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Modeling moisture condensation in humid air flow in the course of cooling and heat recovery

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ABSTRACT

This article, written as a result of theoretical and practical research, describes a survey of condensate nucleation in heat exchangers utilizing low-potential heat of humid air. The authors conducted a series of studies aimed at developing energy-independent technical solutions to protect heat exchangers from the freezing of the moisture condensing on their exchange surfaces in the course of humid air heat recovery. There was a negative dependence of the thickness of the layer of condensate formed on the heat exchange surface, the condensation rate, and the moisture content of warm air. The article presents the outcomes of the numerical study aimed at building a model of the condensation processes that occur due to a flow of moist air when the air is cooled and heat is utilized. The study provided experimental validation for the consistency of the processes related to water vapor condensation in the flow of cooled moist air. The outcomes presented in the article can apply to both the exchange systems using the low potential heat of the atmospheric air, e.g., for room heating or snow melting, and to the exhaust air energy recovery systems of the building ventilation plant.

1. Introduction

The widespread use of air cooled heat exchangers, recuperators, and other systems recovering waste heat of exhaust air from ventilation systems is impeded by the condensation and subsequent freezing of moisture on the surfaces of air cooled exchangers. Frost formation and icing of exchange surfaces exposed to moist air at subzero temperatures results not only in a sharp drop in the heat exchange efficiency but also in the need to use additional power to defrost the apparatuses [1,2]. The aforementioned problems are typical of air cooled systems virtually all over Russia [3].

There are two main ways to protect heat exchanger surfaces from icing: regular heating (defrosting), which means additional energy costs, and chemical protection of exchange surfaces by means of special anti-icing moisture-repellent compositions applied to the heat-exchange surface. The article [4] analyzes these two ways in detail, with emphasis on the chemical methods. The condensation of water vapor in air and on cold heat-exchange surfaces clearly has a strong influence on the rate of frost formation and air cooler icing. A reasoned choice of the geometric parameters of heat exchangers, cooling air flow rates, and conditions for heat exchange with cooling surfaces can significantly reduce the energy required to defrost heat exchangers [5,6]. Hence, creating heat regime models for air cooled exchangers that provide a reasonable picture of the heat exchange process accounting for air vapor condensation in the cooling air and on cold exchange surfaces is, at the moment, an important task. Coping with the task can help to create new, energy-independent solutions to protect exchange apparatuses from frost formation. Many studies have been performed to solve this task; the work described in are most similar to the problems analyzed in our article [7].

2. Modeling the processes of moist condensation in cooling air

The purpose of the studies described herein was to develop energy-independent technical solutions protecting heat exchangers from the freezing of the moisture condensing on their exchange

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surfaces in the course of moist air heat recovery/recuperation. Numerical analyses were used to model moist condensation processes in the air flow exposed to cooling. Calculations were made using a model simulating two air canals exchanging heat via the heat-conducting wall. Fig. 1 provides a general view of the air cooler in question.

To study the thickness of the condensate droplets nucleating on the heat exchange surface and the rate of nucleation, the authors solved the conjugate problem of heat conductivity and gas dynamics using the latest software platform [8] ANSYS CFX 11.0 based on the numerical solution of the Navier–Stokes equation system.

$$\begin{aligned}
&\rho\frac{\partial u}{\partial t}+\rho u\frac{\partial u}{\partial x}+\rho v\frac{\partial u}{\partial y}+\rho w\frac{\partial u}{\partial z}=-\frac{\partial p}{\partial x}+\mu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right]\\
&\rho\frac{\partial v}{\partial t}+\rho u\frac{\partial v}{\partial x}+\rho v\frac{\partial v}{\partial y}+\rho w\frac{\partial v}{\partial z}=-\frac{\partial p}{\partial y}+\mu\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right]\\
&\rho\frac{\partial w}{\partial t}+\rho u\frac{\partial w}{\partial x}+\rho v\frac{\partial w}{\partial y}+\rho w\frac{\partial w}{\partial z}=-\frac{\partial p}{\partial z}+\mu\left[\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right]
\end{aligned}\tag{1}$$

Furthermore, equations of continuity and of state were to be satisfied:

$$\frac{\partial p}{\partial t}+\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0\tag{2}$$

$$p=\rho RT\tag{3}$$

here u, v, w are the sought components of the air velocity vector (by axes x, y, z), p is the pressure, t is the time, μ the dynamic viscosity coefficient of air, ρ is the density, R is the universal gas constant, T is the temperature.

For the simplicity of modeling, we can assume that air is incompressible and isothermal; body forces are ignored.

Direct solution of Eq. (1) taking into account the eddies of all scales (DNS, Direct Numerical Simulation) can be practically implemented only for very low flow rates. Therefore, the authors used a semi-empirical method based on velocity splitting by the time averaged component and the pulse component $u_i(t)=\overline{u_i}+u_i'(t)$ with transition to the so-called Reynolds averaged Navier–Stokes method:

$$\frac{\partial}{\partial t}\left(\rho\overline{u_j}\right)+\frac{\partial}{\partial x_i}\left(\rho\overline{u_i}\cdot\overline{u_j}\right)=\frac{\overline{\partial p}}{\partial x_j}+\frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial\overline{u_i}}{\partial x_j}+\frac{\partial\overline{u_j}}{\partial x_i}\right)-\rho\overline{u_i'u_j'}\right],$$

$$\frac{\partial u_i}{\partial x_i}=0,\quad \frac{\partial u_i'}{\partial x_i}=0,\tag{4}$$

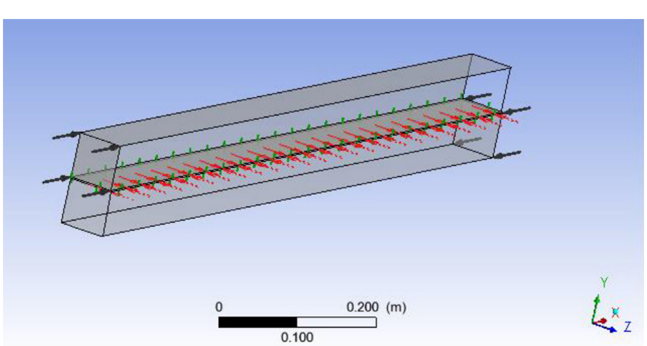

Fig. 1. Air cooler simulation model. Length: 1 m (along the *X* axis). Height of each of the areas of cool and cooling air (along the *Y* axis): 0.059 m. The area width was taken as 0.06 m (along the *Z* axis), and the symmetry condition was established on the border of this area. Sheet thickness is 0.002 m.

where $\bar{p}$ is the average pressure, and indices i = 1,2,3 and j = 1,2,3 correspond to the coordinates x, y, z. Shear (Reynolds) tensions $\rho\overline{u_i'u_j'}$ are six unknowns additional to the averages motion parameters $(\bar{u}_i, \bar{p})$ that are usually approximated using the Boussinesq hypothesis:

$$\rho\overline{u_i'u_j'}=-\mu_t\left(\frac{\partial\bar{u}_i}{\partial x_j}+\frac{\partial\bar{u}_j}{\partial x_i}\right)+\frac{2}{3}\rho k\delta_{ij},\tag{5}$$

where μ_t is the additional viscosity caused by pulsations; k is the averaged eddy pulsation energy (*TKE*). The system is open-loop, so it requires additional conventions ("eddy models").

2.1. First set of numerical experiments

In the first sequence of numerical experiments, the authors analyzed the thermal state of the heat-conducting wall made from different materials (copper and PVC) located between two air flows (cold, −20 °C, and cooling, +20 °C) [9]. The areas analyzed corresponded to the exchanger simulation model shown in Fig. 1. In this set of experiments, the length of the heat exchanger was 0.3 m along the x axis (in the direction of the cooling air flow) [10,11]. Three flow modes were compared (the oncoming flow rates were 2, 1, and 0.2 m/s), and the length of the square section device was 0.3 m. The Reynolds numbers for each of the three flow modes were, respectively: 13500, 6750, 1350.

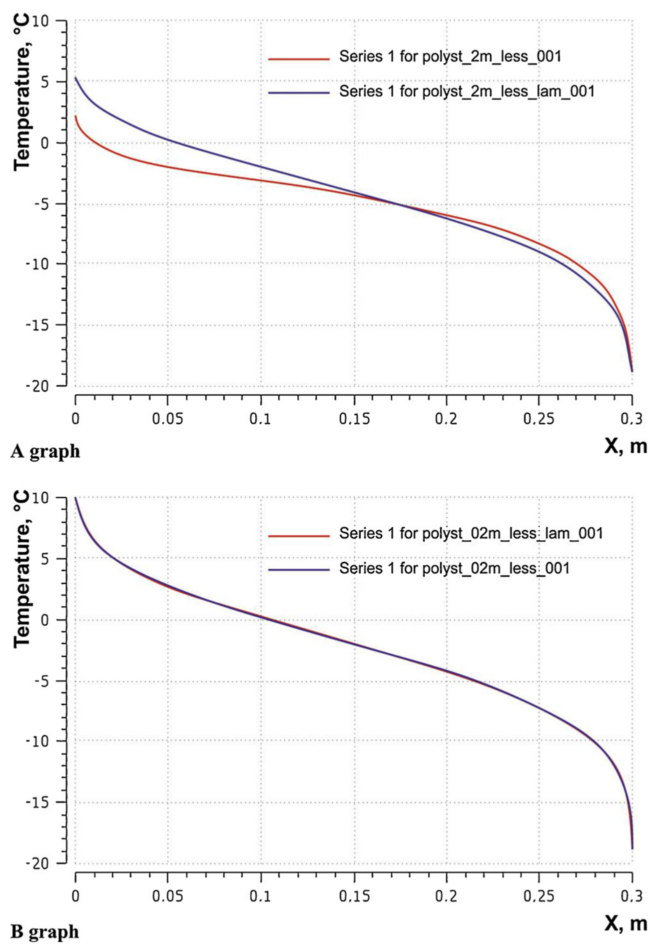

Fig. 2. Comparison of the laminar (red line) and turbulence (blue line) problem definitions. Average temperature of the heat-conducting PVC wall on the cold air side. A graph—turbulent flow mode, 2 m/s; B graph—laminar mode, 0.2 m/s. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)