Publication VII

L. Sivill, P. Ahtila, and M. Taimisto. 2005. Thermodynamic simulation of dryer section heat recovery in paper machines. Applied Thermal Engineering, volume 25, numbers 8-9, pages 1273-1292.

© 2004 Elsevier

Reprinted with permission from Elsevier.

Available online at www.sciencedirect.com
SCIENCE DIRECT®

Applied Thermal Engineering 25 (2005) 1273–1292

APPLIED THERMAL ENGINEERING

www.elsevier.com/locate/apthermeng

Thermodynamic simulation of dryer section heat recovery in paper machines

L. Sivill [a,*], P. Ahtila [a], M. Taimisto [b]

[a] *Laboratory of Energy Economics and Power Plant Engineering, Helsinki University of Technology, P.O. Box 4400, 02015 HUT, Finland*

[b] *UPM-Kymmene Oyj, Jämsänkoski and Kaipola Mills, P.O. Box 35, 42301 Jämsänkoski, Finland*

Received 18 December 2003; accepted 10 September 2004

Abstract

Modern paper machines are equipped with heat recovery systems that are able to recover over 50% of the energy used by the paper machine. The recovered heat is used for the heating of dryer section supply air, process water and machine hall ventilation. Today, the structure of new heat recovery installations is optimised with sophisticated algorithms to match the design point of the paper machine. However, after installation, the real operation point may change over time from the original design point. To study the effect of these changes and to improve the performance of heat recovery, a simulation program was developed based on thermodynamic modelling. In this paper, the simulation program is compared with measurements from three paper machines. Furthermore, practical guidelines are presented for the improvement of dryer section heat recovery based on conclusions from simulated examples.
© 2004 Elsevier Ltd. All rights reserved.

Keywords: Heat recovery; Simulation; Paper machine; Dryer section; Heat exchanger; Condensation

* Corresponding author. Tel.: +358 9 451 3613; fax: +358 9 451 3986.
E-mail address: leena.sivill@hut.fi (L. Sivill).

1359-4311/$ - see front matter © 2004 Elsevier Ltd. All rights reserved.
doi:10.1016/j.applthermaleng.2004.09.002

Nomenclature

A	heat transfer surface area (m^2)
h	convective heat transfer coefficient (W/m^2K)
l	latent heat (J/kg)
$\dot{m}''_c$	mass flux of condensation ($kg/s\,m^2$)
Q	heat transfer rate (W)
s	thickness of heat exchanger material (m)
T	temperature (K)

Greek letters

λ	thermal conductivity (W/m K)

Subscripts

1	exhaust air
2	stream to be heated
s	surface on exhaust air side

1. Introduction

The heat recovery system of a paper machine is basically a heat exchanger network that conveys energy from the humid exhaust air of the paper machine dryer section to different process streams (Fig. 1). The humid exhaust air from the dryer hood is first led to conventional heat recovery

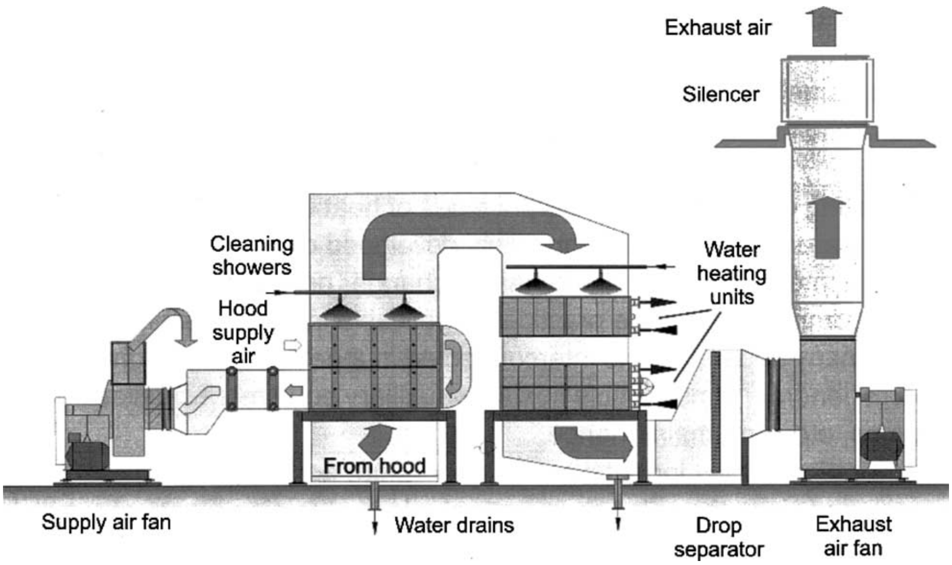


Fig. 1. Modern heat recovery system of paper machine dryer section [1].

(CHR) units, which recover heat to the dry supply air going into the hood. After this, heat is recovered in aqua heat recovery (AHR) units to the circulation water of machine hall ventilation, process water, white water and/or wire pit water, depending on the structure of the heat recovery system.

Hence the fundamental purpose of the dryer section heat recovery is to return part of the energy used by the paper machine back into use in a profitable way. In a modern wide paper machine producing 300 000 t/a, the amount of recovered heat may exceed 20 MW in cold periods, depending on the operating conditions of the paper machine [1]. This corresponds to over 50% of the primary energy brought to the paper machine. Consequently, the efficiency of heat recovery has a significant impact on the whole economy of papermaking. For example, in Finland, the energy costs constitute roughly 10% of the total costs in the pulp and paper industry [2].

For this reason, the optimal design of paper machine heat recovery systems has been under intensive investigation for decades. The first computerised design application developed in the 1980s used a sequential modular approach, in which an iterative cost minimisation routine was connected to the thermodynamic models of the heat exchangers in heat recovery [3]. After this, the time required for processing was reduced by creating simplified regression models for the heat recovery system in the optimisation procedure [4,5]. Maltais [6] has also studied the economic structure of heat recovery installations; however, the heat recovery models he used for simulation were based on gross estimates and experiments, rather than on thermodynamic theory.

In the 1990s, a completely new approach was taken towards the optimal structure of heat recovery as methods varying from pinch analyses [7] to mathematical programming [8] were introduced. The efficient use of these methods required taking a stand on the non-linearity of the optimisation routine, since the heat transfer in heat recovery is very non-linear due to condensation. At present, a new successful heat recovery optimisation method combines mixed integer linear programming with heuristic knowledge and elements similar to pinch analysis [8,9]. Using this method, Söderman and Pettersson [10,11] have recently analysed the sensitivity of the optimal structure of heat recovery systems to different cost factors.

In spite of the advances in the optimisation of new heat recovery installations, paper machines have constantly changing operating conditions; as the machines go through several renovations during their life cycle, the original design parameters may only be appropriate for a short period of time. In addition, the developed methodologies for the optimisation of new heat recovery installations are not particularly compatible for retrofitting because the benefit from making changes depends on the structure and operating conditions of the existing system in question. To be more precise, the optimisation programs for the design of heat recovery systems are based on user-defined cost functions; providing these in a retrofit situation is a challenging task due to restrictions on available space, physical distances, demolition costs, etc. Another problem is that the structure of heat recovery is optimised using user-defined process parameters, which does not really answer the question of what the profitability of changing the process parameters would be. For these reasons, a simulation program was developed, which enables the examination of the performance of an existing heat recovery system in different operation points and with a modified structure [12]. The simulation application does not contain any elements for optimisation and is intended for thermodynamic analyses only.

1.1. Simulation application

The simulation program used for this paper is based on the thermodynamic steady state modelling of the two types of heat exchangers found in modern paper machines. By connecting the individual heat exchanger modules in the desired way, and by feeding the rest of the required input information into the simulation program, the behaviour of an existing heat recovery system can be investigated. The developed simulation application with its detailed description and source code created with Matlab version 6 are given in [12]. The flow charts of the heat exchanger programs are presented in Appendices A and B.

The heat and mass transfer equations used in the simulation application are commonly available in the literature. For example, Soininen [13] has presented an excellent paper on the thermodynamics of dryer section heat recovery. In brief, the surfaces of the heat exchangers are divided into a number of small regions. In the centre point of each region, the program calculates the local conditions and then determines the heat transfer rate. Under condensing conditions when the surface temperature T_s on the exhaust air side is below the dew point of exhaust air, the heat transfer rate Q is calculated using the following relation:

$$Q = h_1 A_1 (T_1 - T_s) + \dot{m}''_c A_1 l(T_s) = \frac{A_2}{\frac{s}{\lambda} + \frac{1}{h_2}} (T_s - T_2). \tag{1}$$

In the equation, the first term on the right hand side refers to convection from exhaust air to the heat transfer surface and the second term to the latent heat released due to condensation. The mass flow rate of vapour condensing on the heat transfer surface, $\dot{m}''_c$, is solved separately with mass transfer functions. The sum of the first and the second terms must be equal to the rate of heat transfer from the heat transfer surface to the medium to be heated. A term for conduction in water is not included because it is assumed that the thickness of the condensate layer formed on the exhaust air side is negligible, as in Soininen's calculations [13]. In regions where condensation does not occur, the term for latent heat in Equation 1 is simply ignored.

1.2. Aims of the paper

In this paper, it is argued that in operating paper machines there is a major opportunity for energy savings in their heat recovery systems and that this opportunity can be used to best effect by making changes to the operation point and/or the structure of heat recovery, depending on the case in question. To prove these arguments conclusively, one first needs to introduce a workable tool which enables heat recovery simulation. Secondly, one should investigate how the different parameters, which must be changeable in a retrofit case, affect the performance of heat recovery. Finally, case studies should be carried out in order to prove the existence of the energy saving opportunities and to demonstrate the magnitude of these energy saving opportunities in real surroundings.

This paper serves the first and the second phases of the argument justification. Results calculated with heat recovery simulation are compared with measurements from three operating paper machines to demonstrate the suitability of the developed simulation program for the simulation of real processes. After this, general guidelines for the improvement of dryer section heat recovery in operating paper machines are presented based on conclusions from simulated examples. The case study results will be published later as the third phase.

2. Simulation results compared to measurements from three paper machines

The three paper machines selected for investigation are located in Kaipola, Finland. Paper machine PM4 produces directory papers 140 000 t/a, PM6 light weight coated paper 270 000 t/a and PM7 newsprint paper 250 000 t/a. Taking field measurements with portable equipment was necessary since most of the required information was not readily available from the mill's automation system. The measurements were made on four selected days during February–April 2001.

The results of the measurements are compared with the corresponding simulation results in Tables 1 and 2. In Table 1, the difference between the actual heat transfer rate and the corresponding simulated values is in the range of 10%. In Table 2, however, the deviation is much larger because the measurements with a limited number of portable equipment could not be taken exactly at the same time. For the same reason, representative long-term averages could not be produced for all the variables. The most important variable affected by this is the inlet temperature of process water, which can rapidly change according to the flow rate of cold additional water that is controlled by the water level in the process water tank. As a result, field measurements taken while the automatic control system is operating distort the results.

Consequently, this is why the validation of the simulation application relied completely on the manufacturer's values shown in [12]. Another reason for using only the manufacturer's values for validation was the expected insufficient measuring accuracy, ±10%, reported by Sundqvist [3]. This size of error was similarly reported in Kaipola for the assessment of air flow rates.

3. Simulated performance of dryer section heat recovery

In the following, heat transfer in heat recovery is demonstrated with the simulation of an example heat recovery stack similar to that in Fig. 2. The structure of the example heat recovery stack is an exact replica of the heat recovery stack in the wet end of Kaipola PM4. In the stack, four CHR units are arranged in a combined cross- and counterflow pattern, while three parallel process water AHR units are located after the CHR units. On top of the process water units, six circulation water AHR units for machine hall ventilation are arranged in series in two parallel rows.

Heat transfer in the example stack is presented in Fig. 3, where T_s denotes the surface temperature on the exhaust air side and Tdp the dew point of exhaust air in the flow direction of exhaust air. The temperature of supply air is presented in the cross flow direction, which is why the final temperature of supply air exceeds the calculated surface temperature at the beginning of the curves. This, of course, does not occur in real life or during simulation, as it is due only to averaging in different directions.

In Fig. 3, heat is transferred from exhaust air without condensation up to point A, at which the surface temperature goes below the dew point. The change of slope in the temperature of exhaust

Table 1
Measured process values in Kaipola 2001 compared to results with conventional heat recovery (CHR) unit simulation

PM4 wet end	18-Apr-01	CHR
Exhaust air	22.9	22.9 kgd.a./s
	81.4/66.7	81.4/69.6 °C
	178/178	178/175.8 g_{H_2O}/kgd.a.
Supply air	13.4	13.4 kgd.a./s
	36/68.2	36/70.6 °C
	23	23 g_{H_2O}/kgd.a.
Heat transfer rate	454	486 kW
Difference	7.1%	
PM4 dry end	**18-Apr-01**	**CHR**
Exhaust air	19.2	19.2 kgd.a./s
	83.5/70.9	83.5/71.6 °C
	186/178.6	186/182.7 g_{H_2O}/kgd.a.
Supply air	12.6	12.6 kgd.a./s
	36/69.7	36/70.9 °C
	23	23 g_{H_2O}/kgd.a.
Heat transfer rate	446	462 kW
Difference	3.5%	
PM6 wet end	**2-Feb-01**	**CHR**
Exhaust air	34.2	34.2 kgd.a./s
	81.3/67.7	81.4/67.6 °C
	133/133	133/133 g_{H_2O}/kgd.a.
Supply air	20.5	20.5 kgd.a./s
	40/66.3	36/68.5 °C
	15.6	15.6 g_{H_2O}/kgd.a.
Heat transfer rate	556	605 kW
Difference	8.7%	
PM6 dry end	**2-Feb-0l**	**CHR**
Exhaust air	49.9	49.9 kgd.a./s
	86.6/70.7	86.6/74.3 °C
	101/101	101/101 g_{H_2O}/kgd.a.
Supply air	21.1	21.1 kgd.a./s
	41.7/72.3	41.7/75.4 °C
	15.1	15.1 g_{H_2O}/kgd.a.
Heat transfer rate	679	736 kW
Difference	8.4%	
PM6 wet end	**16-Mar-01**	**CHR**
Exhaust air	35.7	35.7 kgd.a./s
	80/67	80/66.8 °C
	128/128	128/128 g_{H_2O}/kgd.a.
Supply air	21.1	21.1 kgd.a./s

Table 1 (*continued*)

PM6 wet end	16-Mar-01		CHR
	40.8/66.6		40.8/67.7 °C
	16.3		16.3 g_{H_2O}/kgd.a.
Heat transfer rate	562		589 kW
Difference		4.7%	
PM6 dry end	**16-Mar-0l**		**CHR**
Exhaust air	52.9		52.9 kgd.a./s
	85.6/73.6		85.6/74.6 °C
	106/106		106/106 g_{H_2O}/kgd.a.
Supply air	20.8		20.8 kgd.a./s
	42.8/74.8		42.8/75.6 °C
	17.4		17.4 g_{H_2O}/kgd.a.
Heat transfer rate	694		710 kW
Difference		2.3%	
PM7 wet end	**29-Mar-01**		**CHR**
Exhaust air	31.6		31.6 kgd.a./s
	71.5/58.5		71.5/60.4 °C
	140/129		140/139.5 g_{H_2O}/kgd.a.
Supply air	20.7		20.7 kgd.a./s
	36/57.8		36/57.8 °C
	16.6		16.6 g_{H_2O}/kgd.a.
Heat transfer rate	491		489 kW
Difference		−0.4%	
PM7 dry end	**29-Mar-0l**		**CHR**
Exhaust air	34.3		34.3 kgd.a./s
	76.8/63.5		76.8/64.6 °C
	147/136		147/146.9 g_{H_2O}/kgd.a.
Supply air	17.5		17.5 kgd.a./s
	34.6/61.6		34.6/65 °C
	16		16.6 g_{H_2O}/kgd.a.
Heat transfer rate	488		550 kW
Difference		12.6%	
PM7 wet end	**5-Apr-01**		**CHR**
Exhaust air	31.1		31.1 kgd.a./s
	70.6/58.3		70.6/58.7 °C
	127.5/114		127.5/127.2 g_{H_2O}/kgd.a.
Supply air	20.4		20.4 kgd.a./s
	33.5/56.4		33.5/56.7 °C
	17.4		17.4 g_{H_2O}/kgd.a.
Heat transfer rate	485		491.1 kW
Difference		1.2%	

(*continued on next page*)

Table 1 (*continued*)

PM7 dry end	5-Apr-01	CHR
Exhaust air	30.5	30.5 kgd.a./s
	74.5/60.9	74.5/61.9 °C
	125/102	125/125 g_{H_2O}/kgd.a.
Supply air	15.7	15.7 kgd.a./s
	33.9/60.6	33.9/63.5 °C
	17.0	17 g_{H_2O}/kgd.a.
Heat transfer rate	437	485 kW
Difference	11.0%	

The values for temperature and humidity are presented in the form in/out. The abbreviation d.a. refers to dry air.

air at point A is due to condensation, even though the effect is relatively small because the surface temperature of exhaust air side is only slightly below the dew point. Further on, point B represents the transition from CHR units to AHR units. The surface temperature drops down close to the temperature of water and condensation becomes very intense. After point B, the dew point of exhaust air gradually declines as the exhaust air gets drier. Point C denotes the transition from the process water units to the circulation water units. Thereafter, condensation still continues effectively, although the temperature of the circulation water is already as high as 23 °C when arriving to heat recovery.

One of the main purposes of the previous example is to show that the exhaust air never becomes completely saturated except for the thin boundary layer that develops on the surface of the heat exchanger. This is emphasized to dispell the common misconception that the whole exhaust air flow should reach the saturation temperature before condensation is even possible. The correct statement is that condensation always takes place when the surface temperature on the exhaust air side is below the corresponding dew point. Therefore, the modelling of heat transfer from humid air requires the determination of the surface temperature and the dew point throughout the whole heat exchanger.

In Fig. 3, the effect of convective heat transfer without condensation is only 1.9 MW, whereas the effect of condensation is 7.5 MW. This is illustrated in more detail in Fig. 4, which shows the heat transfer rate as a function of exhaust air temperature. The figure clearly points out the non-linearity of the total heat transfer rate compared to the temperature of exhaust air. The other curve in the figure is included to illustrate the effect of latent heat due to condensation.

In conclusion, it is imperative to understand the mechanisms, which affect the rate of condensation, in order to improve heat recovery in the most effective way. From the perspective of operating paper machines, the AHR units with greater relative effect are more interesting than the CHR units.

3.1. Effect of structural factors on heat recovery

Condensation has little effect on the supply air units because the surface temperature is close to the dew point of exhaust air. This leaves only two structural opportunities to improve the heat

Table 2
Measured process values in Kaipola 2001 compared to results with aqua heat recovery (AHR) unit simulation

PM4 wet end	18-Apr-01		AHR
Exhaust air	22.9		22.9 kgd.a./s
	6.77/-		66.7/60.9 °C
	1787/-		178/117.3 g_{H_2O}/kgd.a.
Process water	81.7		81.7 kg/s
	33.8/44.5		33.8/44.5 °C
Heat transfer rate	3654		3667 kW
Difference		0.4%	
PM4 dry end	**18-Apr-01**		**AHR**
Exhaust air	19.2		19.2 kgd.a./s
	70.9/-		70.9/ °C
	178.6/-		178.6/135.5 g_{H_2O}/kgd.a.
Process water	81.7		81.7 kg/s
	45.2/52.9		45.2/51.7 °C
Heat transfer rate	2630		2206 kW
Difference		−16.1%	
PM6 wet end	**2-Feb-0l**		**AHR**
Exhaust air	34.2		34.2 kgd.a./s
	67.7/45.1		67.7/44.9 °C
	133/66		133/55.6 g_{H_2O}/kgd.a.
Circulation water	173.7		173.7 kg/s
	42/47.6		42/48.3 °C
	23.5		23.5%
Heat transfer rate	3804		4308 kW
Difference		13.2%	
Process water	36.2		36.2 kg/s
	22.2/46.2		22.2/43.6 °C
Heat transfer rate	3625		3238 kW
Difference		−10.7%	
PM6 dry end	**2-Feb-01**		**AHR**
Exhaust air	49.9		49.9 kgd.a./s
	70.7/45.1		70.7/46.5 °C
	101/66		101/59.8 g_{H_2O}/kgd.a.
Circulation water	130.3		130.3 kg/s
	42/47.6		42/49.3 °C
	23.5		23.5%
Heat transfer rate	2853		3220 kW
Difference		12.9%	
Process water	36.2		36.2 kg/s
	22.2/46.2		22.2/44.3 °C

(*continued on next page*)

Table 2 (*continued*)

PM6 dry end	2-Feb-01		AHR
Heat transfer rate	3625		3333 kW%
Difference		−8.1%	
PM7 wet end	29-Mar-01		AHR
Exhaust air	44.3		44.3 kgd.a./s
	58.5/45.5		58.5/47.8 °C
	129/67		129/67.9 g_{H_2O}/kgd.a.
Circulation water	33.0		33.0 kg/s
	43.1/54.9		43.1/54.1 °C
	14		14%
Heat transfer rate	1569		1460 kW
Difference		−6.9%	
Process water	57.5		57.5 kg/s
	23.5/41.4		23.5/47.8 °C
Heat transfer rate	4302		5846 kW
Difference		35.9%	
PM7 dry end	29-Mar-01		AHR
Exhaust air	34.3		34.3 kgd.a./s
	63.5/45.5		63.5/47.2 °C
	136/67		136/63.6 g_{H_2O}/kgd.a.
Circulation water	33.0		33.0 kg/s
	43.1/54.9		42/55.6 °C
	14		14%
Heat transfer rate	1569		1675 kW
Difference		6.8%	
Process water	57.5		57.5 kg/s
	23.5/41.4		23.6/45.3 °C
Heat transfer rate	4302		5203 kW
Difference		20.9%	

The values for temperature and humidity are presented in the form in/out. The abbreviation d.a. refers to dry air.

transfer rate in the CHR units. The addition of surface area would reduce the minimum temperature difference inside the heat exchanger. The other choice would be to reduce the plate distances inside the CHR unit, which increases turbulence in the units. However, as the total heat transfer rate in the supply air units is typically less than 1 MW per stack, the expected value of profit would be insignificant compared to the required investment costs.

The aqua heat recovery units are completely modular, which means extra AHR modules can be installed in an existing heat recovery system in parallel or in series. The addition of surface area by installing parallel units does not have a significant effect, because the mass flow rates of exhaust air, process water and circulation water would be divided between an increased number of heat

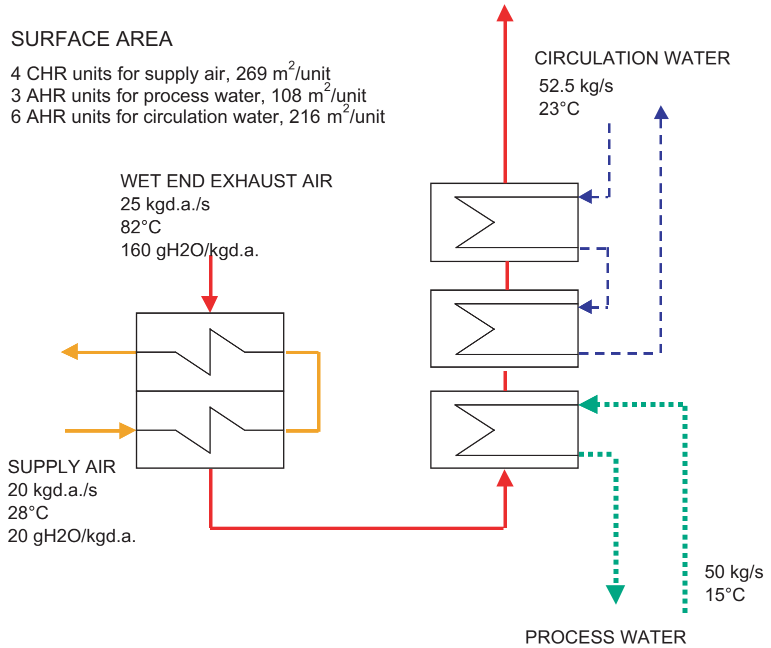


Fig. 2. Structure of an example heat recovery stack. The abbreviation d.a. refers to dry air.

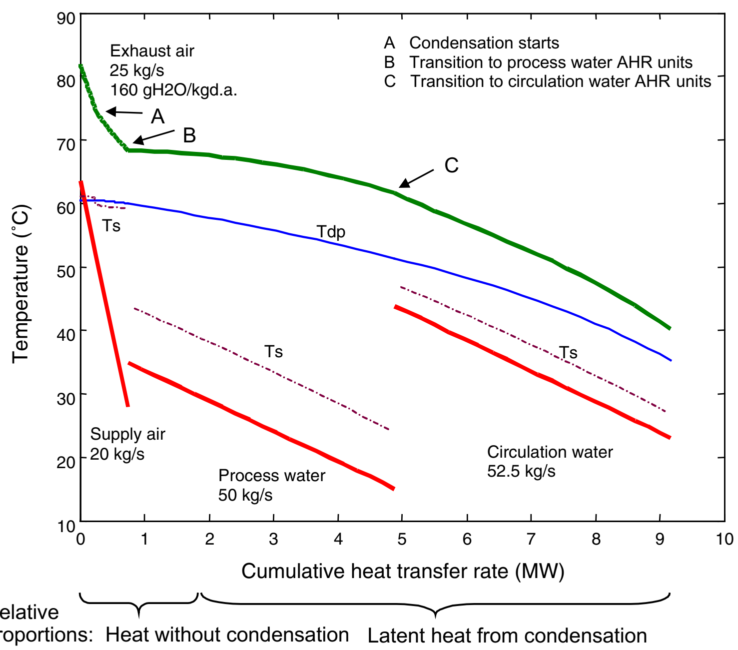


Fig. 3. Heat transfer in the example heat recovery stack with design values.

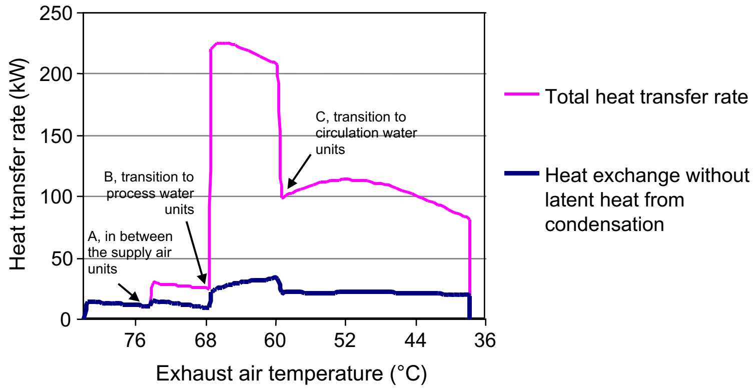


Fig. 4. Heat transfer rate in the example stack divided into heat transfer without condensation and total heat transfer rate.

Table 3
Effect of parallel AHR units on the example heat recovery stack

	Current				
Units parallel/stage	1	2	3	4	5
Process water units	108	216	324	432	540 m^2
Circulation water units	432	864	1296	1728	2160 m^2
Total area	540	1080	1620	2160	2700 m^2
Process water	3.78	4.06	4.15	4.17	4.17 MW
Circulation water	4.28	4.28	4.28	4.28	4.28 MW
Total heat transfer rate	8.06	8.34	8.43	8.45	8.45 MW

recovery units, which reduces turbulence. According to Table 3, the addition of parallel units would increase heat transfer only by 20 kW more than the original example stack.

In contrast to parallel coupling, the addition of extra units in series does improve the heat transfer rate. In Table 4, the addition of extra process water units increases the total heat transfer rate by 1 MW. As a downside, the circulation water would suffer from the change when the heating demand of machine hall ventilation exceeds 2 MW.

This raises the question of whether adding heat transfer surface in series to an existing heat recovery installation is profitable or not. This can be answered only by investigating all the savings and costs related to the case in question, including the investment costs, the consequences of increasing pressure drop, etc. Nevertheless, the dimensioning of typical heat recovery systems used in Finland after the beginning of the 1980s suggests that adding new heat transfer units is unprofitable. Until the early 1990s, the inlet temperature of process water used for the dimensioning of heat transfer surface was typically 5 °C. Due to the closure of water systems in the operating paper mills, the typical inlet temperature is now above 20 °C, even in the coldest periods, indicating that adding new heat recovery units to an existing system is unnecessary. When examining real processes, simulation can be used to confirm this.

Table 4
Effect of serial AHR units on the example heat recovery stack

	Current				
Units parallel/stage	3	3	3	3	3
Process water units in series	108	108	108	216	216 + 108
	324	324	324	648	972 m^2
Circulation water units in series	216	216 + 108	216 + 216	216 + 216	216 + 216
	648	972	1296	1296	1296 m^2
Total area	972	1296	1620	1944	2268 m^2
Process water	4.15	4.15	4.15	6.26	7.41 MW
Circulation water	3.18	3.85	4.28	2.82	1.96 MW
Total heat transfer rate	7.33	8.00	8.43	9.07	9.37 MW

The heat recovery units can also be positioned in series from one stack to another. This type of coupling is possible if the flow rate of process water is small compared to the number of existing process water units. The arrangement enables a very high final temperature.

From these examples, the following conclusions can be drawn:

- increasing the surface area of CHR units is uneconomic,
- the number of parallel AHR units typically is adequate, allowing for moderate capacity improvements during the life-time of the dryer section,
- heat recovery is affected the most by adding or removing AHR units coupled in series,
- the heat transfer surfaces of installed heat recovery systems are usually adequate.

4. Order of heat recovery units in heat recovery

The order of different stages in heat recovery is dictated by the required final temperature and the heating demand of each stream. In modern installations, supply air units are positioned first because they have the highest requirement for temperature. The order of AHR units is typically: white water (55 °C), process water (50–55 °C) and circulation water (46–50 °C). Even as late as the 1980s, it was the custom to place circulation water units before the process water units. Today, the objective is to design heat recovery systems individually according to the requirements of the case in question.

The addition of white water units to an existing heat recovery system is possible if the heat recovery is able to satisfy the whole heating demand of process water and circulation water, and if the humidity of exhaust air is high, in practice over 160 $g_{H_2O}/kg_{d.a.}$. The high humidity level enables a high final temperature for the water.

Process water is needed all year around with a heating demand depending on the inlet temperature of fresh water. Due to the increased closure of the water systems, the process water units may be situated in front of the circulation water units in heat recovery with sufficient heat for machine hall ventilation. The heating demand of machine hall ventilation depends directly on outdoor temperature. In Northern countries, the heat recovery system is typically designed to cover the whole heating demand of machine hall ventilation down to an outdoor air temperature of 20 °C. If circulation water units have been positioned before process water units, the permanence of outdoor temperature dictates whether changing the order is beneficial or not. Changing the order naturally decreases the available heat for machine hall ventilation.

4.1. Effect of operational factors on heat recovery

The most important operational factors affecting the efficiency of heat recovery are the humidity of exhaust air and the inlet temperatures and flow rates of all the streams. In this section, the operational factors are discussed using the simulated results of an example heat recovery system with two stacks of the type presented in Fig. 5. The structure of the heat recovery stacks is identical to the example stack in Fig. 2.

4.1.1. Effect of exhaust air humidity, temperature and flow rate

An increase in exhaust air temperature also increases the recovered heat due to a greater temperature difference. However, the effect is not significant compared to the total heat transfer rate because a rise in the inlet temperature only increases the available sensible heat. Already in Fig. 3, it was demonstrated that latent heat released in condensation gives considerably more energy than the sensible heat.

Production rate, evaporation in the dryer section and the humidity level of exhaust air determine the flow rate of exhaust air. With dry exhaust air, more drying air is needed and the flow rates of exhaust air and supply air increase. Although this promotes turbulence and heat transfer in the supply air units, it has an opposite effect on the AHR units because the low humidity of exhaust air gives less condensation. When using more drying air, the heating of drying air with steam will increase as well. It is therefore imperative to use as little drying air as possible.

The limit for how little drying air can be used is set by the exhaust air humidity. If the exhaust air becomes too humid, the vapour contained in the air may condense inside the hood causing web

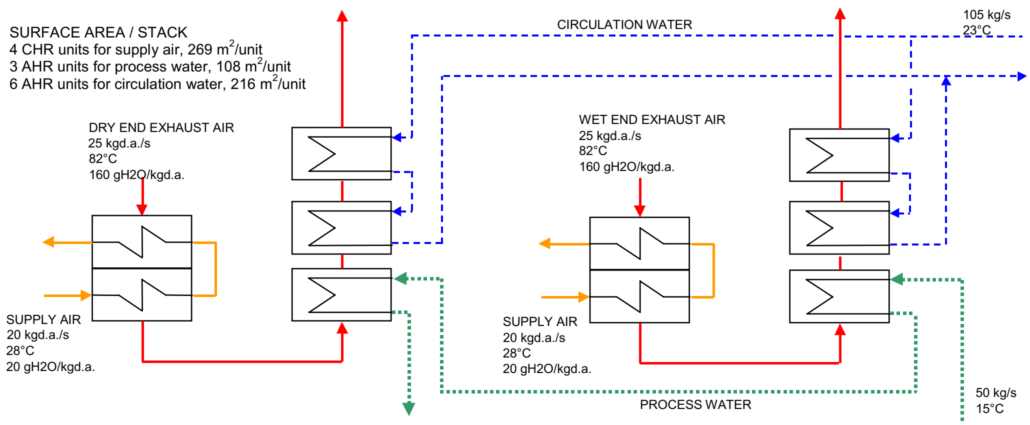


Fig. 5. Example heat recovery system with two heat recovery stacks.

brakes. In modern hoods the design point for exhaust air humidity is typically 160–200 g_{H_2O}/$kg_{d.a.}$, even though the hoods can locally withstand much higher humidities.

In the example heat recovery system, low exhaust air humidity and increased flow rates only bring about 200 kW of extra heat to the CHR units. In contrast, using a higher exhaust air humidity provides 300–1000 kW more heat for the whole heat recovery system, depending on the evaporation rate in the dryer section (Fig. 6).

4.1.2. Effect of water inlet temperature and flow rate

The inlet temperature of water has a significant effect on the heat transfer rate of heat recovery. In fact, it has a more drastic effect than the humidity of exhaust air, as demonstrated in the following example.

In Fig. 7, increasing the humidity from 120 to 160 g_{H_2O}/$kg_{d.a.}$ brings about 300–500 kW of extra heat to the process water units. Compared to this, the relation between inlet temperature and heat transfer rate is the opposite: the lower the inlet temperature, the higher the gain due

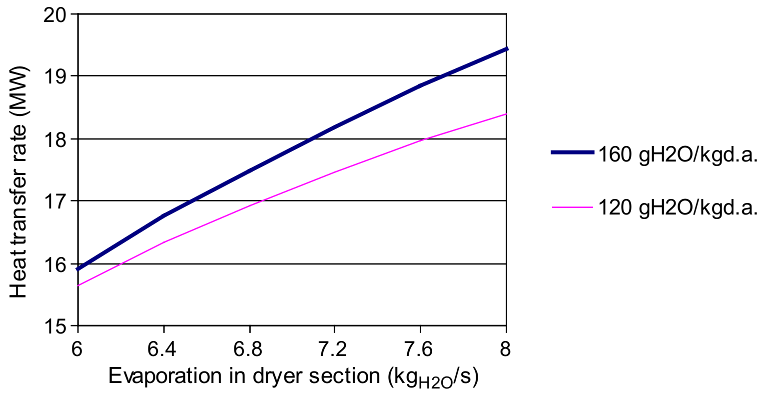


Fig. 6. Total heat transfer rate with two different humidities of exhaust air in the example heat recovery system.

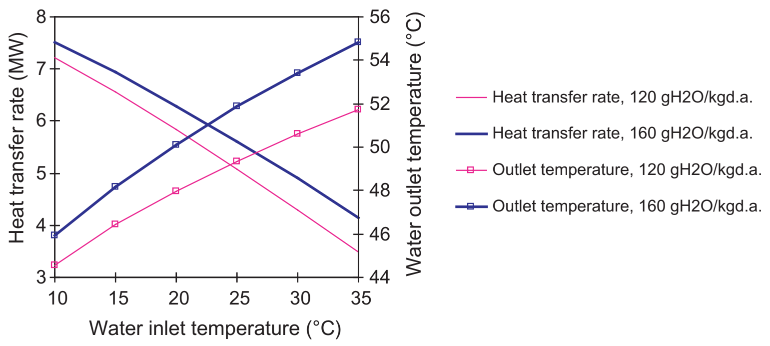


Fig. 7. Heat transfer rate and water outlet temperature in relation to water inlet temperature in the process water AHR units of the example heat recovery system.

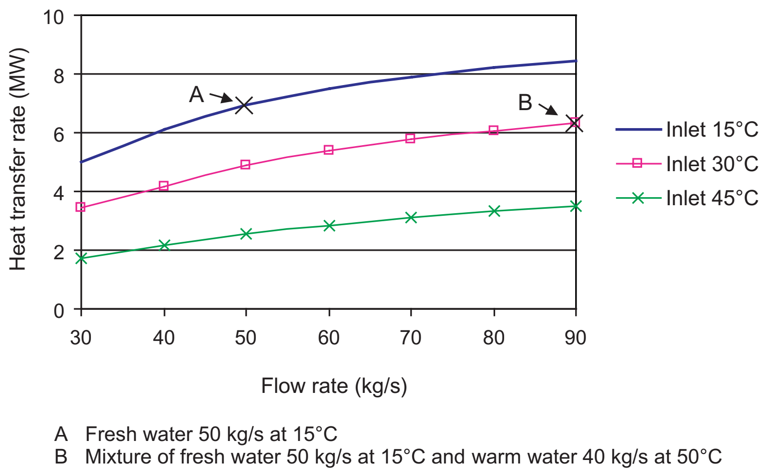


Fig. 8. Heat transfer rate in relation to the flow rate and inlet temperature of process water in the process water AHR units of the example heat recovery system.

to more effective condensation. According to Fig. 7, an inlet temperature of 10 °C gives a heat transfer rate of 7.1 MW, whereas a temperature of 35 °C gives only 3.5 MW with the same humidity level. In conclusion, it is very important not to preheat the water before heat recovery, but to lead the water into the heat recovery system as cold as possible.

The appropriate flow rate of water in heat recovery has a connection to the inlet temperature of the water, given that the water can be recirculated to maximise flow rate and turbulence. In this case, the recirculated water has already gone through heating causing the inlet temperature to rise. In the example system, the demand for additional water is 50 kg/s, which gives about 7 MW of recovered heat with inlet temperature of 15 °C, shown with letter A in Fig. 8. If this cold water was mixed with 40 kg/s of warm water at 50 °C, the mix would be 90 kg/s at 30 °C (letter B in Fig. 8) and the recovered heat would only be 6.2 MW. As a result of the mixing, the recovered heat decreases by 800 kW despite the increase in the flow rate. Hence recirculation loops are not necessarily justified from the energy perspective.

5. Conclusions

The purpose of dryer section heat recovery is to decrease the energy use of a paper machine as economically as possible. This includes providing the optimal structure of heat recovery as well as efficient operation conditions for the system. In reality, there is no guarantee that an installed heat recovery system will operate efficiently, since the original design parameters may be appropriate only for a short period of time. After all, the main purpose of a paper machine is to produce paper; the continuous effort towards a higher production capacity does not automatically mean that energy economy is taken into account. As a rule, energy efficiency can be improved for as long as it is financially profitable and the changes do not compromise the production or cause problems to process control and safety.

The present paper acknowledges this improvement potential by presenting simulation examples of the effect of different internal and external conditions on the performance of dryer section heat recovery. As the simulation examples only give information on the apparent trends, the exact quantitative values in a retrofit situation have to be solved individually with simulation. Results from such a demonstration project will be published later.

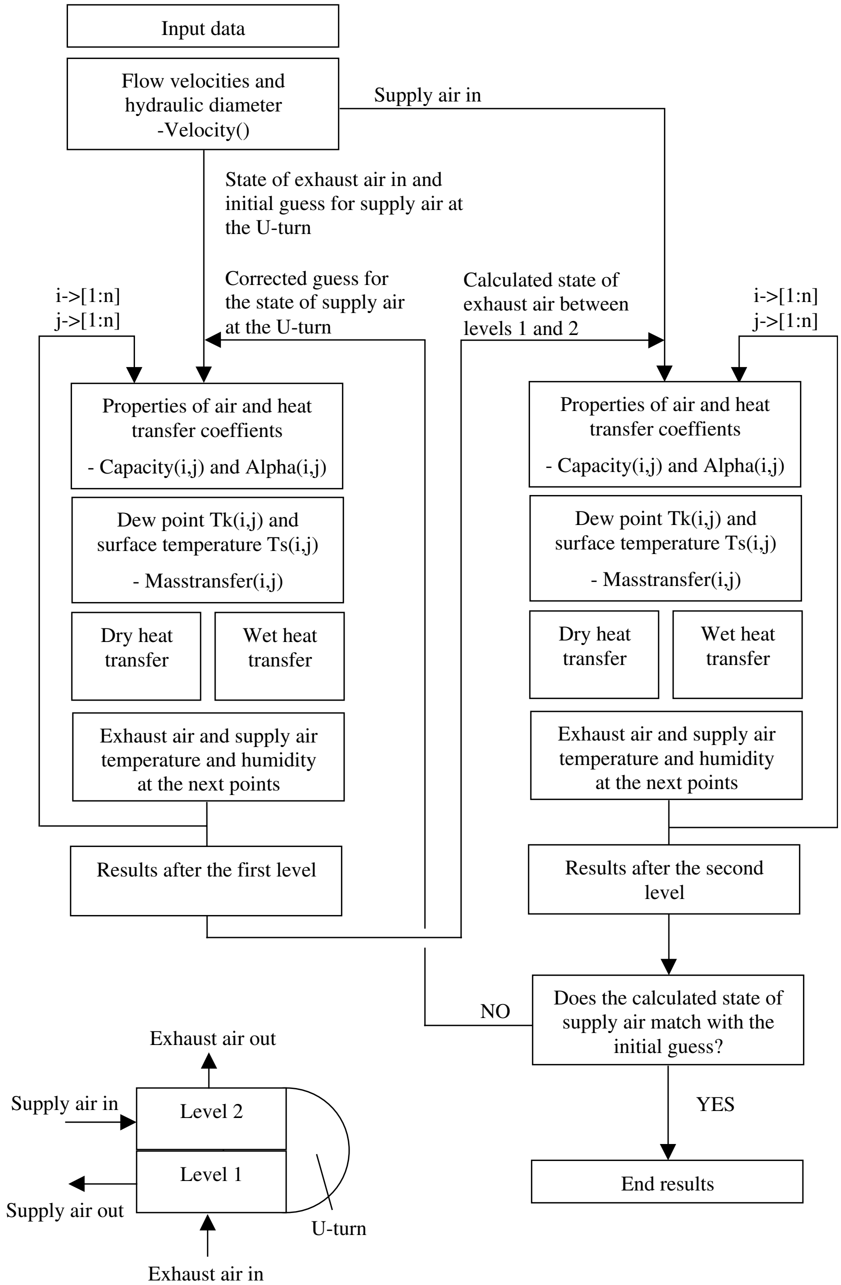
According to simulation, one should start seeking opportunities to improve heat recovery from the development of the operation conditions. Basically, this means that the humidity of exhaust air is set as high as possible by adjusting the control of hood ventilation and, secondly, that the process water and circulation water for machine hall ventilation are brought to heat recovery without preheating with primary energy and without mixing the streams with water that has already been heated. After this, one can continue the search by investigating whether changing the order of the heat exchangers and connecting some heat exchangers in series instead of parallel coupling is beneficial or not. In contrast, because the investments are likely to be financially unprofitable, it is not expected that completely new heat exchangers will be added to existing installations.

To ensure accuracy in the analyses of real processes, gathering data from the automation system of the mill is preferable to field measurements because the values of some variables in heat recovery, especially the inlet temperature of process water, are very dependent on time, due to the operation of the automation system. In addition, the analyses of real processes require taking into account the possible seasonal variation. Unfortunately, not even modern paper machines are equipped with the full repertoire of instrumentation that would allow for an easy access to time averages from all the process variables required. Then again, running a process with high efficiency is more important than getting the data right to the last digit.

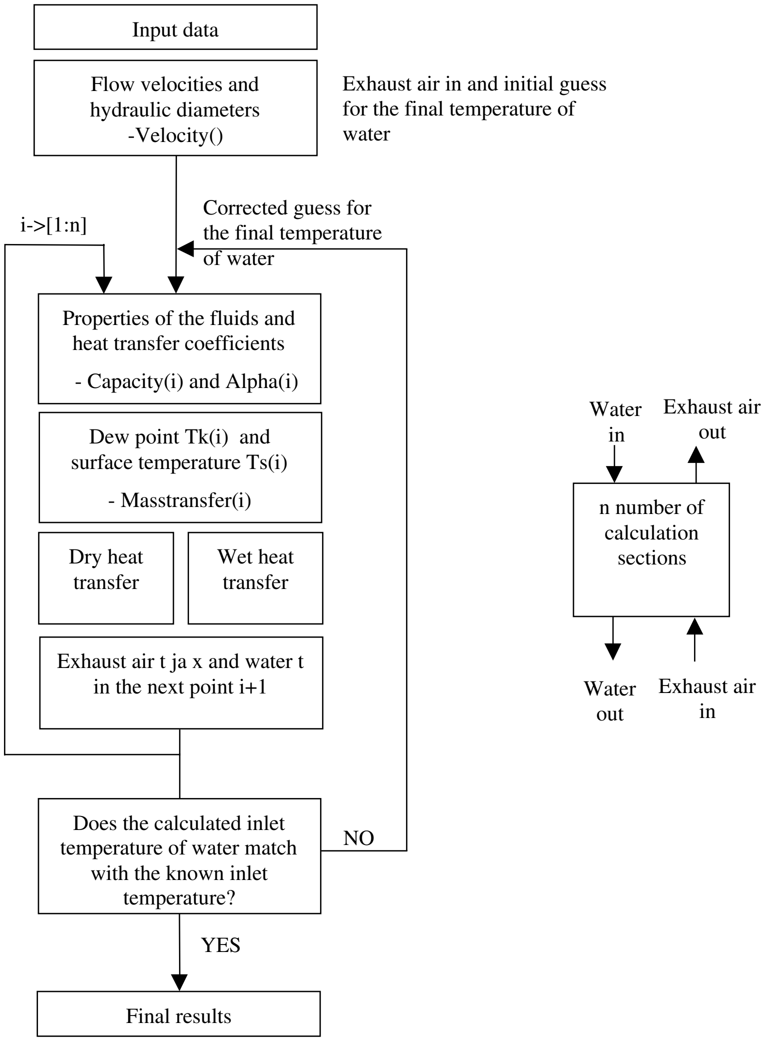
Acknowledgements

The authors thank the personnel of UPM-Kymmene Oyj Jämsänkoski and Kaipola Mills for their helpful comments.

Appendix A. Structure of CHR simulation program



Appendix B. Structure of AHR simulation program



References

[1] H. Sundqvist, Dryer section ventilation and heat recovery, in: M. Karlsson (Ed.), Papermaking, Part 2. Fapet, Helsinki: Drying, Papermaking Science and Technology, 2000.
[2] Review of business conditions in the forest sector 2002–2003, Finnish Forest Research Institute, Vantaa, 2003.
[3] H. Sundqvist, Simulering av värmeåtervinningssystem för pappersmaskiner, Master's Thesis, Åbo Akademi, 1985.
[4] G. Henriksson, Modell och simulator för optimering av värmeåtervinningssystem för pappersmaskiner, Master's Thesis, Åbo Akademi, 1988.

[5] Optimisation of the paper machine heat recovery system, Ministry of Trade and Industry, Valmet Paper Machinery, Turku, 1988.
[6] D. Maltais, Heat recovery on a paper machine hood, in: Proceedings of the 78th Annual Meeting of Canadian Pulp and Paper Association, Montréal, 1992.
[7] K. Penttinen, Process simulation and pinch studies in paper mill processes, Licentiate's Thesis, Department of Chemical Engineering, Lappeenranta University of Technology, 1998.
[8] J. Söderman, T. Westerlund, F. Pettersson, Economical optimisation of heat recovery systems for paper machine dryer sections, in: F. Friedler, J. Klemes (Eds.), Proceedings of the 2nd Conference on Process Integration, Modelling and Optimisation for Energy Saving and Pollution Reduction, PRES'99, Budapest, Hungary, 1999, pp. 607–612.
[9] J. Söderman, P. Heikkilä, Calculation of the heat transfer coefficients with condensation in heat recovery systems, Report 2001-1, Heat Engineering Laboratory, Åbo Akademi University, 2001, ISBN 952-12-0820-1.
[10] J. Söderman, F. Pettersson, Comparison of solutions with varying cost factors in structural optimisation of paper machine heat recovery systems, Report 2001-3, Åbo Akademi University, 2001.
[11] J. Söderman, F. Pettersson, Influence of variations in cost factors in structural optimisation of heat recovery systems with moist air streams, Applied Thermal Engineering 23 (2003) 1807–1818.
[12] L. Kilponen, Improvement of heat recovery in existing paper machines, Licentiate's Thesis, Department of Mechanical Engineering, Helsinki University of Technology, 2002.
[13] M. Soininen, Dimensioning of paper machine heat recovery recuperators, Drying Technology 13 (4) (1995) 867–896.