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Influence of buoyancy forces in multi-storey buildings on the efficiency of a regenerative air handling unit with heat recovery

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Abstract. Various solutions are currently being offered to modernize the ventilation in old residential buildings. In small apartments, it is very appropriate to use decentralized ventilation units - for example with regenerative heat exchangers. Theoretical and experimental studies of such devices show a significant impact of the operational mode of regenerators (determined by the environment) on their efficiency. The aim of this article is to demonstrate the impact of the buoyancy forces in multi-storey buildings on the efficiency of a regenerative air handling unit for decentralized residential ventilation with heat recovery. Numerical solutions are presented for close to the real operating modes, airflow rates and air temperatures of the supply and the exit air, and the temperature ratio.

1. Introduction

Fixed regenerative heat exchangers are used in decentralized ventilation systems for heat recovery of the exhaust air, which is associated with the energy economy of the microclimate systems and the provision of thermal comfort and quality of the indoor air. Recently, local ventilation systems for individual rooms, 1-2 storey buildings, and small houses have started integrating ventilation units into the outside wall of the room, using regenerative heat exchange. Compared to industrial regenerative heat exchangers, which operate at almost constant parameters, regenerators in the field of HVAC operate at variable environmental parameters - temperature, humidity, and pressure difference (ΔP_b) in buildings, leading to different flow rates of hot and cold airflow through the regenerator. This ΔP_b between the indoor pressure and outdoor pressure in the multi-storey building is the pressure of stack, created naturally by air density differences, which drives a ventilation air rate in the building. Unfortunately, this stack effect in multi-story buildings influences the efficiency of the regenerators on the different levels of the dwelling and it is difficult to realize effective work of all regenerator's units. It was noticed in [1] that during the winter there are practically periods in which no heat recovery occurs, and this is confirmed by airflow temperature measurements, [2]. For this reason, it was necessary to study the different airflow into the matrix of regenerator at different floors of the dwelling induced by mechanical fans and natural ventilation in different operating conditions using Computational Fluid Dynamics (CFD) simulations.

The article presents a CFD model of air flows with the boussinesq approximation (i.e. created ΔP_b) in a multi-story building coupled with mechanical ventilation with the regenerator's fans. The computational results were used to investigate the ventilation airflow rates, the thermal and pressure field in the dwelling, and the efficiency of the regenerators. The results of our simulations show the mutual influence between the thermal and aerodynamic mode of the building ventilation and the thermal and aerodynamic mode of the regenerators.

2. Model statement

2.1. Geometry and mesh
The three-dimensional geometries of the matrix and the building are shown in Figure 1. The building has got 5 identical rooms on different levels connected to one common chimney vent with height H=14 m and a free-open square channel 0.2 m x 0.2 m. The main computational domain involves the volume of the rooms and the chimney connected by outlets with a diameter of 0.15 m. The matrixes are presented as single domains connected to the main computational one with inlet diameters 0.093m.

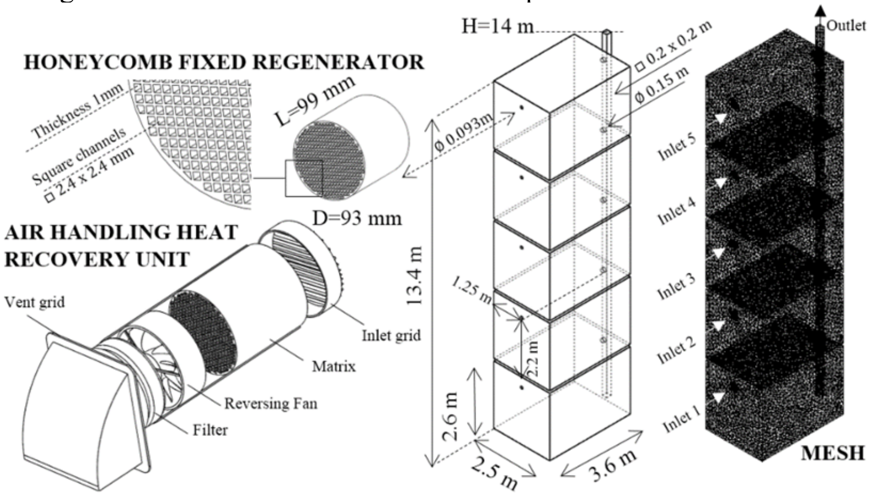

Figure 1. Model statement of the regenerator and the building.

2.2. Computational model and assumptions
The Finite-volume method and software Fluent, ANSYS, [3, 4] was used to solve the differential equations model. The velocity, pressure, and temperature fields of the fluid in the volumes are computed by the system of equations for mass, momentum, and energy conservation. The model describes a three-dimensional airflow induced by mechanical ventilation and buoyancy forces and a radiative underfloor source heating. The boundary conditions are set constant, and the process is steady-state, the heat fluxes of all wall surfaces of the building are (adiabatic), except the radiation heat flux from the floors to the apartments, and also the air change rate by infiltration is zero. The values of atmospheric parameters are constant and the wind around the building is ignored. The physical properties of airflow are fixed except the density whose deviation creates the forces of ΔP_b. The thermodynamic relation between air density and temperature is the ideal gas equation of state. The additional model for turbulent airflow is SST-k-ω coupled with a "P1 model" which includes the computational radiation equation, [5].

The air handling unit (AHU) is modeled as an individual domain with a constant pressure loss coefficient $f_{loss} = 20$, equation (1):

$$f_{loss} = \Delta P/(\rho v^2/2), \qquad (1)$$

where ΔP is the airflow pressure drop, Pa; ρ – the airflow density, kg/m^3; v – the velocity at the minimum flow area, m/s.

The AHU domain is connected to the main airflow computational domain as an interface between it and the cells simulating the fan. The fan is considered to be infinitely thin, but the pressure rise across it is presented as a function of the airflow velocity through it. The correlation between the fan pressure and airflow velocity is given as a polynomial function, equation (2). The swirl tangential and radial velocity of the stream is assumed zero.

$$\Delta P_{FAN} = 2.5126v^2 - 20.845v + 30.849 \tag{2}$$

The constant static pressure of zero Pascals was specified at all wall inlets and the outlet of the chimney. The numerical computations were performed at an external temperature of 275 K and an atmospheric pressure of 101 325 Pa. The sources of heat are placed on the floors of the rooms with a constant surface temperature of 300 K and the flow of heat towards the room is modeled by convection and radiation.

The graph in figure 2 shows a fan curve and three resistance curves of the ventilation system defining three duty-points of the fan - A, B and C. Duty-point B presents the modes of the fan in ideal temperature conditions when the temperature in the dwelling is equal to the external temperature and buoyancy forces are neglected. Duty-points A and C are present when the inside temperature is higher than the outside one, which leads to buoyancy forces and displacement of the resistance curve of the system. Point C is the duty-point of the fan when the airflow direction is towards the room - air supply direction. Point A is the duty-point of the fan when the airflow direction is to the outside - air exhaust direction.

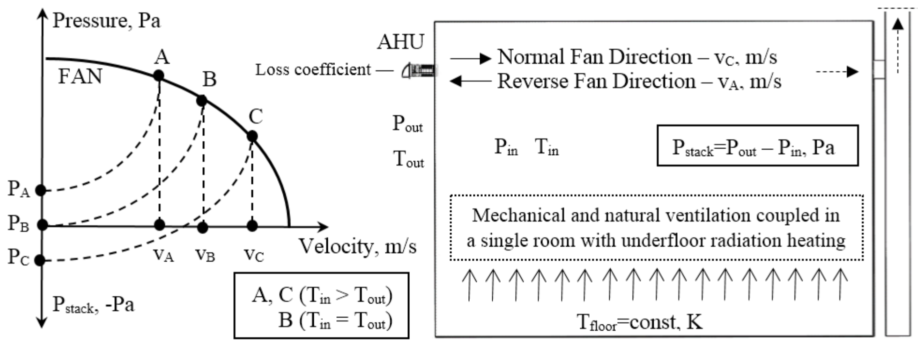

Figure 2. Problem statement of computational model

2.3. Thermal properties of air and honeycomb ceramic matrix

The average dry air thermo-physical properties at pressure p=101 325, Pa for the temperature range 275≤T, K≤300, and the properties of the matrix are given in Table 1.

Table 1. Thermal-physical properties of dry air and honeycomb ceramic matrix.

Fluid / Unit	ρ, kg/m^3	c_p, J/kgK	λ, W/mK	μ, Pa.s
Dry Air	ρ= p/RT	c_p=1004,2	λ=0,02607	μ=1,704e-5
Cordierite	ρ=1500	c_R=900	λ=1,20	–

3. Numerical results

3.1. Results of airflow distributions

The presented calculations describe separately temperature and aerodynamic state of the system for three operational modes of the AHU's fans - supply air direction, exhaust air direction, and not working fan. All states were calculated when the inside temperature is higher than the outside temperature. For each of the flow directions in these temperature conditions, pressure and velocity fields in the building are given. For reasons of comparison, the same distributions were calculated when the fans of the AHU were not working, and only buoyancy forces drove the airflow in the building. For the last case, the distribution of the inside temperature is presented too, to show the effect of natural ventilation in case of free opened inlets (AHUs without ventilation close valves).

The pressure distributions for the three AHU operating modes and the distribution of temperature in the building in the case of a non-working fan can be seen in figure 3. The figure shows the pressure fields in the building for each operational mode of AHU.

The results for the velocity distribution inside the building and the AHU are shown in figure 4. The airflow rates for each room under the respective conditions can also be seen in this figure.

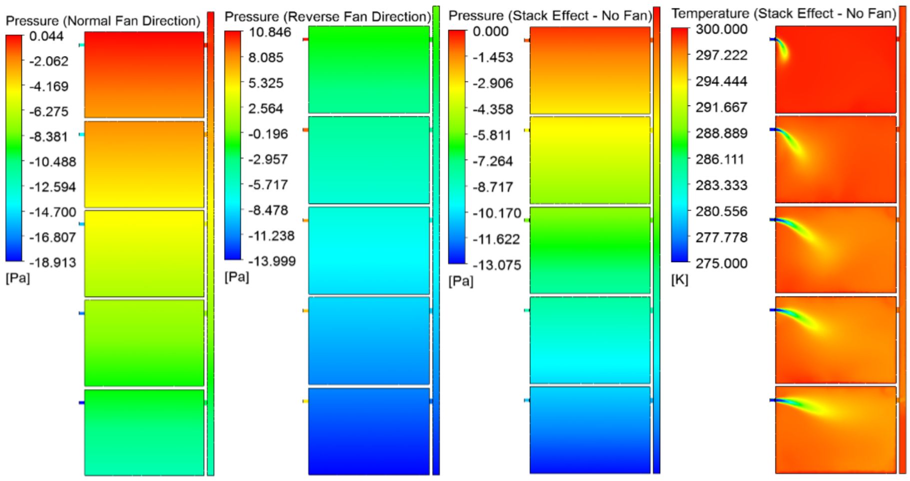

Figure 3. Pressure distributions at different modes of AHU.

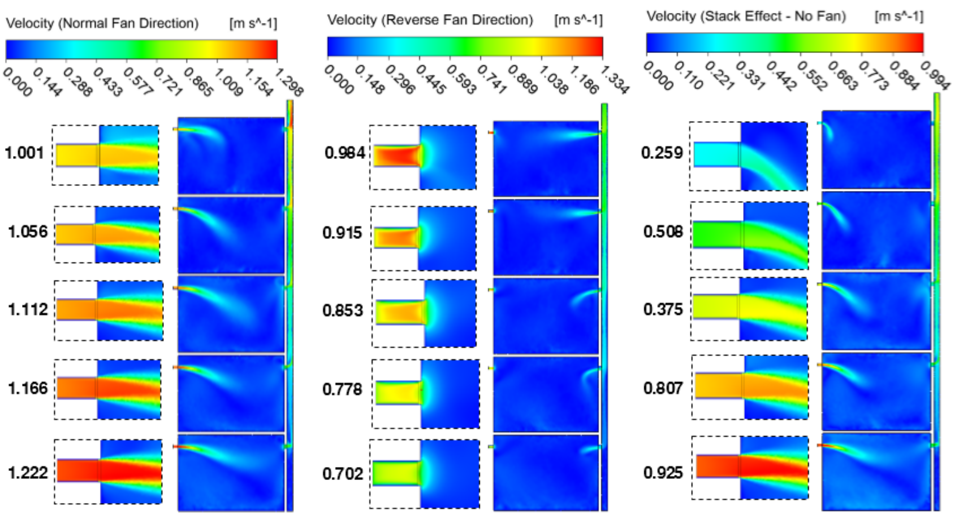

Figure 4. Velocity distributions at different modes of AHU and average inlet/outlet air velocity.

3.2. Results of the flow temperature in AHU

The change of the velocity in the regenerator matrix respectively the magnitude of the airflow rate affects the air change rate in the apartment and, also, it strongly affects the temperature ratio of the regenerator. The variations of the inlet and outlet temperatures of the regenerators on three different levels are presented in figure 5.

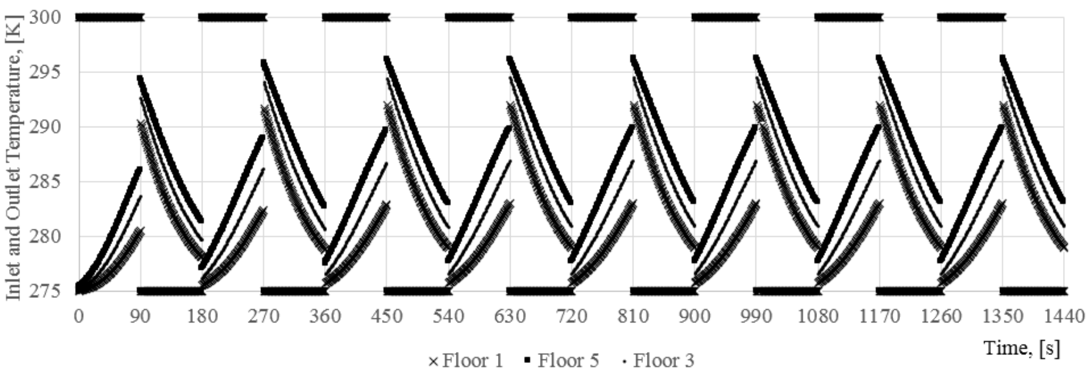

Figure 5. Outlet temperatures during the operating of the regenerator at different floors.

The curves in figure 5 were calculated using the model presented in, [6]. The graphs present the inlet and outlet air temperatures for a mode of the regenerators with half-periods of 90 seconds.

The regenerative air handling unit reaches the quasi-steady-state regime after 8 switches.

The values of the temperature ratios for both directions of airflow in AHU are presented in table 2.

Table 2. Temperature ratio during the hot and cold half-period, %.

Floor	$\Theta_{h,1}$	$\Theta_{c,1}$	$\Theta_{h,2}$	$\Theta_{c,2}$	$\Theta_{h,3}$	$\Theta_{c,3}$	$\Theta_{h,4}$	$\Theta_{c,4}$	$\Theta_{h,5}$	$\Theta_{c,5}$	$\Theta_{h,6}$	$\Theta_{c,6}$	$\Theta_{h,7}$	$\Theta_{c,7}$
1	91,98	31,61	86,69	36,15	85,26	37,14	84,92	37,36	84,85	37,42	84,83	37,43	84,82	37,43
2	86,00	38,58	81,28	41,78	80,14	42,49	79,88	42,65	79,82	42,69	79,81	42,70	79,80	42,70
3	86,00	40,32	77,50	46,11	75,30	47,36	74,80	47,65	74,68	47,71	74,66	47,72	74,65	47,73
4	86,00	42,24	74,22	50,15	71,05	51,86	70,32	52,24	70,16	52,33	70,12	52,35	70,11	52,36
5	80,47	48,41	68,92	55,26	65,99	56,74	65,33	57,07	65,18	57,14	65,14	57,16	65,13	57,16

The temperature ratios were calculated by equations (3) and (4):

$$\Theta_h = (T_{h,i} - T_{h,o}^*) / (T_{h,i} - T_{c,i}) \tag{3}$$

$$\Theta_c = (T_{c,o}^* - T_{c,i}) / (T_{h,i} - T_{c,i}), \tag{4}$$

where the indexes "i " and "o" are inlet and outlet. The indexes "h" and "c" are for hot and cold airflow; $T_{h,i}$, $T_{c,i}$ – the inlet temperatures of the hot and cold airflow, K; $T_{h,o}^*$, $T_{c,o}^*$ – the outlet average temperatures during the hot and cold half-periods, K.

4. Conclusion

The proposed CFD models can be used for numerical experiments to optimize the operation of the regenerator and to find suitable components integrated into the AHU according to the geometry and interior of the building. Unbalanced operating modes of regenerators, influenced by natural ventilation, lead to poor thermal efficiency, different air exchange rates on different floors of the building and cannot be ignored in multi-storey buildings. This leads to the need to select a new control system and components for this type of decentralized ventilation modules.

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