ANALYTICAL INVESTIGATION OF AIRFLOW PATTERNS WITHIN A PAPER MACHINE HALL

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The results of field measurements conducted in the Umka Cardboard Mill hall indicate a certain potential for heat recovery (approx. 4430 kW) in the air accumulated below the hall ceiling and extracted by the ventilation system. Analytical investigation is carried out for finding out the most favourable locations for heat exchangers' intake connections. Heat exchangers are to be used for waste heat recovery. Focus is laid on determining the convection flow rates along the vertical surfaces of the machine, as well as on defining the characteristics of the thermal plumes created above the machine. These characteristics are calculated according to the plume theory, introducing a virtual point source instead of the real source dimensions. The results of mass and heat balance of the hall are also presented in the paper. The same method can be applied in other paper mills and similar industry premises, where large flat machine surfaces are common. The investigation confirms the assumption that the plume formed above the heat exchangers in the coating drying section is the hottest and with considerable high airflow rate (6720 m^3/h above the hottest heat exchanger). It reaches the ceiling level and spreads horizontally, penetrating the nearby sections along the hall. Therefore, it is suggested that this airflow is the most suitable for heat recovery.

Keywords: cardboard machine hall, mass and heat balance, plume theory, airflow pattern

INTRODUCTION

Large quantities of heat and water vapour are released from the paper machine in the machine hall during paper manufacturing. In addition, considerable amounts of steam can be released from some parts of the machine. The vapour and steam are corrosive and destructive for all surfaces of the machine hall, onto which they condensate, including the cardboard machine itself. During the papermaking process, large amounts of excess moist and corrosive vapour must be extracted and ventilated away, as part of the production process, which is often done by fan units mounted in the ceiling.

The results of field measurements conducted in an Umka Cardboard Mill hall indicate certain potentials for heat recovery (approx. 4430 kW) in the air accumulated below the hall ceiling and extracted by the ventilation system.[1] After having analysed the heat potentials, the application of regenerative pebble-bed heat exchangers was considered.[2] The results of the measurements also show that temperature and moisture distribution vary significantly with space.[1] However, for finding the most favourable locations for heat exchangers' intake connections, additional information is required. To this end, an analytical investigation of the airflow patterns in the hall has been carried out, focus being laid on determining the convection flow rates along the vertical surfaces of the machine, as well as on defining the characteristics of thermal plumes created above the machine. The analysis also includes the results of hall mass and heat balance, providing a better understanding of the nature of air movements within the hall and an insight into the forces governing the movements.

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Problem description

The Umka Cardboard Mill is located 15 km from Belgrade, on the right bank of the Sava river. After machine reconstruction in 2006, the mill's production capacity has been increased up to 75000 t/year. Recycled wastepaper fibres are used as a raw material for stock preparation. Dry saturated steam is a basic secondary energy source in the production process. The steam is produced in natural gas-fired boilers at two pressure levels (*i.e.* 3 and 12 bars). Drying cylinders and steam-air heat exchangers are supplied with the steam at 3 and 12 bars, respectively. On the average, the machine's steam load is of 16 t/h.

The cardboard machine of the Umka Cardboard Mill is accommodated in a 196 m long, 15 m wide and 11 m high production hall. The architectural features of the hall can be considered as a single volume of 32000 m^3, with a base area of 3000 m^2. The hall can be divided into 6 sections, matching 6 different sections of the machine, each with a different technological process, as shown in Figure 1: former, pre-drying, smoothing, after-drying, coating drying and handling sections.

According to on-site observations, field measurements and previous studies, the coating drying section is the largest source of waste heat in the hall and the second largest source of water vapour after the former section. The results of the mass and heat balance of the machine and those of field measurements indicate that this section is characterized by the highest temperatures of heat exchangers' surfaces (IR and impingement dryers) – 190-210 ºC,[1] by a fairly low thermal efficiency – around 33.6%,[2] the highest air temperature – 73 ºC,[1] and the largest quantity of water evaporated from the cardboard – 0.54 kg/s.[2] Based on these results, it can be assumed that the hottest and the highest plume is formed over the coating drying section of the machine. Consequently, the space above this section should be the best location for heat exchangers' intake connections. The results of analytical calculations discussed later in this paper confirm this hypothesis.

Air is supplied to the hall through several openings located on the floor, a passage to the nearby hall located on the side wall and randomly opened windows, as shown in Figure 1. The opened windows are located mostly at the lower levels of the hall (1-2 m above the floor). Moreover, warm air is being injected into the space between the ceiling and roof construction, in order to prevent vapour from condensing on the ceiling surfaces during the winter period. One part of this airflow enters the hall through opening 2, located in the ceiling. The cold air from the floor is heated by the hot machine surfaces. It rises due to buoyancy, settles at some level or reaches the ceiling, where it is extracted by the ventilation system of the hall. The ventilation system consists of a series of axial fans, which extract the hot and moisture air through 43 exhaust openings located in the ceiling, as shown in Figure 1.

Analytical modelling is important because the airflow patterns in a room like this are governed by the mechanical ventilation and convection flows from the heat sources present in the room. The formation of horizontal air layers may be expected. The warmest air layers are at the top and the coolest air layers are at the bottom. The air moves easily within a horizontal layer, however, transportation between the layers needs a stronger force (*i.e.* a buoyancy force).[3]

RESULTS OF AIR MASS AND HEAT BALANCE

For a good understanding of the flow structure, it is very important to consider the air mass and heat balance of the hall. The calculations were based on the results of field measurements conducted in a winter day under steady-state conditions.[1] This is a reasonable assumption, since the period of time spent on measurements was negligible, compared to the production process period which lasted for a few days. The capacity of the ventilation system is approx. 165000 Nm^3/h, since some of the ceiling mount ventilators are out of order. The mean temperature and relative humidity of the extract air are of 60 ºC and 35%, respectively. As a result, 9.9 MW of the heat leaves the hall with the extract air, the mass flow rate of which is of 57 kg/s. The characteristics of the supply air are presented in Table 1 and the results of air mass balance – in Figure 2.

The total mass flow rate of the supply air is of 59 kg/s, which means that approx. 2 kg/s of air leaves the hall through the gaps in the building construction. The total airflow heat input and output are of 3028 kW and 9900 kW, respectively. The difference of 6872 kW should correspond to the heat released from the heat sources in the hall. Comparing the moisture content of the supply and extract air in the mass balance, a difference of 1.95 kg/s was found.

The results of air mass balance indicate that approx. 73% of the total mass flow rate enters the hall through the openings located in the handling section. Since the extract openings are uniformly spaced along the ceiling, longitudinal air movements from the handling to the former section are expected.

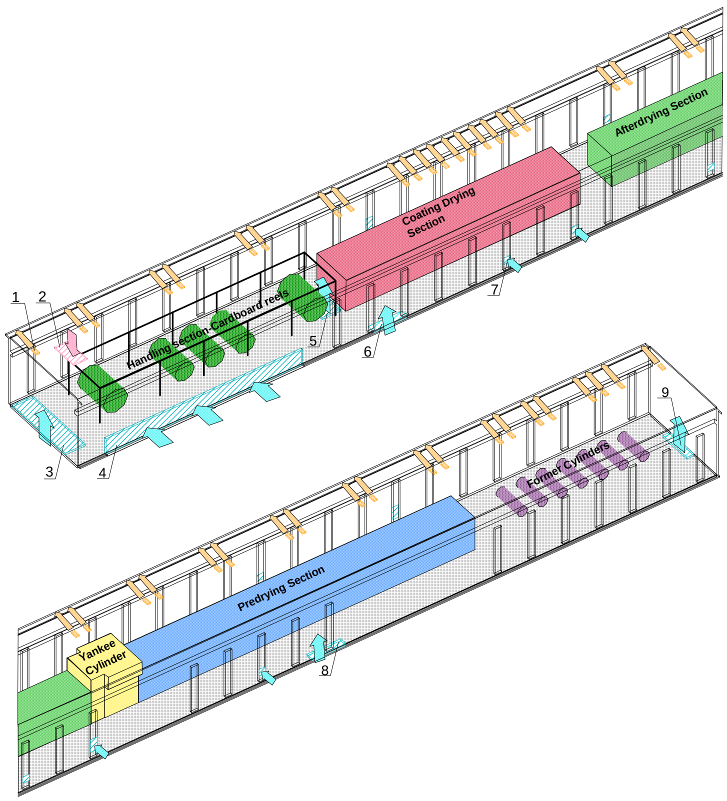

Figure 1: Characteristic sections of the cardboard machine and hall with locations of supply and exhaust openings: 1. Existing exhaust openings; 2. Opening 2; 3. Opening 1; 4. Opening 3; 5. Opening 7; 6. Opening 4; 7. Windows; 8. Opening 5; 9. Opening 6

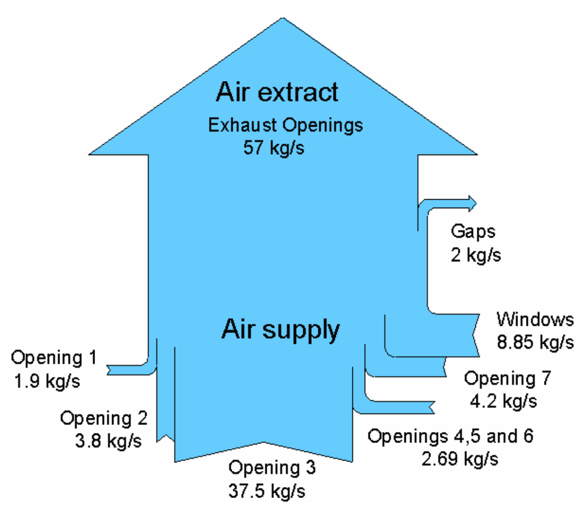

Figure 2: Mass flow diagram; approx. 64% of the total mass flow rate enters the hall through Opening 3; air leakages include 3.5% of the total mass flow rate

Table 1
Characteristics of supply air

Characteristics \ Opening	1	2	3	4	5	6	7	Windows	Total
Opening area, m^2	33	8	125	5	5	5	30	2.35	213
Temperature, ºC	25	50	26.7	26.8	25.8	16.5	26.5	4	-
Relative humidity, %	33	35	38.6	32	35	65	33	78	-
Velocity, m/s	0.05	0.45	0.3	0.04	0.08	0.34	0.12	3	-
Mass flow rate, kg/s	1.9	3.8	37.5	0.23	0.46	2	4.2	8.85	59
Heat flow rate, kW	80	455	2081	10	20	73	186	123	3028

VERTICAL CONVECTIVE FLOWS – PLUME THEORY

If an object is warmer than the surrounding air, the air is heated and the warm air moves upwards, due to buoyancy. The air current created in this way is called natural convection flow or plume. Generally, heat transfer involving motion of air caused by a difference in density is called natural or free convection. As a result of free convection, a flow appears in the form of a boundary layer, moving along the surface or as a thermal plume above a surface. The amount of air in the convection flow also increases with height, due to entrainment of the surrounding air (Fig. 3). Because the driving force in the convection flows is the buoyancy force caused by the density difference, a temperature gradient in the room influences the plume rise height.[3] Therefore, information on the thermal plume characteristics is essential for determining the airflow patterns in a room and for optimizing the waste heat recovery system.

In this case, warm objects, such as machine hood or heat exchangers of the coating drying section, create rising convection flows. Depending on the power and geometry of the heat source, the convection flows will rise all the way to the ceiling or will settle at a lower height.

Calculations have been carried out for both sides of the machine hood. If the flow is assumed to be turbulent, the following basic features can be defined:[4]

-Maximum velocity

$$v_z = 0.1(\Delta T z)^{1/2} \quad \text{[m/s]} \qquad (1)$$

-Thickness of boundary layer

$$\delta_z = 0.11 z^{0.7} \Delta T^{-0.1} \quad \text{[m]} \qquad (2)$$

-Airflow rate

$$q_{v,z} = 0.00275 \Delta T^{0.4} z^{1.2} \quad \text{[m}^3\text{/sm]} \qquad (3)$$

where ΔT is the temperature difference between the surface and the surrounding air, and z is the height from the bottom of the surface.

Information on the convective heat loads from the hood is also necessary to get a reasonable estimation of the thermal plume characteristics above the hood. This load can be estimated from the energy consumption of the heat source Φ_{tot} by $\Phi = k \cdot \Phi_{tot}$, where the value of coefficient k is 0.3-0.5 for large machines and components.[3] In this study, the convective heat loads are calculated using the correlation found in literature,[5] by taking the mean value of the hood and air temperature:

$$\Phi = \overline{\alpha} l \Delta \overline{T} \quad \text{[W/m]} \qquad (4)$$

where $\overline{\alpha}$ – convective heat transfer coefficient for the mean temperature of the surrounding air [W/m^{2}K], l – characteristic length, in this case hood height [m], $\Delta\overline{T}$ – mean temperature difference between the hood and the surrounding air [K].

Further on, the convective heat transfer coefficient can be expressed as:[5]

$$\overline{\alpha} = \frac{\overline{Nu}\,\overline{\lambda}}{l} \quad \text{[W/m}^2\text{K]} \qquad (5)$$

where $\overline{Nu}$ – Nusselt number for the mean temperature of the surrounding air [-], $\overline{\lambda}$ – conductive heat transfer coefficient for the mean temperature of the surrounding air [W/mK], l – characteristic length, in this case hood height [m].

For turbulent flows along flat vertical surfaces, the Nusselt number is a function of other dimensionless numbers (Pr, Gr).[5]

Analytical equations to calculate velocity, temperature and airflow rate in thermal plumes over the point and line heat sources with given heat loads were derived based on momentum and energy conservation equations presented in many textbooks.[4] The characteristics of thermal plumes above line sources can be defined as:[4]

-Centreline velocity

$$v_z = 0.067\Phi^{1/3} \quad [\text{m/s}] \quad (6)$$

-Centrelines excess temperature

$$\Delta\theta = 0.094\Phi^{2/3} z^{-1} \quad [^0\text{C}] \quad (7)$$

-Airflow rate

$$q_{v,z} = 0.013\Phi^{1/3} z \quad [\text{m}^3/\text{sm}] \quad (8)$$

However, the heat sources are seldom a point, a line or a plane vertical surface. Moreover, the convection flows from the horizontal surfaces are very difficult to determine in the same way as for point, line or vertical surfaces. The reason is that the flows behave in a very unstable way and leave the flat surface from different positions at different times, partly depending on the total air movement in the room.[4] The most common approach to account for the real source dimensions is to use a virtual point source located along the plume axis, at a distance z_0 on the other side of the real source surface, as presented[4] in Figure 4.

Several methods for determining the location of the virtual source have been reported in literature.[3,4] For this calculation, it is assumed that the airflow rate at the top of the hood corresponds to the convective airflow from the vertical surfaces of the hood. For a given convective heat flux, the location of the virtual source can be calculated from equation (8), as well as the other parameters from equations (6) and (7). The thermal plumes are influenced by temperature stratification, since the driving force of the plume is the temperature difference between the plume and the surroundings. When the difference diminishes, the plume will disintegrate and spread horizontally in the room. The plume will settle between its equilibrium height z_t, where the temperature difference between the plume and the surrounding air disappears, and maximum height $z_{\max}$, where air velocity in the plume drops to zero. For a line source, the following formulae can be written:[4]

- Maximum height

$$z_{\max} = 0.51\Phi^{1/3}\left(\frac{d\theta}{dz}\right)^{-1/2} \quad [\text{m}] \quad (9)$$

- Equilibrium height

$$z_t = 0.35\Phi^{1/3}\left(\frac{d\theta}{dz}\right)^{-1/2} \quad [\text{m}] \quad (10)$$

where Φ [W/m] and $\frac{d\theta}{dz}$ [^{0}C/m] are convective heat flux from the hood surfaces and vertical temperature gradient, respectively.

SUMMARY OF RESULTS AND DISCUSSION

The results of the analytical investigation are derived for each characteristic section of the hall, since the geometry and temperature of the machine hood vary considerably from section to section.

The temperature of the former cylinders does not differ much from the surrounding air temperature,[1] and the amount of heat released in the press section corresponds to the heat absorbed by the water contained in the pulp during the change of phase (*i.e.*, latent heat). That is why, the former section can be considered as thermally neutral with respect to other sections. The produced cardboard is wound on reels in the handling section, which can be also considered as thermally neutral with respect to other sections.

Air flow rate and convective flux are calculated using equations (3) and (4). The results are obtained for each section and for given hood dimensions. The total air flow rate from the hood top (z = 5 m) in the pre-drying section is of 4.36 m^3/s or 15705 m^3/h, while the total convective flux from the hood in this section is of 255 W/m, or taking into account hood dimensions – of 13 kW. Similarly, the total airflow rate from the hood top in the smoothing section (z = 7.5 m) is of 1.08 m^3/s or 3888 m^3/h. The total convective flux from the hood in this sector is of 1221 W/m or, if taking into account hood dimensions, – of 8.5 kW. The total airflow rate from the hood top in the after-drying section (z =5 m) is of 3.07 m^3/s or 11052 m^3/h, while the total convective flux is of 363 W/m or 11 kW. The total airflow rate from all heat exchangers in the coating drying section (z = 1.75 m) is 1.3 m^3/s or 4,630 m^3/h, and the total convective flux is of approx. 45 kW.

The main characteristic features of the thermal plumes in all sections, such as location of the virtual source, maximum plume height, equilibrium height and airflow rate at maximum plume height, are calculated with equations (8) to (10), and are presented in Table 2.

The total convective heat flow from the machine surfaces in the sections under observation is found to be of nearly 77.5 kW. This amount of heat is negligible compared to the results of the hall heat balance, which imply a 6872 kW total heat release from the heat sources in the hall. The reason for such discrepancies may be the small number of measuring points and, consequently, large errors in deriving the mean values, even if the error could not be big enough to cause such differences. Another reason may be the leakage of hot air from the heat exchangers in the coating drying section into the hall, and from the machine hood into the other sections. Due to the complexity of the production process and of the cardboard machine itself, it is difficult to determine or estimate the rate of air leakage and its influence on the convection flows and overall airflow patterns in the hall. This is going to be a separate study on the detailed energy audit of the cardboard machine. Another possible reason could be the overcapacity of the ceiling mount ventilators, which might cause additional airflow from the bottom to the top of the hall. The present investigation, considering only the airflows occurring due to convective heat transfer from the outer surfaces of the hood to the surrounding air, has provided a good qualitative picture of the overall airflow patterns in the hall and has indicated the possible existence of maintenance (leakage of hot air) and control problems (overcapacity of the ceiling mount ventilators).

The overall convection volume flow rate from the vertical surfaces of the drying hood from equation (3) is of 30600 m^3/h. The airflow rate of the thermal plume created above the hood is increased, due to the entrainment of the surrounding air, resulting in 55000 m^3/h for the maximum plume height.

Table 2
Characteristic features of thermal plumes above the cardboard machine

Section	Pre-drying	Smoothing	After-drying	Total for drying hood	Coating drying*
Location of virtual source z_0, m	1	1.14	1.13	-	0.25
Vertical temperature gradient $d\theta/dz$, ^{0}C/m	3.37	3.25	3.16	-	3.88
Maximum plume height z_{max}, m	1.73	2.95	2.01	-	3.31
Equilibrium height z_t, m	1.19	2.03	1.38	-	2.28
Airflow rate at the maximum plume height, m^3/h	25200	10080	19692	55000	6720

*The results for one of three heat exchangers (impingement dryers) whose surfaces are the hottest

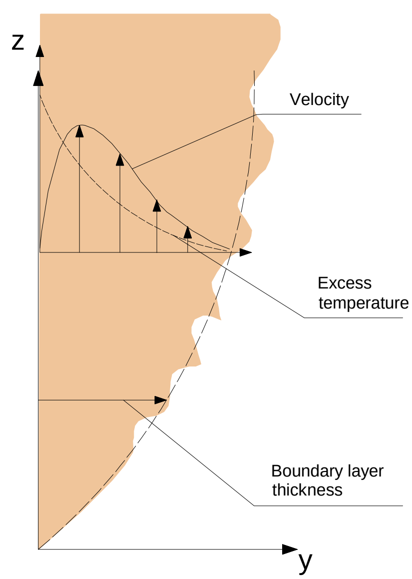

Figure 3: Natural convection flow along a vertical surface;[4] the amount of air in the convection flow increases with height, due to entrainment of the surrounding air

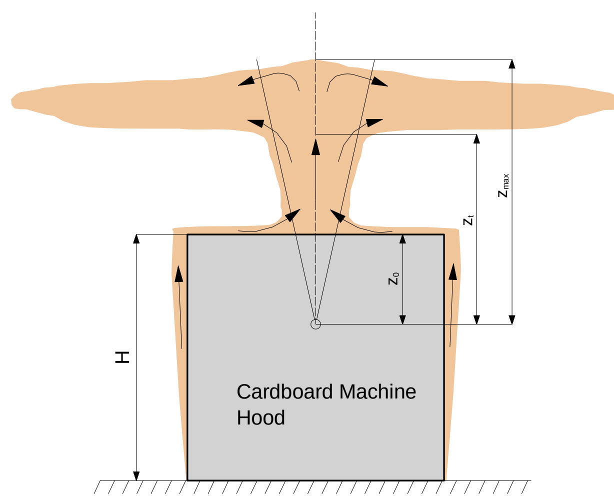

Figure 4: Thermal plume above the hood positioned as virtual source z_0; the plume settles between its equilibrium height z_t, where the temperature difference between the plume and surrounding air disappears, and maximum height z_{max}, where air velocity in the plume drops to zero

The maximum airflow rate over the hottest heat exchanger in the coating drying section is of 6720 m^3/h. The airflow rates created along the hall walls are supposed to be much smaller, due to a lower surface-air temperature difference.[1] For the simplicity of the analysis, it is justified to omit these airflows.

A maximum plume height of 3.3 m above the heat exchanger top was found in the coating drying section. The remaining plumes have smaller heights and therefore do not reach the ceiling. The investigation confirms the assumption that the hottest and the highest plume is formed above the heat exchangers in the coating drying section. It reaches the ceiling level and spreads horizontally, penetrating the nearby sections along the hall. Therefore, this plume is the most suitable for heat recovery.

CONCLUSIONS

The results of the analytical investigation presented in this paper are in good agreement with previous studies and with the on-site observations made in the Umka Cardboard Mill hall. The same method can be applied in other paper mills and similar industry premises, where large flat machine surfaces prevail.

Several conclusions are derived from the analytical investigation, as follows:

- The airflow patterns in the hall are governed by vertical convection flows, due to buoyancy and mechanical ventilation. A leakage of hot air into the hall from some parts of the machine also alters the thermal plumes created above the machine.
- Air velocities are fairy low (below 1 m/s) in both horizontal and vertical direction, except in the vicinity of windows.
- The investigation also confirms that the plume formed above the heat exchangers in the coating drying section is the hottest and the highest (3.3 m above the hottest heat exchanger top) with a considerably high airflow rate (6720 m^3/h above the hottest heat exchanger). It reaches the ceiling level and spreads horizontally, penetrating the nearby sections along the hall. Therefore, this airflow appears as the most suitable for heat recovery.
- Besides utilizing the waste heat from the ventilation extract air in the hall, it is possible to reduce the energy losses from the production process, by minimizing air leakages and by improving insulation around the machine hood and the heat exchangers.

Further investigations with additional measurements and numerical simulation are needed for determining the influence of air leakages on the overall airflow patterns in the hall.

REFERENCES

[1] M. Stanojević, G. Jankes, M. Stamenić, N. Tanasić *et al*., Project No. EE 233009, *Utilization of solid waste and waste heat in process industry*, (in Serbian), Faculty of Mechanical Engineering, Belgrade, 2009, pp. 6-20.

[2] M. Stanojević, G. Jankes, M. Stamenić, N. Tanasić *et al*., Umka Cardboard Mill Project No. EE 233009, *Utilization of solid waste and waste heat in process industry*, (in Serbian), Faculty of Mechanical Engineering, Belgrade, 2007, pp. 13-30.

[3] H. Skistad, E. Mundt, P. V. Nielsen, K. Hagström and J. Railio, in “Displacement ventilation in non-industrial premises”, Rehva, Brussels, 2004, pp. 9-24.

[4] H. Goodfellow and E. Tähti, in “Industrial Ventilation Design Guidebook”, Academic Press, San Diego, USA, 2001, pp. 517-541.

[5] Dj. Kozić, B. Vasiljević and V. Bekavac, in “Thermodynamics Handbook”, (in Serbian), Faculty of Mechanical Engineering, University of Belgrade, Belgrade, 2004, pp. 136-141.