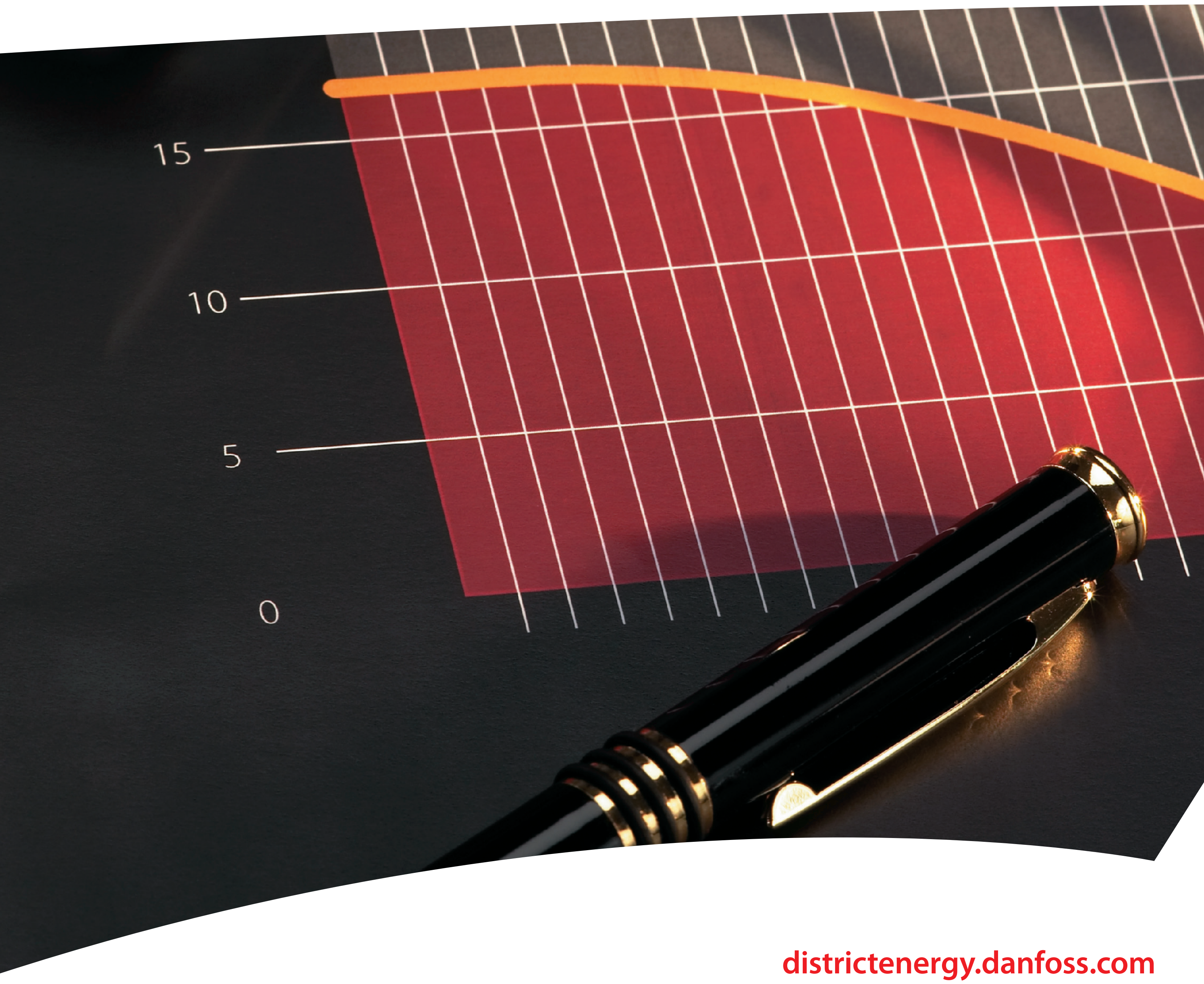
MAKING MODERN LIVING POSSIBLE

Technical paper

Control concepts for DH compact stations

Herman Boysen, Product Application Manager, Danfoss A/S
Jan Eric Thorsen, Application Centre Manager, Danfoss A/S

Published in EuroHeat & Power IV/2004
English edition



districtenergy.danfoss.com

TECHNICAL PAPER

Control concepts for DH compact stations

1. Abstract

As a result of a wide range of functional requirements to DH house stations and flat stations, a variety of control concepts are commercially available on the market today. The specific behaviour, or control performance, of those concepts vary significantly and are depending on many factors. This article describes the basic functionality of four control concepts, a thermostatic controller, a proportional controller and a parallel and serial coupling of those. The focus is on plate heat exchanger control, and the investigation is based on dynamic simulations, i.e. on models verified up against practical measurements. Specific cases, but also the general behaviour of the concepts are described. Control performance is investigated regarding stability, temperature overshoots, stationary temperature deviations and settling time. Finally, some recommendations are made in regard to selecting the control concept depending on demands and supply conditions.

Author(s)


Herman Boysen, Product Application Manager, Danfoss A/S
Danfoss District Energy, Nordborg, Denmark,
+45 7488 4494 · boy@Danfoss.com

Jan Eric Thorsen, Application Specialist, Danfoss A/S
Danfoss District Energy, Nordborg, Denmark,
+45 7488 4494 · jet@Danfoss.com

2. Introduction

The trend is towards small compact multifunctional controllers, requiring lower system prices, improved control functions, space saving and improved flexibility rate. The most common functions to be integrated or combined are thermostatic control, proportional control, differential pressure control, idle control and DHW priority control. Both self-acting and electric controls and combinations of those are used [1/2]. To investigate functionality and potential of control components and concepts, dynamic simulation has been applied.

3. Modelling of components

The computational software tool used for this investigation is the well-known *Simulink* program package, by The MathWorks Inc. [3]. *Simulink* is used for modelling, simulating and analysing of dynamic systems.

The principles for the component models are described in [4/5/6]. Here the presentation is limited to the thermostatic controller. The thermostatic controller model is based on a force balance, including the actuator bellow element, the springs and the O-ring friction, as shown in fig. 2. A linear valve characteristic is implemented. The first order thermal time constant, as a function of the flow rate, is included in the model.

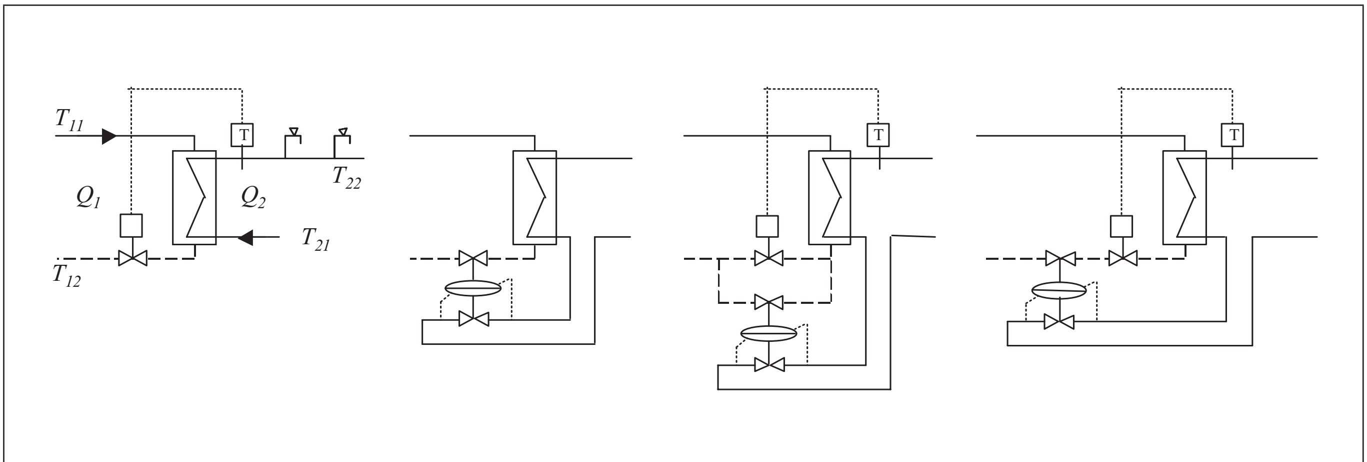


FIGURE 1: Thermostatic controller, proportional controller, and a parallel and serial coupling of those

4. Control concepts investigated

Concept 1:
Heat exchanger control by thermostatic controller

Concept 2:
Heat exchanger control by proportional controller

Concept 3:
Heat exchanger control by parallel coupling of thermostatic and proportional controller

Concept 4:
Heat exchanger control by serial coupling of thermostatic and proportional controller

To compare the results some identical boundary conditions are used for all four concepts.

$dP_{control\ valve} = 1$ bar, $T_{11} = 65$ °C and 90 °C, $T_{21} = 10$ °C. The simulations are to be regarded as specific cases, intending to investigate the concept specific behaviours.

4.1 Heat Exchanger Control by Thermostatic Controller

Fig. 3 and 4 show results for this concept. A linear valve characteristics is implemented,

with a T_{22} deviation, disregarding $Q_2 = 100$ l/h of approx. 6 °C for the $T_{11} = 65$ °C case. For the $T_{11} = 90$ °C the value becomes approx. 4 °C, which is explained by the lower primary flow rate needed, resulting in lower P-deviation. Normally 100 l/h is a low value for tapping flow. The general T_{22} temperature range, reflecting $T_{11} = 65$ °C to 90 °C and disregarding $Q_2 = 100$ l/h, is approx. 8 °C.

Looking at peak temperatures, this concept has the highest peak; see fig. 4 at time 800 sec. Here the effect of low flow, Q_2, around the sensor, resulting in slow sensor becomes visible. Furthermore, there is no feed forward signal, like e.g. a proportional controller, to react immediately. Comparing the settling time, this concept has the longest in general. Especially looking at a step response where the system is only marginally stable, see fig. 4, time interval 500 to 600 sec. All P-controllers, like a thermostatic linear valve, will

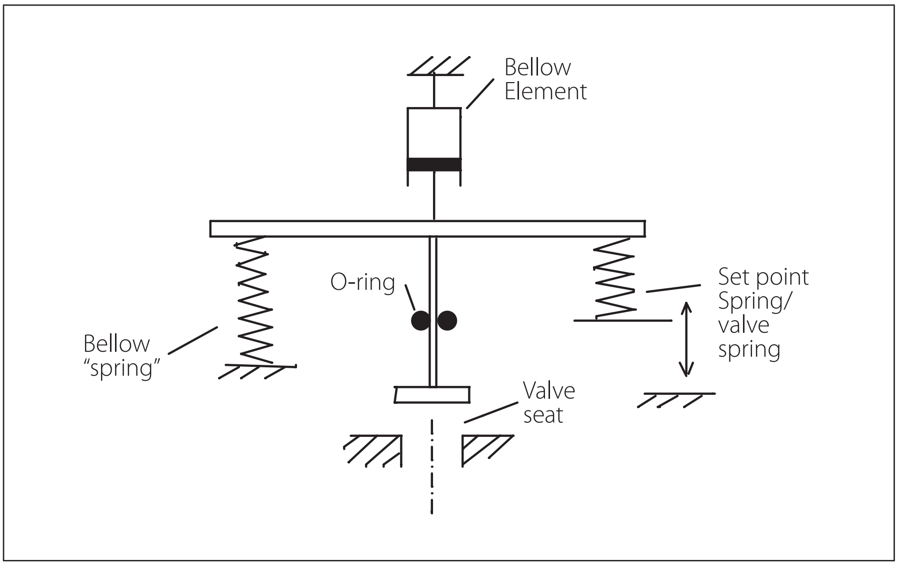


FIGURE 2: Schematic presentation of thermostatic controller

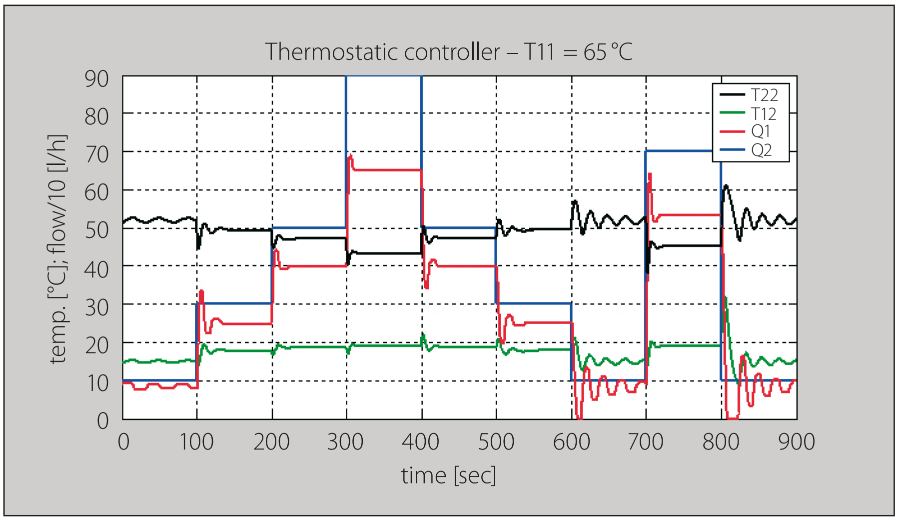


FIGURE 3: Simulation result for thermostatic controller, $T_{11} = 65$ °C

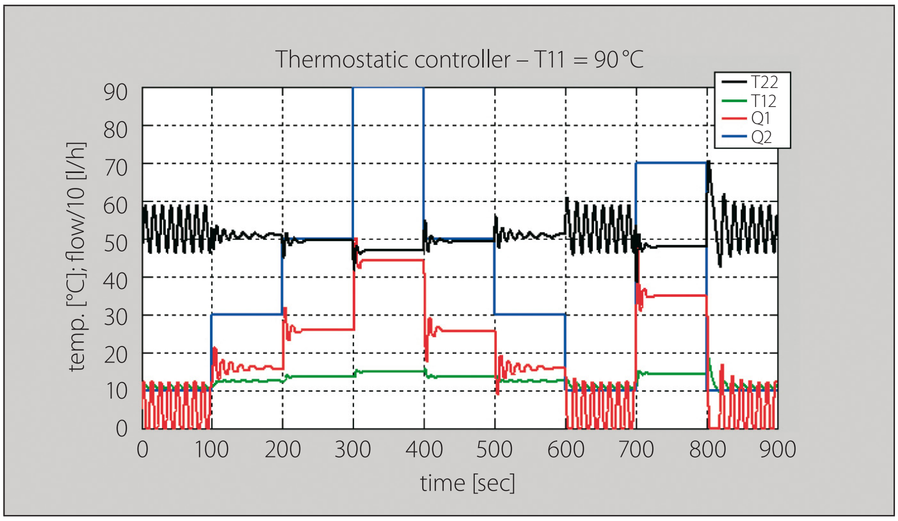


FIGURE 4: Simulation result for thermostatic controller, $T_{11} = 90$ °C

become unstable at a certain low value of Q_2, due to the loop-gain, dT_{22}/dkv-valve, goes towards infinity when Q_2 goes to zero.

4.2 Heat Exchanger Control by Proportional Controller

This concept is stable for all values of Q_2. Looking at the temperature deviation for T_{22}, but disregarding the lowest Q_2 tapping steps of 100 l/h, the result is approx. 4°C for $T_{11} = 65$ °C and for $T_{11} = 90$ °C. The influence from hysteresis on the proportional controller is relatively more visible at low tapping flows compared to higher flows. Furthermore, the general T_{22} temperature level is significantly different comparing $T_{11} = 65$ °C (T_{22} level about 42 °C) and 90 °C (T_{22} level about 58 °C), resulting in a temperature range of approx. 20 °C, $Q_2 = 100$ l/h disregarded. Simulation results for this concept are shown in fig. 5.

Peak temperatures are relatively low and the settling time relatively short. The actual values are highly dependent on the dynamics of the heat exchanger, e.g. thermal capacity, which becomes more dominant at low flow ranges.

4.3 Heat Exchanger Control by Parallel Coupling of Thermostatic and Prop. Controller

Fig. 6 and 7 show simulation results for this concept. Since there are two valves in parallel, the thermostatic valve model implemented has half the capacity compared to concept 1. The proportional valve contribution is slightly lower compared to concept 2. At $T_{11} = 90$ °C the main part of the flow belongs to the proportional controller. At $T_{11} = 65$ °C the flow split is nearer to half and half.

Benefits of this concept are the relatively small tap temperature deviation T_{22} in combination with the wide stable Q_2 tap flow range. Disregarding tap flows of $Q_2 = 100$ l/h, the deviation is approx. 5 °C at $T_{11} = 65$ °C and 2 °C at $T_{11} = 90$ °C. The general T_{22} temperature range, reflecting $T_{11} = 65$ °C to 90 °C, disregarding $Q_2 = 100$ l/h, is approx. 7 °C.

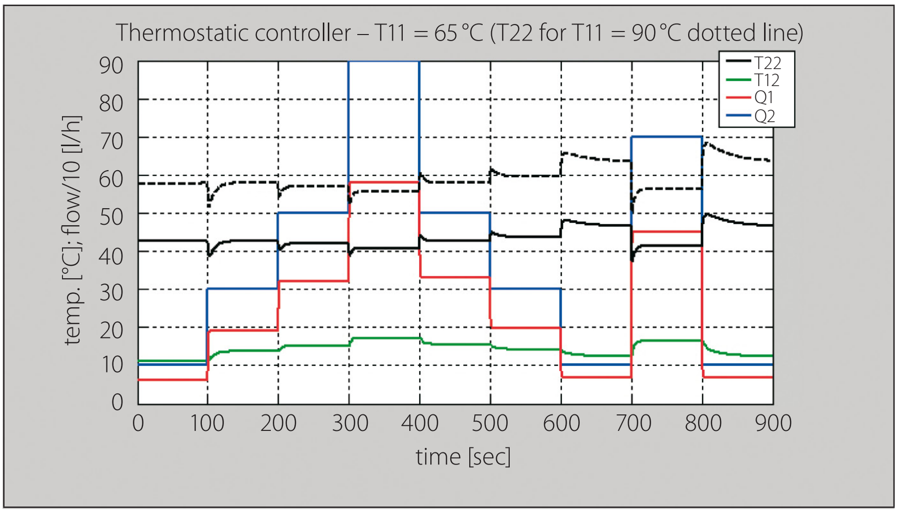


FIGURE 5: Simulation result for proportional controller, $T_{11} = 65$°C and T_{22} for $T_{11} = 90$°C

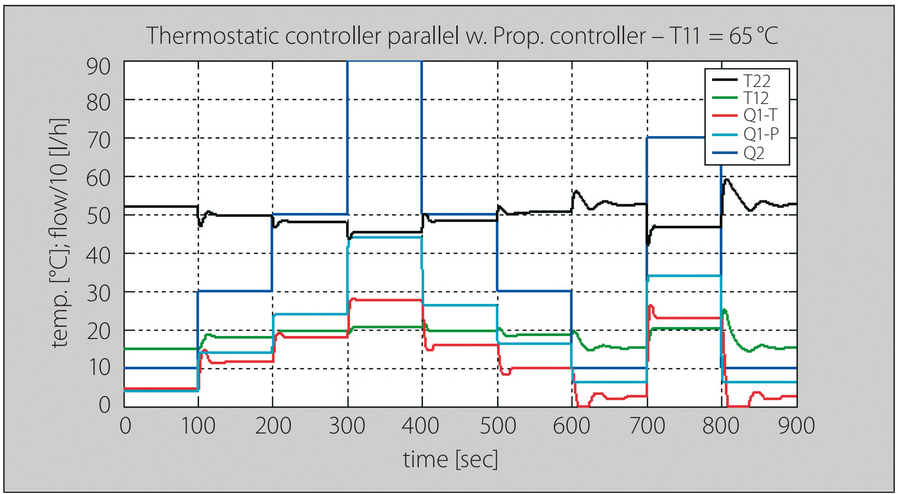


FIGURE 6: Simulation result for thermostatic contr. parallel with prop. controller for $T_{11} = 65$ °C

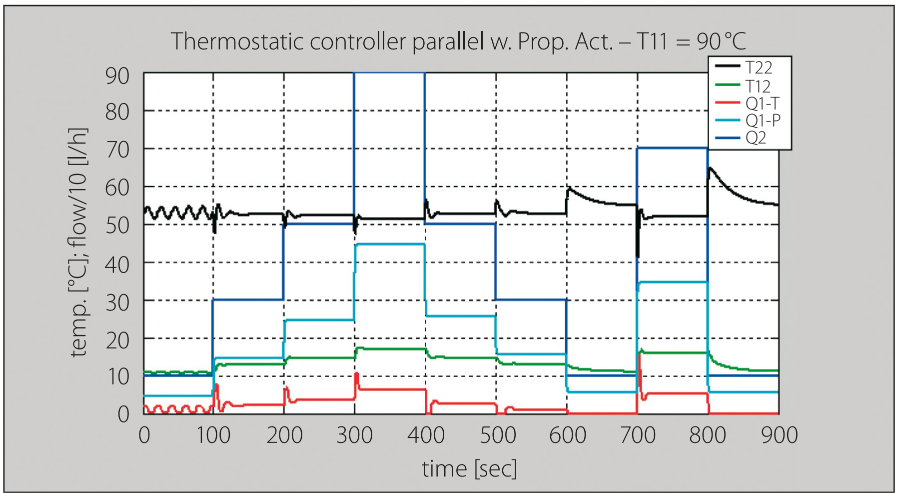


FIGURE 7: Simulation result for thermostatic contr. parallel with prop. controller for $T_{11} = 90$ °C

The situation of a totally closed thermostatic valve is shown in fig. 7 time range 600 to 700 sec. Here the proportional controller is determining the tap temperature, which becomes high due to the "high" proportional controller flow contribution. If the flow part of the thermostatic valve is small, oscillations will occur. But only the smaller part of the flow will oscillate, which results in insignificant tap temperature peak to peak values, see fig. 7 time range 0-100 sec.

Considering temperature peaks and settling time, this concept performs better than concept 1. Regarding tap temperature range in general, this concept is performing likely or slightly better than concept 1 and clear better than concept 2. Comparing peaks and settling time to concept 2, the results depend on what step is regarded. Looking at fig. 7 and 5 at time 300 sec. setting time of concept 3. is shorter, since the thermostat gives a short boost on the primary flow. Looking at fig. 5 and 6 at time 600 sec. the settling time of concept 2 is shorter. The oscillation of the thermostatic valve is the explanation. In general, temperature peaks are of the same range where comparable. The fact, that the initial tap temperature is as different prior to some of the steps, a comparison is not straightforward to do in all cases.

4.4 Heat Exchanger Control by Serial Coupling of Thermostatic and Prop. Controller

Since this concept puts the proportional controller and thermostatic controller in series, the capacities of each one should be increased compared to the other concepts. The simulation results for this concept are shown in fig. 8 and 9. Looking at the temperature deviation for T_{22}, but disregarding the lowest Q_2 tapping steps of 100 l/h, the result is approx. 7 °C for T_{11} = 65 °C and approx. 4 °C for 90 °C. The general T_{22} temperature range, reflecting T_{11} = 65 °C to 90 °C, disregarding Q_2 = 100 l/h, is approx. 10 °C. Concept 4 gives a wider stability range compared to the thermostatic controller (concept 1), since the proportional controller acts as a limiter for the thermostatic controller gain, dkv/dT_{22}, principally resulting in a larger P-band for the thermostatic controller. Most effect of reducing the thermostatic controller gain is, however, achieved at the lower supply temperature T_{11} = 65 °C. At higher supply temperatures or higher supply differential pressure the "damping" effect of the proportional actuator decreases. The concept opens the possibility for decreasing the tap temperature range by increasing the proportional controller contribution. But the consequences are a reduced Q_2 flow range where stability is obtained at the high supply temperature

For the T_{11} = 65 °C case, there is almost no oscillation, compared to concept 1, fig. 3, which indicates the effect of the serial connected proportional actuator. At T_{11} = 90 °C the oscillations are becoming more dominant, i.e. compared to concept 3, see fig. 7. The reason is that the thermostatic controller capacity becomes more dominant. Due to the serial coupling, concept 4 is able to control variations in supply differential pressure, without high T_{22} tap temperatures, which are drawbacks of concept 2 and 3. Regarding peak temperatures and settling times, this concept is to be ranged between concept 1 and 3.

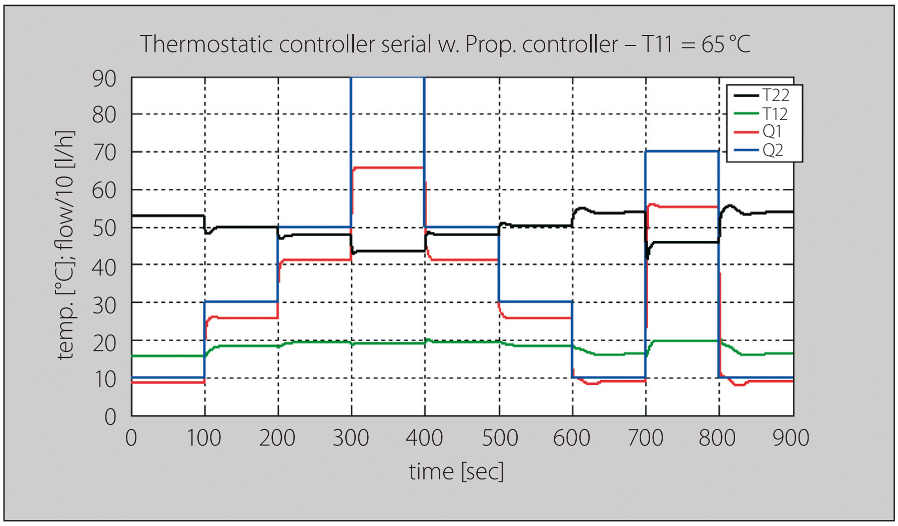


FIGURE 8: Simulation result for thermostatic contr. serial with prop. controller for T_{11} = 65 °C

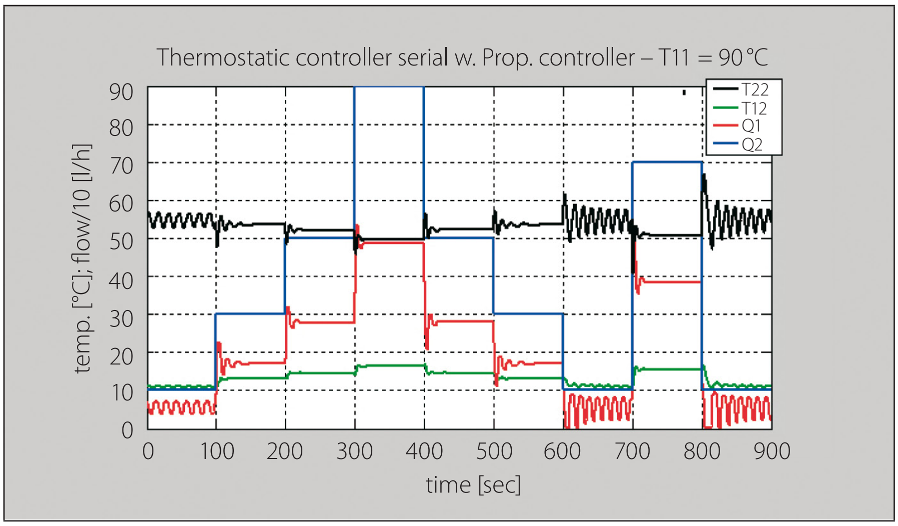


FIGURE 9: Simulation result for thermostatic contr. serial with prop. controller for T_{11} = 90 °C

	Peak temperatures	Steady state error	Stability range	Settling time	dP range
1. Thermostatic contr.	Fair	Good	Fair	Fair	Good
2. Proportional contr.	Best	Poor (1)	Best	Best	Very Poor (2)
3. Parallel coupling	Good	Good	Good	Good	Poor (2)
4. Serial coupling	Fair	Fair/Good	Fair/Good	Good	Good

(1) for T11 constant the performance is good
(2) by introducing a differential pressure controller the drawback will be eliminated

TABLE 1: Control characteristics for the four concepts

5. Control Performance Summary

In table 1, the general characteristics of the concepts are summarized. The concept potentials to handle variations in supply differential pressure is included, and stated in column "dP range". The conclusion should be seen in relation to how each concept performs when designed for covering a typical primary temperature range as specified.

6. Recommandations

As shown in the simulation cases, the four control concepts have their specific behaviour regarding heat exchanger control. If the supply conditions are constant, T_{11} and $dP_{control\ valve}$ the proportional controller concept is performing good in a simple way. If the supply temperature only is changing concepts 1, 3 and 4 are relevant. In this case concept 3 gives the best control performance followed by concept 4.

If the supply differential pressure is changing only, concepts 1 and 4 are relevant, where concept 4 give the best control performance. If the proportional controller is equipped with a setting adjustment possibility, a manual seasonal adjustment due to varying supply conditions is a well-known compromise. To improve the control performance in all four cases at changing supply differential pressure, a differential pressure controller could be introduced. Besides this, the differential pressure controller introduces a flow limiting function, introducing hydraulic balance in the distribution system. As to stand-by operation (no tapping of DHW) the proportional controller in concept 2 and 4 provides immediately zero primary flow, which is an advantage in case where the scaling issue is relevant. In this case a bypass thermostat with a relative low set point (~ 40−45 °C) could keep the system "warm" at stand-by. On the opposite the concepts 1 and 4 give higher comfort (taping temperature reaches set point faster) when tapping after a long stand-by periode.

7. Conclusion

This paper describes by means of case simulations the control performance of four concepts. Depending on the supply conditions and the demand to control performance, different concepts are to be preferred. There is no top of the line concept. The success depends on the demands and supply conditions. Due to this, the different concepts have their market share today. In practical DH house or flat station cases, where specific requirements are to be fulfilled and documented, laboratory tests are normally performed on prototype DH stations. Typical focus is on control performance, capacity considerations, heat loss and stand-by operation.

References

[1] L. Happe, *Danfoss AVTI Multifunctional Compact Controller*, Offprint EuroHeat&Power, No 5009, September 2002.

[2] *Danfoss AB valve*, Presented 2004 at Poznan International Installation Fair (INSTALACJE 2004).

[3] *Simulink, Dynamic System Simulation for MATLAB®*, Version 4, Users Guide, The Math Works, Inc. 2000.

[4] Herman Boysen & Jan Eric Thorsen. *Control Concepts for DH Compact Stations Investigated by Simulations*. Presented at 9th International Symposium on District Heating and Cooling, Finland, August 2004.

[5] J.E. Thorsen, *Dynamic Simulations of DH House Stations*, 7. Dresdener Fernwärme-Kolloquium, Germany 2002. Printed in EuroHeat&Power, June 2003.

[6] Atli Benonysson & Herman Boysen. *Optimum control of heat exchangers*, Presented at 5th International symposium on automation of district heating systems, Finland, August 1995

More information

Find more information on Danfoss District Energy products and applications on our homepage: **www.districtenergy.danfoss.com**

Danfoss A/S · DK-6430 Nordborg · Denmark · Tel.: +45 74 88 22 22 · Fax: +45 74 49 03 95
E-mail: districtenergy@danfoss.com · www.districtenergy.danfoss.com

 Produced by Danfoss A/S, DH-SM/PL © 08/2011