Leak Air in a Double-Wall Chimney System

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Abstract. Operating biomass stoves in modern buildings with tight shells often requires a room-independent air supply. One possibility to arrange this supply is to use a double-wall chimney with fresh air entering through the annular gap. For this setup, a mathematical model has been developed and checked with experimental data. It turned out that for commercially available chimneys, leakage is not negligible and inclusion of leak air in the calculation is crucial for reproduction of the experimental data. Even with inclusion of this effect, discrepancies remain which call for further investigations and a refinement of the model.

1. Introduction
Biomass is an important source of renewable energy, in particular used for heating purposes on small and large scale. Biomasse stoves have to be equipped with a chimney, through which the flue gases are led to the environment, as well as a combustion air supply. Traditionally the combustion air was directly taken from the room where the heating appliance was located.

During the last decades, building shells got much tighter, advancing in the direction of passive houses. Therefore building shells of modern houses are often too tight to enable enough air supply through leakage. Consequently, modern stoves are constructed room sealed with separate air supply in order to enable the operation of stoves in almost air-tight buildings.

One possibility to arrange air supply is by using a double wall chimney: The flue gas exits through the inner pipe, while fresh air enters through the annular gap. Due to preheating, the buoyant force of the air reduces the natural draft of the chimney; an effect which can be significantly reduced by proper insulation. This setup has been investigated in a research project conducted in cooperation with an industrial partner [1]. The project consisted both of measurements performed on different experimental setups and the creation of a mathematical model for the chimney which is outlined in sec. 2.

2. Outline of the Model
The model is quasi-static and effectively one-dimensional, making use of the wealth of heuristic relationships for heat and mass transfer established in engineering [2]. The cylindric chimney is discretized by introducing \( N_{\text{cyl}} \) slices, for which mass and energy balances are employed to calculate pressure and temperature. Some relevant physical quantities in a single cylinder slice are illustrated in fig. 1, where also the notation is fixed.

Convective heat transfer coefficients are determined from the Nusselt numbers, which depend on the properties of the gas flow, characterized by Reynolds and Prandtl number. Pressure in pipe and annular gap is determined to a large extent by buoyancy. In addition, dynamic pressure and friction have some influence, all taken into account in Bernoulli’s equation.
Some quantities used to describe a cylinder slice in a double-wall chimney: $T$ denotes temperature, $\dot{m}$ the mass flow, $\alpha$ the coefficient of convective or conductive heat transfer, $\varepsilon$ the coefficient of radiative emission and $\psi$ the dimensionless number characterizing wall friction. Subscripts $p$ denote the pipe section, $g$ the gap section, $a$ the ambient. Quantities defined on the lower/upper boundary of the slice carry an additional subscript $l/u$. The subscripts $wp$, $wpi$ and $wpo$ denote the pipe wall, its inner and its outer surface, $wg$, $wgi$ and $wgo$ indicate the gap wall in the same way. Mass flows entering/leaving the slice have a subscript $in/out$, mass flows between regions within the cylinder slice are written as $\dot{m}_{a2g}$ and $\dot{m}_{g2p}$.

Other important physical and geometrical quantities not explicitly shown in this figure are the area $A$ of the wall barrel, volume flows $\dot{V}$, gas density $\rho$, gas velocity $v$ and gas pressure $p$, characterized by the difference $\Delta p$ to the ambient pressure $p_a$.

2.1. Energy balance for heat transfer at the outside of the pipe wall

As an example, we set up the balance equation for the heat transfer for the outside of the pipe wall: The heat conducted through the pipe wall is partially transferred to the gas in the gap and partially converted to radiation which is emitted to the inner gap wall,

$$(\alpha A)_{wp} (T_{wpi} - T_{wpo}) = (\alpha A)_{wpo} \left( T_{wpo} - \frac{T_{gu} + T_{gl}}{2} \right) + \Sigma_{p2g} (T_{wpo} - T_{wgi}),$$

where we have employed a condensed notation for subscripts of products, $(\alpha A)_{wpo} = \alpha_{wpo}A_{wpo}$.

In addition we have performed a factorization of the radiation term and defined

$$\Sigma_{p2g} := \frac{\sigma_{SB} A_{wpo}}{\varepsilon_{wpo} A_{wgi} \left( \frac{1}{\varepsilon_{wgi}} - 1 \right)} \left( T_{wpo}^3 + T_{wpo}^2 T_{wgi} + T_{wpo} T_{wgi}^2 + T_{wgi}^3 \right),$$

with $\sigma_{SB}$ denoting the Stefan-Boltzmann constant. The prefactor is chosen to account also for reflection and re-absorption of radiation within the annular gap.

Here, as in the whole model, arithmetic means are used instead of more accurate expressions based on logarithmic temperature differences. This introduces a controllable systematic error which is small compared to other uncertainties. The benefit is that the equation is formally linear with respect to the unknown temperatures.
2.2. Inclusion of Leak Air
The walls are modelled as homogeneously porose; for the volume flows of leak air we set
\[\dot{m}_{a2g} = \sigma_g \rho_g A_{wg} \Delta p_g, \quad \dot{m}_{g2p} = \sigma_p \rho_p A_{wp} (\Delta p_p - \Delta p_g)\] (3)
with the (initially unknown) porosity coefficients \(\sigma_p\) and \(\sigma_g\). These mass flows change the
gas velocities, determined from the continuity equation which, for the (quasi-)stationary case, simplifies to \(\text{div} \, \mathbf{v} = 0\). Including leak air, one arrives at
\[v_{p,\text{out}} \rho_{p,\text{out}} = v_{p,\text{in}} \rho_{p,\text{in}} + \dot{m}_{g2p}, \quad v_{g,\text{out}} \rho_{g,\text{out}} = v_{g,\text{in}} \rho_{g,\text{in}} - \dot{m}_{g2p} + \dot{m}_{a2g}\] (4)
with gas densities calculated from the ideal gas equation.

2.3. Structure of the Complete Model
The resulting equation for the unknown temperatures \((T_{pu}, T_{wpi}, T_{wpo}, T_{gl}, T_{wgi}, T_{wgo})\) is
formally linear and can be solved by simple matrix inversion. However, some entries of the
coefficient matrix and of the inhomogenity depend directly or indirectly on the unknown
numbers. In addition, the equations for the cylinder slices are coupled, since for example
\(T_{pu}^{(k)} = T_{pl}^{(k+1)}\) and \(v_{p,\text{out}}^{(k)} = v_{p,\text{in}}^{(k+1)}\) with \(k\) denoting the index of the current cylinder slice.

Obtaining the final solution requires an iterative approach, during which also velocities,
pressure values, leak air mass flow and heat transfer coefficients are re-calculated several times.
Boundary values like \(T_a, p_a, T_{pl}^{(1)}\) and \(v_{p,\text{in}}^{(1)}\) are obtained from experiment.

3. Comparison to Experimental Data and Outlook
The model works reasonably well for some setups, if a sufficient amount of leak air is permitted,
as shown in figs. 2 and 3. For \(\sigma_p = \sigma_g = 0.0 \frac{m^3}{s \, m^2 \, Pa}\) the simulation yields flue gas temperatures
which are up to 100°C too high. Changing this to \(\sigma_p = 0.00015 \frac{m^3}{s \, m^2 \, Pa}\), which results in a volume
flow of \(V_{g2p} \approx 5 \frac{m^3}{h}\), significantly improves the agreement between simulation and experiment.
(Further parameter-tuning could be done to obtain an even better agreement.)

The remaining deviations can be mainly attributed to the fact that the quasi-stationary
simulation responds faster to changes than the real system, which possesses some thermal inertia.

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**Figure 2.** Flue gas temperature \(T_p(2.15 \text{ m})\) for double-wall chimney with insulated liner

**Figure 3.** Pipe pressure difference \(\Delta p_p(2.15 \text{ m})\) for double-wall chimney with insulated liner
In other setups, the discrepancy between simulation results and experimental data is unacceptably large. This happens in particular for setups with insulated case, as shown in figs. 4 and 5. For this setup also numerical problems (in particular instabilities) have been repeatedly encountered.

![Figure 4](image1.png)  ![Figure 5](image2.png)

**Figure 4.** Flue gas temperature $T_p(3.9\, \text{m})$ for double-wall chimney with insulated case

**Figure 5.** Air temperature $T_g(0.45\, \text{m})$ for double-wall chimney with insulated case

Also here, some deviations can be attributed to the quasi-stationary modelling and could be removed by setting up a fully dynamical simulation. A more serious issue is the yet unsolved problem of heat transfer coefficients for the asymmetrically heated annular gap [3]. For the present studies, the expressions for an annular gap with thermal insulation have been used as an approximation, but this introduces an uncontrollable systematic error, which is a serious source of uncertainty for the results of the model. Here, some progress on theoretical grounds is required to improve the model.

Other improvements may be necessary as well in order to obtain a model with predictive power, but already at the present stage it is clear that for commercially available chimneys leakage is not negligible and inclusion of leak air in the corresponding simulation is a crucial point.

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