

Article

Control of Winter Surface Condensation in a Tissue Paper Production Hall: A Measurement-Calibrated, TMY-Based Comparative Assessment

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Abstract

Winter surface condensation on the walls and false ceiling of paper production halls degrades the building fabric, promotes mould growth and disturbs the process environment, yet the choice between the available remedies—envelope insulation, ventilation intensification and mechanical dehumidification—is rarely supported by a common quantitative basis. This paper reports a measurement-calibrated assessment of the three solutions for a tissue mill in Petrești, Romania, housing two tissue machines in a 34 800 m³ hall enclosed by an uninsulated 200 mm reinforced-concrete wall ($U = 3.89 \text{ W m}^{-2} \text{ K}^{-1}$). Two measurement campaigns established an exhaust airflow of 85500 m³ h⁻¹ (2.46 h⁻¹) and a process-driven humidity-ratio increment across the hall of $\Delta x = 8.0 \text{ g kg}^{-1}$ with one machine running and 12.2 g kg⁻¹ with both, corresponding to a moisture uptake of 18.4 and 27.2 t d⁻¹ respectively—3.7 to 3.9 times the value declared in the process water balance, which indicates substantial hood spillage. A one-dimensional surface-temperature model coupled to a psychrometric zone model was driven by an 8760 h typical meteorological year, yielding 5053 cold-season hours (outdoor temperature below 12 °C). On the existing wall, the internal surface resistance carries 50.6% of the total thermal resistance, so the surface temperature closely tracks the outdoor temperature and condensation occurs in 91–100% of cold-season hours for any indoor temperature between 20 and 35 °C; eliminating it by heating alone would require 45–52 °C indoors. Adding 50 mm of external EPS removes condensation in all 5053 h once the hall is held at 27 °C; at the recommended minimum design temperature of 28 °C, which keeps a margin of at least 1.5 K over the season, it saves about 690 MWh per season and the cost is recovered in 1.9–3.1 years, whereas ventilation intensification (2.3× the present airflow) and hybrid coil-plus-desiccant dehumidification achieve the same technical result at ten-year costs of 3.6–4.1 M€. A sensitivity analysis confirms this ranking. Once the wall is insulated, the dehumidification duty of the plane wall falls to zero and the airflow required to prevent condensation drops below the installed capacity, so the existing ventilation need not be increased: insulation does not compete with the alternatives, it makes them unnecessary.



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1. Introduction

Surface condensation inside industrial buildings is the visible outcome of a simple inequality—the internal surface temperature of the envelope falls below the dew point of the

adjacent air—but its consequences are neither simple nor cheap. Persistent liquid water on walls and ceilings corrodes reinforcement and fixings, saturates finishes, drips onto product and equipment, and sustains microbial colonisation of the surfaces it wets. Aelenei and Henriques [1] showed that condensation risk analysis on building envelopes must be treated as a coupled thermal and hygric problem rather than as a check on a single design condition, because the risk is governed by the joint distribution of surface temperature and vapour content over the whole season. Xue et al. [2] demonstrated experimentally that moisture accumulation in envelopes is self-reinforcing: once a surface has been wetted, its capacity to release the accumulated moisture is reduced, so condensation and mould growth become progressively easier to trigger. Orlik-Koźdoń [3] quantified the sensitivity of both water-vapour condensation and mould germination to indoor climatic conditions, confirming that the indoor humidity level, not the indoor temperature alone, is the controlling variable. The review of moisture ingress in building envelope materials by Lopez-Carreón et al. [4] places surface condensation within the broader family of moisture transport mechanisms and stresses that mitigation must act either on the vapour supply or on the surface temperature, the material itself offering little defence once wetting is established.

The design rules that follow from this understanding are codified in EN ISO 13788 [5], which defines the internal surface temperature required to avoid critical surface humidity through the temperature factor f_{Rsi} , and in EN ISO 6946 [6], which fixes the surface resistances used to compute it. Ilomets and Kalamees [7] applied the same criterion to thermal bridges and showed that the criticality of a junction is decided by the ratio between the internal surface resistance and the total resistance of the assembly: where that ratio is large—that is, where the construction itself is thermally weak—the surface temperature is effectively bound to the outdoor temperature and no realistic increase in indoor temperature can lift it above the dew point. This mechanism, well documented for thermal bridges in insulated buildings, applies a fortiori to entire uninsulated elements such as the bare reinforced-concrete walls typical of 1970s Eastern European industrial construction. Oladokun et al. [8] coupled computational fluid dynamics with a mould growth model and confirmed that the local surface temperature field, rather than the bulk room condition, determines where colonisation begins.

Large industrial and sports halls introduce a further complication: they are strongly stratified. Karpuk et al. [9] measured vertical air temperature gradients in a large industrial hall and found differences in the order of 10 K between the occupied zone and the roof zone, so a single well-mixed node is a poor representation of the enclosure. Lin et al. [10] extended this observation to humidity, showing for ice arenas that thermal and hygric stratification are statistically distinct and that the upper zone can be simultaneously the warmest and the most vapour-laden part of the volume—precisely the combination that maximises condensation risk on the roof and on the upper part of the walls. Limane et al. [11] reached comparable conclusions for indoor swimming pools by CFD, and the systematic review of swimming facilities by Smedegård et al. [12] identified envelope condensation control as the dominant constraint on the ventilation and dehumidification strategy of high-latent-load enclosures. Swimming halls are, in this respect, the closest well-studied analogue of a tissue mill: a large volume with an intense, continuous and spatially concentrated internal moisture source.

In the pulp and paper sector, the moisture source is the drying section. Hubbe [13] reviewed the energy efficiency challenges of pulp and paper manufacturing and identified drying as the single largest consumer, while Laurijssen et al. [14] benchmarked energy use at process-unit level across European mills and reported wide dispersion in the performance of the drying and hood-exhaust systems. Tissue machines are the most extreme case, combining a Yankee cylinder with a high-temperature hood; Yan and Li [15] simulated

heat-pump recovery from paper exhaust hoods and confirmed that the exhaust stream carries a large, low-grade thermal and latent load. Condensation on the building structure is a recognised consequence of inadequate machine room ventilation: Ghosh [16] documented, in two case studies, fog, condensation and water drip-back onto the finished paper, corrosion and poor working conditions, all traced to ventilation systems that were not sized on the actual heat and moisture loads. The TAPPI machine room ventilation guidelines, presented by Nielsen [17] and issued as TIP 0404-50 [18], accordingly size machine room ventilation on these loads and limit the humidity rise above ambient to about 7 g kg^{-1} at the operating floor and about 21 g kg^{-1} beneath the roof. When the hood capture efficiency falls short—through leakage, fouling or under-sized extraction—the uncaptured fraction is released directly into the hall and becomes an internal moisture gain of several tonnes per day. Despite the industrial importance of the problem, published quantitative assessments of condensation control in paper mill halls are scarce, and the hood spillage that drives it is seldom reconciled against an independent measurement of the hall air moisture balance.

Where the moisture cannot be removed at source, the vapour content of the air must be lowered. Ma et al. [19] formulated the choice of supply-air humidity as an optimisation over spatially distributed moisture demands and showed that, where the requirement is set by the most critical location rather than by the mean condition, the design target must be derived from that location—for a condensation-prone hall, from the coldest surface. Mechanical dehumidification by cooling coil is limited by the apparatus dew point: the coil cannot dry air below the saturation humidity ratio corresponding to its surface temperature, whatever the airflow. Solid desiccant wheels remove this ceiling by adsorbing vapour directly on a hygroscopic matrix. Xu and Chen [20] reviewed low dew-point desiccant wheel systems and mapped the trade-off between attainable dew point, regeneration temperature and specific energy; Chen et al. [21] demonstrated cascaded wheels delivering supply air below 0°C dew point; Guan et al. [22] and Ma et al. [23] reported on-site performance of deep-dehumidification wheel installations in lithium-battery plants, where regeneration heat dominates the energy bill. Hybrid arrangements, in which a cooling coil performs the bulk moisture removal and a wheel finishes the drying, were analysed by Jia et al. [24] and, for buildings with high latent loads, by Comino et al. [25], who reported substantial reductions in regeneration energy relative to a wheel operating alone. These studies establish the technical feasibility of deep dehumidification but also its cost: regeneration heat is the most expensive energy in the chain, and its magnitude scales with the moisture actually passed to the wheel.

Finally, a credible comparison between solutions that behave differently across the weather spectrum cannot rest on a single design point. Sun et al. [26] discussed the generation and use of typical meteorological years for coupled building simulation, and hourly TMY data are now routinely available from the PVGIS service of the Joint Research Centre [27] for any European location. Hour-by-hour evaluation over a full year is the only way to convert a physical criterion into an operational metric—hours of condensation per season—and to weight energy consumption by the frequency of the conditions that cause it. Freimanis et al. [28] assessed seventeen insulation systems applied to solid masonry walls and found that the hygrothermal risk transferred to the wall depends strongly on where the insulation is placed, external systems being the safest option for an enclosure whose inner face must remain dry. On the economic side, Basińska et al. [29] showed for building retrofit that insulation measures are frequently justified on energy grounds alone, independently of any secondary benefit.

The present work brings these strands together for a specific and, in Eastern Europe, widespread configuration: a 1975 reinforced-concrete tissue mill hall with an uninsulated envelope, a heavily loaded ventilation system and chronic winter condensation on the upper

walls and the false ceiling. The objectives are (i) to close the hall moisture balance against measurements and quantify the discrepancy with the declared process water balance, (ii) to identify the physical mechanism responsible for winter condensation, (iii) to evaluate three types of solutions—envelope insulation, ventilation intensification and mechanical dehumidification—on a single hourly TMY basis using identical technical, energetic and economic criteria, and (iv) to establish whether these solutions are alternatives or whether one of them dominates.

2. Material

2.1. The Building and the Process

The facility is a thin-paper (tissue) mill located in Petrești, Alba County, Romania (45.921° N, 23.561° E, 278 m a.s.l.), commissioned in 1975. The structure is reinforced concrete, partly precast, with a 30 m central span housing the paper machines and two 6 m lateral spans on four levels. The central span is single-storey plus a technological floor at +6 m; the reinforced-concrete slab comprises a 280 mm plate, 300 × 800 mm secondary ribs and 500 × 1200 mm main beams at 6 m centres. The analysis boundary is the production hall itself: a main volume of 80 × 30 × 14.5 m (2400 m², 34,800 m³) and an adjoining auxiliary volume of 23 × 30 × 6 m (690 m², 4140 m³), giving a total enclosed volume of 38,940 m³ and a total external wall area of 3190 m² (220 m perimeter × 14.5 m height).

The envelope is a 200 mm reinforced-concrete wall without insulation. With $\lambda = 2.3 \text{ W m}^{-1} \text{ K}^{-1}$, $R_{si} = 0.13 \text{ m}^2 \text{ K W}^{-1}$ and $R_{se} = 0.04 \text{ m}^2 \text{ K W}^{-1}$ [6], the total thermal resistance is $0.257 \text{ m}^2 \text{ K W}^{-1}$ and the thermal transmittance $3.89 \text{ W m}^{-2} \text{ K}^{-1}$. The intrinsic resistance of the concrete layer alone is $0.087 \text{ m}^2 \text{ K W}^{-1}$ —one third of the total—so the assembly is, thermally, little more than a pair of surface films. Over the central span the roof is carried by steel lattice trusses; the composition of the roof closure above them and the presence of any insulation are not documented in the available documentation. Below the trusses, a steel-sheet false ceiling separates the hall from a plenum ventilated with outdoor air (Section 2.2). The axial exhaust fans of the hall are mounted below the false ceiling and discharge directly outdoors, without connection to the plenum.

Production is carried out on two tissue machines, denoted PM6 and PM7 in the ventilation documentation. The process is water-intensive; part of the process water reaches the hall air as vapour. The process water balance supplied by the operator is summarised in Table 1. Only 4–5 t d^{−1} from PM6 and 1–2 t d^{−1} from PM7 is attributed to the hall air, with the bulk of the evaporated water being extracted through the machine hoods (65–70 and 100–105 t d^{−1} respectively) and, for PM7, through a dedicated mist extraction system (10–15 t d^{−1}).

Table 1. Declared process water balance of the two tissue machines.

Stream	Unit	PM6 (OVER)	PM7 (TOSCOTEC)
Water leaving with the paper	t d ^{−1}	2–3	4–5
Vapour extracted by the hood	t d ^{−1}	65–70	100–105
Vapour released into the hall	t d ^{−1}	4–5	1–2
Vapour extracted by mist extraction	t d ^{−1}	—	10–15

2.2. The Ventilation System

Fresh air is supplied along the long sides of the hall at machine level and, separately, into the false-ceiling plenum; air is extracted from the short side of the hall immediately below the false ceiling. Each machine is served by its own supply system. The PM6 supply-air-handling unit is fitted with a steam heating coil which is currently inoperative:

the steam supply and condensate lines have been cut, and the coil is not connected to any heat source. The PM7 supply system incorporates two recuperative heat exchangers that preheat the fresh air with the hot exhaust from that machine—one for the air delivered to the hall, one for the air delivered to the plenum.

The hall supply fans are centrifugal CHB 49 (CBI Italy) units, and the plenum supply fan is a CHB 36 (CBI Italy). From the manufacturer's characteristic curves, and allowing for the elevated system resistance caused by paper-dust deposits on intake filters and ductwork, the fans are estimated to operate at 600–750 rpm and to deliver 40,000–45,000 m³ h^{−1} each, i.e., 80 000–90,000 m³ h^{−1} in total. Extraction is provided by two fans per machine (ø1200 mm) plus three lateral fans (ø1000 mm). The plenum supply fan was observed running at 300 rpm, below the lowest speed shown on its characteristic diagram, corresponding to an estimated 10,000–15,000 m³ h^{−1}.

A cross-section of the ventilation arrangement is shown in Figure 1.

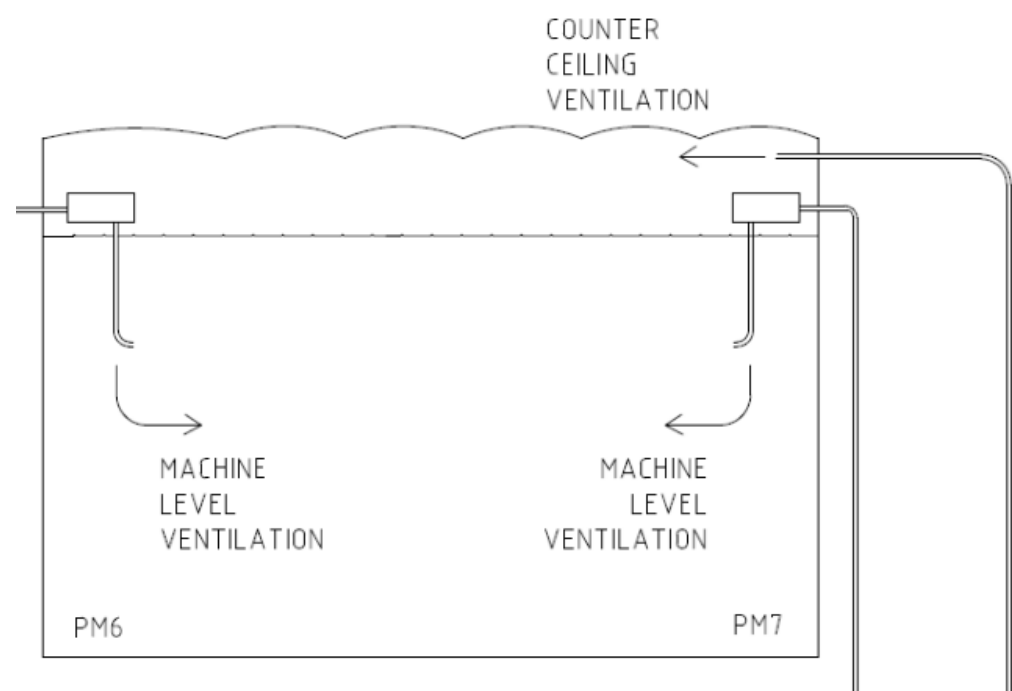


Figure 1. Ventilation system of the production hall, transverse section: fresh air is delivered both at machine level and into the false-ceiling plenum.

2.3. Efficiency Measures Considered

Five measures were taken into account at the outset.

Source reduction. Improving hood capture and machine exhaust is the only measure that addresses the cause rather than its effects, and it is therefore the first priority. It is recommended to the operator as a primary action, in parallel with the envelope insulation: an audit of the PM6/PM7 hoods, establishing the hood air and water balance, the hood balance and the zero-pressure level, should be carried out, along with capturing the open ends of the hoods and ensuring any leakage from the exhaust ductwork is driven back into the hall, followed by sealing the hoods. The audit requires process expertise specific to the machine suppliers and falls outside the scope of the present study; its expected effect is quantified in Section 4.9.

External envelope insulation. Adding expanded polystyrene (EPS, $\lambda = 0.038 \text{ W m}^{-1} \text{ K}^{-1}$) to the outer face of the concrete wall raises the internal surface temperature without altering the indoor moisture balance.

Ventilation intensification. Winter outdoor air carries $1\text{--}3\text{ g kg}^{-1}$ of moisture, i.e., it is drier than the air a cooling coil can produce. Increasing the airflow dilutes and exports the process moisture and lowers the indoor dew point without any dehumidification equipment, at the cost of heating the additional fresh air.

Mechanical dehumidification. Three configurations were considered for treating the upper-zone air: a cooling coil alone, a solid desiccant wheel alone, and a hybrid arrangement in which the coil performs the bulk drying and the wheel finishes it (Figure 2). The coil can be fed either by a dry cooler in free-cooling mode or by a chiller; the wheel is regenerated with air heated to $120\text{--}150\text{ }^{\circ}\text{C}$, potentially recovered from the machine exhaust.

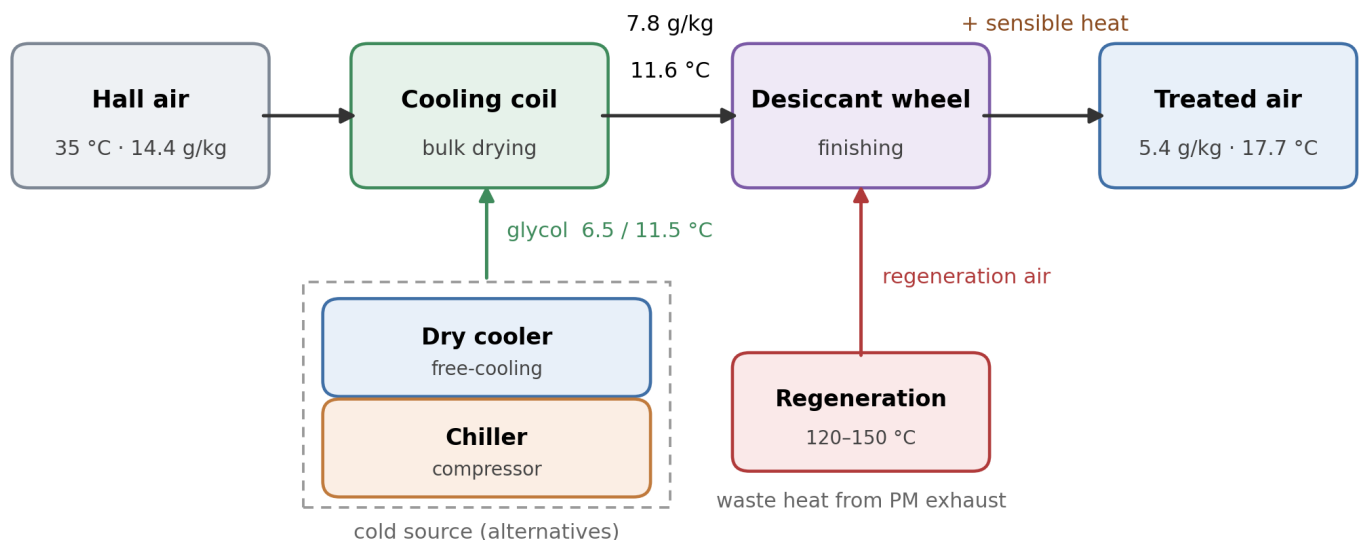


Figure 2. Hybrid dehumidification arrangement: cooling coil for bulk moisture removal, desiccant wheel for finishing, with alternative cold sources and waste-heat regeneration. State points correspond to the winter case.

False-ceiling heating. The temperature of the false ceiling follows the temperature of the plenum air above it. At present, only PM7 contributes recovered heat to the plenum. Extending the same recuperative arrangement to PM6 would widen the achievable range for the ceiling temperature. For an estimated plenum airflow of $10,000\text{--}15,000\text{ m}^3\text{ h}^{-1}$ ($\approx 15,000\text{ kg h}^{-1} = 4.16\text{ kg s}^{-1}$) heated from $-15\text{ }^{\circ}\text{C}$ to $35\text{ }^{\circ}\text{C}$, the required thermal power is $\approx 230\text{ kW}$. Because the source is the hot exhaust of PM6, this measure does not increase the site energy consumption.

3. Method

3.1. Psychrometric Relations

The saturation vapour pressure was computed with the improved Magnus form of Alduchov and Eskridge [30]:

$$p_{\text{sat}}(t) = 610.94 \cdot \exp[17.625 t / (t + 243.04)] \quad [\text{Pa}] \quad (1)$$

with the partial vapour pressure $p_v = (\phi/100) p_{\text{sat}}$, the humidity ratio

$$x = 0.622 p_v / (p - p_v) \quad [\text{kg kg}^{-1}] \quad (2)$$

and the dew-point temperature obtained by inversion of (1),

$$t_d = 243.04 \alpha / (17.625 - \alpha), \quad \alpha = \ln(p_v / 610.94) \quad [^\circ\text{C}] \quad (3)$$

The moist-air density and specific enthalpy were evaluated as

$$\rho = [p / (R_a T)] \cdot (1 + x) / (1 + 1.608 x) \quad [\text{kg m}^{-3}] \quad (4)$$

$$h = c_{pa} t + x (r_v + c_{pv} t) \quad [\text{kJ kg}^{-1} \text{ of dry air}] \quad (5)$$

with $R_a = 287.055 \text{ J kg}^{-1} \text{ K}^{-1}$, $c_{pa} = 1.006 \text{ kJ kg}^{-1} \text{ K}^{-1}$, $c_{pv} = 1.863 \text{ kJ kg}^{-1} \text{ K}^{-1}$ and $r_v = 2500 \text{ kJ kg}^{-1}$. A barometric pressure of 101,325 Pa was used for the single-point calculations (Sections 4.2 and 4.3); the hourly analysis (Section 3.6) used the hourly barometric pressure of the typical meteorological year.

3.2. Moisture Balance of the Hall

The water taken up by the hall air follows from the exhaust airflow and the increase in the humidity ratio between supply and exhaust:

$$\dot{m}_a = \dot{V}_p / (1 + x_{\text{exh}}), \quad \dot{m}_w = \dot{m}_a \cdot \Delta x, \quad \Delta x = x_{\text{exh}} - x_{\text{sup}} \quad [\text{kg h}^{-1}] \quad (6)$$

Exhaust airflows were obtained from velocity traverses at each fan, using the measured duct diameters. Because the process releases essentially the same quantity of water in summer and in winter, Δx was treated as a property of the process and of the installed airflow, and was carried over unchanged to the winter analysis. This assumption rests on the composition of the moisture uptake. About three quarters of it (≈ 20 of 27.2 t d^{-1}) is drying vapour escaping from the hoods, set by the production rate and the hood air balance rather than by the state of the hall air. Two secondary effects act in opposite directions in winter. At the same volumetric exhaust flow, the denser winter air carries about 5% more dry air, which lowers Δx . The drier hall air, with a vapour partial pressure of 2.29 kPa against 4.13 kPa in summer, increases the evaporation from the open water surfaces (the 7 t d^{-1} of Table 1) by 22–56% for surfaces at 50–40 °C. Combined, the two effects give a winter increment of 12.3–13.3 g kg^{−1}, i.e., +1 to +9% relative to the summer value; a wider range is examined in Section 4.10. The same reasoning was applied to the sensible temperature rise in the air across the hall, ΔT .

3.3. Surface Condensation Criterion

For steady one-dimensional heat flow, the total thermal resistance and the internal surface temperature of an opaque element are

$$R_{\text{tot}} = R_{\text{si}} + \sum d_i / \lambda_i + R_{\text{se}}, \quad U = 1 / R_{\text{tot}} \quad (7)$$

$$t_{\text{si}} = t_i - (R_{\text{si}} / R_{\text{tot}})(t_i - t_e) \quad (8)$$

Condensation is predicted whenever $t_{\text{si}} < t_d$. Equation (8) can be rewritten as a weighted mean,

$$t_{\text{si}} = (1 - f) t_i + f t_e, \quad f = R_{\text{si}} / R_{\text{tot}} \quad (9)$$

in which f is the fraction of the total temperature drop consumed by the internal surface film. For the existing wall, $f = 0.13 / 0.257 = 0.506$; adding 50 mm of EPS raises R_{tot} to $1.573 \text{ m}^2 \text{ K W}^{-1}$ ($U = 0.636 \text{ W m}^{-2} \text{ K}^{-1}$, a reduction of 84%) and lowers f to 0.083.

The EPS thickness required to avoid condensation with a safety margin ΔT follows from the condition $t_{si} \geq t_d + \Delta T$:

$$R_{\text{tot,nec}} = R_{si} (t_i - t_e) / (t_i - t_d - \Delta T) \quad (10)$$

$$d_{\text{EPS}} = (R_{\text{tot,nec}} - R_{si} - d_c / \lambda_c - R_{se}) \cdot \lambda_{\text{EPS}} \quad (11)$$

A margin $\Delta T = 1$ K was adopted. Equation (10) becomes singular as t_d approaches t_i : when the indoor dew point exceeds the indoor air temperature, no thickness of insulation can prevent condensation, because the surface temperature can never exceed the air temperature that drives it.

3.4. Fog Formation Criterion

Fog inside the hall was checked separately. Mixing a fraction f_c of cold outdoor air, infiltrated through doors and leaks or introduced unheated, with warm humid indoor air gives

$$t_{\text{mix}} = f_c t_e + (1 - f_c) t_i, \quad x_{\text{mix}} = f_c x_e + (1 - f_c) x_i \quad (12)$$

and fog appears if $x_{\text{mix}} > x_{\text{sat}}(t_{\text{mix}})$, which is possible even when both components are unsaturated because the saturation curve is convex.

3.5. Zone Model and Parametric Analysis

The hall was represented by two zones—a lower zone at machine level and an upper zone beneath the false ceiling—because vertical stratification of both temperature and humidity is well established for enclosures of this size [9,10]. The process moisture is released in the lower zone, rises with the thermal plumes of the machines and accumulates in the upper zone, where the exhaust openings are located; the upper-zone humidity ratio was therefore taken equal to the measured exhaust value. A winter reference case was defined as process moisture release 27.2 t d^{-1} (measured), outdoor air -5°C and 85% RH, lower zone 25°C , upper zone 35°C , and false-ceiling surface 20°C . Each influencing parameter was then varied in turn with the others held at their reference values.

3.6. Hourly Seasonal Analysis

The parametric analysis was extended to a full year using a typical meteorological year generated by PVGIS from the 2005–2023 record for the site coordinates [27], giving 8760 hourly values of dry-bulb temperature, relative humidity and pressure. The cold season was defined by $t_e < 12^\circ\text{C}$, which yields 5053 h per year (58% of the year). The threshold is not a physical limit of condensation but the boundary adopted between the cold (winter) season, which is the subject of this study, and the warm (summer) season. Below about 12°C outdoors the hall must be heated, so its air temperature is controlled and can be prescribed in the model; above it, the process and fan heat ($\Delta T = 8.5$ K) alone bring the hall above 20°C , the hall temperature floats with the outdoor conditions, as in the summer campaigns (Table 2), and a set point can no longer be imposed. The evaluation was therefore restricted to the winter season; the effect of the threshold is examined in Section 4.10. The hourly barometric pressure of the TMY was used in (2) and (3). For each hour the outdoor humidity ratio was computed from (2), with the indoor humidity ratio as $x_i = x_e + \Delta x$, the indoor dew point from (3), the surface temperature from (8), and the hour was flagged as a condensation hour if $t_{si} < t_d$. The metrics reported are the number of condensation hours per season and the minimum margin $t_{si} - t_d$ over the season.

Table 2. Measured moist-air parameters (mean values).

Campaign	Location	t [°C]	ϕ [%]	x [g kg ⁻¹]	ρ [kg m ⁻³]
1 (PM6 only)	Supply	27.0	46	10.0	1.16
1	Exhaust	32.0	60.0	18.0	1.14
1	Inside hall	28.0	70	17.0	1.14
2 (PM6 + PM7)	Supply	32.0	47.5	14.2	1.15
2	Exhaust	38.5	60.6	26.4	1.09
2	Inside hall	35.0	60	21.5	1.11

For the ventilation option, the airflow required in each hour is that which reduces Δx sufficiently to bring t_d below t_{si} without an additional margin ($t_d = t_{si}$); the associated energy is the sensible heat needed to raise the additional outdoor air to the hall temperature. For the dehumidification options, the duty is not the total process generation of 27.2 t d⁻¹ but the residue that the existing ventilation fails to export. The cooling coil was characterised by its apparatus dew point (ADP ≈ 9 °C), which imposes a drying floor $x_{ADP} = x_{sat}(9$ °C) = 7.4 g kg⁻¹; the desiccant wheel by its regeneration power, computed from the mass of water it must adsorb; and the hybrid by the split of the total moisture load between the two components. Free-cooling availability was evaluated from the dry-cooler approach: with an 8 K approach, the required 6.5 °C glycol supply is attainable only below $t_e \approx -1.5$ °C.

3.7. Energy and Economic Evaluation

Seasonal transmission losses through the wall were computed as $U \cdot A \cdot \Sigma(t_i - t_e)$ over the 5053 cold-season hours with $A = 3190$ m². The insulation option was charged with the additional heating required to raise the hall from 25 to 27 °C and credited with the reduction in transmission losses. The same balance was evaluated for the recommended design temperatures of 28 and 30 °C (Section 4.4). Energy was valued at 60 € MWh⁻¹ (thermal) and 150 € MWh⁻¹ (electrical). Investment costs were estimated at feasibility-study level with an accuracy of ± 40 –50%, and a ten-year total cost of ownership was computed without discounting. Discounting was omitted in the base case because the ± 40 –50% uncertainty of the estimates far exceeds its effect over ten years and because the payback of the recommended option is short; a discounted scenario at real rates of 4–10%, with constant real prices and, for the insulation, also over a 25-year service life, is given as a supplementary reference in Section 4.8. The whole wall area was insulated rather than only the upper zone affected by condensation, because the heat saving is proportional to area and the payback period is therefore identical for the lower portion, which consequently finances itself.

4. Results and Discussion

4.1. Airflow and Moisture Balance

Two measurement campaigns were carried out: the first on 17 June with only PM6 running, the second on 2 July with both machines in operation. Velocity traverses at the extraction fans gave a mean of 10 m s⁻¹ at the four machine fans (65,144 m³ h⁻¹) and 6 m s⁻¹ at the three lateral fans (20,358 m³ h⁻¹), a total exhaust airflow of 85,502 m³ h⁻¹. This is consistent with the 80,000–90,000 m³ h⁻¹ estimated independently from the supply-fan characteristic curves, and the extraction measurement was therefore not repeated in the second campaign. The corresponding air change rate is 2.4–2.5 h⁻¹.

The measured air states are given in Table 2. The consistency between the humidity ratio measured at the exhaust and that measured at various points inside the hall (17.0 g kg⁻¹ against 18.0 g kg⁻¹ in the first campaign, 21.5 g kg⁻¹ against 26.4 g kg⁻¹ in the second)

confirms that the exhaust value is a defensible proxy for the upper-zone condition; the in-hall points were affected by adjacent spaces (a partly open door at the walkway, the stairwell at the balcony) and were used only as a plausibility check.

The resulting moisture balances are given in Table 3. With one machine running, the hall air takes up 18.4 t d^{-1} ; with both machines, 27.2 t d^{-1} . These figures exceed the declared release into the hall (Table 1: 5 and 7 t d^{-1} respectively) by factors of 3.7 and 3.9. The near-constancy of the ratio across two different operating states argues against random measurement error and points to a systematic discrepancy, attributable to some combination of an underestimated declared release, malfunctioning hood extraction, poor local capture of the vapour plume, and leakage returning extracted vapour to the hall. Interpreted as a capture deficiency, the measurements imply that about one tenth of the vapour nominally extracted by the hoods and the mist system ($\approx 190 \text{ t d}^{-1}$ for the two machines) ends up in the hall air instead. This finding alone justifies the priority given to source reduction, and it is also the reason why the condensation problem is so severe: the same building with a properly captured process would not condense. The magnitude of this spillage is not unusual: the TAPPI guidelines recommend a hood balance of 60–70% [17,18], and a published audit of a paper machine operating with a hood balance of 42% estimated about 3000 lb h^{-1} ($\approx 33 \text{ t d}^{-1}$) of water vapour escaping into the machine room [31]. The increment measured below the false ceiling (12.2 g kg^{-1}) also lies within the range admitted by the TAPPI guidelines for the roof zone ($\approx 21 \text{ g kg}^{-1}$ above ambient) [17]; it is its combination with an uninsulated envelope that makes it critical.

Table 3. Summer moisture balance of the hall.

Quantity	Unit	Campaign 1	Campaign 2
Exhaust volume flow	$\text{m}^3 \text{ h}^{-1}$	85,502	85,502
Exhaust humidity ratio	g kg^{-1}	18.0	26.4
Supply humidity ratio	g kg^{-1}	10.0	14.2
Humidity ratio increment Δx	g kg^{-1}	8.0	12.2
Dry-air mass flow	kg h^{-1}	96,084	92,882
Air change rate	h^{-1}	2.5	2.4
Moisture uptake	t d^{-1}	18.4	27.2
Declared release into hall	t d^{-1}	5	7
Ratio measured/declared	–	3.7	3.9

4.2. Winter Sensitivity Analysis

With $\Delta x = 12.2 \text{ g kg}^{-1}$ carried over to winter conditions, the reference case ($t_e = -5 \text{ }^\circ\text{C}$, $\varphi_e = 85\%$) gives an upper-zone humidity ratio of 14.4 g kg^{-1} and an upper-zone dew point of $19.7 \text{ }^\circ\text{C}$. At the reference false-ceiling temperature of $20 \text{ }^\circ\text{C}$, condensation on the ceiling is avoided only marginally, and appears as soon as the ceiling falls below $19.7 \text{ }^\circ\text{C}$; on the upper part of the walls, condensation occurs. Over the whole cold season, the upper-zone dew point ranges from 17.1 to $25.1 \text{ }^\circ\text{C}$ (mean $21.2 \text{ }^\circ\text{C}$); this is the range that the underside of the steel-sheet false ceiling must exceed, which is the purpose of the false-ceiling heating measure in Section 2.3. The plenum itself is supplied with outdoor air, heated without moisture addition in the PM7 recuperator, so its dew point remains that of the outdoor air, at most $11.8 \text{ }^\circ\text{C}$ in the cold season. The risk placed on the trusses and on the underside of the roof is therefore low as long as hall air does not leak into the plenum through the joints of the false ceiling, which should be sealed.

The parametric sweeps identified a clear hierarchy of influence. The process moisture release is by far the strongest driver: it acts directly on the upper-zone humidity ratio and hence on the dew point, and higher releases demand correspondingly higher surface temperatures. The outdoor air temperature acts on both zones, since the moisture released by the process is added to that already carried by the supply air; a warmer outdoor air at constant relative humidity carries more absolute moisture and raises the indoor dew point. The outdoor relative humidity has only a weak effect, because in winter the absolute moisture content of outdoor air is small at any relative humidity. The indoor air temperatures—both in the lower zone and in the upper zone—have no effect at all on the dew point, since they do not change the humidity ratio; they act only through the surface temperature, and raising the upper-zone air temperature over the range investigated was found insufficient to lift the wall out of the condensation regime. The false-ceiling temperature ultimately does not affect the dew point but is decisive for condensation on the ceiling itself. In the lower zone, no condensation risk was found under any of the conditions investigated.

4.3. The Mechanism of Winter Condensation

The reason the uninsulated wall cannot be rescued through heating follows directly from Equation (9). With $f = 0.506$, the surface temperature is very nearly the arithmetic mean of the indoor and outdoor air temperatures:

$$t_{si} = 0.494 t_i + 0.506 t_e \quad (13)$$

In summer, this is harmless. At $t_e = 30\text{ }^{\circ}\text{C}$ with the hall at $38.5\text{ }^{\circ}\text{C}$, the wall surface reaches $34.2\text{ }^{\circ}\text{C}$ while the indoor dew point is $29.5\text{ }^{\circ}\text{C}$ —a margin of 4.7 K , and no condensation, exactly as observed in the plant. In winter, the same equation is fatal. At $t_e = -5\text{ }^{\circ}\text{C}$, the indoor dew point is $19.7\text{ }^{\circ}\text{C}$ while the surface temperature of a hall held at $30\text{ }^{\circ}\text{C}$ falls to $12.3\text{ }^{\circ}\text{C}$. To lift the surface to the dew point, the hall would have to be heated to about $45\text{ }^{\circ}\text{C}$, and to about $52\text{ }^{\circ}\text{C}$ at $t_e = -15\text{ }^{\circ}\text{C}$. Winter condensation on this wall is therefore unavoidable by heating alone.

It is worth stating the corollary explicitly: the phenomenon requires *both* factors. The near-zero thermal resistance of the wall is one; the elevated dew point produced by the process moisture is the other. In the absence of this process, the indoor dew point at the referenced outdoor conditions would be about $-7\text{ }^{\circ}\text{C}$, and the bare concrete wall would not condense at all.

Figure 3 shows the surface temperature, the indoor dew point and the indoor air temperature as functions of outdoor temperature for the uninsulated wall and for the wall with 50 mm of EPS, at indoor temperatures of 25 and 35 °C. On the uninsulated wall (panels a and b), the surface line lies below the dew-point line over the whole cold range at both indoor temperatures. With 50 mm of EPS (panels c and d), the surface line is lifted above the dew point; at 25 °C, however, the two lines cross at about an outdoor temperature of 12 °C, beyond which the indoor dew point exceeds the indoor air temperature itself and condensation resumes regardless of insulation thickness—the singularity of Equation (10). At 35 °C, the margin is comfortable across the whole range.

Figure 4 gives the EPS thickness required by Equations (10) and (11). Except in the region where insulation cannot help at all, the required thickness stays below 50 mm; 50 mm of EPS was therefore adopted as the design value.

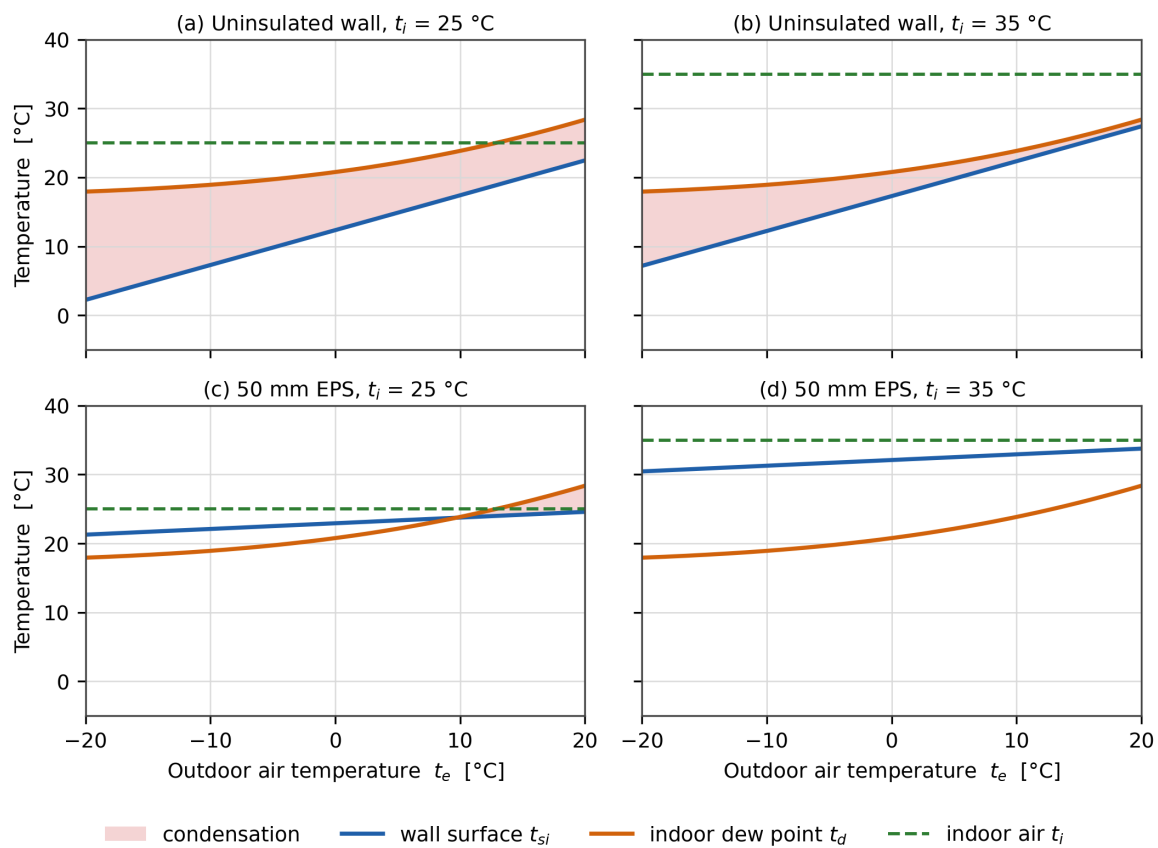


Figure 3. Wall internal surface temperature, indoor dew point and indoor air temperature versus outdoor temperature. (a,b) Uninsulated 200 mm reinforced-concrete wall; (c,d) the same wall with 50 mm external EPS; (a,c) indoor air at 25 °C; (b,d) indoor air at 35 °C. Shaded areas indicate condensation.

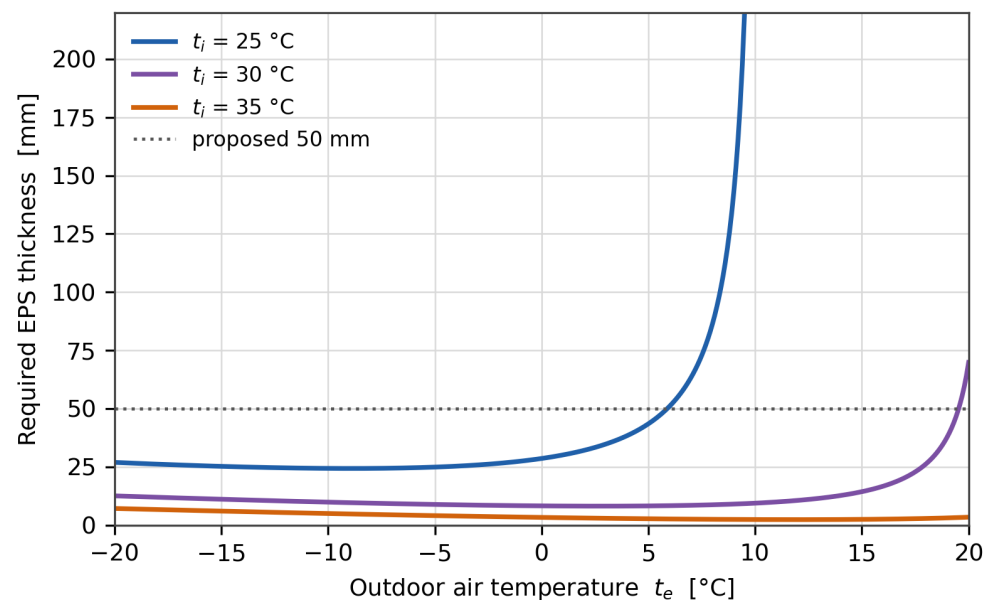


Figure 4. External EPS thickness required to keep the internal wall surface 1 K above the indoor dew point, for three indoor air temperatures. The curves terminate where the indoor dew point reaches the indoor air temperature and insulation ceases to be effective.

4.4. Solution 1—Envelope Insulation

Table 4 gives the hourly evaluation over the 5053 cold-season hours. The result that matters is not that insulation works, but that insulation and heating are complementary rather than alternative. Insulation with a cold hall (20 °C) leaves 5051 condensation hours; heating alone on the bare wall leaves 4605 h even at 35 °C. The combination removes condensation entirely, with a sharp threshold at 50 mm and 27 °C, where the minimum seasonal margin turns positive (+0.6 K).

Table 4. Parametric analysis of insulation over the cold season (5053 h).

Wall	Hall Temperature	Condensation Hours	% of Season	Minimum Margin
Existing (200 mm RC)	20 °C	5053	100%	−16.4 K
Existing	25 °C	5053	100%	−13.9 K
Existing	30 °C	5047	99.9%	−11.4 K
Existing	35 °C	4605	91.1%	−8.9 K
+50 mm EPS	20 °C	5051	100%	−5.6 K
+50 mm EPS	25 °C	239	4.7%	−1.2 K
+50 mm EPS	27 °C	0	0%	+0.6 K
+50 mm EPS	28 °C	0	0%	+1.5 K
+50 mm EPS	30 °C	0	0%	+3.4 K
+50 mm EPS	35 °C	0	0%	+8.0 K

The residual 239 h at 25 °C corresponds to the mildest part of the cold season, when the outdoor air is warm enough to carry appreciable absolute moisture into the hall; they disappear with a 2 K increase in the hall temperature.

The 27 °C threshold is a result of the reference model and carries a minimum seasonal margin of only 0.6 K. Allowing for the uncertainty of the model inputs (Section 4.10) and for thermal bridges (Section 4.9), 28 °C is recommended as the minimum design temperature: it maintains a margin of at least 1.5 K between the surface temperature and the dew point over the whole cold season. A hall temperature of 30 °C, with a margin of 3.4 K, is a conservative value that remains free of condensation under every perturbation examined. The higher set points reduce the net heat saving to about 690 MWh per season at 28 °C and about 430 MWh at 30 °C (simple payback 1.9–3.1 and 3.1–5.0 years), but leave the ranking of the options unchanged. These temperatures refer to the upper zone of the hall, adjacent to the condensation-prone part of the wall, and are compatible with the process: 28–35 °C were measured inside the hall and 32–38.5 °C at the exhaust during the summer campaigns (Table 2); the operator reports temperatures of 27 °C and above in the upper zone as routine and without effect on production; and the TAPPI guidelines prescribe a minimum of 75 °F (≈ 24 °C) at the wet-end operating floor while allowing the air beneath the roof to be up to 25 °F (≈ 14 K) warmer than ambient [17,18].

4.5. Solution 2—Ventilation Intensification

Winter outdoor air, at $1\text{--}3\text{ g kg}^{-1}$, is drier than air leaving a cooling coil at its apparatus dew point, so increasing the airflow is itself a dehumidification strategy. Table 5 gives the airflow required to suppress condensation and the associated seasonal heating energy.

Table 5. Airflow required to avoid condensation and associated cold-season energy.

Wall	Hall Temperature	Required Airflow	Ratio to Present	Air Heating	Energy Surplus
Existing	30 °C	200,213 m ³ h ^{−1}	2.3×	7087 MWh	+4570 MWh
Existing	35 °C	167,724 m ³ h ^{−1}	2.0×	6957 MWh	+3846 MWh
+50 mm EPS	30 °C	62,368 m ³ h ^{−1}	0.7×	1595 MWh	−922 MWh
+50 mm EPS	35 °C	43,283 m ³ h ^{−1}	0.5×	1135 MWh	−1975 MWh

On the bare wall, the measure amounts to doubling the ventilation plant—an additional 115,000 m³ h^{−1} of air handling, ductwork and heating capacity—and a seasonal energy surplus of about 4000 MWh for heating the extra fresh air. On the insulated wall, the airflow required to prevent condensation falls below the airflow already installed: once the envelope is treated, the ventilation does not need to be increased. The existing ventilation system is retained as it is, since it remains necessary to export the process heat, to supply make-up air to the hoods, to maintain air quality and to prevent fog (Section 3.4).

4.6. Solution 3—Mechanical Dehumidification

Table 6 gives the three configurations evaluated on the bare wall at unchanged ventilation airflow.

Table 6. Dehumidification configurations, uninsulated wall, unchanged ventilation airflow.

Configuration	Hall Temp.	Equipment Airflow	Water Split Coil/Wheel	Regeneration	Seasonal Energy
Cooling coil alone	30 °C	not feasible	—	—	—
Desiccant wheel alone	30 °C	152,606 m ³ h ^{−1}	0%/100%	631 kW	1795 MWh
Hybrid coil + wheel	30 °C	197,209 m ³ h ^{−1}	45%/55%	631 kW	1093 MWh
Cooling coil alone	35 °C	not feasible	—	—	—
Desiccant wheel alone	35 °C	108,660 m ³ h ^{−1}	0%/100%	540 kW	902 MWh
Hybrid coil + wheel	35 °C	125,367 m ³ h ^{−1}	51%/49%	540 kW	498 MWh

The cooling coil used alone is excluded on physical grounds. It cannot dry air below the saturation humidity ratio of its own surface, $x_{ADP} = 7.4 \text{ g kg}^{-1}$ for an apparatus dew point of 9 °C, whatever the airflow. On the bare wall, the target imposed by the cold surface falls below this floor for 139 h per year with the hall at 30 °C, and in those hours—the coldest and most critical of the season—the coil cannot meet the requirement.

In series with a desiccant wheel, the coil recovers its usefulness. It removes 45–51% of the water in the range where it is efficient, from the hall humidity down to its own floor, and the wheel finishes the drying below that floor. Because the water taken by the coil requires no regeneration heat—the most expensive energy in the chain—the total consumption falls by 39–45% relative to the wheel alone, in line with the reductions reported for hybrid systems in the literature [24,25].

Free-cooling contributes less than might be expected. Producing 6.5 °C glycol with a dry cooler at an 8 K approach requires an outdoor temperature below −1.5 °C, a condition met in only 681 of the 5053 cold-season hours (13%). Reducing the approach to 6.5 K extends free-cooling operation up to an outdoor temperature of 0 °C. The contribution is real but limited, and the principal merit of the hybrid arrangement remains the reduction in the regeneration duty rather than the elimination of the compressor.

The decisive observation, however, concerns the insulated wall: with 50 mm of EPS in place, the dehumidification duty of the plane wall is zero at every hall temperature analysed, and all three configurations become superfluous; local condensation at thermal

bridges is addressed by the detailing of the insulation and by the design hall temperature (Section 4.9).

4.7. Energy Balance

Table 7 gives the seasonal energy balance of the insulation option, charged with the heating needed to raise the hall from 25 to 27 °C and credited with the reduction in transmission losses over the full 3190 m² of wall.

Table 7. Cold-season energy balance of the insulation option.

Component	Energy [MWh per Season]
Wall transmission losses—present situation (hall at 25 °C)	1306
Wall transmission losses—50 mm EPS (hall at 27 °C)	234
(+) Fresh-air heating to raise the hall from 25 to 27 °C	+246
(−) Saving on transmission losses (3190 m ²)	−1072
Net balance	−826

Insulation does not consume energy: it saves 826 MWh per season even after paying for the additional heating it requires. Transmission losses fall by a factor greater than five. Table 8 places this alongside the net consumption of the other options.

Table 8. Net cold-season energy consumption of the four options.

Solution	Thermal	Electrical	Net Energy
50 mm EPS + hall at 27 °C	−826 MWh	0	−826 MWh
Hybrid coil + wheel (no insulation)	950 MWh	143 MWh	+1093 MWh
Desiccant wheel alone (no insulation)	1795 MWh	0	+1795 MWh
Ventilation at 2.3× (no insulation)	4570 MWh	60 MWh	+4630 MWh

For solutions 2 and 3, the wall remains uninsulated, so the 1306 MWh of transmission losses persist and are not included in the table; these options therefore also forgo the heat saving.

4.8. Economic Assessment

Investment estimates are of feasibility-study accuracy (± 40 –50%) and do not replace tendering. Table 9 gives the capital cost of each option and the ten-year total cost of ownership at 60 € MWh^{−1} thermal and 150 € MWh^{−1} electrical.

Table 9. Investment and ten-year total cost of ownership.

Solution	Main Components	CAPEX [k€]	OPEX [k€ a ^{−1}]	10-Year Cost [k€]
1. Insulation	50 mm EPS, 3190 m ² (25–40 € m ^{−2}), incl. fixing and finish	80–128 (~102)	−50 (gain)	≈−394 (profit)
2. Ventilation	AHU +115,000 m ³ h ^{−1} , ductwork, heating coils, controls	590–998 (~790)	+283	≈3620
3a. Wheel alone	Desiccant wheels 153,000 m ³ h ^{−1} , 631 kW regeneration source, ductwork	2266–3860 (~3060)	+108	≈4140
3b. Hybrid	Wheel (55%), coils + AHU, chiller + dry cooler, glycol circuit, regeneration	2090–3637 (~2860)	+78	≈3640

Insulation is the only option with a negative total cost. The heat saving it produces, about 50 k€ per year, amortises the investment in 1.6–2.6 years depending on the unit execution price, and over ten years yields a net gain of about 394 k€, against costs of 3.6–4.1 M€ for the alternatives (Figure 5). The gap is an order of magnitude and is far larger than the ± 40 –50% uncertainty of the estimates, so the ranking is robust.

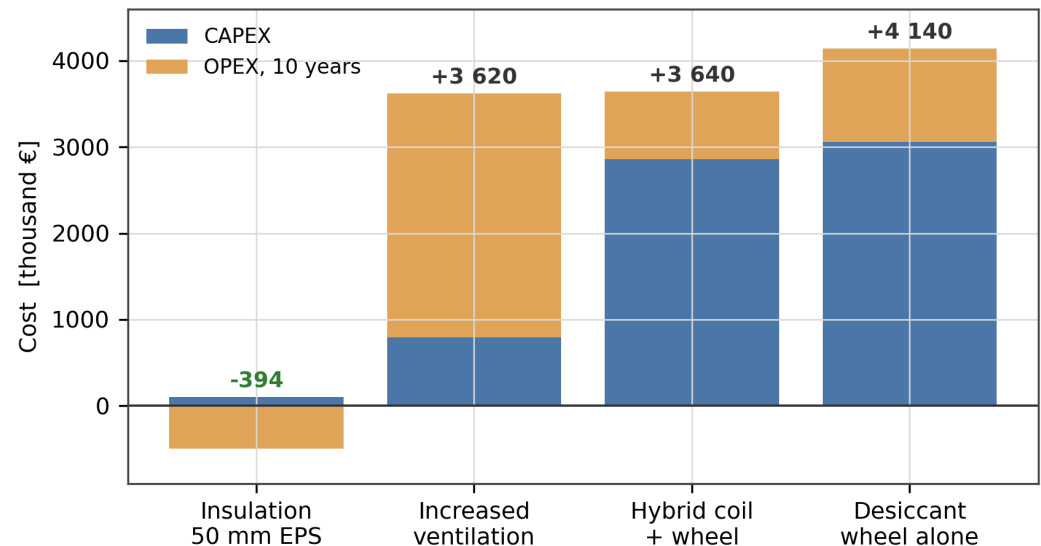


Figure 5. Ten-year total cost of ownership of the four options. Values above the bars are net totals; a negative total indicates a net gain.

It should be stressed that the wall insulation is economically justified as a pure energy-efficiency measure, independently of the condensation problem—a conclusion consistent with the retrofit economics reported by Basińska et al. [29]. That it simultaneously and completely resolves the condensation problem leaves it without a rational alternative.

Discounting does not alter this result (Table 10). At real discount rates of 4–10%, the insulation maintains a positive ten-year net present value of 203–300 k€ for the central investment of 102 k€, with a discounted payback of 1.7–3.1 years over the full investment range; over a 25-year service life, its net present value at 6% is 506–554 k€. The discounted ten-year costs of the alternatives remain 2.5–3.9 M€, and the insulation is the only option with a positive net present value at every rate.

Table 10. Discounted economic evaluation (constant real prices; central insulation investment of 102 k€).

Discount Rate	Insulation: NPV, 10 Years [k€]	Insulation: Discounted Payback [Years]	Ventilation 2.3×: Discounted 10-Year Cost [k€]	Hybrid Coil + Wheel [k€]	Desiccant Wheel Alone [k€]
0% (Table 9)	+394	2.1	3620	3640	4140
4%	+300	2.2	3085	3493	3936
6%	+263	2.3	2873	3434	3855
8%	+231	2.3	2689	3383	3785
10%	+203	2.4	2529	3339	3724

4.9. Discussion and Limitations

Three points deserve emphasis. First, the two causes of the problem are separable and act multiplicatively: an envelope with negligible thermal resistance, and a dew point elevated far above the outdoor value by uncaptured process moisture. Acting on either one is sufficient in principle; acting on the envelope is by far the cheaper of the two available to

the building designer, while acting on the moisture source is the responsibility of the process engineer and should be pursued in parallel. Second, the solutions are not independent alternatives to be traded off: insulating the envelope brings the airflow required to prevent condensation below the installed capacity, so the existing ventilation need not be increased, reducing the dehumidification duty of the plane wall to zero. The two systems nevertheless interact: with the wall insulated, a moderate reduction in the hood spillage allows for a lower hall temperature. Third, the widely held expectation that a large industrial hall can be kept condensation-free by heating it more aggressively is quantitatively wrong for an uninsulated heavy wall, and Equation (9) shows why.

The analysis is subject to several limitations. The winter indoor air temperature was not measured; it was treated parametrically over 20–35 °C, and the condensation threshold emerged at 27 °C, from which the design values of 28 and 30 °C were derived (Section 4.4). Confirming the actual winter indoor temperature by measurement would refine the figures without altering the ranking of the options. The moisture increment Δx and the sensible temperature rise ΔT were calibrated on summer measurements and assumed constant into winter, for the reasons given in Section 3.2; no winter measurements were available. The model nevertheless reproduces the observed seasonal behaviour: no condensation in summer, and water running down the walls and dripping from the false ceiling in winter, with the latter being the reason the study was commissioned. A winter measurement campaign, recording the supply and exhaust humidity ratio and the upper-zone temperature, is recommended to confirm the increment. The envelope model is steady-state and one-dimensional, and neglects the thermal mass and the moisture buffering of the concrete, which would improve the real behaviour somewhat in the short term; thermal bridges are treated below through the temperature factor. The two-zone representation is coarse compared with the CFD treatments applied to swimming halls and ice arenas [10,11], and does not resolve the local surface temperature field that governs where condensation first appears. Finally, the cost figures are order-of-magnitude estimates requiring confirmation by tender.

Thermal bridges. The one-dimensional result applies to the plane wall; at junctions, the surface temperature is lower and condensation can persist after insulation. The effect was quantified with the temperature factor of EN ISO 13788 [5], $f_{Rsi} = (t_{si} - t_e)/(t_i - t_e)$, by computing for every cold-season hour the value that a junction must reach for its surface to remain above the indoor dew point (Table 11). The plane insulated wall has $f_{Rsi} = 0.917$; a junction must reach 0.876 at 27 °C, 0.822 at 28 °C and 0.731 at 30 °C. The critical junctions are the window and door reveals, which must be insulated with the facade, the plinth, the reinforced-concrete columns and beams wherever they interrupt the insulation layer, and the wall–roof junction, i.e., the geometric and structural bridges identified by Ahsan et al. [32] as the typical locations of surface condensation and mould growth. Semko et al. [33] showed for a steel truss in an attic, a configuration close to the present roof, that leaving gaps unfilled by insulation raises the attic air temperature below which condensation occurs on the truss from about −4 °C to about +4 °C. Thermal bridges therefore require careful detailing of the insulation and are an additional argument for the 28–30 °C design values.

Roof assembly. The steel-sheet false ceiling is practically vapour-tight, so no interstitial condensation can develop within it; the condensation observed in the plant is surface condensation on its underside (Section 4.2). Interstitial or pore condensation within the roof closure itself could not be evaluated, because its layers and their vapour resistance are not documented. A survey of the roof construction and an assessment to EN ISO 13788 [5] are recommended, together with the sealing of the joints of the false ceiling.

Table 11. The temperature factor required at junctions of the insulated wall over the cold season, and condensation hours at junctions for the given temperature factor.

Hall Temperature	f_{Rsi} Required	Condensation Hours, $f_{Rsi} = 0.80$	$f_{Rsi} = 0.75$	$f_{Rsi} = 0.70$
27 °C (reference model)	0.876	299	2427	4563
28 °C (recommended minimum)	0.822	25	433	3709
30 °C (conservative)	0.731	0	0	191

Source reduction. The effect of the hood audit and sealing recommended in Section 2.3 is shown in Table 12. For the uninsulated wall to become free of condensation, the moisture uptake of the hall air would have to fall to about 11.7 t d^{−1} at 30 °C, or 9.5 t d^{−1} at 25 °C, a reduction of 57–65% that implies eliminating three quarters or more of the ≈20 t d^{−1} of spillage. This confirms that the same building with a properly captured process would not condense, but also that the sealing would have to be nearly complete to solve the problem on its own. Combined with the insulation, a smaller reduction suffices: lowering the uptake by about 27%, to 19.8 t d^{−1}, would allow the hall to remain at its present 25 °C with the same 1.5 K margin as the 28 °C design value, raising the net heat saving from about 690 to about 1 090 MWh per season. The two measures are therefore complementary and should be pursued in parallel.

Table 12. Effect of a reduction in the moisture uptake of the hall air (hood spillage).

Moisture Uptake [t d ^{−1}]	Reduction	Δx [g kg ^{−1}]	Uninsulated Wall at 30 °C: Condensation Hours	50 mm EPS: Critical Hall Temperature [°C]
27.2 (measured)	—	12.2	5047	26.3
22	19%	9.9	4667	24.3
18	34%	8.1	2392	22.5
15	45%	6.7	121	21.1
12	56%	5.4	2	18.3
7 (declared release)	74%	3.1	0	<20

4.10. Sensitivity Analysis

Each assumption of the model was varied in turn on the full hourly TMY, with the others held at their reference values (Table 13). Two outputs were tracked: the minimum hall temperature giving zero condensation hours on the insulated wall (the critical hall temperature, 26.3 °C in the reference case, reported as 27 °C in Table 4, where the hall temperature was evaluated in steps) and the number of condensation hours on the uninsulated wall at 30 °C. The results are governed by the moisture increment Δx , directly or through the exhaust airflow, by the safety margin and by the internal surface resistance, whose upper value of 0.25 m² K W^{−1} is the one prescribed by EN ISO 13788 for surface-humidity assessment in corners and behind obstructions [5]. The material conductivities and ΔT have practically no influence, since ΔT only matters when the outdoor temperature exceeds the hall set point minus 8.5 K, which does not occur in the cold season. The ranking of the solutions is unchanged in every case: the uninsulated wall condenses in 43–100% of the cold-season hours at 30 °C. At the minimum design temperature of 28 °C, the insulated wall stays free of condensation for Δx up to +10%, which covers the estimated winter increment of Section 3.2, for a 1 K margin and for $R_{si} = 0.25$ m² K W^{−1}; at 30 °C, it stays free of condensation in every case tested.

Table 13. One-at-a-time sensitivity analysis on the hourly TMY.

Parameter	Reference	Range Tested	Critical Hall Temperature, 50 mm EPS [°C]	Condensation Hours, Uninsulated Wall at 30 °C
Humidity-ratio increment Δx [g kg ⁻¹]	12.2	8.0–15.9 (−35 to +30%)	22.5–29.2	2196–5053
Exhaust airflow [m ³ h ⁻¹] (acts through Δx)	85,500	±15%	24.9–28.0	4915–5053
Safety margin on the dew point [K]	0	0–2	26.3–28.5	5047–5053
Internal surface resistance R_{si} [m ² K W ⁻¹]	0.13	0.10–0.25	26.0–27.4	4955–5053
Cold-season threshold [°C]	12	8–16	24.8–28.1	see Table 14
Conductivity of EPS [W m ⁻¹ K ⁻¹]	0.038	0.032–0.044	26.1–26.5	5047
Conductivity of concrete [W m ⁻¹ K ⁻¹]	2.3	1.7–2.5	26.3	4988–5051
Air temperature rise across the hall ΔT [K]	8.5	0–12	26.3	5047
Reference case			26.3	5047

Table 14. Effect of the cold-season threshold on the condensation hours of the insulated wall (50 mm EPS).

Threshold t_e [°C]	Season Length [h]	Condensation Hours at 27 °C	at 28 °C	at 30 °C
8	3753	0	0	0
10	4413	0	0	0
12 (reference)	5053	0	0	0
14	5683	3	0	0
16	6334	85	7	0
whole year	8760	503	276	2

The cold-season threshold was examined separately (Table 14), together with a run of the same model over the whole year. With the wall insulated and the hall at 30 °C, only two condensation hours are predicted over the entire year, so the conclusions do not depend on where the season boundary is drawn; on the uninsulated wall, condensation is predicted in 84–100% of the cold-season hours at 30–35 °C for every threshold examined.

5. Conclusions

A tissue paper production hall affected by chronic winter condensation on its upper walls and false ceiling was analysed on the basis of two measurement campaigns and an hourly typical meteorological year, and three types of remedial solutions were compared on identical technical, energetic and economic criteria.

The measured exhaust airflow of 85,502 m³ h⁻¹ (2.46 h⁻¹) and the measured humidity-ratio increment of 12.2 g kg⁻¹ with both machines running correspond to a moisture uptake of 27.2 t d⁻¹ by the hall air—3.9 times the value declared in the process water balance. About one tenth of the vapour nominally extracted by the machine hoods therefore reaches the hall air, and this uncaptured moisture is what raises the indoor dew point from about −7 °C to about 19.7 °C in the winter reference conditions.

Condensation results from the conjunction of this elevated dew point with an envelope of negligible thermal resistance. Because the internal surface film carries 50.6% of the total thermal resistance of the 200 mm reinforced-concrete wall, the surface temperature is approximately the mean of the indoor and outdoor temperatures. Additional heating alone is consequently not a solution: the bare wall would require an indoor temperature of about 45 °C at −5 °C outdoor and about 52 °C at −15 °C.

All three types of solutions can eliminate condensation completely, but they differ radically in investment and consumption. The cooling coil alone is physically excluded,

since the target imposed by the cold surface falls below its drying floor of 7.4 g kg^{-1} ; in a hybrid arrangement with a desiccant wheel, it takes about half the moisture load and reduces consumption by 39–45% relative to the wheel alone. Ventilation intensification works without any dehumidification equipment, with the winter outdoor air at $1\text{--}3 \text{ g kg}^{-1}$ itself a drying agent, but requires doubling the installed airflow and about 4000 MWh of additional heat per season.

The recommended solution is external insulation of the walls with 50 mm of expanded polystyrene combined with maintaining the upper zone of the hall at a minimum of 28°C , the recommended minimum design temperature, which keeps a margin of at least 1.5 K over the whole cold season; 30°C is a conservative value that ensures the hall remains free of condensation under every perturbation examined. The reference model gives 27°C as the threshold, at which the insulation eliminates condensation in all 5053 cold-season hours, saves 826 MWh per season (about 50 k€ per year) and amortises in 1.6–2.6 years; at 28°C , the saving is about 690 MWh per season and the amortisation 1.9–3.1 years, against ten-year costs of 3.6–4.1 M€ for the alternatives, a ranking that discounts do not alter. The window and door reveals and the other junctions must be insulated with the facade, so that their temperature factor reaches at least 0.82. Once the wall is insulated, the dehumidification duty of the plane wall falls to zero and the airflow required to prevent condensation drops below the installed capacity: the ventilation does not need to be increased and the existing system is retained. In parallel, an audit and sealing of the PM6/PM7 hoods is recommended as a primary action to address the cause; combined with the insulation, a reduction in the moisture uptake by about a quarter would allow the hall to remain at its present 25°C . Insulation does not compete with the other solutions; it makes them unnecessary.

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