# SOLAR WATER PREHEATING FOR OPEN DISTRICT HEATING NETS – OPTIMIZATION OF THE AIR-TO-WATER HEAT EXCHANGER DESIGN

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## 1. Introduction

District heating nets for heat supply are very common in cities of the CIS. In contrast to the EU countries some district heating nets in the CIS are constructed as an open-circuit system regarding the hot water supply (Budig et al. 2008)[1], e.g. in Bishkek, the Capital of the Kyrgyz Republic. In such systems hot water itself is delivered to the consumers via the district heating net, whereas in closed systems only heat is supplied. A simplified scheme of an open-circuit system of the combined heat and power plant (CHP) in Bishkek is shown in Fig. 1. This district heating net supplies 350,000 inhabitants in the city center with the base load of around 2,600 m³ of hot water per hour and the peak load up to 3,000 m³/h. Beside this CHP in Bishkek there are over 60 boiler houses with much smaller open district heating nets for remote city areas.

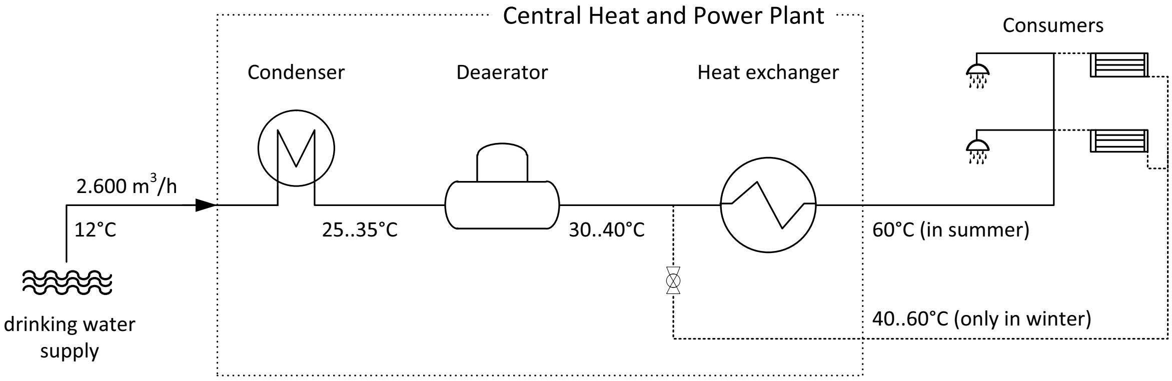
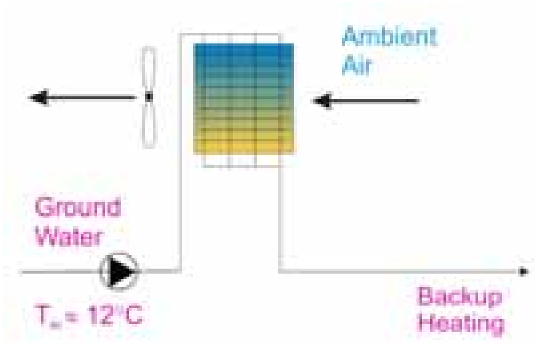



**Fig. 1: Simplified hydraulic scheme of an open-circuit district heating net of the CHP Bishkek (Kyrgyzstan).**

The low water inlet temperature of 12°C, high base load and continental climate (= high solar irradiation and hot summer) in Bishkek are very favorable for preheating of the water using solar energy. For this purpose uncovered solar collectors can be applied very effectively reaching very high annual solar gains of over 1000 $kWh_{th}/m^2a$ and low solar heat costs of less than 1 Euro-Cent/$kWh_{th}$ (Vajen et al. 2008). Uncovered collectors can, however, cover only up to 20% of the heat demand as the outlet temperature is limited. Covered collectors, e.g. flat plate collectors, can be used to achieve higher temperatures and, thus, higher fuel savings. However, they are economically less effective than uncovered collectors because of the higher investment costs, so that a combination of both collector types is economically more feasible if higher temperatures should be achieved.

The extraordinary low water inlet temperatures of district heating nets in Bishkek open more technical options. As the ambient temperature $T_{amb}$ in summer (up to 40°C during the day and 20…25°C during the night) is predominantly much higher than the water inlet temperature $T_{in}$, it is possible to preheat water with the ambient air using fin-and-tube heat exchangers (Frank et al. 2006a), Fig. 2. By this, heat gains can also be achieved during the night. Fin-and-tube heat exchangers (Fig. 3) are widely used for heat transfer between a liquid and a gas in industry and in the residential air conditioning. They have low investment costs and a very compact construction. For water preheating under the climate conditions of Bishkek the air-to-water heat exchangers have similar economical effectiveness as uncovered collectors (Frank et al. 2006b).

[1] About 20% of the CHP plants in the CIS identified by the Budig et al. 2008 have open or partly open district heating nets. The investigations do not consider the size of the nets and smaller district heating nets combined with boiler houses.



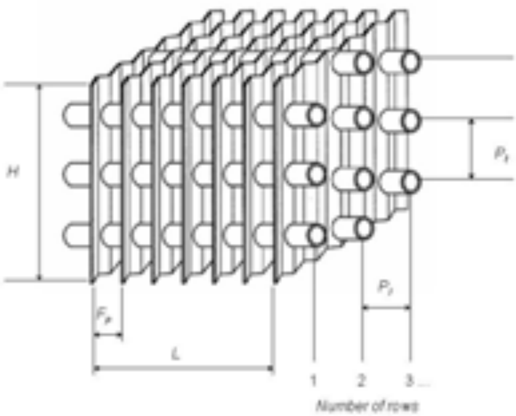




**Fig. 2. Simplified hydraulic scheme of water preheating with the ambient air using an air-to-water heat exchanger**

**Fig. 3. Schematic diagram of a fin-and-tube heat exchanger (taken from Wang et al. (2002) with own changA es)**

A specific problem of implementing air-to-water heat exchangers are the non-standard design requirements for this application. Due to the “back-up” heating by boilers, there are no requirements for heat transfer rate or increase of water temperature. Air flow rate and pressure drops are not restricted too. The air inlet temperature and humidity (ambient air) are changing during day and season. The only known parameters are water inlet temperature and water flow rate, so that even basic thermal design (e.g. calculation of the required UA value) is not possible without additional information or assumptions. Even after setting missing operating parameters and getting the required UA value of the heat exchanger, there is still a lack of design experience regarding the heat exchanger geometry, which strongly affects the heat transfer and pressure drops. Not optimal geometry and operation parameters may lead to higher investment and operating costs of the whole system.

The aim of this study is to elaborate a design procedure of the air-to-water heat exchangers for water preheating in district heating nets with an integrated economical optimization.

## 2. Heat exchanger design

The heat exchanger design (sizing problem) is a complex problem, since the heat exchangers performance and costs depend on many variables (geometrical and operating parameters). As the analytical identification of the optimal design is in the most cases not possible, the sizing problem is usually reduced to the rating problem by specifying the geometry and operating parameters, and then calculating the heat exchanger performance for evaluation. There is, however, a large number of configurations[2] possible if (nearly) all geometry and operating parameters shall be varied. An optimization algorithm can be integrated into the design procedure to find the optimal configuration without calculating all possible configurations. Due to complex dependencies of the heat exchanger performance on its dimensions and operating conditions, a genetic optimization algorithm (Eshelman, L. J. 1991) is applied here, which is more reliable to find the global optimum in comparison to classic (e.g. gradient) algorithms.

The selected objective function for the heat exchanger design optimization is the lowest heat generation costs HGC i.e. the lowest price for the water preheating:

$$HGC = \frac{f_a \cdot C_{inv} + c_{main} + c_{op}}{Q} \quad \frac{€}{kWh_{th}} \qquad (1)$$

with the capital recovery factor $f_a = \frac{i \cdot (1+i)^n}{(1+i)^n - 1}$ (2)

Investment costs $C_{inv}$ include heat exchanger, fan, pump and piping costs:

$$C_{inv} = C_{hx} + C_{fan} + C_{pump} + C_{pipe} \qquad (3)$$

[2] For example, $10^6$ configurations are possible having only 6 variables and 10 values for each variable.

Operation costs $c_{op}$ are calculated by multiplying the annual parasitic energy demand (electricity consumption for boosting air and water flows) and electricity price. The parasitic energy demand and heat gains (transferred from ambient air to water) are determined by annual simulations, since ambient air temperature and humidity are time dependent variables. Transport costs to Kyrgyzstan, possible Kyrgyz import duties and installation costs are not considered due to the wide range of prices and the lack of reliable data.

The optimization procedure is shown in Fig. 4. Heat exchanger configurations are specified and evaluated by a genetic optimization algorithm, implemented in open-source optimization software GenOpt (Wetter 2004). The objective function for each configuration is evaluated by annual simulations in the simulation environment TRNSYS[3] (Klein et al. 2000). Both programs interact with text files.

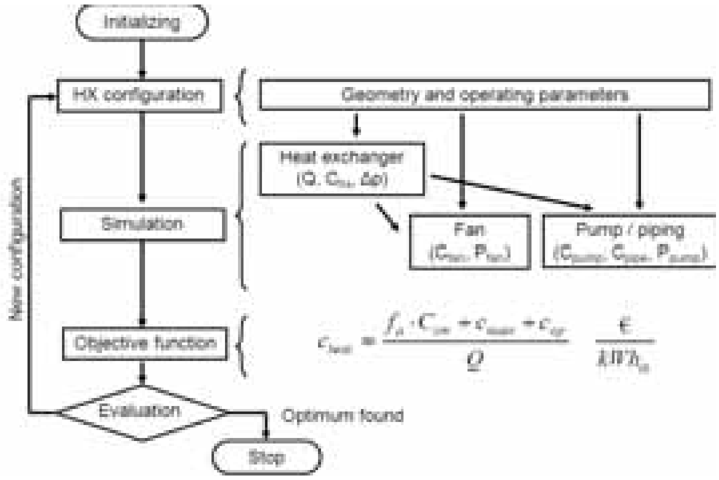



**Fig. 4. Scheme of the optimization procedure.**

The optimization procedure requires a heat exchanger performance model, parasitic energy and cost functions of the system components to calculate the objective function for each configuration. These factors are described in the following paragraphs.

## 3. Heat exchanger model

The heat exchanger model uses empirical heat transfer and flow friction correlations and is described in detail in (Orozaliev et al. 2008). Although models based on empirical correlations usually have higher inaccuracies than those using finite element methods, the latter are not appropriate for an optimization of the whole heat exchanger configuration, because of a too large computing effort. Depending on the optimization algorithm and the complexity of a problem, several thousand calculations can be required.

The model structure is based on the detailed cooling coil model of the ASHRAE[4] HVAC Secondary Toolkit (Brandemuehl 1993, Chillar and Liesen 2004), implemented in TRNSYS as Type1223new. The model accounts for possible condensation of vapor on parts or even all over the heat exchanger surface by considering separately wet and dry parts of the heat exchanger surface. Furthermore, heat losses to the environment are neglected and no capacity effects are implemented.

Type1223new uses airside heat transfer correlations of Elmahdy and Biggs (1979), which is restricted to coils with circular or continuous plain fins and to the coil dimensions used in the experiment (9 samples with plain fins and 12 finned tube heat exchangers). In order to extend the validity range and the generality of the model, other correlations were identified in literature, which had been derived from larger databases. Unfortunately, different correlations cannot be directly compared in terms of heat transfer coefficients, because different data reduction methods were used by the authors, so that it is difficult to choose the "best" one.

Jacobi et al. (2001) reviewed many correlations and recommended for plain fin round-tube heat exchangers

[3] TRNSYS is a flexible tool designed to simulate the transient performance of thermal energy systems
[4] American Society of Heating, Refrigerating and Air-Conditioning Engineers

the correlations of Wang et al. (1998) for dry surface and Wang et al. (1997) for wet surface. Nevertheless, the more recent correlations of the same authors (Wang et al. 2000a and 2000b) have been implemented in the model, which include data of previous reports (in total 74 samples with dry surface and 31 samples with wet surface).

The structure of Type1223new has been also extended to the calculation of pressure drop on both water and air sides. Airside pressure drop is calculated using friction correlations (Wang et al. 2000a and 2000b). Water side pressure drop consists of friction pressure losses in tubes and local pressure losses in bends and is calculated according to the standard equations of Hagen–Poiseuille, Blasius and Nikuradze (Wagner 2001) depending on the flow regime. Further differences to Type1223new are temperature-depending physical water and air properties, evaluated at the mean flow temperature, instead of constant values. Furthermore, convergence procedures had to be adapted to increase the model stability for optimization calculations.

## 4. Heat exchanger cost function

The most common heat exchanger cost function types – as a function of heat transfer rate or heat exchange surface – are not appropriate for fin-and-tube heat exchangers. The heat transfer rate depends not only on the heat exchanger geometry, but also on operating parameters, so that the same heat exchanger can have different heat transfer rates depending on operating parameters. Cost functions as a function of heat exchange surface are available for plate and shell-and-tube heat exchangers (UITA 2002, Hewitt 2002). However, it cannot be applied for fin-and-tube heat exchangers as the ratio between fin and tube surfaces, made of different materials (e.g. aluminum and copper) and having different costs, is not considered. Such a cost function would move the optimization algorithm to increase the tube surface comparing to the fin surface (heat transfer coefficient on the tube surface is higher than on fins) and, thus, falsify the optimization results.

Therefore, another cost function type is proposed here. Taking into account that fins (or aluminum plates), tubes and heat exchangers are industrial mass products and the manufacturing process is highly automated, it can be assumed that the material costs are the most sensitive parameter in the price formation of the heat exchangers. A heat exchanger cost function depending on the material costs has been developed with data of 74 heat exchanger samples with staggered tube layout varying from small heat exchangers (1.25 m x 1 m flow area) to very large ones (11 m x 2 m flow area). All samples are from a large manufacturer (Güntner Group). Unfortunately, heat exchangers of other producers cannot be used due to the lack of necessary information about geometry and/or prices. Heat exchanger prices have been calculated by the producer-specific software Güntner Product Calculator (GPC 2006) for different geometry parameters. To estimate the material costs, the mass of aluminum (fins) and copper (tubes) needed for the heat exchanger was multiplied with aluminum and copper prices from the London Metal Exchange (LME). The developed cost function with prices from 2006 is shown in Eq. 4 and 5:

$$C_{hx} = -0.0005 \cdot C_{mat}^2 + 10.338 \cdot C_{mat} + 432.84 \qquad (4)$$

with material costs

$$C_{mat} = m_{Al} \cdot C_{Al} + m_{Cu} \cdot C_{Cu} \qquad (5)$$

$C_{Al} = 1.7$ €/kg and $C_{Cu} = 4$ €/kg (LME 01/2006)

The cost function is in a good agreement with the source data (Fig 5). The deviation is up to 15% for small heat exchangers and up to 10% for medium and large heat exchangers.

Although, the cost function is derived from prices from only one producer and some geometry parameters could not been varied over a wide range restricted by available standard values (tube diameter, longitudinal and transverse tube pitch), it can still be assumed that the heat exchanger costs will be predicted pretty well because the different producers have very similar prices because of the strong market competition.

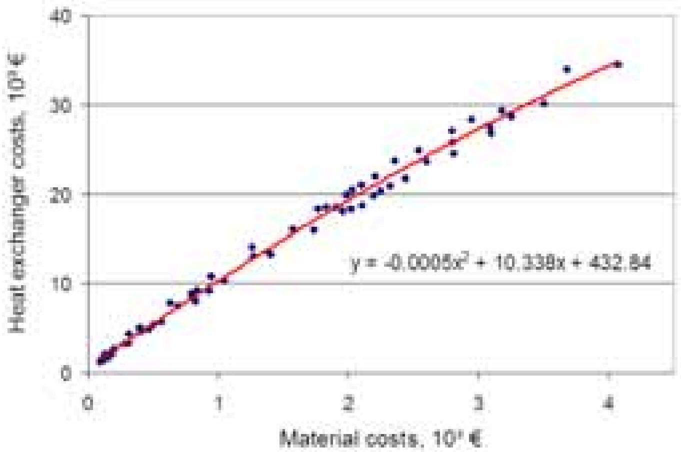



**Fig. 5. A cost function of the heat exchanger $C_{hx}$ (line) and the source data (dots)**

## 5. Fans, pumps and piping

Air or water flow rates and pressure drops determine the required fan or pump. During the optimization process, different heat exchanger geometries and operating conditions need to be calculated. For all these configurations it is necessary to estimate the costs and the parasitic energy demand of fans and pumps. As shown in (Fan 2009) for fans, estimation of the power consumption based on a constant fan efficiency and a cost function depending on operating parameters (flow rate and pressure drop) have large uncertainties over the large range of operating parameters. This makes these approaches not suitable for an optimization of the heat exchanger design. Therefore, a tool for automated selection of an appropriate fan from a special data base was developed in (Fan 2009). The data base contains parameterized characteristic curves (coefficients of the total pressure increase and the efficiency as a function of the flow rate) and costs of about 400 axial fans with an impeller diameter from 630 mm to 1250 mm. Although all fans are from one producer (Wolter 2009), a comparison with fans of other producers (with available information about fan power consumption and prices) showed that prices of the chosen producer are often lower or in the same range as others, so that no fans from other producers need to be included in the data base (Fan 2009).

Another advantage is the possibility to select not only the fan nearest to the operating point, but also the fan with the lowest annual costs (incl. operation costs), Eq. 6. In this case, however, additional information about the capital recovery factor $f_a$, the expected hours of operation per year $t_{op}$ and the electricity price is necessary $c_{el}$ (Table 1).

$$C_{ann} = f_a \cdot C_{fan} + c_{main} + P_{fan} \cdot t_{op} \cdot c_{el} \qquad (6)$$

In a similar way, a data base for pumps has been established containing characteristic curves and costs of about 100 pumps. All pumps in the database are from a large producer (WILO 2009).

For the selection of the pump, additional information about pressure losses in other hydraulic parts (piping) of the system is necessary. The pressure drop in the piping, caused by friction, is calculated according to the equations of Hagen–Poiseuille, Blasius and Nikuradze (Wagner 2001) depending on the flow regime. The local pressure losses in fitting and elbows are assumed to be equal to the friction pressure losses, which is typical assumption in water supply calculations (Recknagel et al. 2005). Furthermore, the selection of the pump is coupled with a selection of the piping diameter, so that a combination of both leads to the lowest annual costs of hydraulics (Eq. 6). Piping costs are calculated using prices for polypropylene pipes (Akatherm 2009) with 50% extra for fittings.

## 6. Optimization results

A heat exchanger for a test plant in Bishkek (Frank et al. 2006a) was redesigned using the described design procedure and compared with the installed one, designed by an established producer. The boundary conditions for dynamic simulations are listed in Table 1.

**Table 1. Boundary conditions for dynamic simulations**

| Parameter | Value | Unit |
|---|---|---|
| Simulation period | May-Sep (frost-free period) | |
| Weather data for Bishkek | from Meteonorm 6.0 | |
| Simulation and data time step | 1 | hour |
| Water inlet temperature | 12 | °C |
| Water flow rate | 6 | $m^3/h$ |
| Control function | $T_{amb} > T_{w\_in}$ | |
| Electricity price $C_{el}$ | 2 | €-ct/kWh |
| Interest rate $i$ | 6 | %/a |
| Years in operation $n$ | 10 | a |
| Maintenance costs $C_{mnt}$ | 1 | % of CAPEX |
| Expected operation hours per year of a fan and a pump $t_{op}$ | 4000 | h/a |
| Piping length | 100 | m |

In the optimization nearly all geometry parameters and the air flow rate have been varied over a wide range. The variation range of some parameters has been restricted by production possibilities (e.g. fin pitch and thickness) or even distribution of the air flow (finned length and height). Tube thickness has not been varied, because the smallest tube thickness will always be selected by the optimization concerning the material costs, pressure drop on the water side and heat conduction in the tube. The smallest tube thickness is determined by the operating pressure. Therefore, the same tube thickness of 0.32 mm has been specified as in the installed heat exchanger. Other fin (e.g. louvered, split fins) and tube (e.g. oval, flat) types have not been considered because of the absence of respective heat exchanger cost functions. The varied parameters are listed in Table 2 together with their variation range, value at the installed heat exchanger (Ref) and optimal values (Opt). A comparison of different properties of the reference and optimal configurations relevant for the evaluation is given in Table 3.

**Table 2. Varied parameters, their variation range and values at the reference and optimal configurations**

| Parameter | Unit | Variation range | | Ref | Opt |
|---|---|---|---|---|---|
| | | min | max | | |
| Air flow rate | $m^3/h$ | 9,000 | 40,000 | 10,000 | 20,000 |
| - capacity flow ratio | | | | 2.1 | 1.0 |
| Fin pitch $F_p$ | mm | 1.5 | 5 | 3 | 1.5 |
| Fin thick | mm | 0.15 | 0.45 | 0.25 | 0.15 |
| Finned length $L$ | m | 0.8 | 1.5 | 1.25 | 1.3 |
| Finned height $H$ | m | 0.8 | 1.5 | 1 | 1.5 |
| Tube rows | | 2 | 9 | 6 | 4 |
| No. of passes | | 2 | 16 | 10 | 6 |
| Tubes per row | | 15 | 90 | 20 | 50 |
| - Transverse tube pitch $P_t$ | mm | | | 50 | 30 |
| Longitudinal tube pitch $P_l$ | mm | 15 | 50 | 25 | 20 |
| Tube outer diameter | mm | 6 | 22 | 12 | 6 |

**Table 3. Comparison of different properties of the reference and optimal configurations**

| Parameter | Unit | Ref | Opt | change |
|---|---|---|---|---|
| Heat generation costs *HGC* | €-ct/kWh | 0.75 | 0.53 | -29% |
| Annual transferred heat *Q* | MWh/a | 82 | 132 | 61% |
| Investment costs $C_{inv}$ | € | 3681 | 4008 | 9% |
| Maintenance costs $c_{main}$ | €/a | 37 | 40 | 9% |
| Operating costs $c_{op}$ | €/a | 79 | 117 | 47% |
| Heat exchanger costs $C_{hx}$ | € | 1841 | 1727 | -6% |
| Fan costs $C_{fan}$ | € | 947 | 1253 | 32% |
| Fan electrical power $P_{fan}$ | kW | 0.50 | 1.06 | 113% |
| Pump costs $C_{pump}$ | € | 563 | 563 | 0% |
| Pump electrical power $P_{pump}$ | kW | 0.70 | 0.70 | 0% |
| Piping costs $C_{pipe}$ | € | 330 | 465 | 41% |
| Outer diameter of piping tubes $D_{out\_pipe}$ | mm | 40 | 50 | 25% |
| Heat exchanger airside pressure drop $\Delta p_{hx\_air}$ | Pa | 71 | 67 | -6% |
| Heat exchanger waterside pressure drop $\Delta p_{hx\_liq}$ | kPa | 34 | 123 | 265% |
| Piping pressure drop $\Delta p_{pipe}$ | kPa | 152 | 53 | -65% |
| Pump head | kPa | 193 | 193 | 0% |

The objective function HGC of the optimal design is approx. 30% lower than in the reference configuration, this is caused by a different geometry and different operating parameters. In the optimal configuration the air flow rate has been doubled and the fin pitch is only half of that in the reference configuration (Table 3). These changes had been expected due to the low electricity price. A higher air flow rate and a smaller fin pitch would lead to a higher heat transfer coefficient on the one hand and a higher airside pressure drop on the other hand. The airside pressure drop of the heat exchanger remained, however, at the same range (Table 3) due to a larger frontal area (product of heat exchanger length and height) and a shorter flow length (product of longitudinal tube pitch and number of tube rows). Nevertheless, a more expensive and powerful fan is required due to a higher air flow rate. The increased waterside pressure drop of the heat exchanger, caused by a smaller tube diameter, was compensated by selecting a larger piping diameter, so that the same pump is used in both configurations.

Although the fin surface is 75% larger in the optimal configuration (approx. 200 m²) than in the reference one (115 m²), the mass of fins is almost the same (42 kg) in both cases due to the different fin thicknesses. The heat exchanger costs decreased slightly because less copper is needed for smaller tube (13.6 kg of copper instead of 16.3 kg in the reference configuration). The total heat transfer area is increased from 120 m² to 200 m². Together with a higher air flow rate and a smaller fin pitch, a larger heat transfer area led to significantly higher annual energy gains *Q* (heat transfer). Total investment costs are approx. 10% higher and operation costs are approx. 50% higher (mostly) caused by a more expensive fan with a higher power consumption.

It has to be emphasized, that the improvement of the heat costs is the result of combination of all parameters. Changing of only one parameter without changing others can even lead to higher heat costs due to the very complex dependencies between parameters (e.g. doubling of the air flow rate in the reference configuration leads to 6% higher HGC).

## 7. Conclusion

This paper describes an air-to-water heat exchanger design procedure for water preheating in open district heating nets. The design procedure includes an integrated economical optimization. The target function for the optimization is the cost effectiveness. In the optimization, different heat exchanger geometry and operating parameters have been varied by a genetic optimization algorithm. The influence of the heat exchanger design on its costs, heat transfer and friction performance has been evaluated economically by determining investment and operation costs of the system.

For the test plant in Bishkek, an optimized heat exchanger design leads to 30% lower heat generation (transfer) costs compared to the installed one, which was designed by a heat exchanger producer. Such a large optimization potential is definitely to some extend caused by special boundary conditions of the non-standard application and by the lack of experience of the design engineer with these constrains.

The design procedure can be further used for other application of fin-and-tube heat exchangers such as air collectors, air conditioning or heat pumps. A critical point in the optimization procedure is the relatively high uncertainty of the heat exchanger model and cost functions. In order to increase the accuracy and reliability of the optimization results, more precise heat exchanger models and cost functions are necessary.

## 8. Nomenclature

| | | | | | |
|---|---|---|---|---|---|
| $C_{inv}$ | € | Investment costs | $HGC$ | €/kWh | Heat generation costs |
| $C_{el}$ | €/kWh | Electricity price | $i$ | %/a | Interest rate |
| $C_{fan}$ | € | Fan costs | $L$ | m | Heat exchanger finned length |
| $C_{hx}$ | € | Heat exchanger costs | $n$ | a | Years in operation |
| $c_{main}$ | €/a | Maintenance costs | $P_{fan}$ | W | Fan electrical power |
| $c_{op}$ | €/a | Operation costs | $P_l$ | m | Longitudinal tube pitch |
| $C_{pipe}$ | € | Piping costs | $P_{pump}$ | W | Pump electrical power |
| $C_{pump}$ | € | Pump costs | $P_t$ | m | Transverse tube pitch |
| $CHP$ | | Combined heat and power | $Q_{use}$ | kWh/a | Annual heat gain |
| $CIS$ | | Commonwealth of Independent States | $T_{amb}$ | °C | Ambient temperature |
| $f_a$ | | Capital recovery factor | $T_{w\_in}$ | °C | Water inlet temperature |
| $D_{out\_pipe}$ | m | Tube outer diameter (piping) | $\Delta p_{hx\_air}$ | Pa | Heat exchanger airside pressure drop |
| $F_p$ | m | Fin pitch | $\Delta p_{hx\_liq}$ | Pa | Heat exchanger waterside pressure drop |
| $H$ | m | Heat exchanger finned height | $\Delta p_{pipe}$ | Pa | Pressure drop in piping |

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