Control of heating plants
## Contents

### 1. Heating boiler control
- 1.1 Introduction \(\text{6}^{6}\)
- 1.2 Boiler water temperature control \(\text{6}^{6}\)
- 1.3 Burner stage switching \(\text{8}^{8}\)
- 1.3.1 Switching with boiler thermostats \(\text{8}^{8}\)
- 1.3.2 Switch-on ratio \(\varepsilon\) (epsilon) \(\text{10}^{10}\)
- 1.3.3 Dynamic switching differential \(x_{\sigma}\) (x sigma) \(\text{11}^{11}\)
- 1.3.4 Switching with integral \(\Sigma K * \text{Min.}\) \(\text{12}^{12}\)
- 1.3.5 Modulating burners \(\text{13}^{13}\)
- 1.4 Systems without boiler pumps \(\text{13}^{13}\)
- 1.4.1 Load behavior \(\text{13}^{13}\)
- 1.4.2 Boiler water volume under variable load \(\text{14}^{14}\)
- 1.5 Systems with main pump and pressure less header \(\text{16}^{16}\)
- 1.5.1 Load behavior \(\text{16}^{16}\)
- 1.6 Boiler temperature compensation \(\text{17}^{17}\)
- 1.7 Maintained boiler return temperature \(\text{18}^{18}\)
- 1.7.1 Limiting control acting on the consumer groups \(\text{19}^{19}\)
- 1.7.2 Limiting controlling acting on a separate control element \(\text{22}^{22}\)
- 1.7.2.1 Bypass configuration in case of a pressure less header and central maintained boiler return temperature \(\text{24}^{24}\)
- 1.7.3 Maintained boiler return temperature via boiler mixing pump with thermostat \(\text{25}^{25}\)
- 1.8 Heating boiler with storage tank \(\text{26}^{26}\)
- 1.9 Boiler with flue gas heat exchanger \(\text{27}^{27}\)
- 1.10 Demand-controlled heat generation \(\text{28}^{28}\)

### 2. Control of multiple boiler plants
- 2.1 Introduction \(\text{29}^{29}\)
- 2.2 Hydraulic parallel or series connection \(\text{29}^{29}\)
- 2.3 Demands on boiler sequence control \(\text{31}^{31}\)
- 2.4 Sequence control changeover criteria \(\text{32}^{32}\)
- 2.4.1 Manual control \(\text{32}^{32}\)
- 2.4.2 Sequence control based on outside temperature \(\text{32}^{32}\)
- 2.4.3 Sequence control based on boiler temperature \(\text{33}^{33}\)
- 2.4.4 Sequence control based on a change in load and control factor \(\varepsilon\) \(\text{34}^{34}\)
- 2.4.5 Sequence control based on consumer flow temperature \(\text{34}^{34}\)
- 2.4.6 Sequence control based on common boiler return temperature \(\text{36}^{36}\)
- 2.4.7 Sequence control based on the maximum flow and return temperature \(\text{38}^{38}\)
- 2.4.8 Sequence control with a “hydraulic switch” (system header) \(\text{39}^{39}\)
- 2.4.9 Sequence control after storage charging \(\text{40}^{40}\)
- 2.4.10 Sequence control based on burner load \(\text{42}^{42}\)
- 2.4.11 Sequence control with the modulation factor of condensing boilers \(\text{42}^{42}\)

### 3. Control and supervision of oil and gas burners
- 3.1 Introduction \(\text{44}^{44}\)
- 3.2 Burner control \(\text{46}^{46}\)
- 3.3 Forced draft burners \(\text{47}^{47}\)
- 3.4 Burners without fans \(\text{48}^{48}\)
- 3.5 Output control for burners with two or more stages \(\text{49}^{49}\)
- 3.6 Modulating burner output control \(\text{49}^{49}\)
- 3.7 Flame supervision program \(\text{50}^{50}\)
- 3.8 Basic structure of the start-up program \(\text{51}^{51}\)
- 3.8.1 Standby \(\text{51}^{51}\)
- 3.8.2 Controlled startup (start-up command) \(\text{51}^{51}\)
- 3.8.3 Prepurging \(\text{53}^{53}\)
- 3.8.4 Preignition \(\text{53}^{53}\)
- 3.8.5 Safety time \(\text{54}^{54}\)
- 3.8.6 Controlled shutdown \(\text{54}^{54}\)
3.9 Peculiarities in the control of low-to medium-capacity forced draft burners
3.9.1 The air pressure switch can often be dispensed with
3.9.2 Controlled oil preheating
3.9.3 Long preignition
3.9.4 Postignition/reignition
3.9.5 Start repetition or lockout in the event of loss of flame
3.9.6 Peculiarities in the control and supervision of atmospheric gas burners
3.10 Modulating control of the flue gas residual oxygen content (\(\lambda\) control)

<table>
<thead>
<tr>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1</td>
<td>Room temperature control</td>
<td>60</td>
</tr>
<tr>
<td>4.1.1</td>
<td>Room temperature control, general</td>
<td>60</td>
</tr>
<tr>
<td>4.1.2</td>
<td>Room temperature control acting directly on the burner</td>
<td>61</td>
</tr>
<tr>
<td>4.1.3</td>
<td>Modulating room temperature control acting on a mixing valve</td>
<td>64</td>
</tr>
<tr>
<td>4.1.4</td>
<td>Room/flow temperature cascade control</td>
<td>65</td>
</tr>
<tr>
<td>4.2</td>
<td>Outside temperature or weather-compensated supply temperature control</td>
<td>67</td>
</tr>
<tr>
<td>4.2.1</td>
<td>Room temperature setpoint correction and night setback</td>
<td>69</td>
</tr>
<tr>
<td>4.2.2</td>
<td>Room temperature compensation authority</td>
<td>70</td>
</tr>
<tr>
<td>4.3</td>
<td>Individual room temperature control</td>
<td>71</td>
</tr>
<tr>
<td>4.3.1</td>
<td>Thermostatic radiator valve heads</td>
<td>72</td>
</tr>
<tr>
<td>4.3.2</td>
<td>Simple individual room temperature control systems</td>
<td>75</td>
</tr>
<tr>
<td>4.3.3</td>
<td>Individual room controllers integrated into building automation and control systems</td>
<td>76</td>
</tr>
<tr>
<td>4.3.4</td>
<td>Individual room control system combined with energy consumption measurement per occupied unit</td>
<td>77</td>
</tr>
<tr>
<td>4.4</td>
<td>Automatic heating limit switching</td>
<td>79</td>
</tr>
<tr>
<td>4.4.1</td>
<td>Yearly automatic heating limit switching (summer/winter changeover)</td>
<td>80</td>
</tr>
<tr>
<td>4.4.2</td>
<td>Daily automatic heating limit switching</td>
<td>80</td>
</tr>
<tr>
<td>4.5</td>
<td>Start and stop time optimization</td>
<td>81</td>
</tr>
<tr>
<td>4.6</td>
<td>Fast reduction / boost heating</td>
<td>82</td>
</tr>
<tr>
<td>4.7</td>
<td>Adaptive (self-learning) heating curve</td>
<td>82</td>
</tr>
<tr>
<td>4.8</td>
<td>Pump control and interlock</td>
<td>83</td>
</tr>
<tr>
<td>4.8.1</td>
<td>Differential pressure-dependent speed control</td>
<td>83</td>
</tr>
<tr>
<td>4.8.2</td>
<td>Speed control based on valve position of the consumers</td>
<td>87</td>
</tr>
<tr>
<td>4.8.3</td>
<td>Pump run-on</td>
<td>87</td>
</tr>
<tr>
<td>4.8.4</td>
<td>Pump cycling</td>
<td>87</td>
</tr>
<tr>
<td>4.9</td>
<td>Frost protection features</td>
<td>87</td>
</tr>
<tr>
<td>4.9.1</td>
<td>System frost protection</td>
<td>87</td>
</tr>
<tr>
<td>4.9.2</td>
<td>Room or building frost protection</td>
<td>87</td>
</tr>
<tr>
<td>4.10</td>
<td>Inspector function</td>
<td>88</td>
</tr>
<tr>
<td>4.11</td>
<td>Manual heating system operation</td>
<td>88</td>
</tr>
<tr>
<td>4.12</td>
<td>Control of district heat transfer</td>
<td>88</td>
</tr>
<tr>
<td>4.12.1</td>
<td>Transfer station</td>
<td>88</td>
</tr>
<tr>
<td>4.12.2</td>
<td>Heat meters</td>
<td>89</td>
</tr>
<tr>
<td>4.12.3</td>
<td>Differential pressure control</td>
<td>91</td>
</tr>
<tr>
<td>4.12.4</td>
<td>Limiting functions</td>
<td>92</td>
</tr>
</tbody>
</table>
### 5. Control of domestic hot water plants

<table>
<thead>
<tr>
<th>Section</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1</td>
<td>Domestic hot water charging with internal heat exchanger</td>
<td>96</td>
</tr>
<tr>
<td>5.1.1</td>
<td>With charge pump and without supply control</td>
<td>97</td>
</tr>
<tr>
<td>5.1.2</td>
<td>Supply control with mixing circuit</td>
<td>98</td>
</tr>
<tr>
<td>5.1.3</td>
<td>Diverting circuit</td>
<td>99</td>
</tr>
<tr>
<td>5.2</td>
<td>Domestic hot water charging with external heat exchanger</td>
<td>100</td>
</tr>
<tr>
<td>5.2.1</td>
<td>Supply controlled with primary side valve</td>
<td>101</td>
</tr>
<tr>
<td>5.2.2</td>
<td>Storage tank supply controlled via primary and secondary side valves</td>
<td>102</td>
</tr>
<tr>
<td>5.3</td>
<td>Special functions of domestic hot water charging systems</td>
<td>103</td>
</tr>
</tbody>
</table>

### 6. Control of heat pump plants

<table>
<thead>
<tr>
<th>Section</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.1</td>
<td>Introduction</td>
<td>104</td>
</tr>
<tr>
<td>6.2</td>
<td>Operating principle of the heat pump</td>
<td>104</td>
</tr>
<tr>
<td>6.3</td>
<td>Heat sources</td>
<td>106</td>
</tr>
<tr>
<td>6.3.1</td>
<td>Heat source: outside air</td>
<td>106</td>
</tr>
<tr>
<td>6.3.2</td>
<td>Heat source: ground</td>
<td>106</td>
</tr>
<tr>
<td>6.3.3</td>
<td>Heat source: ground water</td>
<td>107</td>
</tr>
<tr>
<td>6.4</td>
<td>Heat pump name</td>
<td>107</td>
</tr>
<tr>
<td>6.5</td>
<td>Operating modes</td>
<td>108</td>
</tr>
<tr>
<td>6.5.1</td>
<td>Monovalent operation</td>
<td>108</td>
</tr>
<tr>
<td>6.5.1.1</td>
<td>Special case, monoenergetic operation</td>
<td>109</td>
</tr>
<tr>
<td>6.5.2</td>
<td>Bivalent operation</td>
<td>109</td>
</tr>
<tr>
<td>6.5.2.1</td>
<td>Bivalent alternative operation</td>
<td>110</td>
</tr>
<tr>
<td>6.5.2.2</td>
<td>Bivalent parallel operation</td>
<td>111</td>
</tr>
<tr>
<td>6.5.2.3</td>
<td>Alternative/parallel bivalent operation</td>
<td>112</td>
</tr>
<tr>
<td>6.5.3</td>
<td>Operating mode selection</td>
<td>112</td>
</tr>
<tr>
<td>6.6</td>
<td>Heat pump characteristics</td>
<td>113</td>
</tr>
<tr>
<td>6.6.1</td>
<td>Coefficient of performance ε</td>
<td>113</td>
</tr>
<tr>
<td>6.6.2</td>
<td>Yearly energy coefficient β</td>
<td>114</td>
</tr>
<tr>
<td>6.7</td>
<td>Heat pump controllability</td>
<td>115</td>
</tr>
<tr>
<td>6.7.1</td>
<td>Heat output control directly at the heat pump</td>
<td>115</td>
</tr>
<tr>
<td>6.7.1.1</td>
<td>Hot gas bypass or suction throttling</td>
<td>115</td>
</tr>
<tr>
<td>6.7.1.2</td>
<td>Compressor valve unseating</td>
<td>115</td>
</tr>
<tr>
<td>6.7.1.3</td>
<td>Compressor speed control</td>
<td>115</td>
</tr>
<tr>
<td>6.7.2</td>
<td>Heat pump ON/OFF control</td>
<td>115</td>
</tr>
<tr>
<td>6.7.2.1</td>
<td>Controlled variables for ON/OFF control</td>
<td>116</td>
</tr>
<tr>
<td>6.8</td>
<td>Heat pump operating limits</td>
<td>117</td>
</tr>
<tr>
<td>6.8.1</td>
<td>Vaporization pressure operating limit</td>
<td>117</td>
</tr>
<tr>
<td>6.8.2</td>
<td>Condensation pressure operating limit</td>
<td>117</td>
</tr>
<tr>
<td>6.8.2.1</td>
<td>Determining the maximum permissible condenser inlet temperature</td>
<td>119</td>
</tr>
<tr>
<td>6.9</td>
<td>Heat storage</td>
<td>121</td>
</tr>
<tr>
<td>6.9.1</td>
<td>Heat buffers / storage tanks</td>
<td>121</td>
</tr>
<tr>
<td>6.9.2</td>
<td>Stratified charging and stepped charging of storage tanks</td>
<td>121</td>
</tr>
<tr>
<td>6.9.3</td>
<td>Storage tank charge control</td>
<td>122</td>
</tr>
<tr>
<td>6.9.4</td>
<td>Plants without storage tanks</td>
<td>123</td>
</tr>
</tbody>
</table>

### 7. Control of solar panel plants

<table>
<thead>
<tr>
<th>Section</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.1</td>
<td>Introduction</td>
<td>124</td>
</tr>
<tr>
<td>7.2</td>
<td>Different types of circuits</td>
<td>125</td>
</tr>
<tr>
<td>7.2.1</td>
<td>Solar plant with one solar panel field</td>
<td>125</td>
</tr>
<tr>
<td>7.2.2</td>
<td>Solar plant with two heat exchangers in the storage tank</td>
<td>125</td>
</tr>
<tr>
<td>7.2.3</td>
<td>Solar plant with two solar panel fields</td>
<td>128</td>
</tr>
<tr>
<td>7.3</td>
<td>Control with integration in an overall system</td>
<td>129</td>
</tr>
</tbody>
</table>
1. Heating boiler control

1.1 Introduction

Heating boiler manufacturers specify various limit values, such as:
- Minimum permissible boiler water temperature
- Minimum permissible boiler return temperature
- Minimum flow rate through the boiler

Depending on plant type, various additional requirements must also be met, such as:
- Energy-optimized plant operation
- Return temperature as low as possible in condensing boilers so that condensation can actually take place

A further important point is the burner startup frequency, because short burner running times give rise to poor overall efficiency. This gives rise to a further boundary condition:
- Minimum burner operating time

Partially conflicting conditions

These conditions are partially conflicting. Therefore, for example, a small variation of the boiler water temperature, which is desirable from a control point of view, is not necessarily reconcilable with the requirement for long burner running times.

Various basic solution possibilities for meeting these boundary conditions are explained in the following. Depending on the plant components, types of controllers and systems used, one or other of the solutions may be more suitable, or a combination of the described approaches may have to be considered.

1.2 Boiler water temperature control

A simple and still common way of controlling the boiler water temperature is to use a thermostat. Various types of boiler thermostat may be used, including control thermostats, safety temperature limiters and, sometimes, safety temperature detectors otherwise known as “thermal reset limit thermostats.”

A thermostat consists of a two-position controller with a setpoint transmitter and sensor combined in a single unit.

![Thermostat diagram](image)

Fig. 1-1  Boiler with control thermostat and safety temperature limiter

1. Boiler
2. Burner
3. Control thermostat
4. Safety temperature limiter
5. Two-position controller with setpoint transmitter
6. Sensor
With thermostatic control, the control thermostat (3 in Fig. 1-1) switches the burner on when the boiler water temperature falls below a given value \((x_u)\) and switches it off again when the water temperature reaches the upper switching point \((x_o)\) is reached (Fig. 1-3) Boiler thermostats are normally calibrated to the switch-off temperature (i.e. \(x_o = w\)) and operate with a switching differential \(X_D\) of approximately 6 K. Control thermostats with an adjustable switching differential are also quite common.

![Fig. 1-2 Control thermostat – steady-state characteristic](image)

This produces the following dynamic response:

![Fig. 1-3 Curve for boiler water temperature with thermostat control](image)

To ensure the necessary safety functions, boilers have built-in safety temperature limiters and in some cases, also built-in safety temperature sensors.

The safety temperature **limiter** switches the plant off when the boiler temperature is too high. It has to be released manually, i.e. someone is required to go to the site and physically check the situation which caused the plant to switch off.

The safety temperature **detector** or thermal reset limit thermostat also switches the plant off when the boiler temperature exceeds a predefined limit, but it is automatically switched back on again when the boiler temperature has cooled down sufficiently.

Thermostats are generally designed to be **intrinsically safe**, i.e. if there is a fault (e.g. a disconnected sensor wire) they automatically switch off the boiler.
Control and safety limit thermostat

An additional safety limit thermostat is connected in series with the control thermostat (Fig. 1-5, safety limit thermostat). After tripping, it must be manually reset (when the sensing tube has cooled sufficiently). Such thermostats are generally intrinsically safe, i.e. they shut the boiler down in case of a defect.

Fig. 1-4 Boiler thermostat (possible design)

Fig. 1-5 Double thermostat (with two temperature control/limit thermostats or a safety limit thermostat)

Limit thermostat and safety limit thermostat (for panel mounting)

1.3 Burner stage switching

In order to ensure operation that is stable in terms of control and optimal in terms of energy use, the temperature variations in the primary flow caused by the switching differentials of the boiler controllers should be minimized or, if possible, eliminated.

1.3.1 Switching with boiler thermostats

Where a constantly high flow temperature is not required, reduction of the boiler or flow temperature according to weather conditions gives rise to reduced heat loss (see 1.6). Weather compensation suggests itself especially in cases where the individual burner stages are switched via two-position controllers, which gives rise to higher boiler temperatures at partial load and lower boiler temperatures at full load (Fig. 1-6).
However, the possibility of temperature reduction according to weather conditions is greatly limited by the minimum permissible boiler return temperature. In order for the boiler to be able to produce the required output, the boiler temperature setpoint must be higher than the low-limit controlled boiler return temperature by at least the temperature difference corresponding to the current load.

In order to prevent shutdown or fluctuation of the burner stages at full boiler load, the setpoint should be set approximately a further half of the boiler controller’s switching differential higher (setting according to overshoot s’ in Fig. 1-6) ⇒ longer burner running times.

Example:

Minimum permissible boiler return temperature = 60 °C
Temperature difference between boiler inlet and outlet = 10 K
Boiler controller’s switching differential = 8 K
Boiler temperature setpoint = 60 °C + 10 K + 4 K = 74 °C

Therefore, the minimum boiler temperature setpoint always depends on the minimum permissible boiler return temperature. In order to keep the boiler temperature down, therefore, it is necessary to select as small as possible a temperature difference between boiler inlet and outlet, i.e. based on the maximum permissible boiler flow volume, which usually corresponds to a temperature difference of approximately 10 K. A disadvantage of this is the resultant increased investment costs (controls elements, pumps, valve etc.) and the increased pump energy demand.

The reduced limits for pollutant emissions from boiler plants and the simultaneous efforts to minimize energy consumption have lead, among other things, to the requirement for long burner running times or fewer burner startups. Each burner startup produces additional emissions and increased boiler pollution.
The necessary prepurge times for forced draft burners give rise to additional energy loss through internal cooling, which, in combination with the boiler pollution, contributes to a reduction in the annual utilization ratio. However, longer switching intervals cannot be achieved by the control system alone. An improvement could be made with continuously controlled (modulating) burner operation (see 1.3), or the use of an appropriate storage tank (see 1.8).

1.3.2 Switch-on ratio $\varepsilon$ (epsilon)

Because of the way in which the two-position controller operates, the full thermal capacity is transferred to the controlled plant (e.g. the boiler water system) in the form of pulses (see Fig. 1-3, Burner ON). The pulse duration depends on the load, i.e. the greater the quantity of hot water required from the boiler, the longer the burner will remain on. The switch-on ratio $\varepsilon$ is the quotient of the pulse duration $t_e$ and cycle time $t_z$.

$$\varepsilon = \frac{t_e}{t_z}$$

Based on the switching cycle, the theoretical value of the manipulated variable is equivalent, in the units of the positioning signal, to:

$$y = Y_h \cdot \varepsilon$$

(indicated by the fine line in Fig. 1-7 where $y = 50 \%$), and the average capacity supplied to the system is:

$$P = P_{\text{max}} \cdot \varepsilon$$
1.3.3 Dynamic switching differential \( x_\sigma \) (x sigma)

Every two-position controller has a switching differential, quoted by the manufacturer. This is the static switching differential \( X_0 \) (cf. Fig. 1-2).

In a closed control loop, the fluctuations in the control variable (cf. Fig. 1-6 \( X_t \)) will always be larger than this static switching differential. The actual fluctuation is sometimes referred to as the dynamic switching differential \( x_\sigma \).

The bandwidth of the fluctuation \( x_\sigma \) depends on the controlled system \( (T_t; T_0 \text{ or } T_u; T_g) \) and on the switching differential \( X_0 \) of the controller. The switching frequency must also be taken into account, e.g. in situations with over-frequent burner starts.

The following diagram shows the relationship between the different variables. It is applicable to two-position controllers with no feedback, and to the very common situation of systems operating at 50% load.

![Diagram showing the main characteristics of a two-position controller](image)

**Example for boiler water temperature control:**

\[
X_{SD} = 5 \text{ K} \quad X_\sigma = 25 \text{ K (}\Theta_{HL} - \Theta_{LL}) \Rightarrow X_{SD}/X_\sigma = 0.2 \\
T_t = 1 \text{ min} \quad T = 10 \text{ min} \quad \Rightarrow T/T = 0.1 = S \text{ (degree of difficulty)}
\]

From diagram (see above):

\[
x_\sigma / X_\sigma = 0.28 \Rightarrow \text{bandwidth } x_\sigma = X_\sigma \cdot (x_\sigma / X_\sigma) = 25 \text{ K} \cdot 0.28 = 7 \text{ K}
\]

i.e. the boiler water temperature in this system will fluctuate by 7 K, not merely by the switching differential \( X_{SD} \) of 5 K.

From diagram (see above):

\[
f \cdot T = 0.85 \Rightarrow \text{Cycle time } t_z = T / (f \cdot T) = 10 \text{ min} / 0.85 = 11.8 \text{ min}
\]

i.e. under these conditions, the burner will switch on approximately every 12 minutes.
1.3.4 Switching with integral $\Sigma K \cdot \text{Min.}$

A further possibility for switching burner stages (or sequence-controlled boilers, see chapter 2) is switching based on temperature deviation from the setpoint over time.

Only the most important approaches will be described in the following. Advanced control systems provide more comprehensive and complex solutions, which are described in the appropriate technical documentation.

Simplified operating principle

If the boiler water temperature falls by more than the switching differential SD, the integration of the temperature differential $\Delta T$ in K over time $\Sigma (K \cdot \text{Min.})_1$ begins. Additionally, the respective locking time ($t_{\text{min},1}$) for switching the 2nd stage begins.

The 2nd stage is switched when the locking time has expired and the defined integral value is reached. It continues to run until the desired boiler temperature w is achieved again.

When the 2nd stage is switched off, the switch-off integral for the 1st stage is activated $\Sigma (K \cdot \text{Min.})_2$. When the predefined value is reached, stage 1 is also switched off.

Additionally, such concepts define upper and lower limits at which the stages must be on or off in the sense of a safety circuit.

As can be seen from Fig. 1-9, the various values for the integrals and locking time(s) must be very well adapted to each other and to the system conditions. Control systems provide the possibility of logging important values (e.g. switching frequencies, running times, etc.). This allows the selected values to be optimized on the basis of operating behavior.
Compared with thermostatic control, this solution has the advantage that the output stages are not simply switched on and off on the basis of a temperature differential, but that the system also takes account of the behavior of these temperature changes over time.

1.3.5 Modulating burners
Modern boilers are often equipped with modulating burners whose output can be continuously regulated. Such burners are becoming increasingly available for smaller outputs, and in some products, continuously regulated burner operation is even possible to below 20% output. This enables the boiler water temperature to be controlled within narrower limits over a broad operating range. It is only below this limit that the output still has to be controlled according to one of the methods described above.

In order to be able to achieve sufficiently long running times even below this limit (especially in the case of relatively large boilers) a sufficiently large storage tank suggests itself as a simple but effective solution (see 1.8).

1.4 Systems without boiler pumps
1.4.1 Load behavior
In a system without a boiler pump serving several consumers, the boiler flow temperature $\vartheta_{KV}$ is constant. It should be noted that this “constant value” is the desired, theoretical supply temperature, because, depending on burner control, it fluctuates to a greater or lesser degree in actual operation (see 1.3).

Fig. 1-10 Boiler with several consumer groups (mixing circuit); load diagram
The load diagram (Fig. 1-10) shows the progression of the boiler flow and return temperatures \( \theta_{KV} \) and \( \theta_{KR} \) as well as the respective flow and return temperatures \( \theta_{VL} \) and \( \theta_{RL} \) of the heating groups in the mixing circuit. The progression is also shown by approximation in this diagram.

The actual progression depends on the type of radiators used, and it is slightly curved in the lower range because of the influence of the mean temperature difference on heat output. Fig. 1-11 shows a typical heating characteristic curve for radiator heating.

The boiler return temperature corresponds to the return temperature of the heating groups, i.e. \( \theta_{KR} = \theta_{RL} \).

In the system shown in Fig. 1-10, the heating groups (mixing circuit) transport the necessary boiler flow water through the boiler by means of their own pumps.

The result is that the volume of water flowing through the boiler is matched to the required load.

In order to examine in greater detail how the water volume varies over the heating load range from 0 – 100 %, the temperature characteristic of a radiator heating system showing the actual flow and return temperature curve as in Fig. 1-11 is required.

![Temperature characteristic of a radiator heating system](Fig. 1-11)

The volume of water transported through the boiler at partial load can be determined from the temperature ratios as follows:

Boiler water volume at partial load = \( \frac{\theta_{VL} - \theta_{RL}}{\theta_{KV} - \theta_{RL}} \times 100 \% \)

The flow and return temperatures of the heating group can be determined from the temperature characteristic Fig. 1-11. At 50 % load, the flow temperature is 60 °C, and the return temperature 50 °C.

Boiler water volume at 50 % load = \( \frac{55 \, ^\circ\text{C} - 45 \, ^\circ\text{C}}{80 \, ^\circ\text{C} - 45 \, ^\circ\text{C}} \times 100 \% \)

at 50 % load = 29 % boiler water volume

This can now be transferred to a boiler water volume / load diagram as shown in Fig. 1-12. Further boiler water volumes in different load states can be determined and transferred in the same way.
Fig. 1-12 shows that the curve is not linear, but it becomes increasingly steep with increasing load. The curvature becomes greater the smaller the temperature differential between the flow and return temperatures becomes and/or the lower these are set in relation to the boiler water temperature (e.g. 55/45 °C).

The reduced volume of water flowing through the boiler has the effect of reducing the burner running times. Therefore, this is not an ideal circuit, although it is still frequently encountered in relatively small systems (e.g. in single-family homes).
1.5 Systems with main pump and pressure less header

1.5.1 Load behavior

In case of a system where the boiler is equipped with its own boiler pump and the consumers are connected to a pressure less header (see Fig. 1-13), the boiler return temperature $\theta_{KR}$ is no longer directly dependent on the return temperature of the heating groups, as shown under 1.4.

The load diagram (Fig. 1-13) shows that the boiler flow temperature $\theta_{KV}$ still has a constant progression, and the flow and return temperatures ($\theta_{VL}$ and $\theta_{RL}$) of the heating groups also display the familiar progression (from an ideal point of view).

The boiler return temperature $\theta_{KR}$ rises continuously with decreasing load. It finally reaches the boiler flow temperature $\theta_{KV}$ when the entire water volume circulates through the boiler via the bypass.
In the case of heating boilers, it is customary for the boiler temperature to be weather-compensated, i.e. dependent on the outside temperature. This reduces the boiler losses and means that the controls of the heating groups in the circuit only have to provide fine control to the group flow temperature. Standard heating controllers today usually provide boiler temperature compensation as standard.

Fig. 1-14 shows the progression of the boiler flow and return temperatures ($\vartheta_{KV}$ and $\vartheta_{KR}$) as well as the flow and return temperatures ($\vartheta_{VL}$ and $\vartheta_{RL}$) of the heating groups over the entire load range. The boiler flow temperature $\vartheta_{KV}$ should always be higher than the highest group flow temperature $\vartheta_{VL}$. The lowest boiler return temperature $\vartheta_{KR}$ should be approx. 5 K above the minimum permissible return temperature $\vartheta_{RL,\text{min.}}$.

However, this compensation is only meaningful if the low-limit control does not have to be set very high because of the boiler return temperature control (see 1.7). Additionally, the periodic run-up of the boiler temperature for domestic hot water charging also has a disadvantageous effect (see chapter 5).

![Boiler water temperature compensation](image-url)
1.7 Maintained boiler return temperature

**Boiler corrosion**

The prevention of boiler corrosion due to the flue gases is a special problem in heating systems. This kind of corrosion occurs when the flue gases are cooled below the dew-point temperature of the absorbed water vapor causing it to condense. Since, with the exception of purified natural gas, all fuels contain a greater or lesser amount of sulphur, sulphuric acid can form in the condensate, which corrodes the boiler walls.

The dew-point temperature of the absorbed water vapor in the combustion gases of extra-light and light fuel oils is between 40 and 50 °C. In order to minimize boiler corrosion, it must be ensured that the temperature in the area of the combustion chamber walls does not fall below the boiler water temperature specified by the manufacturer, e.g. 55 °C. Therefore, it is necessary that:

- the actual boiler water temperature is always above this limit
- no boiler return water at temperatures below this limit ever reaches the combustion chamber wall area.

**Non critical in modern, low-temperature boilers**

This problem is no longer as critical in modern, low-temperature boilers, because they are made of corrosion-resistant materials. Therefore, the boiler water temperature can be continuously reduced to 40 °C or even less according to the outside temperature.

**Solutions**

In the case of non-corrosion-resistant boilers, an attempt is made to prevent the occurrence of harmful temperatures in the boiler by selecting an appropriate hydraulic circuit. However, this is insufficient in case of a large and rapid load increase (e.g. during startup and morning boost). Additionally, a separate controller must be provided to limit the boiler return temperature.

The following solutions are suitable for this purpose:

- Modulating limit controller acting on the consumer groups
- Modulating limit controller acting on a separate controlling element
- Boiler mixing pump with thermostat (not recommended)
1.7.1 Limiting control acting on the consumer groups

This limiting circuit is suitable for systems in which all controllers and controlling elements are installed centrally, i.e. in the distribution room, or, that not being the case, the return temperature of the external groups or sub headers is kept to a high level by local limit controllers. Furthermore, there must be no manually controlled groups or groups with no possibility of intervention.

**Prerequisite**
The prerequisite for low-limit control of the boiler return temperature is a pump in the boiler circuit which enables water from the boiler flow side to be directly mixed with the boiler return.

**Operating principle**
The limit controller prevents the boiler return temperature from falling below the setpoint by:
- locking the opening commands for the controlling elements in the consumer groups supervised by the controller,
- slightly closing controlling elements that are open too far so that an excessively large flow volume of cooled consumer return water does not enter the boiler circuit.

The hot boiler water required for the return temperature boost is supplied to the boiler return – depending on the hydraulic circuit – via the bypass of the pressure less header, the bypass lines of the controlling elements or the boiler mixing pump (see 1.7.3). In extreme cases, e.g. in plant startup mode, no consumer return water flows back to the boiler at all. However, as soon as the boiler return temperature rises above the setpoint with increasing plant heating, the group controls are gradually enabled by the low-limit controller.
Therefore, the delayed heatup of the consumer groups is tolerated in the interest of boiler protection. However, this is generally insignificant, because the start of the heatup phase can be brought forward if necessary.

In practice, however, it is not necessary for the low-limit controller to intervene in all group controls by “rationing” boiler water during startup. As many groups as the boiler’s output can support during startup at the lowest outside temperatures can be excluded from limiting control (possibly with staggered start-up). Especially air heating coils in ventilation and air conditioning systems should be excluded from boiler water “rationing,” because they depend on an unrestricted heat supply at all times in order to prevent freezing and a deterioration of comfort levels.

The proper functioning of maintained boiler return temperature depends not only on the type of return temperature control. It also requires correct sizing of the controlling elements in the individual consumer groups and of the bypass (depending on the type of header). The following example shows some considerations in this regard.

The system shown in Fig. 1-16 operates with a boiler flow temperature of 80 °C. The two heating groups operate with flow/return temperatures of 60/40 °C. Additionally, the minimum return temperature to the boiler must not fall below 55 °C according to the manufacturer’s specification.

In order to enable the group valves and boiler bypass to be correctly sized, and to enable hydraulic balancing to be performed properly, the following information is important:

- Water volumes in the hydraulic circuits of the heating groups
- Decisive temperature difference for valve sizing
- Maximum water volume for which the bypass should be sized and adjusted.

![Fig. 1-16 Maintained boiler return temperature; valve and bypass sizing](image-url)
The hydraulic circuit of the heating groups is an injection circuit with a three-port valve, i.e. the group pump delivers a constant mass flow. Since the maximum flow temperature $\theta_{vl} = 60 \, ^\circ C$ is way below the boiler flow temperature $\theta_{kv} = 80 \, ^\circ C$, the heating group and its valve and bypass can be sized accordingly. The maximum required boiler water volume as a percentage of the water volume delivered by the group pump can be calculated from the temperature ratios (in the design state) at the mixing point of the heating group:

$$\text{Maximum boiler water volume} = \frac{\theta_{vl} - \theta_{rl}}{\theta_{kv} - \theta_{rl}} \times 100\%$$

$$= \frac{60 \, ^\circ C - 40 \, ^\circ C}{80 \, ^\circ C - 45 \, ^\circ C} \times 100\%$$

$$\text{Maximum boiler water volume} = 50\%$$

The decisive temperature difference for valve sizing is calculated from the boiler flow temperature and the return temperature in the design state.

$$\Delta T = \theta_{kv} - \theta_{rl} = 80 \, ^\circ C - 40 \, ^\circ C = 40 \, K$$

The maximum water volume flowing through the boiler bypass can be determined by the same method as for the heating groups:

$$\text{Maximum water volume in boiler bypass} = \frac{\theta_{krmin} - \theta_{rl}}{\theta_{kv} - \theta_{rl}} \times 100\%$$

$$= \frac{55 \, ^\circ C - 40 \, ^\circ C}{80 \, ^\circ C - 45 \, ^\circ C} \times 100\%$$

$$\text{Maximum water volume in boiler bypass} = 37.5\%$$
1.7.2 Limiting controlling acting on a separate control element

In systems with numerous, widely spaced controlling elements or in cases where several controlling elements are connected outside of the header (e.g. in community heating systems), it is more appropriate and less expensive to assign a separate controlling element to the limit controller. The advantage is to be found in the simpler electrical installation, because intervention in the individual heating groups is not necessary. If required, it is also possible to use this controlling element for primary control of the main flow temperature (e.g. for a district heating line).

This limiting circuit is also suitable if manually controlled heating groups, groups without the possibility of intervention or sub headers without local low-limit control are connected to the header as well as in systems with a variety of control systems (electronic, pneumatic) or if controllers from a variety of manufacturers are used.

The limit controller prevents the boiler return temperature from falling below the setpoint by setting the separate controlling element such that – depending on header design – less cold consumer return water or more hot boiler water is supplied to the boiler return. Fig. 1-17 shows an appropriate circuit for pressure less header.

Constant mixing with maintained boiler return temperature

If in Fig. 1-17 the boiler return temperature is below the necessary value even at steady state, an additional constant mixing circuit should be installed ⇒ smaller valve than if the entire water volume circulates via the bypass. This makes intervention by the controlling element with the associated reduction in output only necessary in case of sudden, large load increases (e.g. heatup after reduced operation). For installation purposes and for reasons of cost, a fixed boiler bypass should be installed whenever the permanently mixed boiler water volume exceeds 30%.

Fig. 1-17 Maintained boiler return temperature with separate three-port valve (constant bypass if required)
Except in the case of the pressure less header, where the placement of the bypass provides for the selection of a so-called startup priority, as described in the next section, these circuits have the disadvantage that the output of all consumer groups connected to the header is restricted while the limiting function is active. Since in the case of air heating coils, this can give rise to a risk of freezing and a deterioration of comfort levels in the conditioned space, these circuits should not be used if ventilation and air conditioning systems are to be connected to the header.
As already mentioned, startup priority can be hydraulically controlled by the placement of the bypass in the pressure less header.

If the bypass is installed at the front end of the header (Fig. 1-19, A), there is no longer sufficient boiler water for the consumers if the low-limit control intervenes. Therefore, the group pumps draw the lacking water from the return collector via the bypass, which gives rise to a mixture of boiler water and cold return water in the header. Therefore, all heating groups connected to the header are heated with the same reduction in this case as well. Since the limiting control valve opens increasingly as the temperature of the system and, therefore, of the boiler return rises, the proportion of boiler water in the flow distributor increases. By the time the low-limit controller has fully opened the control valve, the full boiler output is available to the consumers.

If, however, consumer groups with start-up priority are to be supplied by the pressure less header, e.g. ventilation groups, which have to reach their full output as soon as possible on startup (frost hazard, comfort), they are connected at the front end of the header, and the bypass is positioned behind them (Fig. 1-19, B and C). During the limiting control phase, the groups placed ahead of the bypass receive more boiler water the closer they are to the boiler. Those behind the bypass receive a mixture of the remaining boiler water and water from the return collector.

If the bypass is at the back end of the header (Fig. 1-19, D), only the groups at the front end receive the desired boiler water during the limiting control phase. The other groups draw water from the return collector via the bypass. These groups heat up with a long delay. If, for some reason, the bypass has to be placed at the back end of the air header, heating coils must be located at the front end, and radiators and floor heating systems further back.
1.7.3 Maintained boiler return temperature via boiler mixing pump with thermostat

In the circuit shown in Fig. 1-20, a circulating pump is installed in a bypass between the boiler flow and return. When it is in operation, it mixes hot water from the boiler flow with the cold consumer return, thus raising the boiler return temperature.

![Diagram of boiler return temperature with boiler mixing pump](image)

**Careful pump sizing**  
The boiler mixing pump must be carefully sized. It should be configured for the flow volume required to raise the boiler return temperature to the desired minimum value. In order to save electrical energy, the pump can be switched on and off by a thermostat which acquires the consumer return temperature. The thermostat must be installed upstream of the inlet of the pump pipe into the boiler return pipe (Fig. 1-20, A) in order to prevent the pump from continually switching on and off. If the boiler return temperature falls below the minimum value, the thermostat switches the pump on. It remains on until the consumer return temperature rises above the minimum value again. A check valve in the bypass pipe prevents undesirable water circulation when the boiler mixing pump is off.

**Circuit not recommended**  
This simple circuit, however, can only keep the boiler return temperature above the desired minimum value under steady-state conditions and during slow load increases. In case of a rapid load increase, boiler water mixing is ineffective, because the temperature of the boiler water itself decreases during temporary full-load operation. It only becomes effective again when the total volume of water in the system has been sufficiently heated. Since this heat-up period can take a relatively long time in large-scale systems, this circuit is not recommended.
1.8 Heating boiler with storage tank

In many boiler designs, the water volume of the boiler is very small in proportion to its output. Therefore, it can be helpful to operate a boiler with a storage tank in order to increase burner running times and, therefore, to improve boiler efficiency. The sizing of the storage tank depends on the specific boundary conditions of the system.

In a system as shown in Fig. 1-21, the boiler is switched on and off according to demand by the sensors with two-position controllers (or storage tank thermostats) in the storage tank.

The boiler flow setpoint should be slightly higher (approx. 2-5 K) than the storage tank temperature. The boiler supply is regulated to the desired temperature (no control thermostat) in order to enable stratification in the storage tank. This also safeguards the boiler inlet temperature (boiler produces $\Delta \theta$).

The hydraulic circuit of the heating groups should ensure that they supply a low return temperature in order not to disturb the stratification in the storage tank.

![Fig. 1-21 Heating boiler with storage tank](image1)

![Fig. 1-22 Heating boiler (with gas burner) and storage tank](image2)
1.9 Boiler with flue gas heat exchanger

In plants of this type, the flue gas is directed through a downstream heat exchanger. In the process, the flue gas is cooled to below the condensation temperature. This allows a portion of the condensation heat as well as the remaining sensible heat of the flue gas to be utilized in order to preheat the boiler water (increased efficiency). The flue gas heat exchanger must be operated with a low return temperature, which makes appropriate hydraulic circuits necessary on the producer and consumer side.

The condensate is corrosive (especially where fuel oil is used; sulphuric acid H₂SO₄ is produced) so appropriate materials must be used not only for the heat exchanger but also for the boiler and stack. For this reason, separate heat exchangers were mainly used in the early stages of development of this technology (from 1980). Nowadays, various standard boilers (for gas and oil), so-called condensing boilers, which have an integrated heat exchanger and which have been optimized in terms of materials and useful heat output are available on the market.

![Fig. 1-23 Boiler with integrated flue gas heat exchanger and external heat exchanger](image)
1.10 Demand-controlled heat generation

Systems with a heating center and several substations are operated in an energy-optimized manner, if possible. In order to achieve this, the individual substation controllers are connected with the heating center in such a way that heat demand signals, setpoint boosts etc. can be communicated to the heating center via a communication network (see Fig. 1-24) or other connection (e.g. relay bus). Modern controls offer a wide variety of possibilities which are described in the respective technical documentation.

The following shows an example of a residential estate with communication networks for coordinating energy production and distribution (e.g. local process bus LPB) and for energy evaluation (e.g. SYNERGYR bus) between the heating center and the individual substations (houses).

Fig. 1-24 Example of a communication network on a residential estate

- LPB junction box
- bus junction box – sealed (e.g. SYNERGYR)
- LPB data bus
- Heat meter bus
2. Control of multiple boiler plants

2.1 Introduction

At first glance, the control of multiple boiler plants appears to pose no significant problems. Oil or gas-fired boilers using water as the heat transfer medium provide controlled systems with degrees of difficulty that can be easily mastered. Load adjustment is also apparently logical: If the heat production that is in operation is no longer sufficient, a lag boiler is brought online, and it is taken offline again when it is no longer needed. If, however, the requirements of energy-optimized operation, high availability and long service life are taken into account, the result is a very sophisticated problem that can only be solved with correct hydraulic integration of the boilers and intelligent control means. This chapter covers the hydraulic and control engineering concepts applied today as a decision-making aid for the planning of multiple boiler plants.

Limitation

The problem can be covered sufficiently well on the example of dual-boiler plants, because the same problem arises in all multiple boiler plants between the boiler(s) in operation and the boiler to be brought online or taken offline. An appropriate indication will be given if a system is not suitable or is highly suitable for plants with more than two boilers.

2.2 Hydraulic parallel or series connection

Parallel connection

In case of parallel connection (Fig. 2-1), the return temperature to all boilers is the same. If each boiler is controlled by its own thermostat (two-position controller), the output is divided on the basis of volume flow amongst the boilers that are online, i.e. if consumer load is 40 % and there are two identical boilers, for example, the output is divided equally between them at 20 % each. Parallel connection is selected more frequently than series connection, not only because of the simpler piping, but also because it can prevent the lag boiler from having to be operated with poor efficiency at low loads. However, parallel connection requires exact balancing of the water flows, which must be proportional to the output share of the individual boilers in order for each boiler to be able to achieve its full output.

Fig. 2-1 Multiple boiler plant with parallel connection
In case of series connection (Fig. 2-3), the return temperature to all boilers is not the same. The flow temperature of the lead boiler can be the return temperature of the lag boiler. The boilers provide different output shares.

If the consumer load exceeds the full load output of the lead boiler (B1), it remains at full load after the lag boiler (B2) comes online, whereas the lag boiler must start at low load. If the consumer load falls back below the capacity of the lead boiler, the result – in the case of thermostatic control of the individual boilers – is uncontrolled switching of the two boilers. Series connection is especially suitable if a heat producer that requires low return temperatures (gas boiler with flue gas condenser, or heat pump) is connected upstream of a conventional boiler.
2.3 Demands on boiler sequence control

The requirements for energy-optimized and environmentally sound boiler sequence control can hardly be met with hardware alone. However, digital control technology, in combination with optimization functions, enables plant operation with minimum emissions, low energy consumption and high availability. The major control and optimization functions are described in the following.

Bringing the lag boiler online and taking it offline again basically involves the following functions:

- Opening/closing the boiler shutoff valve
- Activating/deactivating boiler temperature control
- Starting/stopping the boiler pump
- Release/locking the burner control

Optimal boiler switching meets the following requirements:

Each boiler should

- Be brought online in good time in order to ensure uninterrupted heat supply
- Not be switched too frequently in order to avoid unnecessary start-up and shutdown losses
- Remain online at least long enough for the acidic condensate in the combustion chamber and flueways to dry out completely (corrosion prevention)
- Only be brought online when it is genuinely needed

The question of whether or not a lag boiler is genuinely needed is not decided solely on the basis of the static deviation between actual and set point temperature, but by calculating the integral of the deviation over a defined locking time.

It is also useful to consider the current operating state of the lag boiler:

- A warm boiler can be brought online more quickly than a cold one.
- In the case of boilers whose highest efficiency is in the partial load range, the lag boiler should, wherever possible, be brought online at a given partial load point.
- The lag boiler should not be taken offline again until the lag boiler can definitely cover the load on its own.
- In case of a fault in one boiler unit, switchover to another boiler should occur automatically.
- A defective boiler must also be hydraulically disconnected from the system.

In the case of boost heating, water heating and other heatup processes with a large, temporary heat demand, the boiler temperature set point should be immediately raised to its maximum value, and – depending on the overall load situation – the lag boiler should be brought online without delay.

Depending on the plant concept, boiler type, selected fuel, output distribution or hydraulic circuit, etc., either automatic or manual changeover of the boiler priority may be preferable.

The efficiency of modern boiler designs is relatively constant in the mid and upper output range, but it falls off rapidly in low-load operation. Therefore, the output distribution between the individual boilers should be as even as possible so that each boiler can operate in an optimal efficiency range.
If no energy is required in intermediate seasons or, during night setback, the lead boiler should also be taken offline. A suitable control variable for this is the current outside temperature $\theta_{AA}$ (see 4.4). The boiler(s) can be brought back online on the basis (together with other criteria) of the attenuated outdoor temperature $\theta_{AM}$ (cf. 4.4). The appropriate attenuation time constant (approx. 15 to 30 hours) must be matched to the heat storage capacity of the building.

2.4 Sequence control changeover criteria

The sequence control of multiple boiler plants can be accomplished according to various criteria. The question as to which are the most appropriate criteria depends on the plant situation and operating conditions/requirements, and must be decided individually in each case.

**Commonly used changeover criteria**

The most common sequence control changeover criteria are as follows:
- Manual control
- Outside temperature
- Boiler temperature
- Change in load based on control factor $\varepsilon$
- Consumer flow temperature
- Common boiler return temperature
- Maximum supply and return temperature
- Storage tank temperature(s) or storage tank charging
- Burner load (in case of modulating burners)
- Modulation factor of condensing boilers in operation

**Additional changeover criteria**

It is also possible, especially with DDC systems or modern heating group controllers, to implement demand-dependent sequence control based on actual load. For example, the position of the furthest open valve can be taken as a criterion for bringing an additional output stage (burner or boiler) online.

However, other criteria, such as domestic hot water charging, heating boost, changeover in the event of fault, etc., can also be used for sequence control.

### 2.4.1 Manual control

The temperature of both boilers is controlled, as in single boiler plants, autonomously by the respective boiler temperature controllers, and the lag boiler is brought online and taken offline by means of a manual switch. This type of sequence control may seem obsolete in the age of automation. In large-scale industrial or commercial systems, however, where qualified staff is required to manage the system around the clock anyway, the staff are usually highly experienced and equipped with information on forthcoming demand trends (weather forecast, information on the imminent startup or shutdown of heat consumers, etc.).

So, while this is not an automatic method of sequence control, it is still an intelligent one.

### 2.4.2 Sequence control based on outside temperature

In the case of boiler plants whose heat output is used to more than 90% for space heating, the outside temperature is a suitable reference variable for sequence control. However, the “attenuated” outside temperature must also be used in this case, because the current outside temperature fluctuates too rapidly, which can cause the lag boiler to be brought online and taken offline too briefly at the heating limit. The outside temperature can also be used for locking the lag boiler in case temporary load peaks can occur at outside temperatures above 0 °C.
2.4.3 Sequence control based on boiler temperature

Thermostatic control

This type of thermostatic control is subject to the same problems as already explained with regard to burner stage switching. The next output stage (in this case the lag boiler) must not be brought online until the temperature of the lead boiler has fallen by the amount of the defined switching interval. The necessary switching intervals and differentials automatically give rise to relatively large temperature fluctuations in the main flow. Additionally, such circuits have a very strong tendency to oscillate, which cannot be sufficiently prevented even with the use of complex timing elements. Excessively long switch-on delays often cause a major temperature drop in the lead boiler, which can give rise to harmful flue gas condensation. Finding that the boiler temperatures are controlled to lower values at full load than at partial loads, the operating personnel are often tempted to set the set point of the lag boiler to the same values as the lead boiler. This intervention has the desired effect at full load, but in partial load operation, both boilers switch on and off simultaneously!

Fig. 2-4 Sequence control based on boiler temperature with boiler thermostats and parallel connection

1 Control thermostat
2 Safety temperature limiter

Boiler temperature controllers

Digital boiler temperature controllers provide more elegant solutions to this problem: They allow, for example, any number of switching steps to be operated at a given set point and within the selected switching differential. If the actual value of the boiler temperature falls below the lower switching point, the control deviation is multiplied by the duration, which is mathematically equivalent to an integral \( \Sigma (K \cdot \text{Min.}) \). If the integral exceeds the definable minimum value after the elapsing of a locking time, the active output stage is fixed, and it is controlled with the next higher stage. In the case of decreasing load, an equivalent opposite procedure is applied (cf. 1.3.4).

Using digital technology, it is even possible to divide the continuous correcting span of a modulating burner into individual output stages and to delay the transition from one stage to the next via an inhibit integral. This can be used to prevent a boiler flow temperature control with PI action from changing the burner output too quickly.
2.4.4 Sequence control based on a change in load and control factor $\varepsilon$

The boilers each have their own boiler temperature control system. Additional boilers are brought online or offline on the basis of the maximum boiler output (absolute or relative) and the continuously measured control factor $\varepsilon$ (cf. 1.3.2). In the boiler sequence controller, a full audit of the overall output is maintained, and the boilers which are brought online are those capable of satisfying the heating demand over the long term with the maximum possible efficiency.

Fig. 2-5  Sequence control with boiler temperature sensor and control factor

1  Boiler temperature sensor
2  Safety temperature limiter
3  Boiler sequence controller

2.4.5 Sequence control based on consumer flow temperature

In the case of thermostatically controlled boilers, the consumer flow temperature (measured in the main flow to the header) is only suitable as a switch-on criterion Fig. 2-6 If the active output stage is no longer sufficient, it falls below the controller set point, signaling additional demand. Even in low-fire operation, however, thermostatically controlled boilers share the required output equally amongst themselves, which results in intermittent operation. The flow temperature is kept to the controller set point, providing no information on whether additional boilers that have been brought online can be taken back offline again.

With a step controller with P-action, burner switching and the activation and deactivation of the lag boiler is performed directly in relation to temperature, but the proportional offset and switching intervals give rise to major flow temperature fluctuations. Therefore, this type of control is unsuitable for more than two boilers. The only advantage of P-control is the slightly longer burner running times.

An improvement can be achieved with a PI controller in combination with a step controller Fig. 2-7 The advantageous property of control without deviation from the set point, however, deteriorates due to the necessary use of a step switch. But the flow temperature fluctuations can be kept within narrower limits. Disadvantages are more frequent burner cycling and the fact that plants with PI-control are more susceptible to oscillation. In order to prevent this, switch-on and switch-off delay elements must always be incorporated.
Another point to note with this type of control is the fact that, when the lag boiler is brought online, its flow temperature is immediately raised to the controller set point, and, therefore, the boiler return temperature also rises accordingly via the bypass. Since the lead boiler remains in high-fire operation, its flow temperature also rises parallel to the return temperature (see temperature/load diagram). In this case, it is important to ensure that sufficient distance from the safety limit temperature is maintained. In digital systems, the flow temperature set point can be reduced when the lag boiler comes online, then it can be gradually raised to its original value.

The plant can be operated with a constant or weather-compensated set point. Depending on the plant’s load behavior, this may require major adjustment effort. In order to prevent flue gas condensation, an excessive decrease in the boiler temperature must be prevented. Depending on the plant concept, outside temperature-dependent locking of the lag boiler or of individual burner stages can be applied. This type of control is suitable for gas-fired boiler plants with atmospheric burners and parallel hydraulic connection, because frequent switching is perfectly acceptable in this case.

Fig. 2-6 Sequence control based on flow temperature (1) with PI-controller (2) and step controller with four steps (3); boilers connected in parallel – with safety temperature limiter (4)
A significant improvement in terms of control can be achieved with a hydraulic connection in series (Fig. 2-7) because the PI controller can control the flow temperature via the mixing valves without a set point deviation. Here too, delay elements must be used to prevent the lag boiler from being brought online or taken offline too early. Two-stage burners are especially suitable, because (like the lag boiler) the second burner stage is switched via the control valve signal. The boiler temperature is controlled by the boiler temperature controller, with a fixed or weather-compensated set point.

This type of control is independent of the number of boilers, so it is especially suitable for plants with more than two boilers. To reduce the power consumption of the main pump, the use of a multispeed pump is recommended. Priority selection by reversal of the sequence is also possible with a hydraulic series connection (regardless of the number of boilers).

Fig. 2-7 Sequence control based on flow temperature using a PI sequence controller; boilers connected in series

**2.4.6 Sequence control based on common boiler return temperature**

Because of its direct dependence on load, the boiler return temperature is an ideal switching criterion. However, practical experience has shown that this type of control is very sophisticated in terms of the setting values and the “comprehensibility” of the plant, and is therefore highly demanding in respect of commissioning. Fig. 2-8 is an example of an application with boilers that are hydraulically connected in parallel, and single-stage burners. There are also systems in operation with two-stage burners, and these function perfectly well, but because of the direct dependence on the load, their commissioning requires greater effort (see temperature/load diagram).

In order to prevent oscillation, the usual time delays are also incorporated here. The proportional offset gives rise to relatively long burner running times. Boilers with large water volumes are advantageous, because they can be used as accumulators. The flow temperature can briefly fall to the return temperature set point, which also has a positive effect on burner running times.

In plants where the boilers are hydraulically connected in series and equipped with mixing valves Fig 2-9 the control sensor can be placed in the boiler return instead of in the boiler flow. This allows the advantages of continuous sequence control to be combined with those of return temperature control.
Control according to return temperature requires, especially with parallel connection, careful hydraulic sizing and balancing of the boiler water flows during commissioning. Unequal boiler outputs have a disadvantageous effect on priority changeover (volume flow changes require a changeover to other set points). This type of control is less suitable for more than two boilers.

Fig. 2-8 Sequence control based on return temperature using a P-controller with two-step switch; boilers connected in parallel
2.4.7 Sequence control based on the maximum flow and return temperature

The sequence control described under 2.4.5 has the disadvantage that the main flow temperature to the header decreases as soon as there is no further heat demand from the consumers. This cooling is detected by the sensor and gives rise, via the control, to an unnecessary increase in boiler output until the boiler limit thermostats switch off the burners.

This operating state can be prevented by placing a second sensor in the boiler return, where the higher of the two sensor values is fed to the controller (Fig. 2-9). If the heat demand decreases to zero, boiler water no longer flows into the consumer flow but directly into the boiler return. This means that the boiler return temperature rises above that of the main flow to the header, causing the controller to shut off the burners.

Fig 2-9  Sequence control based on maximum flow or return temperature using a PI-controller with a step controller with four-steps; boilers connected in parallel – sequence control after storage tank charging.
If a heating system is designed with two or more boilers, the water volume delivered by the boiler pumps under partial load conditions can be considerably greater than the total volume drawn off by the consumer circuits. In order to avoid major fluctuations of pressure and flow volume in the boiler circuit and the associated negative effects on the consumer circuits, the boiler circuit is normally short-circuited using a bypass between boiler supply and return (Fig. 2-1 and Fig. 2-3). On start-up, boiler plants with return temperature control initially supply no heating water at all to the consumers, and then, for some time afterwards, they supply too little. The consumer groups draw the volume of water they need to make up the deficiency from the cold return pipe via this bypass. In this phase, the measuring sensor must be placed in the main supply pipe, between the bypass and the manifold in order to be able to detect the heat demand. At low load, however, when both the boiler circuit and the consumer circuits are operating almost entirely in short-circuit mode, there is practically no water circulation. The pipe cools down, and the sensor initiates a heat demand signal, although the boiler output should actually be switched back. In this phase, therefore, the sensor should again be located in the boiler circuit.

The control solution to this problem is based on the location of separate sensors in the main supply to the manifold and in the common return to the boilers. The supply sensor is used to control the main supply temperature and bring the boiler output online, and the return sensor is used to initiate the reduction of the boiler output as soon as its measured value exceeds that of the supply sensor.

Fig. 2-10 Parallel connected multiple boiler plant with return low-limit control and "hydraulic switch" (system header)

1 Hydraulic switch
2 Sensor to measure load status of the plant
3 Sequence controller with step controller and time elements
4 Safety temperature limiter
5 Controller for low return temperature protection
The bypass serves not only to ensure the necessary flow volumes in the producer and consumer circuits but also to provide genuine hydraulic decoupling of the two circuits. If a maximum flow velocity of 0.2 m/s at nominal flow volume is selected for the bypass, the resultant pipe diameter is so large that it could almost be referred to as a bypass duct. Clever sensor placement in the upper part of this duct provides for load-dependent switching of the boiler output levels using a single sensor only. This type of “hydraulic switch” is regarded, primarily in Germany as a standard solution. Elsewhere, a bypass pipe of the same nominal diameter as the main supply pipe is generally considered sufficient.

2.4.9 Sequence control after storage charging

Assuming selection of the right switching criteria and use of the appropriate control strategies, the methods described above produce reliable, well-functioning systems.

However, the efforts to extend burner times, and so to minimize the number of burner switching operations, are not wholly successful and are primarily achieved at the expense of severely fluctuating boiler temperatures.

Where there is a demand for optimum results, these can only be achieved by use of a storage tank. Fig. 2-11 shows a variant of this system. From the hydraulic viewpoint, the more cost-effective parallel connection can be selected, because with this strategy, there are no benefits to be reaped from a connection in series. Single, or possibly two-stage burners may be used, but not modulating burners. Single-stage burners can offer advantages in terms of combustion, provided they are properly matched to the boiler.

Fig. 2-11 Sequence control after storage-tank charging, acting on four burner stages:
- Boilers connected in parallel
- 1 Storage tank sensor (ON Stage 1 lead boiler)
- 2 Storage tank sensor (ON Stage 2 lead boiler and sequence boiler)
- 3 Storage tank sensor (OFF)
- 4 Boiler sequence control unit with step controller, time elements etc.
- 5 Boiler flow temperature control (including maintenance of boiler inlet temperature)
- 6 Safety temperature limiter
The purpose of the storage tank is to maximize burner running times and so to minimize the number of burner switching operations. This is why it is designed as a stratification storage tank with a volume of at least 10 liters per kilowatt of thermal output. Depending on the size of the plant and the available space, two or more storage tanks may have to be planned for. Multiple storage tanks are connected in series.

**Principle of operation**

The following function description refers, by way of an example, to a system with a single storage tank and two boilers as illustrated in Fig. 2-11. If the temperature in the storage tank falls below the set point of the sensor located halfway up the tank (1), the primary boiler and associated burner will be switched on:

- If the output is greater than or equal to the current consumption, the burner will remain in continuous operation or the storage tank will gradually be recharged.
- When the storage temperature reaches the set point defined by the sensor at the bottom of the tank (3), the boiler will be switched off. This always applies to all boilers (or burner stages), whatever the load conditions, i.e. with a fully charged storage tank, the entire heat generating system is disabled.
- If the level of consumption exceeds the output of the lead boiler, the storage tank will be discharged further. As soon as the temperature falls below the set point at the sensor located near the top of the tank (2), the lag boiler will be brought online (possible after a delay).
- Once the second boiler is online, both boilers remain in operation – irrespective of load conditions – until the storage tank is fully loaded. Both boilers are then switched off via sensor (3). In this way, throughout the period for which the heating is in operation, long burner running times and hence, high levels of efficiency can be achieved.

**Hydraulic decoupling via storage tank**

The storage tank also provides perfect hydraulic decoupling between the boiler circuit and the consumer circuits. As far as boiler sequence control is concerned, the discharge time between two sensors provides a load-dependent, variable time integral, which has considerable advantages over a permanently set time-delay relay. The heat production system reacts more rapidly the greater the heat demand is. Oscillation does not occur, because storage capacity has to be charged or discharged between the switch-on and switch-off points. The plant is also easy to master in control terms, and only small temperature variations occur.

**In-built hydraulic decoupling**

The storage tank charging temperature is controlled via the boiler flow temperature set point using a mixing valve in the return. Assuming correct sizing, i.e. if the minimum boiler temperature set point is selected correctly, this control also ensures that the boiler return temperature remains high.

For the purpose of energy optimization, the storage tank temperature can be weather-compensated. The heat storage makes the system completely insensitive to sudden load changes (e.g. heatup processes). The clear advantages of this concept come at the price of increased investment costs and space requirements.
2.4.10 Sequence control based on burner load

This type of sequence control is suitable for boilers with modulating burners. The flow temperature of each boiler is controlled separately by a modulating controller to a fixed or weather-compensated set point. The lag boiler is brought online and taken offline based on burner load (air damper position and time delay). If the combustion efficiency is greater at partial loads than at full load, the lag boiler can be brought online earlier. When both boilers are in operation, they both operate in the modulating load range between approximately 30 % and 100 %. In order to achieve an even load distribution, the boilers must be hydraulically connected in parallel.

This is a simple control in principle. However, the two-position base load behavior of the burners often gives rise to control difficulties in the transition range.

2.4.11 Sequence control with the modulation factor of condensing boilers

So far, we have seen how the supply temperature to the consumers and the common boiler return temperature can be used to bring individual output stages online and offline (e.g. see 2.4.7). However, in the case of condensing boilers – especially when more than two boilers are interconnected – the approach already discussed is not sufficient to achieve a high degree of efficiency under all operating conditions, not only for individual boilers, but also for the configuration as a whole. The type of plant under consideration must satisfy the following criteria:

- Provide the desired supply temperature in accordance with demand, and without significant fluctuation or deviation
- Keep the return temperatures of condensing boilers as low as possible
- Optimize overall efficiency by use of various running-time strategies

A key point with the control methodology discussed here, is that it takes account not only of the supply and return temperature, but also of the modulation factor (i.e. the output currently being produced) of the boiler(s) in operation. For this purpose, the boiler sequence controller determines the boiler modulation factor (via a fan speed) and maintains an audit of the output from all the boilers.

Separate boiler temperature control

Each boiler has a separate boiler temperature control unit. The secondary boiler provides the individual boiler temperature control units with the required boiler temperature set point as necessary (nowadays normally via a bus system).

Various running-time strategies

In order to adapt the boiler sequence control as well as possible to the various operating conditions, different running-time strategies can be used:

- Enable as few boilers as possible
  - Minimizes the power required for the fan and the boiler pumps
- Extend burner running times, thereby minimizing the number of burner starts
  - Reduces emissions caused by burner starts
- Enable as many boilers as possible
  - Useful e.g. when the volume of water consumed is significantly higher than the amount generated
Fig. 2-12 Sequence control with modulation factor from two (or more) condensing boilers (in addition to flow and return temperature)

1 Boiler sequence controller
2 Flow temperature sensor
3 Return temperature sensor
4 Boiler temperature controller
5 Boiler temperature sensor
6 Safety temperature limiter
In an automatically controlled oil-or gas-fired boiler, the burner is basically the controlling element of a control loop, i.e. it operates according to the commands of a controller. That controller can be a simple thermostat in a small boiler, or it can be a precise, energy-optimizing electronic controller in a large boiler.

![Diagram of Automatic Burner as Controlling Element of Boiler Temperature Control Loop](image)

Fig. 3-1 Automatic burner as the controlling element of the boiler temperature control loop

In automatic boiler temperature control (Fig. 3-1), basically the same process occurs repeatedly: As soon as the temperature, i.e. the controlled variable ($x$), falls below the adjusted setpoint ($w$), the controller gives the so-called positioning command ($y$) to the heat producer; in this case, it is the starting command for the burner. The burner then produces heat, causing the temperature to rise, and, as soon as the setpoint is reached again, the controller shuts the burner down again. The temperature starts to fall again ... and so it goes on, back and forth between controlled startup and controlled shutdown. Depending on heat demand, the startup cycles are repeated in more or less rapid succession (= intermittent operation). In case of very great heat demand, the burner can often remain in operation for numerous hours without interruption (= continuous operation).

**Suitability of oil and gas**

Oil or gas is highly suitable for automatic heat generation, because:

- oil and gas can be delivered to the burner easily and cleanly through pipes
- oil and gas can be exactly metered or completely shut off using various types of valves
As little as 1cm³ of fuel oil will make 1m³ of water undrinkable, and the hazards, of gas (explosiveness, toxicity of some gas types) are common knowledge. In addition to these hazards there is also the environmental impact of the combustion gases of both fuels. Keeping both as small as possible is the target of all involved parties in oil and gas heat generation, i.e. the plant manufacturers and operators, safety authorities and legislators. Since about 1990, maximum limits for the pollutant content of flue gases from boiler plants have been specified in all countries where protection of the environment is an issue. Compliance with these limits is periodically monitored by specially trained inspectors. Setting aside the harmfulness of the combustion gases, the following conclusion can be drawn:

Fuel oil and gas are safe, provided the released amounts are properly burned.

“Properly” means that the combustion process must meet the following requirements:

• The burner must be supervised to ensure that it really ignites on startup and that the flame then continues to burn without interruption until controlled shutdown, i.e. that the entire quantity of fuel released is completely burned up

• If the burner fails to ignite or if the flame is lost during burner operation and in case of other plant faults that can cause an impermissible discharge of unburned fuel, the fuel valves must be closed immediately. Therefore, all fuel valves are designed to automatically close in the deenergized state

• In case of serious defects of the combustion system, whether in the burner, in the valves, in the flame supervision system, etc., the entire plant must be shut down. The device causing the lockout must remain interlocked in this fault position. The possibility must also be provided to signal the lockout via an optical or acoustic signal

The consequence of these requirements is as follows:

Every oil or gas burner must be equipped with a flame detector which, in combination with a flame supervision system, monitors the ignition and continued presence of the flame up until the controlled shutdown and which, in case of a fault, not only cuts off the fuel supply but also deactivates the burner and triggers an alarm via a control device.

However, burners do not simply consist of the fuel supply line, valves, ignition equipment and flame supervision system. They are (especially in the case of large burners) highly complex devices which must be started in a specific function sequence – the startup program. So, the burner control equipment must operate according to a schedule.

In practice, the program control and flame supervision systems are normally accommodated in a common housing. This combination of functions is referred to as follows:
3.2 Burner control  The burner control is used for **flame-dependent** startup and supervision of the burner. It is sometimes referred to colloquially as the flame safeguard. A schematic view of its operating principle would be something like the following:

![Schematic view of a burner control](image)

**Fig. 3-2** Operating principle of a burner control

1. Burner control
2. Programming mechanism
3. Flame supervision system
4. Flame detector
   a. Startup command from the plant’s temperature or pressure controller
   b. Startup and standby signals
   c. Flame signal
   d. Burner control signal

**Flame supervision** is performed using a **flame detector** on the input side of the flame supervision circuit and the **flame relay** on its output side. The flame relay’s contacts are linked to the program control in such a way that the burner can only be started up and operated if all conditions for correct flame supervision are met.

![Flame defectors](image)

**Fig. 3-3** Flame defectors for the supervision of blue-and yellow-burning oil and gas flames
3.3 Forced draft burners  

With this burner type, all necessary elements for automatic heat generation are installed in or on the fan housing, or they are in the immediate vicinity of the burner. The minimum equipment consists of the following:

![Diagram of forced draft oil burner](image)

Fig. 3-4 Basic design principle of a forced draft burner bottom with pilot burner

- M  Fan
- P  Oil pump, connected axially to the fan motor
- BV  Fuel valve(s)
- ZBV  Pilot burner valve
- Z  Ignition transformer
- LK  Combustion air damper, permanently set or controlled via motor
- SA  Air damper actuator for motor control
- Q...  Flame detector
- OH  Oil preheater, installed between nozzle and adjustable head in small light-oil burners, as a separate unit in large heavy-oil burners

There are, especially in the case of gas burners, additional components, such as air and gas pressure switches. The electrical signals from these devices are also mainly interrogated by the burner control, i.e. they are taken into account in program control.

In the case of high-capacity heavy-oil burners, the ignition spark is often not sufficiently powerful for direct electrical ignition of the burner. In this case, a low-capacity pilot burner (usually a gas or light-oil burner) is used to ignite the main flame. This type of ignition is referred to as gas-electrical ignition, and burners with this configuration are referred to as interrupted pilot burners. The pilot burner valve (ZBV) is used to release the required quantity of ignition fuel.

Forced draft oil/gas burners (dual-fuel burners) are built for medium to high-capacities and are primarily used in places where gas operation is more economical than oil operation during the gas low tariff period. The oil nozzle is normally located in the center of the so-called gas head. Since different combustion air quantities are required for the same burner output during oil and gas operation, the air damper control or air/fuel ratio control is more complex than in burners for a single fuel type. Additionally, the burner control must be equipped with different startup programs for oil and gas.
3.4 Burners without fans

Burners without fans are built both for oil and gas. The oil burners are the same as those used in oil stoves for individual room heating. They are not controlled and supervised by burner controls, so their functioning will not be described in detail here.

The true domain of burners without fans is gas-fired heat generation. Atmospheric gas burners are ignited by means of direct electrical ignition, a small pilot burner or, in the case of larger burners, using a so-called ignition burner bar. Ignition burner bars ignite the individual burner elements with a small pilot light directly assigned to each.

Gas-fired continuous-flow water heaters are usually ignited by a continuously burning pilot light. In systems of this kind, flame supervision mainly used to be achieved by electromechanical means using so-called safety pilots. Among other things, they ensured that gas was only released when the pilot light was burning. Nowadays, ignition and supervision is normally accomplished using purely electronic device combinations (ignition and supervision in the same housing).

Large atmospheric burners with infinitely variable burner output, on the other hand, are controlled and supervision by burner controls. The startup program differs only in a few minor details from that of a forced draft gas burner.

Alongside the advantages of atmospheric gas burners, which are:

- simple design,
- clean combustion,
- low-pollutant flue gases,
- easy control of burner output (only the gas quantity needs to be varied)

there are the disadvantages of this burner type:

- heat loss due to the continuous pilot light as well as the continuous draft due to the largely open burner system.
The flue gas outlet of the burner does not open directly into the stack but through a funnelled out gas duct, the so-called draft protector (Fig. 3-5 left). This ensures that strong gusts of wind in the stack do not blow out the main flame or pilot light thus endangering the safety of the plant. The continuous flow of air through the device due to the stack draft is also necessary in order to prevent any gas leaking from a gas valve from escaping into an enclosed space (kitchen, bathroom or cellar). In systems that are protected by burner controls, however, energy-saving combustion air and flue gas damper controls are used, where permitted.

3.5 Output control for burners with two or more stages

Output can be roughly matched to the current heat demand by adjusting the fuel and combustion air quantities in steps. When heat demand is low, the burner operates at the first output stage. As heat demand increases, the air quantity is generally increased first by an appropriate amount, then fuel valve 2 is opened (by means of an auxiliary switch on the air damper actuator). When the heat demand decreases again, fuel valve 2 is closed first, then the air quantity is also reduced to level 1. This means that combustion takes place with an excess of air in each transitional phase, which reduces combustion efficiency. Therefore, this type of output control is not energy optimized and is often replaced in practice by single-stage burners that are optimized for the boiler in combination with a storage tank, or by burners with two-stage shift control.

If instead of an on/off valve, an infinitely variable control valve (e.g. a gas butterfly valve) is used to enable the 2nd output stage, then not only the air quantity but also the fuel quantity is gradually (i.e. proportionally) adjusted as the nominal load position is approached. The term “two-stage shifting” is used to describe this kind of control. It prevents operation with excess air during the transition between output stages.

3.6 Modulating burner output control

For technical reasons, the output of such burners can only be varied continuously, i.e. modulated, above a certain fixed limit. For normal forced draft burners, this limit is at around 30...40 % of nominal load. Below this value, or partial load step, fuel/air mixing becomes so unfavorable that complete combustion is no longer guaranteed. So the burner is operated with on/off control in this partial load range.

In modulating operation, appropriate control devices must be used to adjust the fuel and air quantities simultaneously for output control. This is referred to as air/fuel ratio control.

Since, for technical reasons, the fuel and air have to be mixed in a non linear ratio, readjustment of the air/fuel ratio is necessary if the fuel type is changed.
When a burner is started up, physical and chemical processes take place which will only be described here from the point of view of control. We would like to concentrate particularly on the sequence of flame supervision here.

- The ignition of the burner is registered by the flame supervision system at the point in time that it receives the electrical “flame signal” from the flame detector. This must be, at the latest, on expiry of the so-called safety time (see 3.8.5), i.e. at the end of the time phase for burner ignition specified by the burner control. If the flame signal is not present at that time, the control unit immediately cuts off the fuel supply and then initiates lockout. This behavior is a uniform requirement of the competent safety authorities.

- If ignition occurs in good time, the flame monitoring system then checks whether the flame continues to burn correctly, i.e. it checks whether the flame signal remains constantly present without interruption until the burner is shut down.

- If the flame is lost during operation (or in case of failure of the flame signal during this phase), the burner control unit either initiates lockout or attempts a restart. However, restarts are only permitted for very low-capacity oil or gas burners.

It is, however, possible for the electrical flame signal to be triggered in other ways than by the flame, e.g. by other light sources, referred to as “extraneous light,” or by electrical faults simulating a flame signal.

If a simulated, i.e. erroneous, flame signal occurs, the safety of burner operation is no longer guaranteed. Therefore, every burner control tests its flame supervision system for correct functioning at every startup. This is done by checking that a flame signal is not present during off times and/or during the prepurging time. If a flame signal occurs during these times, the burner control immediately initiates lockout. Since most burners are started several times a day, and each time the burner control tests its flame supervision system as described, this test procedure provides a high level of safety.

Burners in continuous operation, where the self-test on burner startup does not occur one or more times a day, are a special case. If a flame-simulating defect occurs in continuous operation, a possible flame failure will not be detected, and the lockout will not occur, allowing unburned fuel to escape. Therefore, in large plants where continuous operation is possible, special self-checking flame supervision systems are implemented.

3.7 Flame supervision program
3.8 Basic structure of the start-up program

The basic structure of the startup program will be illustrated on the example of a large burner in order to be able to present as many of the influencing factors that must be considered during burner startup as possible. In the case of burner control units for small and very small burners, the number of such influencing factors is naturally far smaller, but the program structure remains the same. The program structure can be divided into the following phases:

3.8.1 Standby

In the interval between two burner runs, the burner control goes into the standby position. Its flame supervision circuit monitors the plant and itself for the occurrence of extraneous light and other erroneous flame signals. Lockout is executed if a fault occurs. The control output for the burner air damper delivers a close signal. The outputs for certain electrical check loops of the burner control are energized. (The function of such loops will be described in greater detail in the following section).

3.8.2 Controlled startup (start-up command)

Controlled startup is executed by the boiler thermostat when the boiler water temperature falls below the adjusted setpoint (including switching differential). Provided that all startup conditions are met, the control part of the burner control is energized. In order to check this, the signalling and checking contacts of all elements of the heat generation system are connected in so-called check loops of the burner control (normally multiple signalling contacts in series). All of these loops are linked (“intermeshed”) with the burner control in such a way that burner startup and operation is only possible if the loops are closed at the right time during given program phases. If that is not the case, the burner control initiates the necessary safety measures. There are the following types of check loops:

a) First startup check loop

Check for the correct initial position of various burner elements on startup. This loop is only “active” for a few moments, because the contacts it switches must be opened or switched over in the further course of the startup procedure. If it is not closed during the check time, the startup is not executed, but no lockout occurs.

Loops with this function normally include the following:

- End switch of the air damper actuator for the closed position
- End switches for indicating the closed position of the gas valve actuators
- Signalling contact for the “tight / leaky” signal of gas valve proving
- Auxiliary / check contacts of relays which indicate their correct initial position on startup
- Normally closed contact of the air pressure switch

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b) Second startup check loop  

Check of the operational prerequisites for burner startup and operation. This loop must be closed at the time of controlled startup and remain closed until controlled shutdown, otherwise the startup will be aborted or the burner will be shut down. However, a lockout will not be initiated, but a restart will be executed as soon as the startup conditions are met again.

Loops with this check function include the following:

- Boiler thermostat (in steam boilers the pressostat)
- Boiler temperature switch (whereas the safety limit thermostat is connected directly to the phase supply)
- Water level switch of steam boilers
- Gas pressure switch for minimum gas pressure (an attempted burner start with no gas pressure would be pointless)
- Oil temperature controller or thermostat in case of oil burners with oil preheaters, as well as other signal or check contacts with comparable functions

Loops that have special significance with regard to safety must remain closed without interruption from a defined moment until controlled shutdown. The following should be included in these loops, for example:

- ACTUAL contact of the air pressure switch. The lacking or failure of the air pressure signal would mean that prepurging, which is especially important for gas burners, would not take place. An undiscovered air pressure failure during burner operation could give rise to sooting of the boiler for numerous hours

- Signalling contact of the maximum gas pressure switch. If the maximum permissible gas pressure were to be exceeded, the amount of gas supplied to the burner would increase. Since the quantity of air would remain the same, however, the consequence would be insufficient air, giving rise to an increase in pollutant emissions and sooting

- Contacts of the flame relay of an external flame detector. Such flame detectors supervise the flame in the same way as the built-in flame supervision circuit of a burner control. Unlike one of these, however, the flame relay contacts of the flame detector are wired to externally accessible terminals so that the flame signal can be passed to the programming mechanism. These signal lines are also configured as check loops, and, in large-scale plants, two or more flame detectors are often connected in series in these loops in order to meet special safety requirements

A defect that causes a safety check circuit of this kind to open or close at the wrong time will trigger lockout in every case.
3.8.3 Prepurging

The purpose of prepurging is to clear the combustion chamber of any fuel/air mixture capable of deflagration or even explosion which may have formed during the previous standby period. The cause can be leaky gas valves or fuel oil that has dripped into the hot combustion chamber (and evaporated). At the same time, prepurging also sets in motion the air in the stack, which may have greatly cooled and become “heavy,” in order to lessen the effect of the pressure surge on ignition of the burner.

The prepurging duration is specified in standards. It can be between 10 and 150 seconds depending on burner type and application. In most cases, it is 30 or 60 seconds. During the prepurging time, the air in the combustion chamber and in the flueways up to the stack connection should be exchanged approximately 3 – 5 times. Also during the prepurging time, the flame supervision circuit continues the extraneous light check, in the case of some burner controls with increased intensity and additional self-tests.

When prepurging is completed, the combustion air quantity must be reduced, because the (small!) flame of the pilot burner must be ignited first. This would be difficult, if not impossible, with the full air flow. As soon as the air quantity has been reduced, the burner control activates the ignition transformer.

3.8.4 Preignition

This program phase is referred to as preignition or preignition time, because the ignition transformer is activated before fuel is released. The preignition time is necessary, because the high voltage across the ignition electrodes requires a certain time to first ionize the air molecules (like lightening in the atmosphere) before the actual ignition spark can arc across.
3.8.5 Safety time

The safety time begins at the moment of fuel release. It is the time for ignition of the burner, i.e. formation of the flame, that is predefined by the burner control. On expiry of this time period, the flame supervision system must register a flame signal, or the fuel valve will be immediately closed (in less than 1 second!), and lockout will be executed either immediately or with a slight delay, depending on the type of burner control.

The duration of the safety time is naturally also precisely specified by standards and safety regulations. It depends on the type and hazardousness of the fuel, so it is shorter for gas burners than for oil burners, which are more difficult to ignite and less hazardous. It also depends on burner output, i.e. on the quantity of fuel that is released during the safety time and may not be ignited. Therefore, forced draft gas burners have safety times between 2 and 5 seconds, atmospheric gas burners between 5 and 10 seconds, and oil burners, depending on output, also between 5 and 10 seconds.

Since the ignition of the flame causes a pressure surge, the burner control schedules a further interval for flame stabilization. Immediately afterwards, the burner control energizes the boiler temperature controller and passes the command to it for the further burner operation. This action concludes the burner startup by the burner control. However, it continues to supervise the flame and its check loops so that it can immediately close the fuel valves, shut down the burner and initiate lockout if dangerous operating conditions occur. The period of less than one second that is required to close the fuel valves in case of a fault is also referred to as the “safety time in operation.”

3.8.6 Controlled shutdown

Controlled shutdown is triggered by the boiler thermostat when the boiler temperature rises above the adjusted setpoint, and the thermostat opens its contact and with it the check loop. The opening of the loop is, at the same time, the signal for the control part of the burner control to return the burner and its programming mechanism to the initial position for the next startup. It does not recommence the extraneous light check until the flame is definitely lost. This is not the case until the remaining quantity of pressurized gas in the pipes downstream from the gas valves has been completely released and burned.

The main elements of this program sequence for startup of a large gas burner also apply to an oil burner of similar design. Therefore, dual-fuel burners, i.e. gas/oil burners, can be controlled by the same burner control and supervised using the same flame detector, provided the flame detector (e.g. ultraviolet detector) is able to detect blue-and yellow-burning flames equally well.

In the case of low-to medium-capacity burners, however, the startup program of an oil burner varies considerably in a few points from that of a gas burner.
In low-to medium-capacity oil burners, the fan rotor and oil pump are often powered by the same motor. This means that, if the motor fails, neither combustion air nor fuel will be delivered to the burner, so there is never a risk of boiler sooting due to lack of air. If the oil pump is powered by a separate motor, however, air pressure monitoring is always recommended, because, in case of a fan failure, fuel would still be delivered to the burner, although no air would be delivered.

In the case of heavy-oil burners, the fuel oil must be preheated so that it can be pumped more easily and atomized to a sufficient degree (at very low temperatures, heavy oil becomes so viscous that it takes on a gelatinous consistency!). In the case of light-oil burners with very low capacity, the already low-viscosity oil must also be preheated, because the low burner output (e.g. 1.5 kg oil throughput per hour) requires a very fine bore of the atomized nozzle and an appropriately matched viscosity of the oil.

With such burners, a check for sufficient oil preheating is usually only made on startup. After burner ignition, the radiant heat of the flame is in many cases enough to continuously preheat the small quantity of oil in the nozzle and nozzle pipe to a sufficient degree. On the other hand, in the case of heavy-oil burners, the oil temperature must be verified from startup to controlled shutdown. In order to check for oil preheating, the contact of the oil temperature switch is included in the appropriate start-up and operation check loops of the burner control.

In the case of oil burners whose oil pump is powered by the fan motor, the control program of the burner control is configured such that the ignition transformer is activated at the same time as the fan motor. This long preignition period provides indirect oil valve proving. If the oil valve is leaky, a smaller or larger quantity of fuel oil will already be atomized in the combustion chamber during prepurging.

Because the ignition spark is already present, it ignites the oil, and the prematurely formed flame is detected and interpreted by the flame supervision circuit as extraneous light, which triggers lockout.

Long preignition is not permitted with gas burners, because leaks in the gas supply can mean that an explosive gas/air mixture is present in the combustion chamber which must be expelled during the prepurging phase. Therefore, preignition commences just before the beginning of the safety time, i.e. just before the fuel valve is opened.
3.9.4 Postignition/reignition

Since, in the case of oil burners, the fuel oil must first boil and evaporate before it can react with the atmospheric oxygen, it is more difficult to ignite an oil burner than a gas burner. For this reason, the ignition of the oil burner is often assisted by postignition, i.e. the ignition transformer is not immediately deactivated on appearance of the flame or expiry of the safety time, but it remains in operation for a defined period of time. Other burner controls support flame stabilization via so-called re-ignition. Such burner controls deactivate the ignition transformer when the flame signal is received, even if the safety time has not yet elapsed, but they reactivate it immediately if the flame appears to be lost or even fails completely for a brief period. However, such reignition attempts must never continue for a total duration longer than the permissible safety time for the respective burner.

Fig. 3-7 Startup programs for oil and gas burners
Left: Oil burner control program
Right: Gas burner control program
with short preignition t3; with long preignition t3"
and postignition t3n no postignition

3.9.5 Start repetition or lockout in the event of loss of flame

In the case of forced draft gas burners, the burner control always initiates lockout if the flame is lost during operation in order to prevent all possible hazards. The only exceptions are burners with very small capacity.

In the case of the less hazardous oil burners, in which even a relatively small air bubble in the oil supply pipe can cause loss of flame, so-called start repetition is permissible. If the flame relay drops out during operation, the burner control immediately closes the oil valves, resets its programming mechanism to the initial position (in the case of smaller burners, it already is in this position) and immediately tries to start the burner up again according to the startup program. If a perfect flame is not achieved within the safety time, there is obviously a major defect, so the burner control finally initiates lockout.
The control and supervision of atmospheric gas burners does not include prepurging and all of the checks involved with it. However, burner startup does not begin with the brief preignition but with a delay. This delay is insignificant for burner startup but important in case of a fault. If, for example, the burner fails to ignite and repeated shutdowns occur in the course of troubleshooting, the burner control has to be reset each time before it can execute the next startup attempt. If start-up were to begin with preignition and gas release instead of a delay, dangerous concentrations of gas could soon gather, because more gas may be released than the natural draft is able to extract from the combustion chamber. The delay prevents too rapid a sequence of unsuccessful startup attempts, thus preventing hazardous gas concentrations from gathering in the vicinity of the burner.

Special variants are necessary for large atmospheric gas burners, because certain program modifications are needed in this case. These are, for example, longer safety times or special air damper control programs which limit the quantity of secondary air drawn in by the burner to the necessary minimum (prevention of heat losses).

Fig. 3-8  Control program for an atmospheric gas burner with delay “tw”
During combustion processes (oxidation), fuel molecules bond with oxygen. So-called stoichiometric combustion is achieved if each fuel molecule bonds with an oxygen molecule $O_2$ so that the flue gas contains no more oxygen molecules. The characteristic for the fuel/air ratio is expressed using the lower-case Greek character $\lambda$. Ideal combustion with an exactly stoichiometric fuel/air ratio $\lambda = 1$, i.e. a residual oxygen content of 0 in the flue gas, cannot be achieved with natural fuels such as oil and gas because of premature carbon monoxide (CO) and soot formation (Fig. 3-9).

If there is too little air ($\lambda < 1$), the fuel is not completely burned up. This has a negative effect on efficiency, and the quantities of soot and pollutants, especially highly toxic carbon monoxide (CO) and unburned hydrocarbons (CH), in the flue gas rise very sharply with a decreasing $\lambda$ number.

The greatest efficiencies and, at the same time, the lowest pollutant concentrations are achieved with a slight excess of air, i.e. $\lambda_{\text{opt.}} = 1.03...1.3$ (see diagram).

Since the oxygen content of the combustion air depends on the air’s density, which in turn depends on the air temperature, it changes continually in the course of a heating period. In order to avoid operation with too little air even in unfavorable weather conditions, uncontrolled boilers are normally operated with a considerably greater excess of air than $\lambda_{\text{opt.}}$. This gives rise to a correspondingly great reduction in combustion efficiency. The greater the burner output, therefore, the more the investment in a $\lambda$ control system pays off.
Whereas flue gas analysis used to take about 1 hour in the laboratory, the zirconium dioxide (ZrO₂) probe, in combination with electronic signal amplification, now makes it possible to continually measure the residual oxygen content of flue gases and to compare this measured value at the input of a modulating controller with the selected λ_{opt} setpoint. If a deviation is detected, the controller corrects it by adjusting the fuel/air ratio.

If the plant is equipped with multistage or modulating burners, compensation of the setpoint for the residual oxygen content of the flue gas is required. The reason for this being that a burner requires a greater excess of air at low loads than at full load so that a greater amount of kinetic energy is available for the fuel/air mixing process. Therefore, the residual oxygen content setpoint must be compensated as a function of burner output. Since different quantities of combustion air are required for gas than for oil, plants with dual-fuel (gas/oil) burners must be equipped with two shift controllers.

Digital technology also provides for “smart” solutions in this field.

Fig. 3-10 Oxygen sensor for measuring the residual oxygen content of flue gases from natural gas and light oil boilers (in conjunction with the appropriate controller)
In room temperature control, a distinction is made between individual room control in large buildings (office buildings, hotels, etc.), which is described in section 4.3, and room temperature control in smaller buildings, where the output of heat to the entire building is controlled on the basis of the temperature in a representative living room, the so-called reference room. This type of control is acceptable as long as the temperature behavior of the reference room at least roughly corresponds to that of the remaining rooms. It is also suitable whenever the temperature of a relatively large main room has to be controlled alongside that of subordinate adjoining rooms. This room temperature control principle is applied, for example, in single-family homes, shops, restaurants, sports halls, cinemas etc.

Temperature disturbances occurring in the room are detected directly by the room sensor and corrected.

Since a drop in the heating circuit temperature (e.g. during domestic hot water charging) does not affect the room temperature until after a long delay, the disturbance is not corrected until then. Therefore, by the time the sensor can detect the room temperature change, its cause is usually no longer present. The control valve is then incorrectly opened much too far, causing a room temperature overshoot.

Room temperature controllers are available either with selectable operating modes or in one of the two variants as two-position controllers for direct burner control or as three-position controllers for quasi-continuous control of a mixing valve. The common control concepts are as follows:

- Two-position room temperature control acting on the burner
- Modulating room temperature control acting on a mixing valve
- Room/flow temperature cascade control
With this type of control (Fig. 4-1), the room thermostat (two-position controller) directly activates the boiler’s burner when the room temperature falls below its setpoint. The boiler water temperature and with it the flow temperature rises, the radiators get warmer, and, finally, the room temperature rises above the setpoint, so that the controller de-activates the burner again. Due to the inertia of the room temperature controlled system, this type of control gives rise to continual fluctuation of the controlled variable, whose range of variation increases the larger the room is.

An improvement in control terms can be achieved with so-called “thermal feedback,” illustrated in this example of a two-position controller with a bimetal sensor.

**Thermal feedback**

Fig. 4-1 Room temperature control acting directly on the burner
1 Room thermostat
2 Gas valve
3 Boiler with atmospheric gas burner
4 Safety limit temperature limiter

Fig. 4-2 Two-position controller with feedback
1 Bimetal sensor
2 Setpoint adjustment facility
3 Thermal resistor
4 Snap-action magnet (for precision switching)
At the same time as the bimetal sensor (1) switches the burner on, a small heating coil (3) in the room thermostat housing is activated. This heats up and simulates a rapid rise in the room temperature, causing the burner to switch off again early. However, if the room temperature is still below the setpoint, the heating coil cools off quickly, and the burner is switched on again.

This thermal feedback greatly increases the switching frequency. The fluctuations of the flow temperature caused by the on/off switching of the burner are minimized and then smoothed by the room's inertia. This makes room temperature control within a tolerance band of approximately 1 K possible. However, the high switching frequency is primarily suitable for atmospheric gas burners and not for forced draft burners, in which every burner startup involves losses. Additionally, the boiler can cool to below the flue gas dew-point in the deactivated state. Therefore, this control type is only suitable for boilers whose design and materials are suitable for low-temperature operation.

The graph below shows the control action of a two-position controller with feedback, using the example of a controlled system with multiple storage tanks.

Fig. 4-3  Two-position controller with feedback in multiple storage tank controlled system

- a) Response of the controlled variable x (room temperature) in the heat-up process
- b) Response of the feedback variable x_r (additional temperature through feedback resistor)
- c) Output positioning pulse (y)
Installing thermal feedback in a temperature control system – which acts on the zone pumps – produces, for example, in the following changes:

<table>
<thead>
<tr>
<th></th>
<th>Without thermal feedback</th>
<th>With thermal feedback</th>
</tr>
</thead>
<tbody>
<tr>
<td>Band in which controlled variable fluctuates</td>
<td>3.6 K</td>
<td>0.8 K</td>
</tr>
<tr>
<td>Duration of a switching cycle</td>
<td>34 min</td>
<td>8 min</td>
</tr>
</tbody>
</table>

Room thermostats with a gas diaphragm

These room thermostats do not require thermal feedback because they have a small switching differential (< 1 K). However, this small switching differential does cause – as is the case with room thermostats with thermal feedback – a reduction in the length of the switching cycle, and therefore results in frequent switching operations.

Fig. 4-4 Room thermostat with setpoint selector dial and on/off switch (with a gas membrane as the sensor element)
4.1.3 Modulating room temperature control acting on a mixing valve

This type of control (Fig. 4-5) allows the boiler to be operated at a constant temperature which can be kept above the flue gas dew-point. The consumer flow temperature can then by reduced to the required value by mixing with cold return water. In terms of faulty control behavior, this type of room temperature control is hardly better than the direct burner control described under 4.1.1.

![Diagram of room temperature control acting directly on a mixing valve]

Fig. 4-5  Room temperature control acting directly on a mixing valve

1  Room temperature controller (e.g. CHRONOGYR REV…)
2  Motorized mixing valve
3  Boiler with forced draft burner
4  Safety limit temperature limiter
5  Control thermostat

Room temperature control involves a very slow controlled system. If the controller detects a deviation from the selected setpoint, it delivers a corresponding positioning command to the mixing valve. Accordingly, the actuator fully opens or closes the mixing valve before the room temperature sensor can detect the effect of this intervention. This results in a continuous overshoot and undershoot of the room temperature.
With this type of control (Fig. 4-6), the room temperature control is divided into two control loops, i.e. a slow room temperature control loop and a fast flow temperature control loop. The primary controller (P-action) is adapted for the room temperature control system, and the auxiliary controller (PI-action) for the flow temperature control system. The primary controller detects the room temperature control deviation, from which it determines the reference variable for the flow temperature control. The auxiliary controller controls the flow temperature to the value specified by the room temperature controller. In practice, however, the two controllers are usually combined in a single device.

In the case of cascade control, the flow temperature thus becomes the manipulated variable for control of the room temperature. The room sensor measures the temperature in the room and compares it with the room temperature setpoint. If there is no deviation between the actual and desired value, then the flow temperature is controlled to a predefined setpoint of e.g. 40 °C. However, if there is a deviation, the flow temperature setpoint is adjusted, i.e. a room temperature which is too low by 1 K will result in a flow temperature setpoint which is too high by 20 K, for example. This transmission factor (slope S) is adjustable. It is important to select as high a value as possible, as otherwise there will be a correspondingly large proportional offset (or “error variable”). The auxiliary controller keeps adjusting the control device until the temperature measured by the flow sensor is equal to the new setpoint. In the case of cascade control, disturbance variables in the flow and room temperature controlled systems are detected and controlled.
The room temperature setpoint $w_1$ is set to 20 °C and the associated supply temperature setpoint $w_2$ to 40 °C (= offset for room temperature control). The steady-state characteristic of the master room temperature controller (Fig. 4-7) shows the corresponding supply temperature setpoints ($w_2$) for a cascade slope $S$ of 20. The cascade slope $S$ corresponds to the transfer coefficient $K_P$ of the controller (often referred to as the controller gain). For a room temperature differential of 1 K, this produces a supply temperature setpoint change $\Delta w_2$ of 20 K.

![Steady-state characteristic of a room temperature cascade master controller](image)

Transfer coefficient of controller, $K_P = \frac{\Delta w_2}{\Delta \theta_1}$

In the example shown, a flow temperature correcting span $\Delta w_2$ of 40 K (20...60 °C) for the room temperature controller gives rise to a proportional band $X_p$ of 2 K and a maximum residual error variable $e$ (w-x) of $-1$ K or $+1$ K.
Outside temperature compensated supply temperature control (Fig. 4-8) involves closed-loop supply temperature control (fast controlled system) and open-loop room temperature control. It requires an outside temperature sensor (8) and a supply temperature sensor (7). The relationship between the outside temperature and the room temperature is represented by the heating curve (6) which provides the setpoint for the supply temperature controller (1). The lower the outside temperature the higher the supply temperature must be in order to ensure that the desired room temperature is achieved. Which supply temperature is necessary at which outside temperature is influenced by the type of heat transfer (radiators, floor heating), the thermal insulation of the building envelope and the building’s location (solar radiation and wind factors) and defined by the starting point and slope of the heating curve (see also Fig. 4-9 and Fig. 4-10).

In simple controllers, only the slope of the heating curve can be adjusted.

Fig. 4-8  Outside temperature compensated supply temperature control

1 Supply temperature controller
2 Mixing control valve
3 Boiler with fan burner
4 Safety temperature limiter
5 Boiler temperature control thermostat
6 Supply temperature setpoint adjuster (heating curve)
7 Supply temperature sensor
8 Outside temperature sensor
9 Thermostatic radiator valve
10 Heating-group circulating pump
In buildings with good thermal insulation, the influence of outside temperature variations on the room temperature decreases, whereas the influence of the interference variables such as solar radiation, wind and external heat sources increases. Purely outside temperature compensated supply temperature control can be improved by using additional outdoor sensors to detect wind and/or solar radiation influences and to correct the basic heating curve setting accordingly. In this case, the term “weather-compensated” supply temperature control is used.

**Advantage**
Changes in the boiler water temperature are quickly detected by the supply temperature sensor and corrected.

**Disadvantage**
Disturbances occurring in the room (internal heat gain) cannot be detected and corrected.

Fig. 4-9 shows a graphical presentation of mathematically calculated heating curves in the usual slope range between 0.25 and 4.0. Because the heating curves are not linear, they have a different slope at every point. In order to be able to assign a defined slope, therefore, a linear grid has been defined for the outdoor temperature $\Delta \vartheta_A$ between $+20 \, ^\circ\text{C}$ and $0 \, ^\circ\text{C}$. This indicates the change in the supply temperature setpoint in K for an outside temperature change ($\Delta \vartheta_A$) of 20 K.

In older digital heating controllers, the setting values are often raised by a factor of 10, because older displays only provided for one decimal place. This enabled slopes of 0.25 to 4.0 to be programmed with setting values between 2.5 and 40.

**Non-linear heating curve**
Since the transmission losses of buildings increase in proportion to the difference between the room temperature and outside temperature, but the heat output of the radiators increases more greatly with an increasing difference between the mean radiator and room temperatures, the heating curve levels out with falling outside temperature.
However, the heating curve can still be displayed as a straight line if a logarithmic scale is selected for the outside temperature (Fig. 4-10) so that the curvature is optically linearized.

![Optically linearized heating curve](image1)

**Optically linearized heating curve**

4.2.1 Room temperature setpoint correction and night setback

The selected heating curve can be adjusted in parallel to correct the daytime temperature and to set the night setback. The effect is independent of the slope, so the magnitude of the parallel shift acts directly as a room temperature change. The starting point for the night setback is the selected daytime room temperature level. Fig. 4-11 shows this correction/setback possibility on a manually adjustable hardware controller.

![Manual parallel shift of the heating curve](image2)

In the case of room temperature corrections during the day and especially in the case of night setback, it should be noted that the selected temperature change is not achieved for some time. This duration depends directly on the design (time constant) of the building (heat storage capacity, thermal insulation, window area proportion etc.).
4.2.2 Room temperature compensation authority

The room temperature can also be used as an influencing variable for the heating curve. The room temperature of a reference room is measured for this purpose. If a disturbance occurs in that room (e.g. solar radiation, change in the number of occupants), a parallel shift of the heating curve is executed according to the selected authority of the room temperature deviation. In order for the compensation to have an effect, the authority must be set relatively high. However, since interference variables like solar radiation and internal heat sources do not have the same effect in all rooms, the room temperature compensation authority must be carefully considered. If the heat gain is to be compensated individually, the use of thermostatic radiator valves in all rooms or individual room control (see 4.3) is recommended.
4.3 Individual room temperature control

As discussed previously, there are various influences which result in a change in room temperature. As these interfere with the room temperature control loop, these are also referred to as “disturbance” effects. The main ones are:

- Heat emitted by people
- Heat dissipated by equipment, machinery, lighting etc.
- External influences such as solar radiation, wind etc.

Different types of individual room control are used to counteract the disturbance caused by these influences.

Fig. 4-12 Overview of various individual temperature control systems with central, or outdoor-temperature or room-temperature compensated control of the primary heating circuit

1 Central heating circuit controller
2 Thermostatic radiator valve head (fitted onto radiator valve)
3 Thermostatic radiator valve head with remote sensor
4 Radiator valve fitted with electronic valve head
5 Radiator valve with actuator and time-programmed room temperature controller
6 Individual room controller with room unit (as part of a building automation and control system) acting on several radiator valves with actuators
7 Room temperature sensor in reference room (for room influence)
8 Outdoor temperature sensor (and possibly other sensors for solar and wind influence)
**4.3.1 Thermostatic radiator valve heads**

Weather-compensated supply temperature control in combination with thermostatic radiator valve heads – fitted to the individual radiator valves – can already be considered a simple individual room control system. Moreover, these thermostatic heads make it possible to set a lower room temperature (e.g. in bedrooms) than the temperature defined by the heating curve.

Thermostatic radiator valve heads are P-controllers with a relatively large residual error variable. However, if the primary control is based on weather-compensated control of the supply temperature, the thermostatic heads are required only for fine control of the room conditions. As a result, the residual error variable is only noticeable in the presence of disturbance variables.

![Fig. 4-13 Thermostatic radiator valve heads (on the right with remote sensor)](image)

Depending on installation conditions, thermostatic heads with remote sensors may be used.

![Fig. 4-14 Installing a thermostatic radiator valve head, and installation conditions where a remote sensor is called for](image)
**Principle of operation**

The fluid-filled sensor (1) responds to deviations from the predefined room temperature setpoint. As the room temperature rises, the fluid inside the metal capsule expands, exerting pressure on the bellows. This pressure is transferred to the stem (3), which modulates the valve closed, thereby reducing the heat output from the radiator.

After the valve is closed, an overtravel mechanism (2) with a spring compensates for any pressure caused by further expansion of the bellows after the valve is closed, so that this pressure is not transferred to the valve stem.

When the room temperature falls, the bellows expands again, so opening the valve. The radiator heat output increases again.

The result is stepless operation of the radiator valve, with fine control of the flow of the heating medium to the radiator. In this way, a constant room temperature can be achieved in individual rooms in accordance with the desired setpoint.

The setpoint is adjusted by rotating the setting knob. This causes the thermostatic head (4) to be screwed further in or out of the base unit (5), so changing the basic position of the valve stem.

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Fig. 4-15 Thermostatic radiator valve head (cross-section)

1. Fluid-filled sensor element
2. Overtravel mechanism (compensates for expansion when the valve is closed)
3. Plastic stem for operation of the valve (isolates valve from sensor element)
4. Setting knob
5. Base unit
6. Screwed fitting
**Avoid underestimating hydraulic problems**

When using thermostatic radiator valve heads, it is of primary importance not to underestimate hydraulic problems. The only way to keep these more or less under control is through careful sizing and perfect hydraulic balancing. Even then, however, the mutual influence of individual heating circuits under different operating conditions cannot be eliminated entirely.

**Radiator valves with integrated differential pressure control (MCV MiniCombiValves)**

Radiator valves with integrated differential pressure control (also referred to as MiniCombi valves, MCV) ensure a defined heat output whatever the operating conditions. They may be fitted with thermostatic radiator valve heads or electric actuators.

![Diagram](image)

Fig. 4-16 Room temperature, system pressure and volumetric flow rate curves with normal thermostatic radiator valves (left) and MCV differential-pressure-controlled radiator valves (right)

1 Control valve  
2 Differential pressure controller

The main functions are as follows:

- Simultaneous operation as a control valve to regulate the volumetric flow rate, and as a pressure controller for automatic balancing
- Compensation of differential pressure variations, with complete hydraulic decoupling of the consumers
Radiator valves with (electric or thermostatic) actuators that are controlled by a digital room temperature controller provide for individual room or zone temperature control with scheduled switchover from normal to reduced temperature. Alongside an individual weekly heating schedule and selectable temperature setpoints for normal and reduced operation, some additional functions are integrated, such as:

- PI(D)-control without residual deviation
- Display with numerical values for the current temperature setpoint, bar chart for the current 24 hour heating schedule and symbols for normal and reduced operation
- Manual adjustment of the current temperature setpoint
- Manual switchover between normal and reduced operation
- Permanent normal or reduced operation
- Frost protection
- Temporary valve closing in case of a sudden temperature drop due to opening of a window (window function)
- Pump anti-seizing program for long standstill times
- Manual valve actuation, e.g. by service staff
In buildings with a large number of rooms, such as hotels or office buildings, digital, communicating individual room controllers are installed as part of a centrally controlled and monitored room automation system. This makes time-consuming programming and checking activities in each individual room unnecessary, because this work can be carried out from a central automation level via a building bus (e.g. LON, EIB etc.).

The room temperature control in the individual rooms is based on demand, i.e. the system only controls a room at the comfort temperature in response to a specific demand (e.g. manual operation of the occupancy button, detection with an occupancy sensor or a setpoint command from the automation level).
In addition to heating and cooling functions, today’s systems also include integrated functions for room control of lighting and window blinds.

**4.3.4 Individual room control system combined with energy consumption measurement per occupied unit**

Special digital systems Fig. 4-21 provide for individual temperature control of each room in residential buildings and non-air-conditioned office buildings and, at the same time, record the consumption-related heating costs per occupied unit (e.g. of each apartment). The prerequisite for this is horizontal distribution of the heating circuits with a separate supply and return for each apartment or occupied unit. A heat meter or special control valve with integrated flow and temperature difference measurement is installed in the supply and return of each apartment or occupied unit. The amount of heat energy used is calculated from these measured values and transmitted to the central unit via a data bus. The consumption data is saved to an electronic data storage medium in the central unit and serves as input for consumption-related heating cost accounting.

**Advantages**

Advantages of individual room temperature control with energy consumption measurement:

- The programmable room device allows the room temperature for normal and reduced operation in each occupied unit to be exactly matched to individual needs in accordance with the current heating schedule.
- Manual mode switching using the economy button
- Display of important information for each occupied unit, e.g. setpoints and actual values, room temperature, heating schedule, current heating water demand, meter reading, heat consumption, billing date meter reading, errors and faults.
- The room temperature controllers allow an individual temperature setpoint to be selected and maintained for each room.
Fig. 4-21 Individual room control system combined with measurement of the energy consumption per occupied unit (e.g. apartment)

1 Central energy consumption data unit (central unit)
2 Central heating circuit controller
3 Heat metering and control valve
4 Programmable digital room device with temperature sensor
5 Analog room device with temperature sensor
6 Room temperature controller
7 Radiator valve actuator
8 Cold water meter
9 Hot water meter
10 Meter pulse adapter
11 Gas meter
With regard to the central unit, there are, among others, the following advantages:

- Central acquisition and storage of the data for each occupied unit, e.g. actual values, setpoints, operating states, manual interventions, heating energy consumption, billing date meter readings, faults.
- Central control and monitoring of the entire system
- Central management of the room temperature controls in the occupied units, e.g. via high and low-limit control of room temperature setpoints or of comfort periods.
- Compensation of the supply temperature setpoint per group controller based on the actual heat demand of the assigned apartments (load influence)
- Individual heating cost accounting per occupied unit based on actual heating energy consumption. This creates a major incentive to actually utilise the individual programming capability.

4.4 Automatic heating limit switching

Automatic heating limit switching activates and deactivates the heating system throughout the year according to demand, i.e. as little heating as possible but as much as necessary. It provides for a reduction of energy consumption without affecting comfort, especially in the transitional phases between the cold and warm seasons.

Simple controllers take into account only the “current” outside temperature for automatic heating limit switching; more complex ones use a so-called “damped” outside temperature and also a “mixed” outside temperature for certain functions.

“Damped” outside temperature

The “damped” outside temperature accounts for the fact that, when the heating is not on, the indoor temperature of a building follows the progression of the outside temperature in a damped, delayed manner. This theoretical room temperature progression is calculated by the control system on the basis of the building’s time constant, i.e. its heat storage capacity (light, medium or heavy construction).

“Mixed” outside temperature

The “mixed” outside temperature results from adding the adjustable “damped” outside temperature portion, which depends on the building’s time constant, to the supplementary current outside temperature portion.
4.4.1 Yearly automatic heating limit switching (summer/winter changeover)

The changeover from heating system OFF to heating operation and vice versa is no longer performed manually by the user. Instead, it is switched on and off according to demand, taking into account the heat storage capacity of the building, i.e., based on the “damped” outside temperature. The yearly heating limit is adjustable (e.g., 15 °C). Because so-called yearly automatic heating limit switching takes into account the “damped” outside temperature, the heating system will not be switched on for the sake of one or two cooler days. Therefore, yearly automatic heating limit switching prevents the heating system from being switched on and off too frequently during the transitional period.

4.4.2 Daily automatic heating limit switching

In more complex control systems, the criterion for “heating ON” and “heating OFF” is provided by daily automatic heating limit switching which takes into account the “current” and “mixed” outside temperature. The switching limits for the activation and deactivation of the heating system can be set individually for daytime and night-time operation (day and night limit setpoint).

The heating system is switch on when the “current” and “mixed” outside temperatures fall below the selected limit setpoint for the day or night. However, if one of the two temperatures rises above the selected limit setpoint for the day or night, the heating system is switched off (pump off, control valve closed). Therefore, not only the “current” outside temperature but also the storage capacity of the building is taken into consideration for the activation and deactivation of the heating system.

Direct adjustment for the building’s storage capacity is done by selecting the building time constant, e.g., light construction 18 hours, or medium construction 36 hours (see “damped” outside temperature under 4.4), and indirect adjustment is done by selecting the limit setpoint for the daytime and night-time (greater difference between day and night heating limit with heavier construction, smaller difference with lighter construction).

![Diagram of daily automatic heating limit switching](image)

Fig. 4-23 Daily automatic heating limit switching

- $\theta_{AA}$: Current outside temperature
- $\theta_{AM}$: Mixed outside temperature
4.5 Start and stop time optimization

This optimization feature shifts the switch-on and switch-off times for a given occupancy period for the purpose of energy saving without affecting comfort.

![Diagram of start and stop time optimization](image)

Based on the defined heating schedule (occupancy times), the outside temperature, the room temperature and the building’s storage capacity, the controller automatically calculates the ideal times for the beginning of the heat-up and reduced phases. This means that the system is always switched such that, at the desired time (scheduler setting), the effective room air temperature setpoint is close to being achieved or that, on switchover to reduced operation, it is maintained within a defined band (e.g. 1 K in Fig. 4-24).

Therefore, this optimization ensures that:
- switchover to reduced operation takes place as early as possible
- the reduction to the reduced temperature setpoint takes place without any supply of energy (fast reduction)
- re-heating is kept as short as possible (boost heating)
- the desired normal room temperature is achieved neither too early nor too late.
In systems with room temperature sensors, the heating remains off after switchover from the daytime to the night-time setpoint until the night-time setpoint is reached. Not until then is the temperature controlled to the night-time setpoint.

In systems without room temperature sensors, the controller calculates a duration for which the heating system remains off based on the building's properties.

![Fig. 4-25 Fast reduction and boost heating](image)

After switchover from the night-time setpoint to the daytime setpoint, the supply temperature is raised above the setpoint as defined by the heating curve, in order to heat up the rooms quickly. In modern controllers, boost heating is a self-adjusting process, i.e. its effect depends on the duration and magnitude of the previous setback and can be adapted to the building situation (with or without room temperature sensors). In other words, the process is similar to that described in 4.5, except that the desired setpoint is achieved at the start of the occupancy period.

**4.7 Adaptive (self-learning) heating curve**

The heating curve is automatically adapted to the building’s properties via continual checking of the room air temperature, outside air temperature and supply temperature. Small corrections are made to the heating curve by daily calculation of the mean detected setpoint deviations until it is adapted over the entire outside temperature range. Falsification of the adjustment due to interference variables (open windows, external heat sources etc.) is prevented by special correction limits.
4.8 Pump control and interlock

The controller controls the circulation pump according to demand, i.e. it is only on if heating is required or if frost protection has been triggered.

4.8.1 Differential pressure-dependent speed control

To prevent the pumps in a plant from pumping too much water, pumps with electronic speed control are often used. They match the pumped volume of water to the load conditions of the plant, by controlling the differential pressure. There are two types of differential pressure control for this purpose:

- Control at constant differential pressure
- Control at variable differential pressure

The variants below apply to variable-volume hydraulic circuits, such as those in a community heating network with individual substations (see 4.12.1, the community heat supplier) or in systems with thermostatic radiator valve heads (cf. 4.3). The type of differential pressure control to select depends on how the various consumers in a system respond to low load conditions.

In systems where the consumers respond differently when operating with a low load, speed control at constant differential pressure should be selected. This prevents a deficient supply from any individual consumers operating on low load.

In systems with consumers with the same response when operating on low load, the pump speed can be controlled at a variable differential pressure. This produces greater energy savings than control at constant differential pressure.

The following aspects should be considered:

Pumps for which the speed is not modified have a downward sloping characteristic (1 in 4-26); in other words, the delivery head (Δp) falls as the pumped volume (V) increases. If valves (e.g. thermostatic radiator valve heads) now intervene to alter the plant characteristic curve (e.g. from 3 to 4), then the operating point of the pump shifts accordingly from A to B. In this process, the power consumption is reduced along the power consumption curve 5 from A to B.

The ideal approach, however, would be to take advantage of the laws of proportionality as applied to heating systems, and to shift the operating point along the plant characteristic curve to C. This would also reduce the power consumption even further, i.e. from A on curve 5, to C on curve 6. In addition, this would also reduce the pressure differential arising across the valves.
Fig. 4.26 Change in operating point and power consumption caused by additional resistance and adjustment of pump speed respectively

1 Pump characteristic (for a given speed n)
2 Pump characteristic at reduced speed (response with low load)
3 Plant characteristic at design conditions
4 Plant characteristic modified by additional resistance
5 Power consumption characteristic at speed at design conditions (1)
6 Power consumption characteristic at reduced speed (2)
A Operating point and power consumption of pump at design conditions (speed 1)
B Operating point and power consumption of pump caused by additional resistance (e.g. valves closing) without a change in pump speed (1)
C Operating point and power consumption of pump at reduced pump speed (2)
Control at variable differential pressure follows these basic considerations. The last consumer in the network has a differential pressure at design conditions which must be maintained. This is also referred to as the differential pressure at the “bad” point in the system. The differential pressure is now adjusted by the speed-control function along the straight line (control curve 3 in 4-27) between the differential pressure at the “bad point” and the maximum required differential pressure (A).

This variable speed control function is often applied between the 100 % and 50% of the maximum delivery head $H_{m}$, as these are the values which have proven themselves in practice, and because no significant effort is required on the part of the heating engineer.

In the case of our example, this gives operating point D and the associated power consumption $P_{D}$.

However, the disadvantage of this method is that with a low load, consumers in the first part of the plant may not be supplied with the necessary differential pressure, especially if these consumers respond differently to a low load.
Fig. 4-27 Operating points and potential savings for speed-controlled pumps
1 Pump characteristic (at a given speed n)
2 Plant characteristic at design conditions
3 Control characteristic with control at variable differential pressure
4 Control characteristic with control at constant differential pressure
A Operating point and power consumption $P_A$ of pump at design conditions (speed 1)
B Operating point and power consumption $P_B$ of pump caused by additional resistance (e.g., valves closing) without a change in speed (1)
C Operating point and power consumption $P_C$ of pump with max. reduction in speed
D Operating point and power consumption $P_D$ with variable differential pressure (control characteristic 3)
E Operating point and power consumption $P_E$ at constant differential pressure (control characteristic 4)

Control at constant differential pressure
To ensure that the differential pressure is always sufficiently high, even at the first consumer, the system is controlled at constant differential pressure (control characteristic 4 in Fig. 4-27). This does limit the speed range and the potential energy savings to some extent. In our example, the result is operating point E and the associated power consumption $P_E$. 
It can be seen from Fig. 4-27 that, depending on the type of speed control used, power consumption can be reduced substantially (e.g. to 50% of $P_A$ in the case of $P_E$ or to 35% of $P_A$ in the case of $P_D$). However, with the control characteristics used here, the power consumption cannot be reduced to the theoretical maximum of approximately 20% at $P_C$.

**4.8.2 Speed control based on valve position of the consumers**

Another option for adapting the pump speed to specific system conditions in a given system, is speed control based on the valve position of the consumers.

This type of speed control is useful when the current positions of all the consumer valves can be recorded and evaluated in a building automation and control system. The building automation and control system can then specify the required speed control setpoint. This makes it possible to ensure that the valve with the highest demand is always open as fully as possible (e.g. 95%).

**4.8.3 Pump run-on**

When the controller closes the control valve or switches the burner off, it leaves the pump running, e.g. for 5 minutes. The pump run-on protects the boiler or heat exchanger against overheating due to heat build-up.

**4.8.4 Pump cycling**

This function protects the deactivated pump against seizing by starting it, for example, for 30 seconds every 150 hours (pump kick).

**4.9 Frost protection features**

**4.9.1 System frost protection**

This function protects the system against freezing even if the heating is off (“Protection” mode, ensuring that the system is ready for operation). This is very often a two-stage function.

In systems with an outside temperature sensor, if the current outside temperature falls, for example, below +1 °C, the circulating pump is enabled intermittently (e.g. for ten minutes every six hours) and the controller controls the supply temperature to a minimum value. If the outdoor temperature continues to fall, e.g. to −5 °C, the circulating pump is switched on permanently. When the outside temperature rises above the preset limit value by a switching differential of e.g. 1 K) again, the active frost-protection stage switches off again.

In systems without outside temperature sensors provided the controller has need of a boiler or supply temperature, this temperature, with the appropriate switching limit values (e.g. supply temperature = 10 °C and 5 °C respectively) is used to enable the circulating pump (intermittent switching or continuously on).

**4.9.2 Room or building frost protection**

The building frost protection function protects the rooms from too low temperatures. It acts in all operating modes as a form of minimum room temperature limit control, and can be implemented with or without a room unit.

In heating control systems with room devices, if the room temperature falls below the selected frost protection temperature, e.g. during holiday operation, the heating is switched on and controlled to the set value (e.g. a room temperature of 5 °C).

In control systems without a room device, the controller switches the heating on the basis of the damped outdoor temperature (e.g. < 5 °C).
4.10 Inspector function
The following functions are temporarily activated for inspection measurements:
- The minimum boiler temperature is set to 64 °C, for example, and the high limit to 89 °C, for example.
- The supply temperature is controlled to 44 °C, for example, if no heat demand is present.
- The circulation pump is switched on.
The inspector function is usually terminated one hour after actuation of the respective mode selector switch or when the boiler controller housing is closed.

4.11 Manual heating system operation
The following generally applies to the “manual” mode:
- In case of two-position control: pump ON, burner enabled (monitored by boiler temperature limiter). Electro-thermal actuator deenergized (closed).
- In case of three-position control: Pump ON, electric motor actuator deenergized (operation via hand lever or special pulse buttons on the controller). Boiler temperature control is autonomous.

4.12 Control of district heat transfer
4.12.1 Transfer station
The transfer station is the connecting element between the district heat distribution network and the building consumer circuit. The heat that is delivered is transferred to the building consumer circuit either by direct (Fig. 4-28a) or indirect supply (Fig. 4-28b).

![Diagram of a district heat transfer station](image_url)

**Fig. 4-28 Schematic of a district heat transfer station**

- a) with direct supply
- b) with indirect supply

| A | Distribution network connection | 1 | Heat meter (normally installed on the return side) |
| B | Transfer station | 2 | Differential pressure controller |
| C | Building consumer circuit | 3 | Secondary supply temperature control |
|   |                     | 4 | Heat exchanger |
4.12.2 Heat meters

A heat meter belongs in every transfer station so that the supplied heat quantity can be billed correctly. A distinction is made whether the quantity of heat consumed is billed at a fixed price per kWh or whether the total operating costs of the district heat supply utility are allocated to the heat consumers. (Heating cost accounting). For the per kWh billing type, officially approved and calibrated heat meters are specified, whereas uncalibrated meters can be used for heating cost allocation.

Heat meters that can be calibrated are:
- Ultrasonic heat meters
- Magnetic-inductive heat meters
- Vane-type heat meters
**Installation in supply or return**  Manufacturers normally specify installation on the return side. The main reason is not primarily the lower medium temperature but correct mass flow calculation. The flow measurement of the heat meter detects only the flow volume in m³/h, whereas the heat transfer depends on the mass flow in kg. Therefore, the flow volume must be multiplied by the density of the medium, which varies with temperature. If the heat meter is designed for return-side installation, its arithmetic unit is programmed to use the return temperature for density calculation. If the meter is installed on the supply side, it measures the supply flow volume and calculates the mass flow with the density related to the return temperature, which gives rise to a measuring error. If for some reason a heat meter is installed on the supply side, the manufacturer must reprogram the device accordingly and identify it to ensure that, if the device is exchanged at a later date, the replacement device is also reprogrammed.

**Leakage quantity suppression**  If the flow volume through a heat meter falls below 10 % of the nominal flow rate, the measuring error increases sharply. Measurement can come to a standstill at a certain low flow volume, especially in heat meters with moving meter elements (vane). The flow rate that is no longer measured is referred to as the “leakage quantity.” In order to prevent this operating state from occurring, district heat control systems are equipped with so-called leakage quantity suppression, i.e. as soon as the manipulated variable of the transfer control valve falls below 10 % of its positioning range, the valve is closed.
4.12.3 Differential pressure control

The basic characteristic of a valve indicates the flow volume through it as a function of valve stroke at constant pressure difference across the valve. It is important for district heat supply utilities to know and be able to limit the maximum flow volume demand of each transfer station. Therefore, if the pressure difference across the valve is kept constant, the maximum flow volume through it can also be limited by a mechanical, electromechanical or electronic stroke limiter. This flow volume limitation is normally sealed by the district heat supply utility. Compensation of the pressure fluctuations in the district heat distribution network also improves the controllability of the transfer station.

Since differential pressure control and secondary supply temperature control (acting on a primary-side control valve) are standard for every transfer station, control equipment manufacturers supply so-called combined valves (Fig. 4-31). These are straight-way control valves with built-in differential pressure regulators.

![Fig. 4-31 Combined valve (front view and view of pressure pipes)](image)

![Fig. 4-32 Pressure measurement point and pressure ratios for the combined valve](image)

- $P_1$ = Pressure upstream of combined valve
- $P_2$ = Pressure downstream of volume-regulating section of valve
- $P_3$ = Pressure downstream of combined valve
- $\Delta p_w$ = Working pressure across the volume-regulating section of the combined valve
- $\Delta p_r$ = Pressure drop across differential pressure controller
- $\Delta p_{ges}$ = Pressure differential across the entire combined valve ($\Delta p_{ges} = \Delta p_w + \Delta p_r$)
- $\Delta p_{max}$ = Max. total differential pressure permitted across the almost-closed valve, whilst largely avoiding cavitation
- $\Delta p_{min}$ = Minimum differential pressure required across the fully open valve at nominal stroke, to ensure that the differential pressure controller will still respond reliably
- $p_{stat}$ = Static water pressure in the system pipework
- $\vartheta$ = Water temperature on primary side
- $M$ = Pump
4.12.4 Limiting functions

- Secondary-side supply temperature high-limit control in district heating networks with a supply temperature that is higher than the maximum permitted in the building consumer circuit.
- Primary-side return temperature high-limit control (constant or constant-shifting): this prevents the uneconomical return transport of heat back to the district heating plant.
- Return temperature low-limit control: to protect the heat producers.
- DRT limitation (Differential Return Temperature): if, for example in start-up operation, the difference between the primary and secondary return temperatures in a district heating network exceeds the selected DRT value (e.g. 10 K), the control valve is closed sufficiently far to reduce the primary flow volume until the primary return temperature falls to the DRT value. The purpose and benefit of DRT limitation is illustrated in Fig. 4-33.

A heat exchanger is sized such that it transfers the maximum required heat output during full-load operation. In this example, full-load operation is designed for 120 / 70 °C on the primary side and 80 / 60 °C on the secondary side. It is a known fact that the heat output results from the heat transfer surface area and the mean temperature difference between the primary and secondary circuits. If, in start-up mode, the secondary return has a temperature of 20 °C instead of the design value of 60 °C for full-load operation, the mean temperature difference between the primary and secondary circuits will be considerably greater than in full-load operation, so the heat output rises considerably above that calculated for full-load operation. This can mean that, in start-up operation, the transfer stations closest to the district heating plant consume considerably more heat than the calculated maximum, whereas in stations further away, the heat supply fails during the heat-up phase.
In many systems, the same heat producer is used for domestic hot water heating as for room heating. **Two main types** of domestic hot water charging are implemented:

- Domestic hot water charging with **internal heat exchanger**
- Domestic hot water charging with **external heat exchanger**

These two main types and the respective domestic hot water charging methods are described from section 5.1 onwards.

However, some general information on domestic hot water charging will be provided first.

**Starting/stopping domestic hot water charging**

Depending on the controller or system implemented, domestic hot water charging can be started/stopped via 1 or 2 sensors or via thermostats. If sensors and a controller are used, the setpoints can be adjusted at the controller, and the actual value can be read off. A possible frost protection function can also be realized via the controller. Thermostats must be set locally (accessibility) and provide no additional functions. The most suitable solution should be selected according to the requirements on the plant to be realized and the possibilities of the controllers and systems used.

![Diagram of Domestic Hot Water Charging using Sensors/Thermostats](image.png)

Fig. 5-1 Domestic hot water charging using sensors/thermostats for start/stop

- $B_{1,2} =$ Storage tank sensors; setpoints selectable at the controller ($R$)
- $T_{1,2} =$ Storage tank thermostats; setpoints only selectable directly at the thermostat ($T_{1,2}$)

**Charging temperature**

The charging temperature should be selected as low as reasonably possible (e.g. 58 °C) because lime and mineral precipitation occurs at excessively high charging temperatures (above approx. 65 °C), which has a negative effect on the maintenance costs and service life of the domestic hot water plant.

5. Control of domestic hot water plants

93
**Placement on the manifold**

The heating group for domestic hot water heating is usually located separately on the main manifold or on the manifold of a substation.

![Diagram of heating manifold with domestic hot water group (ON/OFF valve)](image)

As already mentioned in chapters 1 and 2, the placement of a heating group on the manifold is decisive, especially in start-up mode. Therefore, domestic hot water heating groups are usually placed at the front end of the manifold so that they can make full use of the available hot water without giving rise to unnecessary controller activities in the other heating groups. This also means that the entire manifold does not have to be heated up during domestic hot water charging in summer operation (with appropriate bypass configuration; cf. Fig. 5-2 and section 1.7.2.1)

**Producer side setpoint adjustment**

In systems whose heat producers are controlled based on outside temperature (cf. chapter 1), the operating temperature must be raised to the necessary charging setpoint (e.g. 65 °C) for the duration of domestic hot water charging. Especially in extensive systems, such as residential estates with heating centers, it must be ensured that:

- charging of a domestic hot water tank does not commence until the necessary temperature is available at the manifold, e.g. using an enabling thermostat (cf. Fig. 5-3),
- this switchover also takes place in a coordinated manner without forced charging, and that in such a case all connected domestic hot water storage tanks are charged,
- the heat producer is returned to weather-compensated control when charging is complete.

**Summer operation**

If the domestic hot water is also produced by the heating boiler in summer, it must be ensured that:

- the boiler is only started up when the water heater demands heat in a coordinated manner in case of multiple substations with domestic hot water charging,
- the charge pump is not enabled until the boiler temperature or the temperature at the manifold is sufficiently high for domestic hot water charging, e.g. via an enabling thermostat (cf. Fig. 5-3).
The coordination of multiple substations with domestic hot water charging as mentioned above can be achieved using controllers that are interconnected via a communication network. This coordination is also often implemented using an appropriate electrical circuit (NO/NC contacts) or a simple switching bus system (relay bus).

The following functions must be provided:

- If a domestic hot water charging system starts, the others are also forced to start (e.g. by overriding the “ON” thermostat) in order to make use of the available temperature level and to prevent individual domestic hot water charging systems from switching on and off one after the other, causing the heat producer to switch to the higher temperature on each occasion.
- Each individual domestic hot water charging system stops individually when the storage tank is full (e.g. “OFF” thermostat).
- The last domestic hot water charging system to stop also causes the heat producer to return to the normal boiler temperature.

Many controllers have highly complex coordination and forced charging functions (see further down). The appropriate technical documentation should be consulted for information.

### Forced charging

In most systems it is useful or necessary (depending on storage tank volume) for the domestic hot water storage tanks to be charged once or more every day within predefined intervals (e.g. 01:00 – 04:00). Storage tank sizing should take into account the system conditions (charging times, available space, geographical distribution etc.).

### Circulation pump

Many systems require a circulation pump on the consumer side (e.g. see Fig. 5-3) in order to keep the time to availability of hot water at the draw-off points within acceptable limits. In case of flow control (external heat exchanger without storage tank, cf. 5.2.1) the use of a circulation pump is highly recommendable for control reasons.

### Electric immersion heater

The majority of the domestic hot water charging systems described in the following can be supplemented with an electric immersion heater, for example to be able to leave the heat producer off outside of the heating season. The electric immersion heater normally has its own control and safety equipment (e.g. thermostats that have to be set separately).

Domestic hot water heating can also be provided exclusively by an electric heating register. Many standard commercial controllers and systems provide functions for controlling this type of domestic hot water heating, which will not be considered in detail here.
**Heat metering**

The energy required for generating domestic hot water is often recorded with a separate heat meter to provide appropriate input for heating cost accounting. The correct installation location of the heat meter on the manifold is important.

![Diagram of domestic hot water charging system with heat, enabling thermostat and circulation pump](image)

**Fig. 5-3** Domestic hot water charging system with heat meter, enabling thermostat and circulation pump

HM = Heat meter

T₁ = Enabling thermostat for DHW charging

M₂ = Circulation pump

**5.1 Domestic hot water charging with internal heat exchanger**

This type of domestic hot water charging is very frequently used, especially for individual domestic hot water storage tanks. It is very cost-effective, because standard, prefabricated storage tanks can be used.

![Diagram of domestic hot water charging system with internal heat exchanger](image)

**Fig. 5-4** Domestic hot water charging system with internal heat exchanger

**Suitable manifold design**

With this type of domestic hot water charging system, the return temperature rises sharply towards the end of the charging process (cf. Fig. 5-6). Therefore, it is not suitable for manifolds that require a low return temperature to the heat producer, e.g. condensing boilers, plants with storage tanks or heat pump systems.

The most commonly used circuits with domestic hot water charging systems with internal heat exchangers are as follows:

- With charge pump and without supply control
- Supply control with mixing circuit
- Diverting circuit
The domestic hot water storage tank is charged using a charge pump with open-loop control. The domestic hot water temperature is detected using one (or two) thermostats or sensors. An isolating valve (e.g. a straight-way valve S in Fig. 5-5) which has no control function is often also used with this type of domestic hot water charging.

![Diagram of domestic hot water charging system with charge pump, no closed-loop control](image)

This simple method of domestic hot water charging has the disadvantage that the heat exchanger is operated with the available supply temperature (e.g. boiler supply temperature) which can vary greatly according to the quality of the primary control. This means that no protection against calcification is guaranteed, i.e. temperatures > 65 °C can occur in normal charging operation.

![Progression (simplified) of the inlet and outlet temperatures over the charging state](image)

**Fig. 5-6** Progression (simplified) of the inlet and outlet temperatures over the charging state

- $\vartheta_1$ Domestic hot water storage tank charging temperature
- $\vartheta_2$ Outlet temperature of the internal heat exchanger
5.1.2 Supply control with mixing circuit

*Function*

The domestic hot water storage tank is charged by commanding the charge pump and mixing valve. The supply temperature to the storage tank is controlled by the controller using the mixing valve. The controller also starts and stops domestic hot water charging based on the temperature, which is measured by the storage tank sensor.

![Diagram of domestic hot water charging system with mixing valve](image)

**Legend:**
- \( B_1 \) = Supply temperature sensor (control sensor)
- \( B_2 \) = Storage tank sensor (ON/OFF via controller)
- \( Y \) = Mixing valve
- \( M_L \) = Charge pump
- \( M_Z \) = Circulation pump

For trouble-free operation with this type of domestic hot water charging the setpoint (w) for the supply temperature (\( B_1 \)) should be set approximately 2-5 K above the desired storage tank temperature.
5.1.3 Diverting circuit

Function

The domestic hot water storage tank is charged by commanding the diverting valve. This valve switches between the heating circuit and domestic hot water charging system, i.e. charging with absolute priority (cf. 5.3).

Fig. 5-8 Domestic hot water charging system with diverting valve

T = Storage tank thermostat (ON/OFF)
S = Diverting valve
M = Heating group pump (also charge pump)
M₂ = Circulation pump
5.2 Domestic hot water charging with external heat exchanger

This type of domestic hot water charging is used if the return temperature to the heat producer must be kept low (e.g. district heating substation, heat pump system etc.).

Fig. 5-9 Domestic hot water charging system with external heat exchanger (front, insulated)
5.2.1 Supply controlled with primary side valve

**Function** Domestic hot water heating is performed via a heat exchanger (flow control). The secondary side supply temperature (B₁) of the heat exchanger is controlled using the primary side straight-way valve (Y). The primary side pump (Mₚ) is switched on by the ON-command (T₁) from the storage tank, the closed-loop control is activated and so the valve is opened. The secondary side pump is switched on by a charge-enabling thermostat (T₃) if the primary side supply temperature is sufficiently high. This prevents cooling and mixing in the storage tank. The charging process is stopped when the lower storage tank thermostat (T₂) reaches the set temperature.

In order to ensure trouble-free operation, the charging setpoint (w) should be set approximately 2 K higher than the OFF temperature.

**Circulation pump** If this circuit is used without a storage tank, it is recommendable for control reasons (and not just for reasons of comfort) to use a circulation pump.

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**Fig. 5-10** Supply controlled with primary side valve

- T₁ = Storage tank thermostat (ON)
- T₂ = Storage tank thermostat (OFF)
- T₃ = Enabling thermostat (MS ON)
- B₁ = Supply temperature sensor (secondary side)
- Y = Straight-way valve
- Mₚ = Primary side pump
- Mₛ = Secondary side pump (also charge pump)
- M₂ = Circulation pump
5.2.2 Storage tank supply controlled via primary and secondary side valves

The secondary side supply temperature \(B_1\) is controlled via the secondary side valve \(Y_1\) in the first sequence and via the primary side valve \(Y_2\) in the second sequence (cf. Fig. 5-12). This provides for clear stratification in the storage tank and selective utilization of the heat exchanger’s output. The correct design of the heat exchanger for the output and temperature conditions is decisive for optimal operation. Switching of the two pumps and of the control are achieved via two thermostats \((T_1, T_2)\) or two sensors.

Depending on the plant, a maximum charging temperature \(B_2\) or a minimum return temperature \(B_3\) could be additionally specified, which would be integrated into the control.

![Fig. 5-11 Storage tank supply controlled via primary and secondary side valves](image1)

**Function**

- \(T_1\) = Storage tank thermostat (ON)
- \(T_2\) = Storage tank thermostat (OFF)
- \(B_1\) = Supply temperature sensor (secondary side)
- \(Y_1\) = Secondary side mixing valve (1st sequence)
- \(Y_2\) = Primary side mixing valve (2nd sequence)
- \(M_P\) = Primary side pump
- \(M_S\) = Secondary side pump (also charge pump)
- \(B_2\) = Max. charging temp. (primary side, optional)
- \(B_3\) = Min. return temp. (primary side, optional)
- \(M_Z\) = Circulation pump

The diagram in Fig. 5-12 shows the behavior of the primary and secondary side valves \((Y_1, Y_2)\) as a function of the control deviation \(x_w\) as well as the progression of the various temperatures over the charging state.

![Fig. 5-12 Manipulated variables \(Y_1\) and \(Y_2\), as a function of the control deviation \(x_w\) and progression of the various temperatures over the charging state](image2)
5.3 Special functions of domestic hot water charging systems

**Priority control of domestic hot water charging**

With the deliberately tight sizing of boilers, taking into account demand factors etc., and the relatively smaller required outputs for room heating (e.g. better insulation), the output share for domestic hot water heating is increasing. This makes it imperative to take this output share into account in the calculations and also to accommodate this situation in the control of the heating system. One course of action is to throttle or even cut off the output to the room heating system during domestic hot water charging. Due to the inertia of the building, this only gives rise to negligible, i.e. barely noticeable, temperature variations.

Some devices that can be used for heating group and domestic hot water control provide special functions for priority control of domestic hot water charging with respect to one or more heating groups. A distinction is normally made between two different types of priority control:

- **Absolute priority:** Heating circuits are disabled during domestic hot water charging
- **Sliding priority:** Heating circuits are throttled during domestic hot water charging, e.g. mixing valve

**Legionella control**

This is executed at regular intervals (e.g. once a week). The domestic hot water is heated to a higher temperature (at least > 60 °C) than the normal charging temperature in order to prevent the growth of the bacteria that cause legionnaire’s disease. It is important to note that the supply temperature from the heat producer must also be adjusted accordingly, i.e. higher than the usual setpoint for domestic hot water charging.

**High-limit control of domestic hot water return**

This limitation is used in plants with district heat exchangers (cf. chapter 4) in which the water for domestic hot water heating is drawn off on the secondary side of the heat exchanger. It is normally active during domestic hot water charging. The necessary detailed information can be found in the technical documentation of the various controllers.

**DRT limitation**

Many controllers deactivate DRT limitation (return temperature differential high-limit control, cf. chapter 4) in case of domestic hot water charging. The necessary detailed information can be found in the technical documentation of the various controllers.
By utilizing environmental heat, the electric heat pump can normally produce two to three times more heat energy than the electrical energy used to operate it. Therefore, the heat pump enables a highly effective use of electricity for the heating of buildings.

The heat pump exploits the thermodynamic properties of a refrigerant (e.g. Freon R134a) in a closed-cycle process (see Fig. 6-1).

The special property of refrigerants is that they evaporate at very low temperatures. This means that the plentiful quantities of energy in the environment (outside air down to –20 °C, lake or ground water from 4...12 °C and ground from 0...20 °C) are perfectly sufficient as heat sources in terms of the temperature level to vaporize the refrigerant.

The heat source is cooled by several Kelvin in the process. Energy is always required to vaporize a liquid. In this case, the vaporization energy is drawn from the environment. The vaporized refrigerant has absorbed this vaporization energy in the evaporator without a rise in temperature. The low temperature level does not permit this medium to be used directly in heating systems.

The temperature at which a medium evaporates when heat is added to it is the same temperature at which it condenses (liquefies) when it cools, i.e. heat is removed from it. Therefore, that temperature is referred to as the vaporization temperature in the one case and as the condensation temperature in the other.

The vaporization or condensation temperature depends on pressure. With increasing pressure, the vaporization or condensation point also rises in temperature. Based on these physical facts, the next step is highly logical: increase the pressure in order to raise the vaporization/ condensation point to a level at which the condensation process can be used for a heating system.
This is done using a **compressor**, which draws in the now gaseous refrigerant and compresses it. Auxiliary energy (e.g. electricity) is required for this process. In the case of a suction gas cooled compressor, this energy (motor heat) is not lost but is transferred to the refrigerant to be compressed and heats it up.

In the subsequent **condenser**, the heating water cools the hot gas down, causing it to condense and heating up the heating water.

After the condenser, the refrigerant is completely liquefied but still at a high pressure. The pressure is dissipated using an **expansion valve**, and the process cycle begins again.

*Origin of the name “heat pump”* The heat pump gets its name from this physical process: Heat energy absorbed at a low temperature level is “pumped up” to a level at which it can be used for heating purposes.
6.3 Heat sources

The heat source provides the heat pump with the necessary heat of vaporization.

6.3.1 Heat source: outside air

Outside air is highly available, so it is frequently used. However, the following properties must be borne in mind:

- The heat pump and its drive power must be designed relatively large for the coldest day (when the coefficient of performance is smallest; see 6.6.1).
- During mild weather, with correspondingly low thermal heat demand, a major excess of heat pump output is available, which may have to be stored.
- At outside air temperatures in the range +5 °C to –10 °C major icing occurs on the evaporator (condensing airborne humidity freezes on the evaporator surface at a temperature of < 0 °C). This gives rise to a sharp drop in evaporator performance. The ice must be regularly thawed off by an appropriate (energy consuming!) method.
- The air circulation can cause annoying fan noise, which must be reduced by appropriate soundproofing.

6.3.2 Heat source: ground

In order to use the ground as a heat source, either ground collectors (large area pipe network, usually filled with frost-proof liquid, e.g. water/glycol, installed at least 1.5 m below ground surface) or ground probes (deep drilling required) are used.

In order to use ground collectors, an appropriately large plot of land must be available, and investment costs are normally high. The use of ground probes requires drilling, which also gives rise to correspondingly high investment costs.
When using the ground as a heat source, it is very important to ensure that the heat source can regenerate (possibly installing a backup system such as solar collectors) otherwise the ground temperature will decrease too greatly so that the necessary output will no longer be available.

For the same reason, it is important with ground probes to ensure that the amount of heat extracted per meter of probe is not too large, otherwise the yearly energy coefficient will inevitably deteriorate.

With proper sizing and design, the ground is one of the least problematic heat sources for heat pump operation.

6.3.3 Heat source: ground water

The greatest problems with using ground water as a heat source are its availability and quality. However, if it is available in sufficient quantity and quality and at a suitable temperature level, it is the almost ideal heat source for heat pump operation (subject to authorization!).

6.4 Heat pump name

(In German-speaking countries) heat pumps are named according to the following principle:

\[ X - Y - Z \text{ heat pump} \]

where:

- **X**: Heat source heat transfer medium (e.g. air, water, brine etc.)
- **Y**: Heating system heat transfer medium (e.g. water, air etc.)
- **Z**: Compressor drive energy type (electricity, diesel, gas etc.)

Examples:

<table>
<thead>
<tr>
<th>Heat source</th>
<th>Heat pump name</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside air</td>
<td>Air-water electric heat pump</td>
</tr>
<tr>
<td>Ground</td>
<td>Brine-water electric heat pump</td>
</tr>
<tr>
<td>Ground water</td>
<td>Water-water electric heat pump</td>
</tr>
</tbody>
</table>
6.5 Operating modes
6.5.1 Monovalent operation

Monovalent operation (mono = one, singular)

In a monovalent heat pump heating system, the heat pump alone (Fig. 6-7) provides the required thermal heat in all possible operating states. Therefore, the heat pump must be designed for the maximum heat demand of the building heating system. The maximum possible heating water supply and return temperatures must be designed for the maximum permissible condenser outlet temperature (see. 6.8.2).

Fig. 6-6 System with monovalent operation with storage tank and heating groups

Fig. 6-7 Temperature frequency curve for monovalent operation

In case of failure of the heat pump, no alternative heating is available in a monovalent system.
6.5.1 Special case, monoenergetic operation

Since the maximum output of a plant only needs to be available for a relatively short time, a frequently used solution in single-family homes is an air-water heat pump with electrical auxiliary heating system to cover peak loads. This is actually bivalent/alternative operation (see 6.5.2.1) but since **only one type of energy is supplied**, electricity in this case, the term monoenergetic operation is used. Experience shows that a plant requires less energy if the switchover is performed manually. Additionally, night setback should be dispensed with so that boost heating does not become necessary.

6.5.2 Bivalent operation

Bivalent operation (bi = two, twin)

In a bivalent heat pump heating system, the heat pump alone provides the necessary thermal heat during mild weather and average winter weather. In case of very low outside temperatures, the heat energy demand is covered partially (parallel) or completely (alternatively) by an auxiliary heating plant (see Fig. 6-8).

![Fig. 6-8 System with bivalent operation with a heat pump, storage tank and boiler to cover peak heat demand](image)

Therefore, the heat pump need only be designed for part of the maximum heat demand of the building heating system.

The auxiliary heating system can be operated with the heat pump in different ways and must be designed and used accordingly. The following operating modes are distinguished:

- Bivalent alternative operation
- Bivalent parallel operation
- Alternative/parallel bivalent operation
6.5.2.1 Bivalent alternative operation

In this case, the heat pump only operates in mild weather and average winter weather. In case of very low outside temperatures and to cover the maximum heat demand, the heat pump is switched off and the auxiliary heating plant is switched on.

Fig. 6-9 Temperature frequency curve for bivalent alternative operation

The heating water supply and return temperatures must be designed for the maximum permissible condenser output temperature for the load states with heat pump operation (see 6.8.2). For the load conditions with alternative auxiliary heating plant operation, the heating water supply and return temperatures may exceed these maximum limits. However, the hydraulic connection of the auxiliary heating plant to the heating water circuit must be such that no heating water can circulate through the heat pump’s condenser during auxiliary heating plant operation (high-pressure operating limit).

The auxiliary heating plant must be designed for the total maximum thermal heat demand.

During operation, switchover from the heat pump to the auxiliary heating plant must occur as soon as the heat output of the heat pump is no longer sufficient. This is performed by the control equipment based on the outside temperature and/or heat source temperature.
6.5.2.2 Bivalent parallel operation

In this case, the heat pump and auxiliary heating plant are operated simultaneously to cover the maximum heat demand of the building heating system.

The heating plant must be designed for the maximum permissible return temperature (condenser inlet temperature, see 6.8.2.1). The auxiliary heating plant must be hydraulically connected in series with the heat pump in the heating water supply. The auxiliary heating plant is used to raise the condenser outlet temperature to the necessary supply temperature.

The auxiliary heating plant must be designed for the portion of the maximum heat demand that is not covered by the heat pump.

The auxiliary heating plant is brought online as soon as the heat output of the heat pump alone is no longer sufficient. This is performed by the control equipment based on the heating water supply temperature.
6.5.2.3 Alternative/parallel bivalent operation

This is a combination of parallel and alternative operation.

Low to medium thermal heat demand is covered by the heat pump alone. If the heat demand rises above the heat pump’s heat output, the auxiliary heating plant is operated in parallel according to the supply temperature. If the heat demand continues to rise above the heat pump’s operating limit, it is switched off (based on outside temperature or heat source temperature), and the entire maximum heat demand is covered by the auxiliary heating plant.

![Temperature frequency curve for parallel/alternative bivalent operation](image)

The hydraulic integration of the auxiliary heating plant in the system must be such that:
- it is connected in series in the heating water supply during parallel operation.
- no heating water can circulate through the heat pump’s condenser during alternative operation
- The auxiliary heating plant must be designed for the total maximum thermal heat demand.

6.5.3 Operating mode selection

*Correct operating mode*

The choice of the most appropriate operating mode (optimal in terms of energy and cost/benefit ratio) depends on the following criteria:
- Yearly progression of the building’s thermal heat demand
- Yearly progression of the heat pump’s thermal heat output
- Demand-dependent progression of the heating water supply and return temperatures
- Yearly frequency of the various heating load states.
The coefficient of performance $\varepsilon$ (epsilon) provides for a comparison of different heat pumps and is the ratio of the current thermal heat output to the (electrical) energy supplied to a heat pump plant (electrically operated) in order to produce it.

Coefficient of performance $\varepsilon = \frac{\text{Current heating capacity}}{\text{(Electrical) energy supplied}} = \frac{\text{Useful heat}}{\text{Energy consumption}}$

The greater the value of $\varepsilon$ the greater the energy efficiency of the heat pump.

The coefficient of performance is often referred to as the COP value. The term originates from the (American) refrigeration industry, and is also used in relation to refrigeration machines.

The operating concept of a heat pump heating system must take account of the fact that $\varepsilon$ (and therefore the heating capacity of the heat pump) increases with a decreasing difference between the condensation temperature and the vaporization temperature.

This means in practical terms that a heat pump for building heating:
- has the lowest coefficient of performance, i.e. provides the lowest heating capacity, when thermal heating demand is greatest.
- has an increasing coefficient of performance, i.e. provides increasing heating capacity, with decreasing thermal heating demand.
- has the highest coefficient of performance, i.e. provides greatest heating capacity, when thermal heating demand is least.

A certain coefficient of performance $\varepsilon$ applies only to a given, momentary operating state.

![Graph showing the progression of the coefficient of performance as a function of the temperature difference between the condensation and vaporization temperatures.](image)

Fig. 6-12 Example for the progression of the coefficient of performance as a function of the temperature difference between the condensation and vaporization temperatures

1 Coefficient of performance
2 Temperature difference
The yearly energy coefficient $\beta$ provides a measure for the actual efficiency of a heat pump plant.

**Yearly average is important**

The yearly energy coefficient $\beta$ is the yearly average (mean annual value) of the coefficients of performance $\varepsilon$ occurring within an operating year of a heat pump plant. Typical yearly energy coefficients $\beta$ occurring in actual practice are as follows, according to heat source:

<table>
<thead>
<tr>
<th>Heat source</th>
<th>$\beta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside air</td>
<td>2,5</td>
</tr>
<tr>
<td>Ground</td>
<td>3</td>
</tr>
<tr>
<td>Ground water</td>
<td>3,2</td>
</tr>
</tbody>
</table>

The yearly energy coefficient $\beta$ is determined by simultaneous measurement of the yearly electrical energy consumption of the compressor and auxiliary drives (in kWh) and of the yearly heat production (also in kWh) and heat losses of the storage system.

$$\text{Yearly energy coefficient } \beta = \frac{Q_{\text{HP}} - Q_{\text{ST}}}{W_{\text{HP}} + W_{\text{Pumps}} + W_{\text{Control}} + W_{\text{...}}}$$

- $Q_{\text{HP}}$ = quantity of heat produced by the heat pump
- $Q_{\text{ST}}$ = heat losses in the storage system
- $W_{\text{HP}}$ = energy consumption of the heat pump
- $W_{\text{Pumps}}$ = energy consumption of the evaporator and condenser pumps
- $W_{\text{Control}}$ = energy consumption of the control equipment
- $W_{\text{...}}$ = energy consumption of other components, such as defrosting equipment, Carter heater, ...

This requires an appropriate measuring concept (planning phase) for the heat pump plant, and the plant must be equipped with the necessary sensors and meters (electricity and heat meters).
6.7 Heat pump controllability

A heat pump without controllable heat output produces excess heat during partial load heating operation.

The correct choice of heat pump output control must be specified by the heat pump manufacturer and taken into account in plant sizing and design.

6.7.1 Heat output control directly at the heat pump

More detailed information on the methods of heat output control directly at the heat pump is contained in the “Refrigeration engineering” training module (B08RF). Only the effects of the control methods on the yearly energy coefficient are described here.

6.7.1.1 Hot gas bypass or suction throttling

Heat pump output control using a proportionally controlled hot gas bypass or suction valve is pointless, because in either case a reduction of the heat output does not give rise to an even approximately equivalent reduction of drive power consumption. Therefore, both hot gas bypass and suction throttle control give rise to very poor yearly energy coefficients for the heat pump.

6.7.1.2 Compressor valve unseating

In appropriately equipped multiple cylinder piston compressors, individual cylinders can be activated or deactivated in stages. This is achieved by unseating the suction valves of the cylinders to be deactivated (e.g. by electrohydraulic means). However, this type of heat pump output control is not energy efficient because considerable friction losses occur in reduced power operation and the inertia of the inactive pistons must still be overcome. Therefore, compressor valve unseating gives rise to a relatively poor yearly energy coefficient for the heat pump.

6.7.1.3 Compressor speed control

Heat pump output control via multistep (step switch acting on three-phase multispeed motor) or stepless (frequency converter acting on three-phase motor) speed control provides almost optimal energy use.

6.7.2 Heat pump ON/OFF control

Electric heat pumps with installed drive loads of up to approximately 40 kW are normally controlled by ON/OFF control today, because the previously mentioned output control methods do not provide optimal energy use, or they give rise to high investment costs. It should be noted with regard to this type of control that frequent switching of heat pumps reduces the life of mechanical parts, increases losses at standstill and causes frequent electrical network fluctuations due to the high starting currents.

Therefore, in order to prevent excessively frequent switching, the heat pump heating system must have sufficient heat storage capacity which can temporarily store the excess heat produced by the heat pump and also temporarily cover the heating system’s heat demand when the heat pump is off.
To be on the safe side, the heat pump should also be switched with a delay in order not to exceed a maximum permissible number of starts per hour (e.g. max. 3 starts per hour). The permissible frequency of starts is often specified by the electrical utility.

The following controlled variables are normally used for starting the heat pump:

- In heat pump heating systems with a heat buffer or storage tank, the heat pump is started when the buffer or storage tank temperature falls.
- In floor heating systems with no additional heat buffer or storage tank, the heat pump is started when the heating water return temperature falls.

In both cases, the setpoint for these switch-on temperatures can be weather compensated. This gives rise to an improved yearly energy coefficient of the heat pump and, in the case of floor heating systems, to an adjustment of the stored floor heat according to actual thermal heat demand.

The heat pump is normally stopped when the condenser inlet temperature of the heating water rises (condensation pressure operating limit). In case of highly variable heat source temperatures (= variable heat output of the condenser), the switch-off setpoint must be compensated according to heat source temperature (see 6.8.2.1).
6.8 Heat pump operating limits

A heat pump must only be operated within certain operating limits. These are defined by the following, among other things:

- Minimum permissible refrigerant vaporization pressure
- Maximum permissible refrigerant condensation pressure

6.8.1 Vaporization pressure operating limit

The minimum permissible vaporization pressure for a heat pump plant depends on:

- the maximum permissible compression ratio of the compressor used, i.e. on compressor design.
- the minimum permissible refrigerant vaporization temperature for the heat source used (e.g. if water is the heat source, the vaporization temperature must be above 0 °C because of the risk of icing).

Low operating limit

The vaporization pressure is prevented from falling below the low operating limit by the low-pressure safety pressostat. It switches the compressor off if the refrigerant pressure on the compressor inlet side falls below the defined operating limit.

This occurs, for example, if the heat source temperature falls so low that sufficient vaporization heat can no longer be supplied to the refrigerant to maintain the minimum permissible vaporization temperature.

In heat pump plants where the amount of heat available from the heat source is limited, therefore, the heat pump must be switched off by a temperature controller as soon as the heat source temperature falls below the operating limit.

This heat source temperature operating limit must be specified and taken into account by the planning engineer and builder of the heat pump heating system. It determines the switchover point from heat pump operation to auxiliary heating in bivalent heat pump plants.

The following absolute vaporization pressures apply to several common refrigerants at various typical vaporization temperatures:

<table>
<thead>
<tr>
<th>Vaporization temperature</th>
<th>−10 °C</th>
<th>0 °C</th>
<th>+10 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vaporization pressure Freon R407C* or R290 (propane)</td>
<td>3,5 bar</td>
<td>5,0 bar</td>
<td>6,8 bar</td>
</tr>
<tr>
<td>Vaporization pressure Freon R404A*</td>
<td>4,2 bar</td>
<td>5,7 bar</td>
<td>7,7 bar</td>
</tr>
<tr>
<td>Vaporization pressure Freon R134a*</td>
<td>2,2 bar</td>
<td>3,1 bar</td>
<td>4,2 bar</td>
</tr>
</tbody>
</table>

* R407C and R290 supersede R22, R404A supersedes R602, and R134a supersedes R12 whose use has been prohibited in European countries since 1.1.2000 at the latest (earlier in some cases)

6.8.2 Condensation pressure operating limit

The maximum permissible refrigerant operating pressure for a heat pump plant is normally limited to 25 bar for safety reasons (legal requirements, specifications).

The operating pressure is prevented from exceeding this operating limit by the high-pressure safety pressostat which is normally set to approximately 24 bar.
The condensation pressure for normal operation should be approximately 2 bar below the high-pressure switch-off point, i.e. it should not exceed approximately 22 bar.

The following absolute condensation pressures and refrigerant condensation temperatures apply to several common refrigerants:

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Normal operation</th>
<th>High-pressure safety pressostat</th>
<th>Operating limit</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Condensation pressure</td>
<td>Condensation temp.</td>
<td>Condensation temp.</td>
</tr>
<tr>
<td>Freon R407C and R290</td>
<td>22 bar 24 bar 25 bar</td>
<td>56 °C 60 °C 62 °C</td>
<td>53 °C 56 °C 58 °C</td>
</tr>
</tbody>
</table>

The maximum condenser outlet temperatures of the heating water that can be achieved in trouble-free operation depend on the heat exchanger surface area in the condenser and on the refrigerant condensation temperature at 22 bar. The resultant maximum heating water temperatures are as follows in practice:

- Freon R407C, R290: max. 50 °C
- Freon R404A: max. 47 °C
- Freon R134a: max. 70 °C

As already noted, the condensation pressure is prevented from exceeding the operating limit by the high-pressure safety pressostat. It switches the compressor off before the refrigerant pressure on the discharge side of the compressor exceeds the defined operating limit.

This occurs, for example, if the condenser inlet temperature of the heating water rises to such a level that sufficient condensation heat can no longer be extracted from the refrigerant to maintain the maximum permissible condensation temperature.

Therefore, the heat pump must be switched off by a temperature controller* as soon as the condenser inlet temperature of the heating water exceeds the operating limit.

This operating limit is specified by the planning engineer and builder of the heat pump heating system. In the case of heat pump heating systems with variable heat source temperature and, therefore, variable heat output of the condenser, this operating limit is also variable (see 6.8.2.1).

* The detection of the condenser inlet temperature must be sufficiently fast to ensure that, in case of rapid temperature rises, the heat pump is switched off before the high-pressure safety pressostat is triggered. Normal thermostats are usually unsuitable for this purpose. Alternatively, the heat pump could be switch off by a pressure switch in case of rising condensation pressure, e.g. set to 2 bar below the high-pressure safety limit. It must be borne in mind that the pressure switch must be designed for a very large number of operations and must be connected to the closed refrigerant system (interventions in the refrigerant system are unpopular). Additionally, any desired outside temperature compensation of the pressure switch setpoint is not easily possible. Therefore, switch-off with rising condenser inlet temperature is recommended as a general rule.
6.8.2.1 Determining the maximum permissible condenser inlet temperature

In order to determine the maximum permissible heating water return temperature $\theta_{RL\text{ max}}$, the following data must be known:

- Progression of the heating water supply and return temperatures $\theta_{VL}$ and $\theta_{RL}$
- Maximum condenser outlet temperature $\theta_{VL\text{ max}}$ achievable in normal operation (e.g. 50 °C with the use of R407C; see 6.8.2)
- Maximum heating of the heating water $\Delta\theta_{\text{Erwärmung}}$ across the condenser at all possible outside temperatures or with different heat sources and their temperature progressions
- $\theta_{RL\text{ max}} = \theta_{VL\text{ max}} - \Delta\theta_{\text{Erwärmung}}$

Basically, the behavior of the condenser for the following two heat output $Q_H$ cases must be considered:

- **$Q_H = \text{constant}$** (e.g. ground water as heat source)
  - constant temperature difference across the condenser $(\Delta\theta_{\text{Erwärmung}})$ and therefore constant maximum condenser inlet temperature (see Fig. 6-14, ②)
- **$Q_H = \text{variable}$** (e.g. outside air as heat source)
  - the temperature difference across the condenser $(\Delta\theta_{\text{Erwärmung}})$ increases with rising outside temperature (and constant volume of water transported), which means that the maximum condenser inlet temperature becomes lower than the design point (see Fig. 6-14, ③)

**Outside temperature compensation of $\theta_{RL\text{ max}}$.**

This means that the condenser inlet temperature must be compensated according to the outside temperature if it is used for control purposes.
Fig. 6-14 Determining the maximum permissible condenser inlet temperature ($\theta_{\text{RL max}}$) of a heat pump plant with monovalent operation and with variable and constant heat sources

1. Maximum permissible condenser inlet temperature ($\theta_{\text{RL max}}$) with a **variable** heat source
2. Maximum permissible condenser inlet temperature ($\theta_{\text{RL max}}$) with a **constant** heat source
3. Maximum possible condenser outlet temperature

$\delta_{\text{RL}}$ Heating group supply temperature
$\delta_{\text{RL}}$ Heating group return temperature
$Q_{\text{H var}}$ Heat output of the heat pump with a **variable** heat source (e.g. outside air)
$Q_{\text{H cons}}$ Heat output of the heat pump with a **constant** heat source (e.g. ground water)
$Q$ Required heat output (heat demand) of the plant
6.9 Heat storage

Heat storage, an important element

Heat pump heating systems with ON/OFF control must have sufficient thermal heat storage capacity in order that:
- the heating water temperature fluctuations due to the ON/OFF operation have no undesirable effects on the heating system (room temperature fluctuations).
- unacceptably frequent switching of the heat pump can be avoided (service life, requirements of electricity utility).
- the heating system can still be operated during intentionally long interruptions in heat pump operation (e.g. night storage).

6.9.1 Heat buffers / storage tanks

Heat buffer
A heat buffer is a small storage tank (often referred to as a technical storage tank) that is used for hydraulic decoupling of the heat pump and heating system and to prevent unacceptably frequent switching of the heat pump.

Storage tank
Storage tanks are also used for hydraulic decoupling of the heat pump and heating system and for long-term storage of the heat required by a building (to cover gaps in the electrical energy or heat source supply). Additionally, in bivalent heat pump heating systems, the excess heat produced by an uncontrollable solid-fuel auxiliary heating plant can also be stored over long periods in the storage tank.

6.9.2 Stratified charging and stepped charging of storage tanks

Stratified charging
With stratified charging, the storage tank is charged in layers in a single pass with a constant condenser outlet temperature. It is only used in plants with storage tanks because it offers the following advantages over stepped charging:
- Exact control of the storage tank temperature
- Constant supply temperature is guaranteed
- Maximum utilization of the storage tank’s capacity
- Improved stratification
- No effect on the evaporator

Stepped charging
With stepped charging, charging is performed in layers with multiple passes at increasing condenser outlet temperature. It is used in relatively small systems with one heating group in combination with “technical storage tanks” (see above), because it has too many disadvantages for use in other systems.
6.9.3 Storage tank charge control

Storage tanks are normally operated with so-called charge control. This keeps the condenser outlet temperature (storage tank charging temperature) as high as possible at all times. The refrigerant condensation pressure, which in this case is normally controlled to a pressure approximately 2 bar below the switch-off pressure of the high-pressure safety pressostat, is used as the controlled variable for charge control (see Fig. 6-15). This pressure sensor is installed by the heat pump supplier.

Since the condensation pressure can rise very quickly on heat pump startup, the mixing valve ($Y_1$) must be able to rapidly mix in “cold” water. This can prevent an overshoot of the pressure above the high-pressure safety limit.

As an alternative to charge control via the condensation pressure, the condenser outlet or inlet temperature can be used provided the boundary conditions indicated in 6.8.2.1 are observed.

In case of storage tank charging with charge control, the storage tank temperature is always kept at the same high level regardless of the return temperature and the heat output of the condenser.

Fig. 6-15 Basic hydraulic and control circuit of a heat pump heating plant with storage tank

- $B_1, B_2$: Storage tank temperature sensors
- $B_3$: Pressure sensor (refrigerant condens. pressure)
- $B_4$: Temperature sensor as alternative to $B_3$
- $Y_1$: Mixing valve, fast opening (e.g. Siemens Landis & Staefa solenoid valve)

If actuators with different running times for opening and closing, it may be necessary to install the valve as shown in Fig. 6-16 so that cold water can be mixed in sufficiently rapidly in order to prevent disturbance of heat pump operation.
6.9.4 Plants without storage tanks

In small and very small plants, the investment costs, space requirement and heat losses of the technical storage tank are often carefully considered. However, a technical storage tank has so many advantages that dispensing with it only makes sense in the rarest of cases.

A technical storage tank should only be dispensed with if the following prerequisites are met:

- Almost constant heat source output (variations < 5 K)
- Good storage capacity of the heat output system (e.g. slow floor heating system with good damping characteristics)
- No thermostatic valves or only few installed in the system
- Control and hydraulics must be designed, adjusted and optimized as a complete system from source to output.
- Hydraulic balancing is indispensable

Is the floor an adequate substitute for a storage tank?

Among other things, the following points must be taken into account in heat pump floor heating systems with no storage tank:

- The heat pump and floor heating system must be precisely matched in terms of water volume, pressure drop, temperature difference etc.
- The heat stored in the floor cannot be used freely on demand.
- The supply temperature exceeds the return temperature by the uncontrollable quantity \( \Delta \theta_{\text{Heating}} \).
- The return temperature rises with increasing charging of the floor.
- The switching differential of the weather-compensated return temperature two-position controller for switching the heat pump must be optimally adjusted:
  - small enough to prevent room temperature fluctuations
  - as large as possible in order to prevent too frequent switching

Correct adjustment of the control requires appropriately long-term operational experience.
In the past, complicated hydraulic circuits were often recommended and implemented for controlling solar plants with the aim of achieving as high a solar panel outlet temperature as possible. This gave rise to a considerable reduction in the efficiency of such solar panel plants. The requirement for a high solar panel outlet temperature came at the price of considerable energy losses, because the lower the mean solar panel temperature is \( t_{\text{mean}} = 0.5 \cdot (t_{\text{in}} + t_{\text{out}}) \) the greater the efficiency and energy yield. With the deliberate raising of the outlet temperature, the inlet temperature, and therefore the mean solar panel temperature, also rises.

Human beings have a very good sensory apparatus for perceiving high temperatures as hot but none at all for energy. If a solar panel system is operating poorly in terms of energy, we cannot directly detect that fact with our senses. Let us bear that in mind on the following pages.
The traditional design of a solar panel plant is as follows:

![Diagram of solar panel plant]

**Principle of operation**

As soon as the temperature at sensor B₁ in the solar panel exceeds that at the storage tank sensor B₂ by an adjustable amount, the pump starts. The pump stops again if this condition is no longer met. In good controllers, separate temperature differences can be selected for starting and stopping the pump, where the temperature difference for start-up must be considerably greater.

If there is only one heat exchanger in the storage tank, it is always located at the bottom where the storage tank temperature is lowest.

**7.2 Solar plant with two heat exchangers in the storage tank**

If two heat exchangers are present, the heat can be applied selectively according to the storage tank’s stratification. This is a must if the storage tank is very large in comparison to the solar panel surface area (usually approx. 100 l of storage volume are available per m² of solar panel surface area). However, this circuit also provides major advantages with normally sized storage tanks (approx. 100 l of storage volume per m² of solar panel surface area) if hot water should be heated exclusively by the sun for as long as possible during the transitional period without heating demand.
The additional upper heat exchanger allows considerably higher temperatures to be achieved in the upper part of the storage tank, because the volume of water to be heated is considerably smaller. This is very important for domestic hot water heating, because a temperature of at least 40 °C is required for showers. If the output of the solar panels had fallen to a low level, without the upper heat exchanger it would only be possible to heat the entire storage volume to a temperature of, say, 30 °C, which would not be sufficient for showering.

**Principle of operation**

As soon as the temperature at sensor B₁ in the solar panel exceeds that at storage tank sensor B₂ by an adjustable amount, the pump starts. The pump stops again if this condition is no longer met. If the temperature at sensor B₁ in the solar panel is also approximately 4 K higher than that at storage tank sensor B₃ (upper heat exchanger) the valve is switched over, and the solar panel medium flows through the upper heat exchanger first, then through the lower one. If the temperature difference between B₁ and B₃ falls below 2 K, the valve is switched back, and the medium now only flows through the lower heat exchanger.
In order to fully utilize the heat from the roof, however, the medium must always flow through the lower heat exchanger as well so that as much heat as possible is transferred to the water in the storage tank. This also ensures that the solar panel inlet temperature is kept as low as possible, and the efficiency of the solar panels does not deteriorate (as mentioned at the beginning). Therefore, the following “either-or” circuit is incorrect, and the valve at the lower heat exchanger should always be omitted (see Fig. 7-5). This also applies if the storage tank is charged via an external heat exchanger.

**Fig. 7-4** Incorrect “either-or” circuit
Medium should flow through upper and lower heat exchangers in series if temperature is sufficiently high!

**Fig. 7-5** Incorrect “either-or” circuit
Omit valve for bypassing the lower heat exchanger!
7.2.3 Solar plant with two solar panel fields

Solar panel plant with two differently oriented fields

If two solar panel fields with different orientations are present, the following circuit is used:

![Diagram of solar plant with two differently oriented fields]

Fig. 7-6 Solar panel plant with two differently oriented fields
1 Solar panels
2 Pump 230 V
3 Storage tank

Principle of operation
The control criteria are identical to those of the traditional plant, but the control principle is applied twice acting on the respective pump in each case.
In a correctly planned solar plant, the heat is always transferred to a storage tank. Additionally, the lower part of the storage tank is always reserved for solar energy. This means that only the solar panel plant can apply heat to this part of the storage tank. The water is always coldest at the bottom of the storage tank. This guarantees on the one hand that the solar panel plant can be activated and deactivated completely independently of the remaining system and also that the solar panel plant can be activated at the lowest thermal radiation level.

Fig. 7-7 Ready piped solar storage tank
References (Second updated edition / 2004)

- Technical documentation:
  - Thermostats: ETHERCO, CH-Steinhausen
  - Heating boilers and flue gas heat exchangers: Viessmann, DE-Allendorf
  - Domestic hot water storage tanks: Domotec, CH-Aarburg
  - Heat pump process pictures: Siemens Heiztechnik, DE-Kulmbach

- Recknagel Sprenger Schramek “Taschenbuch für Heizung + Klimatechnik”
- Buderus “Handbuch für Heizungstechnik”
- “Impulsprogramm Haustechnik” Bundesamt für Konjunkturfragen, CH-Bern

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<table>
<thead>
<tr>
<th>ASN-No.</th>
<th>Titel</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-91899-de</td>
<td>Das h, x-Diagramm</td>
</tr>
<tr>
<td>0-91899-en</td>
<td>The psychrometric chart</td>
</tr>
<tr>
<td>0-91900-de</td>
<td>Gebäudeautomation – Begriffe, Abkürzungen und Definitionen</td>
</tr>
<tr>
<td>0-91900-en</td>
<td>Building automation</td>
</tr>
<tr>
<td>0-91910-de</td>
<td>Messtechnik</td>
</tr>
<tr>
<td>0-91910-en</td>
<td>Measuring technology</td>
</tr>
<tr>
<td>0-91911-de</td>
<td>Regeln und Steuern von Heizungsanlagen</td>
</tr>
<tr>
<td>0-91911-en</td>
<td>Control of heating plants</td>
</tr>
<tr>
<td>0-91912-de</td>
<td>Regeln und Steuern von Lüftungs-/Klimaanlagen</td>
</tr>
<tr>
<td>0-91912-en</td>
<td>Control of ventilation and air conditioning plants</td>
</tr>
<tr>
<td>0-91913-de</td>
<td>Regeltechnik</td>
</tr>
<tr>
<td>0-91913-en</td>
<td>Control technology</td>
</tr>
<tr>
<td>0-91914-de</td>
<td>Kältetechnik</td>
</tr>
<tr>
<td>0-91914-en</td>
<td>Refrigeration technology</td>
</tr>
<tr>
<td>0-91915-de</td>
<td>Wärmerückgewinnung im Kältekreislauf</td>
</tr>
<tr>
<td>0-91915-en</td>
<td>Heat recovery in the refrigeration</td>
</tr>
<tr>
<td>0-91916-de</td>
<td>Einführung in die HLK- und Gebäudetechnik</td>
</tr>
<tr>
<td>0-91916-en</td>
<td>Introduction to building technology</td>
</tr>
<tr>
<td>0-91917-de</td>
<td>Hydraulik in der Gebäudetechnik</td>
</tr>
<tr>
<td>0-91917-en</td>
<td>Hydraulics in building systems</td>
</tr>
<tr>
<td>0-91918-de</td>
<td>Stetige Leistungsregelung im Kältekreislauf</td>
</tr>
<tr>
<td>0-91918-en</td>
<td>Modulating capacity control in the refrigeration cycle</td>
</tr>
</tbody>
</table>
The information in this document contains general descriptions of technical options available, which do not always have to be present in individual cases. The required features should therefore be specified in each individual case at the time of closing the contract.

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