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Faculty of Mechanical Engineering and Informatics

# Optimal design of heat exchanger devices

PhD Booklet of PhD Dissertation

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Miskolc, 2021

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# 1 Objectives

## 1.1 Introduction

Without heat exchangers, our modern standard of living would be virtually unsustainable, as they are present in all aspects of life, often even invisibly. We can think of the electricity that flows through our power lines (boilers in power plants), our utility equipment such as the coverings of electrical appliances, furniture upholstery, cushioning, packaging materials (process heat exchangers in chemical plants), our heating and cooling equipment in homes (radiators, air conditioners), our processed food (dried, concentrated foods, alcoholic beverages) or even transport (heat exchangers cooling car engines). The list can be extended to include equipment that uses energy that would otherwise be waste, increasing the energy efficiency of consumers.

There are no strict rules for the choice of heat exchanger equipment for a given task, practically everything will be influenced by external conditions. It is essential to think systemically, because in most cases they will not work on their own. When several devices are integrated in one technology, the mass flow of the medium must be considered constant, which requires the geometric dimensions to be defined.

However, when it comes to the sizing of equipment, it is important not to stop at simply examining the thermal conditions. When building the equipment and designing the supporting structure, the most important parameter is the weight of the equipment. The shell of the equipment will not be involved in the heat exchange, from a thermal point of view it is irrelevant, but from a pressure point of view it will be the most important geometrical characteristic. This wall thickness can be determined using the boiler formula, which shows that it is in inverse proportion to the diameter and pressure, and in inverse proportion to the mechanical properties of the steel. Small pressures and diameters can result in extremely small wall thicknesses, but in practice it is not common to make a pressure vessel from a plate smaller than 6 mm. When carrying out the optimisation, this will be a stringent condition, significantly influencing the optimal dimensions.

As before, the optimisation should use several geometric constraints related to operation and construction. Some examples of these are:

- All but some of the finned tube heat exchangers are axisymmetrical. In this respect, it is not advisable to make short, large-diameter devices, with a length or diameter ratio of more than 1-1.5. With a larger diameter, the wall thickness will be larger; the flow cross-section will be larger, which will reduce the flow velocity, thus reducing the heat transfer coefficient. The weight of the tube sheet wall increases and many more pipes have to be welded. As a result of rolling, it may be necessary to create several axially directed welding, which are the most dangerous type of welds for the device.
- Designing appliances that are too slim should also be avoided. They can be more favourable from a strength point of view (for example, the shell can be made from seamless pipe material, thus avoiding the use of welding), but more problematic from a support point of view. If they are supported by two saddles, they can buckle due to the self-weight and the weight of the load, which can be avoided with more saddles, but then you have to take into account the extra cost of these. Their use is also disadvantageous from a flow and thermal point of view. Due to the small diameter, the flow cross-section will be small, thus increasing the flow rate. This increases the heat transfer coefficient up to a certain value, but

significantly reduces the residence time. The only way to increase the heat transfer surface area is to increase the length of the tubes. Together, these will create an extremely high pressure drop, which will increase operating costs.

- The wrong choice of structure can cause operational and maintenance problems.

The temperature dependence of material properties should not be forgotten in thermal calculations. In terms of driving force, a heat exchanger in which the temperature of hot medium decreases from 80°C to 60°C while the cold medium heats up from 30°C to 40°C is no different from another in which it cools from 60°C to 40°C, while the other changes from 10°C to 20°C if a countercurrent is assumed. However, the material properties required for thermal calculations vary with the mean temperature of the media, so the heat transfer coefficient can be slightly affected. However, it is not practical to optimise these temperature values. As I mentioned, these heat exchangers operate in a system, receiving auxiliary energy from other parts of the plant. If the operating parameters of the equipment were to be optimised on this basis, it is very likely that another heat exchanger would have to be installed to achieve the optimum temperature, which would significantly increase costs.

Due to the IT background, it is nowadays essential to use the Computational Fluid Dynamics (CFD) method to study processes, the results of which are compared with those obtained from the measurement of physical processes. Modelling makes the process of innovative design and optimisation faster and more economical. Measurements taken in a specimen are representative of the real behaviour of the device under test. Using the measured results and a realistic geometric model, the parameters of the CFD simulation (element size, boundary conditions, turbulence model used, mesh compression positions) can be determined and treated as real results.

## 1.2 *Objectives*

As described in the introduction, it can be seen that a wrongly chosen equipment with a wrongly defined geometry can lead to uneconomical operation for the user. In my research, I aimed at determining the most accurate computational model of the equipment selection procedure and the resulting heat transfer conditions.

1. For double pipe, shell-and-tube and finned tube heat exchanger designs, I would like to develop an optimization procedure with the lowest operating and material costs, but capable of delivering the designed heat output.

2. For finned tube heat exchanger types, I would like to define an empirical Nu-number correlation, which can be more accurately applied to the specific type under investigation, consistent with the measured results.

3. I would like to achieve the optimum sizing of steel structural elements to support large equipment and to compare the optimised results under different design specifications.

# 2 Research methods

## 2.1 Definition of the objective function

Optimisation of heat exchangers can be mean as the optimisation of a single element of the plant, the optimisation of the whole structure in some aspect, as well as the optimisation of the whole plant. In the literature, proportionally most research is focused on the first case, i.e. the study of a single structural element. This means turbulence discs and baffles on the shell side in case of double pipe and shell-and-tube heat exchanger types, and fins with different sizes, shapes and pitches in case of finned tube heat exchangers.

During my research, I have focused on the second level, the optimisation possibilities of the whole heat exchanger structure. My goal was to mathematically formulate a cost function that takes into account the manufacturing and operating costs of the equipment. The manufacturing costs consist of the costs of the structural material, cutting, edge preparation and welding of the equipment, while the operating costs consist of the maintenance costs and the required pump power.

The key issue in the calculations is the weighting factor between material cost and operating cost. The determination of this factor depends on the design lifetime of the equipment, its role in the system and the physical properties of the media flowing through it. The performance of the equipment is directly proportional to the heat transfer surface and the heat transfer coefficient. In the optimisation process, this thermal performance will be the initial condition to which we want to associate the lowest cost. One possible way to increase the heat transfer coefficient is to increase the medium velocity. As a consequence, the heat transfer surface area can be reduced, which will mean lower manufacturing costs. Conversely, this higher medium velocity will result in higher pressure drop and wear, more maintenance, which will result in higher operating costs. The optimisation procedure takes all these factors into account [1, 2].

During my research, it was essential to investigate the effects of different turbulence enhancing devices or otherwise turbulators, which are summarised in my dissertation. For these studies I did not test the accuracy of the literature data, but used it as a parameter to build the algorithm.

I have used the correlations in optimization algorithms that can be used by practising design engineers in their work. These methods were the Generalized Reduced Gradient method [3], and the Sequential Unconstrained Minimization Technique [4] and [5]. Using the correlations I have constructed, I have shown that it is possible to construct relationships in such a way that they provide a solution.

## 2.2 Numerical simulations

In my research, I have investigated heat transfer operations from three perspectives: experimental investigation, numerical simulation and analytical calculation. Whenever I had the opportunity, I carried out experiments on real equipment and built a numerical model to compare the results with these different investigating methods. Then examined the results obtained using the relationships found in the literature. If this model did not give similar results to the previous ones, I modified the constant factors in the relations to build a better computational model for the task at hand, which I can use in the optimization [6].

For surface heat exchangers, two typical investigations are common: the determination of the heat transfer coefficient between the flowing fluid and the wall and the resistance (pressure drop) of the structure. In my case, this determination is not sufficient for the used turbulence model, because it is different for a liquid-liquid type tubular heat exchanger than for a gas-liquid type finned tube heat exchanger [7, 8].

During my research, I have used two software for numerical simulations, which are ANSYS and SC-Tetra. For finned tube heat exchangers, I have used two different modelling techniques. In one technique, I modelled each fin and the air volumes between the fins individually, which included a large mesh, and modelled the whole system as a single porous volume. This has the advantage of a much smaller mesh size and is easy to use with CFD programs without a graphical interface (with open source software). The goal of my research is to apply this type of modelling technique to louvered fin heat exchangers, which have an extremely large number of fins and very small fin spacings. My results suggest that this simplification technique cannot be applied.

## 2.3 *Experimental investigation of finned tube heat exchangers*

To perform the measurements an experimental setup was built, which is shown in Figure 1. Finned tube heat exchangers are the most commonly used heat exchangers between liquid and gas phase media and this is what I implemented in the measurements: the gas phase was ambient pressure air and the liquid was water. Hot water was circulated in the tubes of the heat exchanger and produced using a domestic water heater, which was insulated. A Pt100 thermocouple was placed in the bushing of the boiler and connected to a temperature controller. This electric circuit was used to set the temperature of the hot water. The system was pressurized in order to circulate it with the circulating pump. The data needed to perform the calculations is the volume flow rate of this circulated water. This was read with a rotameter calibrated for water.

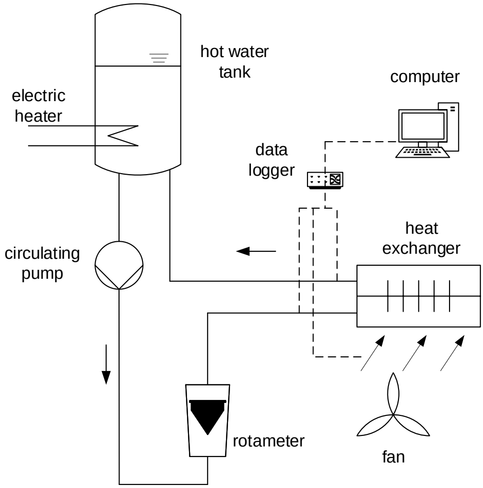


Figure 1: The built experimental set-up

Directly before entering and after leaving the heat exchanger, an entry gland for K-type thermocouples was installed to measure the water temperatures. A Quantum X type MX1609 thermocouple amplifier with type K thermocouples was used for data collection. Two temperatures were measured: the inlet and outlet temperatures of the heat exchanger with 11 Hz sampling rate. Data processing was done with catman®Easy software.

During all measurements I waited for the steady state to develop. The last two minutes of the measured data were taken into account, so I always calculated the arithmetic mean of the 120 measured values, and the material properties were also taken into account at these temperatures, using the polynomial functions in Annex A (for these approximate relations I used the UniSim Design (Honeywell International Inc.) software database).

The equipment is designed to measure several types of heat exchangers. In addition to the computer and data logger, everything is mounted on a stand that can be moved by means of a lifting device. Experiments were carried out with two different designs of finned tube heat exchanger: a heat exchanger with circular fins on a U-shaped tube (Figure 2) and an automotive radiator. In the former case, the equipment was placed in a wind tunnel in the laboratory of the Institute of Energy Engineering and Chemical Machinery, which provided the external air flow. In the latter case, the radiator was tested as a complete unit with two fans of different diameters.

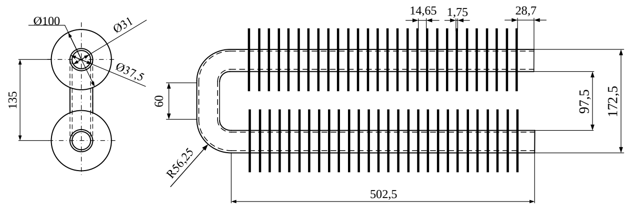


Figure 2: The investigated U-bend shaped heat exchanger

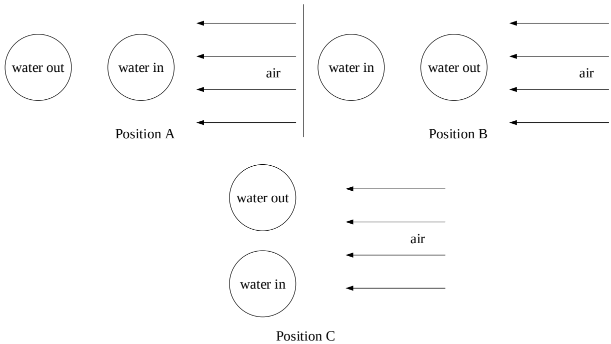


Figure 3: Orientation of heat exchanger relative to the air flow

In case of finned tube heat exchangers, according to the classical theory [9] all equipment counts as cross-flow, since air flows in a direction perpendicular to the fins and thus to the tubes. After evaluation of the measurements, using the correlations found in the literature, the calculated results gave large deviations from the measured data. Consequently, the arrangements shown in Figure 3 were determined with different temperature differences.

I named the Position A as quasi direct current and position B as quasi-countercurrent. For these, I determined the value of $\Delta T_{LOG}$ according to the name, and defined a new empirical Nu number relation. For the C-position, I assumed real cross current, for which I defined the driving force by the temperature difference associated with the counterflow, but created a different empirical relation.

## 2.4 Optimization of steel structures

The studies presented in my thesis so far have focused on the convective heat transfer inside the heat exchanger structure, and have only dealt with the strength problems in a secondary way. As I have written several times, these devices are very rarely used in stand-alone applications, almost always they have to be considered in a system approach. This means that a steel structure has to bear several pieces of equipment (not only heat exchangers, of course, but also other equipment such as vessels, filters, centrifuges, compressors etc.), which means that the steel structure must also be designed to withstand the loads.

Steel structures are fundamental part of any factory or plant, they are practically its skeleton. During the investment, at the beginning of the construction, the installation and manufacturing of this structure has very high resource requirements. In practice, an optimisation process is already under way in the design of the plant; there would be no theoretical limit to building these plants without a major steel structure, using horizontal construction, nor to building production in very tall buildings. The disadvantages of the horizontal construction method are that it would require a very large area, which would reduce the size of the agricultural land, and that the flow of media would not be gravity fed, so that more pumps would be needed, which would increase the energy consumption of the plant excessively (which would be an ongoing expense). For tall frame structures, on the other hand, excessively large section cross-sections would be required, both because of the weight of the structure itself and the mass of the equipment it supports. In neither of these extreme cases is the intervention time for the operating staff negligible and in neither case ideal.

It is also relevant for a stand-alone equipment, for example an outdoor unit of an air conditioning system. Even for smaller installations, the computer room requires significant cooling, with a high capacity chiller and a large surface area fin heat exchanger. These are usually located either in the server room or on top of the control room. In the case of heavy equipment, large cross-sectional sections should be incorporated into the building structure. This mass should include the total mass of the equipment, thus the mass of the steel structure, the charge, the insulation, the fittings and all other associated accessories [10].

In terms of manufacturing aspects and connection point design, the best choice may be I- and box sections. The sizing/optimization process involved comparing several steel grades based on several standard specifications [11, 12, 13, 14]. In all cases, my objective function was the minimum cross-section area, from which the weight minimum and cost minimum follow. In addition to the above, I also examined these minimum cross-sections for several section lengths and loads in the mechanical model. Due to the large number of calculations, I automated the optimization procedure, creating a procedure that can be used by practising engineers. The mechanical model of the compressed and bent-pressed rod I have investigated is shown in Figure 4.

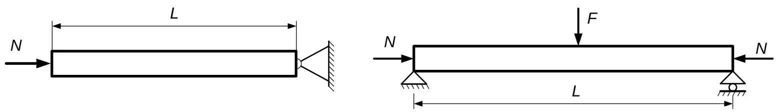


Figure 4: Mechanical model of the investigated beams

After performing the optimisation process of pressed I-sections and box sections and evaluating the results, I was able to conclude that for pressed sections the optimum cross-section areas are not for the highest strength steel and that depending on the type of section, different steel grades will give the lowest values.

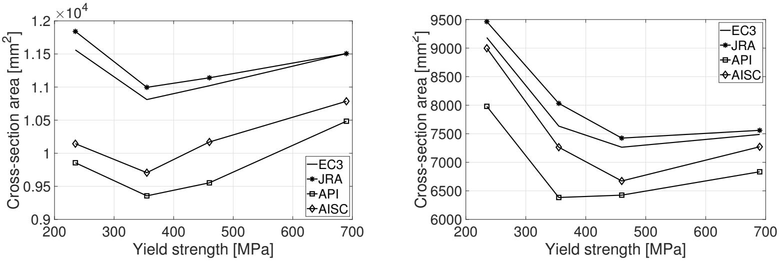


Figure 5: Optimal cross-section areas in the function of the yield strength

# 3 New scientific results

T1. For double pipe heat exchangers, I have created an optimization objective function which includes the two major costs of the equipment, the material cost and the operational cost. In addition to the optimization objective function, I have established the condition functions that provide the safe operation conditions. I have shown that, using these mathematical relationships, the optimisation task can be performed using generally applicable mathematical software.
Related publications: (8) (9).

T2. Using a simulation technique for shell-and-tube heat exchangers, I have shown that baffles in the shell always increase the heat transfer coefficient on the shell side compared to the case without baffles. I have shown that in the case of multi-objective optimization, the thermal optimum point does not coincide with the cost optimum.

The number of tubes that can be placed in a given diameter is a necessary information to perform optimization. I have developed a mathematically precise procedure to determine this number of tubes, since this has a significant effect on both the objective function and the condition functions.
Related publications: (2) (10) (16) (17).

T3. I have shown by investigating the heat transfer conditions of finned tube heat exchangers by measurement and simulation, that by applying the correlations found in literature, large variations in the thermal performance of the equipment are observed. From measurement results, I propose the following relation for quasi direct current and quasi countercurrent,

$$\mathrm{Nu} = 29.59 \cdot \mathrm{Re}_F^{0.2371} \cdot \mathrm{Pr}^{1/3},$$

while for real cross-current the following relation

$$\mathrm{Nu} = 55.40 \cdot \mathrm{Re}_F^{0.1897} \cdot \mathrm{Pr}^{1/3}.$$

In the presented relations, the characteristic geometry is calculated using the Schmidt relation and is applicable to atmospheric pressure air.

Related publications: (1) (4) (6) (7).

T3.1. My investigations for a simplified study of finned tube heat exchangers have shown that even in the case of rectangular fins, the simplification cannot be used, and the heat transfer coefficient for these type heat exchangers must be determined by measurement experiments.

T4. I have shown from results of measurements on two-tube passes finned tube heat exchanger that the generally used empirical correlations are not applicable. As a result of measurements made under several operating conditions, I have determined the

$$\mathrm{Nu}_l = 0.817 \cdot \mathrm{Re}_l^{0.6} \cdot \left(\frac{A}{A_{t0}}\right)^{-0.39} \cdot \mathrm{Pr}_l^{1/3}$$

Nu number correlation for the thermal calculation of this louvered finned heat exchanger. The relationship is applicable to atmospheric pressure air.
Related publications: (3) (5).

T4.1. I have shown that the effect of the humidity of the cooling air should not be neglected in the thermal calculations, and I have demonstrated with the measurement results that the value of the Nu-number cannot be assumed to be constant in the case of laminar flow in the tube side. The heat transfer coefficient on the inner side also changes with the change of the flow velocity in this flow regime.

T4.2. In case of louvered finned tube heat exchangers, I have shown that as the number of fins increases, the heat performance of the equipment increases, but the ratio of heat performance and unit mass has a maximum point, which ensures the optimization of the equipment. The thickness of the fins also has a significant effect on this maximum point, with larger width fins moving it towards smaller fin numbers but increasing its maximum value, while smaller width fins move it towards larger fins but its value will be smaller for these.

T5. I have shown with regard to the optimization of steel structures which can also support heat exchanger structures, that if the column is only loaded to a compressive force in the direction of the beams, it is more appropriate to use a box section than an I-section, as it will require much smaller cross-section areas. I have also shown that, due to the local buckling condition, the smallest cross-sections for I-sections are associated with 355 MPa yield strength, while for box sections, apart from the API results, is associated with the 460 MPa yield strength steel.
Related publications: (11) (12) (13) (14) (15).

T5.1. I have shown that if the compressive load in the direction of the beam is supplemented by a bending moment, the greater the ratio of this moment perpendicular to the axis of the web, the more the use of a box section is recommended. On the other hand, if it can be ensured that this load is applied in the plane of the web, the I-section is recommended. In the case of bent-pressed columns, it has been clearly demonstrated that the use of steels with higher yield strength is recommended, with higher strength characteristics being associated with a smaller load bearing cross-section.

# 4 Possible applications of the results

The basic task of my research was to prove the optimality of the heat exchanger equipment. I have been able to demonstrate this for the double pipe and shell-and-tube heat exchangers that I have chosen to test. I have created a calculation algorithm to determine the number of tubes for shell-and-tube heat exchangers, which will also help design engineers. However I have found a generally applicable empirical correlations for the analytical calculations of various finned tube heat exchangers in the literature that are generally. To test this heat transfer process experimentally, a measuring set-up was built, in which two types of heat exchanger with increased surface area were tested, a classical finned tube type and a louvered finned type. In both cases I found that the general correlation gave an inaccurate result, and as a consequence I had to look at the heat transfer process. In both cases, a new empirical correlation was defined.

I also used the numerical simulation environment to study the heat transfer processes. This gives us double advantages: on one hand, it allows us to measure parameters that cannot be measured in a real plant in real operation (for example surface or volume averaged temperatures, forces, pressures, torques on surface elements), and on the other hand, it allows us to study small changes in geometric and/or process parameters. In my research, I exploited both advantages: the effect of baffles on the optimum thermal and manufacturing costs in the case of shell-and-tube heat exchangers, and the comparison with measured values in case of finned tube heat exchangers.

Steel structures supporting the heat exchanger equipment can also be included in a broader sense in the system under consideration. For these compressed and bent and compressed beams, the required cross-section areas of structural steels of different strengths have been compared, based on different standard specifications. From the results of the optimization, conclusions have been drawn for the cases under consideration, which can provide a useful basis for design engineers.

# 5 *Publications related to the dissertation*


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