Demand-oriented Hydronic Heating System and the Active One-pipe System Design Tool

Zlevor Ondřej[1,*], *Dostál* Jiří[1]

[1]CTU in Prague, University Centre for Energy Efficient Buildings, Buštěhrad, Czech republic

Abstract. This article is focused on hydronic heating systems that use pumps as the control actuators instead of valves. Those systems are called “demand-oriented”, while the systems being controlled by valves are called “supply-oriented”. Reader gets an overview of various versions of both demand- and supply-oriented systems using both one- and two-pipe topologies, supported by a brief historical outline, current state of the art and basic advantages and disadvantages of the introduced hydronic systems. The special interest is given to the one-pipe demand-oriented systems, which offer several benefits in comparison with nowadays widespread systems (mainly supply-oriented two-pipe), however, the computation complexity of design of such systems inhibits their more frequent utilization. This paper also introduces a computation and optimization tool to help with the design of one-pipe demand oriented hydronic heating systems, which eases the designer to size the AHU (Air Handling Units) and to set the mass flows of fluid in the system branches.

1 Introduction

Over millennia, heating of occupied spaces was ensured by a fire burning directly in the heated room or by a hot air conveyed from the fireplace. It all changed at the beginning of the 18th century, when a Swede Marten Trifvald in 1716 designed the first central hydronic heating system, which was utilized in a greenhouse [1]. However, it took almost another 150 years till the central hydronic heating systems spread among resident houses significantly.

In the 18th century dozens of the steam heating systems used to be in operation, in many cases these systems exploited waste heat from steam engines. Radiators were connected by a single pipe which had both the steam supply and the condensate drainage function. Drawbacks of the steam heating systems such as noise, complexity and also the health hazard of users inhibited extension of those systems into resident buildings and at the end of the 19th century they were no more competitive with - at that time - more and more succeeding hydronic heating systems.

Nevertheless, massive spread of hydronic heating came in the second half of the 19th century. At first the gravity systems were built (no circulator pumps existed in those days), since the 50s of the 20th century the electrical circulator pumps had been utilized, which allowed designers to use pipes of smaller diameters and to install the heating systems in a variety of buildings. In the 70s and 80s the number of one-pipe systems increased significantly, mainly due to the material savings during an installation. One-pipe networks from that period were sensitive to inaccuracies in design and realization and to changes of the system, which could rapidly decrease an efficiency and functionality of such systems.

1.1 Present state

Nowadays mostly two-pipe hydronic heating systems are in action, by then several ways of control with different levels of complexity are used. By far the most widespread is the quantitative regulation by changing the hydraulic resistance of the branch with a radiator, which enables the water flow control. The easiest quantitative regulation actuator is a manual valve, however, according to nowadays legislative norms (e.g. [2] in CR) it is no more possible to use it. A solution with an automatic function is a thermostatic valve that autonomously mechanically controls its opening depending on the room temperature. More up-to-date solution is to use Pressure Independent Control Valves (e.g. [3] or [21]), which are utilized mostly by fan-coil units (FCU), but it is possible to find them on the market also in a version for radiators. PICV contains a mechanism for maintaining a constant pressure drop across the control valve, by that the flow through the controlled branch depends only on the valve opening and not on the pressure changes in a system [3].

Another possible way of control is not to control mass flows through radiators with throttle elements, but to use decentralized pumps assigned to each heat exchanger (HX). Such a solution is already on the market, e.g. a low-power pump [4], which contains electronic control system and is designed to be connected to the radiator.

Let’s call systems utilizing quantitative regulation as “passive” or “valve” and systems using the qualitative

* Corresponding author: ondrej.zlevor@cvut.cz

regulation as “active” or “pump”. The company Wilo uses different terminology; systems with the quantitative regulation are called “supply oriented” and with the qualitative regulation as “demand oriented”. It is also possible to meet the term “centralized” for systems with a central pump and decentralized for decentralized pumping systems.

2 Hydronic heating system topologies overview

2.1. Present (passive) one-pipe system

The one-pipe heating system contains the the primary (main) circuit, which goes through the heat source, and the secondary circuits (branches with HXs). Figure 1 depicts several connections of a radiator to the one-pipe hydronic heating system. The variants that allow the HXs to be independently controlled is a connection with a parallel bypass (Fig. 1b) and a version with a bypass valve (Fig. 1c). The diverter tee armature [5] is used to split the flow between the HX branch and the bypass.

The vertical one-pipe heating system were used in the 70s and 80s, e. g. in GDR (East Germany), where the main motive were lower installation costs of such a system. The advantages of the passive one-pipe systems are

- lower costs,
- no need of electrically driven control elements,
- not so dense piping network.

From the point of view of costs are the passive one-pipe systems still reasonable [6], but there are several disadvantages joint with it:

- hard to modify or extend the heating system,
- a little robustness against inaccuracies in design and realization – hard to reach the designed values during the system implementation or modification.

There are studies claiming that the passive one-pipe systems consume 20 % more thermal and even 70 % more electrical energy than similar passive two-pipe systems do [7].

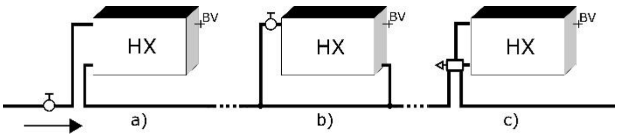

Fig. 1. One-pipe system HX connections. *BV - Bleeding valve*

To ensure that even the last radiator in a set provides the required heat flow it is necessary that the water entering the last HX, already cooled by flowing through the previous HXs, is still of the high enough temperature. It can be reached by increasing the mass flow, which can cause too high mass flow rate in the first HXs of the one-pipe branch and therefore increasing their heat flow. In such a case the return water flowing back to the boiler is not cooled to the designed temperature and in a system with a condensing boiler it makes the system less efficient.

There are attempts to solve this problem, such as circulator pumps driven by the return water temperature [8], or the automatic valves in the return pipe [9]. According to the return water temperature the valve changes its opening and this way by changing the hydraulic resistance of the system it controls the return water temperature. Such a device can keep the return water temperature at the designed level and to keep the boiler efficiency in the required range.

2.2 Passive two-pipe system

In a two-pipe (parallel) distribution system every heat exchanger is supplied from a common supply pipe and returns water into the common return pipe. It is the most utilized hydronic heating system. Compared to the passive one-pipe systems it offers several advantages:

- easy modification and extensions,
- only insignificant dynamic thermal interaction between radiators – influence of radiator heat flow according to the inlet water temperature,
- no need of electrically driven control elements.

Still there are several disadvantages:

- need of hydraulic balancing during implementation,
- pumping energy dissipation on regulation elements (control and balancing valves),
- dynamic pressure interactions among radiators.

If the valve is supposed to have a satisfactory regulation ability (so called valve authority), it is desired that the head loss over the fully opened valve should be similar to the head loss of the regulated branch. It is recommended to use valves with authority round 0.33, in which case the head loss over the regulation valve is approximately half of the head loss of the regulated branch [10]. The usage of control valves increases the head loss, which negatively affects the needed pumping energy.

Figure 3a depicts a scheme of a two-pipe (parallel) direct-return system. In this system, the length of the path from the discharge port of the circulator through the

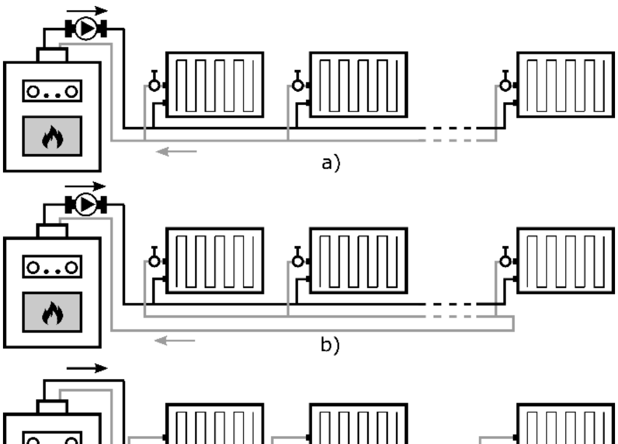

Fig. 2. Two pipe topologies: a) direct-return, b) reverse-return (Tichelmann), c) active

common inlet pipe, branch with the HX and back through the common return pipe, is different for each HX. Therefore the values of the differential pressure losses over each branch pipe are various and it is important to perform the hydraulic balancing. To avoid the hydraulic balancing the two-pipe reverse-return (Tichelmann) layout can be used (Fig. 2b). If the branches have a very similar hydraulic resistance and the system is properly designed, the reverse-return system is self-balancing. Nowadays the hydraulic separators are frequently getting used to avoid interactions between the primary circuit (circuit with the heater) and secondary circuit (circuit with branches with HX).

Zone temperature regulation in case of a passive two-pipe heating system is done by thermostatic radiator valves or by electronic radiator valves controlled by a thermostat.

2.3 Active two-pipe system

In an active two-pipe heating system there is a pump installed to every radiator, that is able to continuously control the mass flow within the radiators. Fig. 2c contains a scheme of such a system. It is necessary to place a check valve to the radiator branch to prevent the back-flow when the pump is turned off. Compared to the valve (passive) two-pipe systems the pumping (active) system has several advantages:

- there are regulation valves in the system, so the pumping energy dissipation is much lower,
- hydraulic balancing is not necessary, the design flows are ensured by pumps,
- the design is simpler – one pump type can operate with a wide scale of radiator sizes.

Disadvantages of an active two-pipe system are:

- still some pressure losses over the check valves,
- pressure interaction can cause regulation oscillations,
- installation costs are nowadays still high, but by utilization with FCUs and compared to prices of the electronic PICV valves it is not a big issue anymore (e.g. the small pump with electronic with the housing is offered for 88 € + 56 € [11], while the price of a PICV starts at 100€ [12]),
- pumps need a wired connection, which represents extra costs in typically wireless applications, such as radiators (not costly compared to the system using servo valves).

Several companies are already offering the active two-pipe technology. From 2001 to 2009 a few research projects ran in a cooperation with the TU Dresden university. These projects were focused on the development and testing of components for heating systems controlled by pumps. The results of the tests accomplished in a testing flat declare 20 % savings of the thermal and 70 % savings of the electrical energy, compared to a heating system controlled by thermostatic valves [13]. However, the amount of thermal energy savings is taken from a comparison of a system, controlled by thermostatic valves with only a one thermostat for the whole house, to a system using a zone regulation by pumps. That is to say, the savings caused by the zone regulation and savings caused by running a pump-controlled system got mixed. The interesting readouts are the electrical energy savings, which is evidently caused by the used topology. A simulation analysis [14] states that despite a lower efficiency (pumping energy/electrical energy input) of the small decentralized pumps compared to the large central one in a passive two-pipe system, the overall pumping energy demands are lower in an active two-pipe system than in the passive system, because of the energy dissipation on the control valves.

The active two-pipe system design is not harder than the passive two-pipe system design. The designed mass flows through the heat exchangers are the same, so also the same pipe sizes and the same radiators can be used. The only thing to do is to add the hydraulic separator to separate the primary and secondary loop, check valves and the circular pumps. The pump speed is regulated continuously which allows the designer to use a one pump type for a wide range of radiators. It makes the design easier and more error- or modification-tolerant.

2.4 Active one-pipe system

Compared to the passive one-pipe system, the active one-pipe system contains a secondary pump assigned to each heat exchanger in every secondary loop, which generates the water flow through the HX. The secondary loops (loops with radiators) are connected to the primary loop through a twin tee fitting. The return water from a HX flows back to the primary loop and is mixed into the bypassing supply water. The supply and return hole in the twin tee are placed at the same coordinate alongside the primary pipe, due to that there is no differential pressure between them. Consequently, the secondary loops are pressure independent on the primary loop – a change of a flow in the primary loop does not affect the flow in the secondary loops. Moreover, if the pump in the secondary loop is turned off, there is no flow in the secondary loop radiator. In such a system there are only thermal interactions between the primary and secondary loops. The radiator heat flows are continuously controlled by the pump speed according to the zone temperature demands.
The advantages of the active one-pipe hydronic system are:

- the system contains generally only two pipes diameters (primary and secondary), therefore the sizing of every single branch piping considering pressure losses is no more necessary,
- secondary loops are hydraulically separated from the primary loop, which eliminates the need of the hydraulic balancing of the system,
- time and material savings (less pipes, connections, valves and the plumber's work),
- one type of a pump in the secondary loop makes it possible to control a wide range of heat exchangers – the system is robust against design inaccuracies,

- compared to the active (pump driven) two-pipe systems there is no pumping energy dissipation on the check valves, so the pump compensates only a small pressure loss of the secondary circuit (so even a small pump is sufficient),
- the amount the overall dissipated pumping energy is the lowest of the introduced topologies,
- a one-pipe system contains less liquid (water, glycol) then a comparable two-pipe system.

The disadvantages are:

- several occupants can have the older passive one-pipe systems related to the higher operational costs and with a low comfort ([7, 15, 16]), which can, as a consequence, harm the reputation of the active one-pipe systems,
- this solution became feasible as late as the wet rotor pumps and BLDC engine control developed, that is to say, there is not much awareness and experiences with the active one-pipe systems,
- temperature interactions among secondary loops have to be taken into account already during the system design; moreover, the design requires iterative methods,
- dynamical thermal interactions appear.

Dozens of active one-pipe systems are already in service in the USA, e.g. [17].

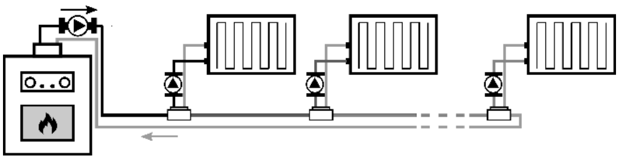

Fig. 3. Caption of the Figure 1. Below the figure.

3 Sizing of heat exchangers in a one--pipe system

One of the reasons, why designers avoid the one-pipe systems utilization is, next to the historical negative connotation, also the laborious design. Every heat exchanger affects by its function the supply water temperature for the following heat exchangers. That is to say, after every single size change of one heat exchanger, it is necessary to recompute values of the other system variables (temperatures, mass flows, heat flows), which solves the systems as an optimization problem and through the iterative recomputing the system variables it finds the optimal HX sizes and the needed water mass flows through the heat exchangers, in order to deliver the desired amount of heat. However, the optimal sizes of the HXs do not correspond to the HXs offered on the market. In sake of finding a real world solution with a utilization of available HXs, choosing always the first bigger and the first smaller HX (in the meaning of the nominal heat flow) from the catalogue list, gives us 2^n sets by permutations of smaller and bigger HXs. Then the tool finds the water mass flows needed to deliver the demanded heat flows for each set of HXs and displays all solutions to the designer.

3.1 Design optimizer

The tool for the one-pipe hydronic network design is available in a Github repository [18]. To be easily accessible, the user interface of the optimization tool was made as a MS Excel file. This file allows the user to fill in the demanded values and afterwards a script implemented in Python language reads data from the MS Excel file, runs the optimization and writes the results into another Excel file. Not every user has a Python installed on his computer, then even though the actual optimization program was written in the Python language, it got built as an executable file to make the tool more accessible.

The solution has two steps:

1. Find the optimal solution by continuous changing of all variables. The optimization variables are $Q_{\mathrm{n}}, \dot{m}_\iota$ (nominal HX heat flows, mass flows through the HX). The mass flow through the primary loop is computed from the sum of all heat flow demands and temperature reduction of the supply water.
2. According to the resulting optimal HX sizes, HX with the nearest lower and higher nominal heat flow are chosen and put together into 2^n sets. Each set is then solved individually – the required mass flows are found, that provide demanded heat flows. The optimization variables are $\dot{m}_\iota, \dot{m}_p$ (mass flows through secondary loops and mass flow through the primary loop)

A more detailed explanation of the optimization phases follows.

3.2 Water temperatures computation

Knowing the HX heat flow, we are able to find the temperature the water is cooled down from

$$Q = \dot{m}c_{\mathrm{p}}\Delta T = \dot{m}c_{\mathrm{p}}(t_{\mathrm{w1}} - t_{\mathrm{w2}}), \tag{1}$$

where Q [W] is the HX actual heat flow, $\dot{m}$ [kg/h] is the water mass flow through the HX, ΔT [K] is the differential temperature of the supply and return water, c_p [J/kg/K] is the specific heat capacity. Knowing the HX supply water temperature (t_{w1} [°C]), we get the HX return water temperature (t_{w2} [°C]) as

$$t_{\mathrm{w2}} = t_{\mathrm{w1}} - \Delta T, \tag{2}$$

from (1) we express ΔT, substitute in (2) and get

$$t_{\mathrm{w2}} = t_{\mathrm{w1}} - \frac{Q}{\dot{m}c_{\mathrm{p}}}. \tag{3}$$

Let's apply (3) to the k-th HX in a set and let's mark all variables associated with the k-th HX with an upper index k, equation (3) then turns into

$$t_{\mathrm{w2}}^k = t_{\mathrm{w1}}^k - \frac{Q^k}{\dot{m}^k c_{\mathrm{p}}}. \tag{4}$$

The supply water temperature entering the HX with the index k+1 is computed as a weighted average of a temperature of the water bypassing the k-th HX through the primary loop and the temperature of the water returning from the k-th HX

$$t_{\mathrm{w1}}^{k+1} = \frac{\dot{m}^k \cdot t_{\mathrm{w2}}^k + \dot{m}^p \cdot t_{\mathrm{w1}}^k}{\dot{m}^k + \dot{m}^p}. \tag{5}$$

3.3 Finding the water mass flow through the HX

For finding the actual heat emission of a HX, which is run under different temperatures (t_{w1}, t_{w2} and t_i) than it is recorded in a datasheet, the formula specified by the EN 442 norm [19] is used

$$\frac{Q}{Q_{\mathrm{n}}} = \left(\frac{\Delta t}{\Delta t_{\mathrm{n}}}\right)^n, \tag{6}$$

where Q [W] is the HX heat emission under the actual conditions, Q_n [W] is the HX heat emission under the datasheet conditions, n is the HX temperature exponent (radiator, according to DIN 4703, corresponds to the exponent value $n = 1{,}30$), Δt [K] and Δt_n[K] are the logarithmic temperature differences (LMTD [20]) under the actual and datasheet conditions.

$$\Delta \mathrm{t} = LMTD = \frac{t_{\mathrm{w1}} - t_{\mathrm{w2}}}{ln\left(\frac{t_{\mathrm{w1}} - t_{\mathrm{i}}}{t_{\mathrm{w2}} - t_{\mathrm{i}}}\right)}, \tag{7}$$

The aim of the problem is to find the water mass flow $\dot{m}$ through the HX, which is able to deliver the demanded heat flow Q under the given temperatures t_{w1} and t_{i}. In (7) we substitute t_{w2} from (3) and get

$$\mathrm{LMTD} = \frac{t_{\mathrm{w1}} - t_{\mathrm{w1}} - \frac{Q}{\dot{m}c_{\mathrm{p}}}}{ln\left(\frac{t_{\mathrm{w1}} - t_{\mathrm{i}}}{t_{\mathrm{w1}} - \frac{Q}{\dot{m}c_{\mathrm{p}}} - t_{\mathrm{i}}}\right)} = \frac{\frac{Q}{\dot{m}c_{\mathrm{p}}}}{ln\left(\frac{t_{\mathrm{w1}} - t_{\mathrm{i}}}{t_{\mathrm{w1}} - \frac{Q}{\dot{m}c_{\mathrm{p}}} - t_{\mathrm{i}}}\right)}. \tag{8}$$

It is not possible to express $\dot{m}$ from (8) analytically, but (8) can be solved for $\dot{m}$ numerically. Let's mark the actual HX heat emission as $Q'(\dot{m})$ and the demanded heat emission stays Q. Let's define the optimization problem, where $\dot{m}$ is the optimization variable and the cost function

$$f(\dot{m}) = \left(\frac{Q}{Q_{\mathrm{n}}} - \left(\frac{LMTD}{LMTD_{\mathrm{n}}}\right)^n\right)^2. \tag{9}$$

The solver finds such a value of $\dot{m}_{opt}$, for that the cost function $f(\dot{m}_{opt}) \approx 0$, so the demanded HX heat emission is satisfactorily close to the actual heat emission:

$$Q'(\dot{m}_{opt}) \approx Q. \tag{10}$$

Knowing the temperatures $t_{\mathrm{wi}}^k, t_{\mathrm{i}}$ and the mass flow $\dot{m}^k$ belonging to the k-th HX, by (4) it is possible to compute the temperature of water leaving the k-th HX. Having a set of *l* heat exchangers, we can extend the optimization process to the whole set.

3.4 Continuous optimization of a set of HXs

The system being solved consists of one primary loop and *l* secondary loops with HXs (Fig. 6). The aim of the optimization problem is to find such HXs nominal heat emissions Q_{n}^k, the primary loop mass flow $\dot{m}^p$ and all secondary loop mass flows $\dot{m}^k$, so that the cost function (13) is minimized.

In the first step, the optimization variables are the mass flows through the HXs ($\dot{m}^k$) and nominal heat flows of the HXs (Q_{n}^k). The primary loop mass flow is determined from the specified temperature difference of the supply and return water and by a sum of all demanded HX heat flows.

$$\dot{m}^p = \frac{\sum_{k=1}^{l} Q_{\mathrm{n}}^k}{c_p(t_{bo} - t_{bi})}, \tag{11}$$

where t_{bo} is a temperature of the water leaving the heat source [°C], t_{bi} the temperature of water returning to the heat source [°C], $\dot{m}^p$ the mass flows through the primary loop [kg/s]. The cost function consists of three components weighted by coefficients w_1, w_2, w_3 and of an equation term representing a soft constraint

$$f = w_1 f_Q + w_2 f_{\dot{m}} + w_3 f_{Qn} + f_{\mathrm{bnd}}, \tag{12}$$

where

$$f_Q = \sqrt{\sum_{k=1}^{l} \left(\frac{Q^k}{Q_{\mathrm{n}}^k} - \left(\frac{\Delta \mathrm{t}^k}{\Delta \mathrm{t}_{\mathrm{n}}^k}\right)^n\right)^2}, \tag{13}$$

penalizes violation of the actual heat flow Q^k from the demanded heat flow of the k-th HX (by the final optimal solution, it is supposed to be $f_Q \approx 0$),

$$f_{\dot{m}} = \sum_{k=1}^{l} \dot{m}^k + \dot{m}^p \tag{14}$$

penalizes the mass flows through the HXs,

$$f_{Q\mathrm{n}} = \sum_{k=1}^{l} Q_{\mathrm{n}}^k, \tag{15}$$

penalizes sizes of the used HXs (the investment cost). The values of the optimization variables are limited by

$$\begin{gathered} 0 < Q_{\mathrm{n}}^k < Q_{max}, \\ \dot{m}_{\mathrm{min}}^k < \dot{m}^k < \dot{m}_{\mathrm{max}}^k. \end{gathered} \tag{16}$$

The upper bound of the nominal heat flow is set as two times the maximal demanded heat flow.

$$Q_{max} = \max(Q_{\mathrm{d}}^k, k = 1\,..\,l) \cdot 2 \tag{17}$$

The lower bound of the minimal water mass flow through the HX is a so-called soft constraint, which is projected into the cost function.

$$\begin{gathered} f_{\mathrm{bnd}} = \left[\mathrm{sgn}\left(\dot{m}_{\mathrm{min}}^k - \dot{m}^k\right) + 1\right] \\ \cdot \left(\dot{m}_{\mathrm{min}}^k - \dot{m}^k\right)^2, \end{gathered} \tag{18}$$

The $\dot{m}_{\mathrm{min}}^k$ is determined by the domain of (8), where the denominator of the expression under the logarithm must be positive

$$t_{\mathrm{w1}}{}^k - \frac{Q^k}{\dot{m}^k c_{\mathrm{p}}} - t_{\mathrm{i}} > 0. \tag{19}$$

From the in equality (19) we get the boundary value as

$$\dot{m}_{\mathrm{min}}^k = \frac{Q^k}{c_{\mathrm{p}}(t_{\mathrm{w1}}{}^k - t_{\mathrm{i}})}. \tag{20}$$

This boundary has no physical sense, but it was introduced because of the mathematical feasibility of the problem.

Results of the continuous optimization are the nominal heat flows and mass flows through the HXs.

3.5 Continuous optimization of a set of HXs

After the optimal values of the nominal HXs and heat flows are found, the second step follows, with an aim to meet a real-world solution, which is as close as possible to the optimal one. Let's have an optimal solution of a set of l HXs from a previous step. The program creates 2^l sets by using the catalogue HXs with the closest higher and lower nominal heat flows. Every such a set is solved as an optimization problem with mass flows $\dot{m}^k$ and $\dot{m}^p$ as optimization variables and with a cost function

$$f = w_1 f_Q + f_{\mathrm{bnd}} \tag{7}$$

where f_Q is same as in the continuous optimization given by an equation (13) and the constraint f_{bnd} is described in (18). Compared to the previous continuous optimization, the nominal HXs heat flows are no more the optimization variables.

4 Example

To demonstrate the one-pipe design tool a one-pipe hydronic heating system consisting of six FCUs was used. Table 1 contains the problem definition. The 4-pipe FCUs from the FW series [22] was used as the FCU catalogue list which defines a set of FCUs, the units in the real-world solution are going to be chosen from. The catalogue values of FCU heat flows were given under the temperatures 90/70/20 °C.

The results of both the optimal and the selected real-world solution are listed in Table 2. The first column contains the FCU numbers, in the second column the desired actual heat flows are shown. The third to sixth column contain the nominal heat flow (Q_{n}), the mass flow and the supply/return water temperatures of the optimal solution. The seventh to tenth column contain the same quantities of the real-world solution. The last column involves the chosen FCUs catalogue names. In the real-world solution all but the second FCU are of higher nominal heat flow than FCUs in the optimal solution (marked as HLHHHH; H for higher, L for lower nominal heat flow). The last row contains the temperature of water returning to the boiler. In both the optimal and the real-world solution it corresponds to the return water temperature specified in the problem setup.

Table 1. Example problem setup.

Supply water temperature	80 °C
Return water temperature	60 °C
Air temperature	20 °C
Specific heat capacity	4186 J/kg/K
HX temperature exponent	1,30
1st HX heat flow	4000 W
2nd HX heat flow	4000 W
3rd HX heat flow	5500 W
4th HX heat flow	5500 W
5th HX heat flow	3000 W
6th HX heat flow	3000 W

Table 2. Example problem results.

		Optimal solution				*Selected real-world solution (HLHHHH)*				
HX num.	$\boldsymbol{Q}$ **[W]**	$\boldsymbol{Q_{\mathrm{n}}}$ **[W]**	$\boldsymbol{\dot{m}}$ **[kg/h]**	$\boldsymbol{t_{\mathrm{w1}}}$ **[°C]**	$\boldsymbol{t_{\mathrm{w2}}}$ **[°C]**	$\boldsymbol{Q_{\mathrm{n}}}$ **[W]**	$\boldsymbol{\dot{m}}$ **[kg/h]**	$\boldsymbol{t_{\mathrm{w1}}}$ **[°C]**	$\boldsymbol{t_{\mathrm{w2}}}$ **[°C]**	**FCU name**
1.	4000	2976	499,06	80,00	73,11	3080	355,65	80,00	70,33	FWM03DF
2.	4000	3194	531,13	76,80	70,32	3080	977,31	76,80	73,28	FWM03DF
3.	5500	4732	779,99	73,60	67,53	5050	442,34	73,60	62,90	FWM04DF
4.	5500	5284	859,71	69,20	63,70	5300	832,59	69,20	63,52	FWM06DF
5.	3000	3253	521,70	64,80	59,85	4490	121,03	64,80	43,48	FWD04AF
6.	3000	3492	555,38	62,40	57,75	4490	149,19	62,40	45,10	FWD04AF
				$\boldsymbol{t_{bi}}$ **[°C]**	60			$\boldsymbol{t_{bi}}$ **[°C]**	60	

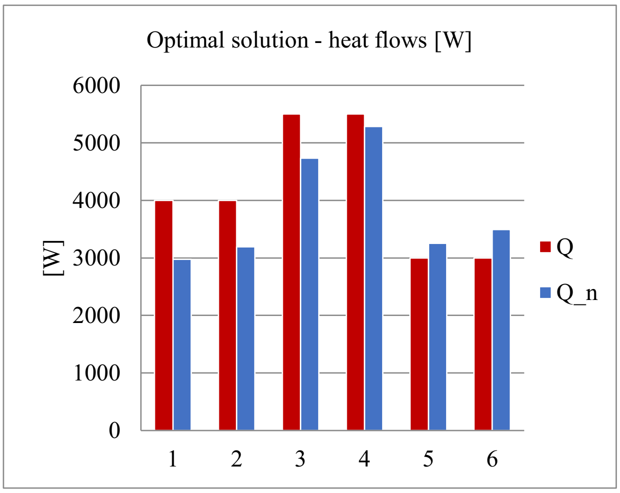

Fig. 4. Example optimal solution: heat flows.

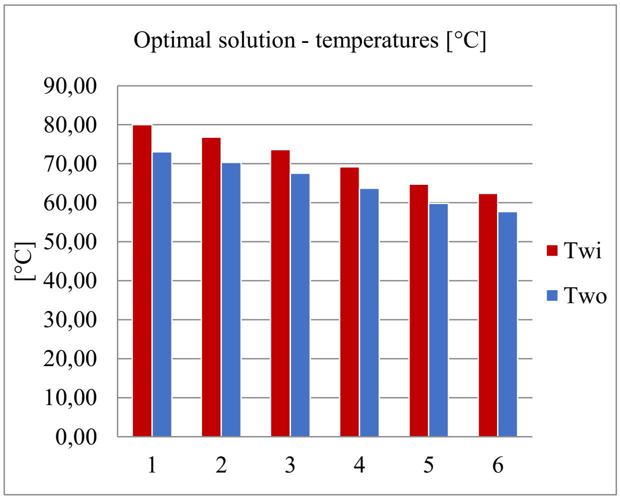

Fig. 5. Example optimal solution: temperatures.

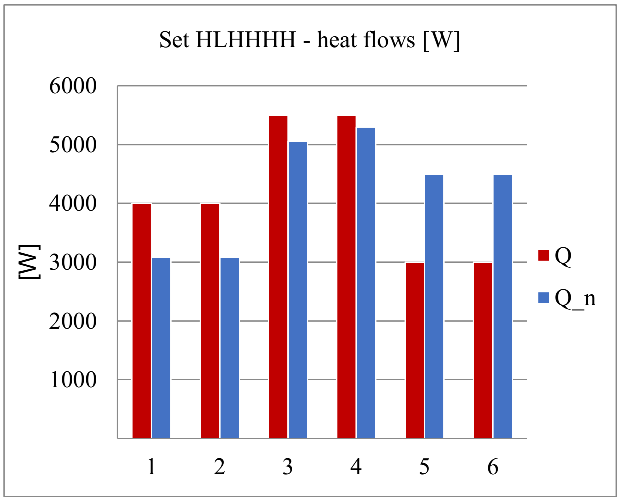

Fig. 6. Real-world solution: heat flows.

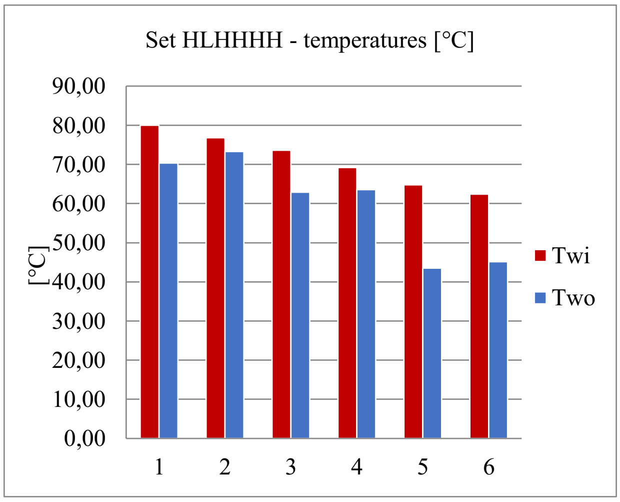

Fig. 7. Real-world solution: temperatures.

5 Conclusion

Even though it may look like there is not much space for innovation in the field of the hydronic heating systems anymore, there are still challenges of the technical development. The active (pumping) heating systems decrease the pumping energy demands and are well suited for the multizone control, which makes them consistent with the nowadays trend of the higher efficiency of the building services. The price of the technology is still an issue, but both the investment, operational and maintenance costs are expected to decrease in the near future, which will help the active systems to become an adequate alternative to the current passive heating systems.

The broader expansion of one-pipe heating systems is inhibited among other also by the computationally laborious design. Nevertheless, the use of computation software could significantly reduce the necessary effort. This paper describes a fundamental mathematical description of a one-pipe hydronic heating system design tool, which is available at [18].

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