurement, and an electromagnetic water flow meter positioned in proximity of temperature probes for density interpolation. Volumetric gas counter equipped with an encoder is used for gas volume flow measurement, inlet gas temperature and pressure are measured for volume flow correction to standard conditions. Cylinders of certified methane G20 99.5% supply the boiler. Air inlet temperature, flue temperature and CO/CO2 emissions are monitored constantly during tests. A monophasic wattmeter was used to measure electrical consumption.

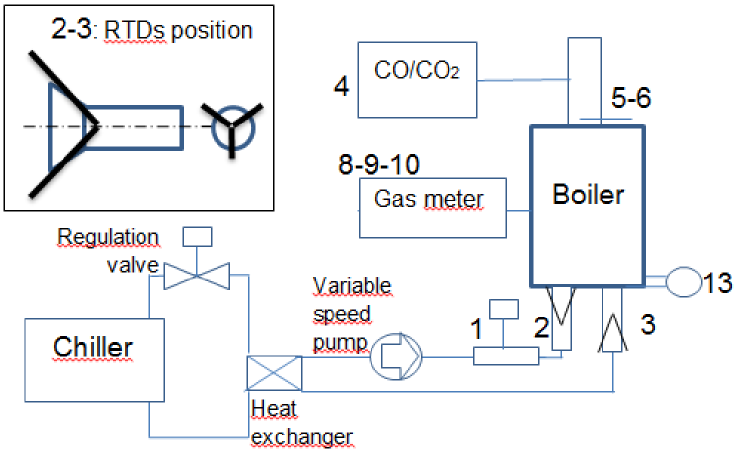


Figure 2. Boiler efficiency measurement apparatus, instruments label refers to Table 2.

N°	Measurement	Instrument	Type	u% (2k)
1	Water flow	Siemens MAG1100+MAG6000	Electromagnetic	1.1
2	Temperature flow	3 PT100 3mm, 4wires	Thermoresistances	0.1
3	Temperature return	3 PT100 3mm, 4 wires	Thermoresistances	0.1
4	CO-CO2	Siemens Ultramat 23		2.0
5	Flue temperature	PT100 3mm, 4 wires	Thermoresistance	0.5
6	Inlet air temperature	PT100 3mm, 4 wires	Thermoresistance	0.5
7	Gross calorific value	Certified cylinder	-	1.0
8	Inlet gas pressure	Siemens Sitrans P DS3	Piezoresistive	0.7
9	Inlet gas temperature	PT100 3mm, 4 wires	Thermoresistance	0.5
10	Gas flow	Elster G4/10 – equipped with encoder	Volumetric+encoder	1.0
11	Air temperature	PT100 3mm, 4 wires	Thermoresistance	0.5
12	Environment pressure	DeltaOhm HD9408TBARO	Piezoresistive	1.2
13	Electrical power	ESAM MT-PA1	Wattmeter	1.0

Table 2. Instruments characteristics and measurement uncertainty

Efficiency results are the average value of three consecutive ten minutes tests with inter-repeatability within 0.5%.

Referring to equation 2 an error propagation analysis on the efficiency was performed. Using instruments measurement uncertainties reported in Table 2, uncertainty on the efficiency resulted equal to 2.2% at nominal heat output conditions and 80-60 °C flow-return temperature.

In particular, considering the following not correlated error propagation formula for a generic function F:

$$F = f(x_i, \ldots, x_M) \tag{23}$$

$$\mathrm{var}(F) = \sum_i^M \left(\frac{\partial f}{\partial x_i} \right)^2 \mathrm{var}(x_i) \tag{24}$$

It is possible to express the variance of the heat output as:

$$\mathrm{var}(\dot{Q}_{out}') = \frac{\mathrm{var}(\dot{V}_{H2O})}{\dot{V}_{H2O}^{\;2}} + B_{H2O}^{\;2} \frac{\mathrm{var}(T_r)}{\rho^2} + \frac{\mathrm{var}(\rho a}{\rho^2} + \frac{\mathrm{var}(T_r) + \mathrm{var}(T_f)}{(T_f - T_r)^2} + \frac{\mathrm{var}(cT_r) + \mathrm{var}(cT_f)}{c^2 (T_f - T_r)^2} \tag{25}$$

Considering a linear interpolation for the water density,

$$\rho = A_{H2O} + B_{H2O} T_{H2O} \tag{26}$$

While the heat input variance is:

$$\mathrm{var}(\dot{Q}_{in}) = \frac{\mathrm{var}(\dot{V}_{gas})}{\dot{V}_{gas}^{\;2}} + \frac{\mathrm{var}(p_{gas})}{(p_{gas} + p_{ref})^2} + \frac{\mathrm{var}(T_{gas})}{(T_{gas} + T_{ref})^2} + \frac{\mathrm{var}(GCV)}{GCV^2} \tag{27}$$

The relative weight of measurement on uncertainty was estimated and reported in Figure 3. The most critical measurements are water temperatures for the heat output and the gas calorific value for the heat input, and particular care should be taken in measuring these quantities. Referring to equation 2, water temperature measurements are carried out using three PT100 probes for each temperature. Gas calorific value could be measured using gas chromatography or by adopting certified cylinders, and in the first case particular care must be taken in order to correctly observe the gas mixture composition.

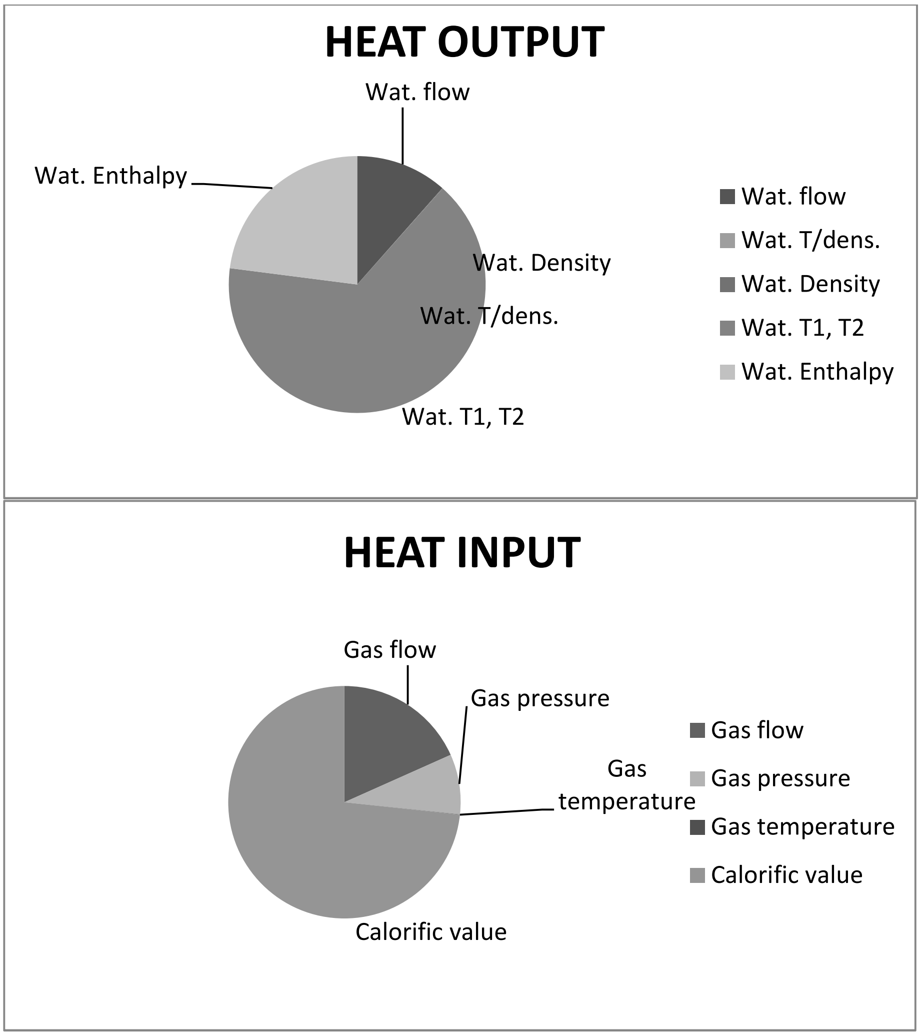


Figure 3. Relative weight of different measurements on the heat input and heat output uncertainty

For a 22kW boiler in Table 3 an example of heat input, heat output and efficiency with relative and absolute uncertainties for a certified calorific value of 2% is reported.

Quantity	Measurment unit	Rel. Unc. (2k)	Abs.unc (2k)
Q_{in}	22080.2 [W]	2.3 [%]	515.9 [W]
Q_{out}	21235.0 [W]	1.5 [%]	319.8 [W]
η_s	96.17 %	2.8 [%]	2.7%

Table 3. Relative and absolute uncertainty on heat output, heat input and water efficiency

By adopting methane certified cylinders with 1% certified uncertainty on GCV it is possible to reduce the heat input uncertainty to 1.6% and the efficiency uncertainty to 2.2%.

In order to measure water temperature transients, according to EN 13203, three radially staggered 1 mm thermocouples were adopted (Figure 5). Small thermocouples show a quicker response during rapid temperature changes when compared to PT100, accordingly to data in literature and Figure 4. (LABNET, 1998)

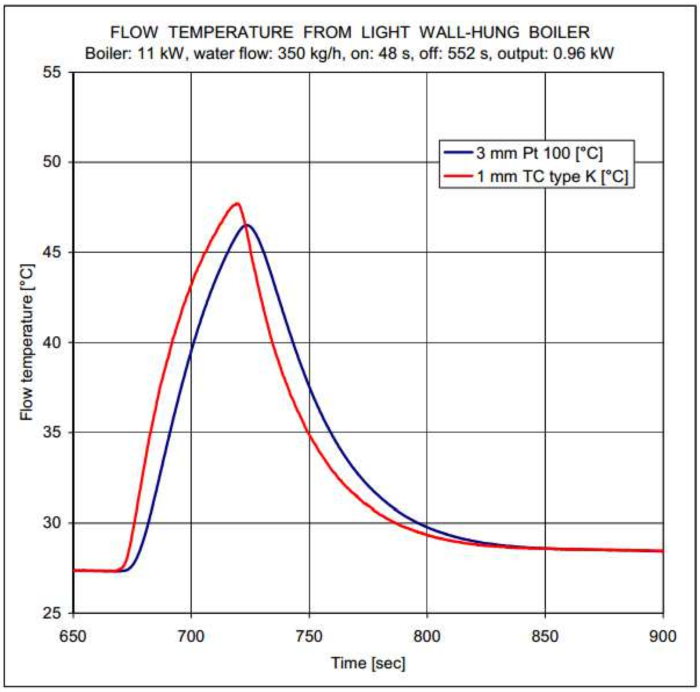


Figure 4. Transient measurement comparison between a 1mm thermocouple and a 3mm thermoresistance

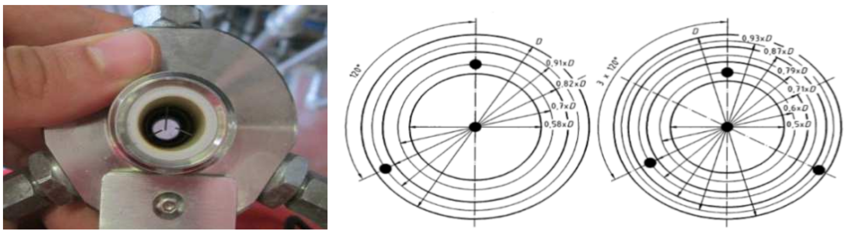
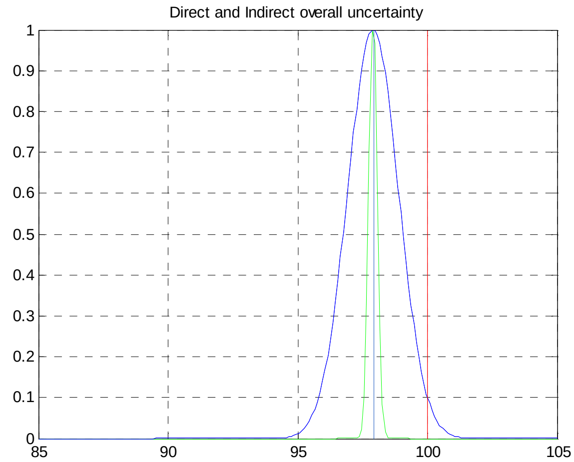


Figure 5. Thermocouples positioning with surface methods.

2.3 Indirect efficiency measurement uncertainty

Indirect efficiency analysis is proposed in literature as both an alternative efficiency measurement method and a procedure to verify correct direct efficiency measurements (ASME, 2013). Indirect efficiency method, also known as “energy balance method”, has a generally lower measurement uncertainty when compared to direct efficiency (also known as input output method). This result could apparently be justified assuming that indirect efficiency requires the estimation of the energy loss only, which is a small fraction of the total energy; this results in a limited effect on the overall efficiency uncertainty. For instance, considering 5% of the energy input as losses and a 10% (2k) relative uncertainty on this estimation, the resulting indirect efficiency would be equal to 95%+-0.5%. Comparing error propagation analysis for direct and indirect efficiency, as presented in Figure 6, indirect efficiency shows lower dispersion.



Efficiency measurement method	Value	Relative uncertainty (2k)
Direct	97.9%	2.6%
Indirect		0.3%

Figure 6. Comparison of direct and indirect estimation of efficiency in terms of measurement uncertainty

When considering the loss term composition for a full load 60/80 °C test, the most important loss term is related to flue gas, a significant part is relative to jacket loss and a negligible part connected to unburned losses (Figure 7).

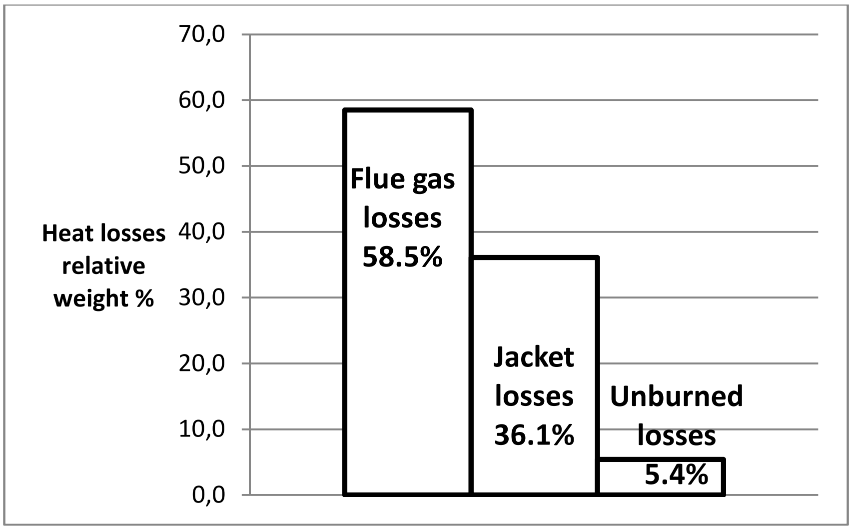


Figure 7. Relative weight of losses terms on the global heat loss of a boiler

When measuring efficiency indirectly, not considering or partially considering loss terms implies a boiler efficiency overestimation. Particular care must be taken in order to correctly measure temperature flue gas and inlet air temperature in order to not generate systematic errors. It is evident that jacket loss quantification is critical because of significant relative weight and complex valuation; furthermore, not considering part of the jacket surface would determine systematic overestimation of the efficiency.
In the following figure advantages and disadvantages of adopting direct and indirect efficiency measurements are reported. Direct measurement is

simpler, directly giving the useful effect quantification yet needs quite accurate measurements; on the other hand, indirect efficiency leads to a very repeatable result and is a useful tool for finding loss causes but needs to complexly estimate small quantities such as jacket loss.

Advantages	Disadvantages
Input/Output Method	
Primary parameters from the efficiency definition (output, input) are directly measured.	Fuel flow and fuel heating value, steam flow rates and steam properties need to be measured very accurately to minimize uncertainty
Requires fewer measurements.	Does not aid in locating source of possible inefficiency.
Does not require estimation of unmeasurable losses.	Requires the use of energy balance calculation methodology for correction of test results to standard or guarantee conditions. Corrections to standard or guarantee conditions can only be made using the energy balance methodology.
Energy Balance Method	
The primary measurements (flue gas analyses and flue gas temperature) can be made very accurately.	Requires more measurements.
Permits corrections of test results to standard or guarantee conditions.	Does not automatically yield capacity and output data.
The as-tested efficiency often has lower uncertainty because the measured quantities (losses) represent only a small fraction of the total energy.	Some losses are practically unmeasurable and value must be estimated.
The effects of fairly substantial errors in secondary measurements and estimated values are minimal.	
Sources of large losses are identified.	

Figure 8. Advantages and disadvantages of input/output method (direct) and energy balance method (indirect) for efficiency estimation (ASME, 2013).

2.4 Energy balance closing

In order to characterize the boiler, energy balance consistence could be checked. The approach undertaken in the present work is not an algebraic verification but a statistical one.

In particular, considering the following energy quantities as random variables:

$$\dot{Q}_{input} - \dot{Q}_{output} = \dot{Q}_{\Delta} \tag{28}$$

Where $\dot{Q}_{input}$ is an uncertainty affected measurement of $\dot{Q}_{in}$ and $\dot{Q}_{output}$ is the sum of $\dot{Q}_{out}$ and $\dot{Q}_{loss}$ uncertainty affected measurements, $\dot{Q}_{\Delta}$ is the energy deviation from the energy balance closing.

Considering a sample size equal to ss, the average values are:

$$\bar{\dot{Q}}_{input} - \bar{\dot{Q}}_{output} = \bar{\dot{Q}}_{\Delta} \qquad (29)$$

The expected value of equation 28 is:

$$Ex(\bar{\dot{Q}}_{input}) - Ex(\bar{\dot{Q}}_{output}) = Ex(\bar{\dot{Q}}_{\Delta}) \qquad (30)$$

So that results in equation 2:

$$\dot{Q}_{in} - (\dot{Q}_{out} + \dot{Q}_{loss}) = 0$$

The variance of $\dot{Q}_{\Delta}$ could be evaluated as follows:

$$sd^2{}_{\Delta} = \frac{\sigma^2{}_{input}}{ss} + \frac{\sigma^2{}_{output}}{ss} \qquad (31)$$

Population standard deviations σ_{input} and σ_{output} can be deduced from relative measurement uncertainties of $\dot{Q}_{input}$ and $\dot{Q}_{output}$.

2.4.1. Statistical based energy balance closure using z-test

In order to evaluate the energy balance closing, having considered balance terms as random variables, a z-test could be set up, using statistical approach measurement uncertainties and sample size would be determinant in analysis results despite of using a simple algebraic approach.
A z-value can be defined as follows:

$$z = \frac{\dot{Q}_{\Delta} - 0}{\sqrt{sd^{2}_{\Delta}}} \qquad (32)$$

Where $\mathrm{Ex}(\bar{\dot{Q}}_{\Delta}) = 0$ (33)

The following hypothesis test could be formulated:

$$H_0:\ \dot{Q}_{\Delta} = 0 \quad \text{versus} \quad H_1:\ \dot{Q}_{\Delta} \neq 0 \qquad (34)$$

Once established a significance level alpha, the hypothesis test gives as result a p-value.
If the p-value is equal to or smaller than alpha, it suggests that the observed data are inconsistent with the assumption that the null hypothesis is true, and thus that hypothesis must be rejected and the alternative hypothesis could be accepted.

Considering a normalized $\bar{\dot{Q}}_{\Delta}$ with respect to $\bar{\dot{Q}}_{input}$:

$$\bar{\dot{q}}_{\Delta} = \frac{\bar{\dot{Q}}_{\Delta}}{\bar{\dot{Q}}_{\mathrm{input}}} \qquad (35)$$

This results in a specific energy deviation from the energy balance closing, which is dimensionally an efficiency deviation.

Considering an alpha level equal to 5% and a sample size equal to 27 for the data shown in Table 4 it follows that zero hypothesis must not be rejected, so that the measurement system uncertainty is too high to discriminate an energy quantity of the order of $\dot{Q}_{\Delta}$ and the balance closure could be considered satisfied.

$\overline{\dot{Q}_{input}}$ [W]	σ_{input} [W]	$\overline{\dot{Q}_{output}}$ [W]	σ_{input} [W]	$\bar{\dot{Q}}_{\Delta}$ [W]	sd_{Δ} [W]	$\bar{\dot{q}}_{\Delta}$ [%]	p-value
22080.2	257.9	21967.0	189.4	113.2	61.6	0,513	0.066

Table 4. Hypothesis test on energy balance closing for a 22kW boiler.

The p-value increases with overlapping curves and with increasing uncertainty, which means that using poor measurement systems affected by substantial uncertainties, would result in erroneously considering the energy balance as always being satisfied. On the other hand, a very accurate measurement system would lead to not accept null hypothesis and identify the quantity $\dot{Q}_{\Delta}$ as a balance or systematic measurement error. Furthermore, lower sample size would increase distributions dispersion, having the same effect of higher measurement uncertainties. In Figure 9 the distribution of energy quantities present in Table 4 is plotted, showing overlapping curves.

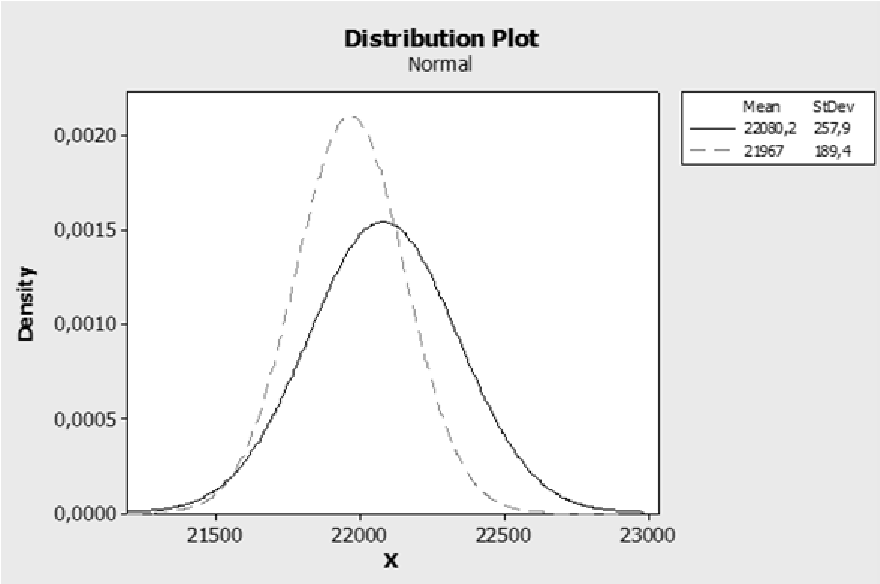


Figure 9. Comparison of the statistical distribution of heat input and global heat output measurements

2.4.2. Measurement system design

When defining the energy measurement system, it is possible to establish the deviation from the energy balance closure that system is able to discriminate by conducting Z-tests according to paragraphs 2.4.1. In particular, once having defined the alpha level, the acceptable energy deviation in absolute terms $\dot{Q}_{\Delta\_acc}$ or efficiency points $\dot{q}_{\Delta\_acc}$, the p-value could be calculated for different uncertainty measurement systems. Those measurement systems showing p-values higher than the alpha level, would be able to discriminate energy deviation equal to or higher than the selected $\dot{Q}_{\Delta\_acc}$.

As an example, considering:

$$\dot{q}_{\Delta\_acc} = 0.75\% \qquad (36)$$

And an alpha level of 10%, and maximum relative uncertainty of 3% for the input and output energy terms, referring to Figure 9, show that only measurement systems contained in the box could be accepted.

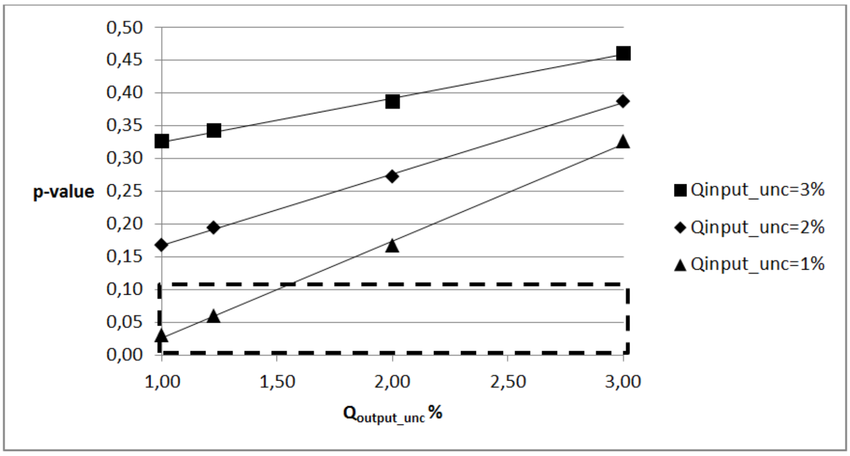


Figure 10. Measurement system design criteria based on statistical closure of energy balance.

As previously described, increasing sample size would have the same effect of improve measurement systems. The proposed methodology is not strictly connected to boilers energy balance yet could be applied to others energy closure problems.

CHAPTER 3: Risk analysis on efficiency declaration

3.1 Water heating efficiency

Erp regulation defines for water heating appliances a water heating efficiency as follows:

$$\eta_{wh} = \frac{Q_{ref}}{(Q_{fuel} + CC \cdot Q_{elec})(1 - SCF \cdot smart) + Q_{cor}} \qquad (37)$$

Where the Q_{ref} depends on the selected water heating tapping profile, Q_{fuel} and Q_{elec} are the fuel and electrical consumption necessary to satisfy the profile, Q_{cor} an ambient correction term:

$$Q_{cor} = -k\left(Q_{fuel}\left(1 - SCF \cdot smart\right) - Q_{ref}\right) \qquad (38)$$

Where k is dependent on the tapping profile, SCF and *smart* are respectively the smart control factor and the smart compliance (European Commission, 2013). Table 5 reports as an example part of three daily tapping profiles (sizes M, L and XL), in particular for the hours from 19:00 to 21:45. Complete daily tapping profiles cover one day (h 00-24).

	M				L				XL			
	Q_{tap}	f	T_m	T_p	Q_{tap}	f	T_m	T_p	Q_{tap}	f	T_m	T_p
h	kWh	l/min	°C	°C	kWh	l/min	°C	°C	kWh	l/min	°C	°C
19:00	**0,105**	3	25		**0,105**	3	25		**0,105**	3	25	
19:30												
20:00												
20:30	**0,735**	4	10	55	**0,735**	4	10	55	**0,735**	4	10	55
20:45												
20:46									**4,42**	10	10	40
21:00					**3,605**	10	10	40				
21:15	**0,105**	3	25						**0,105**	3	25	
21:30	**1,4**	6	40		**0,105**	3	25		**4,42**	10	10	40
21:35												
21:45												
Q_{ref}	**5,845**				**11,655**				**19,07**			

Table 5. Example of three tapping profiles for sanitary water efficiency determination (Erp)

An energy class could be identified, depending on the profile and on the water heating efficiency result, as shown in the following table.

Water heating energy efficiency classes of water heaters, categorised by declared load profiles, η_{wh} *in %*

	3XS	XXS	XS	S	M	L	XL	XXL
A^{+++}	$\eta_{wh} \geq 62$	$\eta_{wh} \geq 62$	$\eta_{wh} \geq 69$	$\eta_{wh} \geq 90$	$\eta_{wh} \geq 163$	$\eta_{wh} \geq 188$	$\eta_{wh} \geq 200$	$\eta_{wh} \geq 213$
A^{++}	$53 \leq \eta_{wh} < 62$	$53 \leq \eta_{wh} < 62$	$61 \leq \eta_{wh} < 69$	$72 \leq \eta_{wh} < 90$	$130 \leq \eta_{wh} < 163$	$150 \leq \eta_{wh} < 188$	$160 \leq \eta_{wh} < 200$	$170 \leq \eta_{wh} < 213$
A^{+}	$44 \leq \eta_{wh} < 53$	$44 \leq \eta_{wh} < 53$	$53 \leq \eta_{wh} < 61$	$55 \leq \eta_{wh} < 72$	$100 \leq \eta_{wh} < 130$	$115 \leq \eta_{wh} < 150$	$123 \leq \eta_{wh} < 160$	$131 \leq \eta_{wh} < 170$
A	$35 \leq \eta_{wh} < 44$	$35 \leq \eta_{wh} < 44$	$38 \leq \eta_{wh} < 53$	$38 \leq \eta_{wh} < 55$	$65 \leq \eta_{wh} < 100$	$75 \leq \eta_{wh} < 115$	$80 \leq \eta_{wh} < 123$	$85 \leq \eta_{wh} < 131$
B	$32 \leq \eta_{wh} < 35$	$32 \leq \eta_{wh} < 35$	$35 \leq \eta_{wh} < 38$	$35 \leq \eta_{wh} < 38$	$39 \leq \eta_{wh} < 65$	$50 \leq \eta_{wh} < 75$	$55 \leq \eta_{wh} < 80$	$60 \leq \eta_{wh} < 85$
C	$29 \leq \eta_{wh} < 32$	$29 \leq \eta_{wh} < 32$	$32 \leq \eta_{wh} < 35$	$32 \leq \eta_{wh} < 35$	$36 \leq \eta_{wh} < 39$	$37 \leq \eta_{wh} < 50$	$38 \leq \eta_{wh} < 55$	$40 \leq \eta_{wh} < 60$
D	$26 \leq \eta_{wh} < 29$	$26 \leq \eta_{wh} < 29$	$29 \leq \eta_{wh} < 32$	$29 \leq \eta_{wh} < 32$	$33 \leq \eta_{wh} < 36$	$34 \leq \eta_{wh} < 37$	$35 \leq \eta_{wh} < 38$	$36 \leq \eta_{wh} < 40$
E	$22 \leq \eta_{wh} < 26$	$23 \leq \eta_{wh} < 26$	$26 \leq \eta_{wh} < 29$	$26 \leq \eta_{wh} < 29$	$30 \leq \eta_{wh} < 33$	$30 \leq \eta_{wh} < 34$	$30 \leq \eta_{wh} < 35$	$32 \leq \eta_{wh} < 36$
F	$19 \leq \eta_{wh} < 22$	$20 \leq \eta_{wh} < 23$	$23 \leq \eta_{wh} < 26$	$23 \leq \eta_{wh} < 26$	$27 \leq \eta_{wh} < 30$	$27 \leq \eta_{wh} < 30$	$27 \leq \eta_{wh} < 30$	$28 \leq \eta_{wh} < 32$
G	$\eta_{wh} < 19$	$\eta_{wh} < 20$	$\eta_{wh} < 23$	$\eta_{wh} < 23$	$\eta_{wh} < 27$	$\eta_{wh} < 27$	$\eta_{wh} < 27$	$\eta_{wh} < 28$

Table 6. Water heating energy efficiency classes of water heaters organized by declared load profiles (Erp)

Ten energy classes could be identified for a wide range of boilers of water heaters; classes are generally separated by 3 or more efficiency points. Within Erp regulation, a market access ban has been set on efficiency in order to exclude poor efficiency boilers from the market. This ban enters in force with the regulation and becomes more stringent starting in 2017. For an XL sized water heater, the minimum allowed efficiency goes from 30% in 2015 to 38% in 2017 (Table 7).

REQUIREMENTS FOR WATER HEATING ENERGY EFFICIENCY

(a) From 26 September 2015 the water heating energy efficiency of combination heaters shall not fall below the following values:

Declared load profile	3XS	XXS	XS	S	M	L	XL	XXL	3XL	4XL
Water heating energy efficiency	22 %	23 %	26 %	26 %	30 %	30 %	30 %	32 %	32 %	32 %

(b) From 26 September 2017 the water heating energy efficiency of combination heaters shall not fall below the following values:

Declared load profile	3XS	XXS	XS	S	M	L	XL	XXL	3XL	4XL
Water heating energy efficiency	32 %	32 %	32 %	32 %	36 %	37 %	38 %	60 %	64 %	64 %

Table 7. Water heating energy efficiency bans for different load profiles (Erp)

3.2 Space heating seasonal efficiency

For boiler space heating appliances a seasonal space heating efficiency could be written as follows:

$$\eta_{seas} = \left(0.85\eta_{s\_1} + 0.15\eta_{s\_4}\right) - \left(F_1 + F_2 + F_3 + F_4\right) \quad (39)$$

Where η_{s_4} and η_{s_1} are respectively the full load stationary efficiency at 60/80 °C of water return/flow temperatures (operating condition 4) and the part load efficiency at 30% of the nominal heat output (operating condition 1). For gas supplied boilers:

$$F_1 = 3\% \quad (40)$$

Others coefficients could be expressed as follows:

$$F_2 = \frac{2.5\left(0.15P_{b\_el\_nom} + 0.85P_{b\_el\_MR} + 1.3P_{eb}\right)}{0.15\dot{Q}_{out\_4} + 0.85\dot{Q}_{out\_1}} \quad (41)$$

$$F_3 = 0.5\frac{\dot{Q}_{stbyls}}{\dot{Q}_{out\_4}} \quad (42)$$

$$F_4 = 1.3\frac{\dot{Q}_{ign}}{\dot{Q}_{out\_4}} \quad (43)$$

Where $P_{b\_el\_nom}$, $P_{b\_el\_MR}$ and P_{eb} are respectively the electrical consumption in conditions 1,4 and boiler standby, $\dot{Q}_{stbyls}$ is the boiler heat standby losses (paragraph 4.1.1) and $\dot{Q}_{ign}$ is the ignition burner consumption. Erp Regulation sets different energy efficiency classes, some of them reachable only using coupled systems (A+, A++, A+++). Furthermore a minimum efficiency ban equal to 86% has been set.
As a consequence, referring to the following table, from the 26th of September 2015, boilers with efficiency class within G and B (lower than 86%) will no longer be allowed to be sold on the EU market.

Seasonal space heating energy efficiency classes of heaters, with the exception of low-temperature heat pumps and heat pump space heaters for low-temperature application

Seasonal space heating energy efficiency class	Seasonal space heating energy efficiency η_s in %
A^{+++}	$\eta_s \geq 150$
A^{++}	$125 \leq \eta_s < 150$
A^{+}	$98 \leq \eta_s < 125$
A	$90 \leq \eta_s < 98$
B	$82 \leq \eta_s < 90$
C	$75 \leq \eta_s < 82$
D	$36 \leq \eta_s < 75$
E	$34 \leq \eta_s < 36$
F	$30 \leq \eta_s < 34$
G	$\eta_s < 30$

Table 8. Seasonal space heating efficiency classes (Erp)

Erp regulations sets ban on other technical aspects of the products such as noisy level, thermal insulation and pollutant emissions.

3.3 Surveillance process

"For the purposes of assessing the conformity with the requirements (…), Member State authorities shall test a single water heater, hot water storage tank (…) and provide the information on the test results to the authorities of the other Member States. If the measured parameters do not meet the values declared by the supplier within the ranges set out (…), the measurement shall be carried out on three additional water heaters (…) The arithmetic mean of the measured values of these three water heaters, hot water storage tanks, (…) shall meet the values declared by the supplier within the range set out (…)." (European Commission, 2013)

Example of tolerances on regulation surveillance are of 8% for η_{seas} and 5% both on Q_{fuel} and Q_{elec} concerning η_{wh}.
A risk analysis has been performed on previously described statistical surveillance process.
Data uncertainty was evaluated considering production and laboratory-related uncertainties. In particular, components and processes most involved in determining boiler efficiency were considered in making assumptions and using experimental data to evaluate their effect on efficiency. Concerning lab-uncertainty, measurement uncertainty was evaluated using error propagation theory for equations 37 and 39, furthermore sources of repeatability factors due to testing method have been considered.
In Figure 11, production related aspects and laboratory factors are listed and associated with η_{wh} and η_{seas}. In particular it is clear that aspects involved in determining water heating efficiency are, also because of the test structure, more numerous and critical, being composed of transients

and temporized phases. For storage products, an additional source of non-repeatability comes from storage related phenomena such as thermal stabilizations and water temperature stratification.

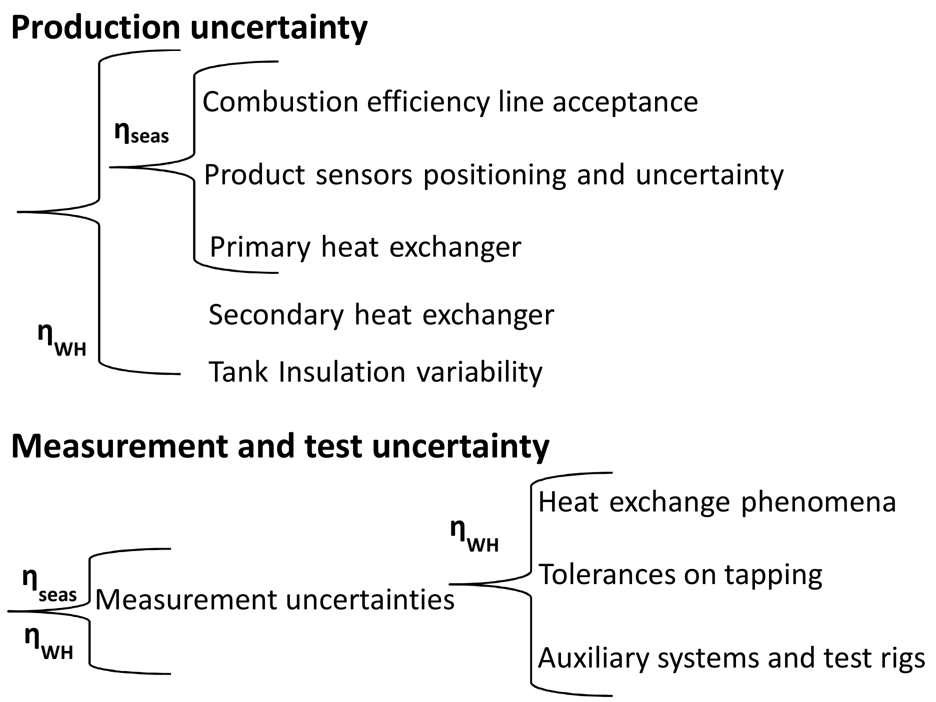


Figure 11. Laboratory and production effects on water heating and space heating efficiencies uncertainty

A calculation was performed in order to quantify production, instruments and test uncertainties, discriminating storage products from non-storage, for water heating and space heating efficiencies. Results are reported in the following table and basically show that global uncertainty is significant. Concerning laboratory uncertainty, this is more critical for water heating measurements, in particular for storage water heaters, where values are reached on the order of 8% (2k). Production effects on efficiency uncertainty prove to be not very influential when compared to laboratory uncertainty. Surveillance lab uncertainty is assumed to be the maximum instrumental allowed by regulation plus test uncertainty.

	u_{rel} (2k)	**η_{WH}**	**η_{seas}**
Lab	Instruments	4.0%	3.5%
	Instruments + test no storage	6.5%	
	Instruments + test storage	8.06%	
Production and process		1.06%	0.80%
External Lab	Max instrumental uncertainty allowed	5.0%	4.0%
	Max instrumental uncertainty allowed+ test storage	8.6%	
	Max instrumental uncertainty allowed + test no storage	7.1%	

Table 9. Laboratory and production uncertainty contribution on water heating and seasonal space heating efficiency

3.3.1. Safety margin on efficiency

Because of uncertainty on efficiency data, in order to avoid conflicts with surveillance bodies the surveillance process itself has been modeled to identify efficiency value safety reductions which could decrease the risk of non-conformity. As reported in paragraph 3.3 and represented in Figure 12 surveillance takes from the market one sample and in case of non-conformity other tree samples.

Products on the market present intrinsic efficiency variability due to production uncertainty and aging effects (such as polyurethane insulation decay). Internal laboratory equipment has a known uncertainty on measurement, while for surveillance laboratories the maximum allowable measurement uncertainty has been considered. The result of the analysis is the statistical distribution of the differences of one sample and another three samples, between internal and surveillance measured efficiency.

Statistical difference is then compared to regulation-allowed difference limits in order to determine the risk of non-conformity.

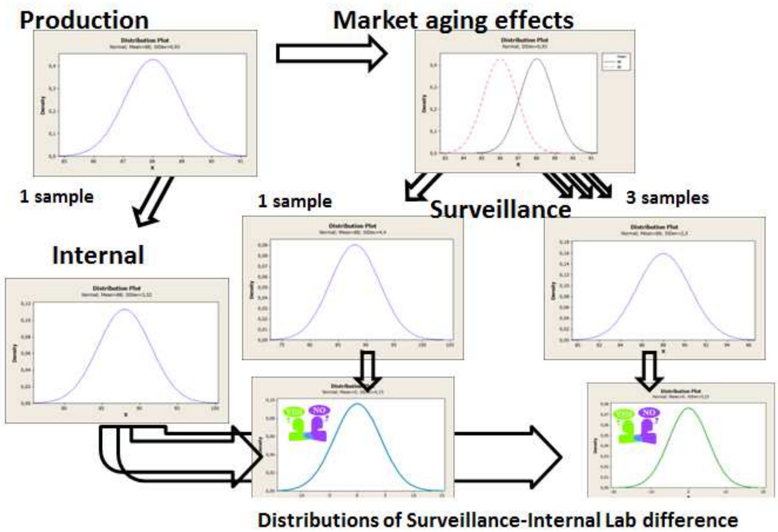


Figure 12. Surveillance on declared data process schematization.

3.3.2. Safety margin analysis results

Concerning space heating efficiency η_{seas}, the surveillance tolerance is equal to 8%, results show that because of the significantly high surveillance margins, the risk of non-conformity is lower than 1%.
Concerning water heating efficiency, regulation currently establish for water heating products two different tolerances in case they are combination heaters (both space and water heaters) or simply water heaters. In the first case surveillance tolerance is equal to the space heating one (8%) while for water heaters it is on Q_{fuel} and Q_{elec} , equal to 5% and results in an approximated 4% surveillance on η_{wh} .
Regarding combination heaters and risk on water heating efficiency, for storage boilers, as shown in Figure 13, a residual risk of about 1% is pointed out. Considering instantaneous appliances, affected by lower uncertainty than storage ones, the risk level is lower still.

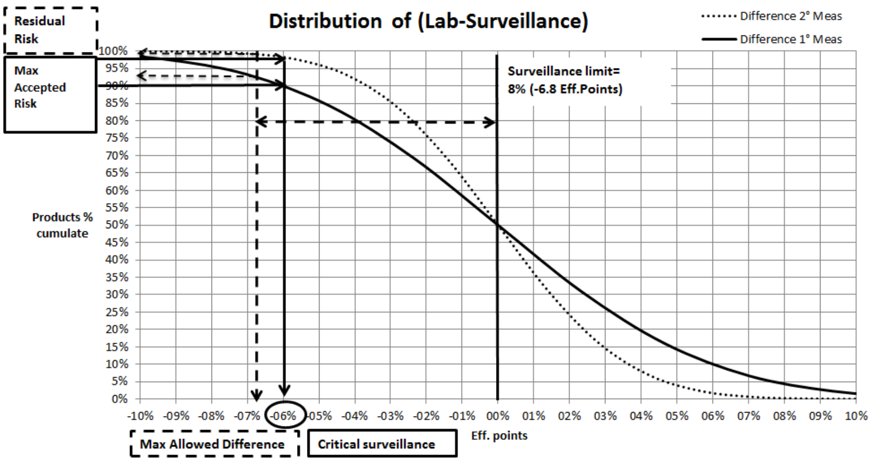


Figure 13. Surveillance tollerance on water efficiency analysis on storage boiler assuming an acceptable risk of non-conformity.

Otherwise, considering a gas storage water heater, because of the more restrictive tolerance (about 4% on the efficiency) the residual risk is of the order of 25% on the first sample and of 12% on three samples measurement. In this case a declaration margin could be necessary.

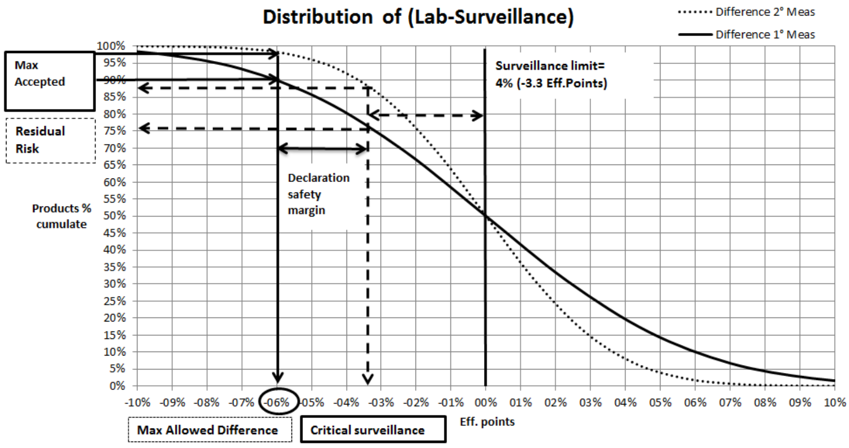


Figure 14. Surveillance tollerance on water efficiency analysis on storage water heater assuming an acceptable risk of non-conformity.

As a consequence of high measurement uncertainty a significant probability of wrong labeling occurs, especially for borderline data. (Kessen, 2007)

CHAPTER 4: Products characterization and efficiency improvement

4.1 Insulation improvement for tanks and Boilers

Insulation improvement is one of the mostly effective and simple way to increase boiler efficiency both in water heating, especially for storage products, and in space heating working conditions.
An experimental test bench has been constructed in order to test insulation of boilers according to chapter 9.3.2.3.1.3 of EN 15502-1 standard (CEN , 2013), and tanks according to chapter 9.4.2.1.3.
The bench is composed of an electrical resistance, a variable flow pump and a heat exchanger. Temperature controlled water is blown through the device under test (DUT); air temperature, water flow rate and electrical absorption are monitored. The electrical resistance and the pump are piloted by a PWM signal, an Arduino UNO board was used. Water and room temperatures are measured using 3mm PT100 and thermocouples type T, water flow is measured using a vortex flow meter, electrical power absorption through a monophasic wattmeter.
Temperatures, water flow and electrical measurements were carried out using NI labview and a Compact DAQ acquisition system. Software was developed in Labview in order to manage the different phases of the test, to check stability conditions established by the standard and drive pump and resistance. Hydraulic scheme of the test rig is shown in Figure 15.

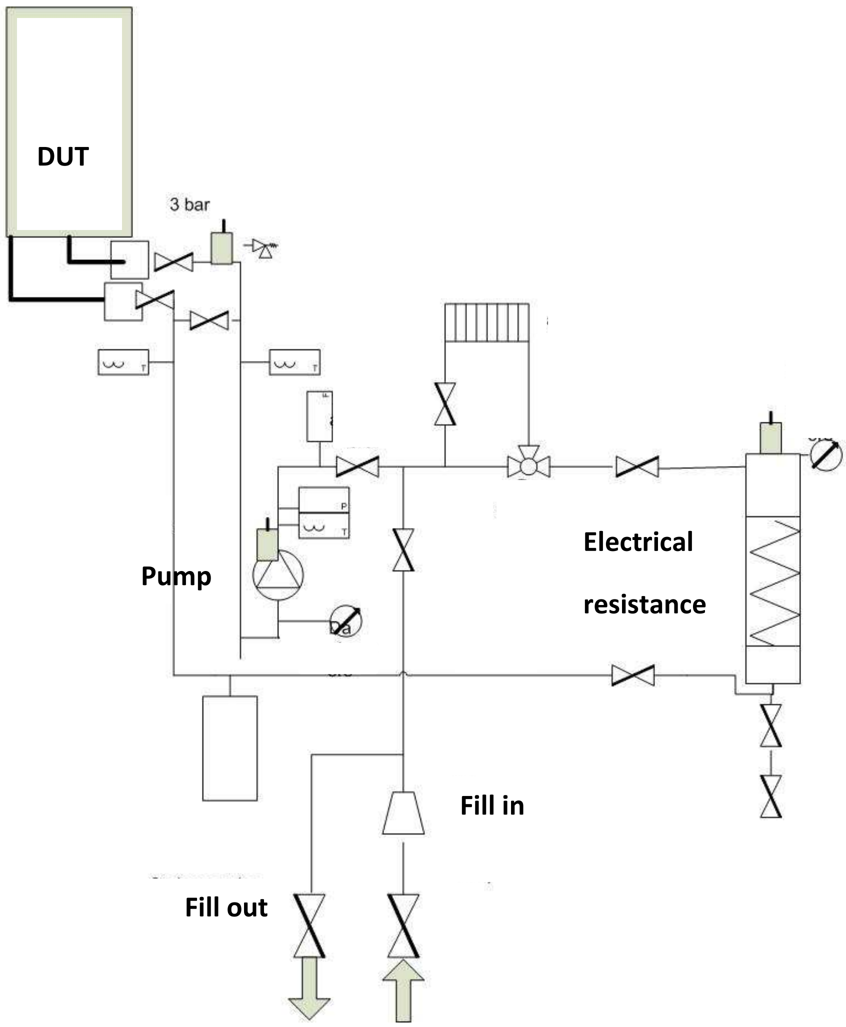


Figure 15. Insulation measurement test rig composed by electrical resistance and pump controlled by PID in order to deliver constant temperature water

The test bench has been located in a climate chamber and air temperature was set to 20°C.

The bench heat loss $\dot{Q}_{bench\_loss}$ could be evaluated measuring electrical power absorbed by the resistance in order to keep bench at a constant temperature, which is equal to the overall electrical absorption ($\dot{Q}_{el\_meas}'$) minus others auxiliaries absorptions (pump and electronics) $\dot{Q}_{el}$.

$$\dot{Q}_{el\_R}' - \dot{Q}_{el} = \dot{Q}_{bench\_loss} \tag{44}$$

After bench characterization, test could be done with DUT and measuring $\dot{Q}_{el\_R}$.Then the DUT loss is equal to $\dot{Q}_{el\_R}$ minus bench losses and electrical auxiliaries absorption. Care must be taken in regulating the pump in order to work at the same power consumption.

$$\dot{Q}_{DUT\_loss} = \dot{Q}_{el\_R} - \dot{Q}_{bench\_loss} - \dot{Q}_{el} \tag{45}$$

In Figure 16 a scheme of the test phases is presented.

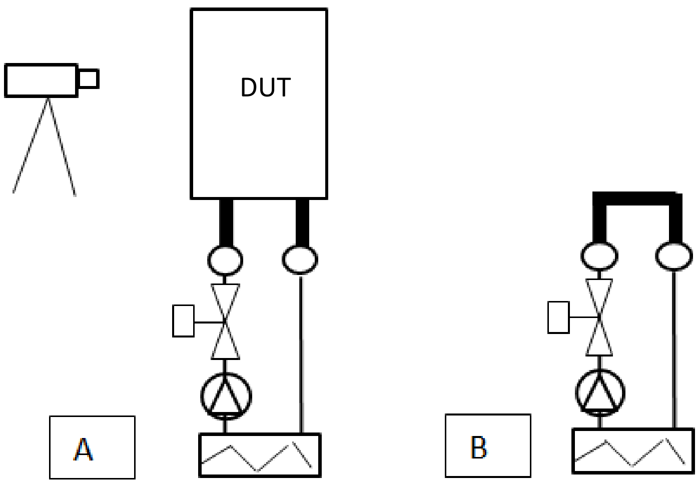


Figure 16. B represents the test bench loss characterization while A the DUT measurement test.

4.1.1. Boiler losses

The boiler standby loss $\dot{Q}_{stbyls}$ is a measure of boiler radiative-convective losses for an air temperature of 20°C and a water temperature of 50°C. (CEN , 2013) The constructed test rig has been used in order to evaluate boilers standby losses for different products, in Figure 17 results for three boilers are reported.

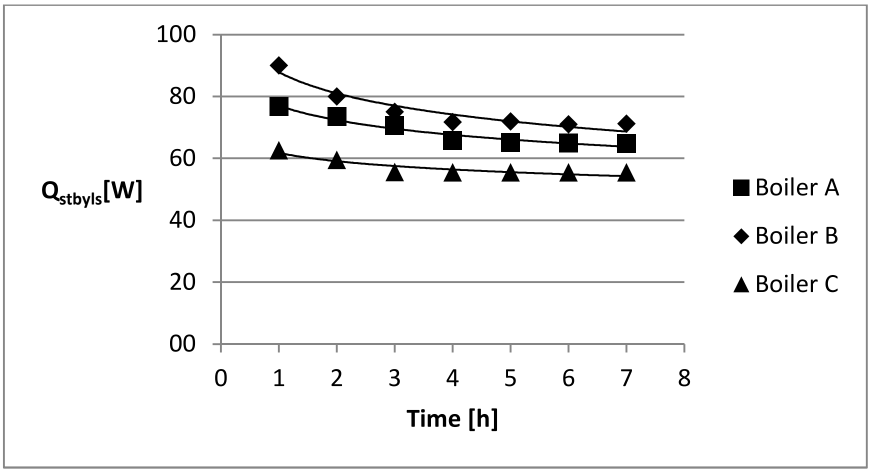


Figure 17. Boiler standby losses results for three differently insulated boilers

Results show that such tests lead to values between 50 and 70 W and, furthermore, tests generally take a significant amount of time to reach steady and stable results. The need of a climate chamber is evident if considering the typical temperature excursion present in a non-controlled environment over so long a time.

4.1.2. Tank losses

The constructed test rig was used in order to evaluate tank losses according to EN 15502-1. The test provides that water at a controlled temperature of 65°C must be provided to the tank until some thermal

stabilization criteria are satisfied. In particular, inlet and outlet temperature difference must be lower than 1 K for more than 15 mins and inlet temperature variation must be within 1 K. Room temperature must be equal to 20°C; some variations are admitted. (CEN , 2013)
After stabilization the tank inlet and outlet are sealed and the tank is left to cool for 24 hours (t_{tank_test}).
After the cooling period the tank water temperature is measured.
Considering a tank volume V_{tank} and a zero-dimensional model, water temperature cooling down inside the tank could be written as follows:

$$\frac{dT}{dt} = \frac{\bar{h}\,A_{tank}}{\rho c_{H2O} V_{tank}} (T_{tank} - T_{air}) \qquad (46)$$

Where $\bar{h}$ represents an equivalent coefficient of heat exchange, which includes convective, radiative and conductive over insulation exchanges.

Considering the following equation:

$$\bar{h}\,A_{tank} = \frac{\rho c_{H2O} V_{tank} \ln\left(\frac{T_{tank\_0} - T_{air}}{T_{tank\_24} - T_{air}}\right)}{t_{tank\_test}} \qquad (47)$$

A cooling coefficient (CR) could be defined as follows:

$$CR = \frac{\bar{h}\,A_{tank}\,t_{tank\_test}}{V_{tank}} = \rho c_{H2O} \ln\left(\frac{T_{tank\_0} - T_{air}}{T_{tank\_24} - T_{air}}\right) \qquad (48)$$

And the tank heat loss for 65°C water temperature and air temperature of 20°C could be calculated as follows (CEN , 2013):

$$\dot{Q}_{tank\_loss} = \frac{\rho c_{H2O} V_{tank} \ln\left(\frac{T_{tank\_0} - T_{air}}{T_{tank\_24} - T_{air}}\right)}{t_{tank\_test}} 45 \qquad (49)$$

In Table 10, measurements of tank losses are presented for two different materials, polyurethane and polystyrene for a 40 liters V_{tank}.

	T_{tank_0}	T_{tank_24}	$\dot{Q}_{tank\_loss}$	CR
Polyurethane	61.3	38.4	77.4	1.1
Polystyrene	65.0	32.9	99.4	1.38

Table 10. Tank heat loss measurement for two different insulation material and a volume of 40 liters

4.1.3. Thermal camera measurements

Thermal camera temperature measurements were carried out in order to evaluate heat loss. A Flir thermal camera for R&D purposes was used; boiler and tank surface temperatures were measured. Thermal camera measurements were compared to surface PT100 and thermocouples measurements. A Matlab code was developed in order to calculate from a thermal image the correspondent heat loss. Every pixel of a thermographic image was modeled as a constant temperature radiative convective heat exchange problem. In particular the jacket losses could be estimated as follows:

$$\dot{Q}_j = \sum_{k}^{pixels} J_k \alpha_k \left(T_{k\_j} - T_{k\_air}\right) \qquad (50)$$

The coefficient of convective heat exchange was evaluated using Churchill-Chu and Mc-Adams correlations based on dimensionless numbers. The calculation uncertainty was assumed to be equal to 20% which is derived from the most relevant uncertainty term: the convective heat exchange coefficient. Other sources of uncertainty are the temperature field discretization, emissivity estimation and radiation related approximations, not perpendicularity errors and temperature measurement uncertainty.

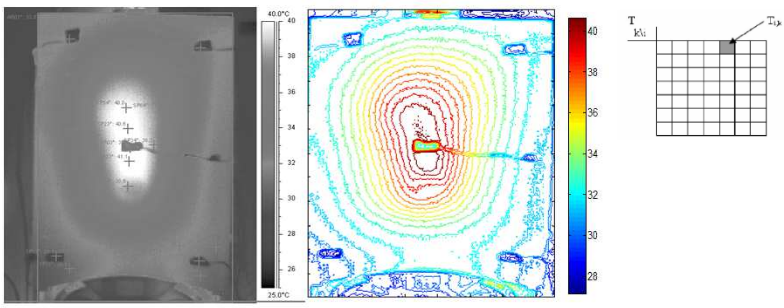


Figure 18. Boiler frontal thermal camera measurement during burning phase and correspondent Matlab elaboration

In the following table surface losses calculation are reported for a 22kW boiler: results show that most of the heat loss is on the front and left surfaces of the boiler, which could be explained by the primary heat exchanger positioning.

The global heat loss for the analyzed boiler, which has no insulation panels, corresponds to about 0.8% efficiency points. The rear side of the boiler was insulated with a 2 centimeter thick black wood panel. (Table 11)

Surface	**Loss [W]**
Top	13.2
Bottom	20.4
Front	59.6
Lateral left	52.2
Lateral right	27
TOT	172.4
Tot [eff. points]	0.78 %

Table 11. Heat losses measurement of the different sides of a 22kW boiler using thermography and correspondent relative efficiency point.

Measurements comparison

Thermal camera measurements have been compared to electrical power measurement results, and water temperature decrease method for a 40 liters tank. Results are shown in Table 12.

Methodology	Heat loss [W]	Difference %
Thermal camera	94.2	-
Water temperature decrease	91.9	2.2
Electrical power	80.9	14.2

Table 12. Comparison of heat loss measurement methodologies for an 40 liters insulated tank

The most reliable and stable result can be obtained using the electrical power method, although this method has some limitations: it cannot be used to measure boiler jacket loss during working conditions and takes a long time. Water temperature decrease method could be used only for tanks and has the advantage of showing exiting water temperature, which is the most immediate and intuitive effect of the lack of insulation. Thermal camera method is in general the quickest way to estimate surface heat loss and it can be used in any working condition but has a significant uncertainty.

4.2 Boiler tank draw-off analytical model

The boiler/tank draw-off modeling is useful to foresee boiler behavior within aspects connected to hot water production capacity and could also be a useful tool for tank-boiler first-order dimensioning.
Tank water distribution is generally a 3D non-stationary mathematical problem which could be solved using tools such as FEA and CFD. Temperature stratification effects, which mostly characterize the water temperature inside a tank, are a spatial phenomenon so that at least two or one-dimensional simulation is necessary. Nevertheless, for some applications, a concentrate parameters tank modeling would be sufficient in order to obtain approximated results.
In the present work an analytical zero-dimensional model of boiler-tank has been performed. Considering Figure 19 the water is supposed to go through the heat exchanger and then into the tank.

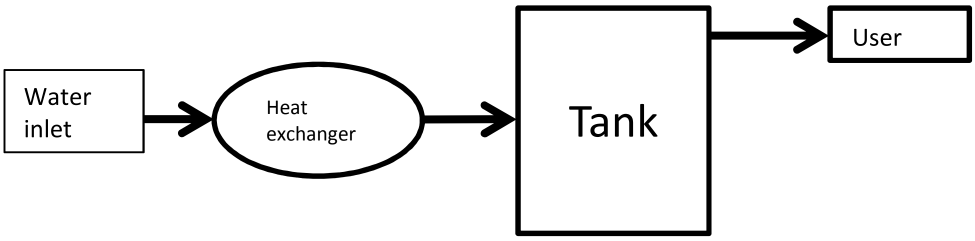


Figure 19. Boiler burner/tank logic scheme for sanitary draw-off

Considering the following equations:

$$\rho c_{H2O} V_{tank} \frac{dT_{tank}}{dt} = \rho c_{H2O} \dot{V}_{H2O} (T_{tank\_in} - T_{H2O\_out}) \tag{51}$$

$$\rho c_{H2O} \dot{V}_{H2O} (T_{tank\_in} - T_{cold}) = \langle \begin{matrix} \dot{Q}_{out} \text{ for\_t} > t^* \\ 0\text{\_for\_t} < t^* \end{matrix} \tag{52}$$

$$T_{tank}(t = 0) = T_{set} \tag{53}$$

$$T_{cold} = f \tag{54}$$

Where T_{tank_in} and T_{H2O_out} are the tank temperature of incoming and outgoing water, T_{cold} the cold water temperature which is general a known function of the time, t^* is the time in which, depending on the boiler control, the burner turns on.

Considering previous equations it is evident that the physical and mathematical connection between T_{H2O_out} and dT_{tank} is not modeled because derives from stratification effects. Indeed, in the presented model it is assumed that these two temperatures are equal. This means that ideally, draw off temperature is always equal to the mean tank temperature. Thanks to this assumption it is possible to analytically solve equation 51.

The specific flow rate is an important parameter for water heaters performance evaluation in terms of hot water draw-off capacity, this quantity can be expressed as follows:

$$SF = \frac{\left(\int_0^{10mins} \rho \dot{V}_{H2O} dt\right)}{10} \frac{\bar{T}_{H2O\_out} - \bar{T}_{cold}}{30} \qquad (55)$$

Where $\bar{T}_{H2O\_out}$ $\bar{T}_{cold}$ are the mean temperature of the delivered hot water and incoming hot water for a 10min test. For volume tanks small enough to replace the initial content of water during the ten minute test, the SF could be properly approximated using the previously shown model. In Figure 20 and 21 for a 40liters and 22kW storage boiler experimental and model results in terms of T_{H2O_out} and SF are reported.

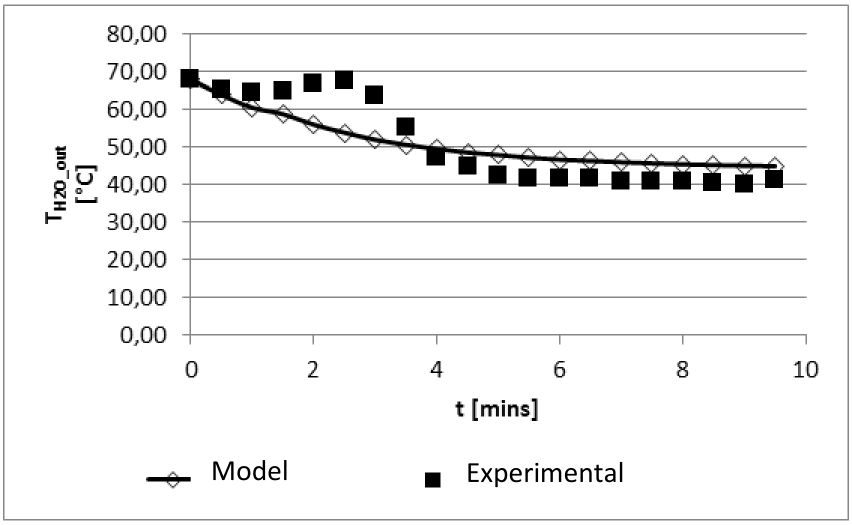


Figure 20. Experimental and model water temperature draw-off for a 22kW and 40liters storage boiler and a 10l/min flow rate. Experimental draw-off show firstly a peak due to tank temperature stratification and following a valley, whereas modeled temperature maintains an average value

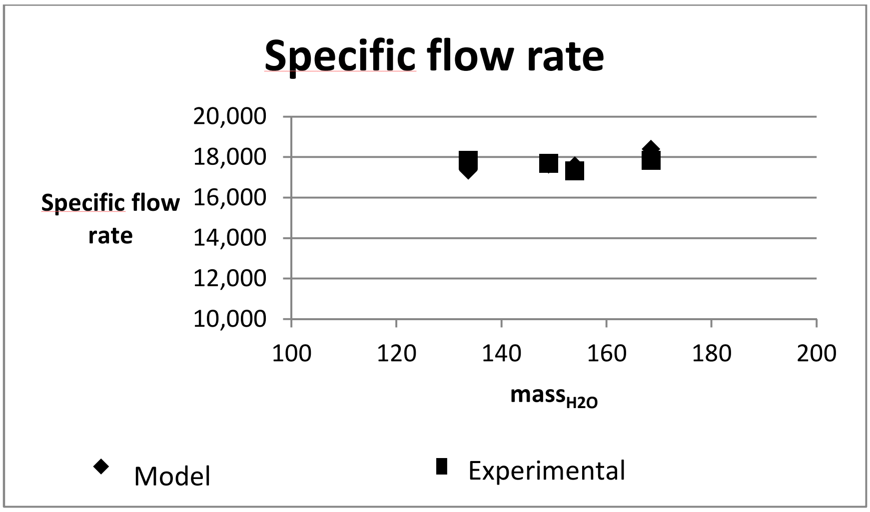


Figure 21. Experimental and modeled specific flow rate is reported for different tapped water quantities for a 22kW and 40 liters storage boiler. Model well approximates SR experimental results.

Results show that by making the previously reported assumptions, the model is able to foresee average energy inside the storage, providing comparable results in terms of specific flow rate. Stratification effects could be included by adding appropriate experimental based characterizations to the analytical draw-off temperature found.

CHAPTER 5: Experimental and numerical studies for new products feasibility

5.1 Conventional boiler for Erp

Conventional boilers are sized in order to not determine condensate formation. Considering the burning process of methane, flue condensation heat represents a significant amount of energy, which corresponds to almost 10% of the total methane chemical energy available. In order to allow boilers to extract this energy amount, a specific design is necessary, especially in terms of heat exchange surface and condensate discharge systems. Condensate indeed is corrosive and, when not correctly evacuated, could lead to quality issues. This leads to adopting a discharge system for condensate evacuation in condensing boilers.

Thus, condensing boilers are generally more efficient than standard boilers but also more expensive and sophisticated. Erp regulation sets a ban limit to the space heating efficiency which is equal to 86%; currently space heating efficiency for standard boilers is of the order of 83%. This means that without any significant product development, standard boilers could no longer be sold in Europe after Erp enters in force. In the present PhD thesis work a technical analysis has been carried out in order to evaluate the possibility of increasing efficiency in conventional boiler enough to satisfy the Erp limitations.

5.1.1. Boiler methane combustion

The equation of combustion for pure methane (CH_4) is the following:

$$CH_4 + 2O_2 \Rightarrow CO_2 + 2H_2O + 37.74\ kJ/m^3 \quad (56)$$

Combustion ideally uses pure oxygen; air contains about 21% oxygen and 79% nitrogen by volume and is readily available while pure oxygen must be processed and generally the cost outweighs the benefit of increased combustion control. Using air instead of oxygen, one cubic meter of methane (at standard temperature and pressure) burns completely with 9.53 cubic meter of air as shown below:

$$CH_4 + 2O_2 + 7.53N_2 \Rightarrow CO_2 + 2H_2O + 7.53N_2 + 37.74\ kJ/m^3 \quad (57)$$

The ratio of 9.53 is known as the stoichiometric air/fuel ratio; the shown heat of combustion is released when the fuel burns. Ideally, the right amount of air to completely burn all the fuel must be provided, yet, to ensure that all the fuel is burned, some amount of excess air is delivered. Flue gas loss increases with excess air because of the higher amount of air discharged at high temperature. On the other hand, a lack of oxygen generates unburned loss. Oxygen present in the flue gas is directly related to air excess adopted so that it could be used to find the optimal air excess as shown in Figure 22.

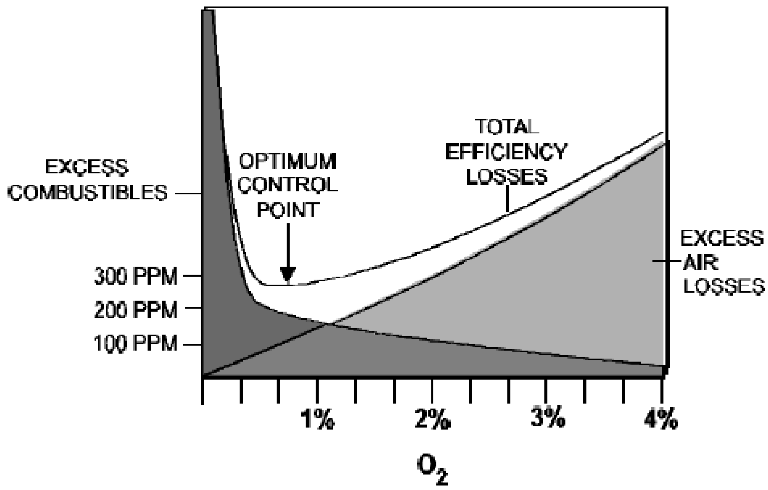


Figure 22. Boiler combustion loss as a function of oxygen concentration in the flue gas, which is correlated to the air excess.

Furthermore NO_x generation increases with air excess, because of higher presence of N_2 and higher combustion temperature.

The air excess (e) represents the percentage of air excess compared to the stoichiometric quantity. Combustion conditions could be evaluated using the Ostwald triangle, which relates e, CO, CO_2 and O_2 concentrations. The graph is reported in Figure 23. By using this graph it is possible to identify and set combustion conditions of the boiler. In particular, in the present analysis, only combustion conditions CO concentrations lower that 1% are considered.

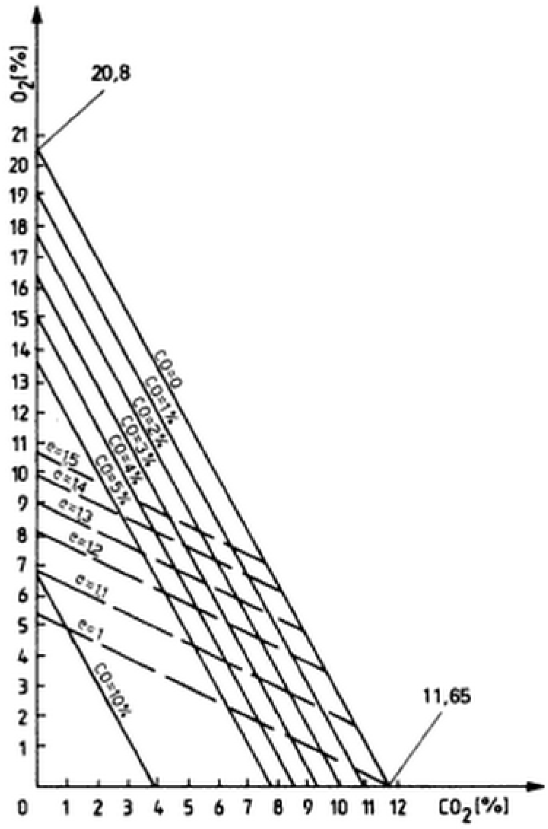


Figure 23. Ostwald triangle for methane – The triangle correlate air excess "e" with CO, CO_2 and O_2 concentrations determining the combustion conditions.

Standard boiler efficiency improvement

Undesired effects have to be avoided when increasing standard boiler efficiency, particularly formation of condensate and excessive unburned emissions (CO etc.). Condensation phenomena generally depend on the temperature reached by wet flue gas, in particular the wet flue starts condensing when reaching dew point conditions. This could happen both with decreasing return water temperature in the boiler and reducing the heat provided. In the second case, indeed, an extra-sizing heat exchanger effect occurs, increasing heat exchange and pushing flue temperature near to condensation limits. This is evident in Fgiure 24, where the effects of condensation are shown in terms of efficiency and reported as functions of water temperature and heat provided.

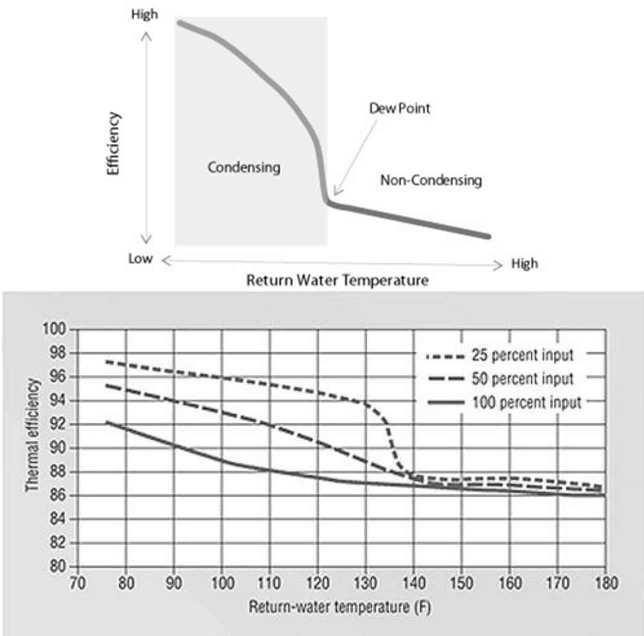


Figure 24. Boiler efficiency increase due to condensation as a function of water return temperature and heat input

In the present thesis work, in order to improve efficiency, a standard 22kW boiler has been experimentally characterized. Fan rpm have been reduced and gas valve adjusted in order to maintain nominal air excesses and reduce heat provided without determinate previously defined undesired effects.
In particular when reducing air provided, CO% and condensate formation were monitored and tests repeated for different return temperatures. The jacket loss was not considered , making the assumption of an ideally insulated boiler, so that indirect efficiency was evaluated.

Different boiler configuration were found to satisfy non-condensate conditions and limited CO concentration. Some of the configuration reported in Table 13 are theoretically able to satisfy the Erp limits.

$\dot{Q}_{in}$ (kW)	CO_2 (%)	CO (ppm)	η_{ind} (%)	Discharge system
17.6	8,33	42	88.2	coaxial 1 m
13.6	7,11	4,9	88.6	coaxial 1 m
9,6	5	8	87.5	coaxial 1 m

Table 13. Indirect efficiency, Heat input, CO ppm and CO_2 % for an increased performance standard boiler supposing no jacket loss.

Tests at return water temperatures of 37°C and 40°C showed undesired presence of condensate, while at 47°C no condensate was found. Neglecting electrical consumption, the seasonal efficiency resulted to be:

$$\eta_{seas} = (0.85\eta_1 + 0.15\eta_4) - (3) = 85.6\% \qquad (58)$$

Which is, despite the approximations, a value compliant with Erp limits. Other tests were carried out on the boiler in order to verify compliance with other standard requirements such as: gas limits cold starts and special gas conditions. (CEN , 2013)
Compliance with G231 cold starts has been found but boiler was not able to succeed in G21 overload test. Maximum flue gas length tests shown furthermore a maximum length of 1 meter for the discharge system. It has

been estimated that such boilers would have a 28% risk of surveillance non-conformity.

5.2 Ultramodulating condensing boilers

In the present work a commercial condensing boiler with nominal heat output of 22kW and maximum modulation equal to 10 is characterized in terms of efficiency and electrical consumption. Efficiency and consumption measurements are carried out, an uncertainty analysis on measurements is performed (De Paepe M., 2013) as stated in paragraph 2.
Afterwards the boiler was set up to work out of the nominal modulation in a so-defined “ultramodulation” range, then boiler characterization was extended. When forcing the boiler in these conditions technical limits were pointed out and analyzed.
On-field working scenarios were simulated for the boiler, in order to analyze annual gas and electrical consumptions. Different approaches could be undertaken to model boiler consumption and energy demands, ranging from unsteady modeling to a days-degree approach.
(De Rosa M., 2014) (Bettanini E., 2013) (ASHRAE, 2005) (ANSI/ASHRAE, 2007)
In the present thesis work a stationary model based on bin method has been implemented in order to simulate ultramodulation behavior on field.

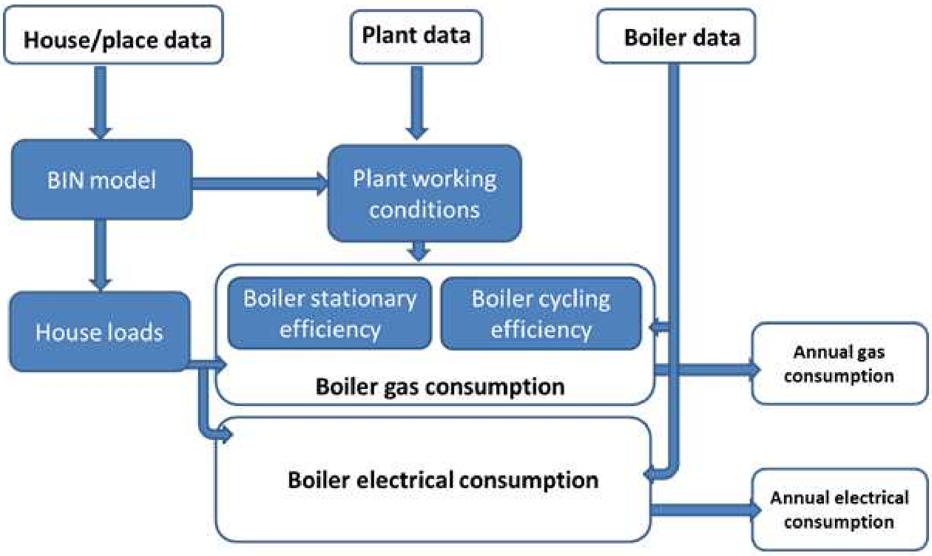


Figure 25. Model overview: on the top model inputs and on the right the model output. Model is composed by different sub-models: bins model, house loads calculation, plant model and boiler consumption submodels

The model developed is composed by the different sub-models which are shown Figure 25.
Model inputs are: city, house losses, plant type and boiler data, while outputs are annual gas and electrical consumption.
The bin model provides annual external temperature frequency to the house loads sub-model on the basis of the selected location. Within the plant sub-model, boiler-plant working conditions are identified as a function of plant type and external conditions.
Eventually, consumption sub-models, based on boiler experimental characterization, evaluate annual gas and electrical consumption as a function of house loads and working conditions.

5.2.1. Annual gas consumption calculation

Bin method identifies for a certain place a set of discrete external air temperatures T_{e_i} and an hourly frequency of occurrence h_i, which are representative of the historical external temperature distribution. Annual temperature frequency was matched to the available heating period of the place set by local regulation (D.P.R. n.412, 1993).

Assuming a room temperature T_{room} equal to 20°C, building specific heat losses H_{eq} , discrete external temperature T_{e_i} which occurs for h_i hours per year, the heat load $\dot{Q}_{load\_i}$ could be calculated as follows:

$$\dot{Q}_{load\_i} = H_{eq}(T_{room} - T_{e\_i}) \tag{59}$$

In terms of energy the load L_i is:

$$L_i = \dot{Q}_{load\_i}\, h_i \tag{60}$$

And the annual energy load L_{year} is equal to:

$$L_{year} = \sum_i^N L_i \tag{61}$$

Where N is the number of discrete external temperature considered for the examined location. The annual year gas consumption C_{year} is the sum of N discrete consumptions C_i dependent on η_{g_i} efficiency conditions:

$$C_i = \frac{L_i}{\eta_{g\_i}} \tag{62}$$

$$C_{year} = \sum_i^N C_i \tag{63}$$

Where η_g is the global plant efficiency:

$$\eta_g = \eta_b \cdot \eta_{em} \cdot \eta_d \cdot \eta_c \tag{64}$$

Where η_b is the boiler efficiency and $\eta_{em}\ \eta_d\ \eta_c$ are the emission, distribution and control plant related efficiencies that are calculated according to Italian technical standard (UNI-TS 11300, 2014).

The degree days (DD) of a certain location are defined as follows (ISO 15927-6, 2007).:

$$DD = \sum_{i}^{n} \left(\bar{T}_{e\_i} - T_{room} \right) \qquad (65)$$

Where n is the number of days considered for the heating period, $\bar{T}_{e\_i}$ is the average daily external temperature.
Five Italian cities were considered in the analysis as representative of five different climate zones. In Table 14 cities, DD, the allowable heating system working periods and related climate zones are reported.

City	**Degree days (DD)**	**Italian climate zone**	**Allowed daily heating hours and period**
Palermo	751	B	1december – 31 march 8hours
Salerno	994	C	15 november – 10 March 10 hours
Rome	1415	D	1 november – 15 april 12hours
Bologna	2259	E	15 october – 15 april 14 hours
Belluno	3001	F	Unlimited, considered: 15 october – 15 april 14 hours

Table 14. Degree- days (DD) allowable daily heating period and climatic zone for five Italian cities.

In Figure 26 the frequency distribution of the external temperature filtered by allowable heating period is reported. Colder cities show wider distributions because of the longer allowable heating period.

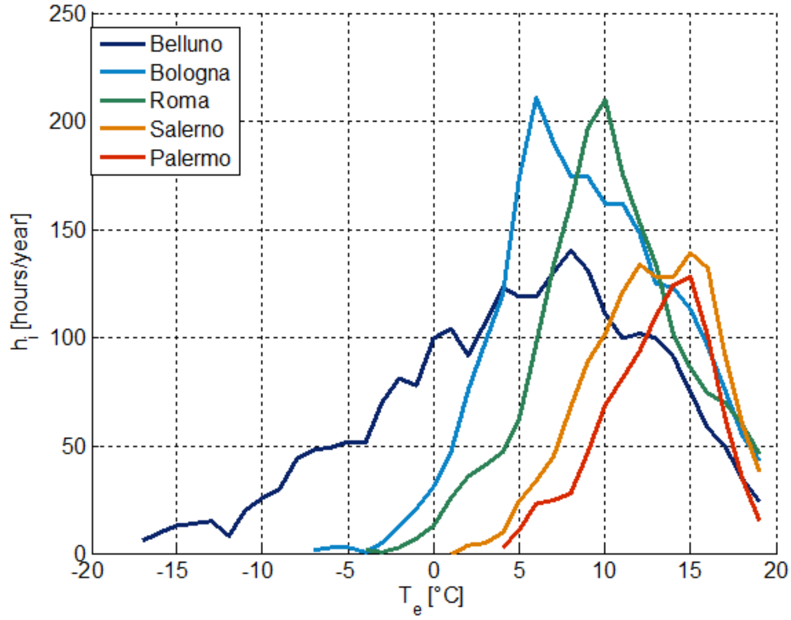


Figure 26. Satistical frequency distribution of external temperatures during considered heating periods for the Italian cities of Belluno, Bologna, Roma, Salerno and Palermo.

5.2.2. Stationary efficiency model identification

According to most of the commercial building energy consumption software, the efficiency was modeled as a polynomial function of heat output and of the return temperature considering a fixed delta temperature on the boiler equal to 20°C:

$$\Delta T_b = T_f - T_r \qquad (66)$$

Efficiency results to be:

$$\eta_s = f(\dot{Q}_{out}, T_r) \tag{67}$$

So that stationary efficiency has been mapped as a function of heat output and return temperature.
So that stationary efficiency is mapped as a function of heat output and return temperature.
Afterwards, boiler manufacturer parameters were forced out of the nominal configuration to let the boiler reach modulation ratios higher than 10, which is the manufacturer setting. When modulating over MR=15 the burner shows flame lifts and a general combustion instability, suggesting the need of a specific gas supply system. This technical limit in currently under investigation. Boiler characterization was extended to ultramodulation range; then, according to eq 67 a multiple linear regression was identified. Regression shows good agreement with experimental data and an error within 3%. Experimental, modeled efficiency and relative errors are shown in Figure 27 for two different water return temperatures.

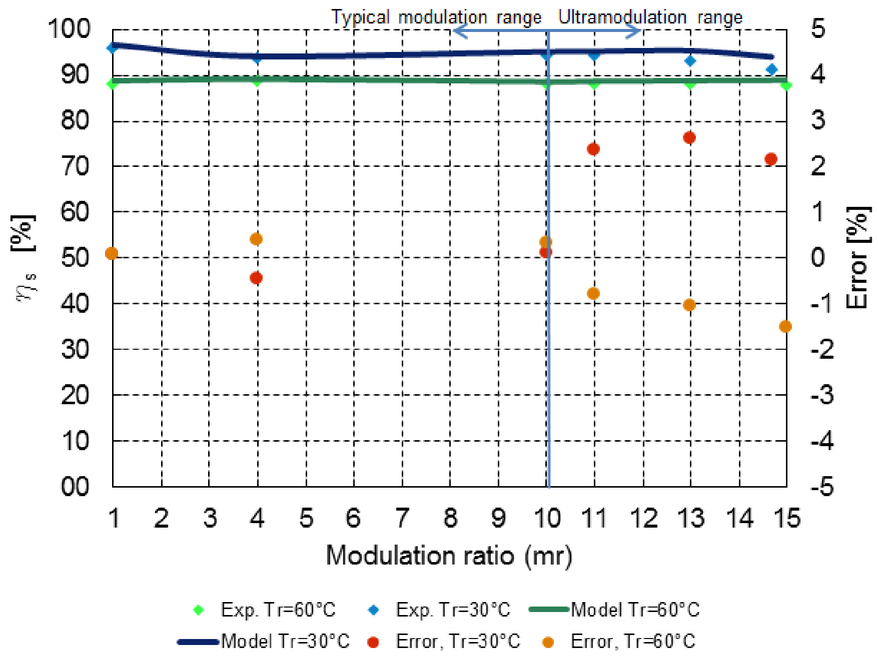


Figure 27. Boiler efficiency experimental, modeled and relative error. Boiler delta water temperature is fixed and equal to 20°C. Data are divided into modulation and ultra-modulation range.

Subsequently, the regression has been used to foresee boiler behavior for MR>15 and up to 40.

5.2.3. Boiler working conditions definition

Real boiler compensation curves were considered as depending on the external temperature. In particular for Te=10°C the compensation curves set Tf =35°C for radiating floor and 38°C for fan coil plants; for both, Tf=20°C when Te=20°C.
Using boiler compensation curve (bcc) the return boiler temperature could be obtained as a function of external temperature:

$$T_r = bcc(T_e) \tag{68}$$

Once definite building equivalent heat losses H_{eq} , load is then a function only of the external temperature and it is equal to the requested heat output:

$$\dot{Q}_{out} = \dot{Q}_{load}(T_e) \quad (69)$$

Consequently, for a certain plant and building, the stationary efficiency η_s is a function of the external temperature only:

$$\eta_s\big|_{H_{eq}} = f(\dot{Q}_{out}, T_r) = f(T_e) \quad (70)$$

5.2.4. Boiler cycling efficiency and ERC

A zero-dimensional energy conservation sub-model has been implemented in order to describe cycling losses and ERC: an equivalent mass m_b, specific heat value c_b and T_b temperature have been considered for the boiler. Cycling behavior has been divided into different phases, starting from thermostat cut-off, the boiler cycle is described as follows:

- After cut-off, the burner stops, while load is still present, the post-purge phase starts. losses are due to jacket losses and orced air flow thought boiler $\dot{Q}_{fa}$, the post-purge duration is fixed by boiler electronic control and it is equal to t_{postp} .

During this phase boiler temperature decreases and is modeled as follows:

$$m_b c_b \frac{dT_b}{dt} = -\left(\dot{Q}_{postp} + \dot{Q}_{load}\right) \quad (71)$$

So that the post-purge heat is:

$$\dot{Q}_{postp} = \left(\dot{Q}_j + \dot{Q}_{fa}\right) \quad (72)$$

Boiler has initially an equivalent OFF temperature T_{b_OFF} , after a period of time t_{postp} temperature is reduced by a quantity ΔT_{postp} and becomes equal to T_{b_postp} :

$$T_{b\_postp} = T_{b\_OFF} - \Delta T_{postp} \tag{73}$$

- Subsequently the post purge period, load, jacket losses and natural air flow $\dot{Q}_{na}$ through the boiler reduce the boiler temperature from T_{b_postp} to an equivalent ON temperature.

This phase has been modeled as follows:

$$m_b c_b \frac{dT_b}{dt} = -\left(\dot{Q}_{nc} + \dot{Q}_{load}\right) \tag{74}$$

Where the natural cooling heat loss is:

$$\dot{Q}_{nc} = \left(\dot{Q}_j + \dot{Q}_{na}\right) \tag{75}$$

The time necessary to reach T_{b_ON} starting from T_{b_postp} is defined t_{nc} .

- Afterwards, having reached T_{b_ON} a pre-purge phase starts, this phase is equivalent to the post-purge one, yet starting boiler temperature is T_{b_ON} .

$$m_b c_b \frac{dT_b}{dt} = -\left(\dot{Q}_{prep} + \dot{Q}_{load}\right) \tag{76}$$

The heat loss in pre-purge phase is:

$$\dot{Q}_{prep} = \left(\dot{Q}_j + \dot{Q}_{fa}\right) \tag{77}$$

The pre purge duration t_{prep} is set by boiler electronics.

- After pre-purge period the boiler is at temperature T_{b_prep} and firing phase can start. No ignition ramp is considered yet the boiler is assumed to reach directly minimum allowable heat output $\dot{Q}_{out\_min}$.

During the firing phase, load, jacket and stack losses are taken into account. Because $\dot{Q}_{out\_min}$ is higher than $\dot{Q}_{load}$, the boiler temperature increases until reaches equivalent OFF temperature T_{b_OFF} and the cycle restarts.

$$m_b c_b \frac{dT_b}{dt} = \left(\dot{Q}_{out\_min} - \dot{Q}_{load}\right) \tag{78}$$

The minimum heat input has been evaluated from the stationary efficiency considering an average boiler temperature for the firing phase:

$$\dot{Q}_{in\_min} = \frac{\dot{Q}_{out\_min}}{\eta_s} \tag{79}$$

The time necessary to reach T_{b_OFF} starting from T_{b_prep} is defined t_{fr} .
The overall boiler efficiency η_b can be then definite considering the energy contributes in the cycle as:

$$\eta_b = \frac{\dot{Q}_{load}\, t_{cycle}}{\dot{Q}_{load}(t_{cycle} - t_{fr}) + \dot{Q}_{postp}\, t_{postp} + \dot{Q}_{nc}\, t_{nc} + \dot{Q}_{prep}\, t_{prep} + \dot{Q}_{in\_min}\, t_{fr}} \tag{40}$$

Where:

$$t_{cycle} = t_{prep} + t_{fr} + t_{postp} + t_{nc} \qquad (41)$$

Is the overall time of the cycle.

ERC trend

Once fixed MR and the building specific heat losses, PLR and ERC became a function of the external temperature only:

$$PLR\big|_{MR,H_{eq}} = \frac{H_{eq}(T_{room} - T_e)}{\frac{\dot{Q}_{out\_nom}}{MR}} = f(T_e) \qquad (80)$$

$$ERC\big|_{MR,Heq} = f(PLR, T_e) = f(T_e) \qquad (81)$$

In Figure 28 ERC is represented as a function of the external temperature parameterized for three different buildings and for MR=1. For H_{eq}= [177, 350, 590] [W/K], (Lazzarin, 2014) which are typical building losses, the ERC shows a slight decrease for lower external temperatures and decreasing sharply for higher ones. The effect of external temperature on ERC is double: it reduces air heat exchange losses although increasing cycling frequency losses. The second effect is predominant in the present calculation and is coherent with experimental data (Landry R. W. et al., 1993).

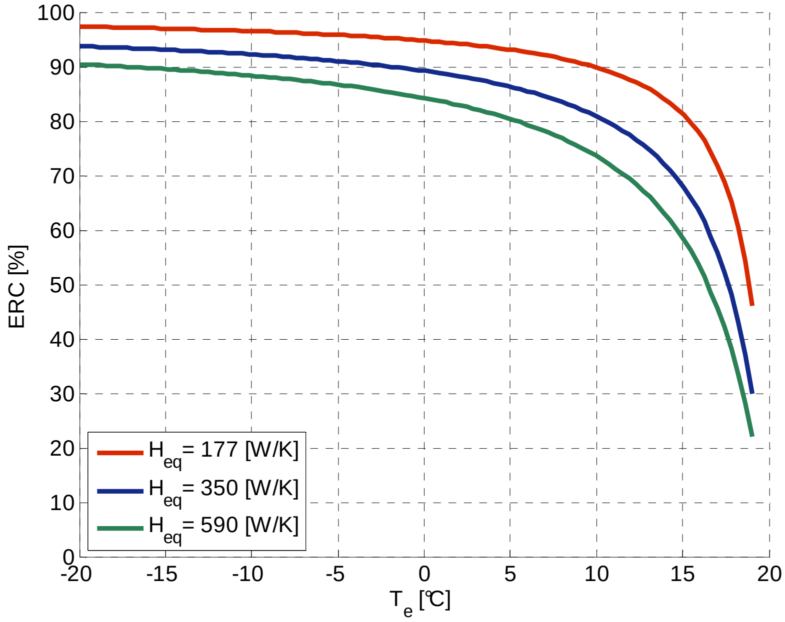


Figure 28. Boiler efficiency reduction from stationary conditions due to cycling losses (ERC) as a function of the external temperature for a fixed boiler and different building specific losses.

5.2.5. Annual electricity consumption calculation

Modeled electrical consumption takes into account boiler pump P_{pump} absorption, valve gas, electronic board and other electrical absorption P_{eb} which are all supposed to be constant with boiler working conditions. Pump power has been considered constant and equal to 40W which corresponds to providing a 16 l/min flow with a 500 mbar prevalence. Electronic board, valve gas and other components power resulted equal to 7 W, while the fan power P_{fan} reduces with increasing mr because modulation is performed reducing fan rpm.

Thus, electrical energy consumption increases with working time of the boiler, while MR, high modulation, on the other hand, reduces fan power absorption. Coefficient of conversion (CC) of the electrical energy into primary energy has been considered equal to 2.5 according to European Directive 2012/27/EU.
The annual electrical primary energy consumption is:

$$E_{p\_year} = CC\left(\left(P_{pump} + P_{eb}\right)t_{b\_year\_ON} + \sum_{i}^{N} P_{fan\_i} t_{b\_ON\_i}\right) \quad (82)$$

Where $t_{b\_year\_ON}$ is the annual working time of the boiler, $t_{b\_ON\_i}$ is the period of time spent by the boiler at each mr_i during the considered year and P_{fan_i} is the related fan power.

Fan electrical power P_{fan} has been measured as a function of RPM within nominal modulation range and in ultramodulation range up to MR=15, a polynomial regression was identified and used to foresee electrical consumption for modulation ratios up to 40.
Experimental measurements, model results and relative errors are reported in Figure 29 for different rpm.

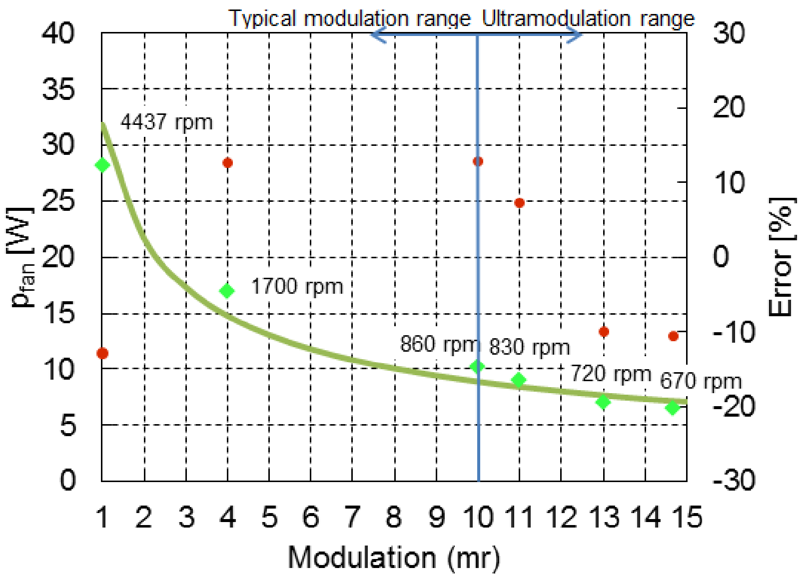


Figure 29. Experimental and modeled fan electrical consumption and relative error for different fan rpm and modulation ratio. Data is divided into modulation and ultra-modulation range.

5.2.6. Ultramodulation consumption analysis

An apartment of 70 square meters built in 1990 was considered, where the ratio between external heat loss surface and internal volume (S/V) is equal to 4.5. The resulting heat loss Heq is 195 [W/K] calculated including ventilation and transmission through the building surface and neglecting solar and free heat contributions. Analysis was carried out in case of heating plant equipped with radiating floor and fan coils.

Referring to Rome and supposing for the characterized boiler the nominal modulation conditions (MR=10), Figure 30 shows as a function of external temperature, the boiler efficiency, annual house loads and the heat provided by the boiler. In particular, in correspondence of an external temperature T_e =9°C, boiler reaches minimum heat output $\dot{Q}_{out\_min}$, for higher temperatures boiler starts cycling and efficiency starts falling, showing a reduction from the steady state.

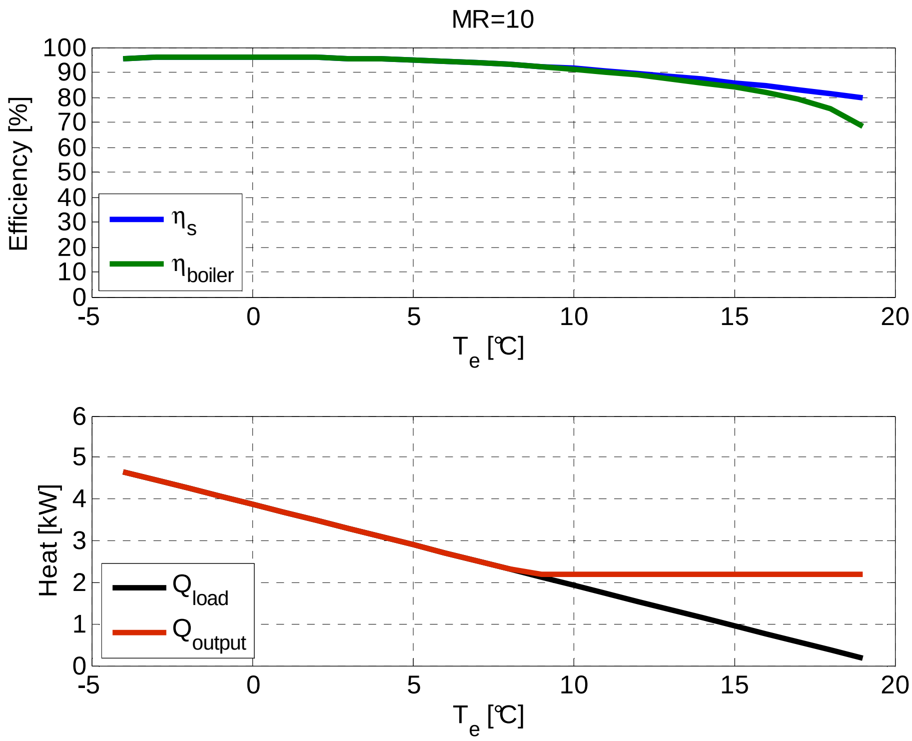


Figure 30. For a boiler with maximum modulation ratio equal to 10: in top figure the efficiency degradation due to cycling losses, in the second one, boiler heat output and loads as a fuction of external temperature.

With regard to Figure 31, the boiler is supposed to reach MR=20, in this case boiler starts cycling at an higher external temperature compared to MR=10, reducing both the cycling frequency and the amount of time in which cycling losses occur.

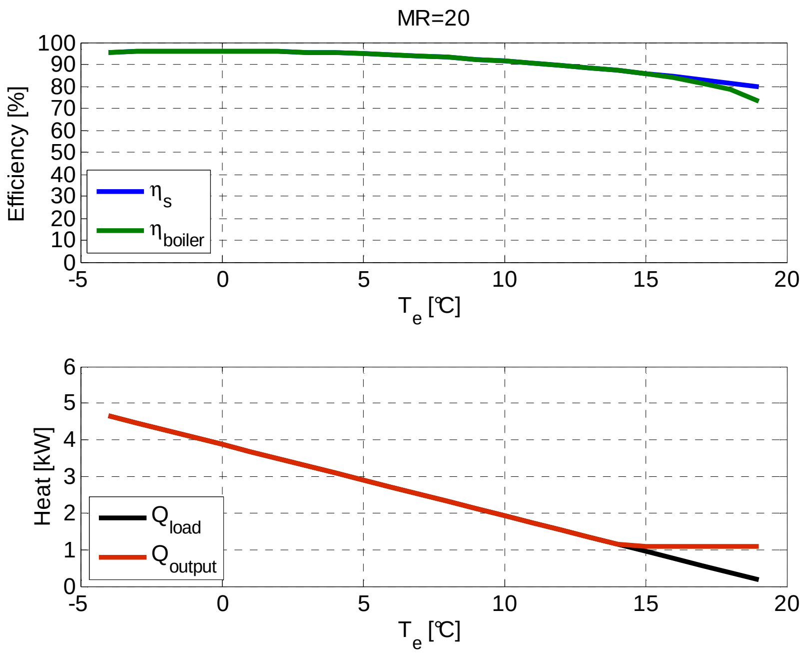


Figure 31. For a boiler with maximum modulation ratio equal to 20: in top figure the efficiency degradation due to cycling losses, in the second one, boiler heat output and loads as a fuction of external temperature

For MR=40, Figure 33, boiler is able to provide the right heat output for almost every external temperature and load request. Indeed, to increase MR from 10 to 40 means for a 22kW boiler a minimum heat output of 0.55 kW; in the considered case the minimum Q_{load} is equal to 0.2 kW at T_e =19°C, a condition that occurs for 46 hours/year during the considered space heating period.

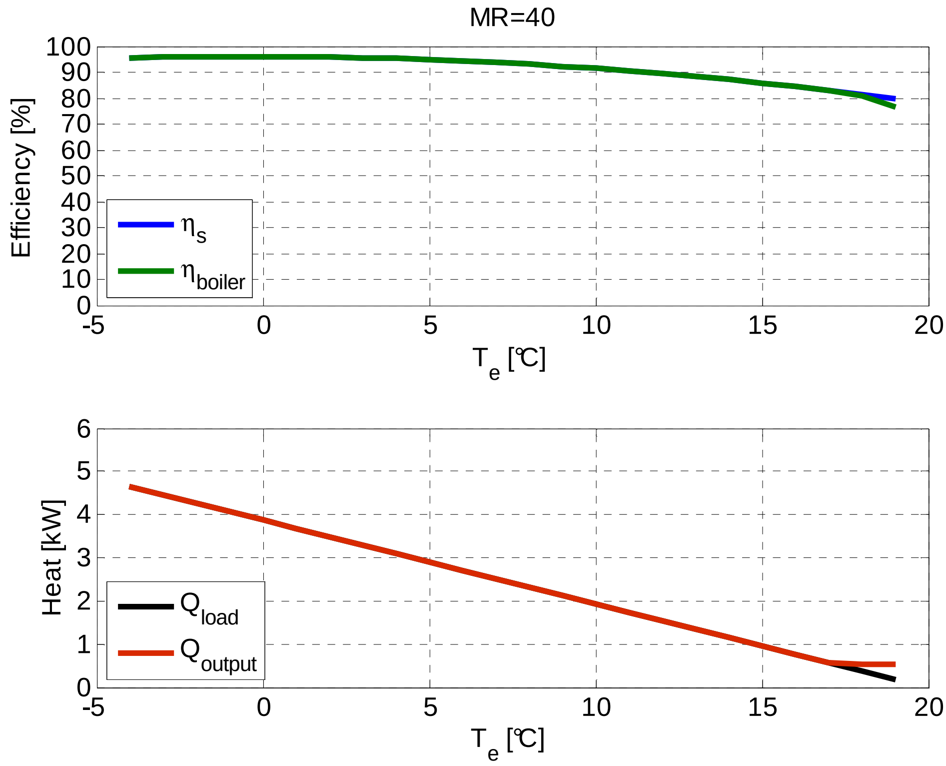


Figure 32. For a boiler with maximum modulation ratio equal to 40: in top figure the efficiency degradation due to cycling losses, in the second one, boiler heat output and loads as a fuction of external temperature.

As a consequence of the higher maximum modulation ratio MR the annual working period of the boiler $t_{b\_year\_ON}$ increases and tends to an "always on" working condition.

To increase MR provides gas savings due to higher boiler efficiency yet, referring to paragraph 1.3, working hours and electrical consumption rise.

5.3 Ultramodulationg boiler modeling results

Annual consumption simulations were performed for different MR, cities and plant type in order to quantify gas savings and electrical consumption increase, consumption are then compared to standard modulation (MR= 10).
Relative differences of consumptions from MR=10 has been evaluated as follows in terms of electrical primary energy δ_{EPE} and total primary energy δ_{PE}.

$$\delta_{EPE} = \frac{E_{p\_year}\big|_{MR} - E_{p\_year}\big|_{MR=10}}{E_{p\_year}\big|_{MR=10}} \tag{83}$$

$$\delta_{PE} = \frac{(E_{p\_year} + C_{year})\big|_{MR} - (E_{p\_year} + C_{year})\big|_{MR=10}}{(E_{p\_year} + C_{year})\big|_{MR=10}} \tag{84}$$

Electrical consumption						
E_{pYear} [kWh]	MR	Belluno	Bologna	Roma	Salerno	Palermo
	10	283	280	210	119	81
	20	304	321	249	161	115
	30	309	330	257	173	123
	40	310	334	260	177	126
δ_{EPE} [%]		+9.7	+19.1	+24.0	+48.3	+55.7
Gas consumption						
Fan coils						
C_{Year} [kWh]	MR	Belluno	Bologna	Roma	Salerno	Palermo
	10	8750	8287	6256	4256	2989
	20	8096	6997	5030	2893	1940
	30	8016	6830	4876	2695	1804
	40	7993	6784	4829	2646	1775
δ_{PE} [%]		-8.1	-16.9	-21.3	-35.5	-38.1
Radiating Floor						
	MR	Belluno	Bologna	Roma	Salerno	Palermo
C_{Year} [kWh]	10	8393	7877	5921	3987	2797
	20	7786	6680	4782	2724	1823
	30	7712	6526	4641	2542	1698
	40	7692	6485	4598	2497	1672
δ_{PE} [%]		-7.8	-16.4	-20.7	-34.9	-37.5

Table 15. Electrical E_{pYear} and gas consumption C_{Year} as a function of the maximum modulation ratio MR, delta consumptions δ_{PE} and δ_{EPE} are reported for radiating floor and fan coils plant.

Results in Table 15 show that with increasing MR the electrical consumption increases significantly in relative terms but does not become comparable to gas consumption in absolute terms. Indeed, considering the overall consumption, significant primary energy savings are possible, ranging from the 7.8% of Belluno to the 38.1% of Palermo. Comparing

radiating floor to fan coil plant, the first shows lower gas consumption because of the better plant efficiency while ultramodulation savings are slightly lower.
Primary energy saving is more significant for milder climates rather than cold ones, since the lower heat loads allow boilers to benefit from higher available modulation ratio.
Actually, should the load be higher than the heat provided in ultramodulation for every condition, boilers simply would work as a typical modulating boiler.

5.4 Electrical-gas consumption trade-off for optimal modulation

Electrical consumption theoretically could became so important to cut out the advantages of ultramodulation. This could happen when electrical consumption becomes comparable to gas consumption, in this case an optimum for the MR could be found such that overcoming this particular MR, primary energy consumption would increase. Considering the previous house scenario and fan coils plant, a parametric analysis on electrical pump power P_{pump} was performed in order to identify optimal MR_{Op} and corresponding critical auxiliary power P_{pump_C} .

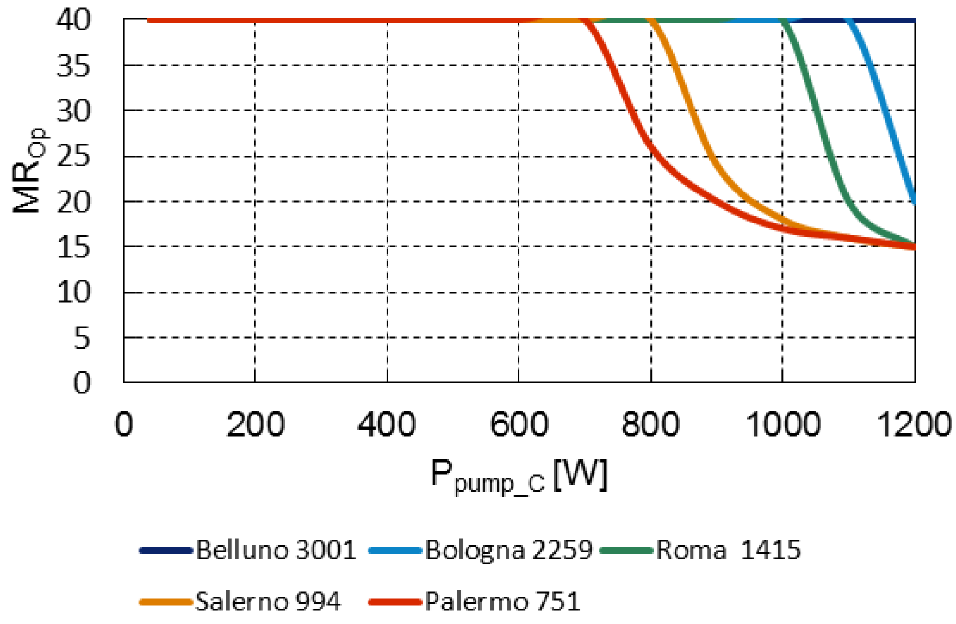


Figure 33. Optimal maximum modulation ratio as a function of electrical power for different Italian cities.

Figure 33 shows that minimum P_{pump_C} necessary to determinate a maximum in MR is about 700W. Furthermore, the value of the critical electrical power P_{pump_C} depends on days degree of the place: in particular lower power is more critical for milder climates, moreover, increasing P_{pump_C}, the MR_{Op} decreases.

The usage of high efficiency pumps for space heating purpose will be mandatory in Europe from September 2015 (Lazzarin, 2014), and could generally reduce pump electrical consumption.

5.5 Savings deriving by electrical resistance adoption

With regard to the analysis reported in Figure 28, for considered external temperatures in which boiler efficiency η_b would drop under 40%, by adopting an electrical resistance of the right power, theoretically would be possible to furthermore reduce overall primary energy consumption.
Indeed, resistance heating efficiency could be write as follows:

$$\eta_{elr\_EPE} = \eta_{elr} \frac{1}{CC} \tag{85}$$

Where η_{el} is the efficiency of the electrical-thermal conversion due to Joule effect, so that η_{elr_EPE} results about 40%. Magnitude of the saving would depend on the house-specific heat loss and on the external temperature distribution and it would be more significant for well insulated houses and milder climates.

5.6 Correlation between savings and day degrees

A relation among degree days, building heat loss and δ_{PE} was identified for different building losses in order to identify trends and roughly estimate ultramodulating savings. The percent reduction δ_{PE} is equivalent to both cost reduction, starting from the primary energy cost, and CO_2 percent reduction. The following relation shows a good correlation with model, in particular the coefficient of correlation R^2 results equal to 0.986.

$$\delta_{PE} = 504 - 0.012DD - 17.9\ln(DD) + 0.060H_{eq} - 67.9\ln(H_{eq}) + (5.1 \cdot 10^{-5})(H_{eq}DD) \quad (86)$$

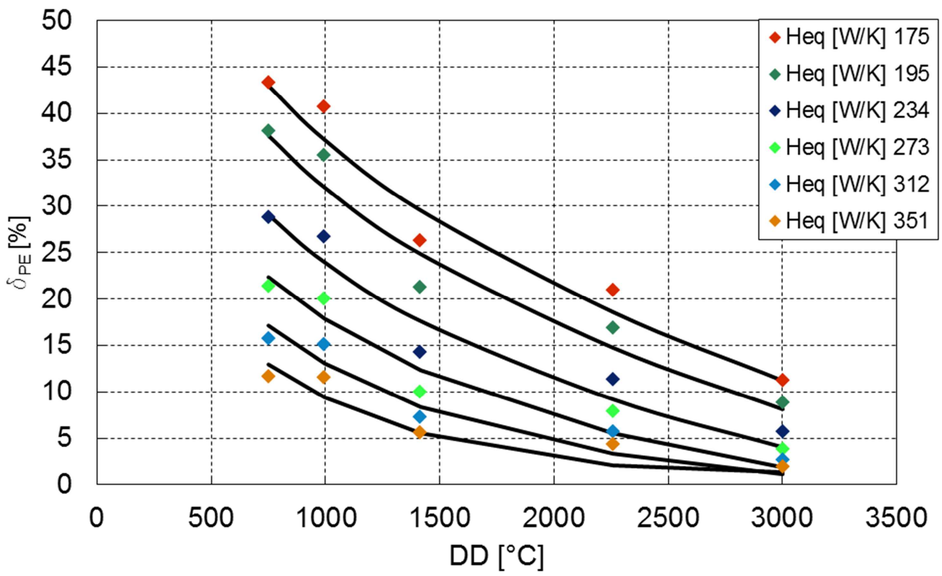


Figure 34. Model results and relation between degree days and total energy saving due to ultra-modulation for different building specific heat losses.

Results in Figure 34 show the relationship among place, building and ultramodulation savings, which could be approximately estimated using the equation 86. Considering Figure 35, it is evident that there an high potential for energy saving within the European territory.

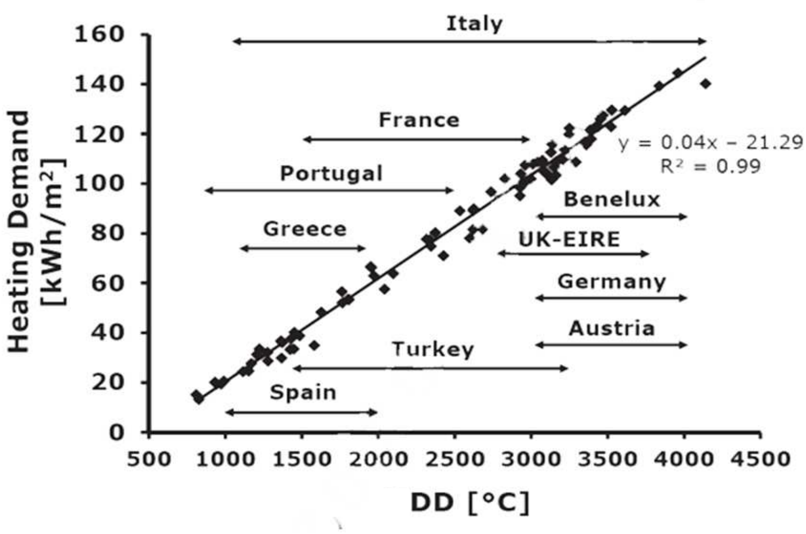


Figure 35. Typical heat demand and correlation with days degree for different European countries. (De Rosa M., 2014)

Conclusions

Boilers efficiency measurement uncertainties and strategies have been analyzed, limits and advantages of direct and indirect efficiency measurements have been pointed out. A statistically-based verification on energy balance closing has been proposed in order to validate efficiency results.
Uncertainty analysis has been extended to space heating efficiency and water heating efficiency according to Erp regulation so that a surveillance risk analysis has been carried out. Three different measurement methods have been proposed and compared to measure radiative and convective heat losses of boilers and tanks. Using such methods, boilers and tank losses have been characterized. An analytical model has been performed in order to foresee storage tank draw-off and energy stored. Furthermore an experimental analysis has been carried out in order to identify maximum efficiency reachable by a standard boiler avoiding condensation, in order to verify Erp compliance showing a borderline result which does not completely exclude the possibility of Erp-compliant non-condensing boilers.
A domestic condensing boiler has been experimentally characterized in terms of efficiency and energy consumption within and over of the nominal working range in terms of burner modulation. An annual consumption model has been developed in order to simulate behavior in the field of such boilers able to modulate from 10 and up to 40.

Simulations show relevant potential for energy savings and environmental impact reduction that are to be more deeply examined. Energy and CO_2 saving potential is considerable and ranges from 8% for colder climates to 38% for milder climates and low emission buildings; savings could be further improved in some cases by adding an electrical resistance. A trade-off analysis shows that with the use of an ultramodulating boiler, electrical consumption increases due to higher operating time, but this has only a minor impact on overall primary energy savings. Furthermore, a relation between ultramodulation savings and the geographic location for different building losses have been identified, in order to approximately estimate CO_2 and money savings.

Further development should consist in performing a complete cost/benefit analysis, to examine in depth the technical aspects of burner ultramodulation and perform long term cycling efficiency measurements both in the field and in laboratory environments. Savings estimation relation could be applied to city or macro area data to quantify macro energy savings.

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