Development of an isothermal 2D zonal air volume model with impulse conservation

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SUMMARY

This paper presents a new approach to model air flows with a zonal model. The aim of zonal models is to perform quick simulations of the air distribution in rooms. Therefore an air volume is subdivided into several discrete zones, typically 10 to 100. The zones are connected with flow elements computing the amount of air exchanged between them. In terms of complexity and needed computational time zonal models are a compromise between CFD-calculations and the approximation of perfect mixing. In our approach the air flow velocity is used as property of the zones. Thus the distinction between normal zones and jet or plume influenced zones becomes obsolete. The model is implemented in the object oriented and equation based language Modelica. A drawback of the new formulation is that the calculated flow pattern depends on the discretization. Nevertheless, the results show that the new zonal model performs well and is a useful extension to existing models.

INTRODUCTION

Basics of zonal models

In the 1970s and 1980s zonal models were mainly based on experimental observation of air flow patterns [1]. Those air flow patterns were used as an input to the model. The aim was to calculate the temperature distribution in different zones. With the possibility to conduct CFD-simulations, this method has seen a revival in the late 1990s using the air flow calculation results as an input for the zonal model [2, 3].

When the air flow pattern is not known it can be predicted from pressure differences between the zones. In the zones, the mass balance is calculated. Furthermore, the balance of thermal energy, contaminant concentrations, etc. can be implemented here.

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

Two zone elements are linked by a flow element. In the flow element, the amount of air exchanged between two zones is calculated by the power law equation. This equation is derived from the Bernoulli equation:

$$\dot{m} = \pm C_d \cdot A \cdot \rho \cdot \sqrt{2 \cdot \left| \frac{\Delta p}{\rho} + g \cdot \Delta h \right|} \tag{2}$$

where C_d is the discharge coefficient, A the common surface of two adjacent zones, Δp and Δh are the pressure and height difference, ρ is the air density and g the gravitational constant. The direction of the air flow depends on the pressure and height difference. Teshome and Haghighat [4] state that a discharge coefficient of 0.83 is commonly used.

Problems with zonal models

The power law equation is valid only when a relatively small flow of air enters a large volume. In the zones, air is modelled not to move, i.e. the velocity is completely dissipated. In the case of rooms simulated by a zonal model this limitation of validity introduces large errors when considering driving air flows with high velocities. For example a jet is considered to be dissipated in the first zone it enters even though experience tells us that it will penetrate far into the room. To take into account driving flows Inard et al. [5] suggest to make a distinction between power law zones described by equation (2) and jet zones. In the jet zones an appropriate jet correlation describes the air flow. Inard et al. [6] extend this idea to thermal plumes.
Another solution is to adapt the discharge coefficients. As the calculated air flow gets higher in directions with lower discharge coefficients the desired air flow pattern can be achieved. Axley [7] suggests a surface drag model to calculate a discharge coefficient based on the shear stress of a portion of air volume. Teshome et al. [2, 4] suggest to use air flows obtained from CFD calculations to calibrate the discharge coefficients.
The only possibility to influence the predicted air flow pattern by adaption of the discharge coefficient is to change them locally, i.e. different flow paths are allowed to have different discharge coefficients. Changing the discharge coefficient globally, maintaining all flow paths equal, will not influence the air flow pattern when a mass flow source provides the air entering the room. Only the pressure differences between zones are affected. Therefore it is suggested to use the value of 1 instead of 0.83 for the discharge coefficient. As the predicted pressures might have unrealistic values, other air properties should not be calculated from them [8].

A numerical disadvantage of the power law equation (2) is that its derivative at zero becomes infinite [9]. Thus, simulations with frequent flow reversals or small pressure differences get slow or do not converge at all. One approach to solve this problem is to linearize the region around zero. However this introduces a discontinuity in the derivative possibly leading to oscillation around the solution.

To resume, previous work has shown three major drawbacks of zonal models. The present work aims to solve these tasks:

1) Failure when driving flows are present in a room
2) Necessity to calculate flow and air properties separately
3) Numerical instability

THE NEW ZONAL MODEL

In the new approach the air flow velocity is a property of a zone. Losses in the flow models are modeled using a coefficient similar to the discharge coefficient.

Instead of calculating an air flow due to pressure drop, the acceleration of the air flow is calculated. The acceleration of the portion of air (mass $\rho \cdot V$) is a result of the forces acting on the boundaries of a flow element. Figure 1 shows the forces acting on the vertical and the horizontal bounds of a flow element. The pressure, the impulse and the gravitational forces are taken into account. The pressure and the velocity information are provided by the linked zones.

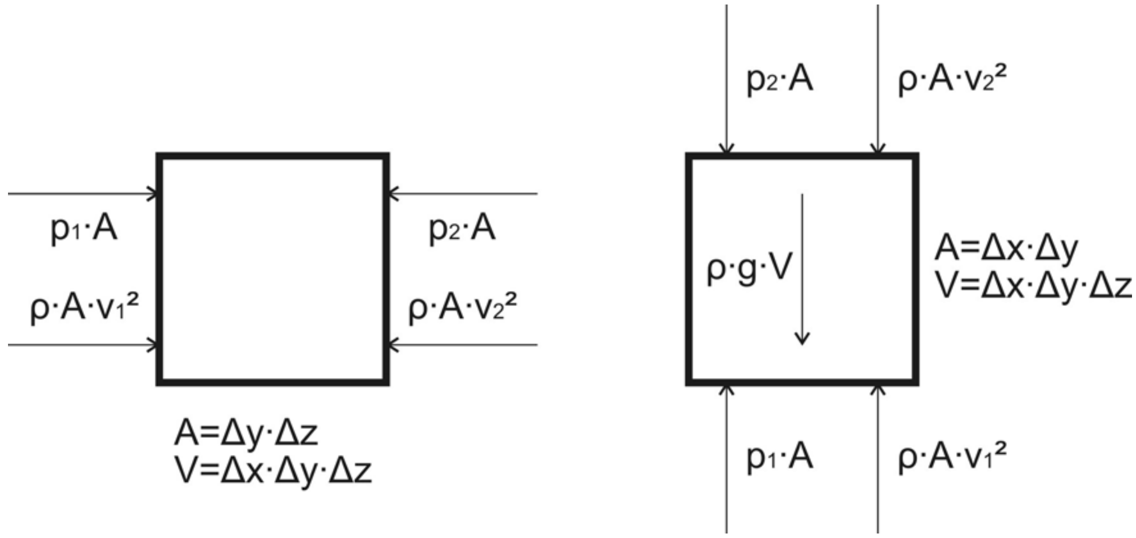


Figure 1. Forces acting on a flow element

When the acceleration is zero steady state has been reached and the sum of the forces is equal to zero. A loss term is introduced to take into account losses. These are supposed to increase with the air flow velocity.
From these considerations, equations (3) and (4) can be set up after division by $\rho \cdot V$:

$$\dot{v} = -\frac{\frac{\Delta p}{\rho} + \Delta\left(v^2\right) + g \cdot \Delta z}{distance} - f_{loss} \cdot sign(v) \cdot v^2 \qquad (3)$$

$$\dot{m} = \rho \cdot v \cdot A \qquad (4)$$

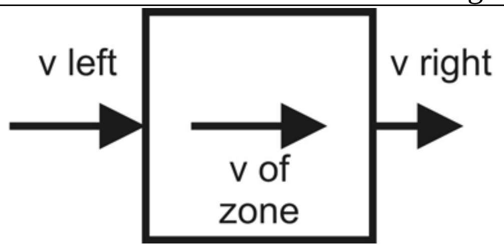
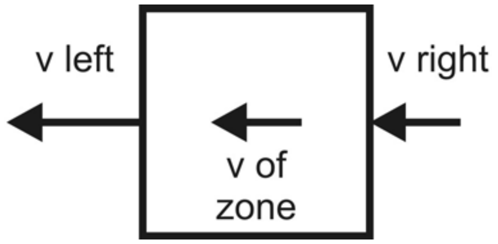
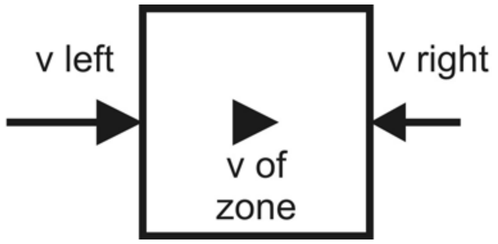
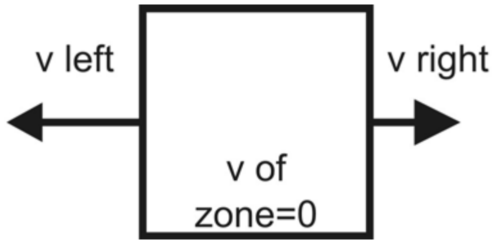
where *distance* is the distance between two zones, f_{loss} is the loss coefficient. The density used is the average density of the two connected zones.

Equations (3) and (4) are implemented in the flow elements to calculate the amount of air exchanged between two zones. They replace equation (2), used in previous formulations. The velocity is computed without the need to solve a square root function. The function is differentiable and flow reversals do not cause discontinuities. The factor f_{loss} describes losses. For an isothermal air flow in an empty room 0.01 applies well.

In the zone elements the properties of the air are computed. Based on the sum of entering and escaping air flows, the change of pressure and density is calculated. At steady state, the same amount of air enters and escapes a zone. For non-isothermal models, the heat balance can be implemented in the zones, too. Furthermore, the balance of other properties can be computed, for example moisture or pollutant concentrations.

The new approach shown in this research work uses the air flow velocity as a further property of a zone. The velocities are computed in horizontal and vertical directions. The admitted velocity depends on the direction of the air flows at the boundaries of a zone. Table 1 shows the four possible cases for the horizontal velocity. The vertical velocity is obtained analogously, exchanging left and right by down and up.

Table 1. Assignment of the velocity in the zones

	If the flow enters on the left and escapes on the right, the left velocity is assigned to the zone’s velocity.
	If the flow enters on the right and escapes on the left, the right velocity is assigned to the zone’s velocity.
	If the flows enter from left and right, the velocity is supposed to be the sum of both velocities. This assumption is needed to keep the continuity of the zone’s velocity with regard to changing flows on the left and right boundary.
	If both flows escape the zone, the velocity in the zone is supposed to be zero.

RESULTS

The validation case is Nielsen’s CFD-Benchmark test room [10]. The room (LxH = 9x3 m²) is built to provide a two-dimensional air flow pattern. Air is supplied by a horizontal slot located under the ceiling at a constant speed of 0.455 m/s. On the opposite side of the room, air is exhausted through a large horizontal slot. At a distance of 3 m and 6 m from the inlet the vertical velocity profiles are measured.

The model is set up in the Modelica-Language. In the *Modelica.Media*-Library, models for air properties and for other fluids are stored. For the isothermal zonal model a simple air model without moisture or pollutants valid from 273.15 to 373.15 K (0 to 100 °C) is used. For calculations in the isothermal case the air has a temperature of 293.15 K (20 °C).

The room is discretized into 6x6 zones. The length of each zone is 1.5 m. The lower zones have the height of the outlet, 0.48 m, the upper zones have the height of the inlet slot, 0.168 m, the other heights are equally distributed – see figure 2. The solution is obtained in about one second on a normal portable PC.

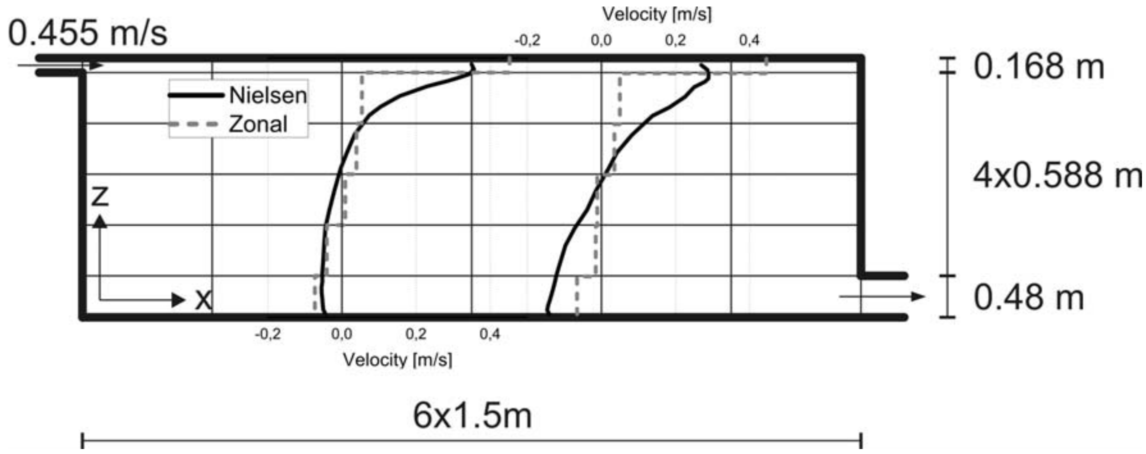


Figure 2. Air flow pattern

The air flow velocity in the upper zone is predicted too high at both measurement locations. At the position of 3 m the recirculation velocity is well predicted, at 6 m it is too low. The impulse conservation has the disadvantage that the inlet air flow velocity is transferred too far into the room.

Influence of the loss factor f_{loss}

To show the influence of the loss coefficient on the air flow pattern the same model as previously described is used with a loss coefficient of 0.1 instead of 0.01. Figure 3 shows that the velocities get lower and the recirculation only takes place in the lowest zone when the loss coefficient is increased. If the loss factor is further increased to 0.2 no recirculation occurs.

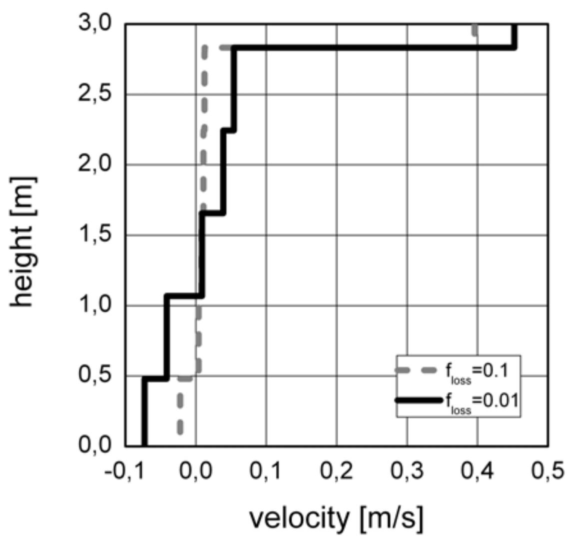


Figure 3. Comparison of loss factors, 3 m from inlet

Influence of the grid

To investigate the influence of the grid, different discretizations of the room are tested. Figure 4 (left) shows the result when the number of zones in z-direction is changed from 6 to 8. The lower and the upper zones maintain their heights (0.48 m respectively 0.168 m). The intermediate zones become smaller when refining the grid (from 0.588 m to 0.392 m). It can be seen, that changing the number of zones does not change the flow pattern fundamentally. Figure 4 (right) shows the influence of refining the grid in x-direction. Velocities tend to get higher with the higher discretization. In x-direction the number of zones is varied from 3 to 18. Best results are obtained using 6 to 12 zones in the x-direction. At lower and at higher discretizations, the recirculation is not well predicted.

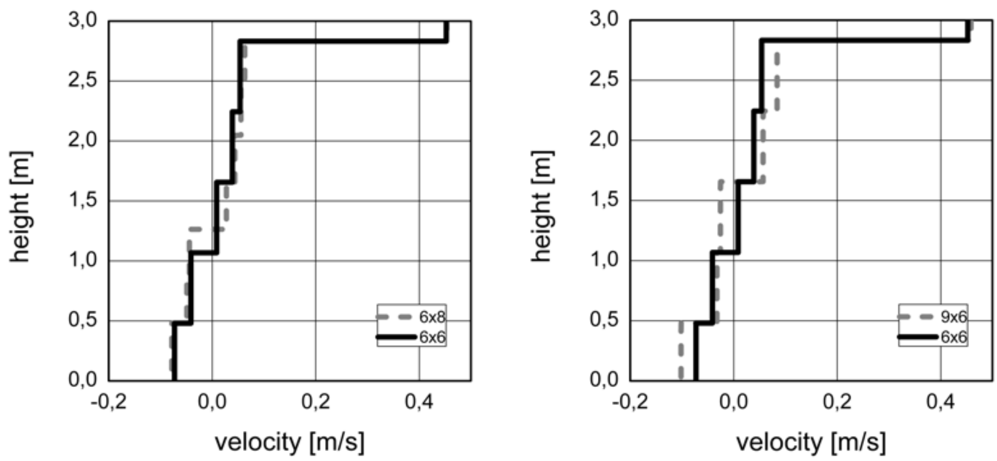


Figure 4. Influence of discretization, 3 m from inlet, f_{loss}=0.01

The impact of the discretization depends on the main flow direction. While the gridding in z-direction hardly affects the result, the gridding in x-direction can falsify it. The ideal size of the middle zones (supposing unity depth) is between 0.5 and 2 m^3.

DISCUSSION

The new zonal model formulation shows that the global air flow pattern can be predicted even though local deviations from measurements exist. While classical formulations dissipate the whole air flow, this formulation does not provide enough dissipation. One way of better taking dissipation into account could be to use the viscosity of air to model losses along a flow path. Another way could be to use flow laws taking into account the boundary layer that will exist in zones near a wall.
The results of the model are dependent on the grid chosen. A loss model taking into account the gridding would thus be useful. This aspect will be focused on in further research.
Even though the representation of the flows is not yet totally satisfying, the zonal model has solved three problems. A change of the loss coefficient f_{loss} has an impact on the flow pattern even though the same value is supposed in each flow path. The computation of unrealistic pressures in the zones is avoided and the computed state of a zone's air can be used in the model. Furthermore, the model is described by continually differentiable functions. Therefore, a solution can easily be found.

The next steps to take in the development of the zonal model are to extend it to non-isothermal conditions. Furthermore, a 3D-extension is desirable to be able to asses spatial distributions of heat sources in rooms.

ACKNOWLEDGEMENTS

This research is benefiting from the work that is being done in the European Community's Clean Sky JTI under grant agreement n° CSJU-GAM-ED-2008-001.

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