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Bypass Design of Heat Exchangers in Air Handling Units as a Function of Expected Operating Hours

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Abstract
*One possible way to provide heating or cooling for buildings is to use **A**ir **H**andling **U**nits. An AHU is equipped with heat exchangers to transfer energy to the supply air. A common AHU has three heat exchangers in line (two for heating and one for cooling). While air is passing through the heat exchangers, pressure losses occur at each heat exchanger regardless of its operating state. In this study the optimized application of bypass for heat exchangers in air handling units is shown to be energetically and economically profitable. Therefore, heat exchangers with and without bypass were investigated by means of experiments and simulations. To minimize the overall energy consumption the size of the bypass is optimized regarding the operating hours. The results show an annual energy-saving potential up to 41 % for a bypass solution.*

Keywords - energy efficiency; heat exchanger; simulation; bypass

1. Introduction

According to Pérez-Lombard [1], in 2004, energy consumption of buildings represented 37 % of the total final energy consumption of the EU. The energy consumption of the building sector was larger than the industry (28 %) and transport (32 %) sectors. Referred to Roulet et al. [2], more than 50 % of the total energy losses in buildings can be due to ventilation losses.

In buildings with high internal loads (thermal or moisture) the infiltration rate is not enough to provide acceptable indoor air quality. Therefore mechanical ventilation (AHU) is used to ensure a sufficient indoor air quality. The AHU is an integrated piece of equipment consisting of fans, heating and cooling heat exchangers, sound attenuators and filters. This paper will focus on the heating and cooling heat exchanger (pre heater, cooler, re-heater) shown in Figure.1.

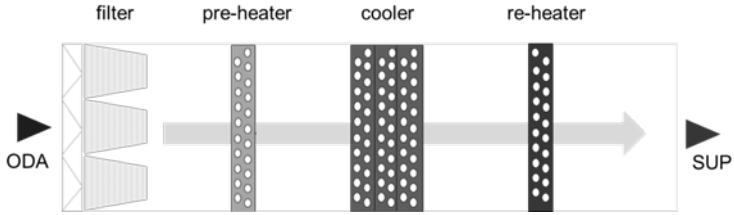

Figure. 1: AHU with heat exchangers
(ODA = outdoor air, SUP = supply air)

While air is passing through the heat exchangers, pressure losses occur at each heat exchanger regardless of its operating state. However, there are many periods throughout the year where one or more heat exchangers are not required. During these periods it would be advantageous to bypass a part of the air around the unoperated heat exchangers so that the air pressure drop is reduced. To reduce the overall energy consumption the size of the bypass needs to be optimized regarding the operating hours of the heat exchanger.

2. Methods

When designing a heat exchanger unit with a bypass, the cross-sectional area of the heat exchanger itself is reduced and accordingly air velocity and pressure drop increase compared to a standard heat exchanger (Figure 3). In case of an open bypass, the pressure drop significantly decreases compared to standard solutions, because the air will flow through the bypass and heat exchanger simultaneously, which results in a large cross-sectional area and low velocity. To preserve the thermal power of the heat exchanger it has to grow in depth and needs to be reconfigured in terms of tube number and fin size. To optimize the overall package a new tool is developed, which considers the thermal and fluid dynamic properties of a variably configurable heat exchanger. Furthermore, the tool provides information about the operating hours of all heat exchangers in an AHU based on weather data and expected usage profile of the supplied rooms. Based on these results, a heat exchanger with an economically sized bypass is calculated according to the expected operating hours.

2.1 Heat Exchanger Tool

In order to study the physical behaviour of the heat exchanger a general model of a fin-tube heat exchanger (Figure 2) is built and analyzed. There are many methods for designing heat exchangers. They differ from one another in their field of application, their physical and mathematical complexity and their accuracy.

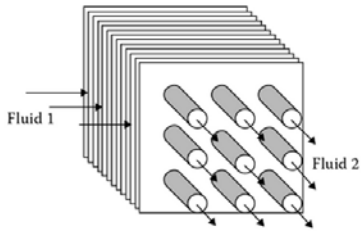

Figure. 2: Example of a cross flow heat exchanger with tubes and fins

In this paper the heat exchanger is modeled according to VDI Heat Atlas [3] part C3, D, G1, L1 and M1. In addition to the idealizations of the flow arrangements, the following simplifying assumptions are made according to [3]:

- The heat exchanger is operated in steady state
- Heat losses to the surroundings, kinetic and potential energies are ignored
- If no phase change occurs, the specific heat capacities are set constant
- The effects of conduction and mixing in the direction of flow are ignored

For the comparative study, the experimental heat transfer rates and pressure drop of 8 row and 4 row heat exchanger reported by [5] are compared with the present model. The presented heat exchanger model is adjusted to obtain both dry and wet heat exchanger conditions. A comparison of the study has been published in [6].

For the determination of the pressure drop Δp_{HE} the following equation applies to all kinds of flow patterns (laminar or turbulent):

$$\Delta p_{HE} = f \cdot N_{Tube} \cdot \frac{\rho_A \cdot v_A^2}{2} \tag{1}$$

with f as the drag coefficient (depending on the nature of the flow problem) and N_{Tube} as the number of tubes in air direction. ρ_A stands for the average density of the air. v_A corresponds to the maximum air velocity in the heat exchanger. The parallel pressure drop Δp_p across the heat exchanger and bypass can be calculated from:

$$\Delta p_p = R_T \cdot \dot{V}^2 \tag{2}$$

with R_T as the total resistance coefficient; it is defined by R_{HE} (drag coefficient for heat exchanger) and R_B (drag coefficient for bypass). $\dot{V}$ is the air volume flow.

$$\frac{1}{\sqrt{R_T}} = \frac{1}{\sqrt{R_{HE}}} + \frac{1}{\sqrt{R_B}} \tag{3}$$

With the knowledge of pressure drop for each component Δp_{HE} and Δp_B the drag coefficient R_{HE} and R_B can be calculated.

$$R_{HE} = \frac{\Delta p_{HE}}{v^2} \tag{4}$$

$$R_B = \frac{\Delta p_B}{v^2} \tag{5}$$

2.2 Experimental Investigation

The aim of the experimental investigation is to highlight the influence of a bypass on air pressure drop and to evaluate the heat exchanger model. Therefore, measurements were conducted on a test bench. A schematic representation of the test bench is given in Figure. 3. The heat exchangers are located in a box insulated by 30 mm thick insulating material in order to reduce the impact by ambient surrounding. To ensure a uniform air flow rate through the heat exchanger, dampers with filters are placed in the box upstream of the heat exchanger. With regards to the hydraulic performance, the pressure drop was determined for several air flow rates (from 8.000 to 12.000 m^3/h). The conditions at the inlet of the heat exchanger were approximately 20 °C with 55 % of relative humidity. Each presented result of the several air flow rates is an averaged number corresponding to a quasi-steady state test of 500 s. Accuracies of the measurement devices are summarized in Table 1.

Table 1: Accuracy of the measurement devices

Measurement	Accuracy
Temperature	± 0.3 K
Differential pressure (0-100 Pa)	± 2 Pa
Relative humidity sensor (0-100 %)	± 4 %

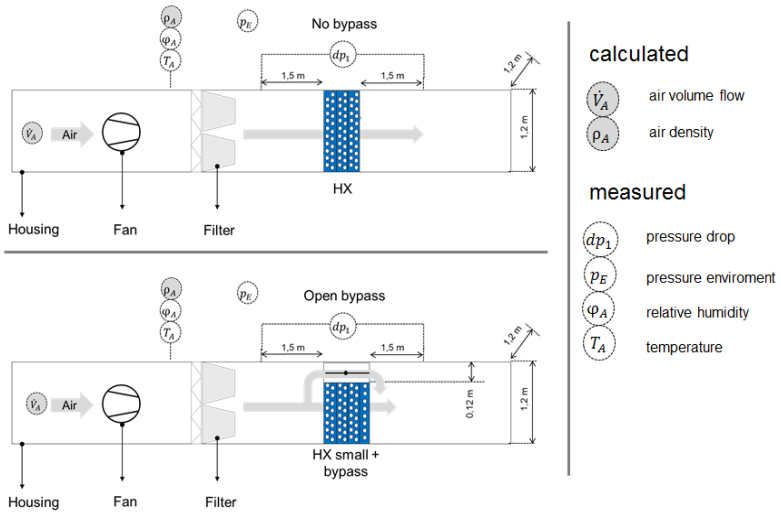

Figure. 3: Schematic test bench with (bottom) and without (top) bypass

The comparison between the predictions by the model and the measurements in terms of hydraulic performance is displayed in Figure. 4. It shows the calculated pressure drop is always larger than the measured one. The mean deviation between the prediction by the model and the measurement for each component is less than 10 %, but the maximum deviation for the combination (heat exchanger and bypass) is up to 80 %. In the non-calibrated model the pressure drop is overestimated, because the applied correlations (Equations 2-5) are just valid for optimal parallel flow. In the studied case the bypass is placed over the heat exchanger. In addition to that the flow areas of the heat exchanger and bypass are of different dimensions. Therefore the correlation for the determination of combined pressure drop is adjusted. The tuning is achieved by multiplying a correction factor which itself is a function of air velocity. The model with new correlations predicts the pressure drop with a mean relative error of 15 %, which is of the order of the experimental uncertainty.

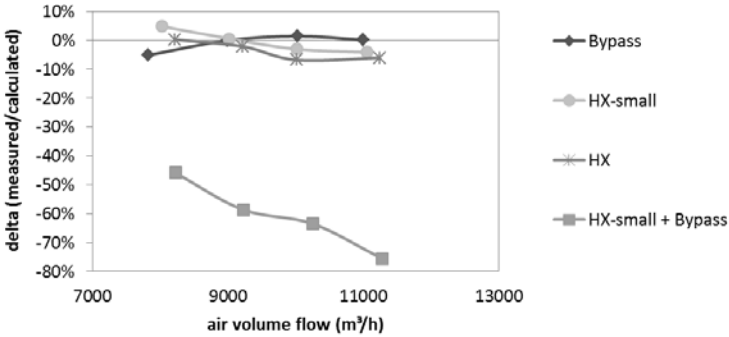

Figure. 4: Relative deviation between the predictions by the model and the measurements

2.3 Weather Data and Predicted Usage

To find the optimal partitioning ratio for the cross sectional areas of the bypass and the heat exchanger it is necessary to calculate the expected operating hours for every operational state (e.g. heating, cooling, humidification). Therefore a software-tool was developed that calculates the operational state and thermal and (de)humidification demands for every 8760 hours of a year.

The tool is designed according to the method of classifying the h,x-diagram presented by Reichert [4]. Reichert defines several classes of outdoor air conditions which all lead to similar air handling processes – for example cold dry outdoor air in winter will generally be processed by the pre-heater, the heat recovery unit, perhaps humidification and re-heating as required by adiabatic humidification.

Base data for these calculations are the Test Reference Years (TRY) for 15 regions in Germany from DWD (Deutscher Wetter Dienst, Germany's National Meteorological Service). These data sets describe typical weather data with an hourly resolution. Using outdoor air temperature, moisture content and ambient pressure the thermodynamic properties of the air are sufficiently defined. Furthermore, a projected usage profile for the air handling unit is defined, covering every hour of the year.

In a first step, the indoor air quality (comfort range) needs to be described via minimum and maximum temperatures and moisture content or relative humidity within the occupied space. Using the internal loads from the usage profile the necessary supply air conditions are calculated and then the most energy efficient way of process is used to get from outdoor air (ODA) to supply air conditions (SUP).

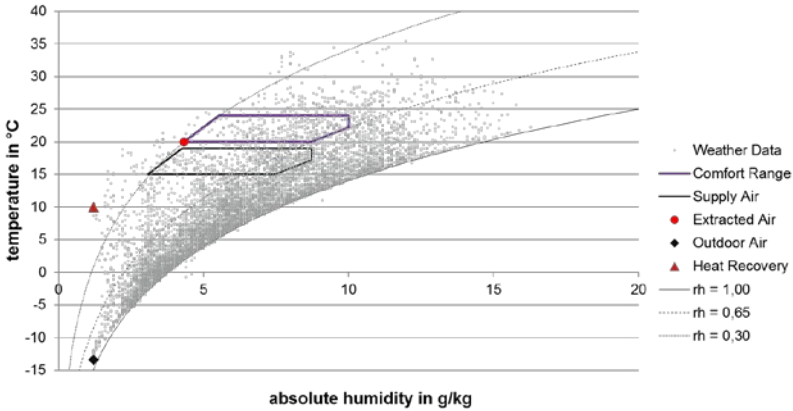

Fig. 5: h,x-diagram showing a weather data set and process steps

Assuming perfectly mixed room air and identical supply and extract air mass flow, the room air conditions are defined equal to extract air (ETA) conditions. Thus, the supply air flow needs to exactly compensate the internal loads and can be calculated via mass and energy balance equations – using air mass flow $\dot{m}_{air}$, temperatures t, thermal load $\dot{Q}_{internal}$ and moisture release $\dot{m}_{steam,internal}$, moisture content x in g_{water}/kg_{air} and mass-specific heat capacity of air $c_{p,air}$.

$$t_{Sup} = t_{ETA} - \frac{\dot{Q}_{internal}}{\dot{m}_{air} \cdot cp_{air}} \tag{6}$$

$$x_{Sup} = x_{ETA} - \frac{\dot{m}_{steam,internal}}{\dot{m}_{air}} \tag{7}$$

Now, with defined supply air conditions it can easily be concluded from the diagram which components are in operation for air conditioning. Fig. 5 shows a weather data set with highlighted process conditions for one time step. Starting at ODA the air temperature reaches about 10 °C after heat recovery, then needs to be further heated and humidified to match minimum SUP conditions and is extracted from the room at minimum comfort condition.

3. Results

Based on the new calculation tool presented and validated in the previous sections, the air pressure drop is computed for three heat

exchangers. The Air Handling Unit, containing the three heat exchangers, is placed in Potsdam/Germany. The internal moisture load is assumed to be 1.27 g/kg and the thermal load is set to a temperature rise of 5 K. Table 1 shows the inputs and results of all three heat exchangers with and without bypass.

For the investigation an air flow rate of 10.000 m³/h is set. This means a velocity of 2 m/s in the Air handling Unit. The width of all heat exchangers remains constant at 1045 mm. Most Air Handling Units operates only during the week and in one shift; therefore an annual operating time of 3.640 hours (14 hours x 5 days x 52 weeks) is determined. The tube diameter (12.5 mm), tube wall thickness (0.35 mm) and fin thickness (0.12 mm) are equal for all three heat exchangers. Due to the danger of frost a fin distance of 4 mm is chosen for the pre-heater. For cooler and re-heater a standard fin distance of 2.5 mm is assumed. The pre-heater is filled with 30 % antifreeze to ensure that the water does not freeze in winter.

Table 1: Results of heat exchangers with and without bypass

		pre-heater		cooler		re-heater	
		no bypass	with bypass	no bypass	with bypass	no bypass	with bypass
Inputs	air volume flow (m³/h)	10.000					
	width heat exchanger (mm)	1.045					
	annual operation hours air handling unit (h)	3.640					
	tube diameter (mm)	12.5					
	tube thickness (mm)	0.35					
	fin thickness (mm)	0.12					
	fin distance (mm)	4		2,5			
	anti freeze (%)	30		-			
Results	max. thermal power req. (kW)	70		112		19	
	annual operation hours heat exchanger (h)	2.288		1.029		1.608	
	tube rows in depth (-)	3	3	6	7	1	1
	heat transfer coefficient (W/m²/K)	56	57	64	67	49	50
	optimal bypass ratio (%)	-	2,6	-	10,3	-	2,6
	air pressure drop (Pa)	31	28	104	62	16	15

The pre-heater has a maximum thermal power of 75 kW with an annual operating time of 2.288 hours. The number of tubes rows (without bypass) is

3, heat transfer coefficient of 56 W/(m²K) and a pressure drop of 31 Pa. Due to the relatively high annual operating time for the pre-heater only a small bypass (optimal bypass ratio 2.6 %) is calculated. This result shows that pressure drop reduction for the pre-heater and thus the energy saving is negligible (from 31 Pa to 28 Pa).

For a cooler a maximal thermal power of 112 kW is calculated. Due to the small number of hot days for the selected region the cooler only operates for 1.029 hours annually. The number of tubes rows without bypass is 6 and the heat transfer coefficient 64 W/(m²K). The low number of operation hours results in an optimal bypass ratio of 10.3 %. The maximal decrease of the air pressure drop for the cooler is up to 41 % (from 104 Pa to 62 Pa). This would mean an electric energy saving of the same amount. To provide the cooling power the cooler has to grow in depth from 6 to 7 rows. Due to the smaller inflow area of the cooler the air velocity is increased (closed bypass) and with that the pressure drop. But on the other hand the higher air velocity results in increased heat transfer coefficient from 64 to 67 W/(m²K) this in turn has a positive effect.

Although the re-heater is 1.608 hours in operation, that means 44 % of total operation time, but the energy saving potential is due to the small row number (1) very low. For the re-heater an optimal bypass ratio of 2.6 % is calculated. This leads to a reduction of pressure loss from 16 to 15 Pa and thus is negligible too.

Table 1 also reveals the fact that the cooler has the highest energy saving potential and a bypass for the other two heat exchangers (pre-heater and re-heater) doesn't provide significant advantages. However, the optimal bypass size depends on the climate region, operating time and the internal loads. To highlight this fact in a real case, the new tool can be used to determine the optimal bypass size of each heat exchanger by an hourly-based simulation for 15 regions in Germany.

4. Conclusions

The aim of this paper was to investigate heat exchangers through modeling and experimental approaches. First, a model was presented using correlations already existing in the literature [3], in order to determine the hydraulic and the thermal performance of the heat exchanger. Secondly, the developed experimental apparatus, the testing conditions and the measured performance were described. Hereafter, a comparison (air side pressure drop) between prediction by the model and measurements was conducted. Although the results were of the same order of magnitude in terms of each component they differ extremely for the combined model (heat exchanger and bypass). Hence an adjusted correlation for the pressure drop was used to increase the accuracy of the model. Thirdly, a model based on [4] was applied to describe the annual operation hours of each heat exchanger.

Finally, a parametric study was carried out to investigate the influence of pressure drop of each heat exchanger by determining the optimal bypass. This is practically interesting in the design of air handling units with heat exchangers. Resulting from these outcomes, it is most likely that heat exchangers with a bypass would be of advantage to most systems, even though additional costs for the bypass hardware have to be taken into account. However, regarding todays technical guidelines, the application of a bypass is always a disadvantage. For example, in German regulations [7] the energy efficiency is rated at the design point considering the maximum air flow resistance. Subsequently, the SFP value (specific fan power in Ws/m^3) is worse than for conventional heat exchangers. At this point the question arises whether the current legal requirements need to be reconsidered, as a higher-grade calculation method evidently shows annual energy savings.

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