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OPTIMIZATION OF THE FIN-AND-TUBE HEAT EXCHANGER DESIGN FOR NON-STANDARD APPLICATIONS

Janybek Orozaliev, Christian Budig, Klaus Vajen
Institute of Thermal Engineering, Kassel University, www.solar.uni-kassel.de
Kassel, Germany

ABSTRACT

This paper describes an optimization procedure of the heat exchanger design, which is based on cost effectiveness. In the procedure, a genetic optimization algorithm varies the heat exchanger geometry and operating conditions. 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 this purpose, a specific approach for setting up of cost functions of the heat exchanger, fans and pumps is presented as well.

It has been shown on a non-standard application, that the heat transfer costs of the optimized heat exchanger design is 30% lower than that of the reference configuration, which was designed by a heat exchanger producer.

NOMENCLATURE

CAPEX	€	Capital expenditures (investment costs)
C_{el}	€/kWh	Electricity price
C_{fan}	€	Fan costs
C_{hx}	€	Heat exchanger costs
C_{mnt}	€/a	Maintenance costs
C_{op}	€/a	Operation costs
C_{pipe}	€	Piping costs
C_{pump}	€	Pump costs
CRF		Capital recovery factor
$D_{out,pipe}$	m	Tube outer diameter (piping)
F_p	m	Fin pitch
H	m	Heat exchanger finned height
HGC	€/kWh	Heat generation costs
i	%/a	Interest rate
L	m	Heat exchanger finned length
n	a	Years in operation
P_{fan}	W	Fan electrical power
P_l	m	Longitudinal tube pitch
P_{pump}	W	Pump electrical power
P_t	m	Transverse tube pitch
Q_{use}	kWh/a	Annual heat gain
T_{amb}	°C	Ambient temperature
$T_{water,\ in}$	°C	Water inlet temperature
Δp_{hx_air}	Pa	Heat exchanger airside pressure drop
Δp_{hx_liq}	Pa	Heat exchanger waterside pressure drop
Δp_{pipe}	Pa	Pressure drop in piping

1. INTRODUCTION

Fin-and-tube heat exchangers (Fig. 1) are widely used for heat transfer between a liquid and a gas in industry and in the residential air conditioning sector, e.g. for water cooling, air cooling or heating. The heat exchanger design is a complex problem, since the heat exchangers performance and costs depend on many variables (geometrical and operating parameters). In practice, many of these variables (e.g. fin pitch, tube longitudinal and transverse pitch, number of tube rows, number of tubes in a row etc.) are selected by designers relying on their personal experience and some recommendations[1]. This procedure might work well for standard applications. But there are more detailed investigations necessary for non-standard applications with different boundary conditions (e.g. low electricity prices, no binding requirement for heat transfer rate or change of air/liquid temperature) because of the lack of experience.

[1] For example, air flow velocity > 2 m/s for even distribution and < 3.5 m/s, otherwise pressure drop too high

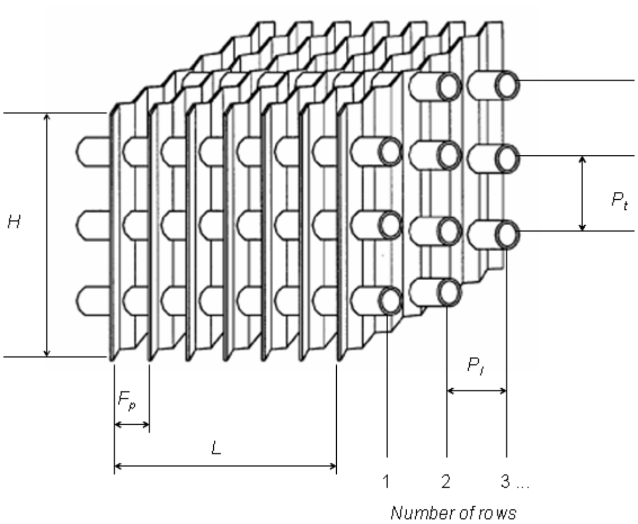

Figure 1. Schematic diagram of a typical fin-and-tube heat exchanger (taken from [1] with own changes)

Furthermore, the heat exchanger geometry and operating parameters strongly affect the heat transfer and pressure drops and, thus, investment and operating costs. For instance, higher flow rates and pressure drops lead to more expensive fans or pumps and a higher parasitic energy demand i.e. higher operating costs. Therefore, not only the heat exchanger itself but also its influence on the whole system including fans and pump has to be considered in the design process to achieve the most cost-effective solution. Preferably investment and operation costs of all components should be used to quantify these influences. However, it is generally problematic to estimate costs, so that often other criteria are applied, e.g. the ratio of Colburn and friction factors [2].

Nevertheless, in this study heat exchanger configurations are evaluated economically involving investment and operation costs of all components and heat exchanger performance. Furthermore, a genetic optimization algorithm is applied to select the optimal heat exchanger design. The optimization procedure, a heat exchanger model (which describes the heat exchanger performance depending on its geometry and operating parameters), parasitic energy and cost functions of the system components and an objective function for the optimization are described in the next sections. The focus of this study is, however, the specific approach of the optimization procedure and setting up of cost functions for evaluation of heat exchanger configurations. The design procedure is illustrated with an example of a non-standard application.

2. APPLICATION

A fin-and-tube heat exchanger is applied to pre-heat cold water (12°C) during the summer for a district heating net in Bishkek, the capital of Kyrgyzstan, using the ambient air (up to 40°C during the day and 20…25°C during the night). The pre-heated water is further heated to the supply temperature of 60°C by conventional boilers (e.g. gas or coal boilers) [3]. 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 also not restricted. 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. ε-NTU calculation) is not possible without additional information or assumptions.

The application has a large potential in Central Asia. There are several thousand small heating plants (up to 50 m³/h water flow rate) and several large combined heat and power plant (approx. 3000 m³/h water flow rate) in operation in the region.

3. HEAT EXCHANGER DESIGN OPTIMIZATION

The heat exchanger design (sizing problem) can be 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 applied 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 [4] 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:

$$HGC = \frac{CRF \cdot CAPEX + C_{mnt} + C_{op}}{Q_{use}} \qquad \frac{€}{kWh_{th}} \tag{1}$$

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

Capital expenditures CAPEX include heat exchanger, fan, pump and piping costs:

$$CAPEX = C_{hx} + C_{fan} + C_{pump} + C_{pipe} \tag{3}$$

Operation costs C_{op} are calculated by multiplying the annual parasitic energy demand (electricity consumption by fan and pump) and electricity price. The parasitic energy demand and heat gain (transferred from ambient air to water) are determined by annual simulations, since ambient air temperature and humidity are time dependant 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.

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

The optimization procedure is shown in Fig. 2. Heat exchanger configurations are specified and evaluated by a genetic optimization algorithm, implemented in open-source optimization software GenOpt [5]. The objective function for each configuration is evaluated by annual simulations in the simulation environment TRNSYS[3] [6]. Both programs interact with text files.

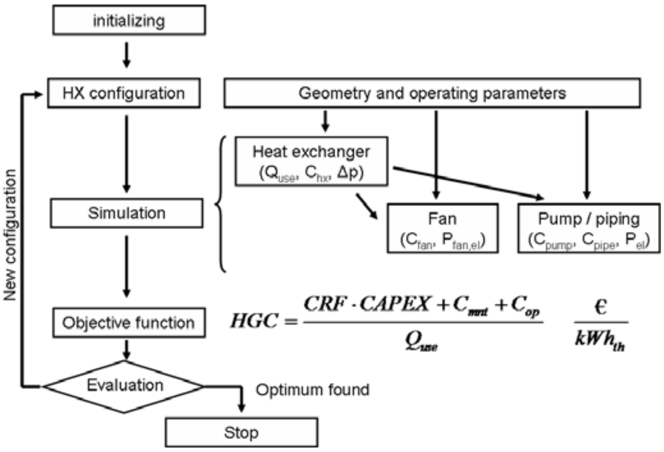

Figure 2. Scheme of the optimization procedure.

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.

4. HEAT EXCHANGER MODEL

The heat exchanger model uses empirical heat transfer and flow friction correlations and is described in detail in [7]. 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 [8, 9], 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 [10], 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, os that it is difficult to choose the “best” one.

Jacobi et al. [11] reviewed many correlations and recommended for plain fin round-tube heat exchangers the correlations of Wang et al. [12] for dry surface and Wang et al. [13] for wet surface. Nevertheless, the more recent correlations of the same authors [14, 15] 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 [14, 15]. 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 [16] 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.

5. 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 [17, 18]. However, it can not 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

[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

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 (1.25m x 1m flow area) to very large ones (11m x 2m 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) [19] 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 $C_{mat} = m_{Al} \cdot C_{Al} + m_{Cu} \cdot C_{Cu}$ (5)

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

The cost function is in a relatively good agreement with the source data (Fig 2). The deviation is up to 15% for small heat exchangers and up to 10% for medium and large heat exchangers.

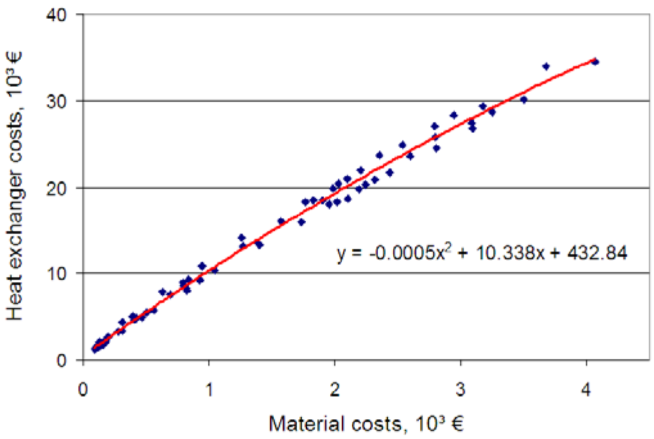

Figure 2. A cost function of the heat exchanger C_{hx} (line) and the source data (dots)

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.

6. 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 [20] 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 [20]. 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 GmbH [21]), 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 [20].

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 CRF, the expected hours of operation per year t_{op} and the electricity price is necessary C_{el} (Table 1).

$$C_{ann} = CRF \cdot C_{fan} + C_{mnt} + P_{fan} \cdot t_{op} \cdot C_{el} \qquad (6)$$

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 AG [22].

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 [16] 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 [23].

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 [24] with 50% extra for fittings.

7. OPTIMIZATION RESULTS

A heat exchanger design for a test plant in Bishkek [3] was optimized 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 [25]	
Simulation and data time step	1	hour
Water inlet temperature	12	°C
Water flow rate	6	m³/h
Control function	$T_{amb} > T_{water,\ in}$	
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.

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).

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	m³/h	9000	40000	10000	20000
- capacity flow ratio				2.1	1.0
Fin pitch Fp	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
HGC	€-ct/kWh	0.75	0.53	-29%
Q_{use}	MWh/a	82	132	61%
$CAPEX$	€	3681	4008	9%
C_{mnt}	€/a	37	40	9%
C_{op}	€/a	79	117	47%
C_{hx}	€	1841	1727	-6%
C_{fan}	€	947	1253	32%
P_{fan}	kW	0.50	1.06	113%
C_{pump}	€	563	563	0%
P_{pump}	kW	0.70	0.70	0%
C_{pipe}	€	330	465	41%
$D_{out,pipe}$	mm	40	50	25%
Δp_{hx_air}	Pa	71	67	-6%
Δp_{hx_liq}	kPa	34	123	265%
Δp_{pipe}	kPa	152	53	-65%
pump head	kPa	193	193	0%

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_{use} (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).

8. CONCLUSION

An optimization procedure of the heat exchanger design is described in this paper and shown on a non-standard application. 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 investigated application, an optimized heat exchanger design leads to 30% lower heat generation (transfer) costs compared to the reference configuration, which was designed by a heat exchanger producer and installed in Bishkek. 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. An optimization potential for standard applications should be less than the potential, estimated in this study.

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. Therefore, this optimization procedure can be used first of all by heat exchanger producers or designers. Furthermore, this procedure shall be applied primarily for non-standard applications or large complex systems, where additional effort for design is required.

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