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Optimum performance of radiator space heating systems connected to achive lowets possible district heating return temperature

P. Ljunggren, J. Wollerstrand,
Lund University/Sweden

Optimum Performance of Radiator Space Heating Systems Connected to achieve lowest possible district heating return temperature

M Sc Patrick Ljunggren, Ph D Janusz Wollerstrand
Department of Energy Sciences, Division for Efficient Energy Systems,
Lund University, Faculty of Engineering
Box 118, 221 00 Lund, Sweden
+46 46 222 92 73, Patrick.Ljunggren@vok.lth.se, www.vok.lth.se

Abstract

The cooling of primary water in district heating substations is largely influenced by the existing oversizing of space heating systems and its components. Performed simulations show that it is possible, with measurements of temperatures and energy consumption, to estimate the actual oversizing of the space heating system in a specific building and, from that information, to control the supply temperature in the system for highest possible cooling of primary water at all loads. Full optimization is possible if the indoor air temperature can be measured or estimated with sufficient accuracy.

Control of an oversized space heating system using the low circulation flow method, as well as low temperature adjustment, results in approximately 12°C improved cooling of the primary water. Further optimization of a low temperature adjusted system can produce another 3-4°C increased cooling. Even a perfectly sized system can be flow optimized with a gain corresponding to almost 2°C.

If a high and a low temperature space heating system is compared, and a flow price is used by the district heating company, there is a significant difference, up to 1 900 €/year for a building with 100 apartments, which reflects a magnitude of possible saving by improving a high temperature system.

Nomenclature

Variables

A	: Area, m^2
c_p	: Specific heat capacity, J/kgK
$LMTD$	: Logarithmic mean temperature difference, °C
$\dot{m}$	: Mass flow, kg/s
n	: Radiator exponent, -
os	: Factor for degree of oversizing, -
$\dot{Q}$	: Heat flow, kW
T	: Temperature, °C
U	: Overall heat transfer coefficient, W/m^{2}K

Indices

0	: Design condition
i	: Indoor
f	: Forward, supply
HX	: Heat exchanger
p	: Primary
r	: Return
rad	: Radiator (system)
s	: Secondary

1 Background

A general goal when designing district heating, DH, substations is to achieve a cooling of DH water that is as big as possible, which, for a given network supply temperature reaching the substation, means a return temperature that is as low as possible. The advantages of an improved cooling of primary water in DH systems are well-known. For example can the heat losses and the pumping work in the system be reduced and the capacity in the network can be increased. Many kinds of heat production, such as combined heat and power, heat pumps, waste heat and flue gas condensation, are also favoured.

The choice of connection scheme in DH substations, i.e. whether cascading of heat exchangers for domestic hot water provision and space heating is applied or if they are connected in parallel, is nowadays considered to be of less significance for the cooling and the DH cost is only marginally affected. Of at least the same interest for the cooling is the level of the temperatures in the secondary systems connected to the substation.

Space heating radiator systems can be connected to DH networks directly or indirectly. In the latter case there will be one or more heat exchanger providing hydraulic separation of the radiator circuit from the DH network. Swedish DH companies have almost universally adhered to the indirect method, and therefore this will be the case also in this study.

1.1 Aims of the study

This study will focus on the performance of heat exchangers for radiator water heating, with an aim of minimizing the return temperature of the DH water. In so doing, we shall consider both deliberate tailoring of heat exchangers for optimum performance and consequences of unintentional oversizing, concerning both heat exchangers as well as radiator system components, which often takes place in practice, given realities of current design practices.

Current committee work within the Swedish District Heating Association looks for cost reduction to be achieved by standardisation of substation equipment, grouping substations into a limited number of classes as related to building sizes. In the authors' opinion it is essential that such standardisation should pay due attention to gains possible to attain by utilising facilities created by modern control methods. More specifically, when attention is per se rightly focussed on cost reduction, it may be tempting to cut down on sizes of heat exchangers. As will become clear, such a reduction would rule out major opportunities for increasing primary water cooling, a development that would run contrary to results of many other efforts made in DH practice.

2 Optimal control of radiator space heating circuits

In theory, the choice of supply temperature in the space heating circuit is rather simple. It is adjusted so that the heat output from the radiators matches the existing heat consumption. The idea is to set desired supply and return temperatures at maximum load, e.g. 80/60 or 60/45°C. Then, for a radiator with known heat transfer characteristics and for a given indoor temperature, the emitted heat per unit radiator area can be calculated. With known design heat power, the necessary total radiator area and radiator circulation flow can be calculated. These parameters, together with the design outdoor temperature, T_{DOT}, are later used for part load calculations. [3]

In order to adjust the heat output to the reduced heat demand, the temperature difference between the radiator surface and the indoor air must be adjusted, which means that the radiators' average surface temperature has to be reduced. In a conventional system, this is achieved simply by lowering the supply temperature. With known radiator heat transfer characteristics and a constant circulation flow in consideration, the requisite supply, and resulting return, temperatures can be calculated.

A drawback with traditional outdoor temperature compensation is that the circulation flow is constant. The cooling of primary water will be optimal at maximum load (T_{DOT}). However, this does not guarantee that the cooling will be optimal at part load. It can be shown that if highest possible cooling is aimed at, the flow should be decreased at part load while the supply temperature should be deceased less than what is done when a constant flow is used. One can then speak of a moving transition, with decreasing load, from a high flow balanced system to a low flow balanced system.

In [3], Frederiksen and Wollerstrand introduced the idea of an optimal control of a hydronic radiator system, connected to a DH network via a heat exchanger. The main idea is that for a given heat load and a given DH supply temperature, as well as a given heat exchanger, there will always be an optimal supply temperature of the space heating circuit, as related to a criterion of a minimum primary return temperature.

In a later publication, [4], Frederiksen quantified the potential temperature gain in a static heat exchange calculation, as shown in Figure 1. As a theoretical premise, the overall heat transfer coefficient, *U*, was assumed to be constant in all variations performed. At the design stage this can be regarded as possible to a great extent.

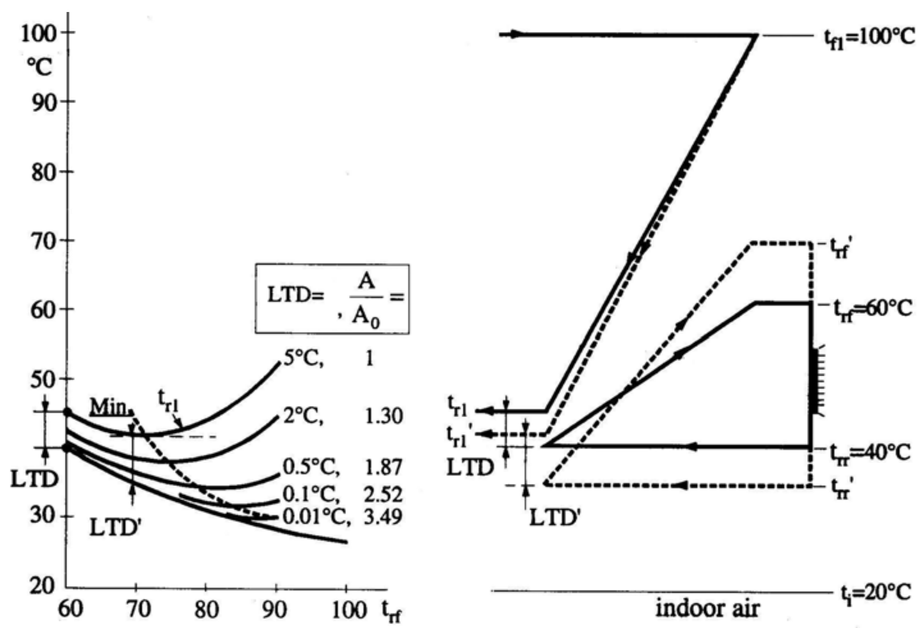

Figure 1 Minimum primary return temperature, t_{r1}, at altered radiator supply temperature, t_{rf}, for increasing heat exchanger area. From [4].

From the figure it can be seen that by combining flow rate optimization with increased heat exchanger sizing, significant gains in primary return temperature can be attained.

Since the idea was originally introduced, major advances have been made in pump technology, such that the pressure differential across a hydronic system can be continuously adjusted. This makes it possible to adapt the circulated mass flow rate at part load to the chosen supply temperature instead of relying on action of thermostatic radiator valves. Together with development of microprocessor technology, these advances have facilitated the introduction of the idea in practice. A Swedish company has rather recently introduced an apparatus that operates along these lines, and it has been tested in a field experimental project, see [1] and [6].

3 Consequences of various ways of oversizing heating systems

Space heating systems, especially the total radiator surface area servicing a building, are almost universally being more or less oversized. The amount of oversizing will typically be at least 10 per cent and very often 100 per cent or even more, [5], [9] and [11]. Additional oversizing can arise as a result from actions taken by the building owner at a later stage in order to save energy, for example replacement of windows, additional insulation or installation of heat recycling in the ventilation system.

What will the consequence of an oversized space heating system be? If the system is run with supply temperatures equal to those that would have been fit if the system had not been oversized, the result will be that the indoor air will be overheated. A system with well functioning radiator thermostats can, at least in theory, throttle flows in order to achieve the correct indoor temperature. A possibility is that the tenants by themselves will adjust the radiator valves in order to lower the indoor temperature, or that they may inform the caretaker, who can adjust the supply temperature, or they may simply open a window now and then to vent out too hot air. Even if a combination of these possibilities will occur, it is most likely that the indoor temperature will be higher than intended.

3.1 Estimating the degree of oversizing

A method to determine the oversizing of the radiators is to measure the flow and the temperatures in the space heating system and then calculate the heat transfer capacity of the radiator, the so-called *UA* value. The following heat transfer equation will apply to the space heating system:

$$\dot{Q}_{rad} = \dot{m}_{rad} c_p (T_{s,f} - T_{s,r}) = UA \left(\frac{T_{s,f} - T_{s,r}}{\ln\left(\frac{T_{s,f} - T_i}{T_{s,r} - T_i} \right)} \right)^n \Rightarrow UA \tag{1}$$

The *UA* value can be calculated at all working conditions, but preferably day or week average values of the measured quantities are used in order to minimize the influence of the system's various time constants. The *UA* value can be considered to be constant and the exponent n, which is termed the radiator constant, is usually set to 1.3 [12].

The right-hand side of the equation above can be applied both for a system that exactly fulfils the design conditions, and to the real system, i.e. the oversized system. The following relation will apply, provided that the indoor temperature is the same:

$$\dot{Q}_{rad,0} = (UA)_0 \cdot LMTD_0^n = UA \cdot LMTD^n = (UA)_0 \cdot (1 + os) \cdot LMTD^n \tag{2}$$

where the parameter *os* indicates how large the existing oversizing is, and sub-index 0 is used for variables at design conditions. From (2) the oversizing can be written as:

$$os = \left(\frac{LMTD_0}{LMTD}\right)^n - 1 \tag{3}$$

If the space heating system is oversized and sufficient compensation in temperature program or circulation flow has not been performed, the indoor temperature will be too high and the equality in (2) will not be fulfilled. We must therefore distinguish between theoretical and actual heat output and we then arrive at the following two equations:

$$\begin{cases} \dot{Q}_{rad,0} = (UA)_0 \cdot LMTD_0^n \\ \dot{Q}_{rad} = UA \cdot LMTD^n = (UA)_0 \cdot (1+os) \cdot LMTD^n \end{cases} \tag{4}$$

$$os = \left(\frac{LMTD_0}{LMTD}\right)^n \cdot \frac{\dot{Q}_{rad}}{\dot{Q}_{rad,0}} - 1 \tag{5}$$

Figure 2 shows the result of a computer simulation of instantaneous supply and return temperatures in the space heating system, primary return temperature, and the indoor air temperature, assuming that the space heating system is 100 per cent oversized. The simulation is based on a dynamic model of the building, i.e. the heat storage capacity of the building has been taken into account. Due to the dynamics of the system, all temperatures exhibit a scattering.

As will often happen in practice, the supply temperature curve has only been lowered to a level corresponding to 75 per cent radiator oversizing, causing the indoor temperature to be higher than the intended 21°C, as can be seen from the lowest line in the diagram. The calculations have been made under the assumption that there is no feedback from the indoor air temperature.

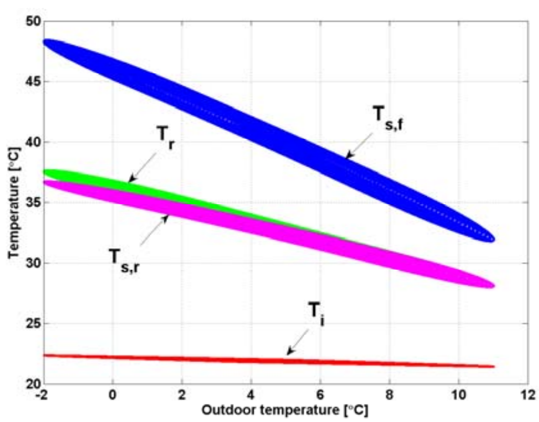

Figure 2 Mode with excessive indoor air heating, simulated instantaneous temperatures.

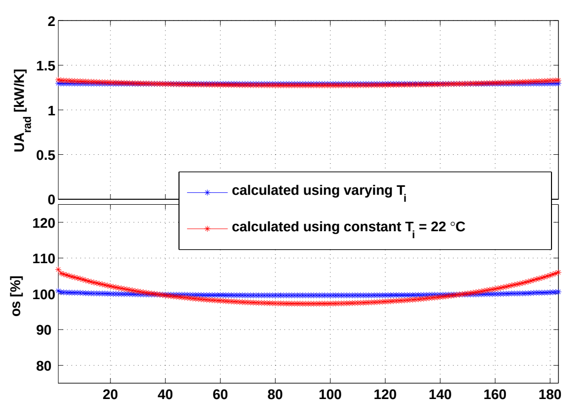

Figure 3 UA value of radiators (top) and degree of oversizing (bottom), under differing assumptions regarding indoor temperature. 'T$_i$ varying' refers to variation according to simulation.

If we do not have the possibility to measure the indoor temperature we must try to estimate it. On top of Figure 3 the radiators' *UA* value is shown, calculated according to equation (1) using both measured and estimated indoor temperature, T_i, respectively. The possibility to calculate T_i from the building's so-called energy signature was also investigated. However, then the internally generated heat (from humans, electric equipment etc.) has to be estimated and it turned out that the result will be same as if T_i is estimated directly. At the bottom of the diagram the degree of oversizing is shown. It has been calculated according to equation (5) for the different methods employed to determine the *UA* value. The diagram includes a simulation made for a control system which relies on a feedback loop from a measured indoor air temperature. As can be seen, the result for such a system is much better. With an estimated indoor temperature the value of the oversizing is not constant which depends on the fact that a supply temperature set-point curve that has not been lowered, or has not been lowered sufficiently, results in an even higher indoor temperature at lower outdoor temperatures. However, it turns out that if the estimated indoor temperature corresponds with the measured temperature on average, then the calculated oversizing, on average, will correspond with the one obtained with measured T_i.

Table 1 gives calculated oversizings on average under different assumptions, as well as resulting indoor temperatures. An estimation of T_i that lies within the real value ± 0,5ºC results in an estimation of the existing oversizing with an accuracy of ± 10 per cent.

Calculated degree of oversizing using measured T_i (21.9ºC on average) is 100 per cent					
Estimated T_i [ºC]	**21**	**21.5**	**22**	**22.5**	**23**
Calculated oversizing with estimated T_i [%]	**85**	**93**	**102**	**111**	**121**

Table 1 Influence of assumed T_i on calculation of the degree of oversizing, 80/60ºC system with 100 per cent oversizing.

3.2 Adjustment of the radiator set-point curve for a given degree of oversizing

Figure 4 below shows calculated temperatures in the space heating heat exchanger as a function of the outdoor temperature. The radiators are designed to work with 80°C supply and 60°C return temperature at T_{DOT} and the heat exchanger to give 63°C return temperature at 100°C primary supply temperature. The solid lines in the diagram indicate parameter values at a constant circulation flow rate. As previously stated, the cooling of primary water is optimal at T_{DOT} when the radiator flow rate is assumed to be constant. From the diagram it can be seen that there is some lowering of the primary return temperature in case of an optimized radiator flow rate, but the gain is not very large. The explanation is that, although optimizing the flow rate results in a generally lowered return temperature in the radiator circuit, the gain is to a great extent offset by a less efficient function of the heat exchanger.

The situation is changed if we consider an oversized system. In Figure 5 the system is now 100 per cent oversized, both the radiators and the radiator heat exchanger, whose size is being doubled by increasing its number of plates. This is not an optimal way of oversizing since the increase in heat transfer capacity by increased heat transfer area is offset by a lower heat transfer coefficient caused by decreased flow velocities. However, this is a typical condition if we assume the unintentional oversizing. The radiators can now operate as if they were designed for approximately 60/40°C. It can be seen that in this case flow rate optimization (dashed lines) produces a bigger gain in terms of a lowered primary return temperature, also at design load.

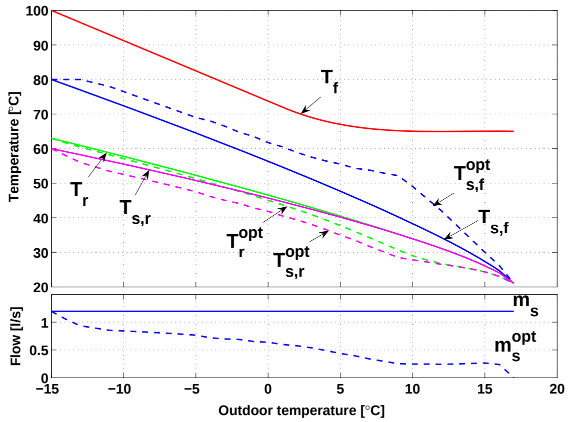

Figure 4 Standard temperature (constant flow, solid lines) and optimized program (variable flow, dashed lines), no oversizing of radiator system.

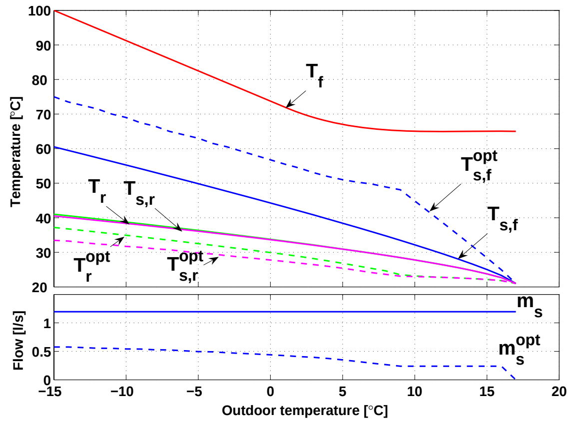

Figure 5 100% oversized radiator system, lowered supply (unchanged flow, solid lines) and optimized program (variable flow, dashed lines).

Another way of compensating for oversized radiator surfaces is to substantially reduce the flow rate to a smaller, constant, size, combined with use of a set-point curve for the supply temperature that would have been appropriate in case of no oversizing of the radiators. This principle is in Sweden generally referred to as ‘low flow radiator system balancing’ or the ‘Kiruna’-method. It has now been a fairly often adopted method in Sweden since the 1970’s [2]. Figure 6 shows the temperature program for a correctly designed system (solid lines) and the differences when a low flow balancing is applied to an oversized system (dashed lines). It turns out that the resulting primary return temperature is of about the same level as with the low-temperature adjusted program (60/40ºC) displayed in the previous diagram, and slightly higher compared to the flow-optimized case within the whole temperature interval.

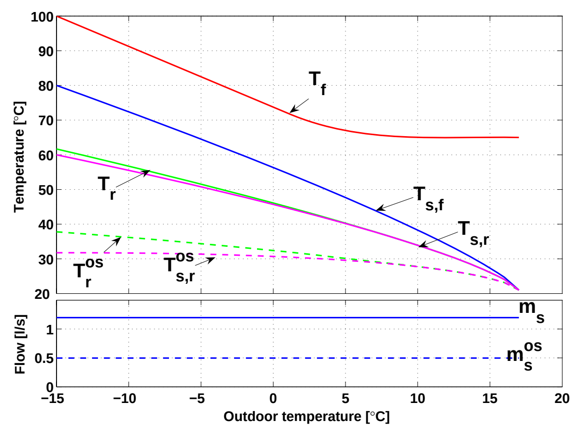

Figure 6 100 per cent oversized radiator system, low flow balancing (design supply temperature and reduced flow).

Thus, the rather simple method of low flow balancing, referring to that it requires no sophisticated control equipment, appears rather favourably. But there is a drawback: If the DH company sets the primary supply temperature to a lowered value, the rather high secondary supply temperature will be an obstacle that may result in excessive primary return temperature, if no correction of the secondary supply temperature set-point curve is employed. Robustness against such changes in primary supply temperature can be achieved if an advanced control algorithm that continuously adjusts the radiator system setting, so as to produce minimal primary return temperature, is used.

The possibility to optimize the supply temperature ‘manually’ (by trial and error), has also been investigated with computer simulations. The method turned out to be working in theory but was obstructed by (the unavoidable) hysteresis in the thermostatic radiator valves.

3.3 Possible gains from radiator system optimization

To be able to make a true quantification of the gain we must take the seasonal duration of the outside air temperature into account. This has been done by simulating the operation for an entire season, under the same differing stipulated conditions that have been studied. The resulting gain in weighted average primary return temperature for the various ways of running the space heating system are shown in Table 2. As can be seen, for given assumptions, both low-flow balancing and low-temperature adjustment of an oversized space heating system result in approximately 12ºC improved cooling of primary water. Flow optimization of a low temperature adjusted system can produce another 3-4ºC improved cooling. Further, even a perfectly designed system can be optimized, with almost 2ºC as a result. If the space heating heat exchanger is 100 per cent oversized, the return temperature will decrease further, with up to 1.5ºC. The column “100% longer” shows what happens if the oversizing of the heat exchanger is made by making it longer, instead of just adding more plates. This will benefit the case with an optimized radiator flow, as can be seen in the table, and will be discussed more in the next section.

	Oversizing of heat exchanger		
	0%	**100% wider**	**100% longer**
0% oversized radiators, constant flow rate	**0**	**-0.5**	**-0.8**
0% oversized radiators, optimized flow rate	**-1.8**	**-3.3**	**-5.8**
100% oversized radiators, low flow balancing, const. flow	**-12.3**	**-13.5**	**-14.6**
100% oversized radiators, lowered set point, const. flow	**-12.1**	**-12.3**	**-12.3**
100% oversized radiators, optimized flow rate	**-14.9**	**-16.2**	**-18.1**

Table 2 Difference (in °C) in annual weighted return temperature from space heating heat exchanger versus reference case with a return temperature of 44.9°C

In a previous study, [7], it was found that the majority of the sold DH in Sweden is accounted for with regard to primary water consumption, in addition to the amount of delivered energy. In this way, a proper cooling is rewarded by the DH company. If a high and a low temperature space heating system is compared, assuming an average flow price, there is a significant difference, up to 1 900 €/year for a building with 100 apartments, which reflects the magnitude of a possible saving by improving a high temperature system.

3.4 Gains attainable by differing ways of up-sizing radiator heat exchanger

For many years now, The Swedish District Heating Association has recommended that heat exchangers are designed for a difference between primary and secondary return temperatures of 3C° at T_{DOT} and a temperature rise on the secondary side in the order of 15ºC, for instance from 45 to 60ºC [10]. Formerly, this difference was bigger, viz. 5ºC. In fact, the lower value implies a heat exchanger that is so large that any further enlargement may seem unreasonable due to diminishing marginal returns versus incremental investment cost. At the most common working conditions, i.e. at part heat load, the temperature differential becomes even smaller than at design condition.

However, this consideration is only valid as long as no change in radiator flow rate is employed. With lower flow rates, the marginal gain indeed becomes larger and therefore it should be considered. In a study on low-flow balancing, by Petersson [8], the gain from adopting a heat exchanger with a larger thermal length, i.e. a bigger heat transfer capacity at a given flow rate, was evaluated. Because of the very low flow rate assumed in that study, the heat exchanger in fact performed far from the magnitude of flow rate it was designed for, and the gain in cooling was found to be rather small. With the kind of optimization suggested in the present work, the flow, especially at bigger loads, is substantially larger than with low-flow balancing and the supply temperature to the radiators is lower.

The results presented in Figure 1 were calculated under the assumption that, when varying flow rates on both the primary and the secondary side of the heat exchanger, the overall heat transfer coefficient, *U*, was kept constant. This can be translated into such modifications of flow channels of the heat exchanger that both primary and secondary flow velocities are kept unchanged, neglecting the rather small influence of temperature on *U*.

Figure 7 shows how the primary return temperature and the radiator supply temperature at design load conditions, and with optimized radiator flow rate, vary with the heat exchanger size. Various assumptions are made about how the added heat exchanger surface has been arranged. It can be seen that increasing the heat exchanger area such that *U* is constant (as in Figure 1) is much better than by making the flow channels bigger, i.e. by widening the apparatus, which in a plate heat exchanger is commonly done by adding the number of plates. Constant *U* resembles the case of increasing the lengths of the plates instead, although, as the diagram shows, these two cases are not exactly identical, due to the fact that at optimal conditions the primary and secondary flow rates will not change in the same proportion when increasing the heat transfer surface area.

In a fourth case, an optimization has been performed to allow for an unsymmetrical heat exchanger, making an adjustment that takes into account the somewhat differing primary and secondary side flow rates. Since the primary flow rate is smaller, the channels have been assumed to be optimized in such a way that they are more narrow on the primary side, so as to cause the heat transfer coefficients on the two sides to become more equal. In practice, for a given heat exchanger operating under differing load conditions with varying primary and secondary flow rates, the respective heat transfer coefficients will of course vary, such that in most cases the primary and secondary heat transfer coefficients will differ to some extent.

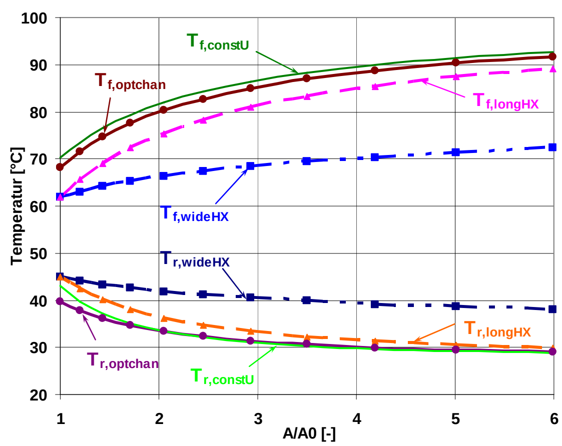

Figure 7 Primary return and secondary supply temperature at design load, assuming optimal radiator flow, as a function of heat exchanger area, under four different assumptions regarding how such change of size is designed for.

Finally, as mentioned in connection to Table 2, where it was illustrated by making the heat exchanger longer, the method of flow-optimization benefits from a ‘smart’ way of up-sizing the heat exchanger.

4 Conclusions

It is possible, with measurements of temperatures and energy consumption, to estimate the actual oversizing of the space heating system in a specific building and, from that information, to control the supply temperature in the system for highest possible cooling of DH water at all loads. Full optimization is possible if the indoor air temperature can be measured or estimated with sufficient accuracy.

The gain depends largely on the design of the flow channels of the space heating heat exchanger. The best result occurs if the channels are asymmetric so that heat transfer coefficient on both primary and secondary side is of the same magnitude at design conditions.

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