International Journal of Energy and Power Engineering
2026, Vol. 15, No. 6, pp. 160–174
https://doi.org/10.11648/j.ijepe.20261506.11

SciencePG
Science Publishing Group

Research Article

Limits and Energy Consumption of Adiabatic Microclimate Control in Textile Shops Under the Climatic Conditions of Bukhara

Nasullo Sadullayev, Firdavs Muzaffarov, Firdavs Murtazoyev, Jasur Hojimurodov*

Department of Electrical and Power Engineering, Bukhara State Technical University, Bukhara, Uzbekistan

Abstract

In textile spinning and weaving shops, air temperature and relative humidity are process parameters. In a hot dry climate, adiabatic humidification of outdoor air in a spray chamber is energetically attractive, but its capability is limited by the outdoor air state. This study quantifies the limits of maintaining the indoor state of a textile shop adiabatically in the climate of Bukhara (Uzbekistan) and the air, water and electricity required. The heat and moisture balances were written jointly in enthalpy form, expressing the outdoor air fraction, the humidification load and the applicability condition of the adiabatic method through the indoor–outdoor enthalpy difference. Monthly Bukhara weather station data for 2021–2025 and a diurnal temperature model were used; the study is a model-based assessment not yet validated by measurements in an operating shop. For an indoor state of 26 ℃ and 60%, the enthalpy difference is 48.8 kJ/kg in January and 7.7 kJ/kg in July and approaches zero during the hot hours of July days. At the design airflow, the indoor state cannot be maintained adiabatically for 522 hours per year for the multi-year mean climate (489–706 hours in the individual years 2021–2024), and for another 306 hours this would require a spray chamber efficiency above 0.85. Humidification consumes 0.68 kg of water in January and 2.32 kg in July per 1 kWh of process electricity. The supply fan power equals 18.5% of the process power, and the fans account for 90–95% of the estimated annual operating cost of 1.9–2.0 million UZS (158–166 USD) per 1 kW of process power. A summer indoor state of 28 ℃ and 60% eliminates the restricted hours completely. A chiller operating only in the restricted hours would consume 64–102 kWh per 1 kW of process power per year, about one-eighth of the chiller electricity of a conventional system with a fixed outdoor air fraction, whereas the adiabatic system uses about three times more water in the warm season. The restricted hours depend mainly on the indoor design state and the outdoor humidity ratio. The results can support the design and the choice of the summer operating mode of air conditioning systems in textile enterprises in hot dry climates.

Keywords

Spinning Mill, Air Conditioning, Spray Chamber, Evaporative Cooling, Enthalpy, Humidity Ratio, Hot Dry Climate, Humidification Load, Sensitivity Analysis

*Correspondence: Jasur Hojimurodov (jasurbekhojimurodov@gmail.com)

Received: 4 November 2026; **Accepted:** 12 November 2026; **Published:** 25 November 2026

Copyright: © The Author(s), 2026. Published by Science Publishing Group. This is an **Open Access** article, distributed under the terms of the Creative Commons Attribution 4.0 License (http://creativecommons.org/licenses/by/4.0/), which permits unrestricted use, distribution and reproduction in any medium, provided the original work is properly cited.

1. Introduction

In textile production, the temperature and relative humidity of shop air directly affect product quality. The moisture content of fibre and yarn comes into equilibrium with air humidity. When relative humidity (RH) drops, yarn breakage, static electricity and dust generation increase, and yarn strength and evenness deteriorate [1, 2]. For this reason, the air state in spinning and weaving shops is maintained throughout the year within a temperature range of 20–26 ℃ and an RH range of 45–65% [3].

This task is usually performed by a central air conditioning system, in which a mixture of outdoor and return air is humidified adiabatically in a spray chamber and supplied to the production zone [3]. Microclimate systems are among the largest electricity consumers of a textile enterprise: in spinning mills, the humidification plant ranks among the largest electricity users after the spinning machines [4, 5]. Functional and neural network models have also been proposed for predicting the power demand of humidification plants [6].

In a hot dry climate, evaporative (adiabatic) cooling of outdoor air requires much less energy than mechanical cooling and is widely used in industrial buildings [7, 8]. The efficiency of this cooling process and the achievable degree of cooling are limited by the wet-bulb temperature of outdoor air and by the efficiency of the apparatus [9]. In textile enterprises, however, RH must also be held at a set value, i.e. the humidity ratio of the supply air is fixed in advance. In this case, the cooling capability of outdoor air is limited by its enthalpy as well. The need to model process heat and microclimate systems jointly has been shown in [10], and the benefits of adiabatic humidification in industrial spaces with strict humidity control have been shown in [11].

In Uzbekistan, deep cotton processing clusters are being established, and spinning and weaving capacities are expanding [12, 13]. A large share of these enterprises is located in regions with a hot dry climate, such as Bukhara. For such a climate, the extent to which an adiabatic system can maintain the indoor state throughout the year, and the amount of air, water and electricity required for this, have not been quantified.

The aim of this study is to determine the limits of maintaining the indoor state of textile shops by the adiabatic method under the climatic conditions of Bukhara and to estimate the air, water and ventilation energy required. It is shown that the outdoor air fraction, the humidification load and the feasibility of adiabatic control are governed by the difference between the enthalpies of indoor and outdoor air, and these quantities are evaluated for Bukhara on a monthly and hourly basis. In addition, the robustness of the results to the model parameters and to the climate data is analysed, and the adiabatic system is compared with mechanical and hybrid cooling in terms of energy, water and cost. The study is a model-based assessment: its results follow from the heat and moisture balances and weather station data and have not yet been validated against measurements in an operating shop.

2. Materials and Methods

2.1. Heat and Moisture Balance of the Shop

A spinning or weaving shop operating continuously is considered. The excess heat of the shop consists of the following components:

$$Q = Q_{\mathrm{eq}} + Q_{\mathrm{light}} + Q_{\mathrm{occ}} + Q_{\mathrm{env}} \tag{1}$$

$$Q_{\mathrm{eq}} = k_q P_{\mathrm{tech}} \tag{2}$$

where Q_{eq}, Q_{light}, Q_{occ} and Q_{env} are the heat gains from the equipment, lighting, occupants and building envelope, kW; P_{tech} is the electric power of the process equipment, kW; k_{q} is the fraction of the energy consumed by the equipment that is released to the shop air as heat. In spinning and weaving shops, the equipment heat is dominant; therefore, $Q = Q_{\mathrm{eq}}$ is assumed below (the influence of the neglected heat and moisture sources is estimated in Section 4.4). The supply airflow required to remove this heat and the fan power are:

$$L = \frac{Q}{c_p(t_{\mathrm{in}} - t_{\mathrm{sup}})} \tag{3}$$

$$N_{\mathrm{f}} = \frac{L\Delta p_{\mathrm{f}}}{\rho \eta_{\mathrm{f}}} \tag{4}$$

$$\omega = \frac{N_{\mathrm{f}}}{P_{\mathrm{tech}}} = \frac{k_q \Delta p_{\mathrm{f}}}{\rho c_p \eta_{\mathrm{f}}(t_{\mathrm{in}} - t_{\mathrm{sup}})} \tag{5}$$

where L is the mass flow rate of supply air, kg/s; c_{p} is the specific heat of air, kJ/(kg·K); t_{in} and t_{sup} are the temperatures of shop air and supply air, ℃; N_{f} is the fan power, kW; Δp_{f} is the total pressure of the ventilation system, kPa; ρ is the air density, kg/m^3; η_{f} is the efficiency of the fan and drive; ω is the ratio of fan power to process equipment power. According to (5), ω does not depend on the size of the shop: it is determined only by the difference between the indoor and supply air temperatures and by the characteristics of the ventilation system.

The state of air is described by its specific enthalpy and humidity ratio:

$$h = c_p t + d(r_0 + c_{\mathrm{v}} t) \tag{6}$$

$$d = \frac{0.622 \varphi p_{\mathrm{s}}(t)}{p - \varphi p_{\mathrm{s}}(t)} \tag{7}$$

$$p_{\mathrm{s}}(t) = 0.61094 \exp\left(\frac{17.625t}{t+243.04}\right) \tag{8}$$

where h is the specific enthalpy of air, kJ/kg; d is the humidity ratio, kg/kg; r_0 = 2501 kJ/kg is the latent heat of vaporization at 0 ℃; c_{v} = 1.86 kJ/(kg·K) is the specific heat of water vapour; φ is the relative humidity; p is the barometric pressure, kPa; p_{s} is the saturation vapour pressure, kPa, calculated by the Magnus formula [14].

If internal moisture sources are neglected, the humidity ratio of supply air must be equal to that of shop air in order to maintain humidity in the shop. Supply air is obtained by mixing outdoor air (fraction x) with return air and then humidifying the mixture adiabatically. Since enthalpy does not change in an adiabatic process:

$$d_{\text{sup}} = d_{\text{in}}; \quad h_{\text{sup}} = xh_{\text{out}} + (1-x)h_{\text{in}} \tag{9}$$

Since the humidity ratios of supply and shop air are equal, the difference between their enthalpies is determined by the temperature difference. The heat balance then takes the form:

$$Q = L\left(h_{\text{in}} - h_{\text{sup}}\right) = Lx\Delta h \tag{10}$$

$$\Delta h = h_{\text{in}} - h_{\text{out}} \tag{11}$$

where Δh is the difference between the enthalpies of shop air and outdoor air. We call it the *enthalpy reserve* of outdoor air: it shows how much heat 1 kg of outdoor air can remove from the shop. Equations (3) and (10) give the required outdoor air fraction, the humidification load and its ratio to the process power:

$$x = \frac{c_p(t_{\text{in}} - t_{\text{sup}})}{\Delta h} \tag{12}$$

$$G_{\text{w}} = Lx\Delta d = \frac{Q\Delta d}{\Delta h} \tag{13}$$

$$\Delta d = d_{\text{in}} - d_{\text{out}} \tag{14}$$

$$g = \frac{3600G_{\text{w}}}{P_{\text{tech}}} = \frac{3600k_q\Delta d}{\Delta h} \tag{15}$$

where G_{w} is the rate of water evaporated for humidification, kg/s; Δd is the humidity deficit of outdoor air, kg/kg; g is the humidification water per 1 kWh of electricity consumed by the process equipment, kg/kWh. According to (13) and (15), the humidification load is proportional to the process power consumption, and the proportionality coefficient depends only on the states of outdoor and indoor air.

2.2. Applicability Condition of the Adiabatic Method

The outdoor air fraction cannot exceed unity. Therefore, (12) yields the condition for maintaining the indoor state by the adiabatic method at the design airflow:

$$\Delta h \geq c_p\Delta t_{\text{d}}; \quad \Delta t_{\text{d}} = t_{\text{in}} - t_{\text{sup,d}} \tag{16}$$

Hours in which this condition is not satisfied are called *restricted hours*. In such hours, the system operates on 100% outdoor air but cannot remove the excess heat completely: the enthalpy of shop air rises to $h_{\text{out}} + c_p\Delta t_{\text{d}}$. The design supply air temperature is determined for the mean conditions of the hottest month (July) and the maximum efficiency of the spray chamber:

$$t_{\text{sup,d}} = t_7 - \varepsilon_{\max}\left(t_7 - t_{\text{wb},7}\right) \tag{17}$$

where t_7 and $t_{\text{wb},7}$ are the mean dry-bulb and wet-bulb temperatures of outdoor air in July,°C; $\varepsilon_{\max}$ is the maximum adiabatic efficiency of the spray chamber. The wet-bulb temperature is calculated by the Stull formula [15].

Even when condition (16) is satisfied, the spray chamber must be able to humidify the air to the required state. The required chamber efficiency is determined from the mixture temperature and the saturation temperature corresponding to the supply air enthalpy:

$$\varepsilon = \frac{t_{\text{mix}} - t_{\text{sup}}}{t_{\text{mix}} - t_{\text{s}}} \tag{18}$$

$$t_{\text{mix}} = xt_{\text{out}} + (1-x)t_{\text{in}} \tag{19}$$

where t_{mix} is the mixture temperature, ℃; t_{s} is the temperature of saturated air with the enthalpy h_{sup},°C. If the calculated ε exceeds $\varepsilon_{\max}$, the chamber cannot humidify the air to the required humidity ratio.

2.3. Climate Data and Calculation Procedure

The state of outdoor air was determined from the monthly mean temperature, mean maximum and minimum temperatures and relative humidity recorded at the Bukhara weather station (39.76 °N, 64.46 °E, 235 m above sea level) from January 2021 to August 2025 [16]. The barometric pressure was taken as p = 98.5 kPa according to the standard atmosphere at the station altitude. For each month, multi-year means were calculated from the 56-month series (Table 1). The humidity ratio of the air was assumed constant during the day, and its temperature was assumed to vary as:

$$t(\tau) = \bar{t} + \frac{t_{\max} - t_{\min}}{2}\cos\frac{2\pi(\tau - 15)}{24} \tag{20}$$

where τ is the hour of the day; $\bar{t}$, $t_{\max}$ and $t_{\min}$ are the monthly mean, mean maximum and mean minimum temperatures, ℃. The calculation was performed for each of the 24 hours of each month and multiplied by the number of days in the month. The following values were adopted: t_{in} = 26°C; φ_{in} = 60%; k_{q} = 0.95; c_{p} = 1.006 kJ/(kg·K); ρ = 1.15 kg/m^3; Δp_{f} = 0.8 kPa; η_{f} = 0.65; $\varepsilon_{\max}$ = 0.85. The calculations were performed in Python 3. The parameters and their sources are summarised in Table A1 (Appendix A), and the hourly intermediate results for July are given in Table A2 to allow independent verification. The generative AI tool Claude (Anthropic) was used for language editing and translation of the manuscript and for drafting the calculation and plotting scripts; all calculations, results and conclusions were checked by the authors.

Hourly observations of the station for 2021–2025 were not available to the authors; therefore, the monthly summaries were combined with the diurnal temperature model (20). Averaging over months and years smooths hot spells, so the restricted hours obtained for the multi-year mean climate should be regarded as a typical-year estimate. The effect of this simplification was checked by calculations for the individual years and with an assumed day-to-day variation of the daily mean temperature (Section 3.5).

Table 1. *Monthly mean outdoor air parameters and enthalpy reserve in Bukhara.*

Month	t, ℃	t_{max}, ℃	t_{min}, ℃	φ, %	d, g/kg	t_{wb}, ℃	h, kJ/kg	Δd, g/kg	Δh, kJ/kg
1	2.1	7.6	−3.0	74	3.33	−0.1	10.4	9.65	48.8
2	6.0	11.5	0.6	66	3.93	2.9	15.9	9.05	43.4
3	11.9	18.0	5.5	60	5.33	7.8	25.4	7.64	33.8
4	19.9	27.0	12.0	42	6.26	12.6	35.9	6.71	23.3
5	24.7	31.6	16.6	36	7.08	15.4	42.9	5.90	16.3
6	30.1	37.5	21.1	25	6.85	17.3	47.8	6.12	11.5
7	31.5	38.8	23.2	26	7.75	18.6	51.5	5.23	7.7
8	28.4	35.9	20.1	29	7.05	16.8	46.5	5.93	12.7
9	22.7	30.8	14.5	32	5.68	13.3	37.2	7.30	22.0
10	14.9	22.1	7.8	50	5.41	9.4	28.7	7.56	30.5
11	9.6	16.3	3.1	62	4.71	5.8	21.5	8.27	37.8
12	3.2	8.7	−1.9	69	3.40	0.6	11.8	9.58	47.4
Year	17.1	23.8	10.0	48	5.57	10.0	31.3	7.41	27.9

t_{wb}: wet-bulb temperature; Δd and Δh are calculated for the indoor state of 26 ℃ and 60% (d_{in} = 12.98 g/kg, h_{in} = 59.2 kJ/kg).

2.4. Hour Classification and Additional Analyses

Each hour of the year was assigned to one of three mutually exclusive categories. An hour is *restricted* if condition (16) is not satisfied. Of the remaining hours, an hour is *chamber-limited* if the required efficiency (18) exceeds ε_{max}; in such hours the shop temperature is held, but the humidity ratio of supply air remains below d_{in}. All other hours are *normal*. In a restricted hour, the additional cooling needed to hold t_{in} at the design airflow is:

$$Q_x = L\left(c_p \Delta t_d - \Delta h\right) \tag{21}$$

where Q_x is the additional cooling load, kW. This quantity was used to estimate a hybrid system in which a chiller supplements the adiabatic system only in the restricted hours. As a conventional reference, a mechanical system with a fixed outdoor air fraction x_{min} = 0.15 and an air-cooled chiller operating from May to September was considered. Its cooling coil load is:

$$Q_{ch} = Q - Lx_{min}\Delta h \tag{22}$$

where negative values of Q_{ch} were taken as zero. The chiller electricity was calculated with a coefficient of performance (COP) of 2.5–4.0, the range used in previous simulation studies [7, 11]. The airflow and fan power of the reference system were taken equal to those of the adiabatic system. The value x_{min} = 0.15 is an assumption of this comparison, close to the smallest outdoor air fraction of the adiabatic system in winter (Table 2).

The sensitivity of the results was assessed by varying one parameter at a time: indoor temperature 25–27 ℃, indoor RH 55–65%, ε_{max} 0.80–0.90, installed airflow ±10%, fan efficiency 0.55–0.75, outdoor temperature ±1 K, humidity ratio of outdoor air ±10% and amplitude of the diurnal temperature variation ±20%. Except for the ε_{max} and airflow cases, the installed airflow was kept at its design value. The climate data were also tested by repeating the calculation for each year of 2021–2025 and by superimposing a normally distributed day-to-day variation of the daily mean temperature with a standard deviation σ = 1–3 K (nine-point Gauss–Hermite quadrature). Because each hour of the mean day represents all days of the

month, the number of restricted hours changes in steps of 28–31 h.

3. Results

3.1. Enthalpy Reserve of the Bukhara Climate

Table 1 and Figure 1 show that in Bukhara the humidity ratio of outdoor air is below the shop requirement throughout the year. The humidity deficit varies from 5.23 g/kg in July to 9.65 g/kg in January; hence, air humidification is needed all year round. In summer the air is dry: in June, RH is 25% and the wet-bulb depression is 12.8 ℃. In a textile shop, however, this potential of adiabatic cooling can be used only within the enthalpy reserve. The enthalpy reserve is 48.8 kJ/kg in January and 7.7 kJ/kg in July. In July, at around 15:00, the air temperature reaches 39 ℃ and the outdoor air enthalpy (59.5 kJ/kg) exceeds that of shop air. In these hours, outdoor air cannot remove heat from the shop, and however much of it is supplied, the indoor state cannot be maintained adiabatically. In Figure 1, this region lies above the line $h = h_{in} - c_p \Delta t_d$. In June, July and August, the diurnal temperature variation enters this region.

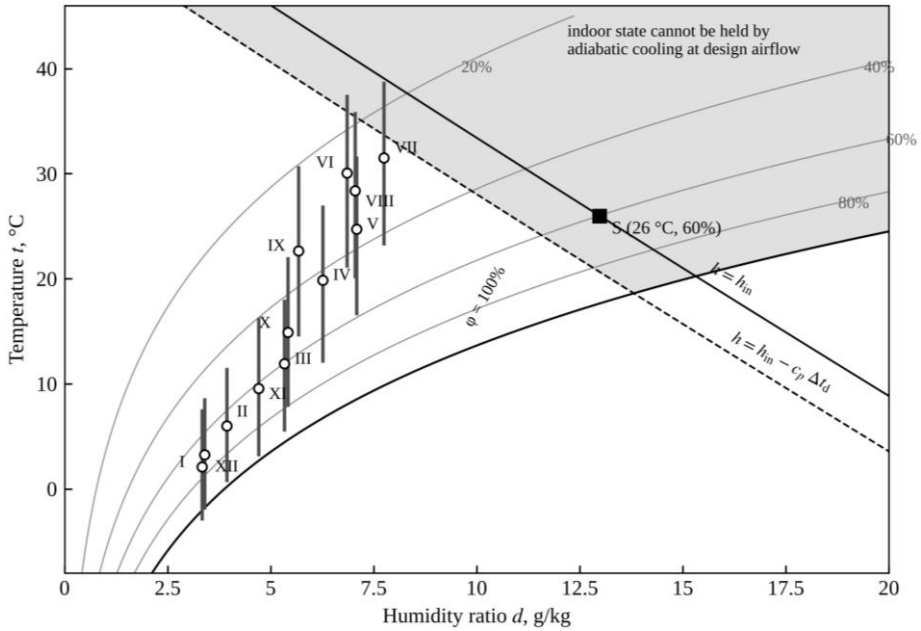

Figure 1. *Monthly mean outdoor air states in Bukhara (I–XII: months; vertical bars: diurnal temperature range) and the required indoor state S on the t–d diagram.*

According to (17), the design supply air temperature is 20.5°C and the temperature difference Δt_d is 5.47 K. This corresponds to 541 m^3/h of supply air per 1 kW of process power, and according to (5), ω = 0.185. Thus, the supply fans alone require additional electricity equal to 18.5% of the consumption of the process equipment. This share is inversely proportional to Δt_d.

3.2. Outdoor Air Fraction and Humidification Load

The monthly results are given in Table 2 and Figure 2. In December–March, owing to the large enthalpy reserve, only 11–16% of the supply air is outdoor air, and the rest is recirculated. In summer, the outdoor air fraction increases and reaches 0.71 on average in July.

Table 2. *Indicators of maintaining the indoor state (26 ℃, 60%) in Bukhara.*

Month	*x*	*g*, kg/kWh	Required ε	Restricted hours	*M*, kg/kW
1	0.11	0.68	0.61	0	507
2	0.13	0.71	0.63	0	484
3	0.16	0.77	0.66	0	586

Month	*x*	*g*, kg/kWh	Required ε	Restricted hours	*M*, kg/kW
4	0.24	0.98	0.75	0	749
5	0.34	1.23	0.83	0	1041
6	0.48	1.83	0.89	150	1624
7	0.71	2.32	0.87	279	1721
8	0.43	1.60	0.88	93	1490
9	0.25	1.14	0.79	0	882
10	0.18	0.85	0.70	0	649
11	0.15	0.75	0.65	0	548
12	0.12	0.69	0.62	0	517
Year	–	1.23	–	522	10 798

x and *g* are calculated from (12) and (15) for the monthly mean conditions; required ε is the maximum value in non-restricted hours, from (18); restricted hours and *M* (monthly humidification water per 1 kW of process power) are obtained from the hourly calculation. The annual *g* is the ratio of the annual water consumption to the annual process electricity consumption.

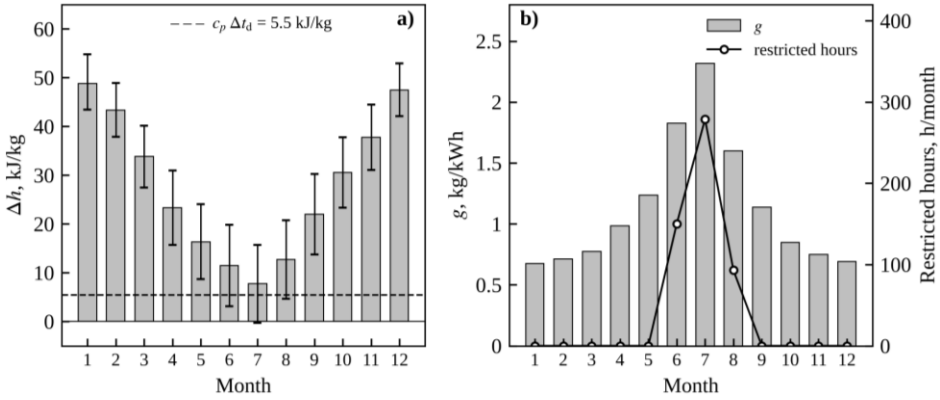

Figure 2. *Monthly values in Bukhara: a) enthalpy reserve of outdoor air (bars: monthly mean; error bars: diurnal range); b) ratio of the humidification load to the process consumption and the number of restricted hours.*

Per 1 kWh of electricity consumed by the process equipment, 0.68–0.71 kg of humidification water is required in winter and 2.32 kg in July. The annual mean is 1.23 kg/kWh, and the annual water consumption per 1 kW of process power is 10.8 t. The summer increase in the load is caused not by the humidity deficit but by the decrease in the enthalpy reserve. In July, the humidity deficit is 1.8 times smaller than in January, whereas the enthalpy reserve is 6.3 times smaller. As a result, more outdoor air has to be supplied to remove the same heat from the shop, and more water is spent to humidify it. The evaporated water absorbs heat from the air: the summer increase in water consumption is due to the water used for adiabatic cooling of outdoor air.

3.3. Restricted Hours and Spray Chamber Efficiency

At the design airflow, condition (16) is not satisfied for 522 hours per year (6.0% of the year): 150 hours in June, 279 hours in July and 93 hours in August. These hours fall between 11:00 and 19:00. If the shop humidity is maintained, the shop temperature rises in these hours by up to 5.6 K above 26 ℃. Increasing the airflow reduces the deviation, but when the enthalpy reserve approaches zero this measure also becomes ineffective. Therefore, in these hours either a temporary deviation of the indoor state is allowed or mechanical cooling is applied.

The second limitation is associated with the spray chamber efficiency. According to (18), the required efficiency does not exceed 0.83 from September to May. In June–August, however, in 306 hours that satisfy condition (16) but are close to the limit, it exceeds 0.85 and reaches 0.87–0.89. In these hours, the chamber cannot humidify the air to the required humidity ratio, and the RH of the shop air falls below the requirement. The missing moisture can be supplied by ultrasonic humidifiers installed in the working zones of the shop [17]. Water

evaporated inside the shop does not change the enthalpy of shop air and therefore does not affect balance (10). Thus, zonal humidifiers restore humidity in hours when the chamber efficiency is insufficient, but they cannot eliminate the restricted hours, in which the enthalpy reserve is insufficient. The moisture missing in the chamber-limited hours is small, about 24 kg per 1 kW of process power per year; at a specific consumption of about 0.06 kWh per kg [18], the electricity of the zonal humidifiers needed to supply it is negligible.

The accounting of all hours of the year is given in Table 3. The two categories do not overlap: the 522 restricted hours are caused by an insufficient enthalpy reserve, whereas the 306 chamber-limited hours occur when the reserve is sufficient but the required chamber efficiency exceeds 0.85. Together with 7932 normal hours, they make up the 8760 hours of the year. In July, the restricted hours fall between 11:00 and 19:00 and the chamber-limited hours occur at 10:00 and 20:00, immediately before and after the restricted period (Table A2).

Table 3. *Classification of the hours of the year (indoor state 26 ℃, 60%; design airflow).*

Month	Hours	Normal	Chamber-limited ($\varepsilon > \varepsilon_{max}$)	Restricted ($\Delta h < c_p \Delta t_d$)
1–5	3624	3624	0	0
6	720	450	120	150
7	744	403	62	279
8	744	527	124	93
9–12	2928	2928	0	0
Year	8760	7932	306	522

Restricted hours do not satisfy condition (16); chamber-limited hours satisfy (16) but require ε > 0.85 according to (18); normal hours satisfy both conditions.

3.4. Effect of the Indoor Design State

The required state of shop air differs between process stages [3]. Figure 3 shows how the number of restricted hours and the annual water consumption depend on the indoor state at a constant installed airflow (541 m^3/h per kW). The number of restricted hours is determined by the enthalpy of shop air. For example, the states of 26 ℃, 50% and 24 ℃, 60% have almost the same enthalpy (53.6 and 53.4 kJ/kg) and the same number of restricted hours, 1353. Increasing the RH of shop air from 60% to 65% reduces the restricted hours from 522 to 217, and raising the summer temperature to 28 ℃ at 60% RH eliminates them completely. Processes requiring low humidity (45–55%) are the most difficult for the adiabatic method. The annual water consumption, in contrast, depends weakly on the indoor state: 8.5–11.7 t/kW (Figure 3b).

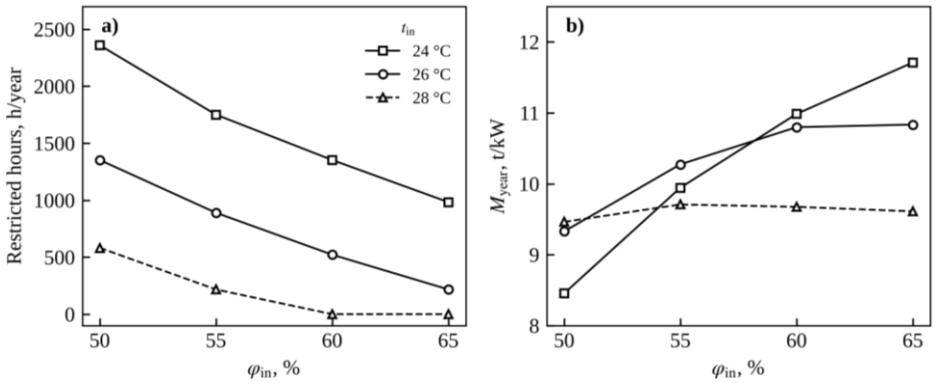

Figure 3. *Dependence on the temperature and relative humidity of shop air: a) number of restricted hours; b) annual humidification water per 1 kW of process power (installed airflow 541 m^3/(h kW)).*

The implications of raising the summer indoor temperature to 28 ℃ at 60% RH (June–August, with the installed airflow unchanged) are as follows. The enthalpy of shop air increases from 59.2 to 65.5 kJ/kg and its humidity ratio from 12.98 to

14.63 g/kg; the July enthalpy reserve increases from 7.7 to 14.0 kJ/kg. The restricted hours disappear, and the chamber-limited hours decrease from 306 to 155. The summer humidification water decreases by 20% (from 4.84 to 3.87 t/kW) and the annual water by 9% (from 10.80 to 9.83 t/kW). The fan power does not change, because the airflow and the supply air temperature difference remain the same (the supply air temperature rises to 22.5 ℃). The effect of 28 ℃ on the thermal comfort and heat stress of workers was not assessed in this study and should be evaluated before such a setting is adopted.

3.5. Sensitivity and Climate-Data Uncertainty

The results of the one-at-a-time sensitivity analysis are shown in Figure 4. The number of restricted hours is most sensitive to the indoor design state: lowering the indoor temperature to 25 ℃ or the indoor RH to 55% increases it from 522 to 890 h, and raising them to 27 ℃ or 65% decreases it to 217 h. Raising the indoor RH to 65%, however, increases the chamber-limited hours to 1408 h. Among the outdoor parameters, a 10% higher humidity ratio of outdoor air increases the restricted hours to 768 h, and a 1 K higher temperature or a 20% larger diurnal amplitude to 706 h. A lower chamber efficiency (ε_{max} = 0.80) reduces the design supply air temperature difference, so the required airflow increases to 613 $m^3/(h \cdot kW)$, ω to 21.0%, and the restricted hours decrease to 460 h; ε_{max} = 0.90 gives 484 $m^3/(h \cdot kW)$, ω = 16.5% and 706 h. A 10% smaller installed airflow increases the restricted hours to 706 h and the chamber-limited hours to 461 h. The fan efficiency affects only the fan power: ω = 16.0–21.8% for η_f = 0.75–0.55. For all parameters considered, the annual humidification water varies within 10.24–11.33 t/kW (±5%).

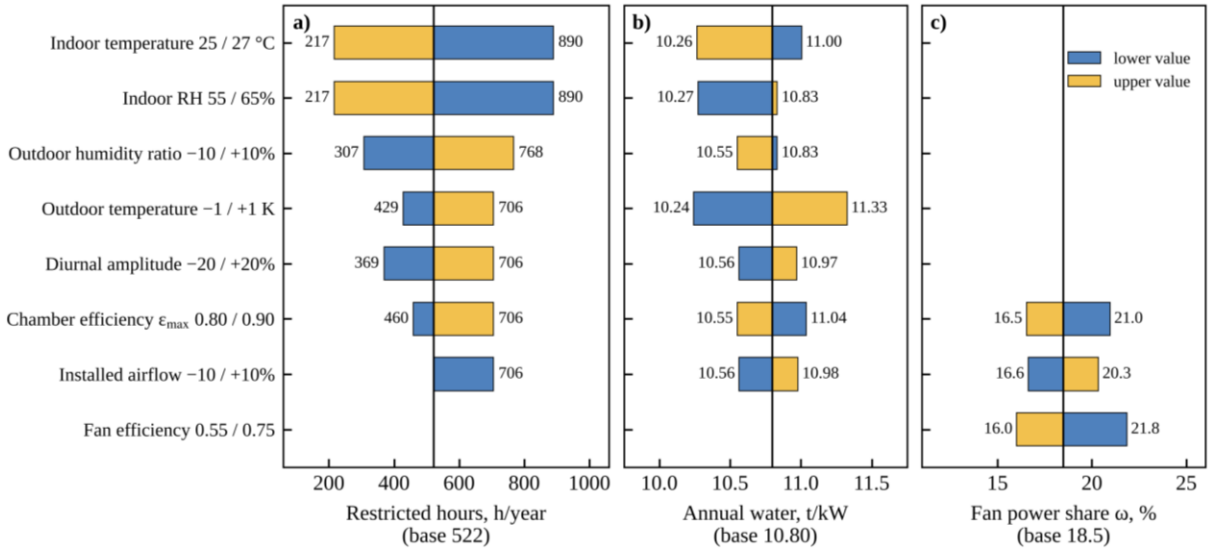

Figure 4. *Sensitivity of the results to the model parameters (one at a time): a) restricted hours; b) annual humidification water per 1 kW of process power; c) ratio of fan power to process power. Blue bars: lower parameter value; yellow bars: upper parameter value.*

The year-to-year variability is shown in Table 4. With the installed airflow fixed at the design value, the restricted hours ranged from 489 h in 2022 to 706 h in 2024; their mean over the complete years 2021–2024 is 606 h, i.e. 16% more than for the multi-year mean climate. The year 2024 was not the hottest, but its July air was the most humid (φ = 29.4%), which reduced the enthalpy reserve to 6.7 kJ/kg. Averaging the climate over years therefore underestimates the restricted hours. The superimposed day-to-day variation of the daily mean temperature has a smaller effect: for σ = 1, 2 and 3 K, the restricted hours are 549, 559 and 562 h, and the annual water consumption remains 10.8 t/kW. In the individual years, the annual water consumption was 10.3–11.2 t/kW.

Table 4. *Results for individual years (installed airflow fixed at the design value).*

Year	July *t*, ℃	July t_{max}, ℃	July φ, %	July Δ*h*, kJ/kg	Restricted hours	Chamber-limited hours	*M*, t/kW
2021	31.7	39.3	26.1	7.4	646	215	11.16
2022	31.1	38.6	26.5	8.4	489	215	10.48

Year	July t, ℃	July t_{max}, ℃	July φ, %	July Δh, kJ/kg	Restricted hours	Chamber-limited hours	M, t/kW
2023	31.7	39.2	25.7	7.7	584	184	10.86
2024	30.9	37.1	29.4	6.7	706	122	10.29
2025*	32.1	39.7	23.6	8.5	584	215	–
Mean climate	31.5	38.8	26.3	7.7	522	306	10.80

*January–August 2025; the annual water consumption is not given for this incomplete year. Mean climate: multi-year monthly means (Table 1).

3.6. Annual Resource Use and Operating Cost (Estimates)

Table 5 gives an estimate of the annual resource use and operating cost of the adiabatic system per 1 kW of process power. The electricity tariff of 1100 UZS/kWh (including VAT) for consumer groups I and II valid from 1 June 2026 [19] and the drinking water tariff for wholesale consumers in the Bukhara region of 9520 UZS/m^3 (including VAT) valid from 1 July 2025 [20] were used, and the costs were converted at 11 912 UZS/USD (4 November 2026) [21]. The evaporated water was taken from the model. Since evaporative systems also lose water through drain and bleed-off, an upper estimate was made with a factor of 1.9, which corresponds to the ratio of total to evaporated water reported for a direct evaporative cooling system [8]. The costs of pumps, water treatment and maintenance were not included.

Table 5. *Estimated annual resource use and operating cost per 1 kW of process power (Bukhara, 2026 tariffs).*

Item	Quantity per year	Cost, thousand UZS	Cost, USD
Supply fans (ω = 0.185, 8760 h)	1619 kWh	1781	149.5
Humidification water, evaporated	10.8 m^3	103	8.6
Humidification water including drain (×1.9)	20.5 m^3	195	16.4
Optional chiller for restricted hours (COP 2.5–4.0)	64–102 kWh	70–112	5.9–9.4
Total, fans and water (without chiller)	–	1884–1976	158–166

For a shop with 1000 kW of process equipment, the values are multiplied by 1000. Pumps, water treatment and maintenance are not included.

The supply fans account for 90–95% of the operating cost. For a shop with 1000 kW of process equipment, the annual cost is 1.9–2.0 billion UZS (158–166 thousand USD). Therefore, measures that reduce fan energy, such as variable-speed control of the fans (Section 4.3), have a larger economic effect than measures that reduce water consumption.

4. Discussion

4.1. Comparison with Previous Studies

In Bukhara, the wet-bulb depression of outdoor air is large (12.8 ℃ in June), which indicates a high potential of evaporative cooling [7-9]. The present analysis shows that when RH must also be held, as in textile shops [3], the usable part of this potential is bounded by the enthalpy reserve, which approaches zero on July afternoons. Therefore, in a hot dry climate, summer deviations of the indoor state are possible even when the spray chamber operates properly, whereas a lack of chamber efficiency mainly affects humidity and can be compensated by humidification inside the shop.

Previous studies of evaporative cooling in industrial buildings evaluated it mainly by comfort criteria, without a prescribed indoor humidity. De Angelis et al. [7] showed that direct and indirect evaporative cooling reduce electricity consumption by about 50% and 40% compared with a vapour-compression system and that internal gains, rather than the building envelope, govern the indoor conditions. Breteau et al. [8] found that direct adiabatic cooling of an industrial building reduces discomfort degree-hours most in a hot dry climate (by 48% in Abu Dhabi) and pointed out the trade-off between cooling, electricity and water use. Jo et al. [11] showed that

adiabatic humidification saves 8–23% of HVAC energy compared with steam humidification in a space with strict humidity control. For spinning mills, the share of humidification and air conditioning in the electricity bill or energy consumption was reported as about 15% [6] and 16% [22]. In the present model, the supply fans consume 18.5% of the process power, i.e. 15.6% of the total consumption of the process equipment and fans, which agrees with these values. The present study differs from [7, 8, 11] in that the indoor humidity ratio is fixed by the process: under this condition, the capability of outdoor air is limited by its enthalpy rather than by its wet-bulb temperature, and the restricted hours, the outdoor air fraction and the humidification load follow directly from the enthalpy reserve through (12), (13) and (16).

4.2. Comparison with Mechanical and Hybrid Systems

An indicative comparison of the adiabatic system with a hybrid and a conventional mechanical system for the warm season (May–September) is given in Table 6. The values were obtained from (21) and (22) with the assumptions stated in Section 2.4. The conventional system holds the indoor state in all hours, but it requires 2156 kWh of chiller cooling per 1 kW of process power, i.e. 539–862 kWh of electricity, and a chiller with a capacity close to the process heat. The adiabatic system needs no chiller, but it consumes 6.8 m^3 of water per kW in the warm season, about three times more than the mechanical system (2.1 m^3), and does not hold the indoor state in 522 h. The hybrid system holds the indoor state in all hours with only 64–102 kWh of chiller electricity, about one-eighth of that of the mechanical system. However, because the enthalpy reserve approaches zero on July afternoons, its chiller must be sized for almost the full process heat (0.99 kW per kW), although it operates only 522 h per year. The fan energy was taken equal for all systems; a mechanical system designed for a larger supply air temperature difference (7 K) would need 22% less fan energy (530 instead of 679 kWh), which does not change the ranking.

Table 6. *Indicative comparison of air conditioning options for the warm season (May–September) per 1 kW of process power.*

Indicator (May–September, per 1 kW of process power)	Adiabatic (this study)	Hybrid: adiabatic + chiller in restricted hours	Mechanical: chiller, x_{min} = 0.15
Indoor state 26 ℃, 60% held	except 522 h	all hours	all hours
Chiller cooling, kWh	0	254	2156
Chiller electricity (COP 2.5–4.0), kWh	0	64–102	539–862
Chiller operating hours, h	0	522	3672
Chiller capacity, kW per kW	0	0.99	0.96
Fan electricity, kWh	679	679	679 (530 at 7 K)
Humidification water, m^3	6.8	6.8	2.1
Reported electricity saving vs vapour compression	about 40–50% [7]	–	reference

Indicative values obtained from (21) and (22); air-cooled chiller; fan energy at the design airflow; spray chamber pumps are not included.

4.3. Implications for Design and Control

For a spinning shop with 1000 kW of process equipment, the results correspond to a design airflow of about 541 thousand m^3/h, a fan power of 185 kW and an annual humidification water consumption of about 10.8 thousand t. If the shop temperature is allowed to rise to 28 ℃ in summer, mechanical cooling is not required.

The results lead to the following recommendations. First, the airflow and the spray chamber should be selected together: at the design airflow, a chamber with ε_{max} = 0.90 instead of 0.85 eliminates the chamber-limited hours, whereas using the higher efficiency to reduce the airflow increases the restricted hours (Figure 4). Second, the outdoor air fraction should be controlled according to (12) by the measured enthalpies of indoor and outdoor air rather than by the dry-bulb temperature, which in summer does not indicate the cooling capability of outdoor air. Third, since the required airflow is proportional to the process heat (3), variable-speed fans can follow the actual load; in a system with quadratic pressure losses the fan power then decreases approximately with the cube of the airflow, e.g. to about 34% at 70% load. Given the cost structure in Table 5, this is the most effective measure. Fourth, for

June–August between about 11:00 and 19:00, the operating mode must be chosen in advance: a temporary temperature rise (up to 5.6 K at constant humidity), a summer set point of 28 ℃, or a trim chiller sized for about 1 kW per 1 kW of process power. Fifth, the humidification water (10.8 t/kW per year, plus drain) requires treatment and bleed-off control, which is important for a water-scarce region.

4.4. Limitations

The study is a model-based assessment. The model has not been validated against measurements in an operating textile shop, because such data were not available to the authors; the results should therefore not be regarded as experimentally confirmed. The agreement of the fan energy share with the values reported for spinning mills [6, 22] is only an indirect plausibility check.

The calculation neglects internal moisture sources, moisture absorption by the fibrous material and heat transfer through the building envelope. Internal moisture sources (occupants, wet processes) reduce the humidification load, so the water consumption is overestimated; in spinning shops with a low occupant density this effect is small. Fibre that enters the shop drier than the equilibrium state absorbs moisture and increases the humidification load: for an increase in moisture regain of 1–2% and a specific electricity consumption of spinning of 3.2–3.8 kWh/kg [22], this amounts to about 3–6 g per 1 kWh of process electricity, i.e. less than 1% of g. Heat transfer through the envelope adds a heat gain in summer and a heat loss in winter. Since condition (16) depends on the ratio of the heat load to the airflow, a summer envelope gain of 5–10% of the process heat has the same effect as a 5–10% smaller airflow and increases the restricted hours from 522 to about 706 h; in winter, envelope losses slightly reduce the outdoor air fraction and the water consumption. In industrial buildings with high internal loads, internal gains rather than the envelope govern the indoor conditions [7]. The humidity ratio of outdoor air was assumed constant during the day; its diurnal variation would change the restricted hours in the same way as the ±10% variation in Figure 4. The climate data are monthly summaries, and the multi-year averaging underestimates the restricted hours of individual years by up to about 35% (Table 4). Thermal comfort of workers at 28 ℃, the dynamic behaviour of the shop and its control system, and the capital costs were not assessed. Verifying these assumptions with hourly measurements at the research facility is a task for further work.

5. Conclusions

1. When temperature and RH are controlled jointly in a textile shop, the adiabatic cooling capability of outdoor air is determined not by the wet-bulb temperature but by the difference between the enthalpies of shop air and outdoor air. The required outdoor air fraction, the humidification load and the applicability condition of the adiabatic method were expressed through this difference.

2. In the climate of Bukhara, for an indoor state of 26 ℃ and 60%, the enthalpy reserve decreases from 48.8 kJ/kg in January to 7.7 kJ/kg in July and approaches zero during the hot hours of July days. At the design airflow, the indoor state cannot be maintained adiabatically for 522 hours per year, and in another 306 hours the spray chamber efficiency is insufficient. In the individual years 2021–2024, the restricted hours ranged from 489 to 706 h.

3. The humidification load is proportional to the process consumption: 0.68–2.32 kg of water, 1.23 kg on average per year, is required per 1 kWh of electricity consumed by the process equipment. The power of the supply fans equals 18.5% of the power of the process equipment. The supply fans account for 90–95% of the estimated annual operating cost of 1.9–2.0 million UZS (158–166 USD) per 1 kW of process power.

4. The number of restricted hours is determined by the enthalpy of shop air. Changing the summer indoor state from 26 ℃ and 60% to 28 ℃ and 60% eliminates the restricted hours completely and reduces the annual water consumption by 9%; its effect on worker comfort requires separate assessment. In hours when the chamber efficiency is insufficient, humidity can be supplemented by zonal ultrasonic humidifiers.

5. A chiller operating only in the restricted hours would consume 64–102 kWh per 1 kW of process power per year, about one-eighth of the chiller electricity of a conventional system with a fixed outdoor air fraction, but it must be sized for almost the full process heat. In the warm season, the adiabatic system uses about three times more water than the conventional one.

6. The restricted hours are most sensitive to the indoor design state and to the humidity ratio of outdoor air; the annual water consumption changes by no more than ±5% within the parameter ranges considered.

Future research will include hourly measurements of the indoor and outdoor air states, airflow, water and fan electricity consumption in a spinning shop in Bukhara to validate the model; the use of hourly weather data; the inclusion of envelope heat transfer, internal moisture sources and fibre moisture sorption; dynamic simulation of the shop with enthalpy-based control of the outdoor air fraction; and the assessment of hybrid schemes combining the spray chamber with indirect evaporative pre-cooling, a trim chiller and zonal ultrasonic humidifiers [17], including worker comfort and the treatment of humidification water.

Abbreviations

RH	Relative Humidity
COP	Coefficient of Performance
UZS	Uzbekistani Som
VAT	Value Added Tax

Author Contributions

Nasullo Sadullayev: Conceptualization, Supervision, Writing – review & editing

Firdavs Muzaffarov: Methodology, Validation, Writing – review & editing

Firdavs Murtazoyev: Formal Analysis, Validation, Writing – review & editing

Jasur Hojimurodov: Conceptualization, Data curation, Formal Analysis, Investigation, Software, Visualization, Writing – original draft

Data Availability Statement

The data supporting the outcome of this research work has been reported in this manuscript. The climate input data are given in Table 1, and the model parameters and hourly intermediate results in Tables A1 and A2; the calculation script is available from the corresponding author upon reasonable request.

Conflicts of Interest

The authors declare no conflicts of interest.

Appendix: Model Parameters and Intermediate Results

Table A1. *Model parameters and input data.*

Parameter	Symbol	Value	Source / note
Indoor temperature	t_{in}	26 ℃ (25–27 ℃)	process requirement [3]
Indoor relative humidity	φ_{in}	60% (55–65%)	process requirement [3]
Heat fraction of process power	k_q	0.95	adopted value
Specific heat of air	c_p	1.006 kJ/(kg ·K)	–
Latent heat at 0 ℃; specific heat of vapour	r_0; c_v	2501 kJ/kg; 1.86 kJ/(kg ·K)	–
Air density	ρ	1.15 kg/m^3	–
Barometric pressure	p	98.5 kPa	standard atmosphere, 235 m
Total pressure of ventilation system	Δp_f	0.8 kPa	adopted design value
Fan and drive efficiency	η_f	0.65 (0.55–0.75)	adopted design value
Maximum chamber efficiency	ε_{max}	0.85 (0.80–0.90)	adopted design value
Design supply air temperature	$t_{sup,d}$	20.53 ℃	from (17)
Design temperature difference	Δt_d	5.47 K	from (17)
Design airflow	L	541 m^3/(h ·kW)	from (3)
Outdoor air fraction of mechanical reference	x_{min}	0.15	adopted for comparison
Chiller coefficient of performance	COP	2.5–4.0	[7, 11]
Outdoor air states	–	monthly means, 01.2021–08.2025	[16], Table 1
Diurnal temperature profile	–	cosine, maximum at 15:00	(20)

Values in parentheses are the ranges used in the sensitivity analysis.

Table A2. *Hourly intermediate results for July (multi-year mean climate, indoor state 26 ℃, 60%, design airflow).*

Hour	t, ℃	h, kJ/kg	Δh, kJ/kg	x	Required ε	Category
00:00	26.0	45.9	13.3	0.41	0.74	normal
01:00	24.7	44.6	14.6	0.38	0.72	normal

Hour	t, ℃	h, kJ/kg	Δh, kJ/kg	x	Required ε	Category
02:00	24.0	43.8	15.4	0.36	0.71	normal
03:00	23.7	43.6	15.7	0.35	0.70	normal
04:00	24.0	43.8	15.4	0.36	0.71	normal
05:00	24.7	44.6	14.6	0.38	0.72	normal
06:00	26.0	45.9	13.3	0.41	0.74	normal
07:00	27.6	47.5	11.7	0.47	0.76	normal
08:00	29.5	49.5	9.8	0.56	0.79	normal
09:00	31.5	51.5	7.7	0.71	0.83	normal
10:00	33.5	53.6	5.7	0.97	0.87	chamber-limited
11:00	35.4	55.5	3.7	1.00	–	restricted
12:00	37.0	57.2	2.1	1.00	–	restricted
13:00	38.3	58.4	0.8	1.00	–	restricted
14:00	39.0	59.2	0.0	1.00	–	restricted
15:00	39.3	59.5	−0.2	1.00	–	restricted
16:00	39.0	59.2	0.0	1.00	–	restricted
17:00	38.3	58.4	0.8	1.00	–	restricted
18:00	37.0	57.2	2.1	1.00	–	restricted
19:00	35.4	55.5	3.7	1.00	–	restricted
20:00	33.5	53.6	5.7	0.97	0.87	chamber-limited
21:00	31.5	51.5	7.7	0.71	0.83	normal
22:00	29.5	49.5	9.8	0.56	0.79	normal
23:00	27.6	47.5	11.7	0.47	0.76	normal

Each row represents 31 days. In restricted hours the system operates on 100% outdoor air (x = 1) and the required state cannot be reached.

References

[1] Nield, R., Ali, A. R. A. Some Effects of Humidity in Open-End Spinning. Part I: The Effect of the Relative Humidity During Spinning on Yarn Properties. Journal of the Textile Institute. 1977, 68(1), 12-17. https://doi.org/10.1080/00405007708631461

[2] Gulsevincler, E., Usal, M. R., Yilmaz, D. The Effect of Humidified Air on Yarn Properties in a Jet-Ring Spinning System. Tekstilec. 2020, 63(4), 294-304. https://doi.org/10.14502/Tekstilec2020.64.294-304

[3] Gvozdkov, A. N., Suslova, O. Yu. Regulation of Heat and Humidity Parameters of the Supply Air Treated in the Air Conditioning System. Izvestiya Vysshikh Uchebnykh Zavedenii, Seriya Teknologiya Tekstil'noi Promyshlennosti. 2019, 3(381), 186-191. (In Russian)

[4] Hasanbeigi, A., Price, L. A Review of Energy Use and Energy Efficiency Technologies for the Textile Industry. Renewable and Sustainable Energy Reviews. 2012, 16(6), 3648-3665. https://doi.org/10.1016/j.rser.2012.03.029

[5] Gunturu, K. P. K., Kota, K. K., Sharma, M. Energy Efficiency Improvement Opportunities in Indian Textile Industries. Textile & Leather Review. 2022, 5, 296-326. https://doi.org/10.31881/TLR.2022.13

[6] Rajasekaran, S., Prakasam, R., Dhandapani, S., David, V. Prediction of Humidification Power in Textile-Spinning Mills Using Functional and Neural Networks. Journal of Energy Engineering. 2003, 129(1), 2-15. https://doi.org/10.1061/(ASCE)0733-9402(2003)129:1(2)

[7] De Angelis, A., Saro, O., Truant, M. Evaporative Cooling Systems to Improve Internal Comfort in Industrial Buildings. Energy Procedia. 2017, 126, 313-320. https://doi.org/10.1016/j.egypro.2017.08.245

[8] Breteau, A., Bozonnet, E., Salagnac, P., Caous, J.-M. Specific Metrics for Direct Adiabatic Cooling of Industrial Buildings and Climate Adaptation. Energy and Buildings. 2025, 332, 115472. https://doi.org/10.1016/j.enbuild.2025.115472

[9] Cheban, D. N., Doroshenko, A. V. The Use of Porous Ceramics for Evaporative and Evaporative-Vapor-Compression Systems. Problemele Energeticii Regionale. 2013, 1(21), 57-64. (In Russian)

[10] Mawson, V. J., Hughes, B. R. Thermal Modelling of Manufacturing Processes and HVAC Systems. Energy. 2020, 204, 117984. https://doi.org/10.1016/j.energy.2020.117984

[11] Jo, M.-S., Shin, J.-H., Kim, W.-J., Jeong, J.-W. Energy-Saving Benefits of Adiabatic Humidification in the Air Conditioning Systems of Semiconductor Cleanrooms. Energies. 2017, 10(11), 1774. https://doi.org/10.3390/en10111774

[12] Khalikov, T., Prus, Yu., Chelombitko, T., Urazov, K., Shirinov, U., Ratynskiy, V. Cotton Textile Industry. E3S Web of Conferences. 2023, 452, 01002. https://doi.org/10.1051/e3sconf/202345201002

[13] Rakhmonov, I., Toirov, M., Obidov, K., Uzakov, N. Textile Process and Electrical Equipment Used in the Development and Production of the Textile Industry in Uzbekistan. E3S Web of Conferences. 2023, 461, 01079. https://doi.org/10.1051/e3sconf/202346101079

[14] Alduchov, O. A., Eskridge, R. E. Improved Magnus Form Approximation of Saturation Vapor Pressure. Journal of Applied Meteorology. 1996, 35(4), 601-609. https://doi.org/10.1175/1520-0450(1996)035<0601:IMFAOS>2.0.CO;2

[15] Stull, R. Wet-Bulb Temperature from Relative Humidity and Air Temperature. Journal of Applied Meteorology and Climatology. 2011, 50(11), 2267-2269. https://doi.org/10.1175/JAMC-D-11-0143.1

[16] Tutiempo Network. Climate Bukhara (Station 381415): Monthly Climate Data. Available from: https://en.tutiempo.net/climate/ws-381415.html (accessed 30 October 2026)

[17] Sadullayev, N. N., Hojimurodov, J. E., Muzaffarov, F. F. Ultrasonic Air Humidifier. Utility Model Patent No. FAP 3132 (Application No. FAP 20260248). Tashkent: Ministry of Justice of the Republic of Uzbekistan; 2026. (In Uzbek)

[18] STULZ GmbH. Ultrasonic Humidifiers. Available from: https://www.stulz.com/en-in/products/detail/ultrasonic-humidifier/ (accessed 11 November 2026)

[19] Cabinet of Ministers of the Republic of Uzbekistan. Resolution No. 243 of 15 May 2026 “On Setting Prices for Fuel and Energy Resources”. Available from: https://lex.uz/uz/docs/-8195731 (accessed 11 November 2026). (In Uzbek)

[20] Spot. uz. Drinking Water and Wastewater Tariffs in Bukhara to Rise from 1 July. 23 June 2025. Available from: https://www.spot.uz/oz/2025/06/23/bukhara-water/ (accessed 11 November 2026). (In Uzbek)

[21] Zamin. uz. Exchange Rates for November 4 Announced. 4 November 2026. Available from: https://zamin.uz/en/economy/226259-exchange-rates-for-november-4-announced.html (accessed 11 November 2026)

[22] Koç E., Kaplan, E. An Investigation on Energy Consumption in Yarn Production with Special Reference to Ring Spinning. Fibres & Textiles in Eastern Europe. 2007, 15(4), 18-24.

Biography



Nasullo Sadullayev is a professor at the Department of Energy Audit, Bukhara State Technical University, Uzbekistan. He holds a Doctor of Technical Sciences (DSc) degree. His research focuses on energy efficiency, energy auditing and reliable power supply, including micro-grids and energy storage systems for remote and low-power consumers and the integration of renewable energy sources. He is the author of monographs on monitoring technical losses in 6–10 kV distribution networks and industrial energy consumption. His recent work addresses energy efficiency indicators for industrial facilities, cooling of photovoltaic modules, and the energy performance of air-cooled heat exchangers.



Firdavs Muzaffarov is an associate professor (docent) at Bukhara State Technical University, Uzbekistan. He holds a PhD degree. His research focuses on renewable energy sources and the energy efficiency of electrical and thermal power equipment. He has published works on the mathematical modelling of vertical-axis wind turbines, the cost-effective supply of alternative energy to consumers without access to the electricity grid, the use of photosensors and photorelays in solar power transmission systems, the causes of failure in DC motors, and the rational use of energy resources.



Firdavs Murtazoyev is a doctoral student (basic doctoral studies) at Bukhara State Technical University, Department of Electrical and Power Engineering. His research focuses on the energy efficiency of air coolers for natural gas, evaporative pre-cooling of the cooling air and two-stage humidification. He has built and operated an experimental rig for evaporative pre-cooling of a finned-tube air cooler in Bukhara and has developed registered software for thermal calculations of gas air coolers.



Jasur Hojimurodov is a doctoral student (basic doctoral studies) at the Department of Electrical and Power Engineering, Bukhara State Technical University. His research focuses on the energy efficiency of textile enterprises, including product energy intensity, the operating regimes of production shops and the energy consumption of microclimate systems with adiabatic and ultrasonic air humidification. He has developed heat and moisture balance models of textile shops for hot dry climates.

Research Field

Nasullo Sadullayev: Power Engineering, Energy Saving, Heat

and Mass Transfer, Renewable Energy, Industrial Energy Efficiency

Firdavs Muzaffarov: Power Engineering, Heat Exchangers, Energy Efficiency, Thermal Engineering, Energy Auditing

Firdavs Murtazoyev: Natural Gas Cooling, Air-Cooled Heat Exchangers, Evaporative Cooling, Heat and Mass Transfer, Industrial Energy Efficiency, Psychrometrics

Jasur Hojimurodov: Energy Efficiency, Textile Industry, Microclimate Systems, Air Humidification, Heat and Moisture Balance