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INDUSTRIAL AIR CONDITIONING SYSTEM FOR PRINTING PRODUCTION: DESIGN, CALCULATION AND MULTI-ZONE OPTIMIZATION

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Abstract. *The article presents research results on the design and optimization of an industrial air conditioning system (ACS) for a printing production facility located in Mykolaiv, Ukraine. Printing enterprises impose strict requirements on indoor microclimate parameters — temperature, relative humidity and air purity — due to the sensitivity of paper-based substrates and printing inks to environmental conditions. This study analyses four configurations of multi-zone central ACS: single-duct straight-through, single-duct with first recirculation, dual-duct straight-through, and dual-duct with first recirculation. Comparative analysis based on thermal and moisture load calculations demonstrates that the dual-channel multi-zone ACS with first recirculation is the most energy-efficient solution for the given facility. The thermal load calculation was performed for summer (outdoor temperature 31 °C, indoor design temperature 25 °C) and winter (outdoor temperature −20 °C, indoor design temperature 20 °C) design conditions. The paper describes the selection of the central air handling unit, the sizing of the air cooler, air heater, adiabatic spray chamber, air filter, and air distribution network. The hydraulic calculation of the pipe network and pump selection are also presented. The results confirm that proper system configuration significantly reduces annual energy consumption for heating, cooling and humidification compared to single-duct alternatives.*

Key words: air conditioning system, multi-zone system, printing production, thermal load, heat recovery, air handling unit, energy efficiency.

1. INTRODUCTION

The printing industry is characterized by high sensitivity of production processes to environmental conditions. Paper, the primary substrate used in offset and digital printing, is hygroscopic: it absorbs or releases moisture depending on the ambient relative humidity, leading to dimensional changes, misregistration of colours, and print quality defects. Printing inks similarly depend on controlled temperature and humidity for optimal viscosity and drying behaviour. Dust particles in the air can cause contamination of printed surfaces and damage to precision printing equipment. Consequently, industrial air conditioning systems in printing plants must simultaneously control temperature, relative humidity, air cleanliness, and airflow velocity within tight tolerances.

The energy performance of air conditioning systems has become a critical concern in Ukraine and globally, given rising energy costs and the imperative to reduce carbon emissions. Modern multi-zone central ACS offer significant advantages over distributed or single-zone systems in large-volume facilities with variable and spatially non-uniform thermal and moisture loads. However, selecting the optimal system configuration requires careful analysis of all operating conditions across the annual cycle.

This paper addresses the design and multi-zone optimization of an industrial ACS for a printing facility in Mykolaiv, providing a systematic methodology that can be applied to similar industrial microclimate engineering projects.

2. LITERATURE REVIEW AND PROBLEM STATEMENT

Multi-zone air conditioning systems have been studied extensively in the context of energy efficiency and indoor environmental quality [1–3]. Fundamental design methodologies are established in normative documents and textbooks on HVAC engineering [4]. The application of central ACS in industrial environments, however, presents specific challenges that differ from comfort conditioning in office or residential buildings.

Printing production occupies a particular niche in industrial microclimate engineering. Studies confirm that relative humidity deviations as small as ±5% from the target value of 50–55% RH can cause measurable paper dimensional changes and colour registration errors in multi-pass printing [5]. Temperature control requirements are typically ±2 °C around a target of 20–25 °C, with tighter tolerances in precision applications.

Previous research on multi-zone ACS configurations has demonstrated the energy-saving potential of recirculation-based systems, particularly in climates with extreme summer and winter design conditions [2, 6]. However, detailed comparative analyses of alternative system schemes for printing applications under Ukrainian climatic conditions remain limited in the published literature. This study aims to fill this gap by providing a rigorous comparative assessment and engineering calculation for a real-world case.

3. DESIGN CONDITIONS AND THERMAL LOAD ANALYSIS

The facility under study is a printing production plant located in Mykolaiv, Ukraine. The design outdoor parameters were established according to Ukrainian State Building Standards: summer outdoor temperature t_{out} = 31 °C (outdoor relative humidity φ_{out} = 40%), winter outdoor temperature t_{out} = −20 °C. The required indoor parameters are: summer t_{in} = 25 °C, φ_{in} = 50%; winter t_{in} = 20 °C, φ_{in} = 45–55%.

The thermal load calculation followed the standard methodology of summing heat gains through building envelope (walls, roof, glazing), solar radiation, internal heat sources (personnel, lighting, printing equipment), and moisture sources (personnel respiration, technological processes). The envelope thermal resistance was calculated accounting for the insulation specifications of the building construction.

Heat gains through the building envelope were calculated using the formula $Q_{env} = k \cdot F \cdot (t_{out} - t_{in})$, where k is the overall heat transfer coefficient (W/m²·K), F is the surface area (m²), and t is the temperature difference between outdoor and indoor conditions. Solar radiation gains through glazed surfaces were determined using orientation factors and the solar radiation intensity for the Mykolaiv climate zone. Internal heat gains from printing equipment (offset presses, drying units) were obtained from manufacturer's technical data sheets and were found to be the dominant component of the summer thermal load, accounting for approximately 58% of total sensible heat gains.

Moisture loads were determined from personnel occupancy (assumed 25 persons at light activity, W = 60 g/h per person) and from technological processes of the printing facility, including evaporation from dampening systems of offset printing units. The resulting total moisture load of 4.2 g/s in summer governs the sizing of the dehumidification capacity of the cooling coil, while the winter moisture load of 3.1 g/s governs the sizing of the adiabatic spray chamber.

Table 1 – Summary of design thermal and moisture loads

Parameter	Summer	Winter
Total sensible heat gain, kW	87.4	−
Total heat loss, kW	−	62.1
Moisture load (total), g/s	4.2	3.1
Required supply air flow, m³/h	18 500	18 500
Outdoor air fraction, %	30	30

4. COMPARATIVE ANALYSIS OF MULTI-ZONE ACS CONFIGURATIONS

Four multi-zone central ACS schemes were analysed:

Scheme 1 — Central single-duct straight-through (once-through) ACS. All supply air is taken from outdoors, processed in the central AHU, and distributed to zone heaters installed in each room. This scheme provides excellent indoor air quality but results in the highest energy consumption for cooling and heating.

Scheme 2 — Central single-duct ACS with first recirculation. A portion of return air is recirculated and mixed with outdoor air before the central AHU. This reduces the thermal and moisture load on the central processing components.

Scheme 3 — Dual-duct straight-through multi-zone ACS. Two parallel air ducts supply warm and cold air independently, mixed at terminal mixing boxes for each zone. Enables precise zone-by-zone temperature control.

Scheme 4 — Dual-duct multi-zone ACS with first recirculation. Combines the benefits of dual-duct zoning with recirculation-based energy recovery, representing the most complex but potentially most efficient configuration.

Comparison of the four schemes was conducted on the basis of: zone temperature control capability, humidity control quality, annual energy consumption index, and system installation complexity. The results are summarised in Table 2.

Table 2 – Comparative characteristics of multi-zone ACS schemes

Criterion	Scheme 1	Scheme 2	Scheme 3	Scheme 4
Zone temperature control	Zonal heaters	Zonal heaters	Mixing box	Mixing box (rec.)
Humidity control	Moderate	Moderate	Moderate	Good
Relative energy consumption	High	Medium	Medium	Low

Installation complexity	Low	Medium	Medium	High
Recommended for printing plant	No	Possible	Possible	Yes

The dual-duct multi-zone ACS with first recirculation (Scheme 4) was selected as the optimal solution. Its key advantages for a printing plant are: individual temperature control in each zone; stable supply air flow rate regardless of zone load variation; the ability to use standard air distribution devices; uniform temperature field in the conditioned spaces; absence of heat exchangers, piping and refrigerant in the served rooms; and phased commissioning as the building expands.

5. EQUIPMENT SELECTION AND HYDRAULIC CALCULATION

5.1. Central air handling unit. Based on the calculated supply air flow of 18 500 m³/h and thermal/moisture processing requirements, a central AHU was selected. The unit incorporates a rotary heat recovery wheel (enthalpy type) with an efficiency of $\eta = 0.72$, a pre-heater (first-stage heating), an adiabatic spray chamber for humidification, a cooling coil (air cooler), a final heater, a G4+F7 two-stage filtration section, and a supply fan.

5.2. Air cooler calculation. The cooling coil was sized to handle the peak summer cooling load. The inlet air conditions are determined by mixing outdoor air (t = 31 °C, d = 12.3 g/kg) with recirculated return air (t = 27 °C, d = 10.8 g/kg) at a 30:70 ratio, resulting in mixed air parameters of approximately $t_{mix} = 28.1$ °C, d = 11.2 g/kg. The required cooling capacity of the coil was calculated as $Q_{cool} = G \cdot (h_{in} - h_{out}) = 5.14 \cdot (57.3 - 38.4) = 97.2$ kW, where G is the mass flow rate of supply air (kg/s) and h is the specific enthalpy (kJ/kg).

5.3. Air heater and adiabatic humidifier. For the winter period, the pre-heater capacity was calculated as $Q_{heat1} = 31.4$ kW (heating outdoor air from −20 °C to +5 °C prior to the spray chamber). The adiabatic spray chamber provides humidification from $d_1 = 1.8$ g/kg (outdoor air equivalent) to $d_2 = 8.5$ g/kg. The second-stage heater provides final heating to the supply air set-point temperature.

5.4. Air filter selection. A two-stage pocket filter (G4 pre-filter + F7 fine filter) was selected to achieve the cleanliness class required for printing production environments. The G4 pre-filter intercepts coarse particles (>10 µm), while the F7 pocket filter captures particles down to approximately 1 µm, meeting the requirements of ISO 16890 ePM1 55%.

5.5. Air distribution network. The supply duct network was designed for a maximum duct velocity of 6 m/s in main ducts and 3 m/s in branch ducts. The total pressure drop of the supply system (including AHU resistance) was calculated as $\Delta P_{total} = 780$ Pa. The return/exhaust duct system was sized for $\Delta P = 340$ Pa. The supply fan was selected with $Q_{fan} = 18\,500$ m³/h at $\Delta P = 820$ Pa (with 5% safety margin), with a motor rated at N = 7.5 kW.

5.6. Hydraulic calculation of chilled water circuit. The chilled water circuit connecting the cooling coil to the chiller was designed for a flow rate of $G_w = 16.7$ m³/h ($\Delta T = 5$ K, chilled water supply/return 7/12 °C). The pipeline diameter was selected as DN 50 with a flow velocity of 1.1 m/s. A circulation pump was selected: $Q_{pump} = 17$ m³/h, H = 12 m w.c., N = 1.5 kW.

5.7. Heat recovery system. The rotary enthalpy heat recovery wheel was sized based on the supply and exhaust air flow balance. Operating at an efficiency of $\eta = 0.72$, the wheel recovers both sensible and latent heat from the exhaust air stream. In winter operation, the wheel preheats and pre-humidifies the incoming outdoor air, substantially reducing the heating and humidification load on downstream components. The estimated annual heat recovered by the wheel is approximately 95 MWh, which represents the single largest energy-saving measure in the ACS design. The wheel is equipped with a variable-speed drive to allow efficiency adjustment based on outdoor conditions and to prevent frosting at low outdoor temperatures.

5.8. Refrigeration plant selection. The cooling load for the chilled water system was determined as $Q_{chiller} = 97.2$ kW (cooling coil) + 12.4 kW (pipe heat gains) = 109.6 kW, with a 10% design margin yielding $Q_{design} = 120$ kW. A water-cooled screw chiller was selected with a rated cooling capacity of 125 kW, COP = 5.2 at design conditions (entering condenser water 30 °C, leaving chilled water 7 °C). The condenser is connected to a cooling tower with a capacity of 160 kW.

6. RESULTS AND DISCUSSION

The designed dual-duct multi-zone ACS with first recirculation demonstrates the following key performance characteristics. In summer design conditions, the system maintains indoor temperature of 25 ± 1 °C and relative humidity of 50 ± 5% simultaneously in all conditioned zones, with independent zone control. In winter design conditions, the system maintains indoor temperature of 20 ± 1 °C and relative humidity of 48 ± 5%.

A comparative energy analysis (Fig. 1) shows that the selected Scheme 4 reduces annual energy consumption by approximately 18–22% compared to the single-duct once-through scheme (Scheme 1), primarily through reduction of heat and cooling loads on the central processing components due to recirculation. The annual consumption of cooling energy is estimated at 215 MWh, heating energy at 168 MWh, and fan electricity at 42 MWh.

The d–h (Mollier) diagram analysis for the summer operating process confirmed that the mixing of recirculated return air (t = 27 °C, d = 10.8 g/kg) with outdoor air (t = 31 °C, d = 12.3 g/kg) at a 70:30 ratio yields mixed air at t_{mix} = 28.1 °C and d_{mix} = 11.2 g/kg. The cooling coil process in the d–h diagram proceeds from the mixed air point to the apparatus dew point at t_{app} = 10.2 °C, from which the supply air is reheated to the supply temperature of 14 °C. This process path is substantially more favourable energetically than cooling 100% outdoor air, since the recirculated air has lower enthalpy than outdoor air under peak summer conditions.

For the winter operating process, the outdoor air at t = −20 °C and d = 0.3 g/kg is first recovered in the enthalpy wheel (exit: t ≈ +2 °C, d ≈ 2.1 g/kg), pre-heated to +5 °C in the first-stage heater, humidified in the adiabatic spray chamber to d = 8.5 g/kg, and finally heated to the supply temperature of +26 °C. The recirculated return air contributes warmth and moisture to the mixed stream at the AHU inlet, reducing the pre-heater and humidifier loads. The calculated process satisfies the requirement of delivering air at 45–55 % RH to the conditioned spaces throughout the heating season.

Zone-level control performance was evaluated for three representative zones within the printing plant: the main pressroom (high equipment heat gain), the paper storage room (lower heat gain, stricter humidity control), and the pre-press studio (moderate gains, precision requirements). The dual-duct mixing box configuration allows each zone to independently select the ratio of warm and cold duct air to achieve the zone set-point temperature, while common duct conditions ensure that supply air humidity remains within tolerance for all zones simultaneously. Simulation of zone response to a step change in solar radiation gain (±20 % of peak) showed that zone temperature returns to within ±0.5 °C of set-point within 8–12 minutes, confirming adequate dynamic response.

The main limitation of Scheme 4 is its higher installation cost due to the dual-duct distribution system and the complexity of mixing box automation. The estimated installation cost premium over Scheme 1 is approximately 25–30%, which is offset by energy savings within a payback period of 6–8 years at current Ukrainian energy tariffs.

The results confirm that multi-zone optimization, combined with heat recovery and recirculation, provides a technically and economically justified solution for industrial air conditioning of printing facilities with demanding microclimate requirements.

7. CONCLUSIONS

1. A systematic comparative analysis of four multi-zone central ACS configurations was performed for a printing production facility in Mykolaiv, Ukraine. The dual-duct multi-zone ACS with first recirculation was identified as the optimal configuration based on zone temperature/humidity control capability, energy performance and operational suitability.
2. Thermal and moisture load calculations for summer (Q_{cool} = 97.2 kW) and winter (Q_{heat} = 62.1 kW) design conditions were completed. The required supply air flow of 18 500 m³/h with an outdoor air fraction of 30% was determined.
3. The central AHU was selected and all key components sized: cooling coil (Q = 97.2 kW), pre-heater (Q = 31.4 kW), adiabatic spray humidifier, G4+F7 two-stage filtration, and 7.5 kW supply fan at 780 Pa system resistance.
4. The energy analysis demonstrates an 18–22% reduction in annual energy consumption compared to single-duct once-through ACS, with an investment payback period of 6–8 years. These findings support the broader adoption of multi-zone recirculation-based ACS in Ukrainian industrial facilities facing high energy cost pressures.
5. Future work should address automatic control system design, integration of the ACS with a building energy management system, and validation of calculated parameters through monitoring of the installed system.

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ПРОМИСЛОВА СИСТЕМА КОНДИЦІЮВАННЯ ПОВІТРЯ ДЛЯ ПОЛІГРАФІЧНОГО ВИРОБНИЦТВА: ПРОЕКТУВАННЯ, РОЗРАХУНОК ТА БАГАТОЗОННА ОПТИМІЗАЦІЯ

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Анотація. *У статті представлено результати досліджень із проектування та оптимізації промислової системи кондиціювання повітря (СКП) для поліграфічного підприємства, розташованого в Миколаєві, Україна. Поліграфічні підприємства висувають суворі вимоги до параметрів мікроклімату приміщень — температури, відносної вологості та чистоти повітря — через чутливість паперових основ і друкарських фарб до умов навколишнього середовища. У цьому дослідженні аналізуються чотири конфігурації багатозонних центральних СКП: одноканальна прямоточна, одноканальна з першою рециркуляцією, двоканальна прямоточна та двоканальна з першою рециркуляцією. Порівняльний аналіз, заснований на розрахунках тепло- та вологонавантаження, демонструє, що двоканальна багатозонна СКП з першою рециркуляцією є найбільш енергоефективним рішенням для даного об'єкта. Розрахунок теплового навантаження виконано для літніх (зовнішня температура 31 °C, розрахункова внутрішня температура 25 °C) та зимових (зовнішня температура –20 °C, розрахункова внутрішня температура 20 °C) розрахункових умов. У роботі описано вибір центральної припливно-витяжної установки, розрахунок параметрів повітроохолоджувача, повітронагрівача, адіабатної форсуночної камери, повітряного фільтра та повітророзподільної мережі. Також представлено гідравлічний розрахунок трубопровідної мережі та вибір насоса. Результати підтверджують, що правильна конфігурація системи значно знижує річне споживання енергії на опалення, охолодження та зволоження порівняно з одноканальними альтернативами.*

Ключові слова: система кондиціювання повітря, багатозонна система, поліграфічне виробництво, теплове навантаження, рекуперація тепла, припливно-витяжна установка, енергоефективність.

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