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Energy efficiency improvement of dryer section heat recovery systems in paper machines – A case study

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ABSTRACT

Modern paper machines are equipped with heat recovery systems that transfer heat from the humid exhaust air of the paper machine's dryer section to different process streams. As a result of process changes, the heat recovery systems may operate in conditions far from the original design point, creating a significant potential for energy efficiency improvement. In this paper we demonstrate this potential with a case study of three operating paper machines. Both operational and structural improvement opportunities are examined. Since the existing retrofit methodologies for heat exchanger networks can not be applied to cases with condensing air, we use thermodynamic simulation models presented earlier to assess the effects of possible changes on the existing heat recovery systems. In order to reduce the required processing time of the simulation models, only a limited number of pre-screened retrofit designs are considered. The pre-screening is carried out on the basis of guidelines presented earlier. The analysis in the case mill revealed savings of 110 GWh/a in process heat with profitable investments. According to the follow-up study, the investments carried out have resulted in 12% lower fuel use and 24% lower CO_2 emissions. The results imply that all operating paper machines should be similarly examined.

1. Introduction

The performance of dryer section heat recovery is essential for the economy of papermaking. A modern wide paper machine uses over 50 MW of primary heat, of which a well-performing heat recovery system is able to recover over 60% in the coldest periods [1]. However, the performance of the installed systems can be far from optimal due to constantly changing operating conditions, renovations and other process changes. Consequently, the operating heat recovery systems can have significant potential for energy efficiency improvement.

In this paper we demonstrate the magnitude of this energy conservation potential with a case study of three existing paper machines. We use a thermodynamic simulation model presented earlier by Kilponen[1] [2] to calculate the performance of heat recovery in the case of several operational and structural improvement opportunities. These opportunities are determined prior to simulation using engineering knowledge and guidelines presented by Sivill et al. [3]. We also review the existing process integration methodologies for the retrofitting of heat exchanger networks (HEN) and the grass-root design of dryer section heat recovery systems to explain why these existing methods can not be applied to retrofit cases with condensing air.

1.1. Structure of dryer section heat recovery systems

The dryer section heat recovery system is a network of heat exchangers in which heat from the dryer section's exhaust air is transferred to its supply air and different water streams e.g., process water, white water and the circulation water of machine hall ventilation. The heat recovery system typically consists of two to four heat recovery stacks, depending on the amount of exhaust air. Each stack is coupled with other stacks in parallel or in series on the water side. The heat exchangers used in modern installations are typically modular plate or combined tube and plate heat exchangers with no contact between the exhaust air and the heated streams (see e.g., Sundqvist [4]). An example of the structure of a modern heat recovery stack is presented in Fig. 1.

1.2. Earlier work

The potential for improving the operation of existing HENs is typically studied prior to retrofitting [5]. After this, process integration methods can be used to determine the most economic HEN modifications. The existing process integration methods for retrofitting can be divided into methods based on pinch analysis [5–12], optimisation methods [13–17] and combinations of the two [18–23]. In connection with the use of pinch analysis, graphical methods have been developed based on exergy [24–27] and advanced composite curves [28–30].

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[1] Sivill since 2004.

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Nomenclature

A	heat transfer surface area (m^2)
h	convective heat transfer coefficient (W/m^2 K)
k	mass transfer coefficient (m/s)
l	latent heat (J/kg)
$\dot{m}''_c$	mass flux of condensation (kg/s m^2)
M	molecular weight of water (kg/mol)
P	pressure (Pa)
Q	heat transfer rate (W)
R	ideal gas constant (J/mol K)
s	thickness of heat exchanger material (m)
T	temperature (K)

Greek letter

λ	thermal conductivity (W/m K)

Subscripts

0	total moist air
1	exhaust air
2	stream to be heated
s	surface on exhaust air side
vap	vapour

Superscripts

$'$	saturated fluid
$''$	per unit area

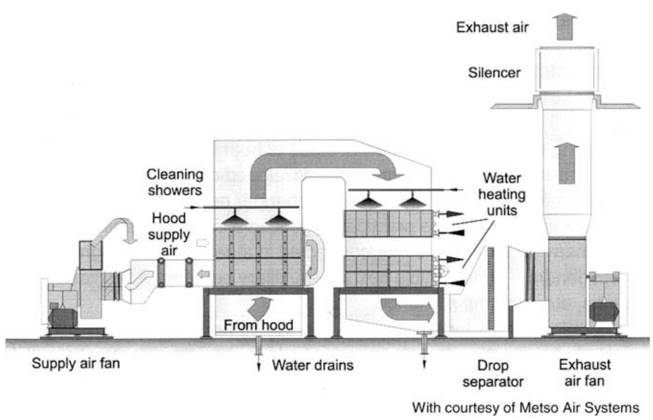

Fig. 1. Modern heat recovery stack of a paper machine dryer section.

All these methods begin with the use of engineering knowledge to set up the initial assumptions for the streams to be matched, and to determine the practical design constraints. Furthermore, all these methods assume constant heat capacities for the streams. This assumption is not applicable to humid air under condensing conditions [3,4,31].

Söderman et al. [32–34] have presented a methodology for the grass-root design of dryer section heat recovery systems based on mathematical programming. This is done by dividing the humid air into small linear temperature intervals. At each interval, the overall heat transfer coefficients are approximated individually for every possible temperature interval match before optimising the structure of the heat exchanger network. In addition, Söderman and Pettersson have studied the influence of the variation of cost factors on the structural optimisation [35] and presented a hybrid method for the synthesis of robust and optimal heat recovery systems [1].

2. Methods

Heat transfer from the dryer section's exhaust air is mainly based on condensation. The following equation can be used to numerically solve the heat transfer rate from the humid air under condensing conditions [2]

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

where the mass flux of condensation is defined from

$$\dot{m}''_c = M \frac{p_0}{RT} k \ln \frac{p_0 - p'_{vap}(T_s)}{p_0 - p_{vap}} \tag{2}$$

Eq. (2) is derived from Fick's law and takes into account the Stefan flow. The most important information in Eqs. (1) and (2) is that condensation always takes place when the surface temperature of the heat exchanger is below the dew point and that the rate of condensation is a function of the local conditions.

The determination of a heat transfer rate for each possible match with condensing air is not a simple task. A systematic method to handle this in retrofit cases has not been presented. In grass-root design, Söderman et al. [32,34] use several assumptions to simplify the heat and mass transfer equations for optimisation. For example, the mass transfer equation is derived for absorption, not condensation, and it therefore ignores the Stefan flow. This leads to the heat transfer rate being underestimated, as demonstrated in the heat exchanger models by Kilponen [2] in Fig. 2. Secondly, the results by Söderman et al. [32,34] do not indicate how the exhaust air humidity before and after each temperature match is handled when summarizing individual temperature matches or when crossing over from one temperature match to another. This is important as the path that has been taken prior to each match affects the humidity available for the following matches. We illustrate this in Fig. 3 with an example calculated using the thermodynamic model presented by Kilponen [2]. The heat available for the next temperature interval depends on which of the cases in Fig. 3, a or b, is chosen as the preceding match. Thirdly, if the air is assumed to be completely saturated after each condensing temperature

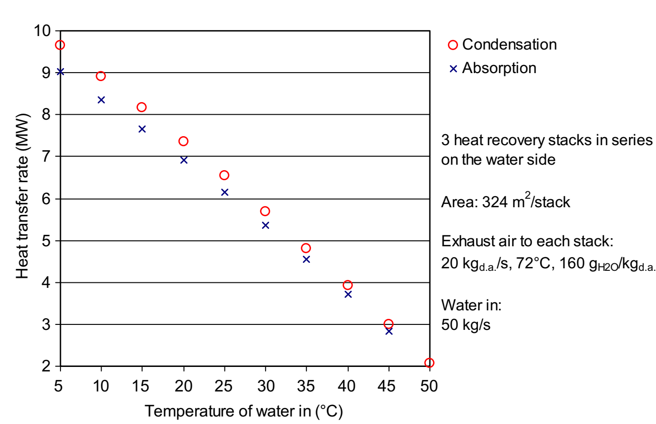

Fig. 2. Heat transfer rates for an example heat recovery system by using mass transfer equations for condensation and absorption: average difference 6%.

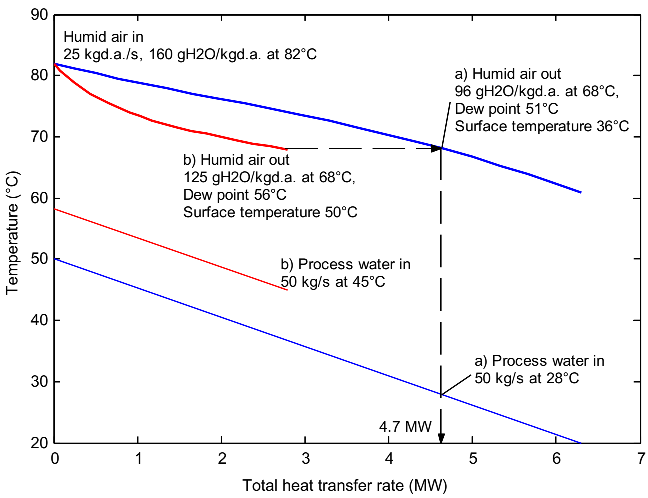

Fig. 3. The humidity of air after each match (here case a or b) at a certain temperature interval of the air (here 82–68 °C) is unique if condensation occurs.

match, the effect of this assumption on the results needs to be estimated as, according to Sivill et al. [3], the air may reach saturation point only on the heat exchanger surface and may remain unsaturated elsewhere.

Due to the previous assumptions used for grass-root optimisation and the lack of specified retrofit methods, we have chosen to rely strictly on the thermodynamic simulation models by Kilponen [2] to assess the heat transfer rates of all the operational improvement and retrofit opportunities in the case study. The use of Eq. (1) in these models requires a significant amount of processing time due to the numerical solving and the iteration required when applied to counter- and cross-flow heat exchangers. Systematically running through every possible design alternative is therefore not an option. For example, the determination of the heat transfer rate at a single operation point lasts several minutes for a whole existing HEN with a contemporary PC. A set of guidelines presented by Sivill et al. [3] is used to restrict the number of possible design alternatives prior to simulation. The structure of the case study is presented in Fig. 4.

3. Results

The case mill located is in the Nordic countries and comprises thermomechanical (TMP) pulp plants, a de-inking plant and three paper machines. For combined heat and power (CHP) production the case mill has a bubbling fluidised bed (BFB) boiler, and one oil-fired boiler provides additional process heat when necessary.

The recovery of exhaust air from the vacuum system and infra red dryers is included as an opportunity for operational improvement. Seasonal changes are taken into account by calculating the results separately for average summer and winter conditions. The performance of each energy conservation opportunity is estimated

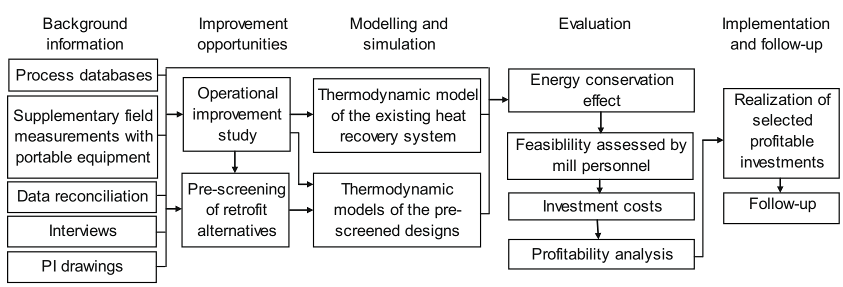

Fig. 4. Structure of the case study.

Table 1
Summary of the energy saving investments in the case mill in 2002.

	Savings in process heat			Annual profit (9.9 €/MWh) (€/a)	Investment costs[a] (€)	Net present value (10 a, 15%) (€)	Payback period (a)	IRR (%)	Realized[b] (GWh/a)
	Winter (MW)	Summer (MW)	Total (GWh/a)						
(a)	7.7	1.5	38.0	375,700	182,100	1,886,000	0.5	206	38.0
(b)	4.4	3.5	32.7	323,300	149,700	1,623,000	0.5	216	22.4
(c)	1.4	1.1	10.4	102,800	61,100	516,000	0.6	168	
(d)	1.5	0.9	9.6	94,900	17,600	476,000	0.2	539	9.6
(e)	1.1	0.6	7.0	69,200	64,900	347,000	0.9	107	7.0
(f)	0.9	0.3	4.8	47,500	55,700	238,000	1.2	85	4.8
(g)	0.5	0.5	4.1	40 500	135,600	203,000	3.3	27	
(h)	0.4	0.4	3.6	35,600	24,000	179,000	0.7	148	
	17.9	8.8	110	1,155,000	691,000	5,797,000	0.6	167	82

(a) PM C heating of process water and fresh water with clear filtrate, changing the order of circulation and process water units in heat recovery.
(b) PM B capacity increase of process water pumps, heating of white water, changing the order of circulation and process water units in heat recovery and humidity of exhaust air to 160 gH_2O/kgd a.
(c) PM C capacity increase of supply air and adjustment of hood ventilation.
(d) PM A connection of cold water to heat recovery and capacity reduction of process water pump.
(e) PM B recovery of condensate and washing water from heat recovery.
(f) PM A recovery of condensate and washing water from heat recovery.
(g) PM A use of exhaust air from vacuum system for the heating of process water.
(h) PM B utilisation of exhaust air from infra red dryers as supply air.
[a] Estimated costs (source: CTS-Engineering Oy, 2002).
[b] PM B humidity could not be increased at (b), estimated potential of the rest of modifications.

Table 2
Relative effect of the realized energy conservation investments on the specific heat consumption in the case mill.

Paper machine	Specific heat consumption in 2001 (MWh/ADt)	Annual potential (GWh/a)	Savings in heat compared to 2001 (%)
PMA	1.4	14	6.8
PMB	1.1	29	9.7
PMC	1.2	38	13.4
Total		82	

ADt = air dry tonne.

individually and as a combination of several opportunities if combining opportunities with a non-competing status have an effect on each other. The results of each design are compared with the modelled performance of the existing heat recovery systems under similar operating conditions. Investment costs were evaluated only for those improvement alternatives which were considered the most feasible by the mill personnel. The investment costs were estimated by CTS Engineering Oy. The costs comprise indirect costs, equipment and machinery, piping, electric appliances, automation, construction works and a 10% cost reserve. Net present value (NPV), internal rate of return (IRR) and payback time (PBT) are used as profitability criteria.

A marginal price of 9.9 €/MWh is used for saved steam based on a price of 8.7 €/MWh for milled peat, which was derived from statistics on average fuel prices in heat production in Finland in 1995–2008 divided by a boiler efficiency of 0.88. Field measurements and the use of averaging are the most important sources of possible errors in the presented results.

Table 1 lists the profitable opportunities and the realized investments between 2002 and 2007. Table 2 presents the effect of the realized investments in relation to the specific heat consumptions of the paper machines. By far the greatest changes affecting the mill's energy use since 2002 are the increase of de-inked pulp production by 30,000 t/a and the decrease of thermomechanical pulping accordingly shown in Table 1. As a result, more steam is produced in the power plant to compensate for the secondary steam that is no longer produced in thermomechanical pulping. In addition, more process water, approximately 60–80 kg/s at 50 °C, is required for the de-inked pulp. According to mill personnel, the increase in de-inked pulp production capacity has increased the demand of process steam on average by 6 MW.

Table 3 and Figs. 5 and 6 show an approximation of what has been achieved in the case mill in 2007 as a result of the energy conservation investments. In Table 3 and Fig. 5, fuels savings are calculated using two routes. Firstly, the business-as-usual fuel use is estimated for 2007 in relation to the capacities and specific heat consumptions in 2001. The difference between this business-as-usual estimate and the realized fuel use in 2007 describes the energy conservation effect. Secondly, the estimated effect of all the realized energy conservation investments is summed up since 2002. These two estimates match with each other even though we ignore the effect of all the process variables affecting the heating demand. On the other hand, the cold seasons in 2001 and 2007 were quite similar and investments to additional heat recovery between the de-inking and thermomechanical pulping are already taken into account in the evaluated effect of the capacity changes.

The follow-up shows that the realized energy conservation investments have enabled changes in the use of raw materials,

Table 3
Effects of energy conservation investments in the case mill in 2007.

Product/process	Specific heat consumption in 2001 (MWh/a)	Capacity change 2001–2007 (ADt/a)	Estimated effect on heating demand in 2007 (GWh/a)
(a) Effect of capacity changes on process heating demand			
PMA	1.4	30,000	6.8
PMB	1.1	20,000	14.3
PMC	1.2	−20,000	19.3
De-inked pulp[a]		30,000	49.7
Thermomechanical pulp[a]		−30,000	
Total			90.1
(b) Forecasted fuels use for 2007 based on capacity changes			
Estimated effect of capacity changes on fuels use in 2007[b]			131 GWh/a
Fuels use in 2001			820 GWh/a
Total			951 GWh/a
(c) Fuels savings in 2007 in relation to forecasted fuels use			
Forecasted fuels use for 2007			951 GWh/a
Realized fuels use in 2007			833 GWh/a
Difference			118 GWh/a
			−12.4%
(d) Energy conservation potential estimated in 2002 for realized investments			
Potential of realized energy conservation investments as process heat			82 GWh/a
Potential of realized energy conservation investments as fuels use[b]			119 GWh/a
(e) Difference between fuels savings estimates			
Estimated fuels savings based on forecasted consumption in 2007			118 GWh/a
Fuels savings based on estimated potential of realized investments			119 GWh/a
Difference			−1 GWh/a
(f) Value of fuels savings in 2007			
Marginal price of peat			8.7 €/MWh
Fuels savings			118 GWh/a
Total			1,027,000 €/a

ADt = air dry tonne.
[a] Annual average effect on steam consumption 6 MW estimated by mill personnel.
[b] Power to heat ratio 0.28 and boiler efficiency 0.88.

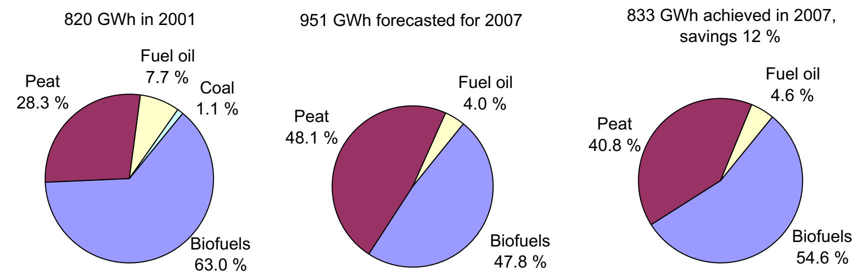

Fig. 5. Fuel use in 2001 and the forecast and actual fuel uses in the case mill in 2007.

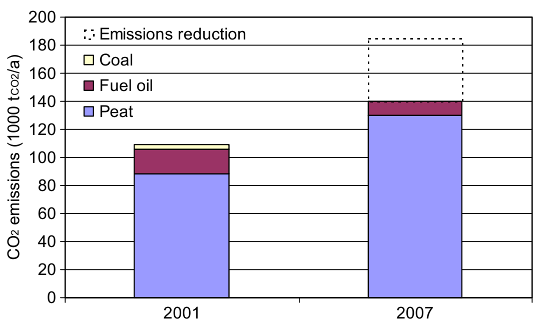

Fig. 6. Carbon dioxide emissions from the case mill power plant in 2001 and 2007 including estimated emissions reduction.

product portfolio and production rates in the case mill without significant changes in the total heating demand.

4. Discussion

The case study includes improvement opportunities that fall into two main groups. Firstly, the heat transfer rate in heat recovery is maximised by enabling as effective condensation as possible, and secondly, the structure of the heat recovery system is modified to ensure that the streams are heated in the economically correct order within the heat recovery system. Since the case study focused only on the heat recovery systems of the paper machines, further possibilities for heat integration can be found by analysing the case mill as a whole. For example, heat from thermomechanical pulping could be transferred to process water after dryer section heat recovery, not before, as is typically done in mills today.

In this study we have applied the thermodynamic simulation models of the heat recovery systems only for an off-line operational improvement and retrofit study. However, these models could also be made suitable for on-line use. Simplified and fast models enable, e.g., fault detection and energy efficiency monitoring. Operational problems in heat recovery could then be detected by following-up the difference between the measured and the modelled output. Furthermore, the energy efficiency of a heat recovery system may be expressed in a new way by comparing the measured heat transfer rate with its objective performance. The simplification can be achieved, for example, by using interpolation based on a database of thermodynamic simulation results calculated in advance or by applying statistical methods and verifying these models with thermodynamic simulation. Diagnostics can be developed even further by constructing neural networks. Although the possibilities to control the dynamic behaviour of HENs was not addressed in this work, further research on the flexibility and controllability of HENs involving humid air is recommended. The development of on-line heat exchanger models for dryer section heat recovery would facilitate the development of new control strategies for HENs.

The case mill follow-up revealed that the design humidity of the dryer section heat recovery systems could not be reached in two paper machines because of operating problems. If this had been known already in the original design, the whole HENs could have been designed differently. For example, the circulation water system for machine hall ventilation could operate at a lower system temperature and the fresh water could be heated with secondary heat from mechanical pulping after dryer section heat recovery. This implies that the design basis for new heat recovery installations should be checked. Further research is required to define which design humidity is appropriate in each case.

The simulation models used in the case study also enable the development of retrofit heat exchanger methods for cases with condensing air. As presented, the use of mathematical programming methods requires a fundamental understanding of the underlying thermodynamics and a careful analysis of the accuracy of the assumptions that are made. The effects of these assumptions can be tested and verified using the thermodynamic simulation models for comparison.

5. Conclusions

This case study revealed a significant potential for profitable energy efficiency improvement in the heat recovery systems of existing paper machines. In the three paper machines examined, the savings correspond to a 7–13% decrease in the specific heat consumption. The simulation models applied in this study open up further possibilities for improving the control and monitoring of the heat recovery systems. The models can also assist in the development future optimisation methods for HENs with condensing air.

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