European Door and Shutter Federation e.V

Calculation Model of Energy Losses of Buildings through Doors

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Hörmann KG

HÖRMANN

Abstract

The aim of this paper is the determination of a transient analytic model for the prediction of energy losses of the buildings through doors with enough level of simplicity to allow a systematic and standardised calculation for door classification.

The emphasis of the study is the ventilation effect when the door is open, also know as air infiltration, as it is the main and paradoxically less known door characteristic in building design.

Mequonic Engineering, S.L. has carried out this project driven by The European Door and Shutter Federation, e.V (E.D.S.F.) to allow a better understanding of the phenomena to contribute to the development of more sustainable doors and buildings.

The model has been validated by specific experiments carried out in collaboration with Hörmann KG.

Contents

1 Introduction **4**

2 Heating and Cooling Loads Principles **5**
2.1 Heat Transmission . . . 6
2.1.1 Heat Loss through door panel . . . 6
2.1.2 Heat loss through below-grade walls and floors . . . 7
2.1.3 Heat loss from at-grade walls slabs . . . 7
2.2 Air Leakage . . . 7
2.2.1 Wind Pressure Effect . . . 8
2.2.2 Stack Effect . . . 9
2.2.3 Total Heat Loss due to Air Leakage . . . 10
2.2.4 Sensible Heating Loss . . . 10
2.2.5 Latent Heating Loss . . . 10
2.3 Air Infiltration . . . 11
2.3.1 Wind Pressure Effect . . . 12
2.3.2 Bulk Density Flow . . . 12
2.3.3 Total Heat Loss due to Infiltration . . . 13
2.3.4 Sensible Heating Loss . . . 13
2.3.5 Latent Heating Loss . . . 14
2.4 Solar Radiation . . . 15
2.5 Long Wave Radiation . . . 16
2.6 Preliminary evaluation of the different effects . . . 16

3 Physical and Mathematical Model **19**
3.1 Bulk Density Flow . . . 19
3.1.1 Mass Flow between Stratified Zones . . . 19
3.1.2 Heat Flow between Stratified Zones . . . 23
3.1.3 Door Height Correction due to Boundary Effects . . . 25
3.2 Energy Losses Model . . . 26
3.2.1 Temperature Profile and Differential Equation . . . 26
3.2.2 Long Wave Radiation Effect . . . 27
3.2.3 Heat Transfer Coefficient . . . 29
3.2.4 Heat Flux . . . 31
3.2.5 Summary of Heat Transfer Coefficient $\mathbf{K}_{\text{loss}}$ and Heat Flux $\mathbf{Q}_{\text{heat}}$ . . . 31
3.2.6 Time $\mathbf{t_h}$ to reach a Reference Temperature $\mathbf{T_{ref}}$ . . . 33
3.2.7 Time $\mathbf{t_b}$ to reach the Equilibrium Temperature $\mathbf{T_{eq}}$ . . . 33
3.2.8 Mean Temperature $\mathbf{T_m}$ . . . 33
3.2.9 Energy Loss Computation $\mathbf{E}_{\text{loss}}$ . . . 34
3.3 Thermophysical Door Cycle Problem and Iterative Model Calculation with Heating . . . 35
3.3.1 1st Interval: Door Open, Heating Off . . . 36
3.3.2 2nd Interval: Door Open, Heating On . . . 38
3.3.3 3rd Interval: Door Closed, Heating On . . . 39
3.3.4 4th Interval: Door Closed, Heating Off . . . 40
3.3.5 5th Interval: Door Closed, Heating On . . . 41
3.3.6 Energy Contribution in time . . . 42
3.4 Thermophysical Door Cycle Problem and Iterative Model Calculation with Cooling . . . 43
3.4.1 1st Interval: Door Open, Cooling Off . . . 43
3.4.2 2nd Interval: Door Open, Cooling On . . . 45
3.4.3 3rd Interval: Door Closed, Cooling On . . . 46
3.4.4 4th Interval: Door Closed, Cooling Off . . . 47

3.4.5 5th Interval: Door Closed, Cooling On 48
3.4.6 Energy Contribution in time 49

4 Calculation example **50**
4.1 Reference data 50
4.1.1 Location and climate 50
4.1.2 Building and door 50
4.1.3 Building intended use 50
4.2 Results 51

5 Experiments **55**
5.1 Background 55
5.2 Test equipment 55
5.3 Test procedure and conditions 57
5.4 Test results 57
5.5 Model experimental adjustment 61

6 Conclusions **64**

7 Annex **65**
7.1 Bulk density flow velocity profile calculation 65
7.1.1 The Equations of Motion, and Various Approximations 65
7.1.2 Application of the Bernoulli Equation 67
7.1.3 Linear Temperature Gradients 67
7.1.4 Linear Temperature Gradients and Linear Pressure Gradients 68

1 Introduction

The use of a building is responsible for the majority of its CO_2 emissions over the course of its life cycle. Energy consumption, as a principal source of such emissions, is mainly associated with the consumption of air-conditioning and heating systems. The latter is determined by losses through the building elevations and openings, represented in large part by doors.

The contribution from doors is usually evaluated as a part of closed fenestration systems, but their behaviour is very different because, unlike the latter, doors are continually being opened and closed depending on their use, and this means a much higher impact on the energy consumption of a building that is widely recognised.

The energy losses through doors are mainly related to the air flow through the door hole when it is open. There has been some approaches to this fact, but up to now there is not a systematic methodology to calculate this energy flow in a practical way with enough precision to evaluate door performance.

There are quite a few algorithms which are used in computer models to describe air flow through large internal and external openings. However, the algorithms are mainly based on simplifying assumptions and the practical range of validity of these models is not well known. These simplifications are not justified in a lot of possible real situations, and it would be necessary to evaluate the error assumed, indicating which situations can be described with confidence by these models.

The key phenomena is the air flow through large external openings. Its description is basically the same as for internal openings as long as there is no wind. Therefore, the research on large external openings was mainly concentrated on wind effects.

Flows induced by fluctuating wind pressures and eddy circulation have been observed in single-sided ventilation. However, it is difficult to characterise fluctuating pressures at the position of the door when the wind characteristics are only known far from the building. An additional challenge is how to relate these fluctuating wind pressures to the total air exchange through the single large opening, and how to deal with cases where internal doors are open.

The ventilation through a single opening is the combined effect of wind and buoyancy, but the available data show a large spread in values. Cross-ventilation is of considerable importance in warm climates but very few data are available. The complexity of the problem in general can only be faced with numerical calculation approaches.

All these wind related topics, in particular single-sided ventilation, need further measurement and modelling efforts to provide better understanding.

The scope of our study is not to develop a model that can describe the phenomenon in all its complexity, but to have a comprehensive simplified model allowing to understand the main factors involved and its relative weight. In this way, we help the door industry to have a better understanding of their products and its use in terms of energy efficiency and sustainability.

In practise, the final goal of this study is the determination of a transient analytic model to be able to predict the energy losses through a door with enough level of simplicity to allow a systematic and standardised calculation.

To have a better understanding of the global problem, first we will analyse the total energy exchange in the door and then we will analyse the air flow problem in detail. After we will describe the global calculation model and a numerical example, finally explaining the experimental tests made supporting the air flow model.

2 Heating and Cooling Loads Principles

There are different types of energy losses in buildings with respect a door. These are the most important ones:

1. **Heat Transmission**: Heat loss due to **Heat Conduction and Convection** when the door is closed.
2. **Air leakage**: Heat loss due to **Leakages** when the door is closed.
3. **Air infiltration or Ventilation**: Heat loss due to massive **Air Flow**.when the door is open.
4. **Radiation**: Heat generation due to natural **Solar Radiation** or **Long Wave Radiation**.

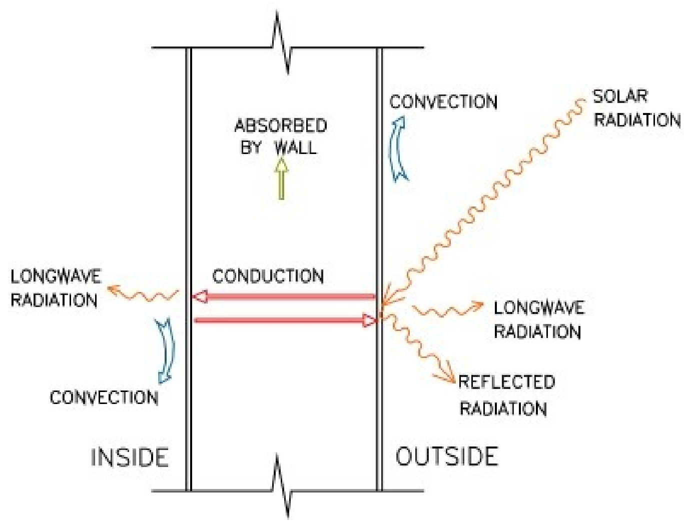

Figure 1: Heat Losses Effects Diagram

As an additional factor, it is the **electrical power** in automatic doors. There is a minor heat generation and energy consumption, but as it is not related to climate and heating charges, we consider it out of the scope of this study.

There are different door typologies depending on the application and how the door panel is moving to open the door hole. Except on pedestrian revolving doors, which remain out of the scope of this study, all types have a vertical plane surface when the door is closed that moves horizontally or vertically to leave a clear door opening.

The door panel or leave moves with a certain vertical or horizontal variable speed, having a transition situation between open and closed stages. Evaluations made in the past show that in general the influence of this transition period is very limited supporting the simplification made that the door is instantly open or closed. This implies the hypothesis that some effects only act when the door is open and other when the door is closed.

Below we explain the main basic equations that govern each of the phenomena to make a first evaluation of their importance:

2.1 Heat Transmission

When a door is closed, there is a heat flow that passes through the door. This is defined as the product of its area and the temperature difference between the inside and outside and by what is known as the thermal transmittance of the door or coefficient of heat transmission U.

The thermal transmittance U gives an indication of the level of thermal insulation of the door when closed.

There is a significant heat transfer through transmission at these regions:

1. Heat transfer through **door panel**
2. Heat transfer between **door and ground** (below-grade walls and floor)
3. Heat transfer between **door and walls** (at-grade walls slabs)

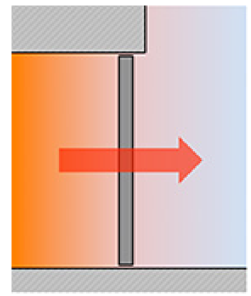

Figure 2: Heat Transmission Losses

2.1.1 Heat Loss through door panel

The heat transferred through walls, ceiling, roof, window glass, floors and doors is all sensible heat transfer, referred to as **transmission heat loss** and computed from:

$$\mathbf{Q_{trans} = UA(T_{in} - T_{out})} \tag{2.1.1}$$

where,

Q_{trans} : Instantaneous Energy Flow due to Transmission, $[W]$

U : Overall thermal transmittance or U-factor, $\left[\frac{W}{m^2K}\right]$

A : Surface area, normal to heat flow, $\left[m^2\right]$

T_{in} : Inside Design Temperature, $[^oC \text{ or } K]$

T_{out} : Outside Design Temperature, $[^oC \text{ or } K]$

The thermal transmittance is calculated by:

$$\frac{1}{U} = \frac{1}{\alpha_{in}} + \frac{1}{k} + \frac{1}{\alpha_{out}} \tag{2.1.2}$$

where,

α_{in} : Convection coefficient of walls inner surface, $\left[\frac{W}{m^2K}\right]$

α_{out} : Convection coefficient of walls outer surface, $\left[\frac{W}{m^2K}\right]$

k : Thermal conductivity of the walls, $\left[\frac{W}{m^2K}\right]$

2.1.2 Heat loss through below-grade walls and floors

$$\mathbf{Q_{trans} = U_{avg}A(T_{in} - T_{gr})} \tag{2.1.3}$$

where,

Q_{trans} : Instantaneous Energy Flow, $[W]$

U_{avg} : Average U-factor for below-grade surface $\left[\frac{W}{m^2K}\right]$

A : Surface area, normal to heat flow, $[m^2]$

T_{in} : Inside Design Temperature, $[^oC]$

T_{gr} : Ground Surface Temperature, $[^oC]$

2.1.3 Heat loss from at-grade walls slabs

$$\mathbf{Q_{trans} = PF_p(T_{in} - T_{walls})} \tag{2.1.4}$$

where,

Q_{trans} : Instantaneous Energy Flow, $[W]$

P : Perimeter (exposed edge) of floor, $[m]$

F_p : Heat loss coefficient meter of perimeter, $\left[\frac{W}{mK}\right]$

T_{in} : Inside Design Temperature, $[^oC]$

T_{walls} : Walls Surface Temperature, $[^oC]$

In general, heat loss through the door panel is of a larger order of the transmission through ground and walls, so it is usually the only term considered in simplified calculations.

2.2 Air Leakage

There is a heat flow associated with the exchange of air mass between the inside and outside when the door is closed. It depends on the characteristics of the air (specific heat, density), the temperature difference between the outside and inside, the area of the door and what is known as air permeability.

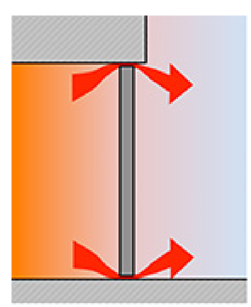

Figure 3: Air Leakage Losses

The coefficient of air permeability L gives us an indication of the level of the leak-tightness of the door when it is closed.

L depends on the pressure difference between the inside and outside which results from two effects:

- Wind
- Stack effect in the interior of the building.

The stack or chimney effect is based on the fact that there are existing air leakages in the upper building at least of the same order of the leakages in the door.

So the Air leakage Airflow losses are then composed by:

$$\mathbf{\dot{V}_{perm}} = \mathbf{\dot{V}_W} + \mathbf{\dot{V}_S} \tag{2.2.1}$$

where,

$\dot{V}_{perm}$: Air leakage Airflow due to All Effects, $\left[\frac{m^3}{s}\right]$

$\dot{V}_W$: Air leakage Airflow due to external Wind Pressure, $\left[\frac{m^3}{s}\right]$

$\dot{V}_S$: Air leakage Airflow due to Stack Effect, in the building, $\left[\frac{m^3}{s}\right]$

2.2.1 Wind Pressure Effect

$$\mathbf{\dot{V}_W} = \frac{\mathbf{L_W} * \mathbf{A}}{\mathbf{3600}} \tag{2.2.2}$$

where,

$\dot{V}_W$: Airflow leakage losses due to external Wind Pressure, $\left[\frac{m^3}{s}\right]$

L_W : Permeability due to Wind Pressure, $\left[\frac{m^3}{hm^2}\right]$

A : Surface area, $\left[m^2\right]$

L_W is calculated with:

$$\mathbf{L_W} = \mathbf{L_R}\left(\frac{\mathbf{P_W}}{\mathbf{P_R}}\right)^{\frac{2}{3}} \tag{2.2.3}$$

where,

L_W : Permeability due to Wind Pressure $\left[\frac{m^3}{hm^2}\right]$

P_W : Wind Manometric Pressure $[Pa]$

P_R : Reference Manometric Pressure, $[Pa]$

L_R : Reference Permeability, $\left[\frac{m^3}{hm^2}\right]$

Reference permeability L_R is obtained by tests at the reference manometric pressure, as specified in the following standards:

- EN 12426, Industrial and Garage Doors: $P_R = 50\ Pa$
- EN 12427, Pedestrian Doors: $P_R = 100\ Pa$

2.2.2 Stack Effect

$$\mathbf{\dot{V}_S = \frac{L_S * A}{3600}} \tag{2.2.4}$$

where,

$\dot{V}_S$: Airflow leakage losses due to Stack Effect, $\left[\frac{m^3}{s}\right]$

L_S : Permeability due to Stack Effect, $\left[\frac{m^3}{hm^2}\right]$

A : Surface area, $[m^2]$

Like wind pressure effect:

$$\mathbf{L_S = L_R \left(\frac{P_S}{P_R}\right)^{\frac{2}{3}}} \tag{2.2.5}$$

where,

L_S : Permeability due to Stack Effect, $\left[\frac{m^3}{hm^2}\right]$

P_S : Stack Effect Manometric Pressure, $[Pa]$

P_R : Reference Manometric Pressure, $[Pa]$

L_R : Reference Permeability, $\left[\frac{m^3}{hm^2}\right]$

The P_S or Stack Manometric Pressure, is the one induced by the Stack Effect, which can be computed as:

$$\mathbf{P_S} = \rho \left(\mathbf{\frac{T_{in} - T_{out}}{T_{out}}}\right) \mathbf{g} \left(\mathbf{H_b - H_{np}}\right) \tag{2.2.6}$$

where,

P_S : Stack Effect Manometric Pressure $[Pa]$

ρ : Density of air $\left(\rho \approx 1,293 \frac{kg}{m^3}\right)$

H_b : Height of the building, of the room or highest window, $[m]$

H_{np} : Height of the Neutral Pressure Plane, $[m]$

T_{in} : Average Inside Temperature Temperature , $[K]$

T_{out} : Exterior Temperature , $[K]$

g : Gravitational acceleration $\left(g \approx 9,81 \frac{m}{s^2}\right)$

H_{np} an be usually calculated as half of the Height of the building:

$$H_{np} = \frac{H_b}{2} \tag{2.2.7}$$

2.2.3 Total Heat Loss due to Air Leakage

Now, knowing the Wind flow due to Air Leakage, we can proceed to compute the Heat Loss. It will be the sum of two components:

1. Sensible Heat Loss Flow, due to the exchange of air of different temperatures.
2. Latent Heat Loss Flow, due to the exchange of the humidity of the air.

In practice:

$$\mathbf{Q_{perm} = Q_L + Q_S} \tag{2.2.8}$$

where,

Q_{perm} : Air Leakage Heat Loss, $[W]$
Q_L : Latent Heat Loss, $[W]$
Q_S : Sensible Heat Loss, $[W]$

2.2.4 Sensible Heating Loss

$$\mathbf{Q_S = c_{p_{air}} \rho \dot{V}_{perm} (T_{in} - T_{out})} \tag{2.2.9}$$

where,

Q_S : Sensible Heating Loss, $[W]$
c_{pair} : Specific Heat of dry air, $\left(c_{p_{air}} \approx 1,0049 \frac{kJ}{kgK}\right)$
ρ : Density of air $\left(\rho \approx 1.293 \frac{kg}{m^3}\right)$
$\dot{V}_{perm}$: Airflow, $\left(\frac{m^3}{h}\right)$
T_{in} : Indoor air temperature, $[^oC]$)
T_{out} : Outdoor air temperature, $[^oC]$)

2.2.5 Latent Heating Loss

$$\mathbf{Q_L = L_V^{H_2O} \rho \dot{V}_{perm} \cdot (\chi_{in} - \chi_{out})} \tag{2.2.10}$$

where,

Q_L : Latent Heating Loss, $[W]$

$L_v^{H_2O}$: Latent Heat of Evaporation at indoor air temperature $\left(L_v^{H_2O} \approx 2257 \frac{kJ}{kg}\right)$

ρ : Density of air $\left(\rho \approx 1.293 \frac{kg}{m^3}\right)$

$\dot{V}_{perm}$: Airflow, $\left(\frac{m^3}{h}\right)$

χ_{in} : Humidity ratio of indoor air, $\left[\frac{kg_{\text{ water}}}{kg_{\text{ dry air}}}\right]$

χ_{in} : Humidity ratio of indoor air, $\left[\frac{kg_{\text{ water}}}{kg_{\text{ dry air}}}\right]$

2.3 Air Infiltration

When the door is open, there is an exchange of air mass of a far greater order than arises due to permeability, although from the physical point of view this is a similar phenomenon.

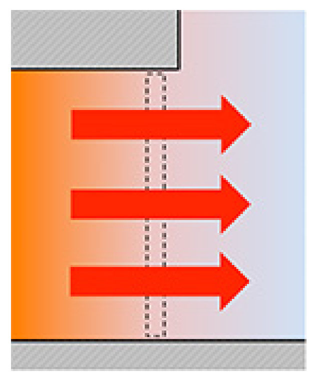

Figure 4: Heat Air Infiltration Losses

For a certain door area, the total amount of energy loss will be determined by how long the door is open. For a given number of cycles it is the speed of the door which determines the total amount of heat loss due to this effect.

During the time that the door is open, the thermal transmittance and air permeability cease to be relevant, as the door panel is removed from the door hole.

As seen with air leakage, air flow exist due to the pressure difference between inside and outside the building, mainly by two effects:

- Wind pressure
- Bulk density flow

When we have a large opening between fluids at different temperature, even in absence of wind, there is a so-called gravitational air flow or bulk density flow due to the density differences.

When the door is open, the stack effect is not considered with the hypothesis that there are not such large openings at the top of the building.

2.3.1 Wind Pressure Effect

The infiltration air quantity due to the wind pressure effect estimation is computed as:

$$\mathbf{\dot{V}_W = C_d \cdot A \cdot v_w = C_d \cdot A \cdot \sqrt{\frac{2\Delta P}{\rho}}} \tag{2.3.1}$$

where,

$\dot{V}_W$: Wind Pressure Air flow , $\left[\frac{m^3}{s}\right]$
A : Area $\left[m^2\right]$
C_d : Discharge Coefficient, usually between $0.5 - 1.0$ [Adimensional]
v_w : Wind Velocity , $\left[\frac{m}{s}\right]$
ΔP : Pressure Diference , $[Pa]$
ρ : Density of air, $\left[\frac{kg}{m^3}\right]$

And C_d can be calculated as:

$$\mathbf{C_d = \frac{1}{\sqrt{1.75 + 0.7 \cdot \exp\left(-\frac{W}{32.5 \cdot H}\right)}}} \tag{2.3.2}$$

where,

C_d : Discharge Coefficient
W : Length of the Aperture , $[m]$
H : Height of the Aperture , $[m]$

2.3.2 Bulk Density Flow

Also know as gravitational flow or buoyancy, this is a more complex phenomena and the equations of this effect will be thoroughly developed in chapter 3.1.

The air velocity distribution is not uniform through the hole section, but we can speak about a net air volume exchange of $\dot{V}_G$.

Taking into account these three effects, the total Air flow entering trough the open door is:

$$\mathbf{\dot{V}_{infil} = \dot{V}_G + \dot{V}_W} \tag{2.3.3}$$

where,

$$\dot{V}_{infil}: \text{ Infiltration Air Flow, } \left[\frac{m^3}{s}\right]$$
$$\dot{V}_{G}: \text{ Air Flow due to the Gravitational Effect, } \left[\frac{m^3}{s}\right]$$
$$\dot{V}_{W}: \text{ Air flow due to the Wind Pressure Effect, } \left[\frac{m^3}{s}\right]$$

2.3.3 Total Heat Loss due to Infiltration

In an analogue way to the air leakage losses, total heat due to both temperature and moisture can be expressed as:

$$\mathbf{Q_{infil}} = \mathbf{Q_L} + \mathbf{Q_S} = \dot{\mathbf{V}}_{\mathbf{infil}} \cdot \rho \cdot (\mathbf{h_{in}} - \mathbf{h_{out}}) \tag{2.3.4}$$

where,

$$Q_{infil}: \text{ Infiltration Heat Loss, } (W)$$
$$Q_L: \text{ Latent Heat Loss, } (W)$$
$$Q_S: \text{ Sensible Heat Loss, } (W)$$
$$\dot{V}_{infil}: \text{ Airflow, } \left(\frac{m^3}{h}\right)$$
$$\rho: \text{ Density of air } \left(\rho \approx 1.293\frac{kg}{m^3}\right)$$
$$h: \text{Interior Enthalpy, } \left(\frac{J}{kg}\right)$$
$$h_o: \text{Exterior Enthalpy, } \left(\frac{J}{kg}\right)$$

Sensible heating and latent heating are also calculated with similar expressions:

2.3.4 Sensible Heating Loss

$$\mathbf{Q_S} = \mathbf{c_{p_{air}}} \rho \dot{\mathbf{V}}_{\mathbf{infil}} \cdot (\mathbf{T_{in}} - \mathbf{T_{out}}) \tag{2.3.5}$$

where,

Q_S : Sensible Heating Loss, $[W]$

c_{pair} : Specific Heat of dry air, $\left(c_{p_{air}} \approx 1,0049 \dfrac{kJ}{kgK}\right)$

ρ : Density of air $\left(\rho \approx 1.293 \dfrac{kg}{m^3}\right)$

$\dot{V}_{infil}$: Airflow, $\left(\dfrac{m^3}{h}\right)$

T_{in} : Indoor air temperature, $[^oC])$

T_{out} : Outdoor air temperature, $[^oC])$

2.3.5 Latent Heating Loss

$$\mathbf{Q_L} = L_{\mathbf{V}}^{\mathbf{H_2O}} \rho \dot{\mathbf{V}}_{\mathbf{infil}} \cdot (\chi_{\mathbf{in}} - \chi_{\mathbf{out}}) \tag{2.3.6}$$

where,

Q_L : Latent Heating Loss, $[W]$

$L_v^{H_2O}$: Latent Heat of Evaporation at indoor air temperature $\left(L_v^{H_2O} \approx 2257 \dfrac{kJ}{kg}\right)$

ρ : Density of air $\left(\rho \approx 1.293 \dfrac{kg}{m^3}\right)$

$\dot{V}_{infil}$: Airflow, $\left(\dfrac{m^3}{h}\right)$

χ_{in} : Humidity ratio of indoor air, $\left[\dfrac{kg_{\text{ water}}}{kg_{\text{ dry air}}}\right]$

χ_{in} : Humidity ratio of indoor air, $\left[\dfrac{kg_{\text{ water}}}{kg_{\text{ dry air}}}\right]$

We also can calculate both heat losses due to infiltration and air leakage adding wind flows due to both effects and computing this addition as the total Heat Loss due to Wind Flow.

2.4 Solar Radiation

When the leaves or panels of the doors are made wholly or partly of glass or other transparent materials, solar radiation directly contributes to the heat inside the building. This is an effect which is generally favourable in winter and unfavourable in summer depending on the latitude.

The value of the flow of heat by radiation depends on the geographic latitude, the orientation of the elevation, the presence of shadows due to neighbouring buildings, the existence of parasols or solar protection elements, etc.

Since access doors are at levels close to ground, the influence of the solar factor is lower than for windows, and in general is of a lower order than other factors, even more so when looking at the effect over the whole year. For this reason, usually the solar factor is not considered in energy classifications of doors.

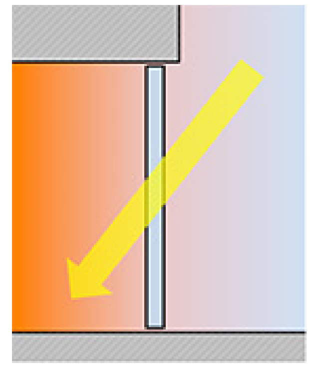

Figure 5: Heat Solar Radiation Losses and Generation

To be able to quantify the effect, we make an approximation to the basic equations that govern the phenomena:

$$\mathbf{Q_{solar} = I \cdot SHGC \cdot A_{pf}} \tag{2.4.1}$$

where,

Q_{solar} : Instantaneous Radiation Energy Flow, $[W]$

I : Solar irradiance $\left[\frac{W}{m^2}\right]$

$SHGC$: Solar Heat Gain Factor, $[adimensional]$

A_{pf} : Total projected area of fenestration, $\left[m^2\right]$

SHGC is the fraction of incident solar radiation admitted through the door, both directly transmitted and absorbed and subsequently released inward, and it can be between 0 and 1. It takes into account the glazing systems properties of reflection, transmission and absorption, as well as shading conditions. It is also known as Solar Factor.

The solar irradiance, which is the radiation power received by a surface, depends on factors like the latitude and the orientation of the door, and it has diffuse and direct radiation components.

The projected area will be:

$$A_{\text{ef}} = A_{glaze} \cos\phi \tag{2.4.2}$$

Where A_{glaze} is the area of the transparent section of the door and ϕ is the incident angle respect the normal of the Area.

2.5 Long Wave Radiation

Long wave radiation is the thermal electromagnetic radiation within or surrounding a body in thermodynamic equilibrium with its environment, or emitted by a black body (an opaque and non-reflective body). It has a specific spectrum and intensity that depends only on the temperature of the body, which is assumed for the sake of calculations and theory to be uniform and constant.

The thermal radiation spontaneously emitted by objects in the form of infrared light can be approximated as black-body radiation.

The net power radiated by doors or walls is the difference between the power emitted and the power absorbed:

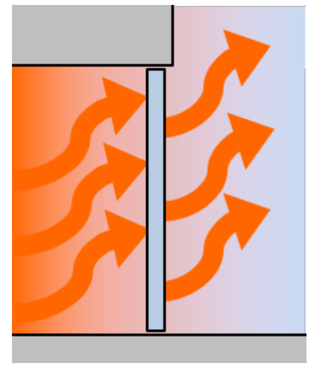

Figure 6: Long Wave Radiation Losses

$$\mathbf{P_{net}} = \mathbf{P_{emit}} - \mathbf{P_{absorb}} \tag{2.5.1}$$

$$\mathbf{Q_{longwave}} = \mathbf{A}\sigma\varepsilon\left(\mathbf{T_s^4} - \mathbf{T_{amb}^4}\right) \tag{2.5.2}$$

where,

A : Area of the body surface (door), $[m^2]$

σ : Constant of Stefan-Boltzmann, $\left(\sigma \approx 5,670 * 10^{-8} \frac{W}{m^2 K^4}\right)$

T_s : Temperature of the body (door) surface, $[K]$

ε : Emissivity $(\approx 0,8 - 0,95)$, [adimensional]

T_{amb} : Ambient temperature, $[K]$

This means that there is an energy flow between the external surface of the door and the ambient outside the building ($T_{amb} = T_{out}$), and between the internal surface and the ambient inside the building ($T_{amb} = T_{in}$).

The amount of this effect is related to the surface temperature, which is also related to the U-value, so it is usual that it is not taken into account for product specification. But, as we will see, it is not negligible in terms of energy losses as it is confirmed by technical literature.

2.6 Preliminary evaluation of the different effects

To have a preliminary idea of the order of magnitude of the different effects, we make a simple evaluation just applying the basic equations (stationary approach) with rough parameter estimations. We consider the following effects:

- Heat transmission, air leakage, solar radiation and long wave radiation when the door is closed.
- Air infiltration and solar radiation when the door is open.

We will calculate the amount of energy due to the different effects in kWh for 1 year.

As a simplification, we do not consider at this introductory stage the bulk density flow.

We calculate a typical case giving usual values for the parameters that are summarised in Table 1.

Parameters	*Symbol*	*Unit*	*Value*
Outside Temperature	T_{out}	$[^oC]$	10
Inside Temperature	T_{in}	$[^oC]$	20
Wind Pressure	P_w	$[Pa]$	12
Door Width	DW	$[m]$	3
Door Height	DH	$[m]$	3
Building Height	$H_{building}$	$[m]$	5
Buiding Area	$A_{building}$	$[m^2]$	1000
Thermal Transmittance	U	$[W/m^2K]$	$2,5$
Air Permeability	L	$[m^3/m^2h]$	6
Reference Pressure	P_R	$[Pa]$	50
Inside Specific Humidity	χ_{in}	$[kg/kg]$	$0,0075$
Outside Specific Humidity	χ_{out}	$[kg/kg]$	$0,0025$
Solar Heat Gain Factor (door closed)	$SHGC$	$[adimensional]$	$0,4$
Solar Heat Gain Factor (door open)	$SHGC$	$[adimensional]$	$0,6$
Irradiance	I	$[W/M^2]$	100
Incident Angle	ϕ	$[^o]$	30
Door Glazing Area	A_{glaze}	$[m^2]$	3
External Door Surface Temperature	$T_{doorout}$	$[^oC]$	12
Internal Door Surface Temperature	T_{doorin}	$[^oC]$	15
Emissivity	ε	$[adimensional]$	$0,9$

Table 1: Typical case parameters for preliminary evaluation.

The values for the constants are taken as described before in this chapter. With these parameter values, applying all the equations above, we have the results we show in Table 2.

Energy losses effect	*Flow* $[W]$	*Energy per year* $[kWh]$
Transmission	$225,0$	$178,4$
Air Leakage	$219,4$	$173,9$
Solar Radiation (door closed)	$93,5$	$74,1$
Long Wave Radiation	$309,4$	$245,2$
Air Infiltration	$207667,1$	$17305,7$
Solar Radiation (door open)	$163,7$	$13,6$

Table 2: Typical case energy results.

We see that infiltration losses are two orders of magnitude over the rest of factors. This balance is strongly dependant on the number of opening cycles and opening time.

If we consider only door closed situation, we see that the order of the rest of the factors is quite similar, including the long wave radiation.

Energy losses effect	*Sensible Heat* $[kWh]$	*Latent Heat* $[kWh]$
Air leakage	$160,7$	$0,2$
Infiltration	$16825,6$	$18,9$

Table 3: Sensible and latent heat comparison.

We also see that latent heat in air leakage has a very limited weight and it be neglected in general. Latent heat in air infiltration has a more significant influence, but its relative weight in the total infiltration losses is quite low, so we will also leave it out of the scope of the simplified model. Only for large humidity differences should be taken into account in accurate calculations.

Regarding solar effect, it can be more influential in pedestrian full glass doors than in industrial doors, but it will be usually be smaller than other factors. For this reason we will leave it out of the scope of the simplified model, but having in mind that it should be included if a more precise calculation is required in certain situations.

3 Physical and Mathematical Model

Once that we have the complete basic equations of the different convergent phenomena related to the energy losses of our building through a door, we will develop a transient analytic ”simplified” model to be able to predict the energy losses, which is the main objective of this study.

Although the previous simple equations are valid as they are for some phenomena (transmission, air leakage), for the air infiltration a deeper approach has to be done to have a more precise model, specially for the bulk density flow.

3.1 Bulk Density Flow

As we will confirm by specific experiments, there is a large heat exchange through a door open even in absence of wind if there is a temperature difference between inside and outside the building. There has been some approximations to this phenomena like [**1**] and [**2**], but its importance to the door field deserves a specific approach.

The classical simplified approach of the so-called gravitational flow is the application of the continuity equation and Bernoulli theorem on both sides of the large opening. This is equivalent to assume the inter-zone air flow to be a steady flow (no variation with time) of an in-compressible, non-viscous fluid of constant density, that is only driven by pressure gradients on both sides of the opening.

We will make an integration of the infiltration flux, partially based in [**2**], where some details now shown can be followed.

3.1.1 Mass Flow between Stratified Zones

In the following, an expression for the mass flow through a large opening separating two zones of different temperature and pressure will be derived.

The general case is considered. There is a static pressure difference at reference level z_0 between zones 1 and 2, and different vertical temperature profiles occur in the two zones. These temperature profiles are assumed linear, though the temperature gradients can be different. The situation is depicted in Figure 7, where some parameters are described.
First we assume that the conditions are such that the neutral level, i.e. the level at which the pressure is equal in zones 1 and 2, is located in the opening, so that bidirectional air flow occurs.

The stack pressure difference between a point at height z and a point at reference height z_0 is calculated with:

$$\Delta P(z) - \Delta P(z_0) = \int_{z_0}^{z} g\left[\rho_2(\xi) - \rho_1(\xi)\right] d\xi \;\; [Pa] \tag{3.1.1}$$

Although density variations due to pressure variations are negligibly small, those resulting from temperature differences should be taken into account, especially when temperature gradients are large. The air density in zone i is inversely proportional to the temperature, namely:

$$\rho_i = \frac{P_{ref} M}{RT_i} \;\; \left[kg/m^3\right] \tag{3.1.2}$$

where P_{ref} is a reference pressure, e.g. the atmospheric pressure, M is the molecular weight of air, and R is the universal gas constant. For a very good approximation we may write:

$$\rho_i = \frac{K}{T_i} \;\; \left[kg/m^3\right] \tag{3.1.3}$$

where K is a constant. The expression for $\Delta P(z)$ now becomes:

$$\Delta P(z) - \Delta P(z_0) = \int_{z_0}^{z} gK \left[\frac{1}{T_2(\xi)} - \frac{1}{T_1(\xi)} \right] d\xi \;\; [Pa] \tag{3.1.4}$$

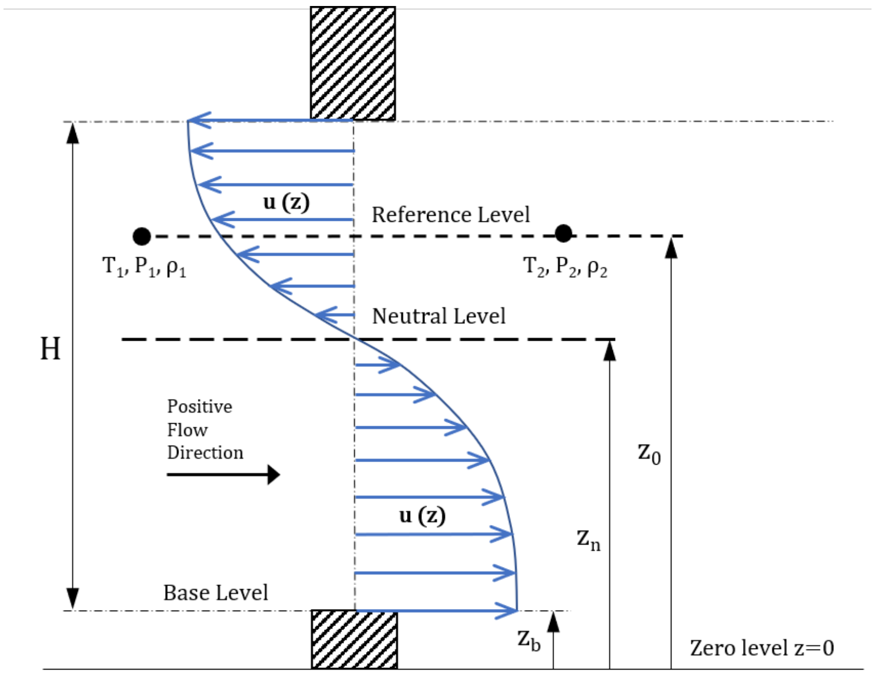

Figure 7: Flow profile and parameter definition

Assuming a linear temperature profile $T_i(z) = a_i + b_i \cdot z$ we obtain:

$$\Delta P(z) - \Delta P(z_0) = \int_{z_0}^{z} gK \left[\frac{1}{a_2 + b_2 \cdot \xi} - \frac{1}{a_1 + b_1 \cdot \xi} \right] d\xi \;\; [Pa] \tag{3.1.5}$$

$$\Delta P(z) - \Delta P(z_0) = gK \left[\frac{1}{b_2} \ln \left(\frac{T_2(z)}{T_2(z_0)} \right) - \frac{1}{b_1} \ln \left(\frac{T_1(z)}{T_1(z_0)} \right) \right] \;\; [Pa] \tag{3.1.6}$$

If the temperature gradient in both zones is not too large, we have to a very good approximation:

$$\ln \left(\frac{T_i(z)}{T_i(z_0)} \right) \approx \frac{T_i(z) - Ti(z_0)}{T_i(z_0)} \tag{3.1.7}$$

The first order approximation is highly accurate. Inserting the linear temperature profile:

$$\ln \left(\frac{T_i(z)}{T_i(z_0)} \right) \approx b_i \frac{z - z_0}{T_i(z_0)} \tag{3.1.8}$$

So in first order approximation:

$$\Delta P(z) - \Delta P(z_0) \approx gK \left[\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right] (z - z_0) \;\; (Pa) \tag{3.1.9}$$

This means that $\Delta P(z)$ changes linearly with the height coordinate z when temperatures in both zones differ at the reference height z_0. Note that the first order approximation it results in a equation which is independent of the temperature gradients in both zones, b_1 and b_2.

If the opening through which the air flows extends from z_b to $z_b + H$, the pressure difference between the two zones at bottom level z_b will be equal to:

$$\Delta P(z_b) = \Delta P(z_0) + gK \left[\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right] (z_b - z_0) \ \ [Pa] \tag{3.1.10}$$

and at the top of the opening:

$$\Delta P(z_b + H) = \Delta P(z_0) + gK \left[\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right] (z_b + H - z_0) \ \ [Pa] \tag{3.1.11}$$

Now, if either $\Delta P(z_b) > 0$ and $P(z_b + h) < 0$ or $\Delta P(z_b) < 0$ and $\Delta P(z_b + H) > 0$ then the neutral level z_n is located inside the opening and bidirectional airflow occurs. If $P(z_b)$ and $\Delta P(z_b + H)$ have the same sign, or if one of them is zero, only unidirectional flow takes place.

According to Bernoulli's Law, a pressure difference $\Delta P(z)$ results in a local air velocity $u(z)$ proportional to the square root of $\Delta P(z)$. Therefore, an infinitesimal volume flow $d\dot{q}$. through an element of height dz in the opening can be written as:

$$d\dot{q} = Wu(z)dz \ \ [m^3/s] \tag{3.1.12}$$

Where W is the opening (door) width.

In Annex 6.1 we develop a complementary approach based on the deduction of $u(z)$ instead of $\Delta P(z)$ where the velocity profile seen in Figure 7 is analytically described.

If we consider the case where $T_2 > T_1$ and where the pressures at reference level z_0 in both zones are such that the neutral level is located inside the opening (so bidirectional airflow will occur), then the mass flow from 2 to 1 is equal to:

$$\dot{m}_{21} = \int_{z_n}^{z_b+h} \rho_2 d\dot{q} = C_d W \sqrt{2\rho_2} \int_{z_n}^{z_b+h} \Delta P(z)^{1/2} dz \ \ [kg/s] \tag{3.1.13}$$

and the mass flow from 1 to 2 is equal to:

$$\dot{m}_{12} = \int_{z_b}^{z_n} \rho_1 d\dot{q} = C_d W \sqrt{2\rho_1} \int_{z_b}^{z_n} \Delta P(z)^{1/2} dz \ \ [kg/s] \tag{3.1.14}$$

where C_d is an empirical constant (discharge coefficient) that can be approximated with the expression (2.3.2).

In these expressions, the error made by placing $\sqrt{2\rho_i}$ in front of the integral sign is negligible because density variations are very small over the integration interval when compared to variations in $\Delta P(z)$. Inserting the linear expression for $\Delta P(z)$ into the integrals gives for $\dot{m}_{21}$ and $\dot{m}_{12}$ the following expressions:

$$\dot{m}_{21} = \frac{2}{3} C_d W \sqrt{2\rho_2} H \frac{C_a^{3/2}}{C_t} \ \ [kg/s] \tag{3.1.15}$$

$$\dot{m}_{12} = \frac{2}{3} C_d W \sqrt{2\rho_1} H \frac{-C_b^{3/2}}{C_t} \ \ [kg/s] \tag{3.1.16}$$

where:

$$C_t = hgK\left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)}\right) = C_a - C_b$$
$$C_a = \Delta P(z_b + h)$$
$$C_b = \Delta P(z_b)$$

Note that in the situation in Figure 7, the pressure difference at the top level of the opening, $C_a = \Delta P(z_b+h)$ is negative, so that $C_a^{3/2}$ is an imaginary number. To keep the value of $\dot{m}_{21}$ real, C_a should be taken in absolute value. It is convenient, however, to write the net mass flow of air through the opening as a complex quantity, i.e.:

$$\dot{m} = \dot{m}_{21} + \dot{m}_{12} = \frac{2\sqrt{2}}{3}C_d W\frac{1}{C_t}\left(\sqrt{\rho_2}C_a^{3/2} - \sqrt{\rho_1}C_b^{3/2}\right) \quad [kg/s] \tag{3.1.17}$$

This expression was first derived by Cockroft (1979). The net mass flow is a complex number, of which the real part is the flow from 1 to 2 and the imaginary part is the flow from 2 to 1.

It must be emphasized that the Cockroft formula for $\dot{m}_{net}$ in the form given above only holds for the special case depicted in Figure 7. There are two reasons why it is necessary to modify the expression.

First, if zone 1 on the left having warmer zone instead of the cooler one, $\dot{m}_{12}$ would take place above the neutral level, and $\dot{m}_{21}$ below it. The integration interval for both contributions would be interchanged, so that in Cockroft's expression, the term containing Ca is now $\dot{m}_{12}$ and the term containing C_b is now $\dot{m}_{21}$. The formula now reads:

$$\dot{m}_{net} = \dot{m}_{21} + \dot{m}_{12} = \frac{2\sqrt{2}}{3}C_d W\frac{1}{C_t}\left(\sqrt{\rho_1}C_a^{3/2} - \sqrt{\rho_2}C_b^{3/2}\right) \quad [kg/s] \tag{3.1.18}$$

However, the real part still gives the flow from 1 to 2 and the imaginary part still gives the flow from 2 to 1.

Second, if the external pressures in both zones differ considerably, the neutral level will shift to a height below or above (i.e. outside) the opening, so that the airflow becomes unidirectional. In this situation, one of the flow terms results from an integration over the entire opening, from z_b to z_b+H, while the other term is cancelled. In the situation of unidirectional flow, the pressure differences at the bottom and top of the opening, C_b and C_a, have the same sign (unless one of them vanishes), so that $C_a^{3/2} - C_b^{3/2}$ is either a real or a pure imaginary number.

By carefully comparing the expressions for $\dot{m}_{net}$ which can be established for the different cases of unidirectional and bidirectional flow, that is by "tuning" the temperature difference and the pressure difference between zone 1 (left) and zone 2 (right), the following very convenient formula for $\dot{m}_{net}$ which holds in all cases can be obtained:

$$\dot{m}_{net} = \dot{m}_{12} + \dot{m}_{21} \tag{3.1.19}$$

Where:

$$\dot{m}_{12} = \sqrt{\rho_1}Re\left(Z_a - Z_b\right) \ \geq 0 \ \ ([kg/s]$$
$$\dot{m}_{21} = -\sqrt{\rho_2}Im\left(Z_a - Z_b\right) \ \leq 0 \ \ [kg/s]$$
$$Z_a = \frac{2\sqrt{2}}{3}C_d HW\frac{C_a^{3/2}}{C_t} \ \ [m^2 Pa^{1/2}]$$

$$Z_b = \frac{2\sqrt{2}}{3} C_d H W \frac{C_b^{3/2}}{C_t} \ \ [m^2 Pa^{1/2}]$$

As the direction $1 \rightarrow 2$ is, by definition, the positive direction, the contribution $\dot{m}_{21}$ should be non positive, which explains the minus sign appearing in the expression. The artificial complex quantities Z_a and Z_b are introduced for convenience and have no physical meaning. In the complex plane, (Z_a - Z_b) is located either on the positive real axis (when there is a unidirectional flow $1 \rightarrow 2$), on the positive imaginary axis (when there is a unidirectional flow $2 \rightarrow 1$), or in the first quadrant of the complex plane (when the flow is bidirectional). When for a given temperature difference between zone 1 and 2 the external pressure difference $\Delta P(z_0)$ is continuously increased from highly negative to highly positive, (Z_a Z_b) describes a smooth continuous curve.

3.1.2 Heat Flow between Stratified Zones

Just as for the mass flow, a convenient expression for the bidirectional heat flow through a large opening between stratified zones can be derived, giving Φ_{12} and Φ_{21} as real and imaginary parts of complex quantities.

Whereas mass flows are calculated by evaluating integrals of the type:

$$\int \rho_i(z) d\dot{q} \ \ (kg/s) \tag{3.1.20}$$

heat flows are calculated by evaluating integrals of the type:

$$\int C_p T_i(z) \rho_i(z) d\dot{q} = C_p C_d W \int \sqrt{2\rho_i(z)} T_i(z) \sqrt{\Delta P(z)} dz \ \ (kg/s) \tag{3.1.21}$$

in an analogous way.

To be able to evaluate these integrals analytically for linear temperature profiles $T_i(z) = T_i(z_0) + b_i(z - z_0)$, the integrand $\sqrt{2\rho_i(z)} T_i(z) \sqrt{\Delta P(z)}$ above, should be of the form $[polynomial] \cdot \sqrt{\Delta P(z)}$ which means that $\sqrt{2\rho_i(z)} T_i(z)$ should be approximated by its "best linear fit", which is (as can be checked easily): $\sqrt{2\rho_i(z_0)} \cdot \left[T_i(z_0) + \frac{1}{2} b_i (z - z_0)\right]$.

The evaluation of the integrals is a rather laborious task, which will not be documented here (the full derivation can be provided to interested readers). However, when these integrals are worked out in the same way as was done for the mass flows, we obtain convenient expressions for the heat flows Φ_{12} and Φ_{21}.

The contribution of the outside air velocity must be also taken into account. According to the results found in [**1**], there is an experimental correction of the wind pressure due to turbulence effects in large openings and this pressure can be expressed as:

$$P_{wind} = \frac{1}{2} \rho \cdot 0.0029 \cdot v_{wind}^2 \tag{3.1.22}$$

Finally the integrated Energy flux results as:

$$\Phi_{tot} = \Phi_{12} + \Phi_{21} \tag{3.1.23}$$

Where:

$$\Phi_{12} = C_p \sqrt{\rho_1(z_0)} Re \left(\tilde{T}_{1a}(z_0) Z_a - \tilde{T}_{1b}(z_0) Z_b \right)$$

$$\Phi_{21} = C_p \sqrt{\rho_2(z_0)} Im \left(\tilde{T}_{2a}(z_0) Z_a - \tilde{T}_{2b}(z_0) Z_b \right)$$

$$\tilde{T}_{ia}(z_0) = T_i(z_0) - b_i H \left[\frac{C_a}{5\,C_t} + \frac{\alpha - 1}{2} \right]$$

$$\tilde{T}_{ib}(z_0) = T_i(z_0) - b_i H \left[\frac{C_a}{5\,C_t} + \frac{\alpha}{2} \right]$$

$$\alpha = \frac{z_0 - z_b}{H}$$

$$Z_a = \frac{2\sqrt{2}}{3}\, C_d\, W \frac{C_a^{3/2}}{C_t}$$

$$Z_b = \frac{2\sqrt{2}}{3}\, C_d\, W \frac{C_b^{3/2}}{C_t}$$

$$C_a = \Delta P(z_b + H) = \Delta P(z_0) + g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right) (z_b + h - z_0)$$

$$C_b = \Delta P(z_b) = \Delta P(z_0) + g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right) (z_b - z_0)$$

$$C_t = C_a - C_b$$

The discharge coefficient as seen:

$$C_d = \frac{1}{\sqrt{1,75 + 0,7 \cdot e^{(-\frac{W}{32.5 \cdot H})}}}$$

And the linear variation of temperature and pressure can be expressed:

$$T_i(z) = a_i + b_i \cdot z$$

$$P_i(z) = c_i + d_i \cdot z + \frac{1}{2}\rho \cdot 0.0029 \cdot v_{wind}^2$$

And the density:

$$\rho_i = \frac{K_{air}}{T_i(z)}$$

$$K_{air} = \frac{101325}{R}$$

$$R = 287.05\, \frac{J}{kg\, K}$$

In practice, a_i and b_i are empirical and define the linear variation off the temperature across the opening height on both sides, as well as c_i and d_i have the same function for the pressure linear variation.

Typical values of these parameters are the following, confirmed from the results described in chapter 4.5:

$$c_1 = c_2 \approx 101325\ Pa \ \ (atmospheric\ pressure)$$

$$d_2 \approx 12 \div 30\ Pa/m$$

In chapter 4.5 we also found that we can approximate:

$$d_1 = d_2 + \frac{0,0125}{z_0^3} \tag{3.1.24}$$

We also confirmed:

$$b_i \approx 0,1 \div 4 \ K/m$$

And a_i can be calculated from with the linear expression of $T_i(z)$:

$$a_i = T_i(z_0) - b_i z_0 \tag{3.1.25}$$

The heat flow Φ_{tot} obtained in this way is a function of T_{in} and T_{out}. In order to simplify the problem by linearizing the equations, we can assumed that:

$$\Phi_{tot} = K_{infil} \cdot (T_{in} - T_{out}) \tag{3.1.26}$$

Then we can extract the constant K_{loss} by infiltration to include in the overall model, which is our objective:

$$K_{infil} = \frac{\Phi_{tot}}{T_{in} - T_{out}} \tag{3.1.27}$$

3.1.3 Door Height Correction due to Boundary Effects

In the previous model, no boundary layer effect was considered, and this means that the maximum velocity value is found on door bottom $z = 0$. Also, it increases indefinitely with the door height H, increasing the energy flux Φ_{tot} in the same way. This takes to increasing errors with increasing door height.

We know that the velocity profile according to known wind profile power law can be taken as follows:

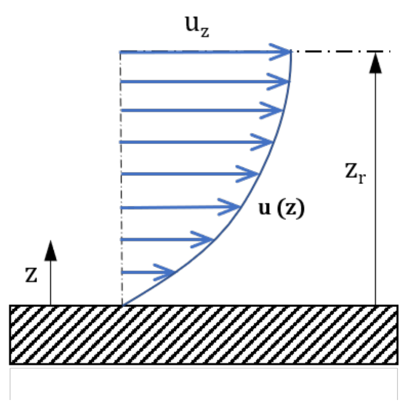

Figure 8: Wind boundary profile

$$u(z) = u_r \left(\frac{z}{z_r}\right)^{1/7} \tag{3.1.28}$$

Where u_r is the wind velocity at the reference height z_r. With this more realistic air flow profile we have $u = 0$ at the bottom ($z = 0$, where a boundary condition exits) and increases until it reaches $z = z_r$, where the wind velocity is $u = u_r$. In the following figure we show the ideal and corrected profile with the described law:

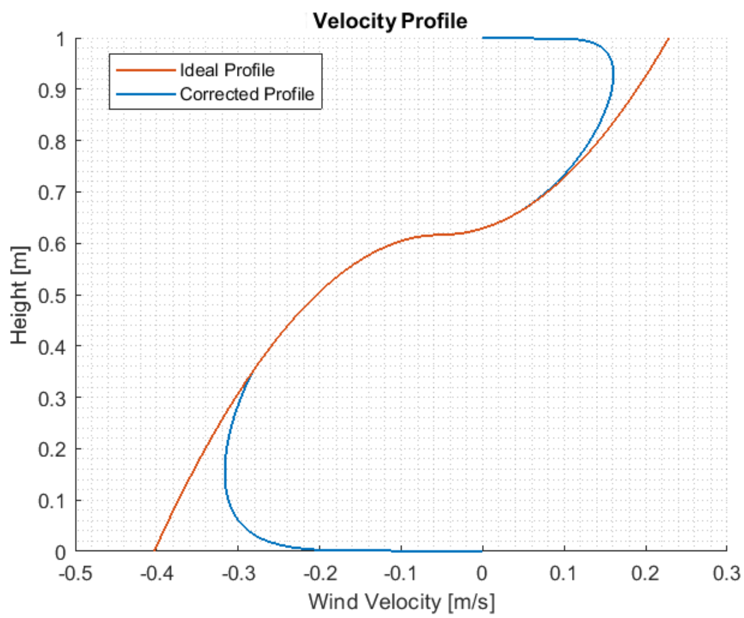

Figure 9: Theoretical wind velocity profiles

The integration of the flow equation including the effect is quite complex, so a different approach was made to have an analytic calculation useful for our objectives. We consider an exponential correction function $\delta(H)$ adjusted with our experimental results:

$$\delta(H) = \frac{7}{8}e^{-0.3073 \cdot H} \tag{3.1.29}$$

This factor is included multiplying the flux, decreasing its value depending on door height.

3.2 Energy Losses Model

As a first step to define the simplified model, we will analyse the thermodynamics equations that govern the global problem.

3.2.1 Temperature Profile and Differential Equation

As we know, if the door is open, we will have a heat loss flux due to a wind infiltration effect. If the door is closed we will have a conduction and air permeability heat loss effect. So let us assume a general heat loss flux (independent if the door is open or closed), and an incoming heat flux due to the Heating. Therefore if we assume that Kloss is constant and doesn't depend on the Temperature at all, we can arrive at the following Differential Equation:

$$-dE_{\text{loss}} + dE_{\text{heat}} = dE_{in} \tag{3.2.1}$$

$$-\dot{Q}_{\text{loss}}dt + \dot{Q}_{\text{heat}}dt = dE_{in} \tag{3.2.2}$$

where $\dot{Q}_{loss}$ is the Energy Flux going inside/outside the building (crossing door):

$$\dot{Q}_{\text{loss}} = K_{\text{loss}}\left(T_{in}(t) - T_{out}\right) \tag{3.2.3}$$

$\dot{Q}_{\text{heat}}$ is the Energy Flux of Heating system and E_{in} is the Internal Energy:

$$E_{in} = C_p \rho V_{ol} T_{in}(t) \tag{3.2.4}$$

If we develop Equation 3.2.2 and integrate at both sides of the equation, we can arrive to the following:

$$-\int_0^t \frac{K_{\text{loss}}}{C_p \rho V_{ol}} dt' = \int_{T_{ini}}^{T_{in}(t)} \frac{1}{T_{in} - T_{out} - \frac{\dot{Q}_{\text{heat}}}{K_{\text{loss}}}} dT' \tag{3.2.5}$$

$$T_{in}(t) = T_{out} + \frac{\dot{Q}_{\text{heat}}}{K_{\text{loss}}} - \left[T_{out} + \frac{\dot{Q}_{\text{heat}}}{K_{\text{loss}}} - T_{ini}\right] e^{-\frac{K_{\text{loss}}}{C_v \rho V_{ol}} t} \tag{3.2.6}$$

Rearranging this equation, we arrive to:

$$T_{in}(t) = T_{eq} - \left[T_{eq} - T_{ini}\right] e^{-\frac{K_{\text{Loss}}}{C_p \rho V_{ol}} t} \tag{3.2.7}$$

where,

$$T_{eq} = T_{out} + \frac{\dot{Q}_{\text{heat}}}{K_{\text{loss}}}$$

We can see that the behaviour of the Equation (3.2.7) is very intuitive, and in fact it is a well known problem (Newton Cooling Law). As we can see, when the time is $t = 0$, we start at T_{ini}, and after a very long time, when $t \to \infty$ the Temperature is T_{eq}. We also see that the larger K_{loss} is (larger heat flux loss), the faster it will arrive at T_{eq}, and the larger the Volume V_{ol} is, the slower it will arrive at T_{eq}.

The next step will be to include in the above equations the effects described in chapter 2.

3.2.2 Long Wave Radiation Effect

According to (2.5) one approach to include this effect in the comprehensive model could be to include it in the Q_{Heat} expression as a constant heat flux. But it cannot be calculated directly, as $\dot{Q}_{\text{long wave}}$ depends on the inside Temperature $T_{\text{in}}(t)$, the Temperature of the door inner surface $T_{\text{door in}}(t)$, the Temperature of the door outer surface $T_{\text{door out}}(t)$ and the outside Temperature $T_{\text{out}}(t)$:

$$\dot{Q}_{\text{long wave}} = -\sigma\varepsilon A(T_{\text{in}}^4 - T_{\text{door in}}^4) - \sigma\varepsilon A(T_{\text{door out}}^4 - T_{\text{out}}^4) \tag{3.2.8}$$

$$\dot{Q}_{\text{long wave}} = -\sigma\varepsilon A(T_{\text{in}}^4 - T_{\text{door in}}^4 + T_{\text{door out}}^4 - T_{\text{out}}^4) \tag{3.2.9}$$

The calculation of of $T_{\text{door in}}$ and $T_{\text{door out}}$ is dependant both on the heat transmission and the long wave effect, and it takes to a non linear complex calculation.

As a first simplification we only consider the effect of heat transmission. According to (2.1.2) we know that the overall heat transfer transmission coefficient U is:

$$U = \frac{1}{\frac{1}{\alpha_{in}} + \frac{1}{k} + \frac{1}{\alpha_{out}}} \tag{3.2.10}$$

The convection coefficients are calculated in well known iterative calculations based on experimental coefficients taking into account a lot of parameters. As a first approach, for this kind of natural convection on doors the values are in the following orders:
where normally,

$$\alpha_{in} \approx 7.7 \ - \ 9 \ \left[\frac{W}{m^2 K}\right]$$

$$\alpha_{out} \approx 14.3 \ - \ 16.7 \ \left[\frac{W}{m^2}\right]$$

With this values we can compute the value of k, as the value of U is given by the door manufacturer. Then it would be:

$$k = \left(\frac{1}{U} - \frac{1}{\alpha_{in}} - \frac{1}{\alpha_{out}}\right)^{-1} \tag{3.2.11}$$

Knowing the parameters α_{in}, α_{out} and K, we can find the surface Temperatures at the door, with the following expressions:

$$\alpha_{in} \left(T_{\text{in}} - T_{\text{door in}}\right) = k \left(T_{\text{door in}} - T_{\text{door out}}\right) \tag{3.2.12}$$

$$k \left(T_{\text{door in}} - T_{\text{door out}}\right) = \alpha_{out} \left(T_{\text{door out}} - T_{\text{out}}\right) \tag{3.2.13}$$

Then, resolving the system, we have $T_{\text{door in}}$, and $T_{\text{door out}}$:

$$T_{\text{door in}} = \frac{\alpha_{in}\alpha_{out}T_{in} + \alpha_{in}kT_{in} + \alpha_{out}kT_{out}}{\alpha_{in}\alpha_{out} + \alpha_{in}k + \alpha_{out}k} = (A + C) \cdot T_{in} + B \cdot T_{out} \tag{3.2.14}$$

$$T_{\text{door out}} = \frac{\alpha_{in}\alpha_{out}T_{out} + \alpha_{in}kT_{in} + \alpha_{out}kT_{out}}{\alpha_{in}\alpha_{out} + \alpha_{in}k + \alpha_{out}k} = A \cdot T_{in} + (B + C) \cdot T_{out} \tag{3.2.15}$$

where,

$$\begin{aligned} A &= \frac{\alpha_{in}\alpha_{out}}{\alpha_{in}\alpha_{out} + \alpha_{in}k + \alpha_{out}k} \\ B &= \frac{\alpha_{in}k}{\alpha_{in}\alpha_{out} + \alpha_{in}k + \alpha_{out}k} \\ C &= \frac{\alpha_{out}k}{\alpha_{in}\alpha_{out} + \alpha_{in}k + \alpha_{out}k} \end{aligned} \tag{3.2.16}$$

As a result of the previous developments, we will have the following:

$$\dot{Q}_{\text{long wave}} = -\sigma\varepsilon A \left(T_{\text{in}}^4 - ((A + C) \cdot T_{\text{in}} + B \cdot T_{\text{out}})^4 + (A \cdot T_{\text{in}} + (B + C) \cdot T_{\text{out}})^4 - T_{\text{out}}^4\right) \tag{3.2.17}$$

We can try to simplify this expression with a linear function dependant on ΔT:

$$\dot{Q}_{\text{long wave}} = -\sigma\varepsilon A \cdot T_{eq}(\Delta T) \tag{3.2.18}$$

As ($\Delta T = T_{in} - T_{out}$) is quite small in a range of ($\Delta T = 0 - 40\ ^oC$), the equation can be linearized without any concern. This linearization would be,

$$\begin{aligned} T_{eq}(\Delta T) = 4\frac{(\alpha_{in}+\alpha_{out})\,UT_{\text{out}}^3}{\alpha_{in}\alpha_{out}} \cdot \Delta T + 6\frac{\left(U\left(\alpha_{in}^2-\alpha_{out}^2\right)+2\alpha_{in}\alpha_{out}^2\right)UT_{\text{out}}^2}{\alpha_{in}^2\alpha_{out}^2} \cdot (\Delta T)^2 \\ +4\frac{\left(U^2\left(\alpha_{in}^3-\alpha_{out}^3\right)-3U\alpha_{in}\alpha_{out}^3+3\alpha_{in}^2\alpha_{out}^3\right)UT_{\text{out}}}{\alpha_{in}^3\alpha_{out}^3} \cdot (\Delta T)^3 \\ +\frac{\left(U^3\left(\alpha_{in}^4-\alpha_{out}^4\right)+4U^2\alpha_{in}\alpha_{out}^4-6U\alpha_{in}^2\alpha_{out}^4+4\alpha_{in}^3\alpha_{out}^4\right)U}{\alpha_{in}^4\alpha_{out}^4} \cdot (\Delta T)^4 \end{aligned} \tag{3.2.19}$$

This expression remains too complex for our simplified calculation purposes. Knowing that $\Delta T << T_{out}$, as a rough approximation we can consider only the first term of the equation:

$$T_{eq}(\Delta T) \approx 4\frac{(\alpha_{in}+\alpha_{out})\,UT_{\text{out}}^3}{\alpha_{in}\alpha_{out}} \cdot \Delta T \tag{3.2.20}$$

If we express the equation the terms of conductivity k:

$$T_{eq}(\Delta T) \approx 4\frac{(\alpha_{out}+\alpha_{in})k\,T_{out}^3}{\alpha_{in}\,(k+\alpha_{out})+k\alpha_{out}} \cdot \Delta T \tag{3.2.21}$$

We can take into account this effect in our general model as it depends only on ΔT, so we can add an additional factor to K_{loss}. Therefore $K_{\text{long wave}}$ would be,

$$K_{\text{long wave}} \approx 4\sigma\varepsilon A\frac{(\alpha_{in}+\alpha_{out})\,UT_{\text{out}}^3}{\alpha_{in}\alpha_{out}} \tag{3.2.22}$$

If we introduce typical mean values of $\overline{\alpha}_{in} \approx 8.4 \quad \left[\frac{W}{m^2K}\right]$ and $\overline{\alpha}_{out} \approx 15.5 \quad \left[\frac{W}{m^2K}\right]$, we finally have:

$$K_{\text{long wave}} \approx 0,734 \cdot \sigma \cdot \varepsilon \cdot A \cdot U \cdot T_{\text{out}}^3 = 4,163 \cdot 10^{-8} \cdot \varepsilon \cdot A \cdot U \cdot T_{\text{out}}^3 \tag{3.2.23}$$

If we calculate with usual T_{out} values, we verify that these losses have the same order than the transmission losses, so we confirm that it is an effect that should be taken into account with door closed, also considering that the simplifications made are in the energy gain side.

To have a better approximation of the surface temperatures for a more accurate result, it is possible to sum $K_{\text{long wave}}$ to the conductivity k in (3.2.13) and make an iterative calculation.

3.2.3 Heat Transfer Coefficient

Let us note that Equation (3.2.7) is a general equation which can be particularised at any time just changing the limits or the effect.

As it is explained in section 2, it is also important to remember that this model assumes the approximation that the door is fully open from the instant that the door panel starts to move, so we can separate the effects between open and closed stages.

If the door is open, K_{loss} will be only determined by the **Heat Flux due to Infiltration**, so:

$$K_{\text{loss}}^{\text{door open}} = K_{infil} = \frac{\Phi_{tot}}{T_{in} - T_{out}} \qquad \left[\frac{W}{K}\right] \tag{3.2.24}$$

If the door is closed instead, K_{loss} will be the **Heat Flux due to Air Permeability, Heat Transmission and Long Wave Radiation**,

$$K_{\text{loss}}^{\text{door closed}} = K_{\text{trans}} + K_{\text{perm}} + K_{\text{long wave}} \tag{3.2.25}$$

As seen, the Transmission coefficient will be:

$$K_{\text{trans}} = A \cdot U \tag{3.2.26}$$

$$U = \left(\frac{1}{\alpha_{in}} + \frac{1}{\alpha_{out}} + \frac{1}{K}\right)^{-1} \tag{3.2.27}$$

And the Air Leakage coefficient, combining the Wind effect and the Stack effect as seen in chapter 2:

$$K_{\text{perm}} = C_p \rho \cdot \dot{V}_{\text{perm}} \qquad \left[\frac{W}{K}\right] \tag{3.2.28}$$

$$\dot{V}_{\text{perm}} = \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_r}\right)^{2/3} \tag{3.2.29}$$

$$P_w = \frac{1}{2} \rho \cdot 0.0029 \cdot v_{wind}^2 \cdot \tag{3.2.30}$$

$$P_s = \frac{\rho g \left(H_b - H_{np}\right) \left(T_{in} - T_{out}\right)}{T_{out}} \tag{3.2.31}$$

where, P_w is the Pressure induced by the Wind, and P_s is the Pressure induced by the Stack effect.

Substituting:

$$K_{\text{loss}}^{\text{door closed}} = A \cdot U + C_p \rho \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_R}\right)^{2/3} + 4\sigma A \frac{\left(\alpha_{in} + \alpha_{out}\right) U T_{\text{out}}^3}{\alpha_{in} \alpha_{out}} \qquad \left[\frac{W}{K}\right] \tag{3.2.32}$$

3.2.4 Heat Flux

Next to the heating, we should add the flux due to **Solar Radiation effect** to the total heat flux. We would have then:

$$\dot{Q}_{\text{heat}} = \dot{Q}_{\text{heating}} + \dot{Q}_{solar} \tag{3.2.33}$$

where,

$$\dot{Q}_{\text{heating}} = C_{\text{heat}} V_{ol} \tag{3.2.34}$$

In the summer, if there is a cooling system, then we will have an equivalent equation for the cooling input.

The heat flux due to solar effect would be as seen before:

$$\dot{Q}_{solar} = I \cdot SHGC \cdot A_{\text{ef}}$$

The value of $SGHC$ changes in case that the door is open, as the transmission factor of the glass disappears and the glazing area equals the door Area. Then we will have two expressions:

$$\dot{Q}_{\text{solar}}^{\text{door closed}} = SHFD_{door} \cdot A_{\text{ef}} \tag{3.2.35}$$

$$A_{\text{ef}}^{\text{door closed}} = A_{glaze} \cos\phi \tag{3.2.36}$$

$$\dot{Q}_{\text{solar}}^{\text{door open}} = SHFD_{hole} \cdot A_{\text{ef}} \tag{3.2.37}$$

$$A_{\text{ef}}^{\text{door open}} = A_{door} \cos\phi \tag{3.2.38}$$

As a result of the considerations made in chapter 2.4, we leave it out of the simplified model object of this study, knowing that is contribution its not negligible in certain situations.

3.2.5 Summary of Heat Transfer Coefficient $\mathbf{K_{loss}}$ and Heat Flux $\mathbf{Q_{heat}}$

We make a summary of the main equations to sum up all the effects commented before:

1. **Infiltration Effect**: Computed when the door is open, in the $\mathbf{K_{infil}}$

$$K_{\text{infil}} = \frac{\Phi_{tot}}{T_{in} - T_{out}} \quad \left[\frac{W}{K}\right]$$

2. **Air Leakage or Permeability Effect**: Computed when the door is closed, is the sum a gradient of Pressure due to Stack Effect and due to the Wind Pressure, we take it into account in $\mathbf{K_{perm\ air}}$

$$K_{\text{perm}} = C_p \cdot \rho \cdot \dot{V}_{\text{perm}} \quad \left[\frac{W}{K}\right]$$

3. **Transmission Effect**: Computed when the door is closed, is the sum a natural convection inside, thermal conductivity, and natural convection outside, this effect is taken into account in the term $\mathbf{K_{trans}}$

$$K_{\text{trans}} = A \cdot U \quad \left[\frac{W}{K}\right]$$

4. **Long Wave Radiation Effect**: Computed when the door is closed, is the balance between the radiation of the inside flux, inside door, outside door and outside flux. It has been linearized and only taken into account the first term ΔT of of this effect. This is taken into account in the term $\mathbf{K_{Trans}}$

$$K_{\text{long wave}} \approx 4\sigma\varepsilon A \frac{(\alpha_{in} + \alpha_{out})\, U T_{\text{out}}^3}{\alpha_{in}\alpha_{out}} \quad \left[\frac{W}{K}\right]$$

5. **Heating Effect**: Computed when the inside Temperature is lower than T_{ref} although the door is closed or open. This is taken into account in the term $\dot{\mathbf{Q}}_{\mathbf{heating}}$.

$$\dot{Q}_{\text{heating}} = C_{\text{heat}} V_{ol} \quad [W]$$

6. **Solar Effect**: Computed when the door is closed and open (with the corresponding coefficients), This should be taken into account in the term $\dot{\mathbf{Q}}_{\mathbf{solar}}$

$$\dot{Q}_{\text{solar}}^{\text{door closed}} = SHFD_{door} \cdot A_{\text{ef}} \quad [W]$$
$$\dot{Q}_{\text{solar}}^{\text{door open}} = SHFD_{hole} \cdot A_{\text{ef}} \quad [W]$$

Finally we have the comprehensive general equations:

$$-\dot{Q}_{\text{loss}} dt + \dot{Q}_{\text{heat}} dt = dE_{in}$$

$$T(t) = T_{eq} - \left[T_{eq} - T_{ini}\right] e^{-\frac{K_{\text{loss}}}{C_p \rho V_{ol}} t}$$

$$T_{eq} = T_{out} + \frac{\dot{Q}_{\text{heat}}}{K_{\text{loss}}}$$

$$\dot{Q}_{\text{heat}} = \dot{Q}_{\text{heating}} + \dot{Q}_{\text{solar}}$$
$$K_{\text{loss}} = K_{\text{trans}} + K_{\text{perm air}} + K_{\text{long wave}} + K_{\text{infil}}$$

We can separate for door closed and open situations:

$$\dot{Q}_{\text{heat}}^{\text{door closed}} = \dot{Q}_{\text{heating}} + \dot{Q}_{\text{solar}}^{\text{door closed}}$$

$$\dot{Q}_{\text{heat}}^{\text{door open}} = \dot{Q}_{\text{heating}} + \dot{Q}_{\text{solar}}^{\text{door open}}$$

$$K_{\text{loss}}^{\text{door closed}} = K_{\text{trans}} + K_{\text{perm air}} + K_{\text{long wave}}$$
$$K_{\text{loss}}^{\text{door open}} = K_{\text{infil}}$$

3.2.6 Time $\mathbf{t_h}$ to reach a Reference Temperature $\mathbf{T_{ref}}$

To integrate the Temperature function and synthesize the model, we have to calculate the time intervals defining the transient heating process.

The Reference Temperature T_{ref} is the temperature at which the Heating or Cooling system switches on. The first objective is to calculate the time t_h to reach it. Let's assume that our building is at an initial Temperature T_{ini}, and there is a heat flux loss due to K_{loss}. So the time passed until temperature drops to T_{ref} is,

$$T_{ref} = T_{eq} - [T_{eq} - T_{ini}]\, e^{-\frac{K_{\text{loss}}}{C_p \rho V_{ol}} t_h} \tag{3.2.39}$$

$$t_h = \frac{C_p \rho V_{ol}}{K_{\text{loss}}} \ln \left(\frac{T_{ini} - T_{eq}}{T_{ref} - T_{eq}} \right) \quad [s] \tag{3.2.40}$$

Where T_{eq} is the Equilibrium Temperature, defined as the temperature at which, being the door open enough time, we reach a balance between heating gain and losses.

Also let us compute the Temperature T_{t_c} at a general time t_c, this will be,

$$T_{t_c} = T_{eq} - [T_{eq} - T_{ini}]\, e^{-\frac{K_{\text{loss}}}{C_p \rho V_{ol}} t_c} \tag{3.2.41}$$

3.2.7 Time $\mathbf{t_b}$ to reach the Equilibrium Temperature $\mathbf{T_{eq}}$

We will compute the time to empty all the building air volume and reach a Temperature $T(t_b) = T_{eq}$. It must be noticed that this occurs at $t_c \to \infty$, but we can assume that $T(t_b) \approx T_{eq}$ with a 5% error criteria or 2% error criteria. We can define the system velocity constant τ, as:

$$\tau = \frac{C_p \rho V_{ol}}{K_{\text{loss}}} \tag{3.2.42}$$

$$t_b^{5\%} = 3 \cdot \tau = 3 \frac{C_p \rho V_{ol}}{K_{\text{loss}}} \quad [s] \tag{3.2.43}$$

or

$$t_b^{2\%} = 4 \cdot \tau = 4 \frac{C_p \rho V_{ol}}{K_{\text{loss}}} \quad [s] \tag{3.2.44}$$

$t_b^{5\%}$ and $t_b^{2\%}$ is the time required to be at a Temperature $T(t_b^{5\%}) \approx T_{eq}$ with an error of 5% and $T(t_b^{2\%}) \approx T_{eq}$ with an error of 2%.

3.2.8 Mean Temperature $\mathbf{T_m}$

Finally, the Mean Temperature T_m is the indoor temperature integrated throughout the cycle. Let us compute it in two cases. The first case is if we start at a Temperature T_{ini} and it reaches T_{t_c}, so T_m will be:

$$T_m^{t_c} = \frac{1}{t_c} \int_0^{t_c} T(t)dt = T_{eq} + \frac{C_p \rho V_{ol}}{K_{\text{loss}} \cdot t_c} (T_{ini} - T_{eq}) \left[1 - e^{-\frac{K_{\text{loss}}}{C_p \rho V_{ol}} t_c} \right] \tag{3.2.45}$$

For the second case, let us particularise this result for $t_c = t_b$, then we will have the following result,

$$T_m^{t_b} = \frac{1}{t_b}\int_0^{t_b} T(t)dt = T_{eq} + \frac{(T_{ini} - T_{ref})}{\ln\left(\frac{T_{ini}-T_{eq}}{T_{ref}-T_{eq}}\right)} \tag{3.2.46}$$

3.2.9 Energy Loss Computation $\mathbf{E_{loss}}$

Let us calculate now the Energy Losses of our system in one door cycle. To do so, we should compute the energy loss of the door, which is

$$E_{\text{loss}} = K_{\text{loss}} \int_0^{t_c} T(t)dt = K_{\text{loss}} \cdot T_m^{t_c} \cdot t_c \quad [J] \tag{3.2.47}$$

and if we consider only until t_b, we will have,

$$E_{\text{loss}} = K_{\text{loss}} \int_0^{t_b} T(t)dt = K_{\text{loss}} \cdot T_m^{t_b} \cdot t_b \quad [J] \tag{3.2.48}$$

3.3 Thermophysical Door Cycle Problem and Iterative Model Calculation with Heating

With all the equations developed, we will now synthesize the transient simplified model for the heating season (winter). In the actual problem, where the door is open and closed and the heating it is not constantly on, we can separate the process in 5 intervals:

1. We start at an initial Temperature T_{ini}, setpoint of the Heating system, and the door is open, being the Heating off ($C_{\text{cooling}} = 0$). It drops until T_{ref} taking a time t_h. At this stage we only have infiltration effect.

2. We start at a reference Temperature T_{ref}. At this instant ($t = t_h$) the Heating is activated ($C_{\text{Heating}} \neq 0$). As the door is open, the inside Temperature drops until $T(t_c - t_h) = T_d$ taking a time $t = t_c - t_h$. The overall door open time is t_c. At this stage we only have infiltration effect.

3. We start at a Temperature $T_d = T(t_c - t_h)$ (after cycle time t_c) where the door was open. At the start of this interval the door is closed, and the Heating ($C_{\text{Heating}} \neq 0$) is trying to recover its initial Temperature T_{ini} taking a time t_q. At this stage we have transmission, permeability and long wave radiation effects.

4. Now we start at an initial Temperature T_{ini} with the door closed, and the Heating is off ($C_{\text{Heating}} = 0$) until we reach the Temperature T_{ref} taking a time t_{h2}. At this stage we have transmission, permeability and long wave radiation effects.

5. Finally the Heating ($C_{\text{Heating}} \neq 0$), with the door closed, starts recovering from the Temperature T_{ref} until the initial Temperature T_{ini} taking a time t_{q2}. At this stage we have transmission, permeability and long wave radiation effects.

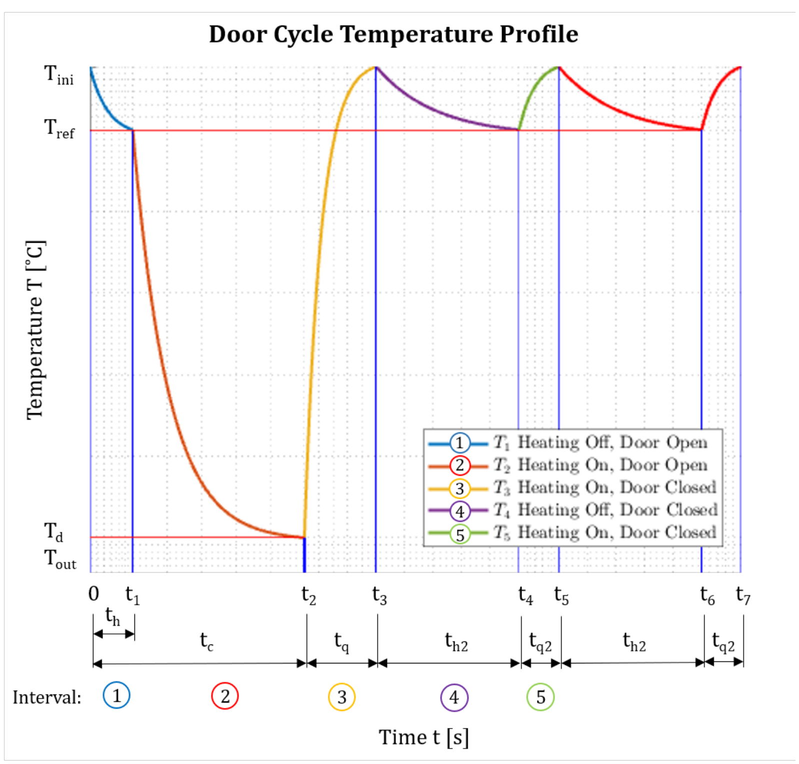

Figure 10: Schema of the characteristic Temperature Profile (not at scale)

3.3.1 1st Interval: Door Open, Heating Off

In this interval, the inside temperature will be:

$$T_1(t) = T_{out} - [T_{out} - T_{ini}]\, e^{-\frac{K_{\text{infil}}}{C_p \rho V_{ol}} t} \tag{3.3.1}$$

Losses in this interval are only related to air infiltration. The coefficient will be:

$$K_{infil} = \frac{\Phi_{tot}}{T_{ini} - T_{out}} \tag{3.3.2}$$

For the calculation of Φ_{tot} some boundary conditions are previously required. In a door, the reference height for temperature measurement is $z_0 = 0,5\,H$, and floor height is usually $z_b = 0\ m$.

The values b_i, c_i and d_i are empirical as seen in chapter 3.1.2. As reference values we can take, for the temperature profile:

$$b_1 = 0,75\ K/m$$

$$b_2 = 1\ K/m$$

And for the pressure profile:

$$c_1 = c_2 = 101325\ Pa$$

$$d_2 = 15\ Pa/m$$

Taking as seen also in 3.1.2:

$$a_1 = T_{ini} - \frac{b_1 H}{2} \tag{3.3.3}$$

$$a_2 = T_{out} - \frac{b_2 H}{2} \tag{3.3.4}$$

$$d_1 = d_2 + \frac{0,1}{H^3}$$

We can calculate then the flow as developed before:

$$\Phi_{tot} = \Phi_{12} + \Phi_{21} \tag{3.3.5}$$

$$\Phi_{12} = C_p \sqrt{\rho_1(z_0)} Re\left(\tilde{T}_{1a}(z_0) Z_a - \tilde{T}_{1b}(z_0) Z_b\right) \delta_{12}$$

$$\Phi_{21} = C_p \sqrt{\rho_2(z_0)} Im\left(\tilde{T}_{2a}(z_0) Z_a - \tilde{T}_{2b}(z_0) Z_b\right) \delta_{21}$$

$$\tilde{T}_{ia}(z_0) = T_i(z_0) - b_i H \left[\frac{C_a}{5\,C_t} + \frac{\alpha - 1}{2}\right]$$

$$\tilde{T}_{ib}(z_0) = T_i(z_0) - b_i H \left[\frac{C_a}{5\,C_t} + \frac{\alpha}{2}\right]$$

$$\alpha = \frac{z_0 - z_b}{H}$$

$$Z_a = \frac{2\sqrt{2}}{3} C_d\, W \frac{C_a^{3/2}}{C_t}$$

$$Z_b = \frac{2\sqrt{2}}{3} C_d\, W \frac{C_b^{3/2}}{C_t}$$

$$\delta_{12} = \frac{7}{8} e^{-0.3073 \cdot H_{12}}$$

$$\delta_{21} = \frac{7}{8} e^{-0.3073 \cdot H_{21}}$$

$$H_{12} = z_0 - \frac{\Delta P(z_0)}{g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right)}$$

$$H_{21} = H - H_{12}$$

$$C_a = \Delta P(z_b + H) = \Delta P(z_0) + g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right) (z_b + h - z_0)$$

$$C_b = \Delta P(z_b) = \Delta P(z_0) + g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right) (z_b - z_0)$$

$$C_t = C_a - C_b$$

$$C_d = \frac{1}{\sqrt{1,75 + 0,7 \cdot e^{(-\frac{W}{32.5 \cdot H})}}}$$

$$T_i(z) = a_i + b_i \cdot z$$

$$P_i(z) = c_i + d_i \cdot z + \frac{1}{2} \rho \cdot 0.0029 \cdot v_{wind}^2$$

$$\rho_i = \frac{K_{air}}{T_i(z)}$$

$$K_{air} = \frac{101325}{R_{air}}$$

$$R = 287.05 \, \frac{J}{kg \, K}$$

We also need:

$$t_h = \frac{C_p \rho V_{ol}}{K_{infil}} \ln \left(\frac{T_{ini} - T_{out}}{T_{ref} - T_{out}} \right) \tag{3.3.6}$$

$$T_{m1} = \frac{1}{t_c} \int_0^{t_c} T(t) dt = \begin{cases} T_{out} + \frac{C_p \rho V_{ol}}{K_{infil} \cdot t_c} (T_{ini} - T_{out}) \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} t_c} \right] & \text{if} \quad t_c < t_h \\ T_{out} + \frac{(T_{ini} - T_{ref})}{\ln \left(\frac{T_{ini} - T_{out}}{T_{ref} - T_{out}} \right)} & \text{if} \quad t_c \geq t_h \end{cases} \tag{3.3.7}$$

Finally, we can compute the heat flux:

$$E_{m1} = \begin{cases} K_{infil} \cdot (T_{m1} - T_{out}) \cdot t_c & \text{if} \quad t_c < t_h \\ K_{infil} \cdot (T_{m1} - T_{out}) \cdot t_h & \text{if} \quad t_c \geq t_h \end{cases} \tag{3.3.8}$$

For a slight better approximation, an iteration could be done with the value T_{m1} calculated, taking $a_1 = T_{m1} - b_1 \cdot h/2$ and recalculate from previous equations and obtain again Φ_{tot}, t_h and T_{m1} to calculate new K_{infil}.

$$K_{infil} = \frac{\Phi_{tot}}{T_{m_1} - T_{out}}$$

3.3.2 2nd Interval: Door Open, Heating On

The inside temperature will be:

$$T_2(t) = T_c - [T_c - T_{ref}]\, e^{-\frac{K_{infil}}{C_p \rho V_{ol}} t} \tag{3.3.9}$$

Where, as the heating system is on, the equilibrium temperature will be:

$$T_c = T_{out} + \frac{C_{\text{heating}} V_{ol} + Q_{solar}}{K_{infil}} \tag{3.3.10}$$

The infiltration losses are defined by:

$$K_{infil} = \frac{\Phi_{tot}}{T_{ref} - T_{out}} \tag{3.3.11}$$

To calculate Φ_{tot} with the same equations as in interval 1 we take:

$$a_1 = T_{ref} - \frac{b_1 H}{2} \tag{3.3.12}$$

$$a_2 = T_{out} - \frac{b_2 H}{2} \tag{3.3.13}$$

$$d_1 = d_2 + \frac{0,1}{H^3}$$

Then the time to empty all the building air volume and reach equilibrium temperature T_c is:

$$t_b^{2\%} = 4\frac{C_p \rho V_{ol}}{K_{infil}} \tag{3.3.14}$$

Then the mean temperature in the interval will be:

$$T_{m2} = \begin{cases} \text{if } T_c \leq T_{ref} \begin{cases} T_c + \dfrac{C_p \rho V_{ol} \left(T_{ref} - T_c\right)}{K_{infil} \left(t_c - t_h\right)} \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} (t_c - t_h)}\right] & \text{if } \; t_c > t_h \; and \; t_b^{2\%} > t_c - t_h \\ T_c + \dfrac{C_p \rho V_{ol} \left(T_{ref} - T_c\right)}{K_{infil} \left(t_b^{2\%}\right)} \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} (t_b^{2\%})}\right] & \text{if } \; t_c > t_h \; and \; t_b^{2\%} \leq t_c - t_h \\ \emptyset & \text{if } \; t_c \leq t_h \end{cases} \\ \text{if } T_c > T_{ref} \begin{cases} T_c + \dfrac{\left(T_{ref} - T_{ini}\right)}{\ln\left(\frac{T_{ref} - T_c}{T_{ini} - T_c}\right)} & \text{if } \; t_c > t_{in} + t_h \\ T_c + \dfrac{C_p \rho V_{ol} \left(T_{ref} - T_c\right)}{K_{infil} \left(t_c - t_h\right)} \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} (t_c - t_h)}\right] & \text{if } \; t_c \leq t_{in} + t_h \end{cases} \end{cases} \tag{3.3.15}$$

Where:

$$t_{in} = \frac{C_p \rho V_{ol}}{K_{infil}} \ln\left(\frac{T_{ref} - T_c}{T_{ini} - T_c}\right) \tag{3.3.16}$$

These several cases contemplate the power of the heating in order to be able or not to increase the inside temperature with the door open.

Finally we have the energy flux:

$$E_{m2} = \begin{cases} \text{if } T_c \le T_{ref} \begin{cases} K_{infil} \cdot (T_{m2} - T_{out}) \cdot (t_c - t_h) & \text{if} \quad t_c > t_h \textit{ and } t_b^{2\%} > t_c - t_h \\ K_{infil} \cdot (T_{m2} - T_{out}) \cdot (t_b^{2\%}) + C_{Heat} \cdot V_{ol} \cdot (t_c - t_h - t_b^{2\%}) & \text{if} \quad t_c > t_h \textit{ and } t_b^{2\%} \le t_c - t_h \\ 0 & \text{if} \quad t_c < t_h \end{cases} \\ \\ \text{if } T_c > T_{ref} \begin{cases} (E_{m1} + K_{infil} \cdot (T_{m2} - T_{out}) \cdot t_{in}) \dfrac{t_c}{t_{in} + t_h} - E_{m1} & \text{if} \quad t_c > t_{in} + t_h \text{ and } t_c > t_h \\ K_{infil} \cdot (T_{m2} - T_{out}) \cdot (t_c - t_h) & \text{if} \quad t_c \le t_{in} + t_h \text{ and } t_c > t_h \\ \emptyset & \text{if} \quad t_c < t_h \end{cases} \end{cases} \tag{3.3.17}$$

For a better approximation, always that we are far from T_{eq}, an iteration can be done recalculating with T_{m2}, computing again $a_1 = T_{m2} - b_1 \, H/2$ and recalculating from Equation (3.4.11) to (3.4.15), obtaining Φ_{tot}, and T_{m2} to calculate a new K_{infil}:

$$K_{infil} = \frac{\Phi_{tot}}{T_{m2} - T_{out}} \tag{3.3.18}$$

3.3.3 3rd Interval: Door Closed, Heating On

In this interval, the inside temperature will be:

$$T_3(t) = T_q - [T_q - T_d] \, e^{-\frac{K_{loss}}{C_v V ol \, \rho} t} \tag{3.3.19}$$

Where:

$$T_d = \begin{cases} \text{if } T_c \le T_{ref} \begin{cases} T_c - [T_c - T_{ref}] \, e^{-\frac{K_{infil}}{C_p \rho V_{ol}}(t_c - t_h)} & \text{if} \quad t_c \ge t_h \\ T_{out} - [T_{out} - T_{ini}] \, e^{-\frac{K_{infil}}{C_p \rho V_{ol}} t_c} & \text{if} \quad t_c < t_h \end{cases} \\ \\ \text{if } T_c > T_{ref} \begin{cases} T_c - [T_c - T_{ref}] \, e^{-\frac{K_{infil}}{C_p \rho V_{ol}}(t_c - t_h)} & \text{if} \quad t_c \le t_h + t_{in} \\ \dfrac{T_{m1} \cdot t_h + T_{m2} \cdot t_{in}}{t_h + t_{in}} & \text{if} \quad t_c > t_h + t_{in} \end{cases} \end{cases} \tag{3.3.20}$$

Where again:

$$t_{in} = \frac{C_p \rho V_{ol}}{K_{infil}} \ln \left(\frac{T_{ref} - T_c}{T_{ini} - T_c} \right)$$

And also:

$$T_c = T_{out} + \frac{C_{\text{heating}} V_{ol}}{K_{infil}}$$

We can assume as an approximation for calculation that the mean temperature is:

$$T_{m3} \approx \frac{T_{ini} + T_d}{2} \tag{3.3.21}$$

As the door is closed, we will have losses due to transmission, leakage and radiation:

$$K_{loss} = K_{\text{trans}} + K_{\text{perm air}} + K_{long\,wave} \tag{3.3.22}$$

$$K_{loss} = A \cdot U \left(1 + 0,734 \cdot \sigma \cdot \varepsilon \cdot T_{out}^3\right) + C_p \rho \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_R}\right)^{2/3} \tag{3.3.23}$$

where,

$$P_w = \frac{1}{2} \rho \cdot v_{wind}^2 \tag{3.3.24}$$

$$P_s = \frac{\rho g (H_b - H_{np})(T_{m3} - T_{out})}{T_{out}} \tag{3.3.25}$$

And with K_{loss} we can calculate:

$$T_q = T_{out} + \frac{C_{\text{Heating}} V_{ol}}{K_{loss}} \tag{3.3.26}$$

$$t_q = \frac{C_v V_{ol}\, \rho}{K} \ln\left(\frac{T_d - T_q}{T_{ini} - T_q}\right) \tag{3.3.27}$$

$$T_{m3} = T_q + \frac{(T_d - T_{ini})}{\ln\left(\frac{T_d - T_q}{T_{ini} - T_q}\right)} \tag{3.3.28}$$

Finally we calculate the energy flux:

$$E_{m3} = K_{loss} \cdot (T_{m3} - T_{out}) \cdot t_q \tag{3.3.29}$$

For a slight better approximation, we can recalculate a new iteration with T_{m3}, computing again from Equation (3.4.23) to (3.4.28) and obtain the new values for T_{m3} and t_q.

3.3.4 4th Interval: Door Closed, Heating Off

The mean temperature in the interval will be:

$$T_4(t) = T_{out} - \left[T_{out} - T_{ini}\right] e^{-\frac{K_{loss}}{C_v V ol\, \rho} t} \tag{3.3.30}$$

We also initially assume that:

$$T_{m4} \approx \frac{T_{ini} + T_{ref}}{2} \tag{3.3.31}$$

As the door is closed we have like in interval 3:

$$K_{loss} = K_{\text{trans}} + K_{\text{perm air}} + K_{long\, wave} \tag{3.3.32}$$

$$K_{loss} = A \cdot U \left(1 + 0,734 \cdot \sigma \cdot \varepsilon \cdot T_{out}^3\right) + C_p \rho \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_R}\right)^{2/3} \tag{3.3.33}$$

where,

$$P_w = \frac{1}{2} \rho v_{wind}^2 \tag{3.3.34}$$

$$P_s = \frac{\rho g (H_b - H_{np})(T_{m4} - T_{out})}{T_{out}} \tag{3.3.35}$$

Then we have:

$$t_{h2} = \frac{C_v V_{ol}\,\rho}{K_{loss}} \ln\left(\frac{T_{ini} - T_{out}}{T_{ref} - T_{out}}\right)$$

$$T_{m4} = T_{out} + \frac{(T_{ini} - T_{ref})}{\ln\left(\frac{T_{ini} - T_{out}}{T_{ref} - T_{out}}\right)} \tag{3.3.36}$$

Finally we have the value of the energy flux:

$$E_{m4} = K_{loss} \cdot (T_{m4} - T_{out}) \cdot t_{h2} \tag{3.3.37}$$

Like in previous intervals, a better approximation can be achieved recalculating with T_{m4} again from Equation (3.4.33) to (3.4.36) and obtaining obtain the new values of T_{m4} and t_{h2}.

3.3.5 5th Interval: Door Closed, Heating On

The inside temperature in the last interval will be:

$$T_5(t) = T_q - [T_q - T_{ref}]\, e^{-\frac{K_{loss}}{C_v V_{ol}\,\rho} t} \tag{3.3.38}$$

where,

$$T_q = T_{out} + \frac{C_{Heating} V_{ol}}{K_{loss}} \tag{3.3.39}$$

In an analogue way like the previous intervals we assume:

$$T_{m5} \approx \frac{T_{ini} + T_{ref}}{2} \tag{3.3.40}$$

The losses will be:

$$K_{loss} = A \cdot U \left(1 + 0,734 \cdot \sigma \cdot \varepsilon \cdot T_{out}^3\right) + C_p \rho \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_R}\right)^{2/3} \tag{3.3.41}$$

where,

$$P_w = \frac{1}{2} \rho v_{wind}^2 \tag{3.3.42}$$

$$P_s = \frac{\rho g (H_b - H_{np})(T_{m5} - T_{out})}{T_{out}} \tag{3.3.43}$$

Then we calculate:

$$t_{q2} = \frac{C_v V_{ol}\,\rho}{K_{loss}} \ln\left(\frac{T_{ref} - T_q}{T_{ini} - T_q}\right) \tag{3.3.44}$$

$$T_{m5} = T_q + \frac{(T_{ref} - T_{ini})}{\ln\left(\frac{T_{ref} - T_q}{T_{ini} - T_q}\right)} \tag{3.3.45}$$

Finally we can calculate the energy flux:

$$E_{m5} = K_{loss} \cdot (T_{m5} - T_{out}) \cdot t_{q2} \tag{3.3.46}$$

As an additional iteration for a better approximation, we can recalculate with T_{m5} again from Equation (3.4.41) to (3.4.45) and obtain the new T_{m5} and t_{q2}.

3.3.6 Energy Contribution in time

Once that we have calculated the energy flux, we can compute the net energy loss along a period of time. We consider one year as a reference.

If we name $N_{heating}$ the number of heating days per year and N_{cycles} the number of opening and closing cycles per year, the total energy along during the whole year would be:

$$\begin{aligned} E_{total\,heating} = &(E_{m1} + E_{m2} + E_{m3}) \cdot N_{cycles} \\ &+ (E_{m4} + E_{m5}) \cdot \frac{1}{t_{h2} + t_{q2}} \cdot (N_{heating} \cdot 24 \cdot 3600 - (t_c + t_q) \cdot N_{cycles}) \end{aligned} \tag{3.3.47}$$

This calculation was made on a 365/24 basis. If we consider that the heating is working a number of days per week N_{week} with a working day time t_{work} in hours, we would have:

$$\begin{aligned} E_{total\,heating} = &(E_{m1} + E_{m2} + E_{m3}) \cdot N_{cycles} \cdot \frac{N_{week}}{7} \cdot \frac{N_{heating}}{365} \\ &+ (E_{m4} + E_{m5}) \cdot \frac{1}{t_{h2} + t_{q2}} \cdot \left(N_{heating} \cdot t_{work} \cdot 3600 - N_{cycles} \cdot \frac{N_{heating}}{365} \cdot (t_c + t_q) \right) \cdot \frac{N_{week}}{7} \end{aligned} \tag{3.3.48}$$

For the calculation of the weight of the single effects, the single loss coefficient K_{trans}, K_{perm}, $K_{longwave}$ and K_{infil} should be applied in the expressions to calculate E_{mi}.

3.4 Thermophysical Door Cycle Problem and Iterative Model Calculation with Cooling

Following the same procedure, we can derive the thermophysical door cycle problem that follow the same equations as those of the presented before (heating season) but for the cooling season (summer). In this case, the temperature diagram is a mirror image with respect to the horizontal axis of the one represented in figure 10.

3.4.1 1st Interval: Door Open, Cooling Off

In this interval, the inside temperature will be:

$$T_1(t) = T_{out} - [T_{out} - T_{ini}]\, e^{-\frac{K_{\text{infil}}}{C_p \rho V_{ol}} t} \tag{3.4.1}$$

Losses in this interval are only related to air infiltration. The coefficient will be:

$$K_{infil} = \frac{\Phi_{tot}}{T_{out} - T_{ini}} \tag{3.4.2}$$

For the calculation of Φ_{tot} some boundary conditions are previously required. In a door, the reference height for temperature measurement is $z_0 = 0,5\,H$, and floor height is usually $z_b = 0\ m$.

The values b_i, c_i and d_i are empirical as seen in chapter 3.1.2. As reference values we can take, for the temperature profile:

$$b_1 = 0,75\ K/m$$

$$b_2 = 1\ K/m$$

And for the pressure profile:

$$c_1 = c_2 = 101325\ Pa$$

$$d_2 = 15\ Pa/m$$

Taking as seen also in 3.1.2:

$$a_1 = T_{out} - \frac{b_1 H}{2} \tag{3.4.3}$$

$$a_2 = T_{ini} - \frac{b_2 H}{2} \tag{3.4.4}$$

$$d_1 = d_2 + \frac{0,1}{H^3}$$

We can calculate then the flow as developed before:

$$\Phi_{tot} = \Phi_{12} + \Phi_{21} \tag{3.4.5}$$

$$\Phi_{12} = C_p \sqrt{\rho_1(z_0)} Re\left(\tilde{T}_{1a}(z_0) Z_a - \tilde{T}_{1b}(z_0) Z_b\right) \delta_{12}$$

$$\Phi_{21} = C_p \sqrt{\rho_2(z_0)} Im\left(\tilde{T}_{2a}(z_0) Z_a - \tilde{T}_{2b}(z_0) Z_b\right) \delta_{21}$$

$$\tilde{T}_{ia}(z_0) = T_i(z_0) - b_i H \left[\frac{C_a}{5\,C_t} + \frac{\alpha - 1}{2}\right]$$

$$\tilde{T}_{ib}(z_0) = T_i(z_0) - b_i H \left[\frac{C_a}{5\,C_t} + \frac{\alpha}{2}\right]$$

$$\alpha = \frac{z_0 - z_b}{H}$$

$$Z_a = \frac{2\sqrt{2}}{3}\, C_d\, W \frac{C_a^{3/2}}{C_t}$$

$$Z_b = \frac{2\sqrt{2}}{3}\, C_d\, W \frac{C_b^{3/2}}{C_t}$$

$$\delta_{12} = \frac{7}{8} e^{-0.3073 \cdot H_{12}}$$

$$\delta_{21} = \frac{7}{8} e^{-0.3073 \cdot H_{21}}$$

$$H_{12} = z_0 - \frac{\Delta P(z_0)}{g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right)}$$

$$H_{21} = H - H_{12}$$

$$C_a = \Delta P(z_b + H) = \Delta P(z_0) + g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right) (z_b + h - z_0)$$

$$C_b = \Delta P(z_b) = \Delta P(z_0) + g K_{air} \left(\frac{1}{T_2(z_0)} - \frac{1}{T_1(z_0)} \right) (z_b - z_0)$$

$$C_t = C_a - C_b$$

$$C_d = \frac{1}{\sqrt{1,75 + 0,7 \cdot e^{(-\frac{W}{32.5 \cdot H})}}}$$

$$T_i(z) = a_i + b_i \cdot z$$

$$P_i(z) = c_i + d_i \cdot z + \frac{1}{2} \rho \cdot 0.0029 \cdot v_{wind}^2$$

$$\rho_i = \frac{K_{air}}{T_i(z)}$$

$$K_{air} = \frac{101325}{R_{air}}$$

$$R = 287.05 \, \frac{J}{kg\, K}$$

We also need:

$$t_h = \frac{C_p \rho V_{ol}}{K_{infil}} \ln \left(\frac{T_{ini} - T_{out}}{T_{ref} - T_{out}} \right) \tag{3.4.6}$$

$$T_{m1} = \frac{1}{t_c} \int_0^{t_c} T(t) dt = \begin{cases} T_{out} + \frac{C_p \rho V_{ol}}{K_{infil} \cdot t_c} (T_{ini} - T_{out}) \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} t_c} \right] & \text{if} \quad t_c < t_h \\ T_{out} + \frac{(T_{ini} - T_{ref})}{\ln \left(\frac{T_{ini} - T_{out}}{T_{ref} - T_{out}} \right)} & \text{if} \quad t_c \geq t_h \end{cases} \tag{3.4.7}$$

Finally, we can compute the heat flux:

$$E_{m1} = \begin{cases} K_{infil} \cdot (T_{m1} - T_{out}) \cdot t_c & \text{if} \quad t_c < t_h \\ K_{infil} \cdot (T_{m1} - T_{out}) \cdot t_h & \text{if} \quad t_c \geq t_h \end{cases} \tag{3.4.8}$$

For a slight better approximation, an iteration could be done with the value T_{m1} calculated, taking $a_1 = T_{m1} - b_1 \cdot h/2$ and recalculate from previous equations and obtain again Φ_{tot}, t_h and T_{m1} to calculate new K_{infil}.

$$K_{infil} = \frac{\Phi_{tot}}{T_{out} - T_{m_1}}$$

3.4.2 2nd Interval: Door Open, Cooling On

The inside temperature will be:

$$T_2(t) = T_c - [T_c - T_{ref}]\, e^{-\frac{K_{infil}}{C_p \rho V_{ol}} t} \tag{3.4.9}$$

Where, as the cooling system is on, the equilibrium temperature will be:

$$T_c = T_{out} + \frac{C_{\text{cooling}} V_{ol} + Q_{solar}}{K_{infil}} \tag{3.4.10}$$

The infiltration losses are defined by:

$$K_{infil} = \frac{\Phi_{tot}}{T_{out} - T_{ref}} \tag{3.4.11}$$

To calculate Φ_{tot} with the same equations as in interval 1 we take:

$$a_1 = T_{out} - \frac{b_1 H}{2} \tag{3.4.12}$$

$$a_2 = T_{ref} - \frac{b_2 H}{2} \tag{3.4.13}$$

$$d_1 = d_2 + \frac{0,1}{H^3}$$

Then the time to empty all the building air volume and reach equilibrium temperature T_c is:

$$t_b^{2\%} = 4 \frac{C_p \rho V_{ol}}{K_{infil}} \tag{3.4.14}$$

Then the mean temperature in the interval will be:

$$T_{m2} = \begin{cases} \text{if } T_c \leq T_{ref} \begin{cases} T_c + \dfrac{C_p \rho V_{ol} (T_{ref} - T_c)}{K_{infil} (t_c - t_h)} \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} (t_c - t_h)}\right] & \text{if } \; t_c > t_h \; and \; t_b^{2\%} > t_c - t_h \\ T_c + \dfrac{C_p \rho V_{ol} (T_{ref} - T_c)}{K_{infil} (t_b^{2\%})} \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} (t_b^{2\%})}\right] & \text{if } \; t_c > t_h \; and \; t_b^{2\%} \leq t_c - t_h \\ \emptyset & \text{if } \; t_c \leq t_h \end{cases} \\ \\ \text{if } T_c > T_{ref} \begin{cases} T_c + \dfrac{(T_{ref} - T_{ini})}{\ln\left(\frac{T_{ref} - T_c}{T_{ini} - T_c}\right)} & \text{if } \; t_c > t_{in} + t_h \\ T_c + \dfrac{C_p \rho V_{ol} (T_{ref} - T_c)}{K_{infil} (t_c - t_h)} \left[1 - e^{-\frac{K_{infil}}{C_p \rho V_{ol}} (t_c - t_h)}\right] & \text{if } \; t_c \leq t_{in} + t_h \end{cases} \end{cases} \tag{3.4.15}$$

Where:

$$t_{in} = \frac{C_p \rho V_{ol}}{K_{infil}} \ln\left(\frac{T_{ref} - T_c}{T_{ini} - T_c}\right) \tag{3.4.16}$$

These several cases contemplate the power of the cooling in order to be able or not to increase the inside temperature with the door open.

Finally we have the energy flux:

$$E_{m2} = \begin{cases} \text{if } T_c \leq T_{ref} \begin{cases} K_{infil} \cdot (T_{m2} - T_{out}) \cdot (t_c - t_h) & \text{if} \quad t_c > t_h \textit{ and } t_b^{2\%} > t_c - t_h \\ K_{infil} \cdot (T_{m2} - T_{out}) \cdot (t_b^{2\%}) + C_{Heat} \cdot V_{ol} \cdot (t_c - t_h - t_b^{2\%}) & \text{if} \quad t_c > t_h \textit{ and } t_b^{2\%} \leq t_c - t_h \\ 0 & \text{if} \quad t_c < t_h \end{cases} \\ \\ \text{if } T_c > T_{ref} \begin{cases} (E_{m1} + K_{infil} \cdot (T_{m2} - T_{out}) \cdot t_{in}) \dfrac{t_c}{t_{in} + t_h} - E_{m1} & \text{if} \quad t_c > t_{in} + t_h \text{ and } t_c > t_h \\ K_{infil} \cdot (T_{m2} - T_{out}) \cdot (t_c - t_h) & \text{if} \quad t_c \leq t_{in} + t_h \text{ and } t_c > t_h \\ \emptyset & \text{if} \quad t_c < t_h \end{cases} \end{cases} \tag{3.4.17}$$

For a better approximation, always that we are far from T_{eq}, an iteration can be done recalculating with T_{m2}, computing again $a_1 = T_{m2} - b_1\, H/2$ and recalculating from Equation (3.4.11) to (3.4.15), obtaining Φ_{tot}, and T_{m2} to calculate a new K_{infil}:

$$K_{infil} = \frac{\Phi_{tot}}{T_{out} - T_{m2}} \tag{3.4.18}$$

3.4.3 3rd Interval: Door Closed, Cooling On

In this interval, the inside temperature will be:

$$T_3(t) = T_q - [T_q - T_d]\, e^{-\frac{K_{loss}}{C_v Vol\, \rho} t} \tag{3.4.19}$$

Where:

$$T_d = \begin{cases} \text{if } T_c \leq T_{ref} \begin{cases} T_c - [T_c - T_{ref}]\, e^{-\frac{K_{infil}}{C_p \rho V_{ol}}(t_c - t_h)} & \text{if} \quad t_c \geq t_h \\ T_{out} - [T_{out} - T_{ini}]\, e^{-\frac{K_{infil}}{C_p \rho V_{ol}} t_c} & \text{if} \quad t_c < t_h \end{cases} \\ \\ \text{if } T_c > T_{ref} \begin{cases} T_c - [T_c - T_{ref}]\, e^{-\frac{K_{infil}}{C_p \rho V_{ol}}(t_c - t_h)} & \text{if} \quad t_c \leq t_h + t_{in} \\ \dfrac{T_{m1} \cdot t_h + T_{m2} \cdot t_{in}}{t_h + t_{in}} & \text{if} \quad t_c > t_h + t_{in} \end{cases} \end{cases} \tag{3.4.20}$$

Where again:

$$t_{in} = \frac{C_p \rho V_{ol}}{K_{infil}} \ln\left(\frac{T_{ref} - T_c}{T_{ini} - T_c}\right)$$

And also:

$$T_c = T_{out} + \frac{C_{\text{cooling}} V_{ol}}{K_{infil}}$$

We can assume as an approximation for calculation that the mean temperature is:

$$T_{m3} \approx \frac{T_{ini} + T_d}{2} \tag{3.4.21}$$

As the door is closed, we will have losses due to transmission, leakage and radiation:

$$K_{loss} = K_{\text{trans}} + K_{\text{perm air}} + K_{long\,wave} \tag{3.4.22}$$

$$K_{loss} = A \cdot U \left(1 + 0{,}734 \cdot \sigma \cdot \varepsilon \cdot T_{out}^3\right) + C_p \rho \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_R}\right)^{2/3} \tag{3.4.23}$$

where,

$$P_w = \frac{1}{2} \rho \cdot v_{wind}^2 \tag{3.4.24}$$

$$P_s = \frac{\rho g (H_b - H_{np})(T_{m3} - T_{out})}{T_{out}} \tag{3.4.25}$$

And with K_{loss} we can calculate:

$$T_q = T_{out} + \frac{C_{\text{cooling}} V_{ol}}{K_{loss}} \tag{3.4.26}$$

$$t_q = \frac{C_v V_{ol}\, \rho}{K} \ln\left(\frac{T_d - T_q}{T_{ini} - T_q}\right) \tag{3.4.27}$$

$$T_{m3} = T_q + \frac{(T_d - T_{ini})}{\ln\left(\frac{T_d - T_q}{T_{ini} - T_q}\right)} \tag{3.4.28}$$

Finally we calculate the energy flux:

$$E_{m3} = K_{loss} \cdot (T_{m3} - T_{out}) \cdot t_q \tag{3.4.29}$$

For a slight better approximation, we can recalculate a new iteration with T_{m3}, computing again from Equation (3.4.23) to (3.4.28) and obtain the new values for T_{m3} and t_q.

3.4.4 4th Interval: Door Closed, Cooling Off

The mean temperature in the interval will be:

$$T_4(t) = T_{out} - \left[T_{out} - T_{ini}\right] e^{-\frac{K_{loss}}{C_v V_{ol}\, \rho} t} \tag{3.4.30}$$

We also initially assume that:

$$T_{m4} \approx \frac{T_{ini} + T_{ref}}{2} \tag{3.4.31}$$

As the door is closed we have like in interval 3:

$$K_{loss} = K_{\text{trans}} + K_{\text{perm air}} + K_{long\,wave} \tag{3.4.32}$$

$$K_{loss} = A \cdot U \left(1 + 0{,}734 \cdot \sigma \cdot \varepsilon \cdot T_{out}^3\right) + C_p \rho \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_R}\right)^{2/3} \tag{3.4.33}$$

where,

$$P_w = \frac{1}{2}\rho v_{wind}^2 \tag{3.4.34}$$

$$P_s = \frac{\rho g(H_b - H_{np})(T_{m4} - T_{out})}{T_{out}} \tag{3.4.35}$$

Then we have:

$$t_{h2} = \frac{C_v V_{ol}\,\rho}{K_{loss}} \ln\left(\frac{T_{ini} - T_{out}}{T_{ref} - T_{out}}\right)$$

$$T_{m4} = T_{out} + \frac{(T_{ini} - T_{ref})}{\ln\left(\frac{T_{ini}-T_{out}}{T_{ref}-T_{out}}\right)} \tag{3.4.36}$$

Finally we have the value of the energy flux:

$$E_{m4} = K_{loss} \cdot (T_{m4} - T_{out}) \cdot t_{h2} \tag{3.4.37}$$

Like in previous intervals, a better approximation can be achieved recalculating with T_{m4} again from Equation (3.4.33) to (3.4.36) and obtaining obtain the new values of T_{m4} and t_{h2}.

3.4.5 5th Interval: Door Closed, Cooling On

The inside temperature in the last interval will be:

$$T_5(t) = T_q - [T_q - T_{ref}]\, e^{-\frac{K_{loss}}{C_v V_{ol}\,\rho} t} \tag{3.4.38}$$

where,

$$T_q = T_{out} + \frac{C_{Cooling} V_{ol}}{K_{loss}} \tag{3.4.39}$$

In an analogue way like the previous intervals we assume:

$$T_{m5} \approx \frac{T_{ini} + T_{ref}}{2} \tag{3.4.40}$$

The losses will be:

$$K_{loss} = A \cdot U \left(1 + 0,734 \cdot \sigma \cdot \varepsilon \cdot T_{out}^3\right) + C_p \rho \frac{A \cdot L_R}{3600} \left(\frac{P_w + P_s}{P_R}\right)^{2/3} \tag{3.4.41}$$

where,

$$P_w = \frac{1}{2}\rho v_{wind}^2 \tag{3.4.42}$$

$$P_s = \frac{\rho g(H_b - H_{np})(T_{m5} - T_{out})}{T_{out}} \tag{3.4.43}$$

Then we calculate:

$$t_{q2} = \frac{C_v V_{ol}\,\rho}{K_{loss}} \ln\left(\frac{T_{ref} - T_q}{T_{ini} - T_q}\right) \tag{3.4.44}$$

$$T_{m5} = T_q + \frac{(T_{ref} - T_{ini})}{\ln\left(\frac{T_{ref}-T_q}{T_{ini}-T_q}\right)} \tag{3.4.45}$$

Finally we can calculate the energy flux:

$$E_{m5} = K_{loss} \cdot (T_{m5} - T_{out}) \cdot t_{q2} \tag{3.4.46}$$

As an additional iteration for a better approximation, we can recalculate with T_{m5} again from Equation (3.4.41) to (3.4.45) and obtain the new T_{m5} and t_{q2}.

3.4.6 Energy Contribution in time

Once that we have calculated the energy flux, we can compute the net energy loss along a period of time. We consider one year as a reference.

If we name $N_{Cooling}$ the number of cooling days per year and N_{cycles} the number of opening and closing cycles per year, the total energy along during the whole year would be:

$$\begin{aligned} E_{total\,cooling} =& (E_{m1} + E_{m2} + E_{m3}) \cdot N_{cycles} \\ &+ (E_{m4} + E_{m5}) \cdot \frac{1}{t_{h2} + t_{q2}} \cdot (N_{cooling} \cdot 24 \cdot 3600 - (t_c + t_q) \cdot N_{cycles}) \end{aligned} \tag{3.4.47}$$

This calculation was made on a 365/24 basis. If we consider that the cooling is working a number of days per week N_{week} with a working day time t_{work} in hours, we would have:

$$\begin{aligned} E_{total\,cooling} =& (E_{m1} + E_{m2} + E_{m3}) \cdot N_{cycles} \cdot \frac{N_{week}}{7} \cdot \frac{N_{cooling}}{365} \\ &+ (E_{m4} + E_{m5}) \cdot \frac{1}{t_{h2} + t_{q2}} \cdot \left(N_{cooling} \cdot t_{work} \cdot 3600 - N_{cycles} \cdot \frac{N_{cooling}}{365} \cdot (t_c + t_q)\right) \cdot \frac{N_{week}}{7} \end{aligned} \tag{3.4.48}$$

For the calculation of the weight of the single effects, the single loss coefficient K_{trans}, K_{perm}, $K_{longwave}$ and K_{infil} should be applied in the expressions to calculate E_{mi}.

4 Calculation example

In this section we will calculate a door example to verify the results of the simplified calculation framework.

To have a reference for comparison, we will use the example in the technical report of the CEN [**19**]. Also, we have added the effect of cooling in summer. Which follow the same equations as those of the presented on the heating and cooling model.

4.1 Reference data

According to the data of the example case in [**19**] we have the following input data:

4.1.1 Location and climate

Location: Paris
Outside temperature in heating season: T_{out} = 10 ºC
Door direction: West-Southwest
Correction factor according to wind direction: C_w = 34,7
Average wind velocity in heating season: v_{met} = 5,0 m/s
Net wind velocity: $v_{wind} = v_{met} \cdot C_w/100 = 5,0 \cdot 34,7/100 = 1,735$
Heating days per year: $N_{heating}$ = 243 days
Outside temperature in cooling season: T_{out} = 28 ºC
Average wind velocity in cooling season: v_{met} = 5,0 m/s
Net wind velocity: $v_{wind} = v_{met} \cdot C_w/100 = 5,0 \cdot 34,7/100 = 1,735$
Cooling days per year: $N_{cooling}$ = 120 days
Solar irradiance in heating season: I_{sh} = 73,6 $kWh/(m^2 \cdot year)$
Peak solar hours per day in heating season: tps_h = 4,53 h/day
Solar irradiance per in cooling season: Is_c = 92,1 $kWh/(m^2 \cdot year)$
Peak solar hours per day in cooling season: tps_h = 8.25 h/day

4.1.2 Building and door

Building height: H_b = 8 m
Building volume: V = 1600 m^3, 8000 m^3, 16000 m^3
Heating power: C_{heat} = 20 W/m^3
Cooling power: C_{cool} = 20 W/m^3
Door height x Door width: H x W = 3 m x 3 m, 4 m x 4 m
Window area: A_g = 1 m^2
Thermal transmittance: U = 1,5 W/m^2K
Air permeability at P_R: L_R = 12 m^3/m^2h
Reference pressure for air permeability: P_R = 50 Pa
Emissivity: ε = 0,9

4.1.3 Building intended use

Inside setpoint temperature in heating season: T_{ini} = 18 ºC
Inside setpoint temperature in cooling season: T_{ini} = 24 ºC
Door cycle time: t_c = 300 s, 120 s, 30 s
Door cycles per year: n = 1000
Working days per week: N_{week} = 5 days
Workday time: t_{work} = 24 hours

4.2 Results

We show the energy losses due to each effect in kWh and % for the different building sizes, door sizes and opening times as found in [**19**], adding also a short cycle case:

$\mathbf{t}_c = 300s$	$A = 3 \times 3$			$A = 4 \times 4$		
	1600 m^3	8000 m^3	16000 m^3	1600 m^3	8000 m^3	16000 m^3
Heating						
Heat Transmission $[kWh]$	414,69	417,15	417,15	736,92	741,38	741,60
Long Wave Radiation $[kWh]$	352,71	354,80	354,80	626,79	630,58	630,76
Air Leakage $[kWh]$	294,18	295,63	295,63	522,79	525,53	525,57
Air Infiltration $[kWh]$	3121,64	6331,81	6400,18	3179,49	8009,16	8411,47
Total [**kWh**]	4182,42	7399,21	7467,58	5064,57	9906,01	10309,07
Cooling						
Heat Transmission $[kWh]$	95,40	95,62	95,62	169,56	169,99	169,99
Long Wave Radiation $[kWh]$	97,62	97,84	97,84	173,51	173,95	173,95
Air Leakage $[kWh]$	51,16	51,26	51,26	90,94	91,13	91,13
Air Infiltration $[kWh]$	1020,61	2066,15	2109,19	1043,35	2693,53	2738,87
Total [**kWh**]	1264,67	2310,82	2353,86	1477,16	3128,50	3173,84
Total						
Heat Transmission $[kWh]$	510,08	512,77	512,77	906,48	911,37	911,58
Long Wave Radiation $[kWh]$	450,33	452,65	452,65	800,30	804,53	804,71
Air Leakage $[kWh]$	345,34	346,89	346,89	613,73	616,66	616,70
Air Infiltration $[kWh]$	4142,24	8397,95	8509,37	4222,84	10702,69	11150,34
Total [**kWh**]	5447,09	9710,03	9821,45	6541,72	13034,51	13482,91

Table 4: Energy losses in kWH, $t_c = 300$ s

$\mathbf{t}_c = 300s$	$A = 3 \times 3$			$A = 4 \times 4$		
	1600 m^3	8000 m^3	16000 m^3	1600 m^3	8000 m^3	16000 m^3
Heating						
Heat Transmission [%]	9,91	5,64	5,59	14,55	7,48	7,19
Long Wave Radiation [%]	8,43	4,80	4,75	12,38	6,37	6,12
Air Leakage [%]	7,03	4,00	3,96	10,32	5,31	5,10
Air Infiltration [%]	74,64	85,57	85,71	62,78	80,85	81,59
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00
Cooling						
Heat Transmission [%]	7,54	4,14	4,06	11,48	5,43	5,36
Long Wave Radiation [%]	7,72	4,23	4,16	11,75	5,56	5,48
Air leakage [%]	4,05	2,22	2,18	6,16	2,91	2,87
Air Infiltration [%]	80,70	89,41	89,61	70,63	86,10	86,30
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00
Total						
Heat Transmission [%]	9,36	5,28	5,22	13,86	6,99	6,76
Long Wave Radiation [%]	8,27	4,66	4,61	12,23	6,17	5,97
Air leakage [%]	6,34	3,57	3,53	9,38	4,73	4,57
Air Infiltration [%]	76,05	86,49	86,64	64,55	82,11	82,70
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00

Table 5: Energy losses in %, $t_c = 300$ s

$\mathbf{t}_c = 120s$	$A = 3 \times 3$			$A = 4 \times 4$		
	1600 m^3	8000 m^3	16000 m^3	1600 m^3	8000 m^3	16000 m^3
Heating						
Heat Transmission $[kWh]$	418,00	419,55	419,55	741,94	745,83	745,87
Long Wave Radiation $[kWh]$	355,53	356,85	356,85	631,06	634,36	634,40
Air Leakage $[kWh]$	296,47	297,34	297,34	526,30	528,61	528,60
Air Infiltration $[kWh]$	1985,86	2568,33	2568,59	2320,99	3344,86	3415,82
Total [**kWh**] 5324,36	2526,04	3055,12	3641,88	3642,14	4218,90	5253,22
Cooling						
Heat Transmission $[kWh]$	95,98	96,17	96,17	170,55	170,97	170,97
Long Wave Radiation $[kWh]$	98,21	98,41	98,41	174,52	174,95	174,95
Air Leakage $[kWh]$	51,47	51,56	51,56	91,47	91,65	91,65
Air Infiltration $[kWh]$	625,91	852,07	852,07	670,19	1104,27	1104,28
Total [**kWh**]	871,46	1098,15	1098,15	1106,52	1541,76	1541,76
Total						
Heat Transmission $[kWh]$	513,97	515,72	515,72	912,49	916,80	916,84
Long Wave Radiation $[kWh]$	453,74	455,26	455,26	805,58	809,31	809,35
Air Leakage $[kWh]$	347,94	348,89	348,89	617,77	620,26	620,25
Air Infiltration $[kWh]$	2611,77	3420,40	3420,66	2991,18	4449,14	4520,09
Total [**kWh**]	3926,58	4740,04	4740,30	5325,42	6794,98	6866,12

Table 6: Energy losses in kWh, $t_c = 120$ s

$\mathbf{t}_c = 120s$	$A = 3 \times 3$			$A = 4 \times 4$		
	1600 m^3	8000 m^3	16000 m^3	1600 m^3	8000 m^3	16000 m^3
Heating						
Heat Transmission [%]	13,68	11,52	11,52	17,59	14,20	14,01
Long Wave Radiation [%]	11,64	9,80	9,80	14,96	12,08	11,92
Air Leakage [%]	9,70	8,16	8,16	12,47	10,06	9,93
Air Infiltration [%]	65,00	70,52	70,52	55,01	63,67	64,15
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00
Cooling						
Heat Transmission [%]	11,01	8,76	8,76	15,41	11,09	11,09
Long Wave Radiation [%]	11,27	8,96	8,96	15,77	11,35	11,35
Air leakage [%]	5,91	4,69	4,69	8,27	5,94	5,94
Air Infiltration [%]	71,82	77,59	77,59	60,57	71,62	71,62
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00
Total						
Heat Transmission [%]	13,09	10,88	10,88	17,13	13,49	13,35
Long Wave Radiation [%]	11,56	9,60	9,60	15,13	11,91	11,79
Air leakage [%]	8,86	7,36	7,36	11,60	9,13	9,03
Air Infiltration [%]	66,52	72,16	72,16	56,17	65,48	65,83
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00

Table 7: Energy losses in %, $t_c = 120$ s

$\mathbf{t}_c = 30s$	$A = 3 \times 3$			$A = 4 \times 4$		
	1600 m^3	8000 m^3	16000 m^3	1600 m^3	8000 m^3	16000 m^3
Heating						
Heat Transmission [kWh]	420,63	420,75	420,75	747,56	748,00	748,01
Long Wave Radiation [kWh]	357,77	357,87	357,87	635,84	636,21	636,22
Air Leakage [kWh]	298,17	298,19	298,19	529,97	530,12	530,11
Air Infiltration [kWh]	624,75	645,67	648,48	802,50	853,46	853,95
Total [**kWh**]	1700,96	1722,30	1725,11	2715,07	2767,44	2767,96
Cooling						
Heat Transmission [kWh]	96,43	96,45	96,44	171,39	171,46	171,46
Long Wave Radiation [kWh]	98,68	98,69	98,69	175,38	175,45	175,45
Air Leakage [kWh]	51,70	51,70	51,70	91,89	91,92	91,92
Air Infiltration [kWh]	207,51	216,48	216,48	262,27	279,66	282,51
Total [**kWh**]	454,24	463,27	463,27	700,80	718,40	721,25
Total						
Heat Transmission [kWh]	517,06	517,20	517,20	918,96	919,46	919,47
Long Wave Radiation [kWh]	456,44	456,56	456,56	811,22	811,66	811,67
Air Leakage [kWh]	349,87	349,89	349,89	621,86	622,04	622,03
Air Infiltration [kWh]	832,25	862,16	864,96	1064,77	1133,12	1136,46
Total [**kWh**]	2155,20	2185,58	2188,38	3415,86	3485,84	3489,21

Table 8: Energy losses in kWh, t_c = 30 s

$\mathbf{t}_c = 30s$	$A = 3 \times 3$			$A = 4 \times 4$		
	1600 m^3	8000 m^3	16000 m^3	1600 m^3	8000 m^3	16000 m^3
Heating						
Heat Transmission [%]	24,73	24,43	24,39	27,53	27,03	27,02
Long Wave Radiation [%]	21,03	20,78	20,74	23,42	22,99	22,98
Air Leakage [%]	17,53	17,31	17,29	19,52	19,16	19,15
Air Infiltration [%]	36,73	37,49	37,59	29,56	30,84	30,85
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00
Cooling						
Heat Transmission [%]	21,23	20,82	20,82	24,46	23,87	23,77
Long Wave Radiation [%]	21,72	21,30	21,30	25,03	24,42	24,33
Air leakage [%]	11,38	11,16	11,16	13,11	12,79	12,74
Air Infiltration [%]	45,68	46,73	46,73	37,42	38,93	39,17
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00
Total						
Heat Transmission [%]	23,99	23,66	23,63	26,90	26,38	26,35
Long Wave Radiation [%]	21,18	20,89	20,86	23,75	23,28	23,26
Air leakage [%]	16,23	16,01	15,99	18,21	17,84	17,83
Air Infiltration [%]	38,62	39,45	39,53	31,17	32,51	32,57
Total [%]	100,00	100,00	100,00	100,00	100,00	100,00

Table 9: Energy losses in %, t_c = 30 s

We verify that the order of magnitude of the different effects is very similar to the precalculation in chapter 2.6. Infiltration is by far the main source of losses in a door.

With very long cycle times and big buildings, the relative weight of the infiltration can be near 90%, but even with fast doors it achieves around 50% of the total losses. In case of increasing larger number of cycles, obviously the infiltration weight is also increased.

The influence of building size increases with the cycle time. In respect of the door area, there is quite a linear relation as expected.

5 Experiments

5.1 Background

There are quite limited experiences in technical literature investigating the global energy performance of a door inside a building. The most complete recent example with a wide approach is [**4**]. Most of this research is based on numerical calculations and only limited testing, but it reveals clearly the importance of air infiltration and related parameters in the overall energy losses.

Some research work it is been done based on CFD and FFD calculations investigating the air flow phenomena through building openings like [**6**] or [**7**], but few recent experiments are published with practical measurements on the field focused on door performance.

In older researches as summarised in [**1**] we do find several experiments specifically on air flow through large openings to characterise the infiltration phenomena like Baranowski et al. (1989), Tang and Robberechts (1989), Pelletret et al. (1988), Allard et al. (1987) and Crommelin and Vrins (1988).

Anyway, in these papers we do not find enough experimental base to evaluate the validity of the bulk density air flow model as described in chapter 3.1. In order to achieve this purpose specifically for the door field, some specific experiments have been done in the framework of this study.

5.2 Test equipment

The basic test consisted in the measurement of the air flow distribution through a door hole that separates two environments or spaces with different temperature conditions.

The tests were performed in the Hörrmann KG Brockhagen testing facilities in Steinhagen, Germany. They are equipped with a dual climate test chamber with the following characteristics:

Hot room:

- Dimensions (Width x Large x Height) = 6 x 4,5 x 4 m
- Maximum temperature = 60 °C
- Heating power = 30 kW
- Air flow = 3000 m$^3/h$

Cold room:

- Dimensions (Width x Large x Height) = 6 x 2,9 x 4 m
- Minimum temperature = -30 °C
- Cooling power = 22 kW
- Air flow = 9000 m^3/h

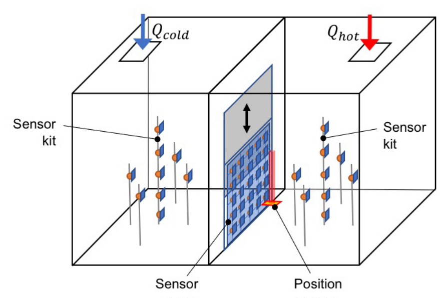

Figure 11: Test equipment distribution

A scaled door of 1x1 meter was used to have a balance between door size and chamber volumes to avoid fast temperature variations inside the rooms allowing air flow measurements in the flow.

To measure the air flow in the door hole, a measurement equipment was designed and manufactured specifically for these tests. The main objective was to be able to have a clear idea of the distribution of the wind speed, temperature and pressure in the door hole area, as well as temperature evolution in the rooms.

It was composed by three modules with distributed sensors:

1. Door sensor array
2. Hot room sensor kit
3. Cold room sensor kit

In total, there were 43 sensors, 25 in the door sensor array.

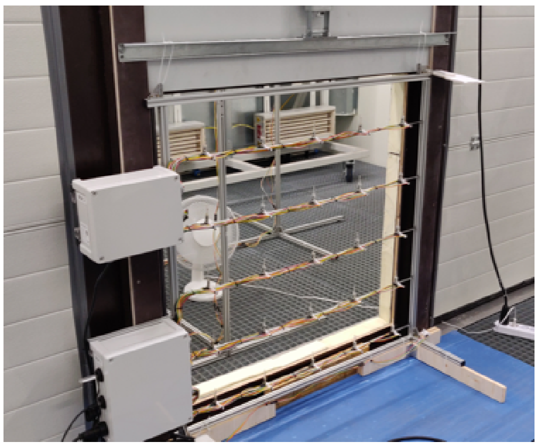

Figure 12: Door sensor array (hot side, door open)

Small wind and pressure free hardware sensors were used to have this multi-point measurement reducing the flow interference at a reasonable cost with enough precision.

Wind sensor rev. P by Modern Device was used for wind speed and temperature measurement. Speed measurement is based on hot wire technology, having analogue output for both wind speed and temperature signals. A study of its performance compared to more complex devices can be found in [**6**], where there are also reference values used to support calibration.

Static pressure measurement was done with BMP180 type sensors. They have a i2C digital output and they also include temperature and relative humidity sensors.

Their main characteristics are the following:

- Wind speed range: 0 - 67 m/s. Precision = 0,001 m/s
- Temperature range: -10 to 85 °C. Precision = 0,1 °C
- Pressure range: 0,3 to 1.3 MPa. Precision = 1 Pa

An additional laser sensor was included to measure the door position.

All sensors were integrated with a controller based in Arduino Mega. Both analogue and digital multiplexors were used to dispose of the large number of I/O required. A software was designed in Arduino platform to record and store the measurement data.

Figure 13: Hot and cold rooms sensor arrangements

5.3 Test procedure and conditions

After proper sensor calibration, the test procedure consisted in the following basic steps:

1. Chamber temperature regulation and stabilization.
2. Data recording start.
3. Door opening and closing cycling.

Several tests were pèrformed with different combinations of cold/hot temperatures and door cycle time, considering long, medium and short cycle times. Some tests were also performed using a small fan was to simulate the presence of wind pressure.

The detailed parameters were the following:

Test group	T_{out} [°C]	T_{in} [°C]	t_{open} [°C]	t_{closed} [°C]	$Wind\ Fan$
A	-5	30	60	180	NO
B	-5	30	30	60	NO
C	-10	30	300	600	NO
D	-10	30	300	600	YES
E	-10	30	180	120	NO
F	-10	30	60	120	NO
G	-10	30	60	120	YES
H	0	20	30	60	NO

Table 10: Test typologies

The following data were collected:

- Time [h:min:s]
- Air velocity [m/s] (analogue)
- Static pressure [Pa] (digital)
- Temperature [°C] (analogue)
- Temperature [°C] (digital)
- Relative humidity [%] (digital)
- Door position [m] (digital)

Analogue measures are taken in terms of electrical tension variation in Volts and converted into m/s.

5.4 Test results

All tests confirmed an air flow velocity distribution in the door hole similar profile to what is described by the simplified model. As an example, we show an instantaneous interpolated result with open door obtained in a test type A.

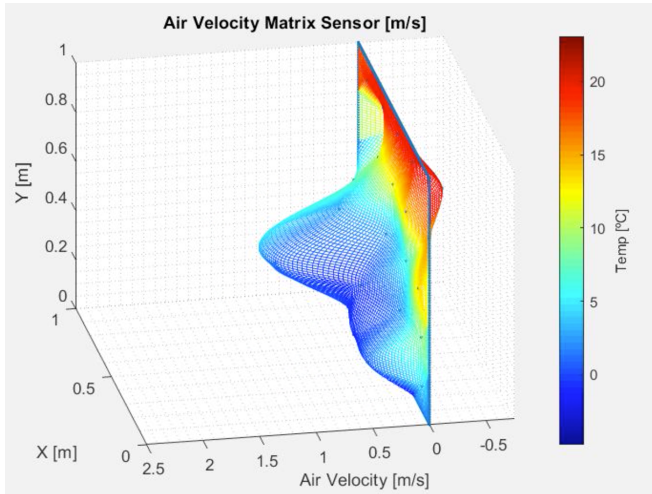

Figure 14: Air velocity distribution, test type A

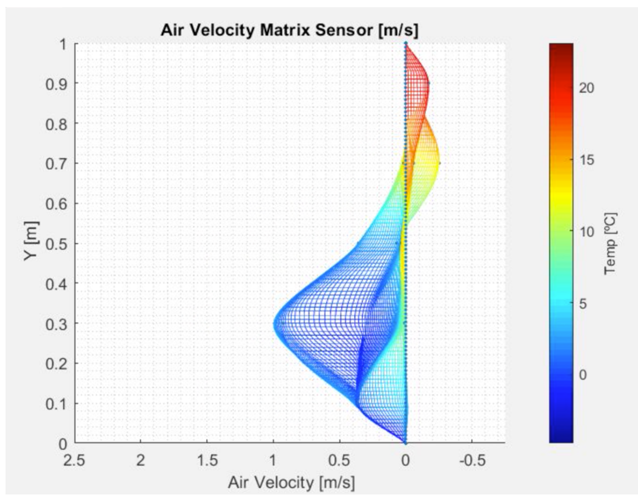

Figure 15: Air velocity distribution, test type A

In every case without wind pressure, the reverse flow was verified. The position of the neutral line was always positioned around 2/3 at total door height. Some verifications made with dry smoke confirmed this fact.

The experimental distribution in the transversal axe is not fully uniform, so an average is made for the simplified model synthesis.

In the following figures we show the transient performance of temperature and performance for three cases:

- Test type C: Long opening time (300 seconds)
- Test type A: Medium opening time (60 seconds)
- Test type B: Short opening time (30 seconds)

We show all sensor temperature and velocity curves and mean values across the door sensor matrix and both rooms.

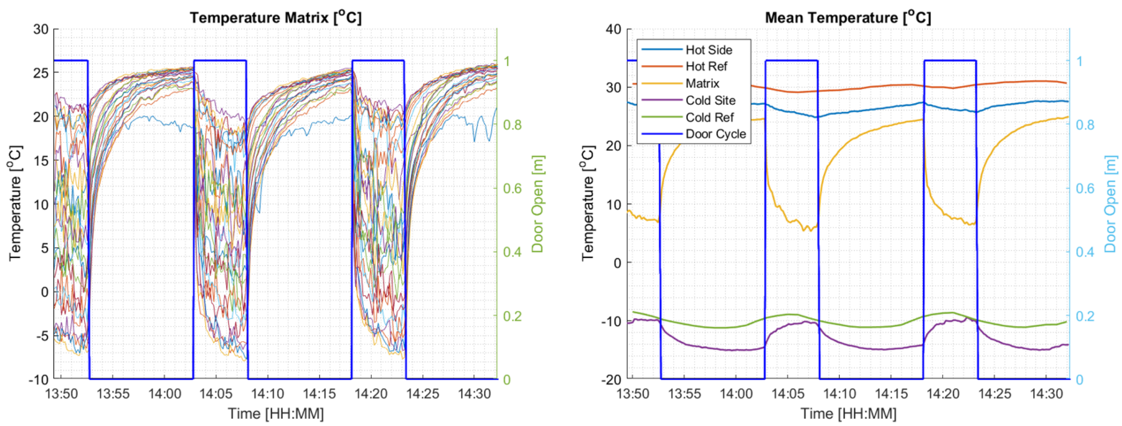

Figure 16: Temperature diagram, test type C

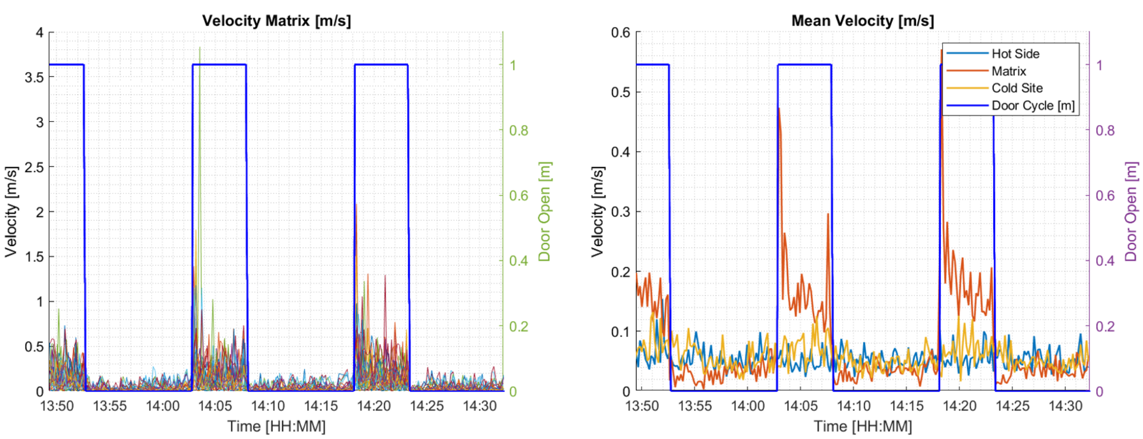

Figure 17: Velocity diagram, test type C

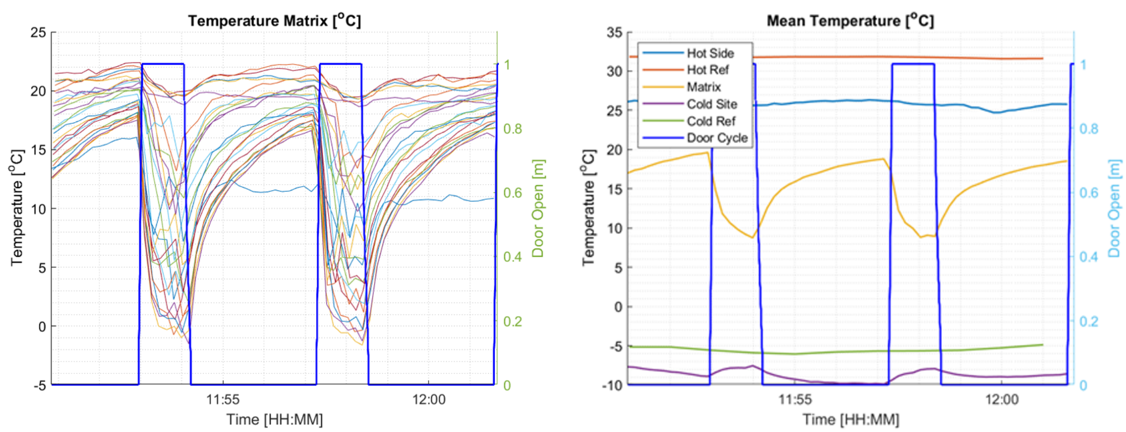

Figure 18: Temperature diagram, test type B

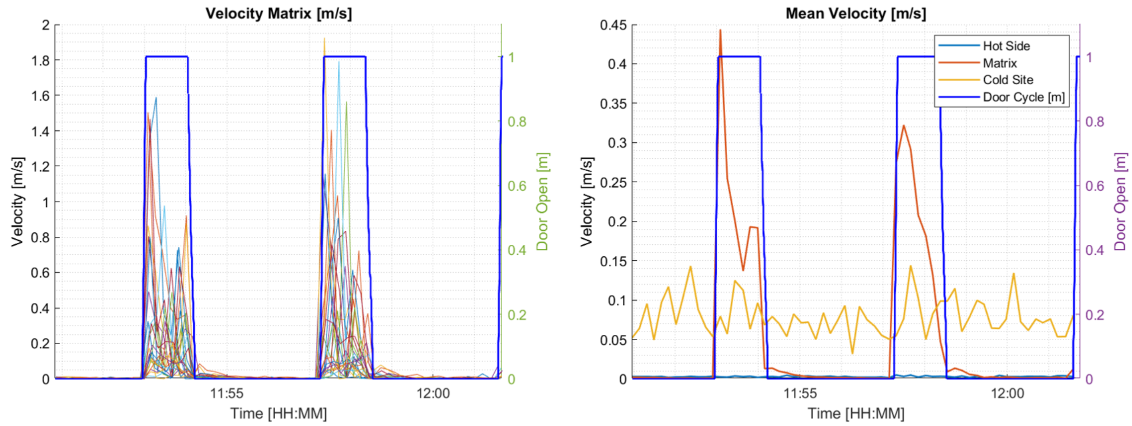

Figure 19: Velocity diagram, test type B

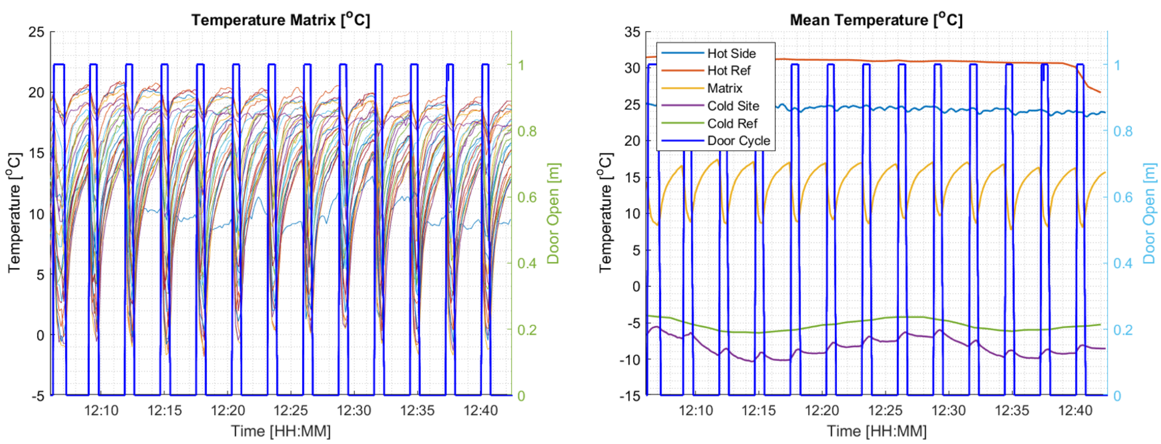

Figure 20: Temperature diagram, test type A

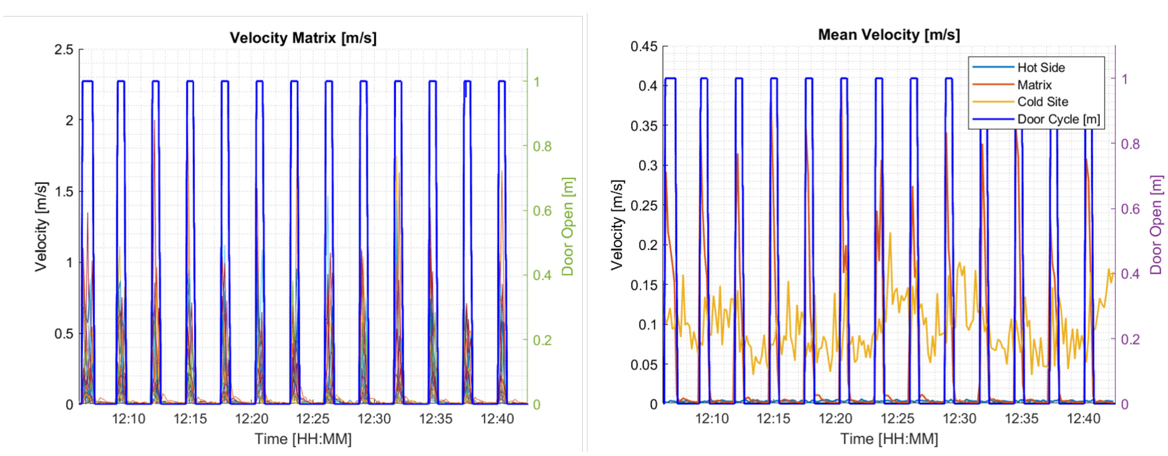

Figure 21: Velocity diagram, test type A

Regarding the main objective of the study, it is confirmed that the air flow distribution is similar in a qualitative way to what is described for the bulk density flow, with opposite flows divided by a neutral line.

In the diagrams is clearly visible the air flow in terms of velocity increase when the door is open in every cycle.

Regarding the overall performance of the system, the results also show a qualitative temperature performance in the rooms as described in modelling.

In our case, as the cold room is smallest, with a lower thermal inertia, we see clearly the temperature decrease in the room, having much lower variation in the hot room.

In the next chapter we will analysis the adjustment of the infiltration model with the experimental data.

5.5 Model experimental adjustment

The objective of the test was the characterization of the bulk density flow in order to validate the infiltration theoretical model as described.

In the following figure we show as an example both theoretical and experimental velocity profiles through the door height for one instant of a significant test and the theoretical velocity equation as in the annex 7.1.4.

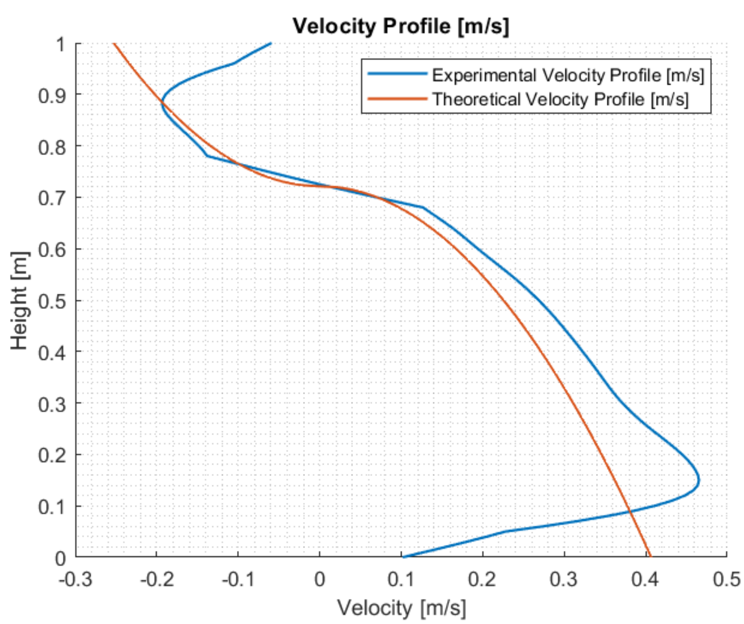

Figure 22: Experimental velocity profile

As it can be seen in the figure, the boundary layer effect is not explicitly included in the theoretical velocity profile, but the correction seen in section 3.1.3 is added in order to have an equivalent flux.

The $\delta(H)$ correction function was adjusted as shown in expression (3.1.29) with the tests results for door height = 1 m.

We also computed the experimental and the theoretical heat flux as a function of the parameters defining the linear temperature and pressure gradients through the door height b_1, b_2, d_1, d_2 (see section 3.1.2) in order to adjust by best fitting these parameters.

With the sensor lectures in terms of temperature and pressure in the rooms (see example in figures 22 and 23), the gradient adjustment was approximated having the best fit between the theoretical and experimental results. The position of sensors at a certain distance of the door hole inside the room required a further approximation.

In practice, the values for the parameters that have been found for the temperature profile are the following :

$$b_1 = 0,75\ K/m$$

$$b_2 = 1\ K/m$$

And for the pressure profile:

$$c_1 = c_2 = 101325\ Pa$$

$$d_1 = 15,1\ Pa/m$$

$$d_2 = 15,0\ Pa/m$$

We see that c_i can be assumed as the atmospheric pressure and that the values of d_i have an small variation. To extend the range of validity to larger doors, a expression of $d_1(d_2)$ was synthesised in (3.1.24).

In the next figure, we show the experimental heat flux and the theoretical heat flux with the adjusted parameters values as described before:

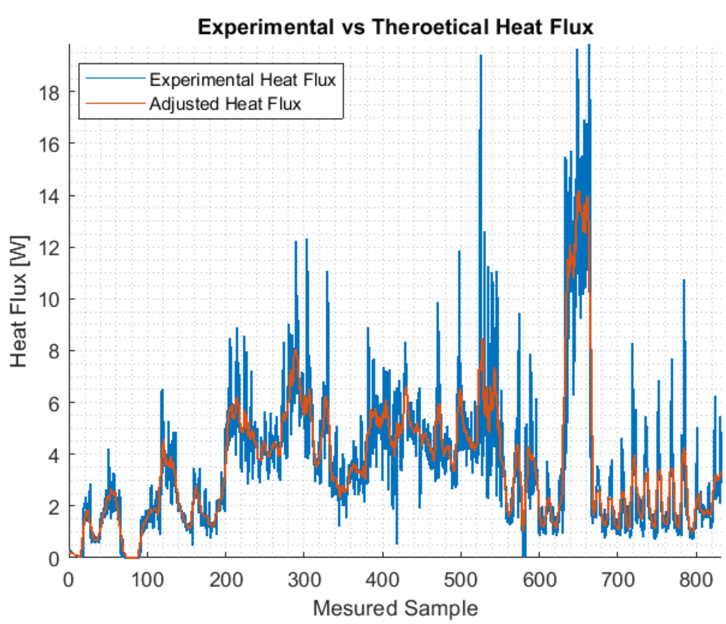

Figure 23: Experimental Infiltration Heat Flux

The assumption of these values fits well in the temperature and pressure range considered in the test conducted, but to confirm the range of validity it would be advisable to extend the range of testing also increasing the number of sensing points inside the rooms next to the door hole.

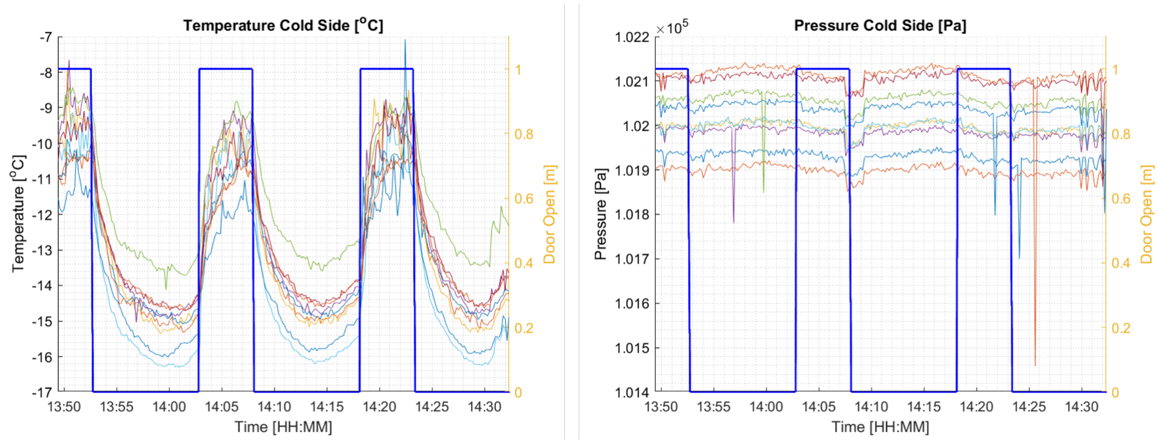

Figure 24: Temperature and Pressure Cold Room, test type A

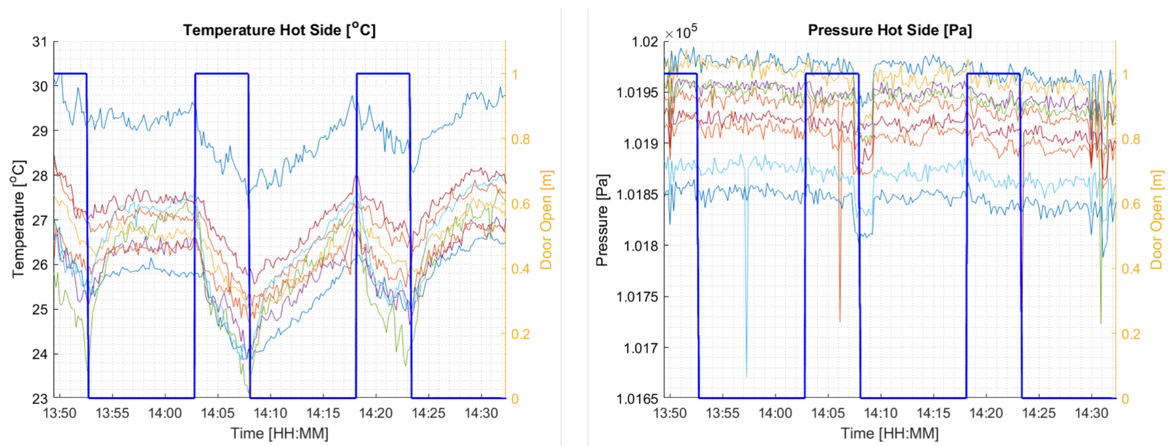

Figure 25: Temperature and Pressure Hot Room, test type A

6 Conclusions

The absence of available analytic calculation frameworks of energy losses through doors have usually taken in the past to underestimations, misunderstandings and improper evaluation of the variables and parameters involved in the phenomena.

We consider that the simplified calculation model developed is an adequate tool to calculate in a simple way the energy losses of building through a door allowing easy programming and integration in digital platforms. The main objective is to help the door industry to design and promote tools to support the energy efficiency and sustainability in architecture and building management. In this sense, the scope of this development is not the detailed simulation of complex real situations in real buildings, as this can only be approximated with numerical calculations and tools.

The importance of air infiltration or ventilation with the door open is clearly shown, helping to evaluate and compare properly the main factors involved in energy losses and changing the usual perspective about the main parameters to be specified for right product prescription for an intended use.

The most complex effect involved, which is the buoyancy or bulk density flow, it has been expressed in a consistent way and integrated in the model. The tests conducted confirm the overall performance predicted by this buoyancy model, but some assumptions have been made to adjust it properly. In this sense, further research would be required to confirm the range of validity of these assumptions in order to limit the expected variation of the results vs real situations.

The synthesis of the long wave radiation is also developed to allow the integration in the model of an effect not usually considered in building energy calculations but certainly present in the reality as confirmed by technical literature.

Although there are experimental approaches validating the single losses effects equations, an additional validation of the global energy losses model would require an specific test program with a detailed knowledge of all parameters involved in the test environment. Some technical challenges are associated to this test design for right parametrization and specification, sensoring and heating/cooling systems measurement.

Having in mind these research lines for the next future, we consider that the model is an important step ahead in the field and that the main objectives of the study are fully achieved.

7 Annex

7.1 Bulk density flow velocity profile calculation

As an alternative and complementary approach to the flow calculation in chapter 3.1, in the following we describe the equations of the air velocity profile explaining the phenomena with successive approximations.

We show how the Bernoulli equation is derived from the Navier-Stokes equation and then how empirical coefficients like the discharge coefficient and non-dimensional numbers like the Froude number, link real world observations to the idealised model.

7.1.1 The Equations of Motion, and Various Approximations

The dynamics of general fluid dynamics is based on the conservation of mass principle (expressed in the continuity equation) and the conservation of momentum principle (expressed in the Navier-Stokes equation). The derivation of these equations can be found in textbooks on fluid dynamics, but the following paragraphs are provided to show the origin of various assumptions leading to the Bernoulli theorem and serve as references for the literature review and discussions in later chapters.

The only external force that will be considered is that of gravity, which exercises a body force ρg per unit volume on each element of fluid (where ρ is the local density and g is the acceleration due to gravity). ρ may vary due to differences in temperature or to concentration but the fluid will be regarded as in-compressible. The compressibility of gases becomes only significant, for example, in deep layers as in atmospheric physics or at high velocities as in jets.

When ρ varies with height, the pressure variation in the fluid is given by the hydro-static equation

$$P = P_0 - g \int_0^z \rho dz' \tag{7.1.1}$$

This shows that the fluid is in equilibrium only when the density, as well as the pressure, is constant in every horizontal plane.

The continuity equation applied to a given volume expresses the fact that the difference between the inflowing mass and the outflowing mass equals the time rate of change of the mass in the volume. For incompressible flows the continuity equation in vector notation is:

$$\nabla \cdot \vec{u} = 0 \tag{7.1.2}$$

Three forces can be distinguished when considering the acceleration dv/dt of a fluid element:

1. Gravity force field $F = -\rho g$
2. Gradient in the pressure field P
3. Friction forces $F(\mu)$ related to the viscosity μ

Applying Newton's second law of motion, the Navier-Stokes differential equation for viscous flow can be obtained:

$$\rho \frac{D\vec{u}}{Dt} = -\nabla P + \rho \vec{g} + F(\mu) \tag{7.1.3}$$

The acceleration vector Du/Dt, concerning an observer moving at the velocity of the liquid (Euler coordinates), can be decomposed as follows:

$$\frac{D\vec{u}}{Dt} = \frac{\partial \vec{u}}{\partial t} + (\vec{u} \cdot \nabla) \cdot \vec{u} = \frac{\partial \vec{u}}{\partial t} + \frac{1}{2}\nabla \vec{u}^2 + (\nabla \times \vec{u}) \times \vec{u} \tag{7.1.4}$$

where for steady-state situations the partial derivative with respect to time is zero. This steady-state description can be used as a good approximation over time intervals where the change of the velocity field is negligibly small.

The modelling of dissipating friction forces in the last term $F(\mu)$ is just what makes fluid dynamics a difficult subject (the interested reader is referred to the textbooks). For perfect fluids (non-viscous flows), the last term in Equation 7.1.3 is neglected the Euler equations of motion are obtained.

It can be shown that only differences of density ρ' from some standard value ρ_0 are relevant in determining the effect of gravity. The Euler equations can be written in terms of the deviations P' and ρ' (one sets $P = P_0 + P'$ and $\rho = \rho_0 + \rho'$) from a reference state of hydrostatic equilibrium:

$$\rho \frac{D\vec{u}}{Dt} = -\nabla P' + \rho' \vec{g} \tag{7.1.5}$$

where,

$$\nabla P_0 = \rho_0 g$$

After division of this equation by the reference density ρ_0,

$$\left(1 + \frac{\rho'}{\rho}\right) \frac{D\vec{u}}{Dt} = -\frac{1}{\rho}\nabla P' + \frac{\rho'}{\rho}\vec{g} \approx \frac{D\vec{u}}{Dt} \tag{7.1.6}$$

It is seen that the density ratio ρ'/ρ appears twice, in the first (inertia) term and in the buoyancy term. When ρ'/ρ_0 is small, it produces only a small correction to the inertia compared to a fluid of density ρ_0, but it is of primary importance in the buoyancy term.

The Boussinesq approximation (Equation 7.1.6) consists essentially of neglecting variations in so far as they affect inertia, but retaining them in the buoyancy terms, where they occur in the combination $g' = g\rho'/\rho_0$. Equations 7.1.3 and 7.1.4 combine for non-viscous flow into the Euler equations:

$$\rho \left(\frac{\partial \vec{u}}{\partial t} + \frac{1}{2}\nabla \vec{u}^2 + (\nabla \times \vec{u}) \times \vec{u} \right) = -\nabla P + \rho \vec{g} \tag{7.1.7}$$

For a steady flow, Equation 7.1.7 can be integrated along a streamline and for any two points on this trajectory the equation can be written as:

$$P_1 + \frac{1}{2}\rho \vec{u_1}^2 + \rho g z_1 = P_2 + \frac{1}{2}\rho \vec{u_2}^2 + \rho g z_2 = Cte \tag{7.1.8}$$

This is the Bernoulli theorem for inviscid rotational flow expressing the total conservation of mechanical energy along a trajectory with three terms representing pressure energy, potential energy and kinetic energy. The constant is in principle varying from streamline to streamline.

Irrotational flow: When integrating Equation 7.1.3 with the additional requirement of irrotational flow,

$$\nabla \times \vec{u} = 0 \tag{7.1.9}$$

The flow is a potential flow, and the Bernoulli equation is obtained without being restricted to a streamline, the constant being the same for all the streamlines in the flow.

When viscous forces can not be neglected as for the case of flow through a pipe or duct, the loss in mechanical energy caused by fluid internal friction results in a loss of pressure, and the Bernoulli equation can then be generalised by adding a pressure drop term ΔP_f to Equation 7.1.8:

$$P_2 - P_1 + \frac{1}{2}\rho\left({u_2}^2 - {u_1}^2\right) + \rho g\left(z_2 - z_1\right) + \Delta P_f = 0 \tag{7.1.10}$$

7.1.2 Application of the Bernoulli Equation

The classical approach of the so-called gravitational flow assumes air flows through large openings to be driven by density fields on both sides of the opening. Each room is considered as a semi-infinite reservoir, all walls are assumed to be in thermal equilibrium with the air enclosed in each cavity, for example, no boundary layer flows, and each streamline is assumed to be horizontal.

The usual way to solve this general problem is to apply Bernoulli equation in the plane and on both sides of the opening, which is limited to non-viscous, in-compressible flow and valid for a stationary flow regime.

Assuming hydrostatic pressures (Equation 7.1.1) on both sides (labels 2 and 1) of the opening, then the velocity in the opening $u(z)$ is obtained from Equation 7.1.8. First assuming $z_2 = z_1$, (for example, a horizontal streamline), the pressure difference $P_2 - P_1$ is known by taking two points far from the opening where the velocities are about equal. Then using $u^2 >> u_2^2 - u_1^2$, the velocity in the opening is:

$$u(z) = \sqrt{\frac{2}{\overline{\rho}_m}\left((P_{2_0} - P_{1_0}) - g\int_0^z \rho_2(z) - \rho_1(z)dz'\right)} \tag{7.1.11}$$

and if $\rho_1(z) = \overline{\rho}_1$ and $\rho_2(z) = \overline{\rho}_2$, we get

$$u(z) = \sqrt{\frac{2}{\overline{\rho}_m}\left((P_{2_0} - P_{1_0}) - gz\left(\overline{\rho}_2 - \overline{\rho}_1\right)\right)} \tag{7.1.12}$$

where $\overline{\rho}_m = \frac{\overline{\rho}_1 + \overline{\rho}_2}{2}$ is the density of the flowing medium. This velocity is the maximum theoretical velocity in a non-viscous fluid. This reasoning can be generalised for ideal fluids (to curved streamlines in the flow through a "window" as in Figure 2.2), by considering the horizontal velocity component rather then the velocity along a streamline.

7.1.3 Linear Temperature Gradients

In the preceding problem the air density in both zones is presumed to be uniform. Uniform air density is a good approximation when the vertical temperature variation over the opening height in each zone is much smaller than the horizontal temperature difference. In fact, because of thermal stratification, or gradients of concentration of any species (humidity, pollutants), this assumption is restrictive and does not allow for the general behaviour of a large vertical opening.

A density ρ depending only on temperature can be written in terms of reference temperature T_0. Pressure is assumed constant so $P(z) = P$

$$\rho = \frac{P}{T\,R_{air}} \tag{7.1.13}$$

$$\rho_0 = \frac{P_0}{T_0\, R_{air}} \tag{7.1.14}$$

where $R_{air} = 287.05\, \frac{J}{kg\, K}$ and is the air gas constant.

So if we assume a linear temperature variation $T = T_0(a + b\, z)$ $[K]$ then,

$$\rho(z) = \frac{\rho_0}{(a + b\, z)} \tag{7.1.15}$$

So finally if we assume a linear temperature variation, the Equation 7.1.11, turns to be:

$$u(z) = \sqrt{\frac{2}{\overline{\rho}_m}\left[(P_{2_0} - P_{1_0}) - g\frac{\rho_{0_2}}{b_2}\ln\left(1 + \frac{b_2}{a_2}z\right) + g\frac{\rho_{0_1}}{b_1}\ln\left(1 + \frac{b_1}{a_1}z\right)\right]} \tag{7.1.16}$$

Where:

$$\overline{\rho}_m = \frac{\rho_1 + \rho_2}{2}$$
$$\rho_1 = \frac{\rho_{0_1}}{(a_1 + b_1\, z)}$$
$$\rho_2 = \frac{\rho_{0_2}}{(a_2 + b_2\, z)}$$
$$\rho_{0_1} = \frac{P_{0_1}}{T_{0_1}\, R_{air}}$$
$$\rho_{0_2} = \frac{P_{0_2}}{T_{0_2}\, R_{air}}$$
$$R_{air} = 287.05\, \frac{J}{kg\, K}$$

7.1.4 Linear Temperature Gradients and Linear Pressure Gradients

In the preceding problem the Temperature in both zones was presumed to be with a linear gradient, and the Pressure was also assumed in both zones constant. But because the linear gradient Temperature many times exist a Pressure gradient also. So let us assume that our Pressure profile is also linearly dependent, $P = P_0(c + d\, z)$ $[Pa]$.

So density ρ depending on temperature and pressure can be written in terms of reference temperature T_0 and P_0.

$$\rho(z) = \rho_0 \frac{c + d\, z}{a + b\, z} \tag{7.1.17}$$

So finally if we assume a linear temperature and pressure variation, the Equation 7.1.11, turns to be:

$$u(z) = \sqrt{\frac{2}{\overline{\rho}_m}\left[(P_2 - P_1) - g\rho_{0_2}\left(C_2\ln\left(1 + \frac{b_2}{a_2}z\right) + C_4 z\right) + g\rho_{0_1}\left(C_1\ln\left(1 + \frac{b_1}{a_1}z\right) + gC_3 z\right)\right]} \tag{7.1.18}$$

where,

$$P_1 = P_{0_1}\,(c_1 + d_1\,z)$$

$$P_2 = P_{0_2}\,(c_2 + d_2\,z)$$

$$C_2 = \frac{(b_2 c_2 - a_2 d_2)}{b_2^2}$$

$$C_4 = \frac{d_2}{b_2}$$

$$C_1 = \frac{(b_1 c_1 - a_1 d_1)}{b_1^2}$$

$$C_4 = \frac{d_1}{b_1}$$

$$\overline{\rho}_m = \frac{\rho_1 + \rho_2}{2}$$

$$\rho_1 = \rho_{0_1}\frac{c_1 + d_1\,z}{(a_1 + b_1\,z)}$$

$$\rho_2 = \rho_{0_2}\frac{c_2 + d_2\,z}{(a_2 + b_2\,z)}$$

$$\rho_{0_1} = \frac{P_{0_1}}{T_{0_1}\,R_{air}}$$

$$\rho_{0_2} = \frac{P_{0_2}}{T_{0_2}\,R_{air}}$$

$$R_{air} = 287.05\,\frac{J}{kg\,K}$$

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