IOP Conference Series: Earth and Environmental Science

PAPER • **OPEN ACCESS**

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To cite this article: Kh Isakhodjayev *et al* 2021 *IOP Conf. Ser.: Earth Environ. Sci.* **939** 012025

IOP Conf. Series: Earth and Environmental Science **939** (2021) 012025 doi:10.1088/1755-1315/939/1/012025

Development of a laboratory nozzle chamber installation for the humidification of buildings

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Abstract. The nozzle chamber, in which water is sprayed into the air stream using mechanical nozzles, is the main unit for these processes in central air conditioning systems (AHUs). The types of nozzles used do not have a sufficiently high effect of interfacial surface forming due to increased metal usage and the broad total dimensions of certain chambers, i.e., they do not have intensive heat and mass transfer. The authors performed testing of the apparatus in the direct iso-enthalpic air cooling mode to improve the performance of the nozzle chamber. Thus, the experiments conducted confirm the relatively high efficiency of FET operation at small values of irrigation coefficient $\mathrm{B} \leq 1.0$. The area highlighted is characterised by the unstable operation of other nozzle types. Therefore, FET nozzles can be operated at irrigation factor values $\mathrm{B} = 0.1 \dots 1.0$. Experiments have shown that this equation is applicable for practical calculations, with a relative error of $\pm$ 6.7 %. The aerodynamic resistance of the spray chamber nozzle chambers is also according to the data not exceeding 160 Pa.

1. Introduction

The key unit for these processes in central air conditioning systems (ACS) is the nozzle chamber. It uses mechanical nozzles to atomize the airstream with vapor. The number of nozzles can be as high as many hundred, resulting in increased metal consumption and wide chamber dimensions overall. This suggests that the nozzle forms used do not have a sufficiently strong interfacial surface forming effect, i.e., do not have an intense transfer of heat and mass [1-5].

At present, in Uzbekistan and abroad, energy saving measures are of particular importance and are one of the main tasks associated with the economic development of the state [6-19].

Ventilation and air conditioning systems (VACS) are among the most energy-intensive in the production and housing and communal services of the country. Therefore, the main measures in the power supply of buildings for various purposes should be to reduce the consumption of heat and electricity in heat supply, ventilation and air conditioning systems [20-22].

Reducing energy consumption in VACS largely depends on the use of highly efficient and reliable devices for heat and moisture treatment of air [23]. Irrigation chambers are one of the main heat and mass transfer devices for air treatment in ventilation and air conditioning systems. Improvement of air conditioning installations using irrigation chambers with transverse nozzles allows improving the quality of the processed air, which increases the efficiency of these devices and, accordingly, reduces energy consumption.

Thus, the problem of the lack of energy-efficient and reliable contact devices for heat and humidity treatment of air in ventilation and air conditioning systems implies a number of measures to study and improve the heat engineering and operational characteristics of these devices.

2. Methods

In order to improve the efficiency of the nozzle chamber, the apparatus was tested in direct iso-enthalpy air cooling mode. The experiments were carried out on the laboratory unit. Schematic diagram of the laboratory setup is shown in Figure 1. Linear dimensions of the nozzle chamber corresponded to $0.36 \times 0.40 \times 0.78\ m$.

The unit has two lines, an airline and a water line. The air line consists of a radial fan (1) with a duct on which a gate valve (2), control section I-I, a micromanometer (12) with a pressure gauge tube (11) and a glass thermometer (13) are located. The air duct is connected to the nozzle chamber (3) where the nozzles (4) are attached to the risers. The lower part of the chamber is a sump (5), which serves for collecting water. The water line of the nozzle chamber consists of a centrifugal pump (6), control valve (7), pressure gauge (8), water meter (9) and by-pass valve (10)

The air flow rate in the ducts and the nozzle chamber is set by means of the slide gate (2). The flow rate to the nozzle chamber is controlled by the bypass valve (10). The control valve (7) serves mainly to regulate the water line pressure. The air flow in the duct is measured by means of a combined pressure receiver with a micromanometer. A glass thermometer (13) is used to measure the air temperature at the nozzle chamber inlet.

The unit works as follows: the air from the room is blown by a fan (1) through a duct into the nozzle chamber (3) and discharged into the adjacent room; a pump (6) is used to recirculate the water in the nozzle chamber; water from the chamber sump is pumped through the pressure gauge (8), water meter (9) and sprayed through nozzles (4) in the air stream; part of the water evaporates and passes as water vapour into the air (adiabatic humidification of the air), another part of the water (the main part) collects back into the sump, etc.

This nozzle chamber is similar in design and operation to the spray chambers of central air conditioners.

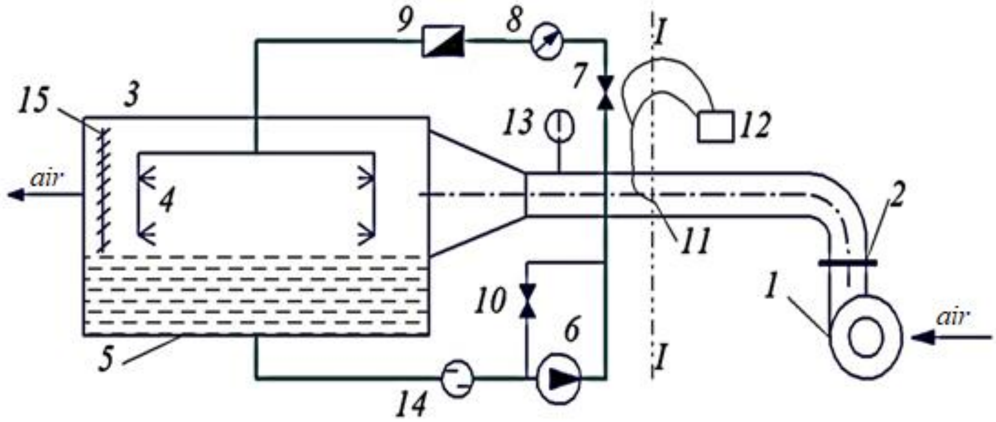

Figure 1. Diagram of laboratory installation: 1 - radial fan, 2 - gate valve, 3 - nozzle chamber, 4 - nozzles, 5 - pan, 6 - centrifugal pump, 7 - regulating valve, 8 - manometer, 9 - water meter, 10 - overflow valve, 11 - combined pressure receiver, 12 - micro-manometer, 13 - glass thermometer, 14 - filter, 15 - drift eliminator

The operation of the nozzle chamber in adiabatic mode is evaluated by the irrigation coefficient (B), the pressure value in front of the nozzles (P_n), the nozzle output (G_n), the efficiency factor (E_a). The formulas for determining the nozzle chamber characteristics are as follows

$$B = \frac{G_W}{G_a}, \tag{1}$$

$$G_n = \frac{G_W}{n}, \tag{2}$$

where G_W – water consumption in the chamber $\left(\frac{kg}{hour}\right)$, G_a – chamber flow rate $\left(\frac{kg}{hour}\right)$, n – the number of working nozzles in the chamber.

The efficiency factor is determined by equation (3)

$$E_a = \frac{t_0 - t_K}{t_0 - t_{M_1}} \tag{3}$$

In order to compare the performance of the ejection type nozzle (FET) with the known eccentric wide-flow nozzle (ESF) 7/10, an additional series of experiments was conducted where the factor space was increased. Thus, the irrigation factor was varied from 0.1 to 2.5. The resulting graphical relationship $E_A = f(B)$ for FET is shown in Figure 2. In the interval 0.1...1.0 the efficiency factor increased sharply. For comparison, Figure 2 shows the dependences of $E_A = f(B)$ for the specified types of nozzles. They are taken from [3, 4].

At B ≥ 1.1 the efficiency of the FET is comparable to that of the wide flare nozzles, at B = 2.5 its efficiency is slightly inferior to the latter.

3. Results and Discussions

As can be seen from Figure 2, the FET nozzle had a high efficiency at low irrigation coefficient values. In particular, at B = 0.6 the efficiency factor $E_A = 0.64$. For typical nozzles in this area efficiency factor is twice lower and corresponds to $E_A = 0.3 \ldots 0.36$. Sufficiently high coefficient of efficiency shows FET at B ≤ 0.6. This area corresponds to the steady operation of the nozzle and can be considered as a working area in the design of the contact apparatus. Practically it can be increased up to B = 1.0. This area is highlighted in color in Figure 2.

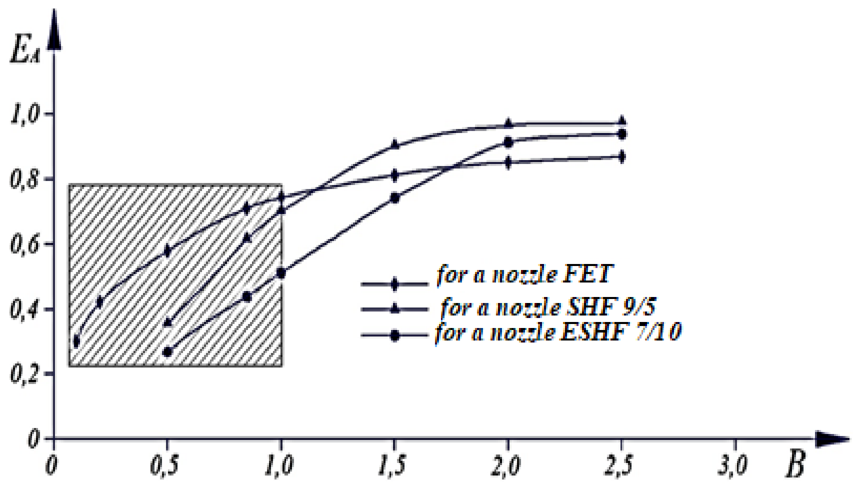

Figure 2. Dependence $E_A = f(B)$ for different nozzle types

Therefore, the experiments conducted confirm the relatively high efficiency of the FET at small values of irrigation coefficient B = 0.1 ... 1.0. The highlighted area is characterized by the unstable operation of other nozzle types. Therefore, FET nozzles can be operated at values of irrigation factor B = 0.1 ... 1.0.

The equation approximating the graphical relationship $E_A = f(B)$ (Figure 2) for FET nozzles can be represented as a logarithmic relationship

$$E_A = 0.74 + 0.19 \cdot e^{ln\,B} \tag{4}$$

From where the irrigation factor (if an efficiency factor is given) can be determined

$$B = exp\left(\frac{E_A - 0{,}74}{0{,}19}\right) \tag{5}$$

$$t_k - t_0 = E_A \cdot (t_w - t_0) + 0.329 \cdot \left(1 - \frac{E_A}{a_1}\right) \cdot (i_k - i_0) \quad (6)$$

All calculations were carried out using MathCAD and Microsoft Excel. The procedure and calculations are given in table.

The general equation describing the heat and humidity treatment of air in typical chambers with ejector-type nozzles can be obtained by substituting equation (4) into equation (6)

$$t_k - t_0 = \left(0.74 - 0.19 \cdot e^{\ln B}\right) \cdot (t_w - t_0) + 0.33 \cdot \left(1 - \frac{0.74 - 0.19 \cdot e^{\ln B}}{E_n}\right) \cdot (i_k - i_0), \quad (7)$$

where $E_n = a_1$ is the reduced enthalpy efficiency factor, which is determined by the formula

$$E_n = \frac{1 - exp[F \cdot ln(1 - E_a)]}{\Phi}, \quad (8)$$

where

$$F = \left(1 + \frac{0{,}725}{\mu}\right)\left[1 + \mathrm{c} \cdot (-\,ln(1 - E_A))^{-0{,}858}\right] \quad (9)$$

It is established from literature [3, 4, 5] that the efficiency of the nozzle depends on the parameters of the design and operation, in particular the diameter of the nozzle and the water pressure before the nozzle. Based on their type, the efficiency of nozzles is calculated by analytical equations. The following form is a mathematical description of this dependency

$$g_n = a \cdot p^m \cdot d^n, \quad (10)$$

where p – is water pressure upstream of the nozzle (kPa), d is nozzle diameter (m), a, m, n are coefficients and degree ratios depending on the nozzle design.

Based on literature data [1÷8], a general form of the design equation for calculating the aerodynamic drag of a nozzle chamber equipped with ejector-type nozzles has been determined

$$\Delta \mathrm{P} = \mathrm{A} \cdot (\rho \cdot \upsilon)^n, \quad (11)$$

where ΔP – is the aerodynamic drag (Pa), (ρ·υ) – is the mass velocity of air flow in the nozzle chamber $\left(\frac{kg}{м^2 \cdot sec}\right)$, A, n – are coefficients.

To derive the type equation (10) for the ejection type nozzle, experimental data from bench tests were used. These are shown in Table 1.

Table 1. Experimental values for nozzle output

Water pressure $P_{\boldsymbol{w}}$, kPa	Nozzle output $\frac{\boldsymbol{kg}}{\boldsymbol{hour}}$, at nozzle diameter $\boldsymbol{d}$					
	2		4		6	
	experience	formula	experience	formula	experience	formula
147	135	153	292	307	468	461
196	182	191	361	382	616	573
245	227	226	430	452	764	679

Table 2. Experimental data for determining aerodynamic drag

№ experiences	Air speed, *м/с*	Air temperature, ^0C	Mass speed, $(\boldsymbol{\rho} \cdot \boldsymbol{\upsilon})$	Resistance, P*a*
1	2.1	16.9	2.52	13.4
2	3.2	17.1	3.84	35.3
3	4.6	17.2	5.57	81.4
4	4.9	17.4	5.88	93.2
5	5.4	17.6	6.48	117.7
6	6.3	18.1	7.42	167.8

A micromanometer was used to experimentally determine the ΔP value at different operating modes.

The pressure measurement points corresponded to the inlet and outlet of the air flow in the nozzle chamber.

The mass velocity of the air was calculated from the linear velocity. The airflow temperature was determined using a dry Assmann psychrometer thermometer. The experimental data are given in Table 2. Based on the data in Table 2, the coefficients in Equation 11 were determined using Microsoft Excel software package. At that, $\mathrm{A} = 1.6$ and $n = 2.3$.

To calculate the aerodynamic drag of a nozzle chamber with ejector-type nozzles, the equation

$$\varDelta \mathrm{P} = 1.6 \cdot (\rho \cdot \upsilon)^{2,3}. \tag{12}$$

A graphical view of this relationship is shown in Figure 3.

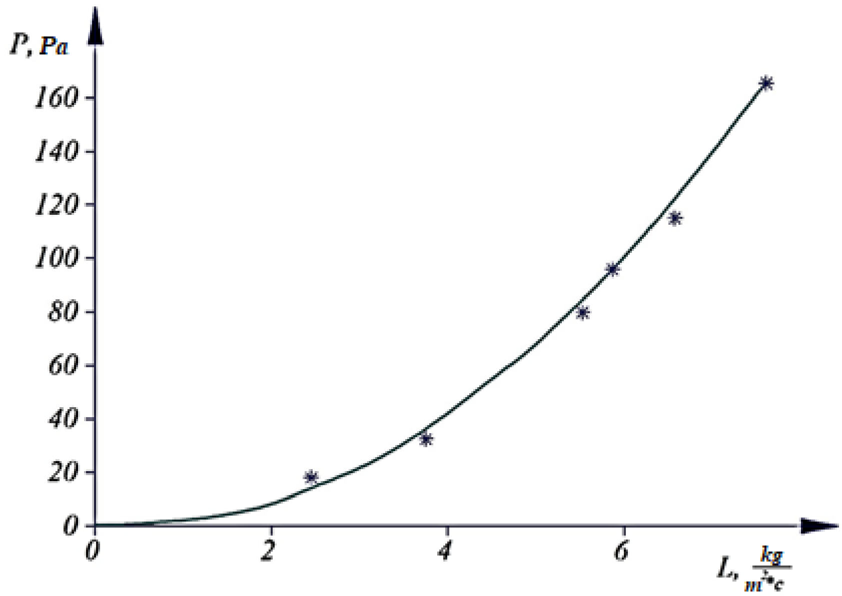

Figure 3. Dependence of aerodynamic drag of the nozzle chamber on the mass velocity of the air flow

In the conditions of hot (temperature t>35÷40 °C) and dry (relative humidity φ<15%) summers in Uzbekistan, these air parameters are far from comfortable, and, therefore, the use of microclimate normalization is necessary. Of particular importance is the cooling and humidification of air in working and residential premises, in passenger transport (buses and train cars). At the same time, air cooling by 6÷10 °C is optimal relative to the environment, bringing φ to 40÷60% [20-22].

4. Conclusions

Experiments have shown that for realistic measurements, this equation is true with a relative error of ± 6.7 percent. Nor does the aerodynamic resistance of the nozzle chamber of the spray chamber according to surpass 160 Pa.

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