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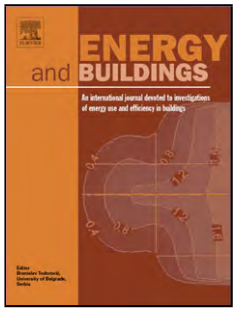

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Energy efficient heat exchanger for ventilation systems

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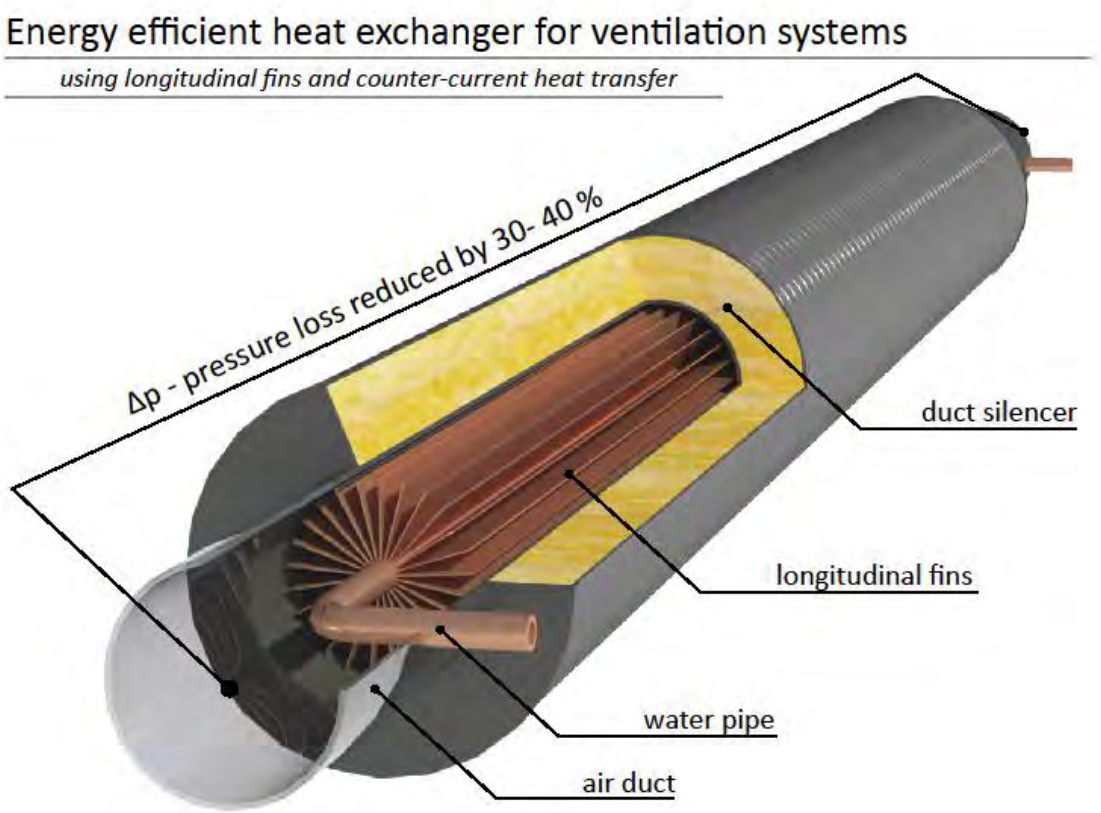
GRAPHICAL ABSTRACT



Highlights

• An air-water heat exchanger with longitudinal fins is presented to increase the energy efficiency of mechanical ventilation systems.
• Theoretical calculations regarding heat transfer and pressure loss are performed and the results are compared with measured data.
• Because of the longitudinal fins the pressure loss is reduced by 30% to 40% compared to a conventional heating coil.
• The counter-current heat transfer makes the heat exchanger more suitable for low temperature heating systems.
• By integrating the heat exchanger into an existing duct silencer it can be realised without additional technical space.

Abstract

The paper investigates whether air-water heat exchanger with longitudinal fins along the air flow direction are an appropriate means to reduce pressure loss in mechanical ventilation systems, compared to conventional heating coils.

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To answer this question, theoretical calculations regarding pressure loss and heat transfer are performed and the results are checked by measurements. It is found that a longitudinal fin heat exchanger with 30 radially arranged fins and an overall length of 830 mm transfers the same heat flow as a conventional heating coil (ø 125). The pressure loss only adds up to about 60 % to 70 %, so a reduction of 30% to 40% can be achieved. Additionally it is expected, that the longitudinal fin heat exchanger is more suitable for efficient low temperature heating systems, because of the counter-current heat transfer. The disadvantage of its increased length can be compensated by integrating the longitudinal fin heat exchanger in a duct silencer, which is required anyway. Hence, the energy efficiency of mechanical ventilation system can be increased.

Key words:

air-water heat exchanger; longitudinal fins; energy efficiency; mechanical ventilation system; reduced pressure loss; counter-current heat exchanger; low temperature heating system; duct silencer

2. Introduction

In today's mechanical ventilation systems heating-/cooling coils consist of pipes, through which heating-/cooling water flows orthogonally to the air flow direction (see Figure 1). The pipes are equipped with fins to increase the area for heat transfer and thereby the heat flow. The fin packages in Figure 1 is arranged in a single row. This case represents a cross-flow heat exchanger. In cases of multiple pipe layings a cross-counter heat exchanger occurs, depending on the hydraulic wiring [1].

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The advantage of this type of heating coil is its low space requirement, explaining why this technology has become established in practice. A disadvantage, however, is the short time period for the heat transfer. At an air flow speed of 2 m/s and a fin length of 4 cm the air is only in contact with the heating surface for 0.02 seconds. Hence, high fin densities and temperature differences are required. This leads to increased pressure loss on the air side and high system temperatures on the heating water side.

Figure 1: heating coil WHR 125 - Helios [Picture: own photo]

The high heating water temperatures needed by the conventional heating coils do not fit with the usual modern low-temperature heating systems. To achieve high energy efficiency for example condensing boilers need heating water temperatures around 55°C/40°C (radiators) and heat pumps around 35°C/28°C (floor heating). Today's conventional heating coils require high heating water temperature ranging from 60°C to 80°C. The highest temperature requirement however determines the heating water temperature supplied by the heat generator.

3. Goal

This paper analyses a heat exchanger with longitudinal fins (Figure 2, Figure 3). The fins (5) are arranged in the air duct (2) or a duct silencer (6) and the fins run parallel to the direction of the air flow (1). This way the heating water can flow through the heating pipe (4) in the opposite direction of the air flow (3), thus creating a counter-current heat exchanger. The counter-current principle advantages low heating water temperatures, and therefore fits better with low temperature heating systems.

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Additionally, it is suspected that the pressure loss on the air side is lower, under conditions of equal heat conduction, since the fins are further apart. So the longitudinal fin heat exchanger promises advantages in terms of efficiency on the air side (low pressure loss) and on the water side (low temperature heat pumps or condensing boilers usable). However, it will probably not be possible to optimize both advantages at the same time but they will have to be traded against each other.

The advantages in terms of the air heating are of course also present for air cooling. Here, higher cooling water temperatures lead to a more efficient performance through increased use of free cooling or higher efficiency of the cooling machine.

One disadvantage of the longitudinal fin heat exchanger is its increased overall length. The increased length is probably the reasons, why this concept isn´t known in mechanical ventilation systems up to know.

To compensate for the disadvantage of increased overall length it can be integrated in anyhow existing volumes like for example duct silencer (see Figure 3 and Figure 11). This constellation is appropriate for smaller ventilation systems for instance in residential buildings, or decentral heating coils in larger RLT-facilities of non-residential buildings. Preliminary sonic measurements show that the insertion loss of a duct silencer is not worsened by the longitudinal fins, but slightly improved. The integration in a rectangular duct silencer is shown in Figure 4.

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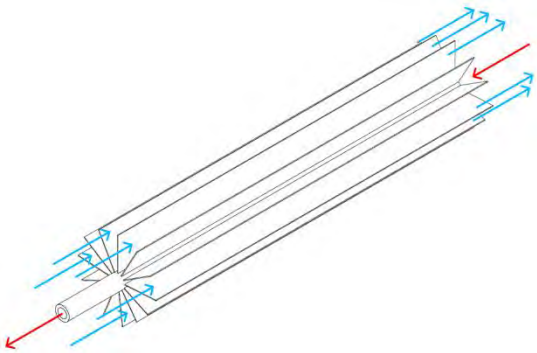

Figure 2: Longitudinal fin heat exchanger with radial fins

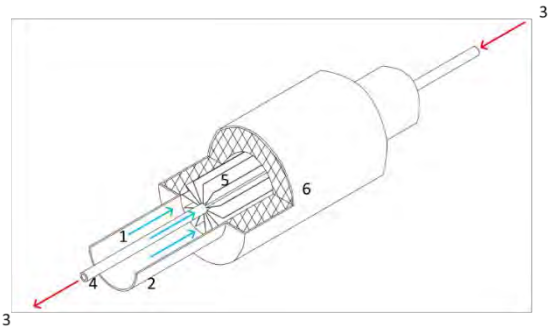

Figure 3: Longitudinal fin heat exchanger integrated into a duct silencer

Of course, the fins can also be arranged sideways (Figure 5) and then be integrated into a rectangular duct silencer (Figure 6) or a splitter attenuator (Figure 8). To improve heat transfer the fins may also be arranged spirally, arranged with some discontinuities, be shifted or be equipped with surface contours to disrupt the laminar sublayer.

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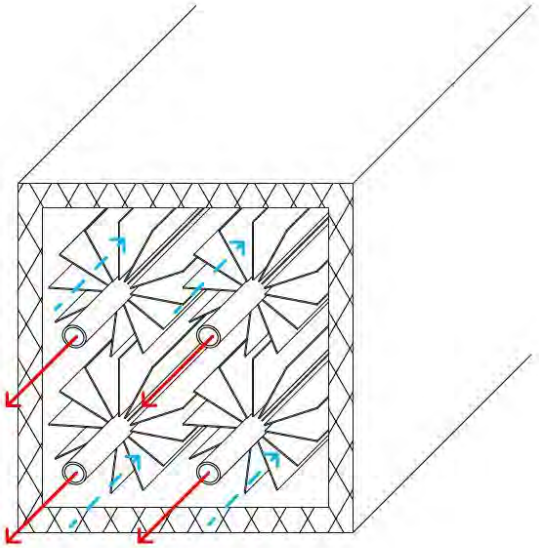

Figure 4: Rectangular duct silencer with radial fins

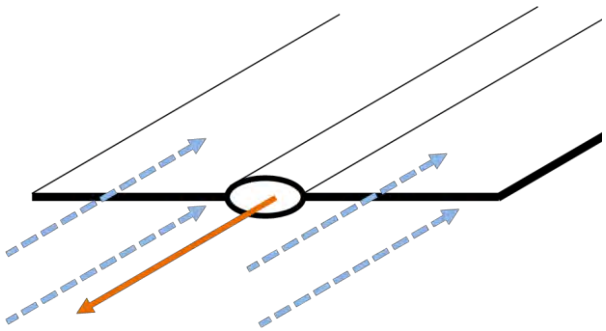

Figure 5: Longitudinal fin heat exchanger with sideways fins

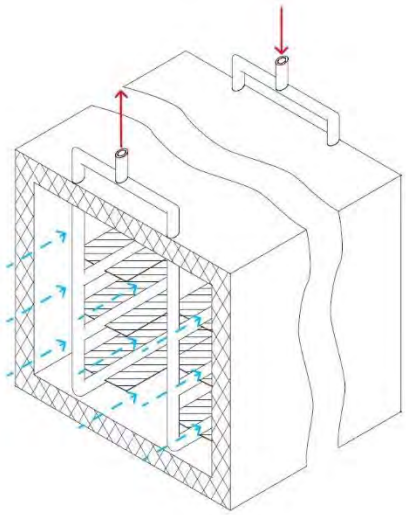

Figure 6 Duct sound damper with sideways fins

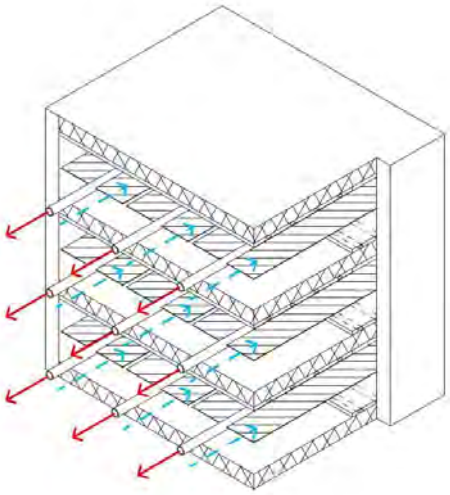

Figure 8: Splitter-attenuator with sideways fins

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4. State of the knowledge

The form of the longitudinal fin heat exchanger integrated into the duct silencer investigated in the following corresponds to a double pipe heat exchanger. For this heat exchanger type there are comprehensive studies on the calculation of the heat transfer and pressure loss as well as the optimization in particular of the heat transmission. Calculation approaches for a water-air heat exchanger with longitudinal fins on the air side can be found in [2], [3], [4] and [5]. Calculation equations for the Nusselt number and friction factor of plain double-pipe heat exchanger are used and the effect of longitudinal fins is considered using a modified equivalent diameter and the fin efficiency. Whereas [2] concentrates on the turbulent current, [3], in particular, adds an attempt to examine the Nusselt number and the friction factor for the transition flow. In [4] measurements are carried out using longitudinal fin double-pipe heat exchangers of brass and equations are drawn up for the Nusselt number and friction factor. These, however, are valid only for turbulent flow. [5] describes equations for the Nusselt number and the friction factor by evaluating 22 sources. On this basis he derives approaches for laminar, transition and turbulent flow. [5] bases his conclusions in particular on [6] and [7] to determine the Nusselt number. In [7] various sources are evaluated within the framework of a new revision of [8] and an equation is developed that includes both the transition and the turbulent flow.

Various studies deal with the optimization of the heat exchanger, in particular the heat transfer. [9] adds aluminium fins with a star-shaped cross-section to the inner pipe of a water-water heat exchanger. The heat transfer increases by 12-51% while the pressure drop increases by 290 – 400%. [10] try to enhance the heat transfer in the inner pipe with air as a working medium by adding free rotating propellers to destroy the boundary layer. Heat transfer was about 250% better but the pressure

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drop rose by 500% to 1000%, depending on the Reynolds number and the number of propellers.

The optimal shape and height of longitudinal fins are treated in [5]. A. E. Bergles shows in [11] the effect of structured and rough surface and interrupted, cut and/or twisted fins. According to this [12] conclude, that regular interruption of the fins are the best solution to improve the heat transfer friction performance in the laminar and transition current. Little effect is visible in turbulent current.

New investigations use numerical CFD simulations to calculate the heat transfer and pressure loss. [13] optimises the heat exchanger with radial find on a three dimensional numerical calculation. [14] deals with dimpled surface structures in heat generators and develops a correlation of Nu-number and friction factor to the Re-number using direct numerical simulations and design of experiments.

There are multiple applications for longitudinal heat exchanger in mechanical ventilations systems. Besides heating or cooling the supply air it can improve the efficiency of a circulation heat recovery system. This systems consist of two air-water heat exchanger and a secondary water circuit with a pump, which transports water to the hot to the cold airflow and vice versa. An advantage of this system is that heat recovery can be realized, even if the two airflows are located apart from each other. Besides that the two air flows stay strictly separated, which is important in case of contamination [15]. Innovative approaches use heat pumps [16] ore heat pipes instead of the secondary water circuit. Heat pipes were invented in 1942 [17] and for the first time used in space technology in 1963 [18]. Within the last years heat pipes become more and more popular. [19] gives a survey of the state of the art. Applications of heat pipes are for example industrial heating, cooling and heat-recovery processes, thermal solar collectors or electrical PV/T collectors. The use of

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heat pipes for heat recovery in mechanical ventilation systems is shown for example in [20]. [21] investigates thermosiphon heat pipes in an air handling unit. Experimental tests were carried out to determine the thermal resistance and efficiency of heat recovery while varying the heat load and the inclination angle of the heat pipes. Based on this mathematical correlations a performance model is developed. [22] studies the efficiency of a heat recovery system using two heat pipe systems and different operating conditions for summer and winter. Compared to a common heat pipe system 2,5% of energy can be saved in summer and 22,1% in Winter.

Up to now heat pipes are usually placed orthogonal to the air flow and are equipped with radial fins. It might be worth investigation, weather the longitudinal fin heat exchanger discussed in this article is able to improve the efficiency of heat pipes.

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5. Calculated Heat Flow and Pressure Loss

The following analyses are confined to the case of radial fins and the integration into a duct silencer (see Figure 3). To evaluate the advantage of the longitudinal fin heat exchanger theoretically, calculations following the methods of longitudinal finned double pipe heat exchanger are presented.

Based on analyses of chapter 4 the methods and equations of [5] are used to calculate the heat flow $\dot{Q}$ and of the pressure loss on the air side Δp_a. It covers all flow types (laminar, transition, turbulent) and is based on the results of different studies. The calculations are sketched in the following section, and we direct readers to [23] or to the relevant literature for the detailed equations used. In Figure 9 the geometric measures for a longitudinal fin heat exchanger in an air duct are presented.

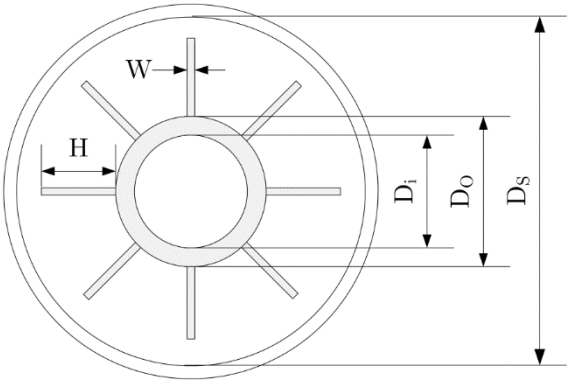

Figure 9: Dimensioning of the longitudinally fined heat exchanger in the air duct

The transferred heat flow is calculated according to [5] as:

$$\dot{Q} = U \cdot A \cdot \overline{\Delta T_{log}} \qquad \text{Eq. 1}$$

with

$\dot{Q}$	W	Heat flow water - air
U	W/(m²K)	Overall heat transfer coefficient water - air
A	m²	Heat transferring surface of the air face
$\overline{\Delta T_{log}}$	K	Logarithmic temperature difference water - air

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The logarithmic temperature difference takes in account the nonlinear change of the temperature via the length of the heat exchanger. The material and fluid data are defined at the mean temperature between inlet and outlet. This simplification is acceptable for this development state of the concept. The inexactness is less than 3% [23]. If necessary numerical simulation can be performed in the future.

The overall heat transfer coefficient U is calculated according to [5] as

$$U = \frac{1}{\frac{1}{\eta_t\,\alpha_a} + \left(\frac{1}{\alpha_w} + \frac{D_o \cdot \ln\left(\frac{D_o}{D_i}\right)}{2 \cdot \lambda_{wall}}\right) \cdot \frac{A}{\pi \cdot D_o \cdot L}} \qquad \textit{Eq. 2}$$

with

η_t	-	Total efficiency of the heat transfer (pipe + fins)
α_a	W/(m²K)	Heat transfer coefficient of the air face
α_W	W/(m²K)	Heat transfer coefficient on the water face
λ_{wall}	W/(m K)	Heat conductivity of the water pipe and the fins
L	m	Length of the finbed water pipe

The total efficiency η_t (pipe + fins) accounts for the part of the surface of the fins and the temperature decline due to the fin height compared with that of the water pipe. It is calculated according to [5] und [8] as follows.

$$\eta_t = 1 - \frac{A_f}{A} \cdot \left(1 - \eta_f\right) \qquad \textit{Eq. 3}$$

with

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$$\eta_f = \frac{tanh(m \cdot H)}{m \cdot H} \quad \textit{Eq. 4}$$

and

$$m = \sqrt{\frac{2 \cdot \alpha_a}{\lambda_{wall} \cdot W}} \quad \textit{Eq. 5}$$

A_f	m²	Surface area of the attached fins
η_f	-	Efficiency of the fins

The heat transfer coefficients $\alpha_a$and α_W are calculated from the particular Nusselt number Nu.

$$Nu = \frac{\alpha\, D_e}{\lambda} \quad \textit{Eq. 6}$$

$$\alpha = Nu \frac{\lambda}{D_e} \quad \textit{Eq. 7}$$

with

λ	W/(m K)	Heat conductivity
D_e	m²	Equivalent diameter

The Nusselt number is usually distinguished according to laminar and turbulent flow types, as well as a transition flow, in which mixed flow types exist. In [5] calculation approaches for the air face according to [6] and [7] are used. The Nusselt number of the water face is calculated according to [8]. The equations are not displayed, since this would exceed the scope of this paper and can be found in the literature referenced in this paper.

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The pressure loss for the side facing the air is calculated according to [5], [7] and [8]. The total pressure loss Δp_a is composed of multiple parts:

$$\Delta p_a = \Delta p_f + \Delta p_{f,ins} + \Delta p_{ins} \quad \text{Eq. 8}$$

$$\Delta p_a = \frac{\rho_a}{2} v_a^2 \left(4 f_f \frac{L}{D_h} + \zeta_{f,ins} + \zeta_{ins}\right) \quad \text{Eq. 9}$$

mit

Δp_a	Pa	Total pressure loss on the side facing the air
Δp_f	Pa	Pressure loss of the finbed pipe and the inner surface of the duct
$\Delta p_{f,ins}$	Pa	Pressure loss due to flow against the abutting fin face
Δp_{ins}	Pa	Pressure loss of built in pipes, which let water flow in and drain out
ρ_a	kg/m³	Density of air (temperature dependent)
v_a	m/s	Speed of air flow
f_f	-	Friction factor of the finbed pipe and the inner surface of the duct
$\zeta_{f,ins}$	-	Sum of the resistance coefficients for the incident flow of the abutting fin face
ζ_{ins}	-	Sum of the resistance coefficients for built in pipes, which let water flow in and drain out

The resistance coefficients $\zeta_{f,ins}$ and ζ_{ins} are obtained from tables and diagrams e.g. according to [16]. The friction factor f_f of the finbed pipe is calculated according to [7] for hydraulic smooth pipes as

Laminar flow

$Re_p < 4000$

$$f_f = \frac{16}{Re_p} \quad \text{Eq. 10}$$

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Transitional area

$4000 \leq Re_p \leq 10^5$

$$f_f = \frac{0.079}{Re_p^{0.25}} \quad \text{Eq. 11}$$

One must bear in mind that the Reynolds number Re_p is calculated using a different equivalent diameter from that used for heat transfer calculations because the inner surface of the duct is taken into account.

6. Measured Heat Flow and Pressure Loss

Heat flow and pressure loss are measured to check the calculations. Because of the available manufacturing possibilities at most 18 fins can be brazed onto the copper pipe. For this reason, two prototypes,

- one with 12 fins and
- one with 18 fins

are manufactured and examined (Figure 10). The longitudinal fin heat exchangers are integrated into a 90 cm long duct silencer with a diameter of DN 125 (Figure 11). Therefore the length of the prototypes comes out to be 83.5 cm. For reasons of manufacturing each prototype is composed of two segments, which are connected with a screw joint. The fins of both segments are aligned. The most important geometric data for the examined heat exchangers are displayed in Table 1. Because the fins are made of copper, the height can be chosen with 42 mm. This way they almost touch the inner surface of the duct silencer. The total efficiency of the finned pipe η_t results still in between 90 and 95% [23].

As a reference system, pressure loss and heat flow are measured for the conventional heating coil (see Figure 1).

In the experimental setup Figure 12 the values shown in Table 1 are measured.

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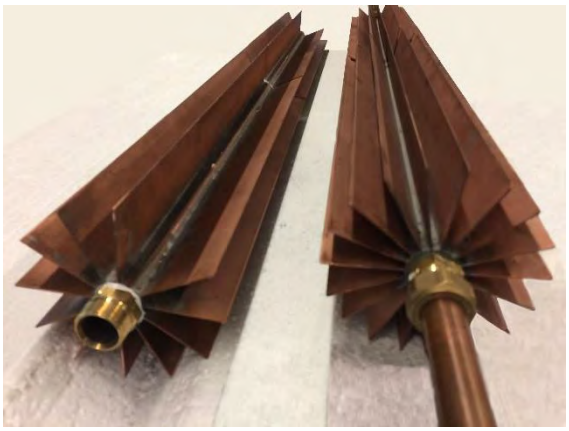

Figure 10: tested prototypes with 12 (left) and 18 fins (right)

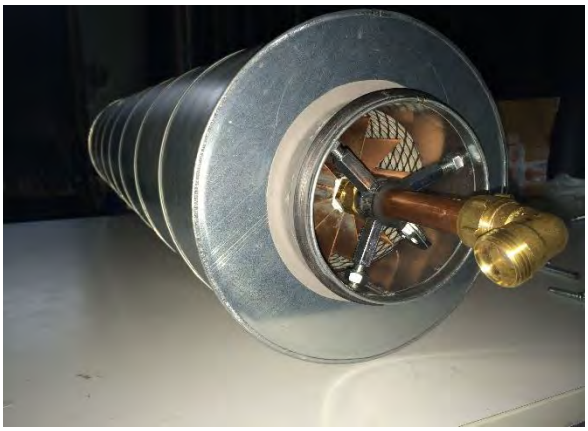

Figure 11: prototype integrated into duct silencer

Number of fins	Number	12 /18
Fin height	mm	42
Fin thickness	mm	1
Length of segment	mm	417.5 (2 segments)
Interior diameter of the water pipe	mm	16
Exterior diameter of the water pipe	mm	26
Heat transfer surface (air side)	m²	0.91/1.331
Interior diameter of the air duct	mm	125
Free flow area for air	m²	0.0112/0.011

Table 1: Geometric data of both test prototypes of the longitudinal fin heat exchanger

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$T_{w,in}$	°C	Inlet temperature of the water	PT 100	0.15 K + 0.002 x [t] Class A DIN EN 60751
$T_{w,out}$	°C	Outlet temperature of the water		
$T_{a,in}$	°C	Inlet temperature of the air		
$T_{a,out}$	°C	Outlet temperature of the air		
v_w	m/s	Flow speed of the water	Ultrasound	+/- 1.6 % v.MW.
$\dot{V}_{a,in}$	m³/h	Air flow (inlet temperature)	differential pressure sensor	2 % to 5 % v.MW.
Δp	Pa	Pressure loss of the heat exchanger	Pressure sensor	0.3 % to 4.5 % v. MW

Table 2: Measured values and measurement equipment

Figure 12: Placing the measuring sensors from Table 2 in the experimental setup

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7. Comparison of Calculations and Measurements

To examine the applicability of the equations for heat flow and pressure losses, the results of these calculations are compared to the measurements performed.

Of course the measurements are subject to uncertainty. The precision of measurements of individual quantities is given in Table 2. In addition, to make the measurements plausible, the heat flow emitted by the water side $\dot{Q}_w$ and the heat flow absorbed by the air side $\dot{Q}_a$ are compared. If measurements and material values are precisely, these should be identical in terms of their absolute values.

$$|\dot{Q}_a| = |\dot{Q}_w|$$

This condition is fairly well satisfied for the conventional heating coil and for the longitudinal fin heat exchanger with 18 fins. Deviations were found for 12 fins. It was found that a temperature profile had formed above the duct cross-section and that the sensor for outlet air temperature had not measured a representative temperature. This mistake was corrected.

Additionally it was taken into account that the longitudinal fin heat exchanger is composed of two individual segments. The segmentation leads to an improvement in heat transfer and, thereby, in increasing temperature, since the laminar boundary layer is disrupted by the gap.

The comparison of calculations and measurements for longitudinal fin heat exchanger with 12 and 18 fins is shown in Figure 13. Measurements are shown as dots. The error bars quantify the uncertainty of the measurements. For the pressure loss the error is too small to be shown. The calculations are completed for the same nodes and are shown as a line in the diagram for contrast. The boundary conditions for the comparison are also displayed in the figure.

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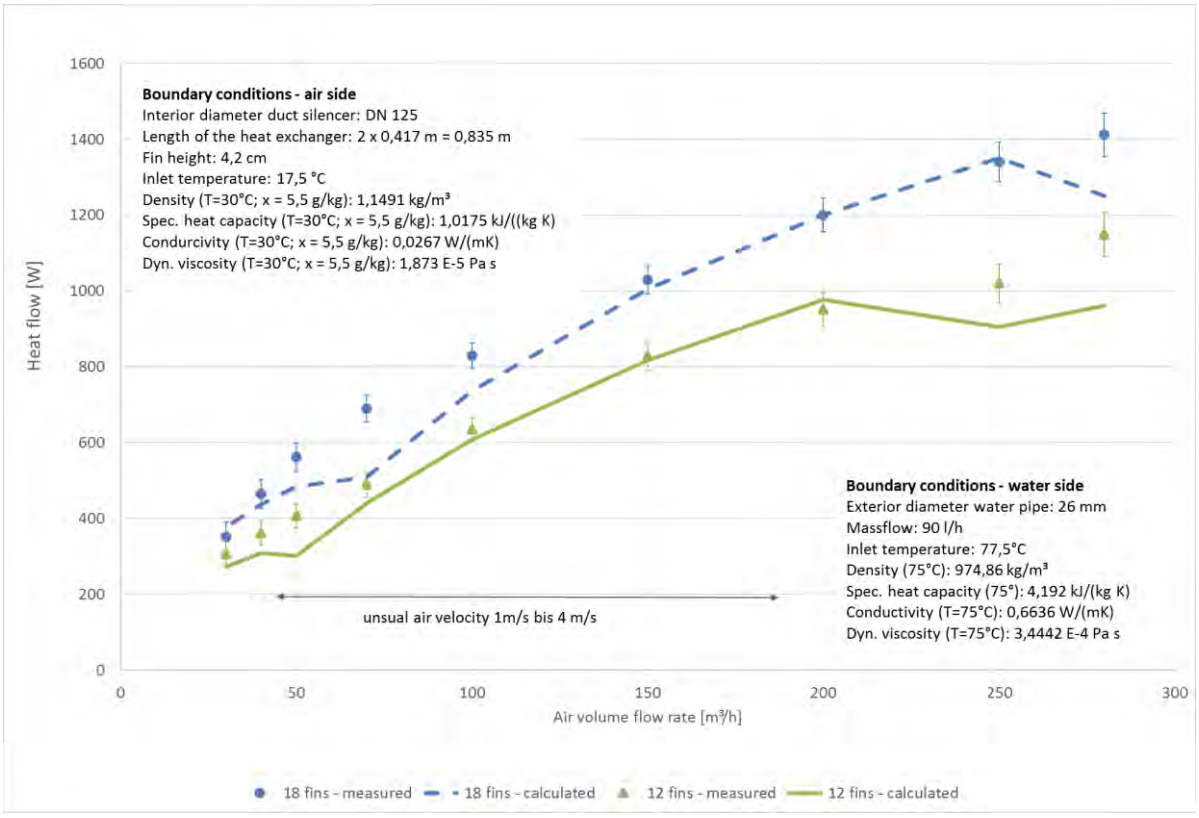

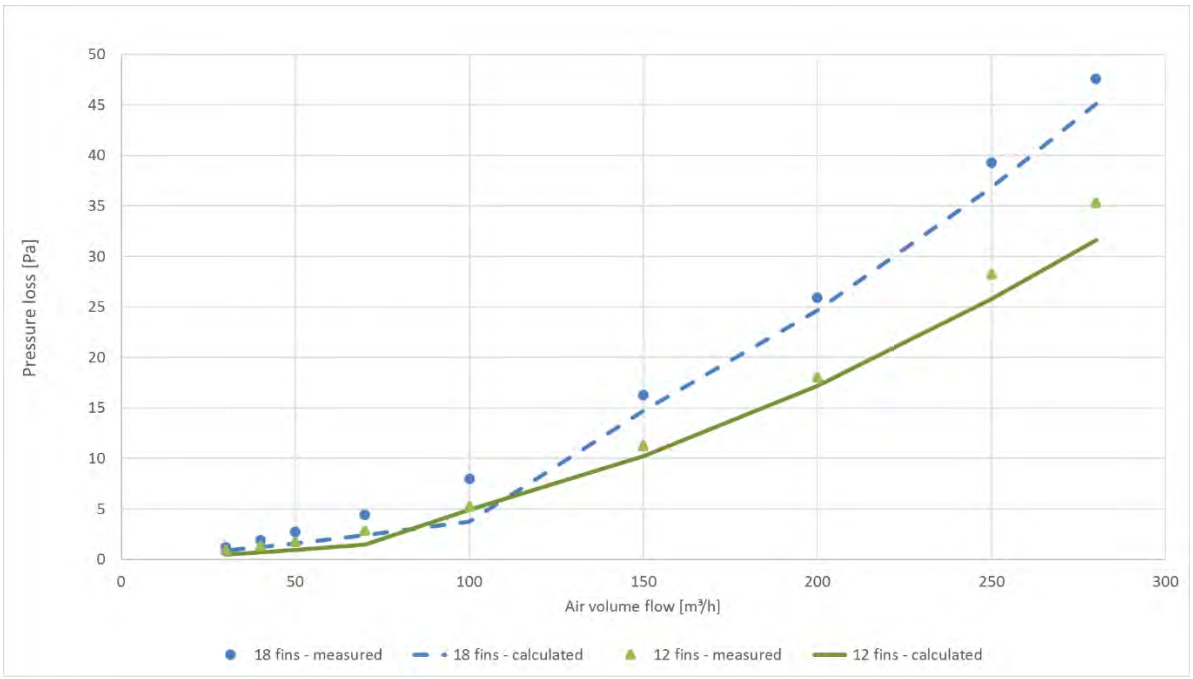

Figure 13: Comparison of calculated and measured values of the heat flow and the pressure loss depending on the air volume flow for the longitudinal heat exchanger with 12 and 18 fins

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The upper diagram displays the transferred heat flow, holding water mass flow constant varying the air volume flow. The transferred heat flow increases as the air volume flow and the number of fins increase. The curves of the calculations display a slope of varying steepness. This results from the different equations used as the flow type changes (laminar, transition, turbulent).

The measurements of the heat flow are quite well represented by the calculations. Differences occur in cases of high air flow and thereby turbulent current. Here, lower values are computed, resulting in a misalignment of the curve. For a transition flow type (18 fins: 70 m³/h to 250 m³/h; 12 fins: 50 m³/h to 200 m³/h) measurements in the upper area of volume flow are very well quantified by the calculation. In cases of low volume flow, the calculations are slightly beneath the measurements. For the laminar disruption range the calculations and measurements fit very well.

The lower figure shows the pressure loss on the air side. It also increases as air volume flow and the number of fins increase. The increase is exponential. The change in the equations used for the calculations also produces a discontinuous curve. Measurements and calculations fit well for the 12 fins. In the case of 18 fins there are some differences for the transition flow.

Overall, the investigations confirm the applicability of the equations used to calculate the longitudinal fin heat exchanger.

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8. Obtaining the reduced pressure losses

One expected advantage of the longitudinal fin heat exchanger is the reduced pressure loss, while keeping the transmitted heat flow equal. To quantify this, the pressure loss of the longitudinal fin heat exchanger is compared with those of a conventional heating coil. The compared heating coil is again the WHR 125 manufactured by HELIOS. The transferred heat flow and the pressure losses were obtained metrologically (see section 6).

To transfer the same heat flow to the air as the conventional heating coil the fin number of the longitudinal heat exchanger is increased to 30. Since no such prototype could be built, the results for evaluation are calculated. Figure 14 shows the calculated transmitted heat flow and the pressure loss of the longitudinal fin heat exchanger with 30 fins as the grey curve. The measurements of the conventional heating coil WHR 125 are shown as red dots.

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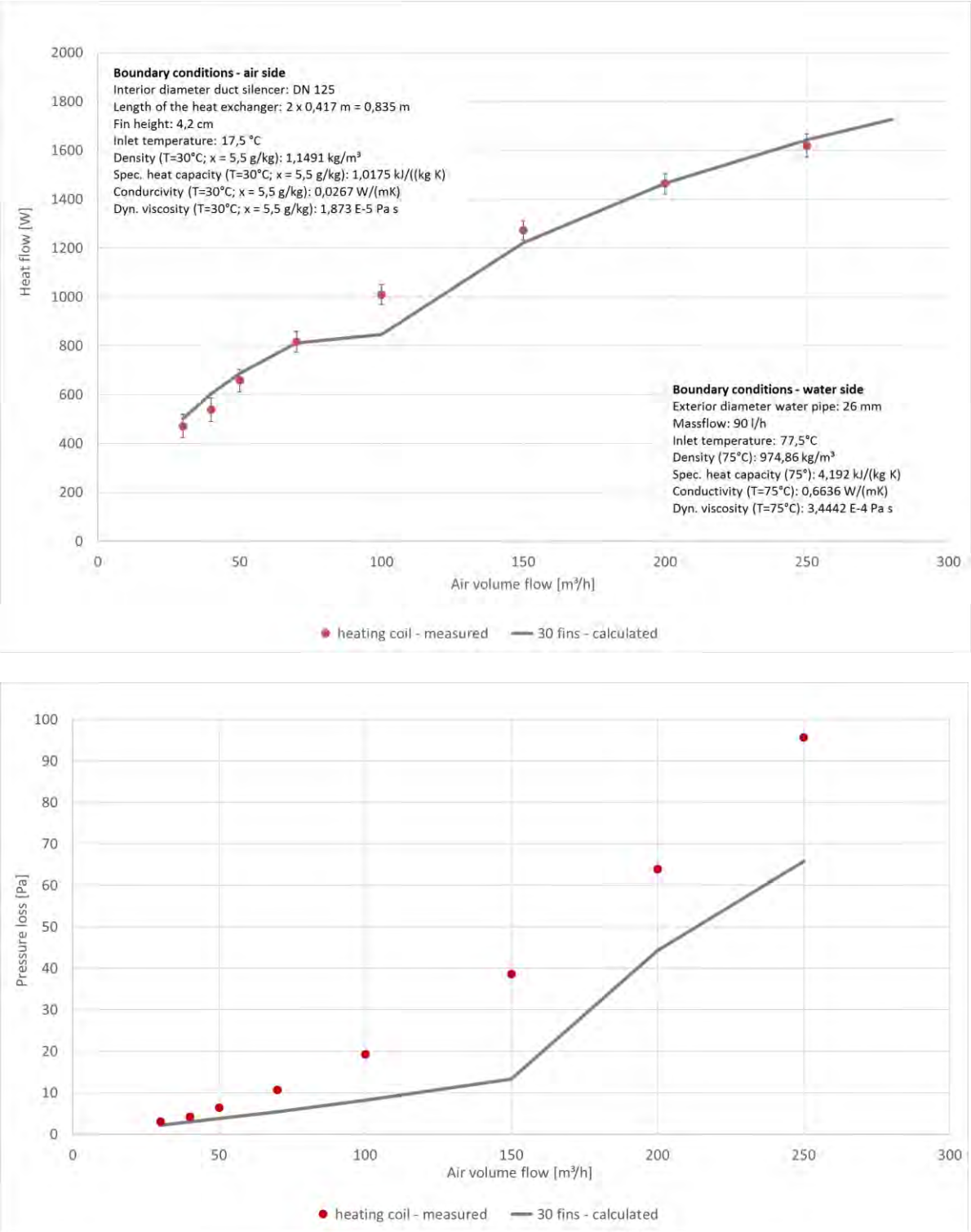

Figure 14: Comparison of the transmitted heat flow and the pressure loss for the longitudinal fin heat exchanger with 30 fins and the conventional heating coil

The upper figure shows that the transferred heat flows are about the same. The lower figure shows the comparison of the pressure losses. Here, the expected advantages of the longitudinal fin heating exchanger become apparent. Because calculations are compared to measurements, the diminished pressure losses cannot

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be quantified exactly. If one qualifies the high savings at 100 m³/h and 150 m³/h, the reduction in pressure loss lies between 30 and 40 %.

The reduction in pressure loss Δp is proportional to the reduction of the fan's electrical power, which is necessary to overflow the air heating coil.

$$P_{el} = \frac{\dot{V}\,\Delta p}{\eta_{tot}}$$

with

P_{el}	W	Electrical power of the fan unit
$\dot{V}$	m³/h	Air volume flow
Δp	Pa	Pressure loss
η_{tot}	-	Total efficiency of the fan unit

Thus, the use of the longitudinal fin heat exchanger can reduce electrical power and electrical energy consumption necessary to overflow the heat exchanger by around 30% to 40%.

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9. Conclusion and Outlook

The investigations show that the pressure loss of the longitudinal fin heat exchanger are reduced by 30 to 40%, with the same heat transfer. Therefore, the expectation of an energy efficient heat exchanger with low pressure loss are met.

The second expected advantage, namely the better integration into low temperature heating systems because of the counter-current flow type, has not yet been explored. Thus far, the results suggest, however, that the heat transfer must be improved for this. If the overall length is not increased, one must increase the heat transfer coefficient on the air side α_a. This can be done through an additional segmentation of the fin pipe, as this limits the spreading of the laminar boundary layer. Preliminary measures show a significant improvement.

The costs and the efficiency of a longitudinal fin heat exchanger compared to the conventional heating coil have not yet been examined. To answer this questions, additional specifications regarding the build and the manufacturing process are needed. Such specifications will be coordinated with manufacturers in the future.

Thus far, the results encourage additional investigations, since the longitudinal fin heating exchanger can improve the energy efficiency of mechanical ventilation systems.

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