Space and combination heaters

Ecodesign and Energy Labelling

Review Study

Task 4

Technologies

FINAL REPORT


Prepared by

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for the

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Cover: Gas-fired central heating boiler [picture VHK 2016-2017]

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The aim of Task 4 is not only adding a further technical analysis to the already comprehensive Task 4 Report from the previous Preparatory Study (2007), covering relevant technical developments over the last decade, this Task 4 report also investigates whether the heating efficiencies that are achieved in real life sufficiently reflect the assumptions, test methods and calculation models that are used in the current Regulations. For this, various field trial monitoring studies into real-life seasonal efficiencies of fuel boilers and heat pumps were analysed.

**Fuel Boilers**

Various monitoring studies indicate, that without incorporating electricity consumption and standby losses the overall real life seasonal efficiency is somewhere between 83.5 and 90%, where according to the Regulation these values should be around 95-97%. Main reason for this discrepancy is the fact that the return temperatures higher than anticipated and that the boiler capacity is too high for the heat load of the house.

**Heat pumps**

Monitoring studies on heat pumps illustrate that the test- and calculation methods used in the Regulation are adequate, but that the assumptions on the system temperatures on the sink side and related value for the seasonal space heating efficiency in existing dwellings are too optimistic. The actual real-life seasonal average supply temperature is the crucial parameter for achieving the projected seasonal space heating efficiency, and is determined by both the emitter capacity and the pump- and temperature controls on the sink-side. These parameters can constitute the difference between a seasonal efficiency of 85% or 174%. Following the findings for existing dwellings, the average supply temperature over the heating season is at least 5K too low (on average over the heating season it should be 43 °C instead of 38 °C), resulting in an average SCOP that is at least 15% too optimistic.

**System features determining generator efficiency**

The most important system features that determine the real life operating efficiency of a heat generator are:

1. Emitter capacity in relation to the heat load (HL/EC-Ratio)
2. Heating schedule controls
3. System temperature and flow controls

The HL/EC-Ratio is no topic in the existing regulation nor in the related guidelines, despite the fact that this HL/EC-Ratio is crucial for achieving the operating efficiency indicated by the calculation methods of the Regulation. It is therefore proposed to explicitly appoint and discuss this parameter, at least in the regulation related guidelines. Goal is that the HL/EC-Ratio becomes an actual topic of discussion when heat pumps and condensing boilers are installed/replaced.

The system- and field trial analyses also indicate that feed temperature controls have a larger effect on the operating efficiency than the currently assumed default of 3%, especially in combination with heat pumps. Furthermore, flowrate controls are not incorporated in the various control classes, although they have significant influence on
the operating efficiency of the heat generator. Also, heating-schedule controls are currently not incorporated in the control classes mentioned in the Transitional Method for space heaters and combination heaters 813/2013 and 811/2013.

Given the importance of heat emitters as well as temperature and flow controls for the heat generator efficiency, it is recommended to include this subject in the future Ecodesign working programme. For heat pumps these system parameters can mean the difference between and electric SCOP of 2.6 and 4.1, i.e. a space heating energy saving of 50%. For gas boilers, discussed hereafter, the difference is not so high but could still amount to 8-10% between a (nominally ‘condensing’) boiler that is condensing in practice and one that is not.

Regarding the existing Regulation, it is recommended to re-evaluate the default temperature regimes at which the heat generators are tested, and to re-evaluate the impact the controls currently have in the existing Regulations. It is also recommended to reassess the different testing and calculation methods for the various heat generators and to try bring them in line with the bin method that is already implemented for heat pumps.

Hydrogen ready

Carbon-free hydrogen from wind and solar driven electrolysis of water as opposed to not carbon-free hydrogen made from natural gas, is indispensable for a carbon-neutral society in 2050. Hydrogen can be an important chemical storage medium for the electricity generation, to be fed in gas-fired power plants on days with low wind and/or sunshine. But also at building level, fed through the existing gas grid, it can be an ideal complementary space heating fuel for electric heat pump hybrids on renewable energy supply-critical days. It is therefore proposed to stimulate the development and marketing of a natural gas boiler that is ‘hydrogen-ready’ (H₂-ready), meaning that with a few adaptations —e.g. a new nozzle, new setting of the electronics, dip-switches on the pre-mix fan—it will be possible to transform a gas boiler, solo or better in a hybrid, into a carbon-neutral hydrogen boiler. Likewise, it should not be too problematic to do the same for gas-fired fuel cells. Given that there are only 31 years till 2050, it is recommended that ‘H₂-ready’ will be adopted under Ecodesign requirement as soon as possible.

Hybrids

It is generally acknowledged that the technology related to heat pumps and fuel boilers is advanced far enough to allow the design and manufacture of meaningful hybrids, that can deliver the requested heat load and are more efficient (and cost effective) than the alternative of a single generator. In that sense, heat pump/ fuel boiler hybrids may have a bright future in the replacement market. Covering the requested heat load will not be an issue for hybrids, especially when the default (24 kW) combination boiler providing domestic hot water as well. But regarding efficiency and cost effectives, these topics are more complicated to assess and largely depend on:

1. Hybrid specific controls, determining whether the heat pump, the boiler of both are used
2. Capacity heat pump in relation to the heat load of the house
3. Requested system temperatures (determined by HL/EC-ratio)
The recently proposed preliminary standards (chapter 7 prEN14825:2018 and prEN15502-2-Y:2019) for determining the seasonal efficiency of hybrids are already further advanced than the method described in the Regulation and use the bin method to determine the contribution of the various operating modes.

Both these methods define an outdoor temperature where the heat pump is switched off ($T_{hp\text{;off}}$) and an outdoor temperature where the fuel boiler is switched off ($T_{fb\text{;off}}$). Above $T_{fb\text{;off}}$ the heat pump does all the work, below $T_{hp\text{;off}}$ the fuel boiler does all the work and in between both generators do their share. It is assumed that these switch-over temperatures are provided by the manufacturer and must be indicated in the technical documentation. The tests are done using these two outdoor temperatures. In real life however, these switch-over outdoor temperatures will most probably not be used to control the operating modes of the hybrid, simply because the heat load of the house and temperature regime are not known. Higher than theoretically assumed heat loads, will lead to higher switch-over points. Also higher than theoretically assumed temperature regimes (which in practise is often the case, see chapter 2) will lead to higher switch over points and different relative shares in operating modes, leading to varying seasonal efficiencies and to a varying cost effectiveness. It is therefore recommended to further develop these draft test standards and incorporate a test indicating the efficacy of the controls determining the operating mode and to do a test using a small and a large heat load for the house. It is also recommended to present the calculated seasonal efficiency of the hybrid together with the design heat load for which it was calculated.
# Acronyms and Units

## Acronyms

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
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<tbody>
<tr>
<td>ASHP</td>
<td>Air Source Heat Pump</td>
</tr>
<tr>
<td>CH</td>
<td>Central Heating</td>
</tr>
<tr>
<td>DHW</td>
<td>Domestic Hot Water</td>
</tr>
<tr>
<td>EC</td>
<td>European Commission</td>
</tr>
<tr>
<td>ECCP</td>
<td>European Climate Change Programme</td>
</tr>
<tr>
<td>ED</td>
<td>Ecodesign</td>
</tr>
<tr>
<td>EEA</td>
<td>European Environmental Agency</td>
</tr>
<tr>
<td>EIA</td>
<td>Ecodesign Impact Accounting (study)</td>
</tr>
<tr>
<td>EL</td>
<td>Energy Labelling</td>
</tr>
<tr>
<td>ENER</td>
<td>EC, Directorate-General Energy</td>
</tr>
<tr>
<td>EnEV</td>
<td>Energie Einsparungs Verordnung (DE)</td>
</tr>
<tr>
<td>ENTR</td>
<td>EC, Directorate-General Enterprise</td>
</tr>
<tr>
<td>ENTRANZE</td>
<td>Policies to ENforce the TRAnsition to Nearly Zero Energy buildings in the EU-27</td>
</tr>
<tr>
<td>EPG</td>
<td>Energie Prestatie Gebouwen (NL)</td>
</tr>
<tr>
<td>EPISCOPE</td>
<td>Energy Performance Indicator Tracking Schemes for the Continuous Optimisation of Refurbishment Processes in European Housing Stocks</td>
</tr>
<tr>
<td>GCV</td>
<td>Gross Calorific Value (of a fuel)</td>
</tr>
<tr>
<td>GIS</td>
<td>Geographical Information System</td>
</tr>
<tr>
<td>GSHP</td>
<td>Ground Source Heat Pump</td>
</tr>
<tr>
<td>HDD</td>
<td>Heating Degree Days</td>
</tr>
<tr>
<td>HVAC</td>
<td>Heating, Ventilation &amp; Air Conditioning</td>
</tr>
<tr>
<td>IEA</td>
<td>International Energy Agency</td>
</tr>
<tr>
<td>NACE</td>
<td>Statistics classification by Economic Activity</td>
</tr>
<tr>
<td>NCV</td>
<td>Net Calorific Value (of a fuel)</td>
</tr>
<tr>
<td>PBIE</td>
<td>Building Performance Institute Europe</td>
</tr>
<tr>
<td>PFHRD</td>
<td>Passic Flue Heat Recovery Device</td>
</tr>
<tr>
<td>pef</td>
<td>primary energy factor</td>
</tr>
<tr>
<td>RT</td>
<td>Réglementation Thermique (FR)</td>
</tr>
<tr>
<td>SAP</td>
<td>Standard Assessment Procedure (UK)</td>
</tr>
<tr>
<td>SCOP</td>
<td>Seasonal Coefficient Of Performance</td>
</tr>
<tr>
<td>SEER</td>
<td>Seasonal Energy Efficiency Ratio</td>
</tr>
<tr>
<td>TABULA</td>
<td>Typology Approach for Building Stock Energy Assessment</td>
</tr>
<tr>
<td>TRV</td>
<td>Thermostatic Radiator Valve</td>
</tr>
<tr>
<td>UHI</td>
<td>Urban Heat Island</td>
</tr>
<tr>
<td>VHK</td>
<td>Van Holsteijn en Kemna (author)</td>
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</table>

## Parameters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>A</td>
<td>floor surface area building [m²]</td>
</tr>
<tr>
<td>cair</td>
<td>specific heat air [Wh/ m³.K]</td>
</tr>
<tr>
<td>Q</td>
<td>heat/energy [kWh]</td>
</tr>
<tr>
<td>q</td>
<td>hourly air exchange [m³.h⁻¹/ m³]</td>
</tr>
<tr>
<td>rec</td>
<td>ventilation recovery rate [-]</td>
</tr>
<tr>
<td>S</td>
<td>shell surface area building [m²]</td>
</tr>
<tr>
<td>SV</td>
<td>shell surface/volume ratio building</td>
</tr>
<tr>
<td>t</td>
<td>heating season hours [h]</td>
</tr>
<tr>
<td>T_in</td>
<td>Indoor temperature [°C]</td>
</tr>
<tr>
<td>T_out</td>
<td>outdoor temperature [°C]</td>
</tr>
<tr>
<td>U</td>
<td>insulation value in [W/K. m²]</td>
</tr>
<tr>
<td>V</td>
<td>heated building volume [m³]</td>
</tr>
<tr>
<td>ΔT</td>
<td>Indoor-outdoor temperature difference [°C]</td>
</tr>
<tr>
<td>η</td>
<td>(heating boiler) efficiency [-]</td>
</tr>
</tbody>
</table>

## Units

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>€</td>
<td>Euro</td>
</tr>
<tr>
<td>ºC</td>
<td>degree Celsius</td>
</tr>
<tr>
<td>a</td>
<td>annum (year)</td>
</tr>
<tr>
<td>bn</td>
<td>billion (1000 million)</td>
</tr>
<tr>
<td>CO₂</td>
<td>carbon-dioxide (equivalent)</td>
</tr>
<tr>
<td>h</td>
<td>hours</td>
</tr>
<tr>
<td>K</td>
<td>degree Kelvin</td>
</tr>
<tr>
<td>kWh</td>
<td>kilo Watt hour</td>
</tr>
<tr>
<td>Unit</td>
<td>Description</td>
</tr>
<tr>
<td>------</td>
<td>---------------------</td>
</tr>
<tr>
<td>m</td>
<td>metre or million</td>
</tr>
<tr>
<td>m²</td>
<td>square metre</td>
</tr>
<tr>
<td>m³</td>
<td>cubic metre</td>
</tr>
<tr>
<td>W</td>
<td>Watt</td>
</tr>
</tbody>
</table>
Table of Contents

EXECUTIVE SUMMARY ................................................................. III

ACRONYMS AND UNITS ................................................................ VI

1 INTRODUCTION ........................................................................ 1
  1.1 Scope .................................................................................. 1
  1.2 Approach ............................................................................ 1
  1.3 Report Structure .................................................................. 1

2 MONITORING REAL LIFE EFFICIENCY HEATING SYSTEMS ......... 3
  2.1 Introduction ......................................................................... 3
  2.2 Analysing results of German monitoring study “Aktion Brennwertcheck” .................. 4
  2.3 Analysing results UK study “In-situ monitoring efficiencies condensing boiler” ........ 12
  2.4 Conclusions monitoring studies condensing boilers ................................................ 14
  2.5 Observations in relation to existing Regulation ......................................................... 14
  2.6 Analysing the results of German monitoring study “Heat Pump Efficiency, by Fraunhofer” .................................................................................................................. 15
  2.7 Analysing the results of monitoring study “Heat Pump Field Trials, by EST” ........... 21
  2.8 Conclusions monitoring studies heat pumps .......................................................... 24
  2.9 Observations in relation to existing Regulation & EN 14825 .................................. 25

3 PRODUCT- & SYSTEM FEATURES DETERMINING SPACE-HEATING SYSTEM EFFICIENCY ..................................................... 27
  3.1 Introduction ......................................................................... 27
  3.2 System components ............................................................. 27
  3.3 Heating System Efficiency ..................................................... 28
      3.3.1 System temperature heat emitters ....................................... 28
      3.3.2 Type of heat generator ..................................................... 32
      3.3.3 Type of heating schedule control ....................................... 34
      3.3.4 Type of system temperature controls ............................... 37
      3.3.5 Type of flowrate controls ............................................... 40
      3.3.6 Future developments: Optimised flowrate and supply temperature control ....... 42
  3.4 Observations in relation to existing Regulations & standards ................................. 45

4 TECHNICAL IMPROVEMENTS FUEL BOILERS ............................. 46
  4.1 PFHRD .............................................................................. 46
  4.2 Turn-down ratio ................................................................... 48
  4.3 Combustion control ............................................................. 52
  4.4 Hydrogen and boilers .......................................................... 53
  4.5 Pump efficiency and control ................................................ 54
  4.6 Observations in relation to existing Regulations & test standards .......................... 55
5 **TECHNICAL IMPROVEMENTS ELECTRIC HEAT PUMPS** ....................... 56
5.1 Improving operating efficiency .......................................................... 56
  5.1.1 Decreasing the temperature lift ..................................................... 56
  5.1.2 Modifications in vapour compression cycle .................................. 57
  5.1.3 Reducing power consumption system components .......................... 72
  5.1.4 Smart controls .............................................................................. 73
5.2 Reducing costs .................................................................................. 77
  5.2.1 Replacement market ...................................................................... 77
  5.2.2 New built and renovation market .................................................. 79
  5.2.3 ASHP ............................................................................................ 80
  5.2.4 GSHP ............................................................................................ 80
5.3 Observations in relation to existing Regulations & test standards ....... 81

6 **DEVELOPMENTS REGARDING HYBRID HEAT GENERATORS** ........ 83
6.1 Introduction ....................................................................................... 83
6.2 Draft Test Standard for Hybrids following TC 113/WG7 .................... 85
6.3 Draft Test Standard for Hybrids following TC 109/WG1 .................... 87
6.4 Observations in relation to Regulation & draft test standards .......... 94
6.5 Products on the market .................................................................... 96

7 **DEVELOPMENTS REGARDING THERMALLY DRIVEN HEAT PUMPS** 103
7.1 Introduction TDHP ............................................................................ 103
7.2 Gas absorption heat pumps ............................................................... 105
7.3 Gas adsorption heat pumps ............................................................... 107
7.4 Other TDHP Technologies ............................................................... 109

8 **COMMENTS Stakeholders on Task 4 items presented in stakeholder**
**MEETING** 112
8.1 Topic: default HP design temperature to 65 °C ............................... 112
8.2 Topic: Emitters capacity and control, new working item ................. 113
8.3 Topic: Heating controls ...................................................................... 115
8.4 Topic: Boiler test conditions ............................................................... 117
8.5 Topic: Ecodesign minimum efficiency limits ...................................... 118
8.6 Topic: Deployment LT-heat pump and TD-heat pump for DHW .......... 119
8.7 Topic: Monitoring / Smart / Connectivity ......................................... 120

**LIST OF TABLES**

Table 1. Explanation of the terms used in SPF and SEFF ....................... 22
Table 2. Avg. System Efficiency and Seasonal Performance Factor for the various heat pumps and measurement phases. ................................. 23
Table 3. Heat load, system temperatures and average emitter capacity over the years .31
Table 4. COPs for a heating system design temperature of 40 °C and a temperature regime 42/38/20 °C ................................................................. 34
Table 5. COPs for a heating system design temperature of 50°C and a temperature regime 52/48/20°C ..................................................................................................................34
Table 6. COPs for a heating system design temperature of 60°C and a temperature regime 62/58/20°C ..................................................................................................................34
Table 7. Switching boiler on/off with a timer .................................................................35
Table 8. Single thermostat with heating schedule ..........................................................36
Table 9. Multiple thermostats with heating schedule ......................................................36
Table 10. Properties of a group of selected refrigerants .................................................69
Table 11. Options for combined operation Zone 2: Parallel (left) and Serial (right) ......92
Table 12. Hybrid product configurations ......................................................................96

**LIST OF FIGURES**

Figure 1. Graphic representation of the heating system ................................................. 1
Figure 2. Age of the condensing boilers that were investigated ........................................ 4
Figure 3. Condensate quantity in relation to outdoor temperature .................................. 6
Figure 4. Condensate quantity in relation to boilers age ............................................... 6
Figure 5. Total condensate quantity in relation to system return temperature .............. 8
Figure 6. Condensate quantity produced in boiler in relation to system return temperature (red line) ................................................................................................. 9
Figure 7. Wrongly dimensioned and adjusted boiler and heating system ..................... 10
Figure 8. Correctly dimensioned and adjusted boiler and heating system ................... 10
Figure 9. System temperatures on a winter day for one of the regular boilers (Ref.346CFR) in the study ......................................................................................... 13
Figure 10. System temperatures on a winter day for one of the combination boilers (Ref.356MJM) in the study ................................................................. 13
Figure 11. Characterisation of the monitoring projects that have been undertaken by Fraunhofer ........................................................................................................ 16
Figure 12. System boundaries for calculating the Seasonal Performance Factor (SPF) ...17
Figure 13. Average values and ranges of the SPF for the various heat pump types and monitoring projects ................................................................................. 18
Figure 14. Average SPF & system temp. for space heating and DHW of air source heat pumps in existing dwellings ............................................................. 19
Figure 15. Average SPF & system temp. for space heating and DHW of ground source heat pumps in existing dwellings ............................................................. 20
Figure 16. System boundary for SPF$_{H4}$ .................................................................... 22
Figure 17. Calculation methods for System Efficiency ................................................... 22
Figure 18. System Efficiencies (SEFF) for the various sites, measured during phase 1 (orange circles) and phase 2 (blue cubes) ......................................................... 23
Figure 19. Components Space Heating System ............................................................. 28
Figure 20. Heating curves for various HL/EC-Ratios ..................................................... 29
Figure 21. Effect of return temperature on efficiency of condensing boilers ..................32
Figure 22. Relation between average heating COP for air and ground source heat pumps (left and right, respectively) and temperature lift (difference between $T_{\text{source}}$ and $T_{\text{sink}}$), based on data taken from industrial surveys and field trials. ..........................................................33

Figure 23. Example of Automatic Flow Control TRV ..................................................................41

Figure 24. Combination boiler with integrated PFHRD from Intergas named Xtreme, with a water heating efficiency of 110% in GCV, determined according to the Dutch EPN .....47

Figure 25. Distribution heat load English housing stock (in blue) and boiler output (in red) in kW..................................................................................................................................................48

Figure 26. Efficiency degradation due to cycling .......................................................................49

Figure 27. Mean boiler run time per cycle ..................................................................................50

Figure 28. Combustion efficiency in relation to air/fuel-ratio.......................................................51

Figure 29. Correlation between CO$_2$ and dew point ..............................................................51

Figure 30. Elco THISION S PLUS Compact 13 V100 + Gas-Brennwert Heizung 3900012 52

Figure 31. Coefficient of performance and temperature difference .................................56

Figure 32. VI with flash tank cycle (left) and VI with economizing heat exchanger (right) .......................................................................................................................................................58

Figure 33. Two compressor heat pumps in series with VI (left), and in parallel/tandem (right) .......................................................................................................................................................59

Figure 34. Ratios of heating capacity @-25°C/8.3°C (blue), COP at 8.3°C (green) and the COP at -25°C (red) for various vapour compression cycle configurations .................60

Figure 35. HP with tandem single speed compressors + EXV.....................................................61

Figure 36. HP with tandem single speed compressors + VI EcHX .............................................61

Figure 37. Schematics and p-h diagram of a two-stage cascade system .................................62

Figure 38. Schematics and p-h diagrams of separated gas cooler cycles$^{14}$ .........................63

Figure 39. R744/CO$_2$ phase diagram .........................................................................................64
Figure 40. Pressure enthalpy chart R744 ..........................................................65
Figure 41. Heat pump system with various heat exchangers and related temperature shift
for working fluid and water that is heated from 5 °C to 70 °C. ..............................66
Figure 42. Efficiencies and capacity of the aquaeco2 heat pump in relation to outdoor
ambient temperature .................................................................................67
Figure 43. System configuration for the aquaco2 DHW-unit ..............................68
Figure 44. Comparison of heating capacity of a group of refrigerant mixtures as function
of outside air temperature ............................................................................70
Figure 45. Comparison COP of a group of refrigerant mixtures as function of outside air
temperatures ............................................................................................70
Figure 46. Comparison COP of two promising refrigerant mixtures as function of outside
air temperature .........................................................................................71
Figure 47. Comparison of heating capacity of two promising refrigerant mixtures as
function of outside air temperature ..........................................................71
Figure 48. Comparison of heat and electricity variability across a year (domestic and
commercial) -2010 ..................................................................................75
Figure 49. Control strategies for smart grid prepared heat pump systems ..........................76
Figure 50. Hydrotop, Integrated heat pump for sloping roofs (source: Dutch Heat Pump
Solutions) .................................................................................................78
Figure 51. Heat pump pipework ...................................................................78
Figure 52. Integrated and pre-installed add-on HVAC modules, containing ASHP system-
components ............................................................................................79
Figure 53. Calculation method for determining the seasonal space heating efficiency hybrid heat generators (Annex IV, Figure 3) .................................................................83
Figure 54. Operating modes for boiler and heat pump .............................................88
Figure 55. Tables B.1, B.2 and B.3 taken from the prEN15502-2-Y:2019 ................89
Figure 56. Table B.4 from the prEN15502-2-Y:2019 ............................................90
Figure 57. Remeha hybrid heat pump components .................................................96
Figure 58. Remeha hybrid components in place ....................................................97
Figure 59. Daikin Altherma hybrid heat pump ....................................................97
Figure 60. Example Altherma operating modes in avg. dwelling in avg. EU climate .......98
Figure 61. Hybrid Indoor Monobloc and gas boiler by Itho Daalderop (Monobloc indoors) ..................................................................................................................99
Figure 62. Hybrid Indoor Monobloc by Enzavu (www.enzavu.nl) .........................99
Figure 63. Hybrid Outdoor Monobloc AroTHERM by Vaillant ...............................100
Figure 64. Installation diagram hybrid heat pump aroTHERM by Vaillant .............100
Figure 65. Fully integrated hybrid by Sime: Murelle Revolution 30 .................101
Figure 66. Components, dimensions and connections Murelle Revolution ..........101
Figure 67. B1 type hybrid non condensing gas boiler with booster heat pump by Sime 102
Figure 68. Temperature levels of thermally driven heat pumps for heating and cooling 103
Figure 69. Ab/Adsorption heat pump system .....................................................104
Figure 70. Basic gas absorption cycle ...............................................................105
Figure 71. Robur 18 kW gas absorption heat pump ..............................................106
Figure 72. E-Sorp modulating 5 - 18 kW heat pump ...........................................106
Figure 73. Adsorption heat pump device .............................................................108
Figure 74. Working principle Coill adsorption heat pump ....................................108
Figure 75. The boostHEAT heat pump boiler principle ......................................110
Figure 76. Thermo acoustic HP principle ............................................................111
Figure 77. BLUE HEART regenerator ...............................................................111
1 Introduction

1.1 Scope
The scope of Task 4 is the Technical Analyses of the Energy using Product and its System. It therefore not only looks into the various types of heat generators, but also into the interaction of the heat generator with the installation/system in which it operates during the use phase. Probably more than with any other Energy Using Product, the System Analysis is a vital subject in the assessment of real life energy efficiency and the improvement potential.

![Diagram of Heating System]

Figure 1. Graphic representation of the heating system

1.2 Approach
For practical reasons, this Task 4 Report will be drafted as an Addendum to the already comprehensive Task 4 Report of Preparatory Study of 2007. In doing so, duplications are voided, as much as possible. First of all monitoring studies will be analysed to gain insights into the validity of applied technical principles in the existing Regulations.

In a next step, technical improvements on existing technologies regarding the various types of heat generators, room temperature control systems, distribution systems and emitter systems will be discussed. And finally, relevant new technological developments regarding one of the above products and systems will be addressed.

1.3 Report Structure
This Task 4 final report contains 9 chapters. After this introductory chapter, the following chapters are addressed:

- Chapter 2 contains results and analyses of some monitoring studies that have been performed since 2007, revealing real-life efficiencies of heating systems and their sensitivity to generator features and some of the other system components.
Chapter 3 lists and explains the product- and system features that have an important influence on the system efficiency; also an indication is given on the relative importance of these different contributing factors.

The following chapters discuss the technical improvements that have been made since the first Preparatory Study in 2007, covering the following:

- Chapter 4. Technical improvements fuel boilers
- Chapter 5. Technical improvements electric heat pumps
- Chapter 6. Developments regarding hybrid heat generators
2 Monitoring Real Life Efficiency Heating Systems

2.1 Introduction

One of the key objectives of the two Regulations on heat generators is to improve the efficiency of heat generators and with it, the overall efficiency of the heating system, resulting in lower energy consumption for space-heating.

This chapter investigates new studies, in particular regarding fuel boilers and electric heat pumps, that have become available regarding real-life efficiency. Results can thus be compared to what is currently written in the regulations.

As a reminder: In the current regulations for fuel boiler (combi) space heaters the seasonal space heating efficiency $\eta_s$ is a function of the efficiency in on-mode $\eta_{on}$ and the sum of a series of deductions for suboptimal temperature control $F(1)$, electricity use $F(2)$, standby heat loss $F(3)$, pilot flame energy use $F(4)$ and — only for cogeneration heaters — a positive contribution $F(5)$ of the electrical efficiency. In formula $\eta_s = \eta_{on} - \sum F(i)$. The efficiency in on-mode $\eta_{on}$ is calculated from test results for full load efficiency $\eta_1$ at a supply/return temperature regime of 80/60°C and a 30% part load efficiency $\eta_4$ at a supply/return temperature regime of 50/30°C as $\eta_{on} = 0.15 \eta_1 + 0.85 \eta_4$. If the boiler is equipped with a temperature control, which is often the case, a positive contribution ranging between 1% (on-off room thermostat) and 5% (multiple room-sensors) may be added.

As mentioned in Task 2, the declared seasonal efficiency values of many condensing boilers are in the range of 93% (gas) to 91% (oil). The sections 2.2 to 2.4 hereafter will explore whether this is in line with the latest research on real-life efficiency and what are the relevant parameters.

In sections 2.5 to 2.7 the same will be done for electric heat pumps, based on monitoring studies in Germany and the UK.

For electric heat pump (combi) space heaters, the formula for seasonal heating efficiency $\eta_{ch}$ takes into account the electric power generation conversion coefficient $CC$, the seasonal coefficient of performance $SCOP$ and deductions for suboptimal temperature control $F(1)$ and the electricity use of ground water heat pumps $F(2)$ In formula: $\eta_s = (100/CC) \times SCOP - \sum F(i)$. The $SCOP$ is calculated as the ratio of the annual reference heat demand $Q_H$ (in kWh heat/a), calculated from the declared (maximum) heating output $P_{designh}$ in kW and a default number of active mode hours $H_{HE}$, and the annual electricity consumption $Q_{HE}$ (in kWh electricity/a). In formula:

$$SCOP = \frac{Q_H}{Q_{HE}} = \frac{P_{designh} \cdot H_{HE}}{Q_{HE}}$$

The electricity consumption $Q_{HE}$ is derived from the electricity consumption in various modes: on, thermostat-off, standby, crankcase heater and off. The on mode electricity use is given by $Q_0/SCOP_{on}$, where $SCOP_{on}$ is calculated interpolation of the heating capacity $P$ and COP at (a maximum of) 7 test points using the so-called bin method. The bin-method uses an array of outdoor temperature values (bin temperatures $T_j$ for bin $j$) and
the number of hours $h_j$ that each rounded-to-integer outdoor temperature occurs in a year. The part load at each bin is calculated against a reference of the climate-dependent design temperature $T_{\text{design}}$, e.g. $-10 \, ^\circ\text{C}$ in an Average climate, for the maximum heat load and against a reference temperature of $16 \, ^\circ\text{C}$ where it is assumed that also given the solar and internal heat load input—there is no heating demand. In other words, the part load is the ratio $(T_j-16) / (T_{\text{design}}-16)$. In case the heat pump cannot meet the heat demand in a certain bin, i.e. $T_j$ is too low for its declared operation limit temperature $TOL$, a real or calculated back-up heater takes over. By default this is an electric resistance heater with a COP of 1, unless the product actually includes a fossil fuel boiler as a back-up with a fossil fuel efficiency of $\eta_s$, which is then multiplied with the power generation conversion coefficient $CC$ to keep all energy calculations in electricity equivalent.

Once the electricity use in on-mode is determined, the electricity consumption in the other modes is calculated from the measured power consumption (in W) in those modes and a default number of hours for each mode. The default hours are given in the regulation and in the designated standard for this calculation EN14825. Thus the total QHE can be calculated, the SCOP can be assessed and eventually the seasonal heating efficiency of the heat pump $\eta_s$ in primary energy, can be established.

### 2.2 Analysing results of German monitoring study “Aktion Brennwertcheck”

In Germany the Verbraucherzentrale investigated around thousand condensing boilers on their condensing capabilities, in a project carrying the name “Aktion Brennwertcheck”. The project was supported by the Bundesministerium für Wirtschaft und Technologie. 90% of these condensing boilers were smaller than 30 Kw; 88% were natural gas boilers, 9% oil and 3% used liquefied petroleum gas as a fuel. The age of the various condensing boilers is indicated in the graph below.

![Age distribution of condensing boilers assessed](https://www.verbraucherzentrale-energieberatung.de/assets/downloads/studien/Aktion_Brennwertcheck_Langfassung_Juli_2011.pdf)

(Source: Verbraucherzentrale 2011

Figure 2. Age of the condensing boilers that were investigated

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Focus of this monitoring project was the measurement of the actual condensate volume per kWh of boiler load, during heating season. The condensate volume in the secondary heat exchanger is a measure of the latent heat that is recovered. The theoretical maximum in condensate production using natural gas (gas quality H) is **160 g/kWh**.

“**Important to note here is the fact that the condensate that is measured for this study, is not only produced in the secondary heat exchanger of the boiler itself, but also in the chimney. This condensate that is formed upwards in the chimney also flows back into the condensate collector but does not contribute to an increased efficiency of the boiler. The longer the chimney, the more condensate (for collective exhaust chimneys, this only applies to the chimney section from boiler to collective exhaust duct). The study report does not mention this phenomenon.**”

Main conclusion of this monitoring project is:

The full saving potential of condensing boilers is not achieved; only one-third of the measured condensing boilers produces an acceptable condensate volume; condensate production of another third of the boilers can clearly be improved and for the remaining third condensate production is manifestly insufficient.

Several reasons for this deficit are mentioned in the study:

1. Heating curves are not properly set (curves are set too high or too steep, leading to higher supply and return temperatures)
2. Temperature difference between supply and return temperature is too small
3. Heating system is not hydraulically balanced
4. Heating system does not use smart-controlled and highly efficient pumps
5. The use of an overflow valve in the distribution system
6. Night set-back is not applied

Figure 3 gives the amount of condensation production in relation to the outdoor temperature. Figure 4 gives the amount of condensation production in relation to age of the boiler.
Figure 3 indicates that the average total condensate production at 0 °C outdoor temperature is around 74 g/kWh, with a large spread of ± 74. The figure also illustrates that the outdoor temperature barely influences the condensate production of the boiler, which in itself is strange because higher outdoor temperatures would need lower supply temperatures, leading to increased condensate production. This clearly indicates that the heating system as whole—including the controls that determine the supply temperature and including smart pumps that determine return temperature—is not properly designed, hydraulically balanced and adjusted to the buildings specific needs.

(Source: Verbraucherzentrale 2011)


Figure 4. Condensate quantity in relation to boilers age
Figure 4 illustrates that heating systems with an older condensing boilers perform worse than systems with a new boiler. The condensate production is on average around 45 g/kWh for older boilers and around 82 g/kWh for the newest boilers. The study report mentions three possible reasons for these findings:

1. Newer boiler have improved efficiencies
2. Newer installations are better designed
3. Over age, the boiler efficiency deteriorates

Undoubtedly these factors play a (be it limited) role in reducing boilers efficiency. However, a fourth very important reason for improved efficiency figures in newer heating systems is not mentioned in the study report, and stems from the reduced heat-load of the newer dwelling (better insulated and more airtight) while keeping emitter sizes more or less the same. This will result in a heating system in which lower supply temperatures suffice while delivering sufficient heat output by means of heat emitters. A fifth reason may be found in the average length of the chimney, which accidently may differ per boiler age, and will lead to a higher condensate production that unfortunately does not affect boiler’s efficiency.

(Source: Verbraucherzentrale 2011)

Figure 5 presents the amount of condensate in relation to the system return temperature. The fact that the boilers produce condensate above return temperatures of 56°C (average dew point of the flue gas) indicates that this condensate is not produced in the secondary heat exchanger of the boiler during steady-state operation, but somewhere upstream in the exhaust pipe. Another explanation is that condensate is produced during heating up phases, when both boiler and ch-water temperature are cool down. For these reasons the R-squared of the picture below will not be very high.
Figure 5. Total condensate quantity in relation to system return temperature

According to figure 5 the average total amount of condensate Y produced in the 1000 condensing boilers that were tested is:

$$Y = -1,4012 \times T_{\text{return}} + 133,68 \ [\text{g/kWh}]$$

If we are to correct this formula for the amount of condensate that is produced upwards in the chimney, the constant in the formula (133,68) must be reduced with the average condensate amount at dew point (return temperature = 56 °C). Y at 56 °C = 55,2 g/kWh, so the corrected formula should be:

$$Y = -1,4012 \times T_{\text{return}} + 78,48 \ [\text{g/kWh}]$$

The red line in Figure 6 represents the corrected average line and is a parallel shift of the original average black line. The theoretical maximum amount of condensate that can be produced in the average condensing boiler (provided all systems components and controls are optimal) is represented by the dashed green line.

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Another factor worth mentioning, is the fact that the capacity of the majority of boilers that were tested in this study, is much too large for the heat load of the dwelling, leading to very high numbers of boiler starts and with it to increased emissions and accelerated wear of the appliance.

Figure 7 illustrates the effect of wrongly sized condensing boilers and a suboptimal system design versus (Source: Verbraucherzentrale 2011). The figure shows high supply and return temperatures (average well above 50ºC) with very little delta T (supply and return temperature very close to each other) which indicates a high flow pump setting. The high return temperatures prohibit condensing operation. Furthermore on/off switching is very frequent.

Figure 8 represents a correctly sized boiler in combination with a good system design, an hydraulically balanced emitter system, and right controls for pump and room temperature (Source: Verbraucherzentrale 2011). The estimated related thermal generator efficiency for the day illustrated in (Source: Verbraucherzentrale 2011)

Figure 7 is 86% and for the day illustrated in (Source: Verbraucherzentrale 2011)

Figure 8 the estimated efficiency is 97%.
Figure 7. Wrongly dimensioned and adjusted boiler and heating system

Figure 8. Correctly dimensioned and adjusted boiler and heating system


The study report further mentions that condensing capabilities do not suffer from this oversizing of the boiler, and that condensate production seems to increase with an oversized boiler.

Though this may be the case, the process of heating up the CH-supply water does not benefit from this increased condensate production. Between subsequent boiler starts the appliance and its chimney cools down, and the flue gasses from the new boiler start will cool further and generate more condensate because of that. This condensation heat however, is used to reheat the appliance and its chimney to a steady-state situation, meaning that the CH-supply water does not benefit from this condensation.

Furthermore, the average efficiency during a start-up cycles (taking around 45 seconds) is obviously lower than during a steady-state situation. The shorter the operation time per cycle, the larger its impact on the average thermal efficiency during this cycle.

So, contrary to what is concluded in the Final Report of the German monitoring study, an oversized boiler will negatively influence the average thermal efficiency of the heat generator. In addition, the emissions will increase due to incomplete combustion during start-up cycles.

**Summary ‘Aktion Brennwertcheck’**

**Condensate production**

The average return temperature of all 1000 heating systems measured in the study, is around 41 °C. Assuming that this represents the average return temperature during periods the boiler is switched on *(if this is not the case and the average return temperature is calculated by averaging the instantaneous return temperatures during both on- and of periods, the average return temperature during on-mode will be higher and as a result the condensate production will be lower!)*, the theoretical maximum amount of condensation that on average can be produced in the boiler, can be calculated with the formula for the green dashed line in Figure 6:

\[
Y = -4.44 \times T_{\text{return}} + 248 \quad [\text{g/kWh}]
\]

At 41 °C, the average amount of condensate that is produced in the boiler itself over all of the sample (following the green dashed line), is around **65 g/kWh**.

When real life data for condensate production is used and this data is corrected for condensate that is produced in the chimney, the formula for the red-line can be used (parallel shift of the black-line in Figure 6, correcting for condensate in the chimney):

\[
Y = -1,4012 \times T_{\text{return}} + 78,48 \quad [\text{g/kWh}]
\]

The average amount of condensate that is produced in the boiler itself over the whole sample, is now around **21 g/kWh**.

**Conversion of condensate production to thermal efficiency**

These condensate production figures can be converted to annual average thermal efficiencies.

Assuming that the condensate production is linear to the system return temperature, and assuming the best achievable efficiency in non-condensing mode is 89% and 98% at full condensing capacity (160 g/kWh at return temperatures of 20° C), the average 21
g/kWh of condensate produced in the boiler corresponds to an annual thermal efficiency figure for the boiler of 90.2% on GCV.

Following the same calculation method, the theoretical maximum of 65 g/kWh would correspond to an annual thermal efficiency of 92.5% on GCV.

The real-life average annual thermal efficiency of the 1000 boilers in this monitoring study will be at best somewhere in between these two values, but most probably close to the 90% on GCV.

2.3 Analysing results UK study “In-situ monitoring efficiencies condensing boiler”

In the UK, the Energy Saving Trust commissioned a project for the in-situ monitoring of efficiencies of condensing boilers\(^4\). The report reviews the first complete year’s data from field trials of 60 condensing boilers. The main conclusions are based on the results from 43 boilers for which a full 12 month data set has been obtained.

**Monitoring system**

The monitoring equipment installed in the house consisted of a gas meter measuring the gas supply to the (combination) boiler, two heat meters (one for supply to CH, one for supply to domestic hot water (DHW)), an electricity meter measuring the power consumption of the boiler, and a set of temperature transmitters for the room temperatures in the various spaces and for the supply- and return system temperature, gas- and flue temperatures. Data were collected on a 5 minute basis from all data-transmitters.

**Results**

The average annual thermal efficiency (all heat out/gas in) of the condensing boilers producing hot CH-water for both space heating and a DHW -storage vessel (not including the losses of the heat distribution system and buffer tank), was 85.3% on GCV, with a standard deviation of 2.5% and maximum and minimum values of respectively 89.2% and 81.2%.

The average annual thermal efficiency of individual combination boilers was 82.5% on GCV with a standard deviation of 4.0% and maximum and minimum values of respectively 89.7% and 68.8%.

These results indicate that the condensing capacity of practically all of these boilers is not used!

Looking at some of the temperature graphs presented in the Final Report of the monitoring study, it becomes clear that system temperatures during space-heating function are too high to facilitate condensation.

\(^4\) In-situ monitoring of efficiencies of condensing boilers and use of secondary heating, Final Report, June 2009, prepared by Gastec at CRE Ltd., AECOM and EA Technology, Commissioned by The Energy Saving Trust
At an average return temperature of around 63 °C the flue gasses in the boiler will not condense. Figure 9 further illustrates that there is not much difference between the system temperatures for heating and dhw-function, indicating that the capacity of the emitter system is too low.

Also in Figure 10 the system temperatures are too high. At average return temperatures of around 52 °C the flue gasses in the boiler will just start to condense.
2.4 Conclusions monitoring studies condensing boilers

Both monitoring studies clearly indicate that the condensing potential of the generators is not yet fully exploited. With average thermal system efficiencies of around 90% in Germany and around 83.5% (average of condensing regular- & combination boilers) in the UK, the largest part of the condensing potential is waiting to be harvested. For all of Europe it is estimated that average thermal generator efficiency is in between these two figures (with UK installing relatively smaller radiators and Germany using relatively better system designs), being around 87%.

This observation however does not mean that the average thermal generator efficiency of the installed base is not improved. By replacing non-condensing boilers with condensing boilers the average appliance thermal efficiency increases with on average 10%, even without exploiting the full condensation capabilities.

By additional improvements in the system design, a large portion of the condensing capacity can still be exploited in existing dwellings. In new buildings it is essential that the total heating system design will be spot on from the beginning, meaning:

1. Increasing the emitter-capacity to heat-load ratio
2. Improving boiler and room temperature controls

See also Chapter 0 for further explanation.

2.5 Observations in relation to existing Regulation

The transitional method calculates the seasonal space heating efficiency in active mode \( \eta_{\text{son}} = 0.15 \times \eta_1 + 0.85 \times \eta_4 = 96.5\% \). \( \eta_1 \) is measured at a return temperature of 60°C and \( \eta_4 \) at a return temperature of 30°C, leading to an assumed average seasonal return temperature of 34.5°C.

Monitoring studies indicate that these values are not achieved in real-life and that the averaged real-life seasonal return temperatures are above 41°C (with Germany around 41°C and UK above 41°C).

The regulation defines the overall seasonal space heating efficiency \( \eta_s = \eta_{\text{son}} - F(1) - F(2) - F(3) - F(4) \). With \( F(1) \) for controls = 3%, \( F(2) \) for electricity consumption (including pump) = 1.5%, \( F(3) \) for standby heat loss = 0.75% and \( F(4) \) for pilot light = 0%, the average \( \eta_s \) for a condensing boiler = 0.15 * 88% + 0.85 * 98% - 3% - 1.5% - 0.75% - 0% = 91.25%.

Because the monitoring studies only measured the thermal efficiency (heat out/gas in) during real-life operation including controls, we can compare the results when \( F(2), F(3) \) and \( F(4) \) are left out of the equation. And since most of the monitored German installations were using weather compensated controls with modulating boilers, the default correction of -3% for controls can (according to the Regulation) be adjusted with +2%, resulting in a total correction for controls of -1%. The efficiency according to the Regulation is now:

\[ \eta_s = \eta_{\text{son}} - F(1) = 0.15 \times 88% + 0.85 \times 98% - 1% = 95.5% \]

The various monitoring studies indicate however, that without incorporating electricity consumption and standby losses the overall seasonal efficiency is somewhere between 83.5 and 90%.

Following the monitoring studies, the two main reasons for this discrepancy are:

1. Heating capacity of the emitter system is too low
2. Pump and temperature controls are sub-optimal
3. Boiler capacity is too big for the heat-load of the house
These observations illustrate that the test- and calculation methods used in the Regulation are overrating the share of the time that the boiler is operating with return temperatures of 30 ºC. The assumption on the seasonal average return temperature (34.5 ºC) is clearly too optimistic.

The actual real-life seasonal average return temperature is the crucial parameters for achieving the projected seasonal space heating efficiency, and is determined by both the emitter capacity and the pump- and temperature controls. These parameters constitute the difference between a seasonal efficiency of 85% or 97%. The crucial influence of these parameters is not adequately highlighted and explained in the existing regulation and related guidelines.

The emitter system is out of the scope of this study, but it can be recommended to add this subject, including the relevant hydraulic and temperature controls for the emitters, to the future Ecodesign working programme. Also in a review of circulator regulation and the new subject of building controls the importance of the system (return) temperature should be scrutinised. Apart from legal measures at EU level, the Member States can of course expand on EPB-measures, installer training and promotion/subsidies to deal with the subject.

Within the scope of this study, the role of temperature controls is underestimated and —in order to give the a realistic insight— and should have more impact e.g. in the factor F(1).

The fact that the boiler is oversized with at least a factor 2.5 is not new and has also been discussed extensively in the 2007 Ecodesign preparatory study. But there has been no improvement and thus it might be timely to adjust the Ecodesign measures, e.g. by introducing a new F(x) factor or to test/calculate efficiency at 40/20/10/5% of the current nominal capacity and/or possibly to be in sync with the heat pump capacities and system temperatures.

2.6 Analysing the results of German monitoring study “Heat Pump Efficiency, by Fraunhofer”

Introduction
An important monitoring study, measuring the real-life performance of Heat Pumps in single family dwellings in Germany is done by Fraunhofer. Since 2006, Fraunhofer ISE from Freiburg Germany, has been investigating electric heat pumps. Within three projects, nearly 250 air-to-water and brine-to-water heat pump systems have been investigated under real operating conditions in houses with various energetic standards. Projects are supported by the German Federal Ministry for Economic Affairs and Energy.

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6 Miara, M. et al, ”Ten years of heat pump monitoring in Germany; Outcomes of several monitoring campaigns. From low-energy houses to un-retrofitted single-family dwellings”, 12th IEA Heat Pump Conference 2017, Fraunhofer Institute, Germany.
WP im Bestand
This project focuses on heat pumps that were installed as replacement for oil boilers in existing buildings. Focus here is –amongst others– investigating efficiencies at higher system temperatures for the heating system.

WP-Effizienz
This project mainly focuses on new dwellings with smaller heat loads and related lower system temperatures. Results of this study can be a good representation of the saving potential of heat pumps in their main application.

WP Monitor (PLUS)
WP Monitor (Plus) is a monitoring project looking more detailed into the differences in efficiencies and their possible causes. Effects of innovative heat pump technologies on efficiencies are also investigated here.

WPsmart im Bestand
This project is still running; its main goal is the identification of the optimal heat pump and system design for space heating and domestic hot water in existing dwellings (incl. hybrid heat pumps). Second goal is the identification of the load shifting potential of electric heat pumps (and their boundary conditions) in the field.

Monitoring system
The consumption of all electric components of the heat pump systems are being measured, amongst which the compressor, the controls and the electric back-up heating element, but also the electric components on the source-side (brine pump, fan or well pump). The electric components on the sink-side (controls and circulator(s) are not included in the SPF-calculations. In case of hybrid solutions also the gas- or oil consumption is measured. Apart from the electric power consumption, all energy-flows, volume flows and temperatures are monitored. The seasonal performance factor (SPF) is determined according to the diagram in Figure 12.

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Figure 12. System boundaries for calculating the Seasonal Performance Factor (SPF)

Results first three monitoring projects
Figure 13 gives a summary of the average SPF-values for the first three projects, as well as the ranges of the individual results, grouped for different heat sources and projects. The comparison takes into account both outside air heat pumps and ground source heat pumps. Ground water heat pumps were omitted due to the little number of examined installations.

Air source heat pumps in existing buildings achieve an average SPF of 2.6, with individual values ranging from 2.1 to 3.3. In new buildings the air source heat pumps achieve slightly higher SPF-values with an average of 2.8, and with a range of 2.3 to 3.4. Newer heat pumps (> 2012) achieve slightly higher SPF in new buildings with an average SPF of 3.2 and with a range of 2.2 to 4.2).

For ground source heat pumps both the average SPF-values and the ranges are higher, mainly due to the fact that source temperatures are higher in the colder periods. in existing buildings they achieve an average SPF of 3.3, with individual values ranging from 2.2 to 4.3. In new buildings the ground source heat pumps achieve higher SPF-values with an average of 3.9, and with a range of 3.1 to 5.1. Newer ground source heat pumps (> 2012) achieve even higher SPF in new buildings with an average SPF of 4.3 and with a range of 3.0 to 5.4).
Intermediate results “WPsmart im Bestand”

Figure 14 shows the averaged data of 19 air source heat pump systems that were monitored in the period from July 2017 until June 2018. The grey-blue bars indicate the Seasonal Performance Factor (SPF); on the left of this bar, the red and intense-blue bars indicate the ratio between heating energy used for space-heating (red bar) and for DHW (blue bar). The grey bar on the left indicates the ratio between electricity used for the compressor (grey) and for the electric heating element (black).

The red numbers represent the averages heating system temperature \((T_{\text{supply}} + T_{\text{return}})/2\); the brown numbers the highest day-average supply temperature. The blue dots represent the averages system temperature \((T_{\text{supply}} + T_{\text{return}})/2\) for the DHW-function.
Figure 14. Average SPF & system temperatures for space heating and DHW of air source heat pumps in existing dwellings.

Legend: Seasonal Performance Factor (SPF); share DHW-heating; share space-heating; electricity used for the electric heating element,....

The color circles in the grey-blue bars give further information regarding the heat pump system:
- Grey circle: hybrid heat pump (either with gas or oil)
- Brown circle: heat input from thermal solar
- Black circle: heat emitter system with only radiators
- White circle: heat emitter system with only floor heating
- No circle: heat emitter system uses both floor heating and radiators

The average values of all heat pump systems is illustrated on the right hand side of the graph, stating an average SPF of 3.1 (with a range of 2.6 to 4.1), a mean system temperature of 36 °C ((Ts+Tr)/2), highest supply temperature for space-heating of 43 °C, and for DHW of 47 °C.

Figure 15 shows the averaged data of 11 ground source heat pump systems that were monitored in the period from July 2017 until June 2018. The green bars indicate the Seasonal Performance Factor (SPF); on the left of this bar, the red and intense-blue bars indicate the ratio between heating energy used for space-heating (red bar) and for DHW (blue bar). The grey bar on the left indicates the ratio between electricity used for the compressor (grey) and for the electric heating element (black).

The red numbers represent the averages heating system temperature \( ((T_s+T_r)/2) \); the brown numbers the highest day-average supply temperature. The blue dots represent the averages system temperature \( ((T_s+T_r)/2) \) for the DHW-function. The black dots (with numbers) represent the average source temperature \( ((T_s+T_r)/2) \). The yellow circle in this graph represents heat input from a thermal solar system.

### Figure 15. Average SPF & system temp. for space heating and DHW of ground source heat pumps in existing dwellings

The average values of all heat pump systems (illustrated on the right hand side of the graph), are 3.7 for the SFP (with a range of 2.8 to 4.6), a mean system temperature of 38 °C \( ((T_s+T_r)/2) \), highest supply temperature for space-heating of 46 °C, and for DHW of 48 °C. The average system temperature on the source-side is 4.2 °C.

As to be expected, the differences in the average SPF values are mainly caused by:

1. Temperature levels of the heat source (the higher the better);
2. Temperature levels of the heat sink (the lower the better);
3. State of technology of the heat pump (improvement in heat pump efficiency resulting from technology development in recent years).

Because DHW-production requires higher supply temperatures, the SPF decreases with higher shares of DHW-heating energy in the total heating energy demand of the dwelling.

For heat pumps systems, the supply temperature on the sink-side is more important than the average system temperature \( ((T_s+T_r)/2) \) on sink-side. This means that – contrary to systems with condensing boilers where the return temperature is more important - heat pump systems benefit from small differences between supply- and return temperature. In other words, heat-emitter systems that can deliver the requested thermal energy at low supply temperatures will achieve the highest SPF.

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**Example 1:**
Dwelling 1 of the group with *air source heat pumps* has an average system temperature of 39 °C and a maximum supply temperature of 54 °C; the average SPF is 2.6. Dwelling 19 of this group has an average system temperature of 30 °C and a maximum supply temperature of 34 °C; the average SPF is 4.1. With a PEF = 2.5, the corresponding seasonal heating efficiencies are 104% for dwelling 1 and 164% for dwelling 19.

**Example 2:**
Dwelling 1 of the group with *ground source heat pumps* has an average system temperature of 54 °C and a maximum supply temperature of 64 °C; the average SPF is 1.8. Dwelling 11 of this group has an average system temperature of 31 °C and a maximum supply temperature of 38 °C; the average SPF is 4.6. With a PEF = 2.5, the corresponding seasonal heating efficiencies are 72% for dwelling 1 and 184% for dwelling 11.

On the source side, the supply- and return temperatures are equally important. For air source heat pumps the source temperature is defined by the outdoor temperature. For ground source heat pumps the type of ground source heat exchanger may influence the temperature regime on the source side.

Noteworthy is also the fact that – apart from two or three heat pump systems – the electrical heating element in all electric heat pump systems is used on a limited scale only (mainly for the weekly anti-legionellae program), meaning that the overall system (including the output of the heat pump) is well designed in the majority of the dwellings.

### 2.7 Analysing the results of monitoring study “Heat Pump Field Trials, by EST”

This field trial on heat pumps was developed by the Energy Saving Trust UK (EST) in 2008 and covers two monitoring phases. The results from Phase 1, which included 83 sites using either air source or ground source heat pumps, was completed and reported in 2010. Wide variations in performance were found in this phase and as a result, the (MCS) installation guidelines have been updated.

Phase 2, undertaken from 2010-2013, includes a comprehensive study of 44 heat pumps and investigates and tries to improve upon variations in performance shown in Phase 1.

**Calculation methods**

The EST-monitoring study utilizes various SPF calculation methods. The method SPF\(_{H4}\) comes close to the method that is used by the Fraunhofer study (see section 2.4), the main difference is the fact that SPF\(_{H4}\) also includes the power consumption of the electric components on the source side (circulators etc.). Heat losses of the heat distribution system and storage vessels are not included.
Next to this SPF_H4 method, the EST-monitoring study also calculates the System Efficiency (SEFF), defined as:

\[ SEFF = \frac{Q_{H, hp} + Q_{H, aux} + Q_{W, OUT}}{E_{S, fan/pump} + E_{hp} + E_{Aux} + E_{immersion} + E_{bt, pump} + E_{fan/pump}} \]

Table 1. Explanation of the terms used in SPF and SEFF

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Explanation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat</td>
<td></td>
</tr>
<tr>
<td>( Q_{H, hp} )</td>
<td>Space heating provided by the heat pump</td>
</tr>
<tr>
<td>( Q_{H, aux} )</td>
<td>Water heating provided by the heat pump to the domestic hot water cylinder</td>
</tr>
<tr>
<td>( Q_{W, aux} )</td>
<td>Water heating, provided by the auxiliary electric heater</td>
</tr>
<tr>
<td>( Q_{W, aux} )</td>
<td>Water heating, provided by the electric immersion to the domestic hot water cylinder (=E_{immersion})</td>
</tr>
<tr>
<td>Electricity</td>
<td></td>
</tr>
<tr>
<td>( E_{S, fan/pump} )</td>
<td>Electricity used by the source pump (for ground-source) or fan (for air-source)</td>
</tr>
<tr>
<td>( E_{HP} )</td>
<td>Electricity used by the heat pump (excluding the ground loop/air inlet and auxiliary heating/immersion)</td>
</tr>
<tr>
<td>( E_{immersion} )</td>
<td>Electricity used to supplement domestic hot water production</td>
</tr>
<tr>
<td>( E_{Aux} )</td>
<td>Electricity used to supplement space heating</td>
</tr>
<tr>
<td>( E_{bt, pump} )</td>
<td>Electricity used by the buffer tank pump (if present)</td>
</tr>
<tr>
<td>( E_{fan/pump} )</td>
<td>Electricity used by the fan or pump of the central heating system</td>
</tr>
</tbody>
</table>
Results.

Figure 18 represents the temperature-corrected system efficiency of the various sites measured during phase 1 (orange circles) and during phase 2 (blue cubes).

The bars on the bottom of the graph indicate the difference between the two measurement phases (after phases 1 several sites were modified due to disappointing measurement results in phase 1).

![Figure 18. System Efficiencies (SEFF) for the various sites, measured during phase 1 (orange circles) and phase 2 (blue cubes)](https://assets.publishing.service.gov.uk/government/uploads/system/uploads/attachment_data/file/225825/analysis_data_second_phase_est_heat_pump_field_trials.pdf)

Averaging these results of individual sites the following picture emerges (see Table 2).

<table>
<thead>
<tr>
<th></th>
<th>Phase 1</th>
<th>Phase 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air source heat pumps</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Average SEFF</td>
<td>1.83</td>
<td>2.16</td>
</tr>
<tr>
<td>Range</td>
<td>1.2 – 2.2</td>
<td>1.7 – 2.7</td>
</tr>
<tr>
<td>Number</td>
<td>22</td>
<td>16</td>
</tr>
<tr>
<td>Ground source heat pumps</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Average SEFF</td>
<td>2.31</td>
<td>2.54</td>
</tr>
<tr>
<td>Range</td>
<td>1.55 – 3.47</td>
<td>1.5 – 3.3</td>
</tr>
<tr>
<td>Number</td>
<td>49</td>
<td>26</td>
</tr>
</tbody>
</table>

The air source heat pumps in Phase 1 have an average System Efficiency of 1.83 and in phase 2 the SEFF is 2.16.

The ground source heat pumps achieve better results with an average SEFF of 2.31 for phase 1 and 2.54 for phase 2. Assuming that the phase 2 values are representative for the UK, they tend to be lower than the German values measured by Fraunhofer (after correcting for power consumption circulators).
2.8 Conclusions monitoring studies heat pumps.

Both monitoring studies determine the combined SPF of heat pumps for the space-heating and DHW-function. Based on these studies it can be concluded that, if heat pumps are to be implemented in a relevant and meaningful manner, a good system design is a prerequisite. System designers and installers therefore play a vital role in this context, because to a large extent, they determine whether the seasonal system efficiency for space-heating and DHW will be around 1.80 (70%) or around 4.5 (180%).

The key factors that determining the seasonal efficiency (in order of priority) are:

1. Correct sizing of the heat pump output in relation to the heat load of the house, thus minimizing the use of the (electrical) back-up heater.
2. Selecting the source type with the higher average temperature (if ground source is technically possible and economically feasible, this options is preferred).
3. Maximizing the heat output of the emitter system at low temperatures, thus lowering the average system temperature of the heating system.
4. Optimising the pump and room-temperature controls, thus ensuring that the supply temperatures are minimised and the individual rooms are only heated when needed.
5. Minimizing the electricity consumption of all electrical components (source-pump of fan, compressor, controls, circulator).

Wrong trade-offs between initial investments costs and system efficiencies can easily lead to low performing heat pump systems. But since system efficiencies are a difficult concept here and because the efficiency-class is already printed on the energy label, discussions on improving system components that could increase the overall efficiency will be difficult.

Primary topic of negotiation for the installation sector will be the initial investment costs.

The heat pumps that are on the market carry energy label classes varying from A+ and A++, indicating that seasonal space heating efficiencies are achieved of between 98 and 125% for A+, and between 125 and 150% for A++. The monitoring studies indicate that –after optimisation of the total heat pump system design– these values can be achieved in new dwellings, namely A+ for air source heat pumps, and A++ for ground source heat pump.

For existing dwellings however, these seasonal space heating efficiency figures are much harder to achieve. Primary reason for this can be found in the average supply temperatures of the emitter system that appear to be higher than assumed when calculating the seasonal space heating efficiency for the Energy Label.
2.9 Observations in relation to existing Regulation & EN 14825

Calculation of the seasonal space heating efficiency for electric heat pumps is done with the formula:

\[ \eta_{sh} = \frac{1}{CC} \times SCOP - F(1) - F(2) \]

With

- CC = Conversion Coefficient
- F(1) = Default Correction for standard suboptimal controls (= 3%)
- F(2) = Default Correction for electricity consumption ground water pumps (= 5%)
- SCOP = Seasonal Coefficient of Performance, in formula: \( \frac{Q_{H}}{Q_{HE}} = \frac{P_{designh} \times H_{HE}}{Q_{HE}} \).

With:

- \( Q_{H} \) = reference annual heating demand
- \( Q_{HE} \) = annual electricity consumption (for heat pump in active mode, thermostat-off mode, standby mode, for crankcase heater and including possible back-up heater).

Both \( Q_{H} \) and \( H_{HE} \) are determined using the BIN-method for a selected climate.

Unless explicitly a LT-heat pump is selected, the default supply temperature for the design point of an air source heat pump (ASHP) is set at 55°C (the assumption apparently is that the default emitter system is sized at a temperature regime of @ 55/45 °C). For an average climate the following test conditions are defined:

<table>
<thead>
<tr>
<th>Condition</th>
<th>Part load ratio [%]</th>
<th>( T_{outdoor_air} ) [°C]</th>
<th>( T_{supply_heating} ) [°C]</th>
<th>( T_{return_heating} ) [°C]</th>
<th>( T_{sys_heating} ) [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>88</td>
<td>-7</td>
<td>52</td>
<td>44</td>
<td>48</td>
</tr>
<tr>
<td>B</td>
<td>54</td>
<td>2</td>
<td>42</td>
<td>34</td>
<td>38</td>
</tr>
<tr>
<td>C</td>
<td>35</td>
<td>7</td>
<td>36</td>
<td>28</td>
<td>32</td>
</tr>
<tr>
<td>D</td>
<td>15</td>
<td>12</td>
<td>30</td>
<td>22</td>
<td>26</td>
</tr>
</tbody>
</table>

Averaging these CH supply, return and system temperatures according to the BIN-method for an average climate, the average seasonal temperature levels according to these test conditions are:

\( T_{supply} : 38°C \), \( T_{return} : 30°C \) and \( T_{system} : 34°C \)

Following this calculation method the seasonal space heating efficiency for an average ASHP in an average climate is around \( \eta_{sh} = 122\% \) (including the -3% correction for controls) with a SCOP of around 3.14.

The monitoring studies determine the combined SCOP for both DHW and space heating. Assuming an average SCOP for DHW of around 2.65 (@ A7/W55 source: Eurovent database), and assuming that 1/3 of the required energy is used for DHW and 2/3 for space heating the overall average SCOP for combined operation would be around \((2.65 \times 0.33 + 3.14 \times 0.67) = 3.0\)

German monitoring studies show, that in existing dwellings (with a share of electricity used for DHW of on average 15%) SCOP values of ASHP’s for combined DHW and space heating operation vary from 2.1 to 3.3 with an average of 2.6. When the installations in existing dwellings are further optimised (mainly optimised controls and emitter capacity), an average SCOP of 3.1 can be achieved, with a spread ranging from 2.6 to 4.1. The average system supply temperature for space heating was 43°C, with a spread ranging van 34°C to 50°C. The dwelling with the lowest seasonal average supply temperature of 34°C achieves the highest SCOP, namely 4.1. The dwelling with the highest seasonal average supply temperature of 50°C achieves the lowest SCOP, namely 2.6.

The UK monitoring study reveals average SCOP values of ASHP’s for combined DHW and space heating of around 3.0.
heating operation (after system modifications) of around **2.45**, with a range from 2.0 to 3.7. These observations illustrate that the test- and calculation methods used in the Regulation are adequate, but that the assumptions on the system temperatures on the sink side and related value for the seasonal space heating efficiency in existing dwellings are **too optimistic**. The actual real-life seasonal **average supply temperature** is the **crucial** parameter for achieving the projected seasonal space heating efficiency, and is determined by both the emitter capacity and the pump- and temperature controls on the sink-side. **These parameters can constitute the difference between a seasonal efficiency of 85% or 174%**. Following the findings for existing dwellings, the average supply temperature over the heating season is at least 5K too low (on average over the heating season it should be 43°C instead of 38°C), resulting in an average SCOP that is at least 15% too optimistic.

In other words, for a realistic rating the existing buildings the heat pumps should be rated according to the High Temperature (rated $T_{\text{supply}}$ 65°C, $\Delta T=10K$) regime and not the current Medium Temperature regime (rated $T_{\text{supply}}$ 55°C, $\Delta T=10K$).

Furthermore, given the importance of emitters as well as temperature- and flow controls for the space heating appliances, it is advisable to include this subject in the future Ecodesign working programme. As mentioned, for heat pumps this can mean the difference between and electric SCOP of 2.6 and 4.1, i.e. a space heating energy saving of 50%. For gas boilers, discussed hereafter, the difference is not so high but could still amount to 8-10% between a (nominally ‘condensing’) boiler that is condensing in practice and one that is not.

**Note:** Same line of reasoning goes for the ground source heat pumps, with the difference that all SCOP values are higher due to the higher source temperatures.
3 PRODUCT- & SYSTEM FEATURES DETERMINING SPACE-HEATING SYSTEM EFFICIENCY

3.1 Introduction

The results of various monitoring studies on real-life heat generator efficiencies (see section 2) clearly indicate that the heat generator itself is not the only component that determines its own generating efficiency. This means that, merely replacing an existing heat generator by a new and more efficient one, does not necessarily imply that the full saving potential of the new generator is achieved. To unlock the full saving potential of both condensing boilers and heat pumps, a system approach is required. For new buildings this can easily be done, because all necessary components need to be specified and procured. For existing buildings however, this will be more difficult since various system components are already in place, and the preferred approach often is that only the heat generator is to be replaced.

In the following paragraphs of this chapter the influence system components have on the generator efficiency or on the total system efficiency is further assessed.

3.2 System components

A hydronic space heating system consists of a heat generator that heats the water, which is then transported to the various heat emitters through a distribution- or pipe systems. When the heated water has entered the various heat-emitters, they subsequently transfers the heat into the rooms, either by convection, radiation or both. Controls determine the supply temperature and when which emitters are heated and to what extend they are heated.
3.3 Heating System Efficiency

The real life overall efficiency of a hydronic heating system is primarily determined by (in order of priority):

- System temperature heat-emitters
- Type of heat generator
- Type of heating schedule control
- Type of system temperature controls
- Type of flow controls

3.3.1 System temperature heat emitters

The heating system design temperature is the average temperature at which the emitter system inserts enough heat into the room (or building) to achieve and maintain the requested temperature in that room during the coldest day. This average temperature is equal to the sum of the supply- and return temperature divided by 2, or in formula:

\[ T_{sys} = \frac{(T_{sup} + T_{ret})}{2} \]
By lowering the system temperature, the generator efficiency will increase. This is true for both condensing boilers (they will condensate more) and for heat pumps (they will achieve higher COPs).

The system temperature however is not determined by the heat generator. The system temperature that is actually **needed** to adequately heat the individual rooms is determined by the heat output capacity of the heat-emitters in relation to the heat load of the room. The higher the emitter capacity (EC) in relation to the heat load (HL), the lower the system temperature will be. In other words, the heating curve that is needed, is determined by the heat load to emitter capacity ratio: **HL/EC-Ratio**

The two strategies to reduces system temperatures therefore are:

1. Reduce heat load by minimizing transmission losses, infiltration losses and ventilation losses (or in other words, increasing insulation and air-tightness levels of the building shell and applying smart ventilation).
2. Increase emitter capacity by installing a larger heat emitter capacity. For floor heating systems this means decreasing the pipe-distance of the floor heating system, increasing the insulation layer under the floor heating system and lowering the thermal capacity of floor above the floor heating system. By doing all of this, the heat output per m² can be increased and the system temperature can remain low. Following the EN442, the radiator capacity is determined at 75/65/20 °C. The heat output at lower temperatures is determined by calculation. A type 22 steel panel radiator with an heat-output at 75/65/20 °C of 1000 Watts for instance, will have a heat output at 55/45/20 °C of approximately 500 Watts, and

(Source: Viessmann)

**Figure 20. Heating curves for various HL/EC-Ratios**
at 45/35/20 °C of around 300 Watts (ratio of 3.3 with 75/65/20 °C temperature regime). So if the heat load of a room is 1 kW at for instance -10°C outdoor temperature, a radiator capacity at 75/65/20 °C of 3,3 kW is needed, to achieve a temperature regime of 45/35/20 °C on the coldest day. If the outdoor temperature rises, the instantaneous heat load diminishes, resulting in an even lower system temperature.

To predict the actual system efficiency with a certain heat generator, the ratio between heat-load (HL) and emitter capacity (EC) at system design temperature (and a room temperature of 20 °C) of the actual room or dwelling, is a good indicator.

\[ R_{HL:EC@40} = \frac{HL}{EC @ 40°C} \]

If this ratio is 1.0, the dwelling with its emitter system is perfectly equipped for low system temperatures. The ratio of 1 indicates that system temperature will be around 40 °C during the coldest day, and lower during higher outdoor temperatures. This means that when condensing boilers or heat pumps are used as the heat generator, the system efficiencies can potentially achieve the highest levels.

The higher this ratio between heat-load and emitter capacity at 45/35/20 °C, the lower the achievable system efficiency. And vice versa, the lower this ratio, the higher the achievable system efficiency.

In new dwellings it is easier to achieve low \( R_{HL:EC@40} \) values, because heat loads can be minimised through well thought-out building techniques and smart ventilation systems, to values of around 2 – 4 kW for e.g. a 100 m² dwelling. This 2 to 4 kW heat output can comfortably be achieved using low temperature (40/35/20 °C ) floor heating systems (providing around 50 W/m² floor area) in the entire house. If floor heating in the living room is combined with radiator heating in the bedrooms, special attention is requested when sizing the radiators, which needs to be done using this same temperature regime. Unfortunately, installer custom is to size the radiators for low temperature, which in the technical jargon of today corresponds to a system temperature of 50 °C (55/45/20 °C). As a result the system temperature for the radiators requires a second heating loop with higher temperatures than the floor heating system. This reduces system efficiency.

In existing dwellings it is a lot more difficult to achieve low \( R_{HL:EC@40} \) values. Initially heat loads will be considerably higher, and although improvements of the building shell will lead to reduced heat loads, the values that are achieved in the new built are not always feasibly in renovation projects. With thorough renovations, heat load values of 5 to 10 kW may be achieved in an average 100m² existing building (see Table 3 for average heat loads, applied systems temperatures and emitter capacity @ given system temperature, over the years). The table shows that by improving building insulation and air-tightness, the heat load is reduced and the system design temperatures are lowered as a logical consequence from around 80°C to around 50°C, using the same size of heat emitters.
Table 3. Heat load, system temperatures and average emitter capacity over the years
(source Viessmann, Vaillant et al.)

<table>
<thead>
<tr>
<th>Year of construction</th>
<th>Heat load dwelling [W/m²]</th>
<th>System design temperature</th>
<th>Avg. emitter capacity @ design temp. for avg. 100m² dwelling</th>
</tr>
</thead>
<tbody>
<tr>
<td>Old (before 1960)</td>
<td>≥ 250</td>
<td>80°C</td>
<td>&gt;25 kW</td>
</tr>
<tr>
<td>1960 – 1977</td>
<td>130 - 250</td>
<td>80°C</td>
<td>20 kW</td>
</tr>
<tr>
<td>1977 – 1982</td>
<td>70 – 130</td>
<td>70°C</td>
<td>13 kW</td>
</tr>
<tr>
<td>1982 – 1995</td>
<td>60 – 100</td>
<td>70°C</td>
<td>10 kW</td>
</tr>
<tr>
<td>1995 – 2002</td>
<td>40 – 60</td>
<td>65°C</td>
<td>6 kW</td>
</tr>
<tr>
<td>2002 – 2009</td>
<td>30 – 50</td>
<td>60°C</td>
<td>5 kW</td>
</tr>
<tr>
<td>2009 –</td>
<td>25 – 40</td>
<td>50°C</td>
<td>4 kW</td>
</tr>
<tr>
<td>Passive house</td>
<td>≤ 15</td>
<td>40°C</td>
<td>&lt; 2 kW</td>
</tr>
</tbody>
</table>

This means that, in existing dwellings, using existing radiator sizes, system design temperatures may at best drop to around 50 to 55 °C. If the emitter capacity is further increased, system temperatures can drop to values of around 40 °C. It is therefore essential that during renovation projects explicit attention is given to the capacity of emitter system, if $R_{HL:EC@40}$ values of around 1.0 are to be achieved.

Emitter capacity can be increased by:

- Adding additional radiators;
- Replacing existing radiators by larger radiators or by LT-radiators/convectors (specially designed for delivering high heat-output at lower system temperatures);
- Adding floor- or wall heating.

Preferably the emitter capacity is increased to a level that a system temperature of 40 °C suffices, meaning that radiators/convectors and floor- and/or wall heating can be supplied with the same system temperatures (no second heating loop with higher system temperatures!).

In summary:

If one really wants to achieve low system temperatures in existing dwellings and thus significantly improve operating efficiencies for both condensing boilers and heat pumps, reducing the heat loads of the dwelling may not be enough. An additional evaluation of the emitter system capacity (and possibly an increase) may be needed to achieve continuously low system temperatures, even during cold periods when air source heat pumps have difficulties achieving good COP-values.

By reducing the $R_{HL:EC@40}$ ratio to values of around or below 1.0, the system efficiency can be increased. By knowing the actual $R_{HL:EC@40}$ ratio of a dwelling, the system efficiency with a certain type of heat generator, can more accurately be predicted.

Example:
In the UK the installation sector usually tends to install the smaller radiators, leading to high $R_{HL:EC@40}$ values. As a logical result of this habit, system temperatures are high and the generating efficiencies for boilers and heat pumps will be lower compared to e.g. Germany (see results of monitoring studies in section 2). This habit has also led to the need of extra radiator casings to reduce the surface temperatures of radiators and minimise the risk of skin burns (a.k.a. LST-radiators, a typical UK-product).
3.3.2 Type of heat generator

All heating systems, regardless the type of heat generator, will achieve better system efficiencies at low system temperatures. Not only are the heat losses in the distribution system reduced, also the generator efficiency will increase, especially when using condensing boilers and heat pump. However, the interaction between generator efficiency, the design system temperature and the associated supply- or return temperature is different for the two generator types.

Condensing boilers

Condensing boilers benefit from low return-temperatures. The lower the return temperature, the more condensate the boiler produces. Above return temperatures of 56 °C no condensation occurs in the boiler itself (condensate flowing back from the flue ducts or chimney however may be observed, but this condensation does not contribute to the generator efficiency). Below return temperatures of around 56 °C the boiler itself starts producing condensate, simultaneously increasing its efficiency (see Figure 21 for the relation between return temperature and boiler efficiency). The efficiency may increase from 87% on GCV without condensation, to around 97% with full condensation (return temperatures are approaching room temperature levels).

The actual occurring return temperature is determined by both the system temperature and the flow. Example: at a certain system temperature of for instance 40 °C, the associated supply and return temperature is determined by the flow through the emitter. A high water flow will reduce the delta T and may result in a supply temperature of e.g. 41 °C and return temperature of 39 °C. A low water flow will increase the delta T and may reduce return temperatures to as much as 30 °C with an associated supply temperature of 50 °C. In both cases, the thermal energy induced by the emitter into the room may be more or less the same, but the generator efficiency will be higher in the latter.

Figure 21. Effect of return temperature on efficiency of condensing boilers

The actual occurring return temperature is determined by both the system temperature and the flow. Example: at a certain system temperature of for instance 40 °C, the associated supply and return temperature is determined by the flow through the emitter. A high water flow will reduce the delta T and may result in a supply temperature of e.g. 41 °C and return temperature of 39 °C. A low water flow will increase the delta T and may reduce return temperatures to as much as 30 °C with an associated supply temperature of 50 °C. In both cases, the thermal energy induced by the emitter into the room may be more or less the same, but the generator efficiency will be higher in the latter.

The type of emitter also influences the maximum delta T that is feasible. For floor heat heating for instance, a lower delta T (up to 4 °C) is preferred in order to prevent colder floor sections. For convectors or steel plate radiators a large delta T (up to 20 °C) is feasible.

Since the resistance in the distribution system is continuously changing because of closing or opening TRVs (thermostatic radiator valves) it is difficult to control the return temperature. A sophisticated pump- and supply temperature control system is needed to continuously induce the right amount of flow and supply temperature, while minimising the return temperatures.

**Heat pump**

A heat pump’s generating efficiency benefits not only from high source temperatures, but also from low sink temperatures, or in other words low heating system supply temperatures. In general, smaller temperature-lifts strongly increase the COP levels that can be achieved by the heat pump.

So contrary to condensing boilers, the pump- and supply temperature control system should be optimised here to minimise the delta T on the sink-side.

Figure 22. Relation between average heating COP for air and ground source heat pumps (left and right, respectively) and temperature lift (difference between $T_{\text{source}}$ and $T_{\text{sink}}$), based on data taken from industrial surveys and field trials.\(^{11}\)

Figure 22 illustrates the relation between temperature lift and COP. To get a feel of the relation between heating system supply temperature and COP some examples are given in the table below.

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\(^{11}\) Ian Staffell et al, A review of domestic heat pumps, Energy & Environmental Science, Volume 5, 2012
### Table 4. COPs for a heating system design temperature of 40 °C and a temperature regime 42/38/20 °C

<table>
<thead>
<tr>
<th>$T_{outdoor}$</th>
<th>$T_{supply}$</th>
<th>$T_{lift}$</th>
<th>Avg. COP ASHP</th>
<th>Avg. COP GSHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10 °C</td>
<td>42°C</td>
<td>52°C</td>
<td>2.2</td>
<td>3.0</td>
</tr>
<tr>
<td>0°C</td>
<td>38°C</td>
<td>38°C</td>
<td>3.2</td>
<td>4.2</td>
</tr>
<tr>
<td>7°C</td>
<td>32°C</td>
<td>25°C</td>
<td>4.3</td>
<td>5.5</td>
</tr>
<tr>
<td>10°C</td>
<td>30°C</td>
<td>20°C</td>
<td>4.7</td>
<td>6.0</td>
</tr>
</tbody>
</table>

### Table 5. COPs for a heating system design temperature of 50°C and a temperature regime 52/48/20°C

<table>
<thead>
<tr>
<th>$T_{outdoor}$</th>
<th>$T_{supply}$</th>
<th>$T_{lift}$</th>
<th>Avg. COP ASHP</th>
<th>Avg. COP GSHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10°C</td>
<td>52°C</td>
<td>62°C</td>
<td>1.7</td>
<td>2.4</td>
</tr>
<tr>
<td>0°C</td>
<td>48°C</td>
<td>48°C</td>
<td>2.5</td>
<td>3.3</td>
</tr>
<tr>
<td>7°C</td>
<td>42°C</td>
<td>35°C</td>
<td>3.3</td>
<td>4.5</td>
</tr>
<tr>
<td>10°C</td>
<td>40°C</td>
<td>30°C</td>
<td>3.8</td>
<td>4.9</td>
</tr>
</tbody>
</table>

### Table 6. COPs for a heating system design temperature of 60°C and a temperature regime 62/58/20°C

<table>
<thead>
<tr>
<th>$T_{outdoor}$</th>
<th>$T_{supply}$</th>
<th>$T_{lift}$</th>
<th>Avg. COP ASHP</th>
<th>Avg. COP GSHP</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10°C</td>
<td>62°C</td>
<td>72°C</td>
<td>1.4</td>
<td>1.9</td>
</tr>
<tr>
<td>0°C</td>
<td>55°C</td>
<td>55°C</td>
<td>2.1</td>
<td>2.8</td>
</tr>
<tr>
<td>7°C</td>
<td>50°C</td>
<td>43°C</td>
<td>2.8</td>
<td>3.6</td>
</tr>
<tr>
<td>10°C</td>
<td>45°C</td>
<td>35°C</td>
<td>3.3</td>
<td>4.5</td>
</tr>
</tbody>
</table>
Table 4, Table 5 and Table 6 illustrate, that COP values can increase considerably over the whole supply temperature range when the heating system design temperature is reduced. The COP-gains are bigger for the ground source heat pumps than for air source heat pumps. It is estimated that a 10 °C reduction of the system design temperature (from 50 to 40 °C) can increase the seasonal space-heating efficiency for an average 100m² dwelling by at least + 40%. This means that if the seasonal space-heating efficiency was 110% @ system design temperature of 50 °C, it becomes 150% @ system design temperature of 40 °C.

These figures indicate that there is a significant saving potential related to the optimisation of the system design. The figures also indicate that a delta T of 2 °C between supply and return temperature instead of 20°C can make all the difference in this context.

### 3.3.3 Type of heating schedule control

In most cases, the rooms in a dwelling do not need to be kept permanently at the preferred indoor temperatures (e.g. 21 °C for the living, 23 °C for the bathroom, 18 °C for the bedrooms). During absence, temperature set-backs are allowed. These set-back temperatures may depend on the responsiveness of emitter system applied and range from 1 to 4 degrees.

Heating schedules may help minimising energy losses due to unnecessary heating of one or more rooms in a dwelling. In addition to manually switching the heating system on or off, there are various other ways in which heating schedules can be implemented, each one with its specific pros and cons:

**A) Switching boiler on/off with a timer**

Switching the boiler on and off is a rather basic way of implementing a heating schedule. The setback temperature is not controlled which may (depending on the heat load of the house) lead to longer heat-up periods, temporarily higher system temperatures and lower system efficiencies. This method assumes that the house is unoccupied during a certain part of the day. With system A1 (see table below) the temperature in ‘the other rooms’ are a derivative and cannot be controlled. At best they can be maximised by using TRVs. With system A2 the feed temperature is controlled by the heating curve which enables the control of the temperature in the other rooms by using a TRV, but only during the periods that the boiler is switched on.

**Table 7. Switching boiler on/off with a timer**

<table>
<thead>
<tr>
<th>Type</th>
<th>Heat generator</th>
<th>Supply temperature</th>
<th>Room temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Timer switches boiler on or off (e.g. on: from 06:30 to 09:00 and from 15:30 to 23:00)</td>
<td>When boiler is on, ( T_{supply} ) is determined by set point room thermostat and design ( R_{HL:EC@40} ) of ref. room</td>
<td>When boiler is on, ( T_{ref} ) is determined by set-point room thermostat. ( T_{set-back} ) is not controlled</td>
</tr>
<tr>
<td>A1.</td>
<td>idem</td>
<td>When boiler is on, ( T_{supply} ) is determined by heating curve and ( T_{outdoors} ) (value from sensor or via internet)</td>
<td>When boiler is on, ( T_{ref} ) is determined by position of TRV. ( T_{set-back} ) is not controlled</td>
</tr>
<tr>
<td>A2.</td>
<td>idem</td>
<td>When boiler is on, ( T_{supply} ) is determined by set point room thermostat, ( T_{ref} ) is determined by set-point room thermostat, ( T_{set-back} ) is not controlled</td>
<td>When boiler is on, ( T_{room} ) is derived from ( T_{supply} ), design ( R_{HL:EC@40} ) of other rooms and position radiator valve</td>
</tr>
</tbody>
</table>
B) Single thermostat with heating schedule

With the boiler continuously in stand-by mode, the thermostat - in which a heating schedule is implemented - switches the boiler on and off. This heating schedule also facilitates the implementation of set-back temperatures, preventing the room temperatures from falling too low. This improves thermal comfort and may help increasing the system efficiency.

These programmable thermostats generally allow a more detailed daily temperature programs (using multiple switch points and temperature levels) than possible with type A heating schedule controls.

In case of a room thermostat (type B1) the temperature of room in which the thermostat is installed (reference room) will determine when the boiler is switched on, meaning that the temperature in reference room is controlled. The temperature in ‘the other rooms’ are a derivative and cannot be controlled. At best they can be maximised by using TRVs.

Type B2 allows for an additional outdoor sensor to be connected to the system, in which case a weather compensated feed temperature is used. The room thermostat can be used to correct upon these weather compensated feed temperatures, for situations where the heating curve is set too high. The temperature schedule of the room thermostat is also used to reduce the heating curve when a set-back temperature for the reference room is applicable.

Type B3 consist of a boiler using only a weather compensated control system and a clock-program that can be implemented on a thermostat that is mounted/attached onto the boiler. This heating schedule with temperature set-backs result in related parallel displacements of the heating curve.

Table 8. Single thermostat with heating schedule

<table>
<thead>
<tr>
<th>Type</th>
<th>Heat generator</th>
<th>Supply temperature</th>
<th>Room temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>reference room</td>
</tr>
<tr>
<td>B1.</td>
<td>Boiler is continuously in stand-by mode and is switched on by the room-thermostat with heating schedule</td>
<td>$T_{supply}$ is determined by set point room thermostat and design $R_{HL:EC@40}$ of the reference room</td>
<td>$T_{ref}$ is determined by set-points room thermostat</td>
</tr>
<tr>
<td>B2.</td>
<td>Boiler is continuously in stand-by mode and if necessary switched on to achieve requested supply temperature</td>
<td>$T_{supply}$ is determined by heating curve, $T_{outdoors}$ and feedback from room thermostat</td>
<td>$T_{ref}$ is determined by set-points room thermostat</td>
</tr>
<tr>
<td>B3.</td>
<td>Boiler is continuously in stand-by mode and if necessary switched on to achieve scheduled supply temperature</td>
<td>$T_{supply}$ is determined by heating curve and $T_{outdoors}$ and the heating schedule in boiler controls</td>
<td>$T_{ref}$ is by $T_{supply}$, design $R_{HL:EC@40}$ of the room, position TRV, and heating schedule</td>
</tr>
</tbody>
</table>
C) Multiple thermostats with heating schedule

A system with multiple programmable thermostats allows for a separate heating schedule for each individual room. These ‘heating schedule per room’ control solutions represent the most sophisticated way to control temperature in the individual rooms of a dwelling or building. All rooms are kept at the optimised set-back temperatures when no one is present, and during presence the individual preferences regarding room temperature are achieved. Provided the heating schedules are truly implemented and comply with the actual use of the rooms, these systems achieve the highest thermal comfort and do not heat rooms unnecessarily.

Table 9. Multiple thermostats with heating schedule

<table>
<thead>
<tr>
<th>Type</th>
<th>Heat generator</th>
<th>Supply temperature</th>
<th>Room temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>reference room</td>
</tr>
<tr>
<td>C1.</td>
<td>Boiler is continuously in stand-by mode and if necessary switched on to achieve requested supply temperature</td>
<td>( T_{\text{supply}} ) is determined by heating curve and ( T_{\text{outdoors}} ) (weather compens. control)</td>
<td>( T_{\text{ref}} ) is determined by set-points of the programmable TRV</td>
</tr>
<tr>
<td>C2.</td>
<td>Boiler is continuously in stand-by mode and if necessary switched on to achieve requested supply temperature</td>
<td>( T_{\text{supply}} ) is determined by dynamic feedback from all room thermostats</td>
<td>( T_{\text{ref}} ) is determined by set-points room thermostat (communicating with motorized valve radiator)</td>
</tr>
</tbody>
</table>

Type C1 works with weather compensated control system to determine the supply temperature. The TRV in each room follow their individually programmed heating schedules for opening or closing the TRVs. Type C2 utilises programmable room thermostats in each room that communicate with motorized valves either in the distribution manifold or in the emitter itself. The supply temperature is determined using an algorithm that converts the feedback coming from all room thermostats to an optimised feed temperature.

Heating schedule controls may to a large extent influence the energy consumption for space heating. The saving potential largely depends on the actual use and correct implementation of the heating schedules. Wi-Fi and blue tooth communication with smartphones will facilitate and simplify the implementation of correct heating schedules. Self-learning algorithms combined with actual presence detection methods may prove to be the best way to implement proper heating schedules.

Note: Heating schedule controls are not included in the existing Regulation on space heaters.

3.3.4 Type of system temperature controls

The fact that the emitter system capacity is designed for a specific system design temperature of for instance 40 °C does not automatically imply that the correct supply- and return temperatures will be implemented in real life. The way in which boiler feed temperature and pump are controlled, will to a large extent determine the actually occurring supply and return temperature and with it, the heat generating efficiency.
Room thermostat controlled systems

System temperature

In heating systems that use a room thermostat, the design $R_{HL:EC@40}$ ratio of the reference room (where the room thermostat is located) will determine the system temperature. If the design $R_{HL:EC@40}$ ratio is 1, the system temperature will be around 40 °C if it is -10 °C outside. The heat load of the reference room will vary, due to:

- different set point for room temperature (e.g. 18 °C instead of 21 °C);
- varying outdoor temperatures;
- varying solar radiation entering the room;
- varying internal gains.

This means that the actual $R_{HL:EC@40}$ ratio will vary as well and will in most cases be lower than the design $R_{HL:EC@40}$ ratio. For this reason the term ‘dynamic $R_{HL:EC@40}$ ratio’ is introduced.

If a modulating room thermostat in combination with a modulating boiler is used, the system temperature will converge towards the lowest possible value given the dynamic $R_{HL:EC@40}$ ratio of the reference room.

If an on/off room thermostat and/or an on/off boiler is used, the system temperature will oscillate around the lowest possible value for the dynamic $R_{HL:EC@40}$ ratio, leading to on average reduced thermal comfort, slightly higher system temperatures and lower system efficiencies due to cycling and the slightly higher system temperatures. Depending on the type of room thermostat (mechanic on/off, electronic on/off, Time Proportional and Integral (TPI) on/off control or a modulating room thermostat), the temperature oscillation will be high or low, resulting in a larger or smaller drop in system efficiency.

The temperature that is achieved in all the other rooms of the dwelling, depend upon the heating schedule for the reference room. If there is no heating demand in the reference room, the other rooms will not be heated either. If the reference room is heated, the system temperature for the other rooms will be the same as for the reference room. Depending on the design $R_{HL:EC@40}$ ratios of the other rooms, they can or cannot be heated sufficiently with the given system temperature.

The fact that the other rooms are not always sufficiently heated when needed, may lead to improper use of this control system. Example: the set temperature is the reference room is temporarily increased (to e.g. 24 °C) in order to sufficiently heat the other rooms. This leads to unnecessary heating of the reference room and to higher system temperatures than strictly needed.

Supply- and return temperatures

With a system temperature that is determined by the dynamic $R_{HL:EC@40}$ ratio in the reference room, the actual supply and return temperature is determined by the pump and the flow resistance in the distribution system. Because thermostatic radiator valves (TRV) may continuously open or close, the flow resistance of the distribution system also is a dynamic parameter. The combined effect of pump settings and dynamic flow resistance will lead to either a high flow velocity over the emitters, resulting in a small delta $T$, or a low flow velocity resulting in a big delta $T$ over the emitter.
This means that depending on the setting of the pump and the dynamic flow resistance, the delta T can be somewhere between 3 °C and 30 °C.

Two extreme examples of room temperature controlled systems:

**Example 1**
If the reference room is for instance the hall behind the front door, and the hall has a relatively high heat load (inferior door sealing) and a small radiator because of limited available space, the $R_{HL:EC@40}$ ratio will be high. The small radiator will have difficulties heating the hall to the requested room temperature, meaning that the room thermostat does not switch off the heat generator. The boiler remains in operation until the maximum supply temperature is reached. In this case supply temperatures may rise to 80 °C, even with moderate outdoor temperatures. Regardless the pump setting or dynamic flow resistance, the return temperature will seldom be below 56 °C. Condensation will not occur.

**Example 2**
If the living room is the reference room, and the $R_{HL:EC@40}$ ratio of the living room is relatively low (< 1.0, system design temperature is < 40 °C), because large radiators are used and the room is well insulated and situated in between apartments, little effort from the radiators is required to achieve the requested room temperature. Already at low system temperatures, sufficient thermal energy is induced into the room and the room thermostat will switch off the heat generator before higher system temperatures are achieved.

Provided the emitter system is hydraulically balanced and the pump is ΔP-controlled, the average delta T may be (depending on ΔP settings) around 20 °C (leading to average return temperatures of below 30 °C, maximising condensate production in condensing boilers) or around 4 °C (leading to low supply temperatures, maximising COP in HP systems).

**Conclusion**
If low system temperatures are the aim in room thermostat controlled systems, the design $R_{HL:EC@40}$ ratio of at least the reference room must be ≤ 1. Flow-resistance in the system and pump controls will in the end determine what the difference between supply and return temperature is. From a generator efficiency point of view, condensing boilers prefer a high delta T, while heat pumps prefer a low delta T.

**Weather compensated systems**

**System temperature**
In weather compensation controlled systems, the heating curve determines the supply temperature. The heating curve in fact, represents the relation between outdoor temperature and heating system supply temperature. Once the outdoor temperature is known, the supply temperature can be calculated and the heat generator will heat the outgoing CH-water to up until this value.

Difficulty with these weather compensation controlled systems is the fact that even if the optimal heating curve is selected, it still represents the highest HL/EC-Ratio of the
various rooms (they are never all the same). Furthermore, the heat loads in a room may change due to more solar radiation entering the room, more internal gains or due to lower set temperatures. This could in theory be used to further reduce the heating curve. Finally the heating curve is often manually set by the installer. A low slope of the heating curve will result in low system temperatures and a higher risk that room are not heated sufficiently heated. A high slope will results in high system temperatures but no risk whatsoever of not heating the rooms sufficiently. And since TRVs are often installed on the emitters there is no risk that excess room temperature will occur.

So in real life there is a serious risk that, even if low $R_{HL:EC@40}$ ratios are applicable for the various rooms, the installer will still select a higher heating curve than needed. Whether the correct heating curve is implemented up until today largely depends on the installer and his willingness to carefully fine-tune the lowest possible heat curve. Unfortunately installers predominantly apply the ‘better safe than sorry’ policy.

An improvement on a heating curve that is set too high can be achieved by using an additional room sensor that monitors the room temperature and adjusts the heating curve when needed.

**Supply- and return temperatures**

With a system temperature that is determined by the heating curve, the actual supply and return temperature is determined by the pump and the flow resistance in the distribution system. Because thermostatic radiator valves (TRV) may continuously open or close, the flow resistance of the distribution system also is a dynamic parameter. The combined effect of pump settings and dynamic flow resistance will lead to either a high flow velocity over the emitters, resulting in a small delta T, or a low flow velocity resulting in a big delta T over the emitter.

This means that depending on the setting of the pump and the dynamic flow resistance, the delta T can be somewhere between 3 °C and 30 °C.

**Conclusion**

If low system temperatures are the aim in weather controlled systems, the $R_{HL:EC@40}$ ratio of all the rooms must be $\leq 1$, and the heat curve must be very carefully tuned to the lowest slope possible.

Flow-resistance in the system and pump controls will in the end determine what the difference between supply and return temperature is. From a generator efficiency point of view, condensing boilers prefer a high delta T, while heat pumps prefer a low delta T.

**Multiple room temperature controlled systems**

Multiple room temperature control systems follow the principle that the heat demand in each room is measured. A central controller determines the overall heat demand based upon the differences between set point and actual room temperature, and optimizes the feed temperature following the changes in the delta Ts in the various rooms, and the speed in which these changes occur.

These control strategies are now mainly focussed on thermal comfort (achieving the requested room temperatures with as little variations possible).
In the near future, self-learning algorithms may be applied that - next to thermal comfort - focus on system efficiency. This can be achieved by minimising supply temperatures and by optimising the delta T between supply and return per room, depending on the type of heat generator and type of emitter.

**3.3.5 Type of flowrate controls**

Flowrate control is crucial for achieving the correct delta T over the various emitters of a hydronic heating system. If the flow over an emitter increases, the delta T decreases and vice versa.

Heating systems with condensing boilers and radiator- or convector type emitters, benefit from a higher delta T, resulting in lower return temperatures and improved condensation. Heat pumps systems on the other hand prefer a lower delta T because at lower supply temperatures the COP of the heat pump increases. When floor (or wall) heating is used, not only a limited system temperature is requested (for comfort reasons), but also a limited delta T is preferred to prevent colder sections in the floor heating surface.

Unfortunately, scant attention is given to proper flow control solutions particularly concerning heating systems in existing dwellings. The default situation in a large section of the existing market consists of non-hydraulically balanced emitter system with fixed speed circulators. The opening and closing of radiator valves cause continuously changing flow resistance and continuously changing flow over the emitters leading to large variations in supply- and return temperatures. This leads to suboptimal heating system efficiencies in existing premises.

The following situations occur:

**No control/no balancing**

The pump (in the existing market often a fixed speed circulator) and the accidently created flow resistance determine the actual delta T over the emitters. Some emitters receive large flows and heat-up quickly, while others may receive only small water flows and barely succeed in maintaining required room temperatures. Opening and closing TRVs will continuously change the flow resistance of emitters and with it the supply- and return temperatures. These systems have no control over the delta T. It is estimated that around two/third of the existing market is stuck with such a rudimentary system.

**Manually hydraulic/hydronic balancing of emitter system**

The emitter system is hydraulically balanced by adding and manually adjusting restrictors in each individual emitter in such a way that all emitters more or less receive the requested flow in periods when all emitters are switched on. By doing this the overall flow resistance in the distribution system is increased (as is the pump pressure), but now all emitters can be properly heated. However, in part load situations (which are most of the time) only a few radiators will be opened, which leads to different flows and different delta T, compared to a situation where all radiators are opened. Opening and closing TRVs will further influence the flow resistance and the supply- and return temperatures. Depending on the type of pump-control these heating systems have only limited control over the delta T that occurs over the emitters.
**TRV with Automatic Flow Control (AFC)**

When heat-emitters are outfitted with TRVs that have an integrated flow limiter, the maximum flow can be set for each individual emitter. As long as the pump pressure is high enough, all opened TRVs will only allow the maximum amount of water to pass through. With a delta P-controlled pump, the rpm off the pump will automatically be adjusted when TRVs are opened or closed, while the system pressure remains the same. Emitter systems that use these TRVs with integrated AFC will have a fairly good control over the delta T that occurs over the emitters.

![Automatic Flow Control TRV](image)

**Figure 23. Example of Automatic Flow Control TRV**

Heimeier introduced their version of the AFC technology in 2017 under the name Eclipse. AFC technology automatically regulates the maximum flow irrespective of the differential pressure. Thanks to the automatic flow control technology only the flow rate has to be defined. The flow rate is directly set at the valve with a setting key or an end wrench. The smaller size of the TRV fits all renovation applications and provides more mounting options, including straight, angle, double angle, right and left body housings.

**Pump controls**

As indicated, pump controls may to a large extent influence the occurring supply- and return temperatures.

When fixed speed circulators are used (older installations), the supply and return temperatures fully depend on the occurring and dynamically changing flow resistance in the distribution system. Even with a perfectly hydraulically balanced emitter system, the delta T may be set correctly when the valves of all emitters are fully open, but when one or more valves are closing (> 95% of the time), flowrates will increase and delta Ts will drop. But in most cases the pump is not set for correct delta Ts when all radiator valves are opened, due to DHW-function that may require higher pump settings.

Nowadays more and more high efficiency pumps are used that, aside from a higher motor efficiency, also contain speed control software and auto adapt algorithms that allow automatic adjustment of pump speed. Pumps speeds are commonly controlled using one of the following options:

**Constant pressure control**

Pressure difference across the pump remains constant at a value that is equivalent to the delta P of the whole system during design (= maximum) flow rates.
Proportional pressure control
Pressure across the pump reduces in proportion to flowrate towards a value that e.g. equal to 50% of delta P at maximum flowrate.

Remote ΔP-sensor control
Pressure across the pump reduces towards the pressure difference across the most remote ΔP-controlled sub-branch of the distribution system.

Properly controlled variable speed pumps not only contribute to a reduced power consumption of the pump, they also improve upon the dynamically achieved delta T over the emitters, provided the system is hydraulically balanced. This significantly improves the overall system efficiency.

Although variable speed pumps combined with emitters with TRVs and automatic flow control result in fairly good delta T control, a relatively high system pressure is needed to achieve this.

3.3.6 Future developments: Optimised flowrate and supply temperature control
Various papers and articles describe the fact that there is still room for further improvement concerning flowrate- and temperature controls, following the line of reasoning described below.

Weather compensated controls will always result in heating curves that deliver higher supply temperatures than strictly needed, because they do not react upon the continuously varying (and in 99% of the cases lower than the expected ) heat loads of the various rooms. But also room-thermostat controlled heating systems may lead to higher supply- and return temperatures that requested, especially when boiler capacity (even in part load) is too high and switched on and off and/or when simple on/off room thermostats are used. Because of this, TRVs are necessary to maximise the heat-input of the emitter and prevent room temperatures from getting too high, simply because the amount of heat supplied is higher than the actual heating demand of the individual rooms. But unfortunately TRVs are also a disturbing factor. They continuously change the flow resistance of the system, which leads to large variations in temperature difference between supply and return. In addition, to facilitate the minimum requested flowrate of a pump, by-pass provisions are necessary to deal with situations where all the TRVs are closed. By-pass provisions have the disadvantage that they increase the return temperature, which can considerably reduce system efficiencies.

If however, the dynamic heat load of each individual room can be measured and monitored and this exact amount of heat can be supplied to each individual room (applying optimised flowrates to achieve the preferred delta Ts), TRVs and by-pass valves would no longer be needed. The overall system would continuously operate at its best system efficiency possible (within the constraints imposed by the emitter capacity), with the lowest possible system temperatures and ΔT-optimised flowrates.

The text section below is a first attempt of the project team to conceptualise such a control system to facilitate further discussions.
Smart Distribution Manifolds

The control principle is based on:

- the use of a heating schedule per individual room, using setback temperatures (preferably using an app) that are optimised for the dwelling concerned
- accurate measurements of the temperature in each individual room, including the rate in which it changes (temperature gradient)
- continuous measurement of the flowrate and supply temperature for each individual room
- with test cycles the optimal flowrate per room is determined and fixed
- with learning curves (or test cycles) the minimal feed temperature that fits the instantaneous heat-load of the individual room are determined (such learning curves provide information on actual RHL:EC@40 ratio of the individual rooms and can be used to calculate the initial feed temperatures)
- the room with the calculated highest instantaneous feed temperature request is the one that determines the instantaneous feed temperature coming from the boiler.
- for rooms for which this instantaneous feed temperatures would be too high (out of a defined range) and may lead to higher room temperatures than required by the heating schedule, the flowrate for that specific room can be slightly reduced, by adjusting the position of the modulating valve; this will reduce the amount of heat that is fed into that specific room and results in lower system temperatures and lower return temperatures.
- a control algorithm is aimed at keeping the individual room temperature as close to the set point as possible by continuously varying the feed temperature and if necessary the flowrate by adjusting the $\Delta P$ over the pump.
- the use of a $\Delta P$-controlled pump; if these is no need for elevated feed temperatures (no heat demand) the pump is switched off.

The obvious component incorporating such control provisions would be a distribution manifold.

Prerequisite here is that:

- All rooms have their own supply- and return pipe connection to a (kind of) distribution manifold, that is outfitted with a modulating valve, a flow sensor and a temperature sensor measuring the feed temperature.
- In case a room uses more than one emitter, they will all be connected to this single supply and return pipe coming from the distribution manifold for the room concerned; it is also necessary that the emitters in one room are hydraulically balanced against one another.
- A wireless (or wired) room temperature sensor (also serving as unit for programming/ displaying local room temperature schedule) provides information on the requested and the actual achieved room temperatures.
- An outdoor sensor might help to better predict the requested supply temperatures (alternatively a value can be obtained from the internet through Wi-Fi).
- The boiler is able to supply low heat outputs, preferably to values below 1 kW.

In addition pump and modulating valves can be controlled in such a way that the flow per room either leads to higher delta Ts (increasing the generating efficiency of condensing boilers) or lower delta Ts (increasing the generating efficiency of heat pumps).
Advantages of such a Smart Distribution Manifold are:

1. Lowest possible system temperatures are always automatically achieved (within the constraints of the heat load of the dwelling and installed emitter capacity).
2. There is no more need for the installer to set the heating curve; a self-learning algorithm during operation (or a learning cycle before commissioning the heating system) can do the job.
3. The control system can generate information to the installer and/or consumer on the specific room or rooms that are limiting the overall system efficiency, by indicating the rooms with the highest HL/EC-Ratio. Further insulating those rooms or installing emitters with a higher heat output in those rooms can lift these restrictions regarding system efficiency. Also, indications of the saving potential (in %) of improved HL/EC-Ratios can be generated by the control system.
4. Delta Ts over the emitters are controlled, maximising the generating efficiency of the generator that is used.
5. Flow resistance within the emitter system is optimised in relation to the delta T required.
6. The use of multiple room thermostats enables that all rooms can be heated according its own room temperature schedule; presence detection sensors (of any kind) may be used to replace or complement the clock schedules in order to improve upon the heating schedules.

From a technical point of view there is no barrier to realise these types of solutions. From an economic point of view, it is estimated that - compared to weather compensated control systems - these solutions can lead to savings of up to 10% for condensing boilers and up to 40% for heat pump systems, assuming a achieved average overall reduction in return/supply temperatures of around 10 °C. Savings on energy costs for space heating, indicating that additional purchase costs for these solutions may be as high as €300 to €400 to achieve pay-back periods of around 5 years.

### 3.4 Observations in relation to existing Regulations & standards

System features that determine the operating efficiency of a heat generator are:

1. Emitter capacity in relation to the heat load (HL/EC-Ratio)
2. Heating schedule controls
3. System temperature and flow controls

**Ad 1).**
The HL/EC-Ratio is no topic in the existing regulation or in the related guidelines, despite the fact that this HL/EC-Ratio is crucial for achieving the operating efficiency indicated by the calculation methods of the Regulation. It is therefore proposed to explicitly appoint and discuss this parameter, at least in the regulation related guidelines. Goal is that the HL/EC-Ratio becomes an actual topic of discussion when heat pumps and condensing boilers are installed/replaced.

**Ad 2 and 3).**

Heating schedule controls are currently not incorporated in the control classes mentioned in the Transitional Method for space heaters and combination heaters 813/2013 and 811/2013.

Other topics for discussion regarding the existing control-classes (Class I to VIII) are:
- Feed temperature controls have a larger effect on the operating efficiency than the currently assumed default of 3%, especially in combination with heat pumps;
- Weather compensation controls generally results in higher average system temperatures than room thermostat controlled systems (provided the reference room has correct sized emitter capacity);
- Flowrate control is not incorporated in the various control classes, although they have significant influence on the operating efficiency of the heat generator;
- The effect of modulating thermostats and modulating heat generators is limited when minimal heat output of the generator is larger than the instantaneous heat load of the dwelling.
4 Technical improvements fuel boilers

Improvements that will be addressed here involve:

1. Technical solutions to further utilise the latent heat that remains in the flue gasses during DHW- and space heating operation by applying tertiary heat exchangers (a.k.a. PFHRDs);
2. Improvements in turn-down ratio;
3. Improvements in combustion control;
4. Improvement in pump controls.

4.1 PFHRD

Passive Flue Heat Recovery Devices are flue gas to domestic cold water heat exchangers that are used together with combination boilers and have the purpose to further cool down the flue gas temperatures in order to facilitate condensation and improve generator efficiency. PFHRDs can both be integrated in the boiler design and be added as a separate product, in which case the boiler manufacturer must confirm that the add-on PFHRD can be combined with the condensing boiler (PFHRD may influence the resistance on flue gas side and water side of the boiler).

In DHW-mode, the combination boiler operates at a temperature regime of 80/60 °C. At this return temperature no condensation occurs. By adding another heat-exchanger that on the water-side uses the cold water inlet and on the gas side the hot flue gasses produced by the boiler in DHW-mode, the flue gasses will condensate, leading to increased tapping efficiencies. These increased efficiencies can be measured using the EN13203-standard.

In space heating mode, the flue gasses preheat the PFHRD. The cold water that is stored inside the PFHRD is preheated accordingly. Depending on the flue-gas temperatures in space-heating mode and on the water content of the PFHRD, small or large amounts of thermal energy are stored in the PFHRD. And as a result, less energy is needed for the next DHW-tapping.
In the PrEN13203-7:2018 a method is proposed to measure the indirect contribution of space heating operation to the water heating efficiency of combination boilers. This PrEN recommends a temperature regime of 43/37 °C at continuous 30% part load as the representative EU-wide operating mode for space heating to determine this indirect contribution.

In the light of the results of various monitoring studies (Section 2) this seems a rather conservative representation. In the latest monitoring project from Fraunhofer (WPsmart im Bestand (still running)) for which the emitter systems have been prepped for high performing heat pump systems, the heating systems just achieve these temperature regimes, indicating that the EU-average will be higher than this. Also the boiler monitoring studies prove that condensate production is limited, indicating that heating system return temperatures are higher than 37 °C.

With regards to the 30% boiler load, for many dwellings the 30% part load will not be enough to reheat the house after a night setback. If the house is to be reheated in a comfortably short period, the full boiler load is often used, indicating that also the amount of flue gasses increase that can help preheating the dhw-storage in the PFHRD. But even based on the conservative PrEN13203-7 system temperatures and continuous 30% part loads, a PFHRD with small integrated cold water store (< 3 litres) can already lead to an additional water heating efficiency of up to 30%.

Of this 30% figure, a maximum of around 10% (one third) of this increase is achieved by elevating the efficiency in tapping mode. The remaining part is achieved by preheating of the cold water stored in the PFHRD during space heating mode. In dwellings with high heat loads and high system temperatures due to high $R_{HL:EC@40}$ ratios, higher numbers (than 20%) are possible, but then larger storage provisions, pumps and anti-legionella equipment will become necessary.

In general it can be stated that tertiary heat exchangers in all cases are a good solution for increasing the regular water heating efficiency. By integrating or adding a store to the PFHRD, cold sanitary water can be preheated, thus reducing the energy needed for the tapping pattern. The latter option is worthwhile in all (existing) dwellings with higher heat loads and higher system temperatures due to higher $R_{HL:EC@40}$ ratio. A 30% increase in
water heating efficiency corresponds to savings of around 3 to 4 kWh for tapping pattern L (around 1 – 1.2 MWh per dwelling per annum).

### 4.2 Turn-down ratio

Heat load of new-built average dwellings (100m²) is decreasing to below 5 kW. Comparable size existing dwellings (depending on their age and renovation levels) have heat loads between 5 – 10 kW.

![Figure 25. Distribution heat load English housing stock (in blue) and boiler output (in red) in kW](image)

These heat loads are calculated at the design temperature (for coldest day situations). On an average day heat loads might be half or even less when only one or two emitters are active, in which case only 1 or 2 kW of heat is needed.

The average instantaneous combination boilers have heating capacities of 24 kW or more and the standard turn down ratio is around 30% to maximum 15%, providing a minimum capacity of around 7 - 3.6 kW, meaning that the boiler is cycling on and off most of its operating time. Short cycling reduces the boiler’s thermal efficiency. It also increases emissions, increases maintenance and shortens equipment life. In the study 'Insitu monitoring of efficiencies of condensing boilers, by Gastec', the relation between boiler efficiency and operation time per cycle was measured. Figure 26 illustrates this relation between efficiency degradation and cycling.

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12 Cambridge Housing Model, DECC, 2011
Bennett\textsuperscript{13} reports in his study that short operational cycle times (≤ 3 min.) not only have a negative impact on the efficiency, they also influence the other emissions from the start-up sequence (CO, NO, and total hydrocarbon (THC), including methane). A study on start/stop emissions showed that these emissions increase significantly during cycling. With on cycles of around 150 seconds THC emissions increase from 0.8 mgC/kWh (in steady state) to 95.6 mgC/kWh during cycling. In other words, THC emissions are 120 times higher during start/stop operation compared to steady state.

\textsuperscript{13} Bennet, G., et al, Space heating operation of combination boilers in the UK: the case for addressing real world boiler performance, research supported by EPSRC and by Bosch Thermotechnology, 2018
70% of the investigated 338 boilers have on cycles of less than 8 minutes, of which around 25% have on-cycles of less than 4 minutes.

A higher turn-down ratio therefore seems a good solution. The downside however is that in order to achieve very high turn-down ratios of below 1:10, the amount of excess air that is added to the combustion process to maintain flame stability and ensure complete combustion is further increased. When fuel and oxygen from the air are in perfect balance - the combustion is said to be stoichiometric. The combustion efficiency increases with increased excess air - until the heat loss in the excess air is larger than the heat provided by more efficient combustion. From that point onwards, not only combustion efficiency drops, but also the dew point is reduced, leading to further reduction of condensation\textsuperscript{14}. 

\textsuperscript{14} Conners, D., Boiler modulation – is more better?, HPAC magazine, February 23, 2018.
A typical excess air level for natural gas-fired condensing boilers is around 30% (=5% \(O_2\)) at which combustion around 9% \(CO_2\) is produced. This correlates to a flue gas dewpoint of 54.4 °C. If additional excess air is added with the goal of improving the turndown ratio, leading to a \(CO_2\) concentration of around 6%, the flue gas dewpoint is further reduced to 46.6.

This means that high turn-down ratios are not by definition the best solution here. Beyond a certain point is has its downsides.

Another way of solving this problem of oversized boiler capacity due to DHW-demands as concluded in Task 3 of this study (resulting in increased emissions and maintenance and reduced efficiencies during space heating operation in low energy dwellings) might be more appropriate. For instance, solutions where the requested tapping pattern can be achieved through well designed combinations of small burners (<10 kW) and storage tanks, aka storage combis. The current product range of integrated condensing storage combis however is very limited and mainly consists of storage vessels (ranging from 40 to over 200 litres) combined with a condensing boiler with capacities ≥ 24 kW.
Only a few integrated condensing storage combis are offered on the market with a boiler capacity of below 14 kW. The Elco Thision S Plus Compact 12V100 for instance with a maximum heat output of 13,9 kW and a 100 litre storage vessel is one of them (see Figure 30). This boiler however still has a minimum heating capacity of 3,9 kW. This means that this boiler will still be cycling when installed in a dwelling with a heat load of 1 – 3 kW during 80% of the heating season.

Non integrated solutions, using a separate dhw-tank and a condensing boiler are an alternative option, which will of course lead to higher installation costs. But unfortunately also the number of single condensing boilers that is offered on the market with a heating capacity below 10-12 kW is rather limited.

![Figure 30. Elco THISION S PLUS Compact 13 V100 + Gas-Brennwert Heizung 3900012](image)

### 4.3 Combustion control

Declining local reserves of natural gas has led to more gas being imported. Since this supply of gas comes from different reserves around the world and is intended for a variety of markets, its composition can vary considerably, and thus imported natural gases generally have a different composition than the gases traditionally distributed. Furthermore, the drive towards a ‘greener’ natural gas infrastructure by the steady increase in the introduction of renewable gases into the gas grid adds to the diversity of composition. Hydrogen from power to gas, syngas (hydrogen/carbon monoxide mixtures) from biomass gasification and substantial fractions of carbon dioxide in fermentation gases add fuel components that do not occur in the normal range of natural gas compositions.

Various boiler manufacturers use flame ionisation as control principle for their premixed burners. This principle is based on the detection of the ions that are formed during hydrocarbon combustion. To a certain extent, these domestic gas boilers are able to burn gases of different composition with automatic adjustment of the excess air ratio. However, differences in the ability to control the combustion process were observed along the modulation range of the boiler. At lower thermal loads, the boilers are more fuel flexible. Furthermore, these adjustments of fuel/air-ratio do not always operate as expected, especially when hydrogen is added to natural gas. The concentration of ions for a hydrocarbon-hydrogen mixture is lower and may disturb the system. Also the flame position may change, indicating that the location of the electrode would have to change.

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Actual fuel gas analysis using a sensor (e.g. CO sensor or oxygen sensor), most probably are less sensitive to moderate hydrogen addition to natural gas, compared to the flame ionisation technology.

4.4 Hydrogen and boilers

Although the production and deployment of hydrogen in condensing boilers is—at this point in time—economically questionable, field trials are already being prepared in order to gain experience. Task 3 indicates that Power-To-Gas (P2G), i.e. carbon-free hydrogen from wind and solar driven electrolysis of water as opposed to not carbon-free hydrogen made from natural gas, is indispensable for a carbon-neutral society in 2050. Hydrogen can be an important chemical storage medium for the electricity generation, to be fed in gas-fired power plants on days with low wind and/or sunshine. But also at building level, fed through the existing gas grid, it can be an ideal complementary space heating fuel for electric heat pump hybrids on renewable energy supply-critical days. In fuel cells it can be a local alternative for electricity generation. Task 3 also mentions other electro fuels (‘e-fuels’) where electricity is used to create hydrogen-based synthetic fuel from water, such as ammonia (NH₃), e-methane, e-methanol, etc..

In a recent KIWA study, it was found that in terms of network- and boiler materials, there should be no specific problems with the use of hydrogen in networks. Relevant metals and most plastics (PE, PVC, NBR) have been tested in hydrogen-pilots and are found to be no more problematic than with natural gas. There is the potential problem of leakage: Hydrogen molecules are smaller than methane molecules and may leak sooner and—once leaked and in certain circumstances—may have a higher explosion risk. Pilot-projects—some for over 10 years in Denmark—did not reveal any specific problems, but especially for in-house gas lines KIWA recommends to wait for the extra certainty that ongoing pilot-projects with hydrogen-boilers may bring. In any case and like with natural gas, a non-sulphur odorant should be added to help detection. As regards pollution, KIWA reckons that it will take some years before the hydrogen gas will have washed away the nitrogen and other compounds from the pipe’s wall. For that reason, in case of fuel cell applications, KIWA believes a solid oxide fuel cell (SOFC) is a better solution than a PEM fuel cell as the latter is very sensitive to pollution that may be present in hydrogen transported through existing gas networks. In terms of costs for adapting the gas network to hydrogen, KIWA estimates 700 million euros mainly for the replacement of all the gas meters in the 7 million Dutch dwellings.

As a conventional combustion fuel in a boiler, hydrogen does not behave fundamentally different from natural gas (methane) but in today’s boilers it cannot be used directly: For the same capacity it requires three times the flowrate of natural gas. This means that when using 100% hydrogen in natural gas boiler, there is a significant risk of flash back

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16 Liquid at room temperature and thus easier and safer to store than hydrogen, which requires 200-350 bar tanks.
17 For a recent discussion of technologies and costs see Malins, C., What role is there for electrofuel technologies in European transport’s low carbon future?, Cerulogy report, November 2017.
18 KIWA, Toekomstbestendige gasdistributienetten, study for Netbeheer Nederland, Apeldoorn (NL), July 2018.
19 For networks there is still a small question mark whether POM—a technical plastic used in certain valves—will resist, because test are less than 2 years old.
20 Sulphur is detrimental for the use of hydrogen in fuel cells and should be avoided to keep that option open.
21 PEM=Proton Exchange Membrane
and damage to the burner. Also the current safety mechanism (ionisation) of a gas boiler cannot be used. But hydrogen combustion boilers have recently been developed by manufacturer REMEHA for pilot-projects in the Netherlands and it is not a large adaption from the existing gas boiler: You probably need a different pre-mix fan (larger capacity and or larger capacity range), an adaptation to the gas valve and/or nozzle and of course some adaptation of the CPU-electronics.

The above applies to 100% hydrogen. According to Swansea University, in concentrations of up to 30% hydrogen can be mixed in with natural gas to form Hydrogen Enhanced Natural Gas (HENG) and could then be used in a conventional gas boiler. For that reason, the study team believes it must be possible to make a natural gas boiler ‘hydrogen-ready’ (H$_2$-ready), meaning that with a few adaptations —e.g. a new nozzle, new setting of the electronics, dip-switches on the pre-mix fan— at a limited cost of e.g. €300 it will be possible to transform a gas boiler, solo or better in a hybrid, into a carbon-neutral hydrogen boiler. Likewise, it should not be too problematic to do the same for gas-fired fuel cells.

Given that there are only 31 years till 2050, it is recommended that ‘H$_2$-ready’ will become a mandatory Ecodesign requirement as soon as possible.

As is customary in these situations to give time for the market actors to prepare, it is proposed to first introduce the ‘H2-ready’ concept in the Energy Labelling, i.e. giving a 20% energy efficiency bonus leading to an A+ rating for a single boiler, and then after 2 years make ‘H$_2$-ready’ mandatory.

4.5 Pump efficiency and control

The efficiency of circulators has increased over the years. The induction motor is replaced by a permanent magnet motor and the terminal box contains a three phase frequency converter with motor control, system control and intelligent adaptive functions. The improved motor efficiency leads to savings on electricity consumption of almost 50%. The use of speed controlled pumps once again halves this already reduced power consumption. The Commission Regulation on Circulators (622/2012(EC)) played a crucial role in the uptake of these energy efficient circulators in heating systems and boilers.

Next challenge is to fully utilise the capabilities of these smart efficient circulators to optimise the supply and return temperature in relation to the minimal required feed temperature and in relation to the type of heat generator used (see also section 3.3.6).

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22 https://www.sciencedaily.com/releases/2018/06/180611133412.htm
4.6 Observations in relation to existing Regulations & test standards

Contrary to heat pumps (where the rated capacity is fitted to the space heating load), the capacity of condensing combination boilers is based on the maximum instantaneous DHW heat load, which on average is around 24 kW and more. The result is that boilers are cycling, leading to lower operating efficiencies, reduced lifetime expectancies and higher emissions. The existing Regulation holds no penalty whatsoever for this existing installation practice.

Point of discussion is whether to apply a similar approach as for heat pumps, where the generator capacity is initially fitted to the space heating load. For higher boiler-capacity to heat-load ratios, a penalty or correction factor for operating efficiency and emissions can be introduced. Alternatively the test conditions can be adjusted in such a way the boiler loads for space heating concurs with real-life heating demands (similar to tests for heat pumps).

The saving potential of storage-type PFHRDs is significant, i.e. around 25 – 30% of dhw-production of all condensing combination boilers! And also for flow-through PFHRDs the impact is not negligible (around 8-9%). It is therefore advised to consider the uptake of PFHRDs into the Regulations for combination boilers. A test method (PrEN 13207) is already in place and although proposed system temperatures and boiler loads are low with respect to the real-life operating modes - the standard could be used for assessing the improvements on the annual tapping efficiency. Reservations regarding the hygienic aspects of PFHRDs must be handled according to existing national legislation and guidelines concerning legionella prevention (storage tanks already have to comply). They may or may not lead to a reduction of the saving potential, as legionella prevention measures in some countries are not considered critical or depend on the storage volume (see Task 1).

Other types of combustion control (other than flame ionisation) that use flue gas sensors, are more flexible for future gas qualities, especially when hydrogen is to be mixed with the original gas qualities. To highlight this flexibility for future gas qualities, the combustion control type might be included in the technical fiche.
5 TECHNICAL IMPROVEMENTS ELECTRIC HEAT PUMPS

The ongoing R&D activities can be characterized by the continuous pursuit for improvement in the operating efficiency of electric heat pump combined with developments that are focussed on the reduction of costs for components, manufacturing and/or installation.

In the existing market the physical capability of replacing a small combination boiler by a heat pump often is a real challenge. Only a limited number of dwellings have a (large) boiler or heating room that offers enough space for a heat pump and a storage vessel. More dwellings however have their small combination boiler installed in either the attic, a small closet or in a kitchen cupboard in which case a simple replacement is no option. Alternative installation options are developed to solve these problems.

5.1 Improving operating efficiency

Efforts aimed at improving the operating efficiency of heat pumps can be divided into four main categories:

1. Decreasing temperature lift through modifications on the sink and/or source side
2. Modifying vapour compression cycle to increase COP
3. Reduction of power consumption HP system components
4. Smart controls

5.1.1 Decreasing the temperature lift

The COP of a heat pump is largely determined by the temperature difference between condenser (sink for heating) and evaporator (source), also known as temperature lift (see Figure 31).

Figure 31. Coefficient of performance and temperature difference
Since the source temperature (air temperature for ASHP or soil temperature for GSHP) is a given fact, an obvious way reduce the temperature lift of the heat pump for space heating applications, is to lower the design temperature of the emitter system. This can be done by lowering the RHL:EC@40 ratio, either by installing high capacity radiators (LT-radiators) and/or large radiation surfaces, or by reducing the heat load of the dwelling (improving insulation/airtightness/ventilation-system), or both. If by doing this, the design temperature can be reduced from 55 °C to 35 °C, the temperature lift of the heat pump system is reduced with 20 °C, resulting in COP-values that are increased by a factor of around 1.75 throughout the year!

5.1.2 Modifications in vapour compression cycle

Other ways to increase the operating efficiency may be based on modifications of the vapour compression and/or expansion-cycle of the heat pump, which help increase the temperature lifts and related COP-levels. These cycle-configurations are also used to design multi-temperature applications such as combination heat pumps, that can serve both the LT emitter system and the HT DHW-system.

By doing this, the possibilities for applications that needs higher temperature lifts (space heating in cold climates and DHW) can be extended.

Examples vapour compression cycle configurations:

a. Variable frequency driven compressor
b. Vapour Injection
c. Multiple compressor systems
d. Cascade systems
e. Expansion valve cycles
f. Ejector cycles
g. Separated gas coolers
h. Refrigerant and mixtures

Ad 5.1.2.a) Variable frequency driven compressor

Variable frequency driven (inverter) compressors, whereby the heat pump output is adjusted, are already broadly being applied. The inverter compressor uses a variable frequency drive to slow down or speed up the motor that rotates the compressor. This method varies refrigerant flow by actually changing the speed of the compressor. The turndown ratio depends on the system configuration and manufacturer. It modulates from 10% (depending on the compressor model) up to 100% at full capacity with a single inverter.

Compared to single or dual speed compressors, the inverter compressor achieves higher efficiency levels, results in more comfortable heating systems and is more versatile in its applications. In fact, variable speed compressors offer the highest part load efficiencies over any other modulation technology. This technology is specifically designed to be highly efficient in both residential and commercial applications.
Ad 5.1.2.b) Vapour Injection (VI)

A Vapour Injection (VI) cycle needs a suitable compressor. In a VI system, liquid from the condenser is expanded to a middle stage between the condensing and evaporating pressures, after phase separation, the vapour is injected to the compressor injection port, and the liquid is further expanded and goes to the evaporator. The refrigerant flow rate and pressure entering the intermediate compressor port can be easily controlled using thermostatic expansion valves. The VI cycles are able to reduce compressor discharge temperature effectively by directly injecting low enthalpy refrigerant vapour into the compressor compression cylinder. They also increase the evaporating and heating capacities due to the lower enthalpy liquid refrigerant entering the evaporator after the inter-stage phase separation.

VI cycles can be classified into two fundamental configurations: (a) Flash tank cycle and (b) Economizing heat exchanger cycle. Figure 32 shows the schematics of a VI cycle for each configuration.

![Figure 32. VI with flash tank cycle (left) and VI with economizing heat exchanger (right)](image)

In a VI cycle with flash tank, two-phase refrigerant is separated into saturated liquid and vapour by a flash tank after the first expansion. It has the advantage of feeding 100% of saturated vapour to the compressor injection port.

The two-stage cycle with economizing heat exchanger allows part of the liquid refrigerant at the condenser outlet to pass through an expansion valve before entering the economizer HX to further sub-cool the mainstream refrigerant coming from the condenser. The superheated intermediate pressure refrigerant leaving the economizer HX enters the intermediate compressor port. As a result, the separation with economizer HX will never be 100% as compared to the flash tank separation due to the limited surface area involved.

Proper designed VI-cycles can help increasing the heat output by around 30% while simultaneously increasing the COP with around 30%.

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Ad 5.1.2.c) Multiple compressor systems

Multiple compressor systems are systems that use two or more compressors in the same heat pump. Advantages of multiple compressor systems either in series or parallel are the simple compression power control for each compression stage. Challenges here are oil management regulation, oil migration and potentially high amounts of refrigerants in the system.

Various studies are performed investigating the potential of multi compressor heat pump systems.

Bertsch and Groll (2008) studied two-stage compression, as shown in Figure 33 (illustration on the left side), which used compressors in series, i.e. low stage compressor (booster compressor) and high stage compressor, and an inter-stage economizing heat exchanger.

The two-stage compression with inter-stage economizing usually runs both stages at low ambient temperatures and only runs the high stage at moderate ambient temperatures. This configuration effectively lowers the discharge temperature and maintains a good efficiency at low ambient temperatures. It has a larger capacity modulation potential than a single VI compressor, but the cycle is more complex and auxiliary oil management is needed to ensure adequate oil return to all compressors.

As described in Shen (2014), to augment heating capacity at low ambient temperatures, another option is to use multi-capacity compressor(s), i.e. a variable-speed (VS) compressor, or put two compressors in parallel (tandem compressors, as shown in Figure 33 on the right). Only partial capacity of the multi-capacity compressor(s) options is used to meet the building cooling load and provide heating at moderate temperatures (such as a VS compressor at low speed, or a single compressor in a tandem set). Full capacity operation is used to boost heating capacity at low ambient temperatures. For multi-capacity compressor(s) in a single-stage system (HP-scheme on the right hand side) the

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heat exchangers must be sized adequately for full capacity operation and the discharge temperature must be managed to avoid exceeding maximum limits.

The Oak Ridge National Laboratory theoretically assessed the performance of seven different types of improved cycle configurations, using the single speeds HP as reference (see Figure 34 below).

![Figure 34. Ratios of heating capacity @-25°C/8.3°C (blue), COP at 8.3°C (green) and the COP at -25°C (red) for various vapour compression cycle configurations](image)

The COP-values at -25 °C include (if applicable) the use of back-up electric resistance heating. So the higher the capacity ratio, the lower the electricity consumption of the back-up heater, and the higher the COP @ -25 °C.

Explanation types of configuration:

- **VI+EcHX**: Single speed compressor with VI, using an economising heat exchanger
- **VI+FlashTank**: Single speed compressor with VI, using a flash tank
- **VS_R@4500**: Variable speed compressor achieving rated heating capacity @ 4500 rpm
- **VS_R@3600**: Variable speed compressor achieving rated heating capacity @ 3600 rpm
- **VS_R@2700**: Variable speed compressor achieving rated heating capacity @ 2700 rpm
- **Tandem_R@low**: Single speed compressor achieving rated heating capacity with single compressor
- **TandemVI_R@low**: Single speed compressor with VI achieving rated heating capacity with single compressor

If the power ratio must be at least 75% (meaning that at -25 °C 75% of the installed heating capacity can be achieved), only the last four configurations comply. Both tandem configurations have a higher COP at -25 °C, because the variable speed compressor has an efficiency drop when running at top speed.

In a next step, two prototypes were selected for laboratory testing: I. Tandem_R@low (using two single speed compressors with an electronic expansion valve (EXV) for discharge temperature control), and II. TandemVI_R@low (using an equal tandem with VI-compressors with inter-stage economizing and discharge temperature control).

![Figure 35. HP with tandem single speed compressors + EXV](image1)

![Figure 36. HP with tandem single speed compressors + VI EcHX](image2)
**Test results**

Laboratory evaluation showed that the tandem, single-speed compressor ASHP system is able to achieve heating COP = 4.2 at 8.3 °C, COP = 2.9 at -8.3 °C, and 76% rated capacity and COP = 1.9 at -25 °C. The tandem, vapour-injection ASHP is able to reach heating COP = 4.4 at 8.3 °C, COP = 3.1 at -8.3 °C, and 88% rated capacity and COP = 2.0 at -25 °C.

**Ad 5.1.2.d)**

**Cascade systems**

A cascade system consists of two or more heat pump cycles, where the intermediate heat exchangers connect the cycles. A cascade system offers the possibility of having different a different working fluid in each cycle. Each fluid can be selected for optimum performance in the specified temperature range.

A cascade arrangement enables cycles to reach an extended operating range, e.g. high temperature lifts between heat source and sink ranging from -70 °C to +100 °C without any oil migration issues. On the other hand, the temperature difference in the cascade heat exchanger degrades the system performance in terms of energy efficiency.

![Figure 37. Schematics and p=h diagram of a two-stage cascade system](image)

**Ad 5.1.2.e and f)**

**Expansion valve and ejector cycles**

These cycle modifications are mainly used to increase temperature lifts and related operating efficiency for cooling purposes.

**Ad 5.1.2.g)**

**Separated gas coolers**

Heat pump cycles with separated gas cooler sections present a less commonly encountered system for multi-temperature purposes. It offers an elegant way for applications, such as single phase heating (e.g. water heating).

Separated gas coolers are heat exchangers in which the refrigerant remains in gas-phase, so no condensation occurs here. The advantage of using a separate desuperheater or gas cooler is the fact that higher temperatures can be achieved. Even if the available energy of gas desuperheating is lower than for condensing, it is possible to obtain hot water with a small heat exchanger because the "temperature pinch" is avoided. By utilizing only the sensible energy of the gas, the leaving water temperature can approach the inlet discharge gas temperature. In contrast, in a condenser the constant condensation energy makes it impossible to obtain water temperatures more than a few degrees above the condensation temperature.

### Ad 5.1.2.h) Refrigerant and mixtures

Refrigerants can be selected on the basis of their operating range. Ideally they have the following properties:

- Low boiling point and low freezing point.
- Low specific heat and high latent heat. Because high specific heat decreases the refrigerating effect per kg of refrigerant and high latent heat at low temperature increases the refrigerating effect per kg of refrigerant.
- The pressures required to be maintained in the evaporator and condenser should be low enough to reduce the material cost and must be positive to avoid leakage of air into the system.
- High critical pressure and temperature to avoid large power requirements.
- Low specific volume to reduce the size of the compressor.
- High thermal conductivity to reduce the area of heat transfer in evaporator and condenser.
- Non-flammable, non-explosive, non-toxic and non-corrosive.
- No negative effects, when any leak develops in the system.
- High miscibility with lubricating oil and it should not have reacting properly with lubricating oil in the temperature range of the system.
- COP in the working temperature range.
- Readily available and it must be cheap also.

For heat pumps that are intended to replace combination boilers in existing dwellings and need to deliver higher water temperatures, certain refrigerants can be selected that suit this purpose best. Special attention in this context is given to the transcritical vapour compression cycle using CO₂.

5.1.2.h – 1. CO₂ heat pumps

Transcritical CO₂ vapour compression cycle – the theory
The triple point of carbon dioxide is high and the critical point is low compared to other refrigerants. The chart in Figure 39 shows the triple point and the critical point on a phase diagram. The triple point occurs at 4.2 bar and -56.6 °C. Below this point there is no liquid phase.

The critical point occurs at 31 °C, which is below typical system condensing temperatures for part or all of the year, depending on the climate. Above the critical point the refrigerant is a transcritical fluid. There is no phase change when heat is removed from a transcritical fluid while it is above the critical pressure and temperature. In a refrigeration system transcritical R744 will not condense until the pressure has dropped below the critical pressure.

![Figure 39. R744/CO₂ phase diagram](https://emersonclimateconversations.com/2015/05/14/co2-as-a-refrigerant-properties-of-r744/)

No other commonly used refrigerant has such a low critical temperature (other refrigerants condense as heat is removed on the high side of the system). Many R744 (CO₂) systems operate above the critical point some or all of the time. This is not a problem; the system merely works differently and is designed with these needs in mind.

The pressure enthalpy chart in Figure 40 shows an example of a simple R744 system operating subcritically at a low ambient temperature and transcritically at a higher ambient temperature. The chart shows that the cooling capacity at the evaporator is
significantly less for transcritical operation. It is important that appropriate control of the high side (gas cooler) pressure is used to optimize the cooling capacity and efficiency when transcritical. For example, increasing the high side pressure will increase the cooling capacity when operating above the critical point. High pressure (>100 bar) is therefore a necessity for transcritical vapour compression cycles with high temperature lifts.

![Figure 40. Pressure enthalpy chart R744](image)

At SINTEF\(^{28}\) an evaluation of different heat pump water heater systems was made, comparing the following HPWH-systems:

1. Heat pump with condenser and desuperheater
2. Heat pump with subcooler, condenser and desuperheater
3. Heat pump with suction gas heat exchanger, condenser and desuperheater
4. R744 (CO\(_2\)) heat pump with a single counter-flow gas cooler

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\(^{28}\) Stene, J., High-efficiency CO2 heat pump water heater systems for residential and non-residential buildings, SINTEF Energy Research, Trondheim, Norway.
HPWH systems no. 1, 2 and 3 were simulated with both R134a and R290 (propane), since these working fluids have a sufficiently high condensation temperature (60-70 °C) when using standard components with 25 bar pressure rating. The CO2 HPWH was equipped with a 60 bar plate heat exchanger as evaporator (SWEP), a reciprocating compressor with 150 bar pressure rating (Bitzer), a 140 bar counter-flow plate heat exchanger as gas cooler (SWEP), a counter-flow tube-in-tube suction gas heat exchanger, an automatic expansion valve (back-pressure valve) and a low-pressure receiver (LPR). The expansion valve and the LPR were used for optimization of the pressure in the gas cooler at varying operating conditions in order to maximize the COP.

The COP was calculated for the different HPWH systems as a function of the evaporation temperature, t0 (Hjerkinn, 2007). In the calculations it was assumed a total UA value of 2100 W/K for both the condenser/heat exchanger combinations and the CO2 gas cooler, 5 K superheated vapour from the evaporator, 5 °C inlet city water temperature and 70 °C DHW temperature. The overall isentropic efficiencies for the compressors were calculated on the basis on typical efficiency curves from laboratory measurements.

The most energy efficient HPWH was System 4, the CO2 system, which in average achieved about 20% higher COP than System 2 and 3. This was mainly due to higher compressor efficiency and the excellent temperature fit in the gas cooler between the CO2 and the water, which minimized the average CO2 temperature during heat rejection. In addition to the high energy efficiency for the CO2 HPWH, CO2 has the advantage of being a non-flammable, nontoxic and environmentally friendly fluid with zero GWP factor when used as a working fluid.

**CO2 heat pump for DHW in practice**

DHW heat pumps using CO2 are on the market since 2001. It is estimated that in Japan alone over 500,000 of these DHW heat pumps are sold. In that sense CO2 DHW HP is a relatively mature technology and have gained popularity over the past decade within the water heating industry because of their ability to deliver significantly more heat for the same amount of electricity compared to traditional electric storage water heaters (SWH). Today most of the CO2 heat pumps are designed for Domestic Water Heating (DWH), with Japanese companies like Panasonic, Daikin, DENSO, Sanden, Itomic and Mitsubishi some of the most active promoters of the technology. Whilst the Japanese market can be considered mature, CO2 heat pumps represent a small market in Europe. However, more and more Japanese companies are entering the European market, redesigning their CO2
heat pumps to fit the European way of life, climate, housing structures as well as EU and national energy and safety standards. Local CO₂ heat pump manufacturers are also emerging, with companies like Stiebel, enEX, ICS, Thermea, Kylma, CTC, JCA, and Viessmann adding CO₂ heat pumps to their product ranges. Outside the domestic market, commercial real estate owners see CO₂ heat pumps as a promising, high performance option for high temperature sanitary hot water for hotels, restaurants, hospitals, schools and other public buildings.

Example: Sanden offers the ‘aquaeco2’, a 4.5 kW monobloc CO₂ heat pump unit (product is based on the EcoCute HP from Japan), that can be used for domestic hot water only, or for combined operation of dhw and space heating. Achieved efficiencies and capacities in relation to outdoor ambient temperatures are given in the figure below.

![Figure 42. Efficiencies and capacity of the aquaeco2 heat pump in relation to outdoor ambient temperature](image)

Given the inlet water temperature (green line) and the DHW-system configuration presented in Figure 42, the following COPs are achieved:

- at A2/W60 °C (the average outdoor temperature for cold climate) the COP is around 3.0;
- at A7/W60 °C (average climate) the COP is around 3.5,
- and for a warmer climate (A14/W60 °C) the COP is around 3.8.

Higher inlet water temperatures will decrease the COP indicating that it is very important to have a good stratification in the hot water tank and use the coldest water possible.

---

These DHW CO₂ heat pumps are expensive when compared to instantaneous gas water heaters, combination boilers or electric storage tanks. In the US the Oak Ridge National Laboratory\textsuperscript{30} investigated some options to reduce the cost by replacing the separate and expensive water fouling prone plate heat exchanger by a low-cost wrap-around gas cooler (an inexpensive tube wrapped around the storage tank). Prototypes have been built and tested to further optimise the performance of these prototypes. Nawaz, K. et al (2017)\textsuperscript{31} found that such alterations also reduce efficiency levels of the DHW heat pump. To the extent known, this R&D work has not (yet) led to more economical DHW CO₂ heat pumps.

**CO₂ Heat Pump for DHW & Space Heating**

Clearly, CO₂ heat pumps can be favourable for DHW-applications. For space heating purposes (requiring a lower temperature lift on the sink side), heat pumps using other refrigerants can achieve higher COP-values. Nevertheless, integrated solutions with CO₂ for space heating and DHW show promising results, as well, in particular with high DHW fractions, which may occur in ultra-low energy houses. Development potentials are seen both at the component level and concerning the system integration. It is still difficult to get components for CO₂ heat pumps, in particular compressors for the small capacity range. Moreover, small compressors shall be further developed regarding the performance data, which is essential for a good system performance. Ejector technology could further enhance the performance \textsuperscript{32}.


\textsuperscript{32} IEA HPP Annex 32, Economical heating and cooling systems for low energy houses, 2011, Carsten Wemhoener (editor).
5.1.2.h – 2. Refrigerant mixtures

The use of mixtures of refrigerants with the aim of increasing the COP and heating capacity at lower temperatures by taking advantage of the thermal glide, is an option that has received limited attention so far. Goal of refrigerant-mixtures is to obtain a performance improvement by realising heat exchange with temperature variations at the refrigerant side, rather than at constant temperature as is the case in conventional cycles. At CanmetENERGY Canada, a study was done to find refrigerant-mixture that enable good heating performance at low outdoor air temperatures.

The properties of the group of selected refrigerants investigated in this study are presented in Table 10 (the properties of new refrigerants are obtained by REFPROP (Lemmon E.W., 2013)). R-410A is used as a reference refrigerant in this study.

Table 10. Properties of a group of selected refrigerants

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>GWP</th>
<th>Safety group</th>
<th>Critical temp. (°C)</th>
<th>Critical pressure (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-410A</td>
<td>2088</td>
<td>A1</td>
<td>70.17</td>
<td>4.77</td>
</tr>
<tr>
<td>Propane</td>
<td>20</td>
<td>A3</td>
<td>96.74</td>
<td>4.25</td>
</tr>
<tr>
<td>R-32/CO₂ (80/20)</td>
<td>540</td>
<td>-</td>
<td>70.83</td>
<td>6.61</td>
</tr>
<tr>
<td>Propane/R-32 (95/5)</td>
<td>52</td>
<td>-</td>
<td>94.69</td>
<td>4.47</td>
</tr>
<tr>
<td>Propane/R-32 (10/90)</td>
<td>610</td>
<td>-</td>
<td>70.89</td>
<td>5.43</td>
</tr>
<tr>
<td>Propane/CO₂ (95/5)</td>
<td>20</td>
<td>-</td>
<td>95.40</td>
<td>4.63</td>
</tr>
<tr>
<td>Butane/R-32/R-125 (10/45/45)</td>
<td>1880</td>
<td>-</td>
<td>77.88</td>
<td>5.26</td>
</tr>
<tr>
<td>Propane/R-32/CO₂ (10/80/10)</td>
<td>542</td>
<td>-</td>
<td>69.77</td>
<td>6.08</td>
</tr>
<tr>
<td>Propane/R-32/R-125 (20/50/30)</td>
<td>1391</td>
<td>-</td>
<td>65.76</td>
<td>4.99</td>
</tr>
</tbody>
</table>

Through simulations the selected refrigerants are compared in a heat pump with counter-flow heat exchangers. The cycle components are designed for the best performance of R-410A and in all comparisons, the size of the compressor, evaporator and condenser are kept the same.

Results (see Figure 44 and Figure 45):

Propane and its mixtures have the highest COP but also the lowest heating capacity. Amongst all the mixtures studied, the mixture of R-32/CO₂ (80/20) has the highest heating capacity (approximately 30% higher than R-410A). The COP of this mixture is slightly smaller (less than 2%) than that of R-410A. By mixing these two refrigerants, not only will the flammability effect of R-32 be reduced, but also the high pressure effect of CO₂ will be minimized. This mixture is non-toxic with a GWP of approximately 540 which is 25% of the GWP of R-410A.
Figure 44. Comparison of heating capacity of a group of refrigerant mixtures as function of outside air temperature.

Figure 45. Comparison COP of a group of refrigerant mixtures as function of outside air temperatures.

Figure 46 and Figure 47 below present, respectively, the COP and heating capacity of two of these refrigerant mixtures as a function of the outside air temperature for a wide range of outside air temperatures. The mixture of R-32/CO₂ (80/20) has a much better COP, which is approximately 17% better than Propane/CO₂ (50/50). The heating capacity of Propane/CO₂ (50/50) is only slightly better (4.5%) than R-32/CO₂ (80/20). Therefore,
it can be concluded for residential purposes R-32/CO₂ (80/20) has a better performance than most of the other refrigerants studied.

![Figure 46. Comparison COP of two promising refrigerant mixtures as function of outside air temperature](image1)

![Figure 47. Comparison of heating capacity of two promising refrigerant mixtures as function of outside air temperature](image2)
5.1.3 Reducing power consumption system components

The third principle for improving operating efficiencies of electric heat pumps deals with reducing the auxiliary power consumption of all system components and complementary functions (back-up heaters, defrosting, etc.).

\[
SPF_{HA} = \frac{\text{Heat provided (HP + electric backup)}}{\text{Electricity (HP + source fan or pump + sink pump + backup)}}
\]

Correct sizing of the heat pump is crucial in this context. If this is not done correctly, the backup heater needs to come in to provide the requested heat, leading to lower operating efficiencies than calculated on the energy label. Since investment costs increase with the heat output, consumers and installers tend to opt for the lower heat outputs without fully recognizing that resulting reduced operating efficiencies will lead to longer pay-back periods or no pay back at all.

The power consumption of the fan or pump that is used on the sink side of the heat pump will influence the operating efficiency, indicating that the flow resistance of the heat exchanger on the sink side and the efficiency of pump or fan itself, are important parameters. Optimising these parameters – also under part load conditions – will lead to higher operating efficiencies. The same goes for the circulator that is used on the sink side. The higher circulator efficiencies, also under part-load, the higher the operating efficiency of the system.

Finally the defrosting cycle is an important parameter influencing the annual operating efficiency of a heat pump system. Depending on the situation, the energy needed for defrost cycles may reach over 20% of the power input of the heat pump system, indicating that its influence on the seasonal performance can be significant.

The most basic and commonly used defrost control strategy is the timed defrost. The evaporators are defrosted below a certain outdoor temperature, according to a predefined defrost schedule in which a timer both initiates and terminates the defrost cycle. Since this strategy is not based on the actual amount of frost present on the evaporator, it is possible that the defrost cycle is either insufficient in duration such that the frost is not completely removed during the defrost cycle or it is excessive in duration such that significantly more heating is supplied than that which is required to remove the frost.

The efficiency of the timed defrost strategy may be improved by using a temperature-controlled termination. In this technique, the defrost cycle is terminated when the evaporator temperature has reached a predetermined temperature, or when the defrost cycle has reached a predetermined duration, whichever occurs first. The temperature termination allows the defrost cycle to end earlier than that which would have occurred with only a timer. However, since the evaporator temperature is only used to terminate a defrost cycle, the possibility still exists that a defrost cycle will be initiated when none is required.

Although not very energy efficient, the timed initiation/termination defrost control strategy is often used – especially in older heat pumps – due to its low cost, simplicity, robustness and ease of maintenance.
However, various technologies are researched and being developed to minimise the energy needed for the defrost cycle by:

- Applying demand driven defrost techniques using either electric resistance heaters or reverse cycle defrosting (starting at the right moment and applying the right duration of the defrost cycle); reverse cycle defrosting using PCM is also being investigated and may lead to further reduction of defrosting time.
- Applying surface treatment of the evaporator (super hydrophobic surfaces yields lower frost thickness and frost mass leading to shorter defrost cycles).
- Applying a high voltage electric field or magnetic field to reduce frost formation of the he-surfaces.
- Applying oscillation and ultrasonic vibration (to remove frozen water droplets).
- Applying hybrid defrosting techniques combining the previous.

### 5.1.4 Smart controls

Next to frequency controls (adjusting compressor speeds to react on head load variations, see section 5.1.2 Variable frequency driven compressor), the following control parameters are important for achieving high operating efficiencies with electric heat pumps:

- Programming Heating Schedule (for DHW and Space Heating operation)
- Smart Supply Temperature Control
- Defrosting Initiation and Termination Control Strategies
- Smart Grid Controls

#### 5.1.4.a Programming Heating Schedule

Combination heat pump systems (producing both DHW and space heating) are the preferred option in the market. Contrary to fuel combination boilers, the heat pump output is matched around the heat load of the dwelling, meaning that maximum output is considerably lower than fuel combination boilers (for which the output is often matched to the instantaneous water heating load). DHW storage tanks are needed to be able to comply with DHW-patterns and because the loading of a DHW-storage tank takes relatively long time (due to smaller hp-capacities), it is extremely important to select the right schedules for heating the storage tank. To prevent comfort problems on the space heating side it is best to schedule the heating of the DHW-storage tank during the temperature set-back periods. But also to prevent the back-up heater from stepping in, and to prevent supply temperatures from running too high, it is important to apply well selected heating schedules.

Options are to implement a default heating schedule that can be adjusted by the installer if necessary or by the user/consumer. Better of course is to make this a self-learning and self-optimising heating schedule program that will have better fit with the various occupancy schedules of the inhabitant and may lead to higher overall efficiencies.
5.1.4.b
**Smart Supply Temperature Control**

Smart supply temperature control is crucial for achieving higher operating efficiencies with heat pumps. Lower supply temperatures will lead to higher COPs due to the reduced temperature lift within the vapour compressions cycle. Current control practices however, are not focussed on minimising supply temperatures wherever possible, but on achieving the requested room temperatures by applying a failsafe and robust heating curve.

As explained in Section 3.3.6 intelligent distribution manifolds can help maximising the operating efficiencies of the heat pump system, and provide feedback on the efficacy of the emitter system. Such solutions for controlling flow and system temperatures are considered extremely important to achieve the full saving potential of the electric heat pumps.

5.1.4.c
**Defrosting Initiation and Termination Control Strategies**

(See section 5.1.3)

5.1.4.d
**Smart Grid Controls**

Countries that use fuel boilers for space heating will generally have an electricity grid that is not prepared for a large scale switch to electric heating with heat pumps. As the Figure below shows, peak heat demand is very much higher than peak electricity demand (about 6 times as high) and heat demand is much more variable across the year. Meeting heat demand directly by electricity would impose enormous strains on the grid unless further changes were made to the system. As far as peak demand is concerned, the difficulties could be mitigated by a combination measures such as:

- Severe improvement of insulation values of dwellings
- Improvement of airtightness
- Smart ventilation with heat recovery
- Applying heat pumps instead of direct electric heating

By doing this, the extra demand needing to be met might be reduced to perhaps twice current electricity peak levels. But even then it would be a major challenge.

In addition to this, renewable power is introduced into the grid on a non-structural basis. Depending on wind and solar radiation the electricity production may vary. Without any buffer capacity this would imply that fuel fired powered power plants would need to be turned down to partial loads, which is not always possible and in any case difficult.

*Smart Grid Ready* appliances with bidirectional IP-based communication and demand side management type of steering algorithms for charging the storage tanks (that are generally incorporated in the heat pump system), can help reducing the problem and may also help operating the heat pump unit in a more economical way (switch on when prices are low).
Integration of a heat pump system into a smart grid will change the way heat pump systems are designed and controlled. A heat pump system without or with a small thermal storage will contribute little to the problems described. Smart Grid Ready heat pump systems are therefore provided with large storage tanks, preferable one for dhw and another for space heating purposes.

Several angles can be selected where the control strategy is concerned:\(^{33}\):

1. **Focused on the grid**
   Heat pump operation is aimed at providing ancillary services to the grid to allow a stable and cost efficient operation of the electric grid. Such services include voltage control, congestion management and spinning and non-spinning reserve (to balance generation and demand and ensure a stable frequency).

2. **Focused on renewable energy**
   Heat pump operation is aimed at an increased utilization rate of renewable electricity, a reduction of feed in peaks and smoothing of the residual load curve. Special attention is paid to the integration of wind and PV electricity as the two main fast growing sources of fluctuating renewable electricity generation.

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3. **Focused on price**

Heat pump operation is aimed at charging the thermal stores while the electricity prices are the lowest. Prices are used to influence the behaviour concerning electricity consumption in order to prevent grid problems.

![Diagram showing control strategies for smart grid prepared heat pump systems](image)

**Figure 49.** Control strategies for smart grid prepared heat pump systems

In order to successfully integrate heat pumps in a smart grid, it is critical to have a holistic view on the energy systems affected. Analysing the smart grid barely from the electric perspective will lead to missing how heat pump system efficiency and indoor comfort will be affected by potential changes in heat pump control. Oppositely, if the focus is only on heat pump efficiency without considering the characteristics and expectations needed in the future electric system, this will lead to considerable costs and waste of resources in the power system. Hence, a holistic perspective is required to analyse, design and operate the future energy system.

Studies show that heat pumps can be used to ease the transition towards a renewable interconnected energy system. It is highlighted that altered heat pump operation might come at the cost of efficiency. High storage temperatures, operation far from optimum compressor speed or frequent switching of the heat pump units. The potential flexibility of heat pump systems, which should be considered already in the planning phase, is mainly dependent on the building physics and the resulting thermal demand profile, heat pump and storage type and size with respect to the demand, and the control strategy applied.
5.2 Reducing costs

Since heat pump installations require a considerable investment, especially when compared to a like-for-like replacement of a gas fired combination boiler, manufacturers try to reduce the component- and manufacturing costs wherever possible. But because many of the heat pump components are already highly optimised mass produced OEM products (also used in air conditioners and refrigeration products) the cost reduction potential is limited. More cost savings can be achieved by further integration of the various components and by minimising the amount of work that is requested from the installer. Depending on the location and the type of the heat pump, various types of product integration may help reducing the installation costs.

Rough estimates for the total cost for an ASHP-installation (including installation costs and VAT, excluding the heat emitter system) vary from €4000 to €8000 (depending on the heat load of the dwelling and brand/type ASHP). Around 50% of these costs relates to non-equipment costs (mainly labour).

For GSHP these total costs (including installation costs and VAT, but excluding the heat emitter system) are estimated to be around €7000 to €15000, depending on heat load and brand/type ASHP. Around 35 to 60% of these costs relates to the equipment and installation of the ground coil (ground source heat exchanger). Also here a large share of the costs (>50%) relates to non-equipment costs (labour and the use of earth-moving and/or drilling machinery).

Reducing labour (installation) costs is therefore considered a priority for both types of heat pumps.

5.2.1 Replacement market

In cases a boiler or combination boiler is to be replaced by a heat pump, the location of (combination) boiler largely determines the feasibility and costs of such an operation.

Kitchen cupboard installation

When the existing (combination) boiler is mounted into a kitchen cupboard, replacement by a heat pump is often not possible, unless a new installation location can be found somewhere in the dwelling that is big enough to accommodate a large storage vessel and the condenser. A part of the existing pipework needs to be redone, the ducting for exhaust gas and supply air removed and related openings in wall and/or roof need to closed. This all means that – in addition to the already higher heat pump / storage vessel costs- installation costs will be a considerable part of the total investment.

All in all, this means that in most cases, a single electric heat pump is not an option. A hybrid solution with a compact heat pump and a gas boiler integrated into one compact unit will have better opportunities in this segment. Such a hybrid solution will also help to prevent that the electricity grid becomes overloaded and further capitalizes on the already existing gas grid.

Attic installations

Dwellings that have their existing (combination) boiler installed in the attic, may have more opportunity to replace it by an all-electric heat pump. Depending on the size of the attic, the space that is needed might be sufficient, although space for an upright storage
tank might be limited. In such cases a horizontal tank or two smaller tanks in series might be used.

Installation costs can be further reduced by integrating all requested components in one prefabricated unit (monobloc) that can directly be mounted into the roof (see Figure 50).

![Figure 50. Hydrotop, Integrated heat pump for sloping roofs (source: Dutch Heat Pump Solutions)](image)

The outdoor unit (evaporator with fan, air supply and exhaust openings and its protection against rain and hail), are integrated in the sloping roof and ready to use. The insulation of the roof is continued on the outside of the integrated rooftop unit (see Figure 51), ensuring that the insulation of the building shell is not interrupted. Pipework near the heat pump will need some adjustments, but the pipework to the emitter system and dhw tapping points can remain.

![Figure 51. Heat pump pipework](image)

**Closet installations**
In case the (combination) boiler is mounted in a closet, replacement by a heat pump is possible, as long as the inside dimensions of the closet are large enough (floor surface preferably larger than 70 x 60 cm) to hold a storage tank. In such cases the storage tank and possibly also the inside-unit of the heat pump can be mounted inside the closet. Serviceability can become an issue here. In addition to this a suitable location for the outdoor unit must be found where the requirements regarding outdoor noise production are limited.
Installation might be easier and cheaper when the indoor- and outdoor unit of the heat pump are combined in a Monobloc, in which case a certified air-conditioning installer is not required and heating installer can do the complete job. The distance between storage tank and Monobloc is preferably limited and the supply and return pipes need to be very well insulated.

Pipework to the emitter system and dhw tapping points can remain. Ducting for exhaust gas and supply air and related openings in wall and/or roof need to be removed and closed.

**Utility room installations**

Residential buildings that use a utility room (e.g. multi-family housing) generally provide the best options for replacing a (combination) gas or oil boiler with a (combination) electric heat pump. Also in this case, installation might be easier and cheaper when the indoor- and outdoor unit of the heat pump are combined in a Monobloc, ensuring that a certified air-conditioning installer is not needed and heating installer can do the complete job. Conditions are that the distance between storage tank and Monobloc is limited and supply and return pipes are very well insulated.

### 5.2.2 New built and renovation market

When dwellings are newly built or completely renovated, the specific requirements for heat pump installations may be taken into account from the early start.

For renovation projects, fully integrated ‘add-on HVAC-modules’ might be an economical solution to create the extra space that is needed for the installation components, while saving on installation costs. For apartment buildings that extra space might be found in dedicated façade elements in which extra space is created for the heat pump components (see Figure 52).
The noise coming from the evaporator can be reduced by applying a silencer (bottom left illustration). Provided the silencers are properly designed, its effect on the efficiency of the heat pump will be little to nothing.

For new built dwellings, all the necessary means for applying heat pump systems can already be included in the design phase of buildings. The best options for applying ground source heat pump are in the new built sector, because here the necessary mechanical excavation for inserting the ground couples heat exchanger can be done before (or during) the actual construction of the building and before gardens are designed and created.

5.2.3 ASHP

Air Source Heat Pumps are in most cases the preferred option because they are the cheapest and easiest heat pump to install, especially when monobloc solutions can be used. Important drawback of the ASHP however is the fact that the source temperature is the lowest (air temperature in wintertime), when the highest heat output is needed (high CH-system temperature in wintertime). Since the COP of a heat pump is directly related to the temperature lift \((T_{\text{condenser}} - T_{\text{evaporator}})\), this leads to lower SCOP-values, this could compromise the operational- efficiency and costs of the system. It is therefore of the utmost importance that, in colder climate zones, sufficient attention is given to the overall system design and subsequent evaluation of the overall system efficiency. If electricity is the only energy source, the ASHP will always be a better solution compared to direct electric heating. But where fuels are available, SCOP values for the electric heat pump system need to be above 2.4 to outrank high efficient heating systems based on gas or oil. This goes for ASHP that are used for space heating, but even more so for DHW-applications, where sink temperatures of around 55 °C are requested.

It is expected that with installation specific integration and optimisation (attic, closet, monobloc, prefab ad-on modules) labour costs for installation can be further reduced with around €500 to maximum €1500.

5.2.4 GSHP

Ground Source Heat Pumps (GSHP) generally operate at higher efficiencies during colder periods (air temperatures below 10 °C) due to a smaller temperature lift, created by the higher source temperature (ground temperature at 6 m depth is around 10 °C). With a correctly dimensioned emitter system using the right controls, GSHP will deliver higher operating efficiencies, leading to lower operational costs.

The need for a ground source heat exchanger however, reduces the possibilities for applying GSHPs. It also has a significant impact on the total acquisition and installation costs. Costs for ground collectors may vary from 250 – 400 €/kW for horizontal collectors to 550-750 €/kW for vertical (drilled) collectors.

The options for cost reduction by further integration of components is limited compared the ASHP that allows for a relocation of the source-side heat exchanger. For GSHP this is not the case. Additional saving on acquisition and installation costs can therefore mainly be expected from increased installation efficiency, economy of scale and reduced margins in the supply chain.
5.3 Observations in relation to existing Regulations & test standards

The test standard (EN 14825) distinguishes four design system temperatures (or temperature applications) for the heat pump (High = 65 °C, Medium = 55 °C, Intermediate = 45 °C and Low = 35 °C) and—depending on the climate region (Average, Colder or Warmer) and heat pump type—three to five source temperatures at which the heat pump can be tested. The Regulation states that the ‘Medium temperature application (55 °C)’ and the ‘Average Climate’ are mandatory. The manufacturer may select additional temperature application(s) and climate regions for which the heat pump is to be tested.

For each design system temperature _ climate combination, a set of tests under part-load conditions needs to be performed to determine the Declared Capacity (DC) and the COP for the various test points. For the sink side (indoor heat exchanger) these test points mimic the reduced heat load over the heating season, by gradually lowering the system temperatures and capacity of the heat pump. For the source side (outdoor heat exchanger) the test temperatures mimic the outdoor temperatures over the heating season. Additional test points are requested for the bivalent temperature (Tbiv) and the operating limit temperature (TOL). For all the test points the heating capacity and COP are determined. The COP-values coming from these tests are used to calculate the SCOP (Seasonal Coefficient of Performance).

These comprehensive test methods and calculations for heat pumps seem well equipped to deal with future improvements that can be expected in heat pump performance. In that sense they are adequate, future proof and more representative than those for fuel boilers.

Because of costs, manufacturers limit the number of design system temperature _ climate combinations to one (55 °C/Average) or maximum two if one wants to promote a specific application of the heat pump, such as a low temperature heat pump, in which case an extra combination is added (e.g. 35 °C/Average). The Regulation dictates that the seasonal heating efficiency ηs,h is determined at a predefined system design temperature of 55 °C (and optionally 35 °C) for an average climate, and presented as the seasonal space heating efficiency e.g. 126% @55 °C and (optionally) e.g. 174% @35 °C, on the basis of which an energy efficiency class is allocated. This approach has a number of drawbacks:

Topics for discussion

1. The current approach suggests that at 55 °C, the averaged system supply temperature over the heating season (following the BIN-method for average climate) is 38 °C, which in real life - due to limited emitter capacities, suboptimal supply temperature controls and suboptimal pump controls - is not the case. Monitoring studies show, that in optimised existing dwellings average supply temperature are above 43 °C. Furthermore this approach gives the impression that the design system temperature can be determined by selecting the right heat pump type (low or non-low temperature heat pump. In any case, it does not clearly signal that the real-life seasonal heating efficiency largely depend upon the heat-load/emitter-capacity ratio and the applied supply-temperature controls and pump controls; these parameters determine whether the seasonal space heating efficiency using the same heat pump is either 85% or 174%.

2. Different energy efficiency classes are used for non-low-temperature heat pumps and for low-temperature heat pumps. The seasonal space heating efficiency in Class A++ can therefore range from 125% (non-low-temp hp) to 175% for a low-temperature hp. This is a huge difference (50 percent points) within one Class. Looking only at the energy label, there is no real urge to go for the low-temperature heat pump, it’s all Class A++.

It is advised to use the same classification for the different heat pump types and to make them smaller where possible (preferably smaller than 25%).
3. Not the heat pump but the HL/EC-Ratio of the dwelling determines the design system supply temperature. It is proposed to discuss the options for an approach whereby the seasonal space heating efficiency is explicitly linked with values for the HL/EC-Ratio (instead of temperature application classes) and where system designers and installers are encouraged to assess this actual HL/EC-Ratio in their projects. By doing this, the key-parameter is clearly identified and turned into an understandable feature that can be influenced, thus increasing the real-life seasonal system efficiency. Assessment tools and measurement tools can be developed to assist with the determination of the HL/EC-Ratio.

4. Heating schedule controls are currently not incorporated in the control classes mentioned in the Regulation for (combination) heat pumps 813/2013 and 811/2013. Other topics for discussion regarding the existing control-classes (Class I to VIII) are:
   - feed temperature controls have a larger effect on the operating efficiency than the currently assumed default of 3%, especially in combination with heat pumps;
   - weather compensation controls generally result in higher average system temperatures than room thermostat controlled systems (provided the reference room has correct sized emitter capacity);
   - flowrate control is not incorporated in the various control classes, although they have significant influence on the operating efficiency of the heat generator;
   - the effect of modulating thermostats and modulating heat generators is limited when minimal heat output of the generator is larger than the instantaneous heat load of the dwelling.

5. Testing for all four design system temperatures and all three climates would require around (4 * 3 * 6 = ) 72 test points. With these 72 test points the complete map with COP / heat-output / system temperature -values for a specific heat pump is known, allowing for a mathematical assessment of the seasonal heating efficiency with every given HL/EC-Ratio. Unfortunately not all of these tests are performed because related test costs are too high. It is proposed to discuss and investigate to what extend the number of test-points can be reduced to achieve such a representative efficiency map over the full range of design system temperatures, climate zones, heat outputs.
# 6 DEVELOPMENTS REGARDING HYBRID HEAT GENERATORS

## 6.1 Introduction

Both the Delegated Regulations (EU) No 811/2013 and (EU) No 813/2013 do not use the term *hybrid*, but speak of a non-preferential supplementary heater of any kind, that can be used together with any kind of preferential heater.

In this chapter the term *hybrid* is specifically used for heat pumps that are combined with a fuel (combination) boiler. It is expected that these specific types of hybrid heat generators may be able to solve several of the issues that are currently encountered in the replacement market.

Current Regulation No 811/2013 allows for an assessment of the seasonal space heating efficiency of such *hybrid* heat generator solutions under the Package Label (see Regulation (EU) No 811/2013, Annex IV, Figure 3)

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<table>
<thead>
<tr>
<th>Seasonal space heating energy efficiency of heat pump</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature control</td>
<td>2</td>
</tr>
<tr>
<td>From fiche of temperature control</td>
<td>3</td>
</tr>
<tr>
<td>Class I = 1 %, Class II = 2 %, Class III = 1.5 %, Class IV = 2 %, Class V = 3 %, Class VI = 4 %, Class VII = 3.5 %, Class VII = 5 %</td>
<td></td>
</tr>
<tr>
<td>Supplementary boiler</td>
<td>3</td>
</tr>
<tr>
<td>From fiche of boiler</td>
<td>3</td>
</tr>
<tr>
<td>Seasonal space heating energy efficiency (in %)</td>
<td>3</td>
</tr>
<tr>
<td>Solar contribution</td>
<td>4</td>
</tr>
<tr>
<td>From fiche of solar device</td>
<td>4</td>
</tr>
<tr>
<td>Collector size (in m²)</td>
<td>4</td>
</tr>
<tr>
<td>Tank volume (in m³)</td>
<td>4</td>
</tr>
<tr>
<td>Collector efficiency (in %)</td>
<td>4</td>
</tr>
<tr>
<td>Tank rating</td>
<td>4</td>
</tr>
<tr>
<td>Tank rating A¹ = 0.95, A₂ = 0.91, Q = 0.95, D = 0.90, D-G = 0.81</td>
<td></td>
</tr>
<tr>
<td>Seasonal space heating energy efficiency of package under average climate</td>
<td>5</td>
</tr>
</tbody>
</table>

---

*Figure 53. Calculation method for determining the seasonal space heating efficiency hybrid heat generators (Annex IV, Figure 3)*
Calculation example No. 1 (without solar contribution, without storage tank):

Step 1.
Preferential Heater is an ASHP with $P_{\text{rated}} = 4\text{ kW}$ and a seasonal space heating efficiency of $126\%$

Step 2.
Modulating room thermostat with modulating heater

Step 3.
Non-preferential heater $P_{\text{sup}}$ is a $24\text{ kW}$ condensing boiler with an $\eta_{s,h} = 94\%$

(Minimum boiler output is selected here)

Calculating Step 3 value:

\[
\frac{P_{\text{rated}}}{P_{\text{rated}} + P_{\text{sup}}} = \frac{4}{28} = 0.143
\]

According Table 6, for a package without storage tank the value for parameter II = 0.592

Value for Step 3 is: $(94 - 126) \times 0.592 = -18.9\%$

Step 5
Seasonal space heating efficiency = Step 1 + Step 2 + Step 3
Condensing gas boiler $110.1\%$

Calculation example No. 2 (without solar contribution, without storage tank):

Step 1.
Preferential Heater is an ASHP with $P_{\text{rated}} = 4\text{ kW}$ and a seasonal space heating efficiency of $126\%$

Step 2.
Modulating room thermostat with modulating heater

Step 3.
Non-preferential heater $P_{\text{sup}}$ is a $8\text{ kW}$ condensing boiler with an $\eta_{s,h} = 94\%$

(Minimum boiler output is selected here)

Calculating Step 3 value:

\[
\frac{P_{\text{rated}}}{P_{\text{rated}} + P_{\text{sup}}} = \frac{4}{12} = 0.333
\]

According Table 6, for a package without storage tank the value for parameter II = 0.217

Value for Step 3 is: $(94 - 126) \times 0.217 = -6.9\%$

Step 5
Seasonal space heating efficiency = Step 1 + Step 2 + Step 3
Condensing gas boiler $122.1\%$

The first calculation above indicates that the existing methods for assessing the seasonal space heating efficiency for a hybrid heat generator package (where both a heat pump and a boiler is installed) will lead to an increased seasonal space heating efficiency of 16\% compared to combination boiler only solution. Given the extra costs that are needed (around €4000,- for an additional ASHP without storage tank), this is a small efficiency increase, especially for dwellings with smaller heat loads where the heat pump delivers most (>80\%) of the heat).

Hybrid solutions have the potential to overcome many of the obstacles that hamper the uptake of heat pumps in the existing market, including:
• Heat load of existing dwellings is too high for economically and technically feasible heat pump solutions.
• Electricity grid does not allow large scale uptake of heat pump only solutions.
• Limited installation options (available space) for heat pump only solution in existing dwellings.

To exploit this potential for hybrid generators both the product and the test-/assessment methods for determining the seasonal space heating efficiency need to be improved. The hybrid product needs to be integrated as much as possible in order to minimise product-size and related installation costs. The test/assessment methods must result in a realistic determination of the seasonal space heating efficiency, allowing for a correct evaluation of all product specific solutions for integrating and controlling both heat flows coming from boiler and/or heat pump. The assessment method that is currently in the Regulation (see calculation examples above) does not comply in this respect. It simply assumes that the separate seasonal efficiency of each appliance is achieved for the share of the heat load that is covered by the appliance.

6.2 Draft Test Standard for Hybrids following TC 113/WG7

In the latest prEN 14825:2018(E) N668, a chapter 7 is added related to the testing of hybrid heat pumps, which are defined as: “encased assembly or assemblies designed as a unit consisting of an air/water(brine)/DX-to-water(brine) electrically driven heat pump with a second generator using fossil fuel, and managed by a common controller providing an optimized operation of the heat generators for space heating”

Two test methods are proposed to determine the $SCOP_{on}$ for hybrid heat pumps: a separated test method and a combined test method.

**Separated test method**

In the separated method the test of the heat pump is conducted with the boiler hydraulically connected but without fuel supply. The tests are similar to the tests for a single heat pump and according to Chapter 6 of the standard ‘Part load conditions for space heating’. Only testing temperature conditions equal or greater than $T_{hp;off}$ shall be conducted. Below that point the heat pump capacity is declared zero. For the determination of the seasonal space heating efficiency of the boiler $\eta_{son}$ and $\eta_{sup}$ and the related corresponding parameters ($\eta_{1}$, $\eta_{4}$, $P_{1}$, $P_{4}$, $P_{stby}$, $el_{max}$, $el_{min}$, $P_{sb}$, $P_{ign}$) the gas boiler shall be tested according to EN15502-1 and the liquid fuel boiler according to EN304 and EN303-2. In this method, $SCOP_{on}$ is calculated using the formula below

$$SCOP_{on} = \frac{\sum_{j=1}^{n} h_{j} \times P_{a}(T_{j})}{\sum_{j=1}^{n} h_{j} \left[ \frac{P_{a}(T_{j}) - P_{sup}(T_{j})}{\text{COP}_{a}(T_{j})} + \frac{P_{sup}(T_{j})}{\eta_{son} \times CC} \right] \times \left( 1 + \frac{F(2) + F(3) + F(4)}{\eta_{sup}} \right)}$$
Combined test method

Using the combined method, the tests are conducted with the complete appliance in operation, following the part load test point as described in Chapter 6 of the standard. Depending on the type of boiler, the system shall be supplied with all the accessories necessary for its installation (according to EN15502-1 and EN304).

The annual fossil fuel consumption in kWh for the dedicated climate and selected heating system can be calculated as the sum of $Q_{\text{fossil}}$ times $h_i$ over all bins.

When using the combined test method, the unit is considered as a black box for which the heating capacity, electricity and/or fossil fuel consumptions are measured. For each test at the temperature condition $T_j$, the COP$_d$ of the unit is defined as the ratio between the heating capacity of the unit and the total energy power input according to the formula below.

$$\text{COP}_d(T_j) = \frac{P_h(T_j)}{P_{\text{elec}}(T_j) + Q_{\text{fossil}}(T_j)/\text{CC}}$$

Where
- $T_j$ is the temperature condition;
- $P_h(T_j)$ is the heating capacity of the unit for the corresponding temperature $T_j$, expressed in kW;
- $P_{\text{elec}}(T_j)$ is the total electricity power input of the unit for the corresponding temperature $T_j$, expressed in kW;
- $Q_{\text{fossil}}(T_j)$ is the fossil fuel power input for the corresponding temperature $T_j$, expressed in kW;
- CC is the conversion coefficient, equal to 2.5.

The SCOP$_{on}$ is calculated with the formula below:
\[
\text{\text{SCOP}_{\text{on}}} = \frac{\sum_{j=1}^{n} h_j \times P_h(T_j)}{\sum_{j=1}^{n} (h_j \times \text{COP}_{\text{bin}}(T_j))}
\]

Where

- \( j \) is the bin number;
- \( n \) is the total number of bins;
- \( h_j \) is the number of bin hours occurring at the corresponding temperature \( T_j \);
- \( T_j \) is the bin temperature;
- \( P_h(T_j) \) is the heating load for the corresponding temperature \( T_h \) expressed in kW;
- \( \text{COP}_{\text{bin}}(T_j) \) is the \text{COP} value of the unit for the corresponding temperature \( T_j \).

When the SCOPon is known following the tests and calculations described above, the seasonal space heating efficiency \( \eta_{\text{sh}} \) can be calculated with the formula’s described in chapter 8 of the prEN 14825.

### 6.3 Draft Test Standard for Hybrids following TC 109/WG1

CEN/TC 109/WG1 is also working on a new standard for testing hybrid gas generators. Their proposal (the draft prEN 15502-2-Y:2019) is currently being discussed in the Ad Hoc Joint Working Group together with CEN/TC 113/WG7.

Where the TC 113/WG7 followed the approach to maintain the multiple test points given by the prEN 14825, TC 109/WG1 tries to reduce the number of test points and related test costs. This method is also referred to as "Simplified Method".

The submission of a New Working Item (NWI) is being prepared, based on the draft prEN 15502-2-Y:2019.

The following text is used for describing the Scope:

The hybrid product is composed of:

- A gas boiler as primary heat generator which could supply the heat demand in all operating conditions (type C1 up to C9 and types B2, B3 and B5, according to FprEN1749:2018).
- An electric or gas heat pump, as a secondary heat generator, which cannot fulfil the heat demand in all operating conditions.
- A control unit that optimises the operation of the generators.

Due to its composition, the hybrid generator shall be able to supply 80 °C feed temperatures.

In chapter 7.Controls, paragraph 7.4 further states that the controls shall be designed to optimise the seasonal primary energy consumption; controls shall monitor both efficiencies of boiler and heat pump to select the optimum hybrid operational mode.
Three operational modes are defined:

1. Zone 1: efficiency of the heat pump > efficiency of boiler and heat pump output is enough to cover the heat demand
2. Zone 2: efficiency of the heat pump > efficiency of boiler but heat pump output is not sufficient to cover the heat demand; boiler is now used as backup heater ($P_{sup}$).
3. Zone 3: efficiency of boiler > efficiency of heat pump; boiler is used alone.

![Example of hybrid operation](image)

**Figure 54. Operating modes for boiler and heat pump**

Normative Annex B of the prEN 15502-2-Y:2019 describes the method for calculating the seasonal space heating efficiencies.

**Ad B.3 Determination of representative test point**

The manufacturer declares in his technical documentation at what outdoor temperature the fuel boiler is switched off ($T_{fb,off}$, above which heat is only provided by the heat pump), and at what outdoor temperature the heat pump is switched off ($T_{hp,off}$, below which heat is only provided by the boiler).

*With these two switch-over points the heating hours of the climatic season can be divided into three zones (see)*. For each zone the average outdoor temperature ($T_{out}$) shall be calculated, using climatic data from EN 14825, with the following formula (1):

$$T_{out} = \frac{\sum h_j * p(T_j) * T_j}{\sum h_j * p(T_j)}$$  \hspace{1cm} (1)

Where:

- $j$ = bin number
- $T_j$ = bin temperature
- $h_j$ = number of bin hours for temperature $T_j$
- $p(T_j)$ = partial load of bin $j$, calculated according formula (2)

$$p(T_j) = \frac{T_j - 16}{T_{designh} - 16}$$  \hspace{1cm} (2)

Where:
\[ T_{\text{designh}} = \text{reference design temperature conditions for heating for average, colder and warmer climates (following prEN14825 section 3.1.64 of the terms and definitions), being } -10 \, ^{\circ}\text{C outdoor for average, } -22 \, ^{\circ}\text{C for colder and } +2 \, ^{\circ}\text{C for warmer climates. Indoor temperatures are fixed at } 20 \, ^{\circ}\text{C.} \]

**Climate condition**

The flow temperature of each of the three zones is determined via interpolation or extrapolation using Tables B1, B2 or B3.

<table>
<thead>
<tr>
<th>Table B.1 - Average Climate Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>[ T_{\text{flow}} ] at various design system temperatures</td>
</tr>
<tr>
<td>[ T_{\text{in}} [^{\circ}\text{C}] ]</td>
</tr>
<tr>
<td>----------------------------------------</td>
</tr>
<tr>
<td>20</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table B.2 - Warmer Climate Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table B.3 - Colder Climate Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>

*Figure 55. Tables B.1, B.2 and B.3 taken from the prEN15502-2-Y:2019*

**Return temperatures**

The return temperatures shall be calculated using the formula

\[ T_{\text{return}} = T_{\text{flow}} - \Delta T \tag{3} \]

The temperature difference \[ \Delta T \] will be calculated using formula

\[ \Delta T = \Delta T(T_{fb,off}) \times \frac{pl(T_{out})}{pl(T_{fb,off})} \quad \text{or} \quad \Delta T = \Delta T(T_{\text{designh}}) \times \frac{pl(T_{out})}{pl(T_{\text{designh}})} \tag{4} \]

Where:

\[ \Delta T(T_{fb,off}) = 5 \, ^{\circ}\text{C} \]
\[ \Delta T(T_{\text{designh}}) = 20 \, ^{\circ}\text{C} \]
The temperature difference calculated shall not be greater than the values given in Table B.4 of the prEN 15502-2-Y:2019.

<table>
<thead>
<tr>
<th>Temperature application</th>
<th>Max. ΔT [K]</th>
<th>Minimum T_return [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low</td>
<td>12.5</td>
<td>21.0</td>
</tr>
<tr>
<td>Intermediate</td>
<td>17.5</td>
<td>23.0</td>
</tr>
<tr>
<td>Medium</td>
<td>20.0</td>
<td>24.4</td>
</tr>
<tr>
<td>High</td>
<td>20.0</td>
<td>24.6</td>
</tr>
<tr>
<td>Very high</td>
<td>20.0</td>
<td>25.2</td>
</tr>
</tbody>
</table>

Figure 56. Table B.4 from the prEN15502-2-Y:2019

**Zone weighting**

The weights of the three zones are defined by the ratio between the equivalent hours of each zone and the total equivalent hours. The averaged part load over the total heating period is calculated according to formula (5). The energy weight of each heating zone is calculated according to formula (6), were \( i \) relates one of the zones 1, 2 or 3.

\[
H_e = \sum_{j=1}^{n} h_j \times pl(T_j) \tag{5}
\]

\[
W_i = \frac{\sum_i h_j \times pl(T_j)}{H_e} \tag{6}
\]

Where

- \( H_e \) = averaged part load over the heating season (i.e. energy generated over total heating period divided by \( P_{\text{design}} \))
- \( W_i \) = the energy weight of heating zone \( i \)

If the emitter system is hydraulically disconnected from the hybrid boiler (there is a hydraulic separator inside the hybrid boiler and the heating system has a dedicated pump) only ‘Method A’ is applicable.

**Checks**

It shall be checked that the heat pump can satisfy the required thermal load at \( T_{fb;off} \). If not, \( T_{fb;off} \) is increased until the heat pump can deliver the required thermal load.

It shall also be checked that the heat pump can achieve the \( T_{\text{flow}} \) as prescribed for the selected temperature application.

**Ad B.4 Direct method (Method A)**

Using the direct method, tests are conducted with the complete appliance in operation at the conditions that were calculated for each of the three zones (\( T_{\text{out}} \), Partial load ratio (\( plr \)), \( T_{\text{supply}} \), \( T_{\text{return}} \)). The total electric power consumption shall be measured (\( Q_e \)), and the
total gas input shall be measured ($Q_g$). The efficiency for the representative point $i$ per zone is calculated using formula (7)

$$\eta_i = \frac{P_{\text{design}} \cdot plr_i}{Q_{g,i} + 2.5 \cdot Q_{e,i}}$$ (7)

Ad B.5 Separated method (Method B)

The tests are conducted with the gas boiler only and with the heat pump only at the conditions of the related zones 1 and 3.

The efficiency of the gas boiler shall be measured according to EN15502-1, at the conditions of zone 1 ($P_{\text{design, hybrid}} \cdot plr_1$, $T_{\text{out},1}$, $T_{\text{flow},1}$, $T_{\text{return},1}$). The efficiency $\eta_1$ is calculated using the formula (8)

$$\eta_1 = \eta_{FB,1} - \frac{2.5 \cdot Q_{e,1}}{P_{\text{design, hybrid}} \cdot plr_1}$$ (8)

The efficiency of the heat pump shall be measured according to EN14511, at the conditions of zone 3 ($P_{\text{design, hybrid}} \cdot plr_3$, $T_{\text{out},3}$, $T_{\text{flow},3}$, $T_{\text{return},3}$). In this case $Q_{e,3}$ is included in the determination of $\text{COP}_3$. The efficiency $\eta_3$ is calculated using the formula (9)

$$\eta_3 = \frac{\text{COP}_3}{2.5} = \eta_{\text{SHP,3}}$$ (9)

The tests that need to be performed for zone 2 depend on the way in which boiler and heat pump are hydraulically connected: either parallel or in series.
Table 11. Options for combined operation Zone 2: Parallel (left) and Serial (right)

<table>
<thead>
<tr>
<th>Parallel Operation</th>
<th>Serial Operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat pump and boiler have the same $T_{return,2}$</td>
<td>FB return temperature is equal to HP flow temperature</td>
</tr>
<tr>
<td>Heat pump flow temperature shall be determined based on technical documentation.</td>
<td>HP flow temperature shall be chosen as to simulate the hybrid boiler control algorithm</td>
</tr>
<tr>
<td>Heat pump capacity $C_{HP,2}$ shall be measured.</td>
<td>Heat pump capacity $C_{HP,2}$ shall be measured.</td>
</tr>
<tr>
<td>Fuel boiler (FB) flow temperature depends on HP flow temperature.</td>
<td>Fuel boiler shall generate the missing power to achieve the reference partial load of zone 2.</td>
</tr>
<tr>
<td>Fuel boiler shall generate the missing power to achieve the reference partial load of zone 2.</td>
<td>Water flowrate through the boiler ($\dot{m}_{FB,2}$) shall be measured</td>
</tr>
<tr>
<td>- Water flowrate through the boiler ($\dot{m}_{FB,2}$) shall be measured.</td>
<td>- FB flow temperature depends on FB flowrate</td>
</tr>
</tbody>
</table>

Formula’s for determining $T_{flow,FB,2}$ in case of parallel operation:

$$T_{flow,FB,2} = T_{flow,2} \frac{\dot{m}_2}{m_2} - T_{flow,HP,2} \frac{\dot{m}_{HP,2}}{m_2 - \dot{m}_{HP,2}}$$  \hspace{1cm} (10)

Formula’s for determining $T_{flow,FB,2}$ in case of serial operation:

$$T_{flow,FB,2} = \frac{\dot{m}_2}{\dot{m}_{FP,2}} (T_{flow,2} - T_{flow,HP,2}) + T_{flow,HP,2}$$  \hspace{1cm} (11)

Where

$$\dot{m}_2 = \frac{P_{design,hybrid} \cdot p_{lr_2}}{c \cdot (T_{flow,2} - T_{return,2})} \quad \text{and} \quad \dot{m}_{HP,2} = \frac{c_{HP,2}}{c \cdot (T_{flow,HP,2} - T_{return,2})}$$  \hspace{1cm} (12)

With

- $c$ = specific heat of water: 4186 kJ/[kg.K]
- $\dot{m}_2$ = total water flow rate in kg/s
- $\dot{m}_{HP,2}$ = water flow rate through heat pump in kg/s
\[ \eta_{hp,2} = \frac{COP_2}{2.5} \]  

**Checks**

It shall be checked that the heat pump can satisfy the required thermal load at \( T_{fb;off} \). If not, \( T_{fb;off} \) is increased until the heat pump can deliver the required thermal load.

It shall also be checked that the heat pump can achieve the \( T_{flow} \) as prescribed for the selected temperature application.

**Calculations**

**Calculating the weight of the heat pump and the boiler in Zone 2**

The weight of the heat pump in Zone 2 in calculated with formula (14)

\[ W_{HP} = \frac{C_{hp,2}}{P_{designh} * plr_2} \]  

**Calculating the efficiency of a hybrid in Zone 2**

The efficiency of a hybrid boiler in Zone 2 is calculated according to the formula (15)

\[ \eta_2 = \frac{1}{W_{hp} \eta_{hp,2} + \left(1 - W_{hp}\right) \eta_{FB,2}} \]  

**Calculation of seasonal space heating efficiency**

The seasonal space heating efficiency of the hybrid boiler in ON mode is calculated according to the following formulae (16)

\[ \eta_{son,hybrid} = \frac{1}{\frac{W_1}{\eta_1} + \frac{W_2}{\eta_2} + \frac{W_3}{\eta_3}} \]  

The seasonal space heating efficiency is then calculated as follows:

\[ \eta_{S,hybrid} = \eta_{son,hybrid} - \sum F(i) \]
6.4 Observations in relation to Regulation & draft test standards

When hybrid generators are addressed in the Regulation or in the Standards, the term ‘preferential’ or ‘primary’ appears, indicating that one of the heat generators in the combined system is the preferred or primary one. The Regulation speaks of preferential heat pump space-heaters and preferential boiler space-heaters, which are both space-heaters that can be combined with other type of heat generators. The prEN 14825:2018 defines the hybrid as an assembly of an electrically driven heat pump and a secondary heat generator using fossil fuel. The prEN 15502-2-Y:2019 describes the hybrid as a product composed of a primary heat generator - being the fuel boiler (that is able to supply all of the heat demand) - and a secondary heat generator, being an electrical or gas driven heat pump. Unfortunately there is no clear definition of what the term preferential or primary exactly means in this context and what it is referring to.

The assumption of the study team is that by using the term preferential, one tries to indicate that one of the two generators is doing the biggest part of the space heating. But there is no real need for doing that, or in other words, it does not serve a purpose. The purpose of applying two (or more) generators (a hybrid) is improving the overall seasonal efficiency. So which generator is doing what part of the heat load fully depends on which generator offers the highest efficiency at what operating conditions. Obviously the heat output that can be delivered by a single generator is a limiting factor here. This means that the logical order for determining the preferred operating mode is:

1. What is the instantaneous heat load that is needed, at what outdoor temperature and what indoor temperature regime
2. Provided the requested heat load can be delivered by both generators at requested temperature regimes, what is the operating mode delivering the highest efficiency?
   Generator A, generator B or combined operation.

Potential of hybrids

It is generally acknowledged that the technology related to heat pumps and fuel boilers is advanced far enough to allow the design and manufacture of meaningful hybrids, that can deliver the requested heat load and are more efficient (and cost effective) than the alternative of a single generator. In that sense, heat pump/ fuel boiler hybrids may have a bright future in the replacement market. Covering the requested heat load will not be an issue for hybrids, especially when the default (24 kW) combination boiler providing domestic hot water as well. But regarding efficiency and cost effectives, these topics are more complicated to assess and largely depend on:

1. Hybrid specific controls, determining whether the heat pump, the boiler of both are used
2. Capacity heat pump in relation to the heat load of the house
3. Requested system temperatures (determined by HL/EC-ratio)

This all implies that the assessment method in the Regulation EU 811/2013, where the rated capacity of both generators is used to weigh the share of both generators and using a ‘II’ factor for calculating the attribution of both generator efficiencies (see section 6.1) is not sufficiently representative.

The recently proposed preliminary standards (chapter 7 prEN14825:2018 and prEN15502-2-Y:2019) are more sophisticated and use the bin method to determine the contribution of the various operating modes. Both these methods define an outdoor temperature where the heat pump is switched off ($T_{hp;off}$) and an outdoor temperature where the fuel boiler is switched off ($T_{fb;off}$). Above $T_{fb;off}$ the heat pump does all the work, below $T_{hp;off}$ the fuel boiler does all the work and in between both generators do their share. It is assumed that these switch-over temperatures are provided by the manufacturer and must be indicated in the technical documentation. The tests are done using these two outdoor temperatures.

In real life however, these switch-over outdoor temperatures will most probably not be used to control the operating modes of the hybrid, simply because the heat load of the house and temperature regime are not known. Higher than theoretically assumed heat loads, will lead to
higher switch-over points. Also higher than theoretically assumed temperature regimes (which in practise is often the case, see chapter 2) will lead to higher switch over points and different relative shares in operating modes.

In practice this means that these controls will be more sophisticated and based on the instantaneous temperature lift, rather than a single outdoor temperature switch-over point. It also means that these switch-over controls are a crucial component in achieving the ultimate efficiency of the hybrid. In the end, the actual efficacy of these controls will determine whether the hybrid is employed in the best possible manner, given the various heat loads and temperature regimes it will encounter in the various dwellings it will be installed.

And because – due to the capacity of fuel boilers - hybrids can easily be applied in dwellings with widely varying heat loads, the relative share of the various operating modes (share of the heat pump) will vary as well, leading to varying seasonal efficiencies and to a varying cost effectiveness.

For hybrids it is therefore proposed to:

1. Develop a test to define the efficacy of the controls that determine which operating mode is to be applied. By varying the supply- and return temperatures in the heat pump test points and observe the switching behaviour of these controls, an indicator could be identified representing the hybrid-operating-mode-control efficacy. Such an indicator can be used as a multiplier when calculating the seasonal efficiency of the hybrid.

2. Because in a test, fixed switch-over points are used to determine the seasonal efficiency of the hybrid, this also implies that a specific design heat load of the house is assumed here. This rated design heat load should be communicated together with the seasonal efficiency, indicating that the calculated seasonal efficiency is directly related to the assumed heat load. Because of its higher efficiency figures, manufacturers will select favourable switch-over points, corresponding to small design heat loads. E.g. a 28 kW hybrid (with a 24 kW fuel boiler and a 4 kW heat pump) is tested at a design heat load of 5 or 6 kW, yielding high seasonal space heating efficiency figures.

3. Because these 28 kW hybrids can also be used in dwellings with much higher heat loads, it is advised to also test/calculate the seasonal efficiency at the maximum design heat load (in this example 28 kW), and include such information in the technical fiche. Using a linear interpolation one can determine what seasonal space heating efficiencies can be achieved in dwellings with different heat loads.
6.5 Products on the market

Multiple product configuration (or packages) is offered on the market under the name hybrid heat pump, hybrid boiler or hybrid space heater. Table 12 summarizes the product configurations that are possible for hybrid space heaters.

When the heat pump part of the hybrid is also requested to help heating the sanitary water, a storage tank is added to this package.

Table 12. Hybrid product configurations

<table>
<thead>
<tr>
<th>Configuration No.</th>
<th>Heat pump</th>
<th>Fuel boiler</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Outdoor unit</td>
<td>Indoor unit</td>
</tr>
<tr>
<td>1</td>
<td>Separate</td>
<td>Separate</td>
</tr>
<tr>
<td>2</td>
<td>Separate</td>
<td>Integrated indoors unit &amp; boiler</td>
</tr>
<tr>
<td>3</td>
<td>Integrated Monobloc outdoors</td>
<td>Separate</td>
</tr>
<tr>
<td>4</td>
<td>Integrated Monobloc indoors</td>
<td>Separate</td>
</tr>
<tr>
<td>5</td>
<td>Integrated Unit indoors</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Integrated Unit outdoors</td>
<td></td>
</tr>
</tbody>
</table>

Configuration no. 1 and 2 requires F-gas certified installers, whilst configuration 3 to 6 can be done by traditional heating installers. Because of this and because they involve more installation work, the total costs for configurations 1 and 2 will probably be higher than configurations 3 to 6. Configurations 5 and 6 have the biggest market potential because they are fully integrated, have the smallest size, lowest installation costs, and can be installed in the same location as the boiler that it is going to replace.

Hybrid Configuration 1

This configuration can be created with any kind of heat pump and boiler product from any kind of manufacturer. The installer makes the package is often also responsible for the controls that determine the operating-mode of the hybrid. This solution is thought to be less cost effective and susceptible for suboptimal operating modes when the controls are not tested and co-delivered by a manufacturer. Heat pump packages that are especially designed for hybrid operation and include well designed and tested controls for determining the optimal operating mode, can tackle these problems.
The hybrid package consists of an outdoor unit (supplied by e.g. Toshiba), a small indoor unit named Elga including controls for selecting the optimal operating mode and a room thermostat. The small indoor unit (condenser section) can be mounted next to the boiler. The indoor unit Elga communicates with any type of boiler through the OpenTherm protocol.

![Figure 58. Remeha hybrid components in place](image)

The heat output of the heat pump section is 3.8 kW at A7 °C/W45 °C. The sound pressure level of the outdoor unit at 5 metres (full load, free standing) is 47 dB(A).

**Hybrid Configuration 2**

With this configuration the location where the boiler is installed can remain the same, provided the new boiler (with integrated condenser condenser) is about the same size. The controls that determine the operating-mode of the hybrid is also integrated in this boiler/condenser unit and is tested and optimized by the manufacturer. An F-gas certified installer is needed to install the outdoor-unit (evaporator) and connect the related pipework to the integrated indoor unit.

![Figure 59. Daikin Altherma hybrid heat pump](image)

On the back side of a small condensing combination boiler, the condenser section of the heat pump is located, resulting in a slightly increased depth of the combination boiler.

The heat pump module is supplied as either a 4 or a 7 kW unit (at A7 °C/W45 °C), and the condensing combination boiler module has a capacity of 7.9 to 31.9 kW (min.-max.). The sound pressure level of the outdoor unit is 48 dB(A)
According to the manufacturer, for an average dwelling with a heat load of 14 kW with the average European climate, the largest part of the required heat output can be covered in the two operating modes using the heat pump (heat pump operation only and combined heat pump boiler operation). In this example a total of 70% of the required thermal energy is produced by the heat pump and 30% by the gas boiler.

For this hybrid unit Daikin developed a patented algorithm that continuously evaluates whether or not the efficiency of the heat pump is higher than that of the condensing gas boiler. Based upon this evaluation the operating mode is selected. These operating modes include a true hybrid operating mode, in which the water is preheated by the heat pump before it is further heated to its requested temperature by the gas boiler.

**Hybrid Configuration 3 or 4**

With these configurations, the complete heat pump product is pre-assembled and integrated in a Monobloc, that can either be installed outside or inside the house. With these configurations there is no need for an F-gas certified installer, because all connections dealing with refrigerant are already made by the manufacturer. This Monobloc must be installed in addition to the condensing fuel boiler. Installation inside the house requires a considerable additional space. But also when installed outside the house a suitable space must be found, that is acceptable from the perspective of noise and does not require long connection lines. Controls for determining the operating modes are preferably supplied by the manufacturer.

Figure 59 shows on the left side a heat pump monobloc next to a condensing gas boiler and on the right side the dimensions (in mm) of the combined unit. Connections for the air ducts that are needed to supply the heat pump section with outdoor air are visible on top of the monobloc. The boiler section has its own (and considerably smaller) connections for air supply- and flue ducts.
The hybrid indoor monobloc (HP-Cube) has a rated capacity of 2.5 kW and the gas boiler (Base Cube) a capacity of 6.7 – 24 kW.

**Figure 61. Hybrid Indoor Monobloc and gas boiler by Itho Daalderop (Monobloc indoors)**

<table>
<thead>
<tr>
<th>Warmtepomp</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Verwarming +7/35W</td>
<td>5,02 kW (L,8-6,2)</td>
</tr>
<tr>
<td>Verwarming met E-BackUp +7/35W</td>
<td>6,10 kW</td>
</tr>
<tr>
<td>Verwarming -7/35W</td>
<td>3,60 kW</td>
</tr>
<tr>
<td>Koeling +28/6W</td>
<td>5,1 kW</td>
</tr>
<tr>
<td>COP</td>
<td>5,4</td>
</tr>
<tr>
<td>EER</td>
<td>4,1</td>
</tr>
<tr>
<td>Watertemperatuur verwarmen</td>
<td>25 / 55</td>
</tr>
<tr>
<td>Watertemperatuur koelen</td>
<td>6 / 20</td>
</tr>
<tr>
<td>Werkingsgebied verwarmen (t-buiten)</td>
<td>-20 / +35</td>
</tr>
<tr>
<td>Werkingsgebied koeling (t-buiten)</td>
<td>+10 / +35</td>
</tr>
<tr>
<td>Koudeouderelg / inhoud</td>
<td>R410 / 0,75kg</td>
</tr>
<tr>
<td>Geluid</td>
<td>38 dB(A)</td>
</tr>
<tr>
<td>Diameter luchtkaanal (rond)</td>
<td>250 mm</td>
</tr>
<tr>
<td>Voeding</td>
<td>230V / 1F / 50Hz (16A)</td>
</tr>
<tr>
<td>Opgenomen vermogen (max.)</td>
<td>1,7 kW</td>
</tr>
<tr>
<td>Afmeting (L x B x H)</td>
<td>656 x 518 x 798</td>
</tr>
<tr>
<td>Gewicht</td>
<td>65 kg</td>
</tr>
</tbody>
</table>

**Hydromodule**

| Tapwatercapaciteit | 8 liter/min |
| Tapwaterstijd | ca. 40 min |
| Afmeting (L x B x H) | 595 x 700 x 1500 |
| Gewicht | 175 kg |

**Figure 62. Hybrid Indoor Monobloc by Enzavu (www.enzavu.nl)**
The heat exchanger that separates the refrigerant flow from the CH-water is not integrated in this outdoor aroTHERM Monobloc, but is an additional component that is installed indoors (see installation diagram in figure 62).

The aroTHERM heat pump Monobloc is supplied as 5, 8, 11 or 15 kW HP-units. The additional Vaillant condensing gas boiler can be selected with various heat outputs, starting from 3.9 – 20.3 kW.
**Hybrid Configuration 5 or 6**

Promising hybrid configurations for the replacement market are those, that have all components completely integrated and pre-mounted in one product and which is are only slightly bigger than the fuel boiler that it replaces.

Several manufacturers are considering such an approach. Some manufacturers have already conceived their first products following this configuration.

**Figure 65. Fully integrated hybrid by Sime: Murelle Revolution 30**

This fully integrated hybrid heating unit is 900 mm high, 600 mm wide and 391 mm deep. The connections for outside air - needed for the heat pump - are 200 mm. The hybrid uses a separate 80 mm exhaust pipe for the removal of combustion exhaust fumes.

**Figure 66. Components, dimensions and connections Murelle Revolution**
The 4 kW heat pump section uses two evaporators, one for the exchange of thermal energy with the outside air, and another one for the exchange of thermal energy with the combustions air. The heat pumps operates either alone or in series with the 24 kW gas boiler during space heating and preheats the CH-water whatever operating mode is beneficial to the overall efficiency. In water heating mode, only the boiler is used for the production of a XL tapping pattern.

Another special hybrid using this no. 5 configuration, is the Uniqa Revolution, also developed by Sime. This hybrid consists of a type B1 non-condensing combination boiler and a small electric heat pump (booster) that uses the combustion air (drawn from the room it is installed) to preheat the CH-water. By doing such Sime claims that the seasonal efficiency can be as high as condensing boilers, without the need for condensate drains and special flue arrangements.

![Figure 67. B1 type hybrid non condensing gas boiler with booster heat pump by Sime](image-url)
7 DEVELOPMENTS REGARDING THERMALLY DRIVEN HEAT PUMPS (TDHPs)

7.1 Introduction TDHP

Thermally driven heat pumps (TDHPs) work at three temperature levels. Driving heat $Q_2$ is supplied at a high temperature level. Useful cold (cooling operation) or low temperature heat (heating operation) $Q_0$ is supplied at a low temperature level. The sum of the heat supplied is released at a medium temperature level. $Q_1$ is the useful heat in heating operation. In cooling operation, it is usually released to the environment. However, medium and low temperature heat can also be used simultaneously for heating and cooling purposes.

![Temperature levels of thermally driven heat pumps for heating and cooling](image)

In principle, all closed cycle TDHP types can be operated in heating and cooling mode. However, when we talk about thermally driven heat pumps, we usually refer to absorption or adsorption heat pumps. Having the highest process efficiencies (coefficient of performance, COP) these sorption heat pumps are far more widespread than TDHP processes like steam jet, double organic Rankine (ORC), thermo-acoustic, thermo-electric, Stirling, Vuilleumier, Pulse tube, or Gifford-McMahon processes. These cycles are incidentally used for applications with additional specific requirements such as, for example, very low useful temperatures. There are also combined (compression-sorption hybrid cycles) or successive heat and mechanically driven processes like Rankine and vapour compression.

---

The efficiency of TDHPs is defined by

$$\text{COP}_c = \frac{\dot{Q}_e}{\dot{Q}_2} \quad \text{for cooling operation,} \quad (1)$$

$$\text{COP}_h = \frac{\dot{Q}_1}{\dot{Q}_2} = 1 + \text{COP}_c \quad \text{for heating operation and} \quad (2)$$

$$\text{COP}_{h,c} = \frac{\dot{Q}_1 + \dot{Q}_0}{\dot{Q}_2} = 1 + 2 \cdot \text{COP}_c \quad \text{for combined heating and cooling operation.} \quad (3)$$

Thermally driven heat pumps are a small yet promising and emerging segment of the heat pump market. Thermally activated heat pumps use renewable energy based on the principle of sorption. As in the compression cycle, auxiliary energy is necessary. Sources of auxiliary energy can be fossil fuels, biomass, solar thermal energy or waste heat from industrial processes.

To be operated, the sorption heat pump has to be connected to a high temperature heat source, a low temperature heat source, and a medium temperature heat sink. Possible system configurations are presented in Figures 69. The reject heat in heat pump operation is used to provide room heating or domestic warm water with a temperature level of typically 20 to 70 °C. As for compression heat pumps the low temperature heat source is usually an environmental heat source: air (directly used or indirectly, e.g. preheated by a solar collector) and different ground sources are available. Sewage water is an interesting alternative. The big advantage over compression heat pumps is that the low temperature heat source needed for the same heating duty is only about half the size. Driving heat is provided most commonly by an internal gas burner but can also be provided by other heat sources as e.g. district or waste heat.

![Driving heat](image)

**Driving heat**
- gas
- district heat
- waste heat (CHP units, industrial processes) etc.

![Heat distribution system](image)

**Heat distribution system**
- panel heating
- radiators
- fan coils
- domestic hot water etc.

![Low temperature heat source](image)

**Low temperature heat source**
- air (dry cooler or solar collector)
- ground probes/collector/water
- sewage water etc.

![Ab/Adsorption heat pump system](image)

**Figure 69. Ab/Adsorption heat pump system**

Thermally driven heat pumps can provide heating and cooling services.

Ongoing R&D is expected to overcome a number of challenges that the technology still faces:

- Widening the operating parameters to optimise performance in existing buildings;
- Scaling down absorption systems to residential scale (capacity);
- Developing more compact systems (size);
- Intelligent integration with other heat sources.
7.2 Gas absorption heat pumps

The gas absorption heat pump technology represents the most developed and mature TDHP-technology. Gas absorption heat pumps use gas both as source of heat to be upgraded and as energy source to drive the heat pump process. This differentiates them from engine-driven heat pumps. The heat from gas is typically produced with a full premix burner.

In the basic absorption process, ammonia is evaporated by the free energy (e.g. outside air) and flows to an absorber, where it forms a solution with water. Heat is generated and is transferred from the absorber to the heating system. The ammonia-water solution is pumped at increased pressure to the generator where heat is added through for example a gas burner. The ammonia vapour formed in the generator flows to the condenser, where it is condensed and energy is transferred to the heating system. A lean ammonia-water solution recirculates from the generator to the absorber. Liquid ammonia flows after a pressure reduction from the condenser to the evaporator where it is vaporized again. Other refrigerants are possible in the absorption process, but ammonia-water is used in heat pumps for space heating.

(source Robur)

Figure 70. Basic gas absorption cycle.

The heat which the burner produces triggers various physical processes in the closed circuit of the gas heat pump - in contrast to an electric heat pump or gas engine driven heat pump, no compressor is needed.

The absorption gas heat pump technology is already a mature product with high efficiency. It is adapted for the replacement of existing boilers (minimal change of existing system) and also suitable for buildings with radiators that might require higher temperatures. Most gas absorption heat pumps use a refrigerant that has a smaller global warming potential than popular refrigerants for vapour compression cycles (Ammonia R717 has a GWP of 0, R32 GWP is 675, R410A GWP is 2088 kg CO₂ eq.) .

The main and largest manufacturer of gas absorption heat pumps is Robur. Their products are also sold on the market under different names by various boiler manufacturers. Their latest product (introduced in 2016) is a modulating 18 kW gas absorption heat pump, perfectly suited for existing residential dwellings for both space
heating and domestic hot water production (with DHW-storage), having a A++ label for space heating at medium temperature regime (55 °C).

<table>
<thead>
<tr>
<th>Technical data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum thermal input</td>
</tr>
<tr>
<td>Heating capacity at A7/W35</td>
</tr>
<tr>
<td>GUE a) at A7/W35</td>
</tr>
<tr>
<td>Heating capacity at A7/W50</td>
</tr>
<tr>
<td>GUE a) at A7/W50</td>
</tr>
<tr>
<td>Max outlet water temperature heating</td>
</tr>
<tr>
<td>Max outlet water temperature DHW</td>
</tr>
<tr>
<td>Nominal max electr. power (pump excl)</td>
</tr>
<tr>
<td>Weight</td>
</tr>
<tr>
<td>Sound pressure (@ 5m)</td>
</tr>
<tr>
<td>Size (w x d x h)</td>
</tr>
</tbody>
</table>

a) Gas Utilization Efficiency according to EN 12309

Figure 71. Robur 18 kW gas absorption heat pump

E-SORP Innovation GmbH, is another company developing a modulating 5 – 18 kW gas fired absorption product into a market-ready TD heat pump.

The key component here is the GAX generator/absorber; the value of the entire system depends on its performance measurements as per the standard in force (EN 12309) demonstrating that, for applications with a low supply temperature 35°C or high supply temperature 55°C for the heating system, the maximum seasonal efficiency of gas consumption was 167% or 152%, respectively.

The results of the project show that at low capacity, 5 to 18 kW, a fully modulating gas-fired absorption heat pump can make a useful contribution to reducing primary energy consumption.

Figure 72. E-Sorp modulating 5 - 18 kW heat pump
7.3 Gas adsorption heat pumps

Gas adsorption heat pumps represent a technology that is still in its development phase. Two German heating manufacturers (Vaillant and Viessmann) have launched zeolite heat pumps on the market for the low output range (up to 15 kW)\textsuperscript{35}. The products use either geothermal energy or solar thermal heat or a combination of both as a partial heat source. The products consist of two modules: a heat pump and a condensing boiler. The heat pump covers the base load for heating the building. The heat pump process requires high temperature heat to drive it. This is provided by the integrated condensing boiler which also covers high heat demands on cold days and domestic hot water heating.

The heat pump module utilises water as the refrigerant. Non-toxic and environmentally friendly zeolite acts as the adsorbent. The microporous ceramic mineral is able to adsorb significant amounts of water vapour in the dry state and thus release latent heat. This process operates in a cyclic manner using two phases:

1. Evaporation and adsorption; in this phase the water refrigerant trickles onto the surface of a stainless steel tube heat exchanger, which acts as an evaporator. The environmental heat is extracted from either a ground- or solar thermal collector. The environmental heat evaporates the water refrigerant. A decisive aspect is that low temperatures of about 4 °C are sufficient here because the process takes place in a high vacuum. The refrigerant vapour flows from the evaporator into the sorber. This has a large heat transmission surface that is coated with zeolite. The zeolite adsorbs the water vapour, and in doing so releases latent heat in a temperature range of up to 80 °C. This process enables the supplied environmental energy to be raised to a useable temperature level. The heat is fed via the heat exchanger into the heating system.

2. Desorption and condensation; As soon as the zeolite is saturated with water, desorption starts as the second phase. For this purpose the gas condensing unit heat the zeolite sorber so that the bound water is evaporated (desorbed). The water vapour condenses on the inside of the double-wall container, which acts as a capacitor and releases the heat previously absorbed by the burner as condensation heat back to the heating network. The heat input is terminated when the zeolite sorber is dry.

Field tests

The developers tested the practicability of the heating device in two successive field tests across Germany. According to this, the zeolite adsorption devices offered a 27% higher efficiency (when operating at 40/30 °C temperature regime) compared to utilization rates of state of the art condensing boilers.

\textsuperscript{35} Heating with gas adsorption heat pumps, Projekt info 03/2015, BINE Information Service, research project was funded by BMWi
Another adsorption type heat pump is being developed by Cooll (http://www.cooll.eu). The adsorption compressor consists of two pressure vessels containing activated carbon that cyclically are being heated and cooled.

(Source: Viessmann)

Figure 73. Adsorption heat pump device

Figure 74. Working principle Cooll adsorption heat pump
7.4 Other TDHP Technologies

*BoostHEAT* develops a TDHP using a newly designed thermally driven compressor. The device can achieve high heat output temperatures and seasonal gas utilization efficiencies (GUE) of up to 200%.
Production of a 20 kW boostHEAT device is planned to start in end of 2018 / beginning 2019. The 50 – 250 kW product versions will be assembled starting from beginning 2020.

**BLUE HEART** (www.blueheartenergy.com) is developing a market ready TDHP based on thermo acoustic waves that are send through a closed HP-circuit filled with helium under pressure.

Electrical driven drivers (1) send an acoustic wave through the pump. At the point where Helium is compressed heat is exchanged by a heat exchanger (2). At the point where the helium is expanded, heat from the source is added using a second heat exchanger (4). Between the two heat exchangers a regenerator is placed (3). Within the regenerator a thermal cycle arises. In this way the regenerator creates a temperature difference or a so called thermal pump or heat pump. The heat exchangers (2) and (4) are connected to either the source or heatsink, depending on the demand of the consumer, either heating or cooling.
Between the heat exchangers 2 and 4 a regenerator is placed. The regenerator is made of a porous material which functions as a buffer between the cold and warm heat exchangers. Due to the soundwaves a continuous cycle of compressing and expanding of Helium gas occurs within the regenerator.

The expectation is that BLUE HEART will introduce their first products for residential buildings second half of 2019.
8  Comments Stakeholders on Task 4 Items Presented in Stakeholder Meeting

8.1 Topic: default HP design temperature to 65 °C

This topic refers to the proposal to set the default design temperature for non-LT heat pump systems to 65 °C instead of 55 °C. This would imply that the average bin-weighted system temperature \((T_{sup} + T_{ret})/2\) would increase from 34.2 °C to 38.6 °C and result in a SCOP figure that would be slightly lower and consequently in a seasonal space heating efficiency that would be slightly lower. According to the study-team this would be more in line with real-life operating situations where CH system temperatures and controls are often not optimised for achieving the assumed test conditions.

Comments stakeholders

A minority of stakeholders would support the idea of reflecting more real-life test conditions, but the vast majority is against this proposal. A few of stakeholder probably misunderstood the proposal, based on the nature of their reaction.

Heat pump manufacturers and associations state that there is no ground for increasing the default design temperature for heat pumps, because the results of Fraunhofer study (which are one of the reasons for this proposal) are not correctly interpreted. Moreover it would give the wrong signal by implying that (non-LT) heat pumps would in principle be suited for high temperature heating systems. In addition, supply temperatures of 65 °C (at -10°C outdoor) cannot be realised by many heat pumps; it will reduce both the efficiency and the lifetime of the product. As it is, the amended EPB-Directive actively promotes the reduction of heat demands in dwellings and the current default design temperature of 55 °C is therefore fine for current and future building stock. HP-test temperatures should be in line with this.

Instead of adjusting the default design temperature for heat pumps, HP manufacturers and associations are in favour of promoting more strongly the conditions that are needed for improved system operation, being: reduction of heat load, high efficient emitters and optimised system controls. Some countries already have some kind of EPB-check dealing with these topics, before heat pumps and related subsidy schemes are deployed.

Fraunhofer further explains that the interpretation of the WP-Smart field trials by VHK are not fully accurate because the boundary conditions of these field trials cannot directly be compared to the EN14825 boundary conditions for 55 °C design temperature. The average of the climate zones of the WP-Smart field trails does not correspond to average climate conditions for the Medium temperature application of EN 14825.

Member states indicate that EPB-legislation will lead to lower heat loads (through improved insulation and airtightness) and that the existing plans for building improvement therefore automatically will lead to lower system temperatures. HP test temperatures should be in line with this and the default design temperature of 55 °C therefore has to remain. Belgium however, in principle does not oppose to a higher system design temperature, provided it does not have a negative impact on HP-sales.
Further research would then be requested to assess the actual impact of such an amendment. UBA-BAM points out that hydraulic balancing of the emitter system as a general strategy, is a prerequisite to reduce system temperatures.

Response
The comprehensive analysis regarding the system design temperatures in Task 4 is primarily intended to illustrate that further reduction of system temperatures and improved pump- and temperature controls offer a huge saving potential.

Current test conditions however anticipate real-life operating conditions that are often not the case. As a result there are many examples of underperforming HP-systems and unfortunately such examples often reach the press. The proposal to apply a slightly higher average system temperature does not imply a degradation of the heat pump. It means the same heat pump is placed in somewhat less favourable conditions, leading to a more realistic assessment of the seasonal efficiency.

The effect of heat pump controls can be made more visible in the seasonal efficiencies established through testing by adopting the Compensation or Dynamic tests as proposed a.o. by UBA-BAM (see also Task 1 and Task 6).

Having said that, the study team understands stakeholders comments that in order to improve upon the existing situation, it is not the heat pump that needs improving. The three key parameters that determine the system temperature (heat load, emitter capacity, controls) need to be addressed more prominently, because seasonal space heating efficiency figures can dramatically be improved when full attention is given to these topics.

8.2 Topic: Emitters capacity and control, new working item
As can be concluded from the analysis of the relation between SCOP values and CH system- or sink temperatures in Task 4, the emitter capacity plays a crucial role in achieving low(er) system temperatures. Optimising emitter capacity and related controls for flow and temperature may bring savings of up to 10% for condensing boilers and up to 50% or more for heat pump systems (expressed as seasonal efficiency). Since the emitter system is out of the scope of this study and in the light of projected saving potential it is recommended to add this subject to the future Ecodesign Working Programme.

Comments stakeholders
Only a few comments were received concerning this topic.

EHI is ready to contribute to future discussions on emitters’ systems. Emitters’ systems are important technologies for EHI and emitters manufacturers are represented within the association. Therefore, EHI stands ready to participate in future discussions on any proposed inclusion of emitter systems in the Ecodesign work programme.

For UBA-BAM it is not clear what the scope of the work item would look like and what would be potential Ecodesign requirements. They ask for more information. Furthermore, proper dimensioning of heat emitters could also be subject to the national EPBD implementations.
ECOS is in favour of a regulation that promotes better sizing and better temperature controls of emitters and welcome a new working plan to tackle this issue.

Belgium recommends a cautious approach since emitters are already covered by CPR: Space heating appliances are considered to be a construction product (EU/305/2011 Annex IV). Construction products that bear the CE marking require a Declaration of Performance (DoP) as of 1st of July 2013 by EU/305/2011 Article 3. The CE marking and DoP are required for products for which ‘harmonised standards’ exist, and which manufacturers and MS are obliged to use to establish required product performance. The list of ‘harmonised standards’ published in the OJ. The manufacturer has to comply with the essential requirements summarised in annex ZA (EU/305/2011 Article 4).

Response

As explained in section 3 of this Task, the heat-load to emitter-capacity ratio determines the actual design system temperature. The higher this ratio, the lower the system temperature will be. As already stated, the amended EPB-Directive will lead to lower heat-loads of both new and existing dwellings and commercial buildings. This will automatically result in lower system design temperatures, as it has done over the years. Where in the sixties the heat-load to emitter-capacity ratio led to system design temperatures of around 80 °C (90/70/20 °C) using radiators, the reduced heat-load of newly built houses leads to design temperatures of around 50°C (55/45/20 °C) using practically the same size of radiators. In that respect, sizing of radiators has not changed that much. It appears to be a more habit driven phenomenon than a carefully considered parameter of the heating system, meaning that certain emitter dimensions are acceptable and others not.

One could say that the general practice is that emitters sizes remain more or less the same resulting in system design temperatures of around 50 °C and when really low temperature emitters are requested, floor- or wall-heating systems are selected, providing up to 50 W/m² of surface at a temperature regime of 40/35/20 °C. The point however is, that in the replacement market where radiator/convector type emitters are the standard, floor and wall heating systems are not the ideal solution. Installing radiator/convector type emitters with a higher capacity will also result in lower system temperatures, but with the conventional type of radiators this would imply that radiator dimensions would need to be 2 to 3 times bigger which is not always possible and furthermore not preferred by the house owner. Radiator/convector type emitters with a high heating capacity at low temperatures (45/35/20 °C) can solve this problem, but for the time being the focus in both the development and the application of such LT radiators/conectors is limited. Also test standards (EN 442 and EN 16430 series) are not prepared to test heat outputs at low temperature regimes (lowest test temperature refers to an excess temperature of 30 °C, corresponding to an average flow temperature of 50 °C).

Next to emitter capacity, the flow and temperature controls play an important role. Flow temperatures that are higher than strictly needed as well as suboptimal flowrates will lead to reduced generating efficiencies. Further optimisation by improving related control systems will help maximising the savings that can be achieved with reduced system design temperatures.

As indicated, increasing the emitter capacity and improving flow and temperature controls and thus reducing the design system temperature by around 10 °C (or more),
represents a huge energy saving potential. It increases the generator efficiency with up to 10% for condensing boilers and up to 50% for heat pumps.

A new Ecodesign Preparatory Study on low temperature emitters could help further identifying bottlenecks and opportunities in achieving this large energy saving potential and may lead to new and targeted information requests regarding the low temperature heat emitting capacity of radiators/convectors. Explicit information on for instance the heat emitting capacity @ 40 °C per unit of emitter surface area (or volume) and explicit information on the effect temperature and flow controls have on system temperatures, may help initiating a heat-emitter selection process whereby system designers and installers are encouraged to select the correct emitters and controls for achieving the lowest feasible levels in system design temperature.

8.3 Topic: Heating controls

The study team argues that (also in the light of the topics mentioned above) controls have a larger impact on seasonal efficiency than the currently assumed default of 3%. It is proposed to increase the default malus from 3% to 8% and include not only temperature controls but also flow-controls. Study team also assumes that heating controls will remain in the Lot 1 Regulation because of its direct relation to and specific dependency on the heat generator.

Comments stakeholders

Apart from EU.BAC the vast majority of stakeholders fully agrees that the ‘heating controls’ topic must remain within the Lot 1 Regulation.

EU.BAC states that —although for the mid-term heating controls may remain under Lot 1— for the longer term EC and stakeholder should prepare to cover all controls under the Lot 38. Until that time current assessment methods and values are preferably to be maintained and not be changed in order to prevent unnecessary administrative burden for control manufacturers. EU.BAC agrees that controls have a larger impact than the currently default malus of -3% especially when flow controls are also included, but proposes to leave this assessment to the Lot 38 preparatory study.

EPEE-EHPA-Daikin are in principle against the proposal to reduce the malus and ask for further scientific evidence before deciding in favour of this proposal. As it is, the F(1) value is also applied in Regulation 21 which would then also need to change, which is problematic. These stakeholders are not in favour of the ensuing fact that the SCOP-value must increase to achieve the same seasonal efficiencies as with the existing regulation.

EHI does not agree with the proposal to increase the malus for controls to -8%. EHI is of the opinion that todays’ efficiency testing conditions for boilers and heat pumps correctly reflect their average use by consumers. There is thus no sound justification to lower the malus below -3%. This opinion is supported by a table showing the summary of one year boiler load monitoring data, indicating that boiler with turn down ratios up to 20% on average operate at 20 – 40% of their maximum capacity. EHI members furthermore disagree with the claim in the VHK study that weather compensation controls generally result in higher average system temperatures than room thermostat controlled systems. Their summarizing position is that the higher energy savings potential from more advanced temperature controls – and corresponding correction factors – remains to be
further detailed based on manufacturers’ experiences. EHI will further assess this issue and provide a proposal, where relevant. EHI emphasizes that in any case controls must remain within Lot 1!

If backed with further evidence, ECOS can agree with increasing the malus for controls. ECOS also proposes to include relevant technical specs of the sensors/controls that are actually used, in the future assessment method. ECOS further considers that the current Regulation makes a difference between boiler integrated controls and separate controls; in the latter a package label can be used leading to higher seasonal efficiency figures. Such a discrimination should not be the case. EHPA shares this view.

UBA-BAM agrees that in the context of an extended product approach, the effect of temperature controls should be fully integrated in the Ecodesign Assessment Method. With respect the surveillance they ask for a check on whether separately supplied controls are actually used.

Response
The decision to leave the energy assessment of the heating controls under the Lot 1 Regulation or relate them to the new Lot 38 is up to the EC. But given the direct relation to and specific dependency on the heat generator the study team has a preference for Lot 1.

To give an example
The integrated boiler controls and algorithms determine what boiler capacity will be used to heat the CH-water. A majority of the boilers use an algorithm that looks at the temperature increase (gradient) of the water coming out of the boiler compared to incoming CH water and adjusts the boiler capacity accordingly. If room thermostats are used to control the temperature in a room, the boiler will be switched off either by the room thermostat when the temperature set point is reached, or by the maximum thermostat that limits the flow temperature to a pre-defined maximum value. If the heating system is weather controlled with TRVs maximising the output of the emitters, the CH-supply temperature is determined by the heating curve, in which case the maximum CH-supply temperature is not governed by the maximum thermostat but by the outdoor temperature (heating curve).

In both cases the boiler capacity that will be addressed is determined by the flowrate and the preferences regarding allowed supply temperature gradient. High flowrates and big allowable supply temperature gradients will results in higher boiler capacities being addressed and vice versa. In other words, if flowrate and allowable supply temperature gradient are not governed smart enough, the envisaged savings related to lower supply temperatures and low capacity boiler operation will not be achieved. In that sense, the table with monitoring data on turn-down ratio’s indicating that boiler primarily operate around their minimum capacities has limited relevance. Anyhow, this example clearly shows that smart flow and temperature controls can only be achieved when they are interacting with smart boiler controls.

Evidence
Evidence for supporting the proposal to increase the malus for controls in this stage of the review study can only be gained from a basic physical approach and common sense.

In the heating season, every room in a dwelling loses heat. To maintain the temperature at the requested levels (set point temperature when present and temperature setback
when absent), the ultimate way would be to supply the exact same amount to the room that is lost, preferably at the lowest possible supply temperature to maximise generator efficiency and minimise distribution losses. Existing heating systems and their controls do not (yet) achieve this, as explained in section 3.3 of the task 4 report. As it is, flowrates, boiler capacities and heating curves are higher than strictly necessary, and in addition room temperature, system temperature and heating schedule controls are suboptimal leading to higher energy consumption that strictly needed. All in all 8% is – for the time being - considered an appropriate value for the malus related to these control topics (which according to EU.BAC may later on even prove to be to moderate when indeed flowrate controls and heating schedule controls are also included).

The assumption (EHPA and ECOS) that integrated temperature controls are not valued, opposite to a separately delivered temperature control and related package label is, according to the study team, not correct. The package label is also mandatory for boiler products with integrated temperature controls. Therefore current measure are not considered discriminatory.

8.4 Topic: Boiler test conditions

The study team states that the current boiler test conditions are overrating the share of time that condensing boilers operate with return temperatures of 30 °C and that the inherent assumption that the seasonal average return temperature is 34.5 °C is clearly too low (monitoring studies reveal an average real-life seasonal return temperature of around 41 °C in Germany and even higher in the UK).

Since boiler are also often oversized by a factor of around 2.5 (heat load of a dwelling is around 5 – 8 kW while 24 to 28 kW is being installed), this practice should be reflected in the proposed adjusted test methods. Proposal is to test the condensing boiler according to a 65 °C system design temperature for the average heating profile, or (provided explicitly requested because of LT-system design temperatures) to a 35 °C system design temperature, including oversize correction factors.

Comments stakeholders

MARCOGAZ claims that there is no oversizing of combi-boilers in relation to the heat load of the house and that this alleged problem is solved by boiler modulation ranges of 1 – 100%. So test- and calculation method must remain unchanged. Effort should be made however, to instruct and educate installers to avoid oversizing.

EHI is of the opinion that the proposal does not reflect real life situations and that current test conditions are adequate, referring again to the table showing the summary of one year boiler load monitoring data.

Response

To the knowledge of the study team that are no (or very few?) condensing boilers with a turn-down ratio of 1%. Please provide information of these product when available.

Furthermore, to prove that today’s test at 30% boiler capacity complies with real life minimum heat loads of a dwelling, by showing monitoring results of boiler products that are explicitly designed to operate in the 30% boiler capacity-range, indicating that in real-life heat loads don’t go must lower than 30% of the boiler capacity, is not very convincing.
8.5 Topic: Ecodesign minimum efficiency limits

Based on the new PEF new minimum energy efficiency limits are proposed, which in principle means that existing values are merely corrected for the increased PEF (limits are not increased). For non-LT electric heat pumps the minimum energy efficiency limit will go from 110% to 130%. For LT-heat pumps from 125% to 150%. As a result of that, also the labelling classes need adjustment.

Comments stakeholders

MARCOGAZ signals that thermally driven heat pump TDHP (amongst which gas heat pumps) will be banned when these new limits (≥ 130%) will be applied to all heat pumps irrespective of the type. These product however are important for customers who want to replace their gas boiler for a more efficient gas fired heat pump. The adjusted MEP-values may not disable this heat pump technology.

EHI has similar comments and objections. Under the current ErP legislation framework the minimum efficient limit (110%) is ambitious while reachable for all TDHP categories and the energy labelling class is representative of the major improvement in energy efficiency and operating cost savings for end users (A+). These two fundamentals aspects need to be maintained under the future regulation. The ranking on the label should also be in line with the principle expressed in the Energy Label framework Regulation, recital 17. Indeed, the scale should rank heating technologies that use renewable energy, in classes above A.

In case the thorough assessment of the introduction of a new Conversion Coefficient (CC) would confirm the need for a new CC value with consequent impact on Ecodesign requirement and labelling, then EHI recommends to:

- Set a specific minimum Ecodesign threshold for TDHPs; it could be set at “etas = 110%” (consistent with the current Ecodesign minimum energy efficiency limit for heat pumps).
- Define the labelling threshold classes in order to have these technologies in the top range of the classification scheme. In this case a specific Ecodesign limit for their NOx emissions could also be explored.

Remark: these considerations are based on the assumption of today’s ErP tests conditions and test methods (referring to EN 12309), which reflect real-life results (as acknowledged in VHK report) and today’s contribution of temperature controls.

Assess further the impacts of proposed energy label class changes for low-temperature heat pumps.

As explained above, any change in the Ecodesign limits and Energy Label class boundaries would need further impact assessment on the different technologies. It is also true for the proposed change in the class boundary of low-temperature heat pumps, which is not based on technology development or any least life cycle analysis.

Such assessment should also be comprehensive and look at the impact of all the changes proposed in the study, if they were to be introduced in a revised Regulation.

EHPA/GHP indicates that - under the existing ErP frame and associated measures - tens of thousands of TDHP-systems have been sold across Europe in residential building blocks and “light commercial” applications and several new residential products are currently under development for introduction across Europe. Especially GEHP are also
used for industrial and commercial applications. The choices that the European Commission will make by adopting a revised ErP could accelerate the adoption of TDHP technology or could completely exclude this technology from the market.

The current proposal of the consultant of the European Commission strongly puts at risk the existence of TDHP technology in Europe. Mainly, two issues arise from the proposed draft of options for the revision:

1. minimum thresholds of efficiency proposed for TDHP are currently unreachable, and
2. positioning of TDHP technology in the class labelling will be limited to classes A, not representative of the major improvement in energy efficiency and operating cost savings for end users compared to condensing boilers.

Response
The proposed adjustments on minimum energy efficiency limits concern the electric heat pumps. For thermally driven heat pumps (gas heat pumps) other efficiency limits will be proposed, appropriate for this TDHP-technology, resulting in a valuation that put thermally driven heat pumps above condensing boiler technologies.

Whether this technology will be in the top-range of the labelling classes (A+++) is however dependent on its actual seasonal efficiency figure in relation to the labelling class efficiency scale that does not discriminate between technologies.

As indicated, the proposed adjustments on minimum energy efficiency limits and related changes in labelling classes, primarily relates to changes in the PEF-value (meaning that MEP-values remain what they were but are recalculated due to the changed PEF). Any additional changes in labelling classes, by re-using empty labelling classes (because of ecodesign requirements), can help to further differentiate between products in the lower labelling classes. This of course has to be discussed with the Commission and stakeholders and options need to consider the announced review of labelling classes and removal of "+" classes by 2026.

8.6 Topic: Deployment LT-heat pump and TD-heat pump for DHW

Comments stakeholders
EHI claims that Lot 1 & 2 Regulations do not consider TDHP for water heating applications. Nevertheless, savings from TDHP for water heating applications have been demonstrated to be as large as the one achievable in space heating applications. Several manufacturers are already designing appliances or selling appliances for this specific application. Thermally driven heat pumps should thus be included in the scope of the Regulations 812/2013 and 814/2013.

SWEDEN asks for the labelling to be revised so that LT heat pumps that are able to produce sanitary hot water can be labelled. This is not the case with the current labelling.

Response
To the knowledge of the study team the current Regulations do not exclude thermally driven heat pump (TDHPs) from the scope.
Specific labels for DHW-products based on LT-heat pumps are indeed missing. The study team does not see any reason to exclude LT-heat pumps from labelling these DHW-products. This can be proposed as an extension of the DHW-labels.

8.7 Topic: Monitoring / Smart / Connectivity

Comments stakeholders

APPLiA states that **energy monitoring** gives the user feedback on consumption and can lead to important savings. Certain functions, such as volume flow limitation, can save water and therefore energy. This is not taken into account. They propose to include this benefit in the calculations. AI in the future can contribute to energy saving and it is important to reward these features to incentivise manufacturers to invest in it.

APPLiA further asks to consider SG Ready/PV Ready products. Please consider that it will become more relevant in the future because the national incentives will end after a certain number of years following the installation (20 years in Germany). The produced electricity will more likely be used in house and less sold to the grid. WG 4 is working to define test and calculating method that can also provide differentiation. See Annex 3.

UBA-BAM explicitly acknowledges the intention to merge the measurement and calculation methods to real life operation. This is a good way to avoid overrating the energy efficiency of space (and water) heaters. However, deficits in real life operation comprise impacts that cannot – or just to some extent – be reflected by measurement and calculation methods: dimensioning of boiler, dimensioning of heat emitters, control settings, hydraulic balancing etc. However, Ecodesign can address these effects indirectly by requiring energy efficiency monitoring (see their separate paper on this topic). It should be noted that such a measure would create a market for monitoring services, since the required hardware becomes available with the products themselves. The benefit for the user is manifold as energy savings are realised due to optimised operating conditions, early feedback on malfunction and installers being incentivised to sell well dimensioned appliances as well as to set optimal control parameters.

Response

Monitoring systems that directly optimise the operating conditions such that the overall system efficiency demonstrably is increased may be valued under the both Regulations. For monitoring systems that merely provide information and are dependent on ad-hoc human intervention before operating efficiencies are increased, this might be a bit more complicated.

The same goes for SG-Ready and PV-Ready appliances, which in principle are not primarily intended to increase system efficiency, but to solve potential grid problems.