A model of the temperatures in a truck cabin

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vid Tekniska Högskolan i Linköping
av

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A model of the temperatures in a truck cabin
En model över temperaturerna i en lastbilshytt

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Sammanfattning
Abstract

Temperature variations in a truck cabin are rather slow and therefore timeconsuming to measure. Because of Scania's great number of truck cabin models, much work, time and money is saved if a mathematical model can predict how the temperature characteristics of the cabins change with different sizes and materials.

A model has been implemented in Matlab/Simulink. The model has a flexible structure and is rather detailed to make it adaptable to different cabin designs regarding materials and sizes.
The output of the model is the cabin's global air temperature. In addition to that there are a large number of temperatures and heat flows that can be displayed. The inputs to the model are the temperature and amount of air provided by the fan, the outdoor temperature, the heat and sun radiation incident on the cabin, the wind speed relative to the truck and the mass flow between the cabin air and the outdoor air caused by diffusion.
The model can give a good view of how a real system would behave and is useful for e.g temperature regulator design and heat loss investigations.

The model behaves like expected from a physical point of view. However the model's simulation results do not correlate with the measurements as good as desired. The cause of this is likely that the measuments are incorrect. Futher verifications are needed with more accurate measurements.

Nyckelord
Keywords

model, temperature, heat, truck cabin

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1 Introduction

1.1 Background and Purpose

Temperature variations in a truck cabin are rather slow and therefore time-consuming to measure. Because of Scania's module program where the size of the cabin can be altered between several different sizes, much work, time and money is saved if a mathematical model can predict how the temperature characteristics of the cabins change with different sizes and materials.

The purpose of this work is to make a model of the temperatures in a truck cabin.

The model will mainly be used to simulate the indoor temperature to ensure that a temperature regulator will have acceptable performance for the different cabin sizes Scania are producing.

1.2 Goal

The model should make it possible to determine if a temperature regulator, developed and approved for a certain truck cabin model, also could be used in other truck cabins of different sizes considering stability, time to reach the desired temperature and other performances. Therefore, the model should be easy to adapt to different cabin designs regarding materials and sizes.

1.3 Limitations

Below follows some important limitations of the work

- Only the global cabin air temperature is to be considered.
- To correctly calculate the absolute temperature is not a demand.
- Full verification is not included but verification should be made to make the model plausible.
- The temperature range is 220-330 Kelvin.
- The model should be implemented in Matlab/Simulink

1.4 Readers Guide

Chapter 2 gives a theoretical survey of the heat transport processes of interest for the model in this thesis. Chapter 3 treats the main approximations. Chapter 4 gives an introduction to the main parts of the model and their notation. In Chapter 5 simulations are compared to measurements and the deviations between them are to some extent explained. Chapter 6 contains the results and Chapter 7 documents the lower levels of the model and also includes some Matlab code.

2 Theory

The one-dimensional heat-flow in steady state from one layer with temperature T_1 to another layer with temperature T_2 can in the general case be put on the form

$$\dot{Q}_{12} = \alpha \cdot A \cdot (T_1 - T_2) \tag{2.1}$$

A is the area of the cross section and α is the heat transfer coefficient with the dimension W/Km^2. The main problem in the study of heat transfer is to find the α values in different heat transport processes and under different conditions.

Heat is generally transferred through four different processes. These are conduction, convection, radiation and mass transport. A treatment of the processes will follow below.

References in this chapter are
J.P Holman, Bo Pierre, C Kittel

2.1 Conduction

2.1.1 Physical mechanism

Heat transfer by conduction is the way heat is transferred in solids. The energy is spread by a diffusion process where the electrons and the lattice vibrations in solids are excited by frequent collisions. For metals the electron part is dominant and for dielectric matter (insulators) the lattice vibrations is dominant though the electrons are much harder bound.

The probability for a small volume in the solid to absorb or give away energy depends on the difference in temperature between the volume and its neighbourhood. The amount of energy or heat transported is then proportional to the temperature gradient and we get the Fourier's law

$$\dot{q} = -\lambda \cdot \nabla T \tag{2.2}$$

where $\dot{q}$ $[W/Km^2]$ is the heat flux.

The thermal conductivity coefficient λ is defined with respect to the steady-state flow of heat down a long rod as

$$\lambda = -\frac{\dot{q}}{\frac{dT}{dx}} \tag{2.3}$$

λ has the dimension $W/_{Km}$.

The thermal conductivity coefficient is temperature dependent but in the temperature range for which the model will be used that is 220-320 Kelvin the coefficient is approximated as a constant.

2.1.2 The heat equation

The internal energy for a piece of mass is

$$U = C_v \cdot m \cdot T = C_V \cdot \int_V \rho \cdot dV \cdot T \tag{2.4}$$

The change in internal energy for a closed system region is determined by the amount of heat-flow through the surface and the power generated in the region.

$$\frac{dU}{dt} = \dot{Q} + P \tag{2.5}$$

where $\dot{Q}$ is the heat flow into the volume and P the power generated in the volume.

Taking the time derivative on the equation (2.4).

$$\frac{dU}{dt} = \frac{d}{dt}(C_v \cdot \int_V \rho dV \cdot T) = \int_V C_v \rho \frac{\partial T}{\partial t} dV \tag{2.6}$$

where the last relation holds because C_v, ρ and the co-ordinates are taken to be time independent.

The heat flow into the region is

$$\dot{Q} = -\int_S \dot{q} \cdot \hat{n} dS = -\int_V \nabla \cdot \dot{q} dV \tag{2.7}$$

where $\dot{q}$ is the vector field of the heat flux
The total amount of generated heat inside the region is

$$P = \int_V h dV \tag{2.8}$$

where h is the heat generated per unit of time and volume

Using the equations (2.5)-(2.8) we have

$$\int_V C_v \rho \frac{\partial T}{\partial t} dV = \int_V (-\nabla \cdot \dot{q} + h) dV \tag{2.9}$$

This equation holds for any chosen balance region only if the integrand of the left-hand side equals the right-hand side. That is if

$$C_v \rho \frac{\partial T}{\partial t} = -\nabla \cdot \dot{q} + h = \lambda \nabla^2 T + h \tag{2.10}$$

where the Fourier's law $\dot{q} = -\lambda \nabla T$ (2.2) has been used

Finally grouping the constants as

$$\kappa = \frac{\lambda}{C_v \rho} \tag{2.11}$$

give the heat equation.

$$\frac{\partial T}{\partial t} = \kappa \nabla^2 T + \frac{\kappa h}{\lambda} \tag{2.12}$$

2.1.3 Steady state solution

In steady state $\frac{\partial T}{\partial t} = 0$ and if we have no internal heat source we obtain

$$\kappa \nabla^2 T = 0 \tag{2.13a}$$

in one dimension that is

$$\kappa \frac{\partial^2 T}{\partial x^2} = 0 \tag{2.13b}$$

the solution to equation (2.13b) is $T(x) = Cx + D$

Using the boundary conditions

$$T(0) = T_o \tag{2.14a}$$
$$T(L) = T_L \tag{2.14b}$$

where L is the distance between the two temperature layers. We can derive C and D as

$$D = T(0) = T_0 \tag{2.15a}$$
$$C = \frac{T(L) - T(0)}{L} = \frac{T_L - T_0}{L} \tag{2.15b}$$

The heat-flow will look like

$$\dot{q} = -\lambda \nabla T = -\lambda \frac{\partial T}{\partial x} \hat{x} = -\lambda \frac{T_L - T_0}{L} \hat{x} = \alpha (T_0 - T_L) \hat{x} \tag{2.16}$$

where

$$\alpha = \frac{\lambda}{L} \tag{2.17}$$

The heat flow through a cross section with area A is now on the form $\dot{Q} = \alpha \cdot A \cdot (T_0 - T_L)$ (2.2). Typical values of α for a wall with a thickness of 1 cm are between 3.6 (plastic foam) and 4500 (steel). At steady state a temperature difference of 1 Kelvin between the surfaces then requires a heat flow of 3.6 W and 4500 W respectively if the cross section area is 1 m^2.

2. 2 Convection

Convection is the process that is responsible for the heat exchange between a surface and a surrounding fluid, in this case the air. When the air molecules come in contact with the surface they will absorb or emit heat depending on the temperature difference between the air and the surface.
There are two different types of convection depending on if the fluid as a whole have any velocity component and that way is forced on the surface, or if the fluid comes in contact with the surface in a more natural way that is by diffusion. The two types are called Forced convection and Natural convection.

2.2.1 Natural convection

The thin layer of air that is in contact with a solid's surface will have the same temperature as the surface. If there is a temperature difference between the surface and the air a small distance away from the surface, there will by the ideal gas law also be a density difference between the air layers. This density difference results by diffusion in a net transport of surface air molecules away from the surface. Their place is then taken by other air molecules that will interact with the surface and a heat transport takes place.

The heat transfer by natural convection was treated by Lorenz in 1881.
He studied a vertical surface with constant temperature T_s surrounded by a colder fluid with temperature T_{air}. The fluid flow close to the surface was assumed to be purely vertical without any horizontal velocity components. From the heat balance for an infinitesimal piece of the fluid, and with temperature independent material parameters, Lorenz derived by integration the following formula for the heat transfer coefficient.

$$\alpha = 0.548 \cdot \left(\frac{g\beta \Delta T \rho C_p \lambda^3}{vH} \right)^{1/4} \tag{2.18}$$

where

$H =$ height of the surface
$\Delta T = T_s - T_{air}$ = The temperature difference between the surface and the surrounding media.
$\beta = \frac{1}{V} \cdot \frac{\partial V}{\partial T} = \frac{1}{T_{air}}$ = heat expansion coefficient for ideal gas
$g =$ gravity
$\rho =$ density of the fluid

C_p = heat capacity of the fluid
λ = coefficient of thermal conductivity of the fluid
v = cinematic viscosity

The equation (2.18) can also be expressed non-dimensionally as

$$\frac{\alpha H}{\lambda} = 0.548\left(\frac{g\beta\Delta TH^3}{v^2}\right)^{1/4} \cdot \left(\frac{v}{a}\right)^{1/4} \tag{2.19}$$

or

$$Nu = 0.548 \cdot (Gr \cdot \Pr)^{1/4} \tag{2.20}$$

with

$$Nu = \frac{\alpha H}{\lambda} = \text{Nusselts number} \tag{2.21}$$

$$Gr = \frac{g\beta\Delta TH^3}{v^2} = \text{Grashofs number} \tag{2.22}$$

$$\Pr = \frac{v}{a} = \frac{\mu C_p}{\lambda} = \text{Prandtls number} \tag{2.23}$$

where

a = thermal diffusivity
μ = viscosity

Over the years it has been found that the average free-convection Nusselt number can be represented in the following functional form for a variety of circumstances:

$$Nu_f = C\left(Gr_f \cdot \Pr_f\right)^m \tag{2.24}$$

where the subscript f indicates that the properties in the dimensionless groups are evaluated at the film temperature

$$T_f = \frac{T_{air} + T_s}{2} \tag{2.25}$$

This table from J.P Holman (p 341) recommends the following values on C and m for different cases of interest.

Geometry	**Gr*Pr**	**C**	**m**
Vertical planes	$10^9 - 10^{13}$	0,10	1/3
Upper surface of heated plates or lower surface of cooled plates	$8 \cdot 10^6 - 10^{11}$	0.15	1/3
Lower surface of heated plates or upper surface of cooled plates	$10^5 - 10^{11}$	0.27	1/4

Typical values of Pr*Gr , alpha and the heat-flow into a vertical plate with area 1 m^2 is using the temperatures

Tair = 300 K
Twall = (300 - T) K

T K	$\alpha\ \left[W/m^2K \right]$	Pr*Gr	Q $\left[W/m^2 \right]$
2	1.8	$1.8 \cdot 10^8$	3.6
5	2.0	$4.7 \cdot 10^8$	10
10	2.7	$9.5 \cdot 10^8$	27
25	3.4	$2.4 \cdot 10^9$	85

2.2.2 Forced convection

The convection is said to be forced when the surrounding fluid flows along the surface. To get a picture of the heat transfer process by forced convection we have to study the fluid flow close to the surface. Because of the viscosity, the flow close to the surface is always laminar even when turbulence appears further out. In such a flow, where no mixture of the different media layers occurs, the heat transfer is due to conduction. Hence the heat transfer rate can be expressed as

$$\dot{q} = \alpha(T_s - T_{air}) = -\lambda \left(\frac{dT}{dx} \right)_s \qquad (2.25)$$

where

$$\alpha = \frac{-\lambda \left(\frac{dT}{dx} \right)_s}{(T_s - T_{air})} \qquad (2.26)$$

and $\left(\frac{dT}{dx}\right)_s$ is the temperature gradient at the surface.

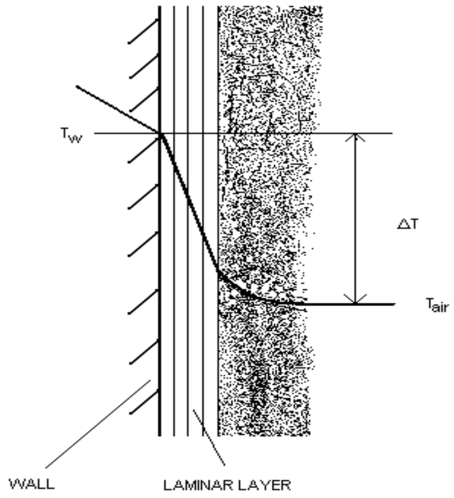

Figure 2.1 shows the temperature distribution for forced convection

If we assume that the whole temperature fall ΔT happens by conduction within an effective thermal boundary layer Δx, we will get the thermal boundary-layer thickness through the relation

$$\dot{q} = \alpha \cdot \Delta T = \lambda \cdot \frac{\Delta T}{\Delta x} \tag{2.27}$$

and

$$\Delta x = \frac{\lambda}{\alpha} \tag{2.28}$$

The magnitude of Δx and indirectly α depends on a number of factors, dimension and form of the surface, conductivity, viscosity etc., and is difficult to calculate. Of great importance is if the airflow is turbulent or not. Initially, the boundary layer development is only laminar but at some critical distance from the leading edge, depending on the flow field and fluid properties, small disturbances in the flow begin to become amplified, and a transition process takes place until the flow also becomes turbulent.

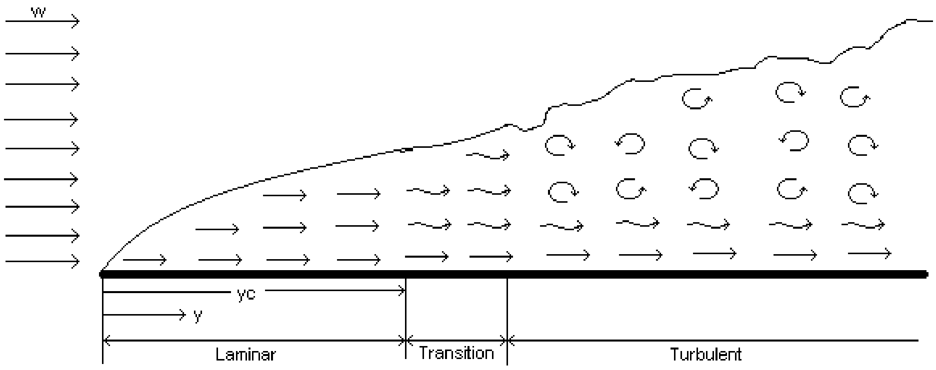

Figure 2.2 shows how the air behaves depending on the distance from the leading air.

The transition from laminar to turbulent flow occurs when

$$\frac{wy}{v} = \frac{\rho wy}{\mu} > 5 \cdot 10^5 \qquad (2.29)$$

where
$w =$ free-stream velocity
$y =$ distance from leading edge
$v = \mu/\rho =$ cinematic viscosity

This particular grouping of terms is called the Reynolds number, and is dimensionless.

$$\mathrm{Re}_y = \frac{wy}{v} \qquad (2.30)$$

The normal range for the beginning of transition is between $5 \cdot 10^5$ and 10^6, depending on surface-roughness condition and the ”turbulence level” of the surface.

The Prandlt number v/a has been found to be the parameter that relates the relative thickness of the hydrodynamic and thermal boundary layer. It is thus the connecting link between the velocity field of the fluid and the temperature field.

It can be shown that thermal boundary-layer thickness is proportional to the dimensionless Prandtl’s number Pr as

$$\Delta x \propto \mathrm{Pr}^{-1/3} \qquad (2.31)$$

For a plate heated over its entire length one obtains by momentum and energy laws

$$\frac{\alpha_y \cdot y}{\lambda} = Nu_y = 0.332 \cdot \Pr^{1/3} \mathrm{Re}_y^{1/2} \tag{2.32}$$

which expresses the local value of the heat-transfer coefficient in terms of the distance from the leading edge y of the plate and the fluid properties. The average heat-transfer coefficient and Nusselt number may be obtained by integrating over the length L of the plate.
Doing so one obtains

$$< Nu_L >= 0.664 \cdot \Pr^{1/3} \cdot \mathrm{Re}_L^{1/2} \tag{2.33}$$

as the average Nusselt number in the laminar case.
The empirical relation

$$< Nu_L >= \frac{< \alpha > \cdot L}{\lambda} = \Pr^{1/3} \left(0.037 \cdot \mathrm{Re}_L^{4/5} - 871\right) \tag{2.34}$$

is used for turbulent flow that is when $\mathrm{Re}_L > 5 \cdot 10^5$.

Observe that α is independent of any temperature.
Typical values with y=1 m and a temperature difference of 10 degrees

w [m/s]	w [km/h]	α $[W/m^2K]$	Re	Q $[W/m^2]$
0	0	0	0	0
1	3.6	3.9	$6.0 \cdot 10^4$	39.2
2	7.2	5.5	$1.2 \cdot 10^5$	55.5
5	18	8.8	$3.0 \cdot 10^5$	87.7
10	36	16.3	$6.0 \cdot 10^5$	163
15	54	30.7	$8.0 \cdot 10^5$	307
20	72	44	$1.2 \cdot 10^6$	440
25	90	56.8	$1.5 \cdot 10^6$	568

2.3 Radiation

Surface electrons absorb part of the electromagnetic radiation incident on the surface and then emit radiation when they are spontaneously deexcitated.

2.3.1 Thermal radiation

Thermal radiation is electromagnetic radiation emitted by a body as a result of its temperature. For the total thermal radiation from a blackbody, Stefan-Boltzmann's law applies.

$$\dot{q}_R = \sigma T^4 \qquad (2.35)$$

$\dot{q}_R$ is the total radiation energy per unit of time. The heat-flow is proportional to T^4 where T is the absolute temperature of the surface. σ is the Stefan-Boltzmann constant with the value $5.67 \cdot 10^{-8} \left[W / m^2 K^4\right]$.

Slightly modified the Stefan-Boltzmann law also holds for grey radiation, that is when the intensity spectrum is similar to the black but with a smaller absolute value. In this case

$$\dot{q}_R = \varepsilon_{grey} \sigma T^4 \qquad (2.36)$$

The emissivity ε is naturally less then one and it is independent of wavelength and temperature.

For the general case we have the relation.

$$\dot{q}_R = \varepsilon \sigma T^4 \qquad (2.37)$$

Generally the emissivity is no longer a constant but depends on the temperature T.

When radiant energy strikes a material surface, part of the radiation is reflected, part is absorbed, and part is transmitted. If r is defined as the reflectivity, a as the absorbability and t as the rate of transmission we have the relation

$$r + a + t = 1 \qquad (2.38)$$

Most solid bodies do not transmit thermal radiation, so that for many applied problems the rate of transmission may be taken as zero.

Then

$$r + a = 1 \tag{2.39}$$

At equilibrium the energy absorbed by the body must be equal to the energy emitted; otherwise there would be an energy flow into or out of the body which would raise or lower the body's temperature.

If two bodies with temperature T is in thermal contact in that way that they exchange radiation only with each other, they will maintain their temperatures. Otherwise the entropy will decrease and that is not allowed according to the laws of thermodynamics. Letting one of the bodies be a black body and writing the net energy flow into the black body which must be zero otherwise the temperature will change.

$$\dot{Q} = -\sigma T^4 + (1-a)\sigma T^4 + \varepsilon\sigma T^4 = (\varepsilon - a)\sigma T^4 = 0 \tag{2.40}$$

which only holds for all T if

$$\varepsilon = a \tag{2.41}$$

so the emissivity is equal to the absorptivity. This is the Kirchhoff law.

Imagine two surfaces with areas 1 m^2 exchanging radiation only with each other and that one surface (surface 1) is perfectly black.

The heat-flow from surface 1 to surface 2 is

$$\dot{q} = \sigma(aT_1^4 - \varepsilon T_2^4) = a\sigma(T_1^2 + T_2^2)(T_1 + T_2)(T_1 - T_2) = \alpha(T_1 - T_2) \tag{2.42}$$

with

$$\alpha = a\sigma(T_1^2 + T_2^2)(T_1 + T_2) \tag{2.43}$$

If the surface 2 has the temperature T_2 = 250 K. Then we will have the following table for the two cases with $a = \varepsilon = 0.2$ and $a = \varepsilon = 0.7$ which is representative numbers for steel and glass.

$T_2 = 250K$	$\varepsilon = 0.2$ (steel)		$\varepsilon = 0.7$ (glass)	
T_1	$\alpha\left[W/Km^2\right]$	$\dot{q}\left[W/m^2\right]$	$\alpha\left[W/m^2\right]$	$\dot{q}\left[W/m^2\right]$
250	0.71	0	2.50	0
260	0.75	7.5	2.63	26.3
270	0.79	15.8	2.79	55.8
280	0.84	25.2	2.96	88.8
290	0.89	35.6	3.14	125.0
300	0.95	47.5	3.33	166.5

2.3.2 Solar radiation

Solar radiation is a form of thermal radiation having a particular wavelength distribution. At the outer limit of the atmosphere the total solar irradiation when the earth is at its mean distance from the sun is 1395 W/m^2. Not all the energy reaches the surface of the earth, because of strong absorption by carbon dioxide and water vapour.

Since solar radiation is concentrated at short wavelengths, as opposed to much longer wavelengths for most ”earth-bound” thermal radiation, a particular material may exhibit entirely different absorbency and transmittence properties for the two types of radiation LW (long wave radiation) and SW (short wave radiation). The classic example of this behaviour is a green house that in principal does not differ from a truck cabin. Ordinary glass transmits radiation very readily at wavelengths below 2 micrometer; thus it transmits the large part of solar radiation incident upon it. This glass, however, is essentially opaque to long-wave radiation above 3 or 4 micrometer. Practically all the low-temperature radiation emitted by objects inside the cabin is of such a long-wavelength character that it remains trapped in the cabin. Thus the glass allows much more radiation to come in than can escape, thereby producing the well-known heating effect.

The total absorptivity for solar radiation can be quite different from the absorptivity for blackbody radiation at some moderate temperature like 25 ^oC

Examples of how the absorbtivity depends on the wavelength

Surface	SW	LW
Aluminium, highly polished	0.25	0.04
Cast iron	0.94	0.21
Stainless steel no. 301 polished	0.37	0.60
Gravel	0.29	0.85
Flat black lacquer	0.96	0.95
White paint, various types	0.12-0.16	0.90-0.95

2.4 Mass transports

The mass flow is the dominant part of the cabin's heat gain and heat loss when the fan is on. There is also a spontaneous exchange of air between the cabin air and the outdoor air by diffusion. The heat flow looks like

$$\dot{Q} = C_p \dot{m}_f (T_m - T_c) + C_p \dot{m}_d (T_{out} - T_c) \tag{2.44}$$

where $\dot{m}_f$ is mass flow put in by the fan and $\dot{m}_d$ is the mass flow caused by thermal diffusion. $\dot{m}_d = \dot{m}_d (T_c, T_{out}, \dot{m}_f)$ and is difficult to determine.

The α values are identified as

$$\alpha = C_p \dot{m} \qquad [W/K] \tag{2.45}$$

The air mass, put in by the fan, is approximately between 0.04 and 0.16 Kg/s depending on the fan speed. $C_p = 1010 [J/KgK]$ so α has values between 40 and 161 when the fan is on.

The air volume of the cabin is 6.5 m^3 for the topline cabin model and the mass is $m = \rho \cdot V = 1.293 \cdot 6.5 \approx 8.35$ kg which means that 0.6 % to 2.5 % of the air is exchanged every second. That is 36 % to 150 % every minute.

3 Approximations

3.1 One dimensional heat flow

The cabin is made up by a number of walls that can be subdivided into wall elements with almost uniform temperature distribution at the surfaces. Because the thickness of the wall elements are thin compared to their cross section areas, the heat transfer from the neighbouring wall elements is so small compared to the amount of heat that flows through the cross section that it can be neglected. Thus only one dimensional heat flow is used in this model.

The assumption is justified by the observation that the heat flow is proportional to the area. Hence if the wall piece is one square metre with a thickness of one millimetre the heat flow parallel to the wall is only one thousandth of the flow orthogonal to the wall.

3.2 Finite number of states

The non-linearity in the convection and radiation heat transfer makes it very difficult to calculate the temperature distribution and a simulation computer program will be used. If we transform the heat equation into the state space form through a change in basis we will end up with an infinite number of states because every point in space is associated with a temperature and is of importance for the future behaviour of the system. The number of states must be reduced before the use of a simulation program is possible.

3.2.1 A wall with three states

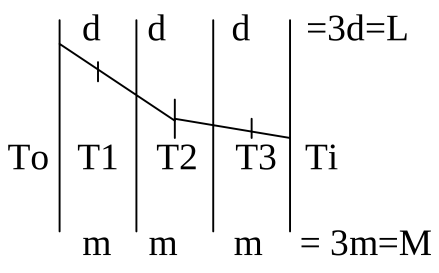

Figure 3.1 shows the temperature distribution for a homogenous wall at a non-steady state.

In steady state the temperature distribution will form a strait line as shown in chapter 2 equation 2.14
The wall has three temperature states at the positions 1/6, 1/2 and 5/6 of the wall thickness.

The temperature distribution is approximated in that way that it is supposed to be linear between the three state positions. Beyond the two outer states the temperature distribution continuos in a strait line and the surface temperatures is approximated by

$$T_o = \tfrac{3}{2}T_1 - \tfrac{1}{2}T_2$$
$$T_i = \tfrac{3}{2}T_3 - \tfrac{1}{2}T_2 \qquad (3.1), (3.2)$$

The amount of energy contained in the wall is approximated as

$$E = C_p \frac{M}{3}(T_1 + T_2 + T_3) = C_p m(T_1 + T_2 + T_3) \qquad (3.3)$$

which is exact in steady state.

The energy balance equation for a layer looks like

$$\tfrac{d}{dt}(amount\_of\_heat\_in\_the\_layer) = (power\_in) - (power\_out) \qquad (3.4)$$

that is for layer i, using $U_i = C_v \cdot m_i \cdot T_i$ (2.4) and the steady state approximation $\dot{Q}_{ji} = \alpha \cdot A \cdot (T_j - T_i)$ (2.1).

$$C_p \cdot m \cdot \dot{T}_i = \sum_j \dot{Q}_{ji} \qquad (3.5)$$

For the three layers we get the following system of linear first order differential equations

$$C_p m \dot{T}_1 = \tfrac{\lambda \cdot A}{d/2}(T_0 - T_1) + \tfrac{\lambda \cdot A}{d}(T_2 - T_1)$$
$$C_p m \dot{T}_2 = \tfrac{\lambda \cdot A}{d}(T_1 - T_2) + \tfrac{\lambda \cdot A}{d}(T_3 - T_2) \qquad (3.6)$$
$$C_p m \dot{T}_3 = \tfrac{\lambda \cdot A}{d}(T_2 - T_3) + \tfrac{\lambda \cdot A}{d/2}(T_i - T_3)$$

We regroup the parameters as

$$\tfrac{\lambda \cdot A}{C_v \cdot m \cdot d} = \tfrac{\lambda \cdot A}{C_v \cdot m \cdot d} \cdot \tfrac{d}{d} = \tfrac{\lambda}{C_v \cdot \rho} \cdot \tfrac{1}{d^2} = \{L = 3d\} = \tfrac{9 \cdot \kappa}{L^2} \qquad (3.7)$$

Using that and putting the equations on matrix form gives

$$\dot{T} = AT + Bu \tag{3.8}$$

where

$$A = \tfrac{9\cdot\kappa}{L^2}\begin{bmatrix} -3 & 1 & 0 \\ 1 & -2 & 1 \\ 0 & 1 & -3 \end{bmatrix} \tag{3.9}$$

$$B = \tfrac{9\cdot\kappa}{L^2}\begin{bmatrix} 2\,0 \\ 0\,0 \\ 0\,2 \end{bmatrix} \tag{3.10}$$

$$u = \begin{bmatrix} T_0 \\ T_i \end{bmatrix} \tag{3.11}$$

According to Glad and Ljung's book Reglerteori [3] the solution to (3.8) is

$$T(t) = e^{A(t-t_o)}T(t_0) + \int_{t_0}^{t} e^{A(t-\tau)}Bu(\tau)d\tau \tag{3.12}$$

To be able to make a comparison with the heat equation solution below a calculation has been made showing what will happen if the wall is at a uniform temperature T = 0 K and then a temperature step is applied at one of the surfaces. Formally

$$T = \overline{0} \text{ and } u = \overline{0} \text{ for } t < 0 \text{ and } u = \begin{bmatrix} \Delta T \\ 0 \end{bmatrix} \text{for } t \geq 0 \tag{3.13}$$

The solution to (3.8) with the conditions (3.13) is

$$\begin{aligned} T_1(t) &= \Delta T\left[\tfrac{1}{6}(1-e^{-4\cdot 9\cdot\frac{\kappa}{L^2}t}) + \tfrac{1}{3}(1-e^{-3\cdot 9\cdot\frac{\kappa}{L^2}t}) + \tfrac{1}{3}(1-e^{-1\cdot 9\cdot\frac{\kappa}{L^2}t})\right] \\ T_2(t) &= \Delta T\left[\tfrac{-1}{6}(1-e^{-4\cdot 9\cdot\frac{\kappa}{L^2}t}) + \tfrac{2}{3}(1-e^{-1\cdot 9\cdot\frac{\kappa}{L^2}t})\right] \\ T_3(t) &= \Delta T\left[\tfrac{1}{6}(1-e^{-4\cdot 9\cdot\frac{\kappa}{L^2}t}) - \tfrac{1}{3}(1-e^{-3\cdot 9\cdot\frac{\kappa}{L^2}t}) + \tfrac{1}{3}(1-e^{-1\cdot 9\cdot\frac{\kappa}{L^2}t})\right] \end{aligned} \tag{3.14}$$

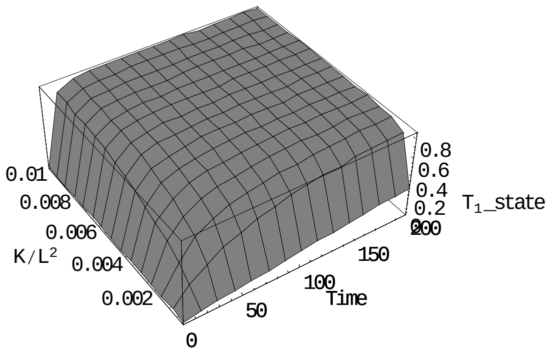

Figure 3.3 shows the step response for T_1 as a function of κ/L^2 when $\Delta T = 1$. The final value is 5/6.

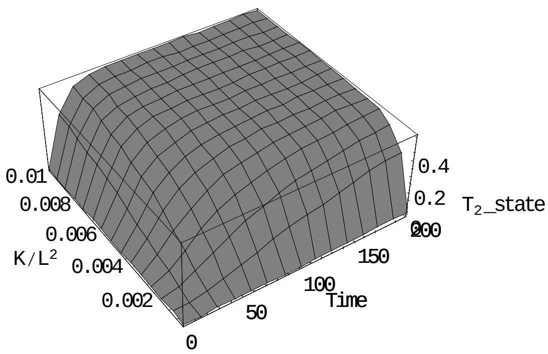

Figure 3.4 shows the step response for T_2 as a function of κ/L^2 when $\Delta T = 1$. The final value is 1/2.

3.2.2 The heat equation solution

To be able to see if a three-state wall is a good approximation, a calculation of the temperature and heat flow according to the heat equation, is made.

From the beginning the wall has a uniform temperature T_0, but then a temperature step ΔT is applied at one side while the other side is maintained at temperature T_0. Below follows a calculation of what effect this will have and later on it is compared to the step response of the state equation approximation.

The heat equation without internal heat source looks like

$$\frac{\partial T}{\partial t} = \kappa \nabla^2 T \tag{3.15}$$

constraints $t > 0$, $0 < x < L$.

The boundary conditions are

$$\begin{aligned} T(0,t) &= T_0 + \Delta T \\ T(L,t) &= 0 \end{aligned} \quad \text{for } t > 0 \tag{3.16}$$

And the initial condition is $T(x,0) = T_0$ for $0 \le x \le L$ (3.17)

Assuming that the temperature distribution can be written on the form

$$T(x,t) = X(x)\cdot\theta(t) + \xi(x),\ \xi''(x) = 0 \text{ for all x} \tag{3.18}$$

where the variable separation technique is used.
Using the result from chapter 2 that the solution in steady state will be on the form

$$T(x) = Cx + D \tag{3.19}$$

we derive, using (3.16) that

$$T(x,\infty) = -\frac{\Delta T \cdot x}{L} + \Delta T + T_o$$

and if $\xi(x) = T(x,\infty)$ we will have a new equation for $T_{X\theta} = X(x)\cdot\theta(t)$

The boundary conditions becomes $T_{X\theta}(0,t) = T_{X\theta}(L,t) = 0$ (3.20)

And the initial condition is $T_{X\theta}(x,0) = \frac{\Delta T \cdot x}{L} - \Delta T$ (3.21)

Inserting $T_{X\theta}$ into the heat equation gives

$$X(x) \cdot \theta'(t) = \kappa \cdot X''(x) \cdot \theta(t) \quad (3.22)$$

and dividing with $\kappa \cdot X(x) \cdot \theta(t)$ on both sides gives

$$\frac{\theta'(t)}{\kappa \cdot \theta(t)} = \frac{X''}{X} = \gamma \text{ , } \gamma = \text{separation constant.} \quad (3.23)$$

$$(3.19) \Rightarrow \begin{matrix} X'' - \gamma X = 0 \\ X(0) = X(L) = 0 \end{matrix} \Big\} \Rightarrow \Big\{ \begin{matrix} X_n(x) = \sin(\frac{n\pi x}{L}), n = 1,2,3... \\ \gamma_n = -\frac{n^2\pi^2}{L^2}, n = 1,2,3,... \end{matrix} \quad (3.24)$$

By (3.22) $\theta' + \frac{\kappa n^2\pi^2}{L^2}\theta = 0 \Rightarrow \theta_n(t) = C_n e^{-\kappa n^2\pi^2 t / L^2}$ (3.25)

Set $T_{X\theta}(x,t) = \sum_{n=1}^{\infty} X_n(x) \cdot \theta_n(t) = \sum_{n=1}^{\infty} C_n \sin(\frac{n\pi x}{L}) e^{-\kappa n^2\pi^2 t / L^2}$ (3.26)

By (3.20)
$$\frac{\Delta T \cdot x}{L} - \Delta T = \sum_{n=1}^{\infty} C_n \sin(\frac{n\pi x}{L}) \Rightarrow$$
$$C_n = \frac{2}{L}\int_0^L (\frac{\Delta T \cdot x}{L} - \Delta T) \sin(\frac{n\pi x}{L}) dx = \frac{-2 \cdot \Delta T}{n \cdot \pi} \quad (3.27)$$

So $T_{X\theta}(x,t) = \sum_{n=1}^{\infty} \frac{-2 \cdot \Delta T}{n \cdot \pi} \sin(\frac{n\pi x}{L}) e^{-\kappa n^2\pi^2 t / L^2}$ (3.28)

and finally

$$T(x,t) = -\frac{\Delta T \cdot x}{L} + \Delta T + T_o + \sum_{n=1}^{\infty} \frac{-2 \cdot \Delta T}{n \cdot \pi} \sin(\frac{n\pi x}{L}) e^{-\kappa n^2\pi^2 t / L^2} \quad (3.29)$$

Some figures showing the step response when $\Delta T = 1$

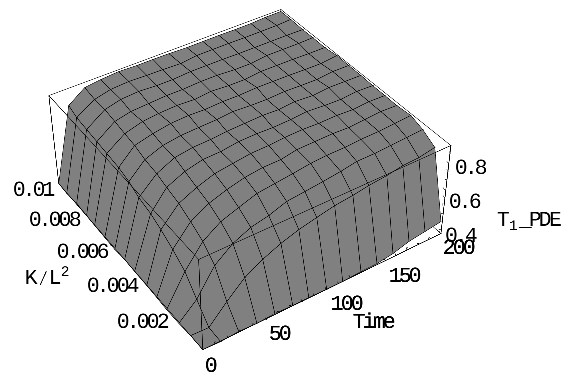

Figure 3.5 shows T(5/6,t) with κ / L^2 as a parameter.

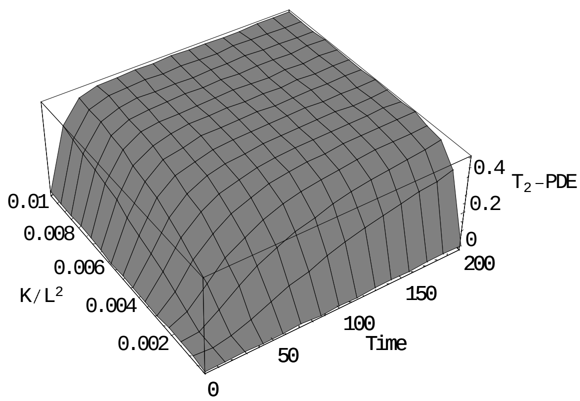

Figure 3.6 shows T(1/2,t) with κ / L^2 as a parameter.

3.2.3 Comparisons between the heat equation solution and the state space solution

Below there are some plots showing the difference between the PDE (partial differential equation) solution and the state equation approximation solution. Observing that both solutions (3.14) and (3.29) are directly proportional to the step ΔT we conclude that the deviation between the exact analytical solution and the state space approximation is directly proportional to the step height ΔT. The deviation between the two solutions will then be of the form $T_{difference} = \Delta T \cdot f(t)$. In the figures below $\Delta T = 1$.

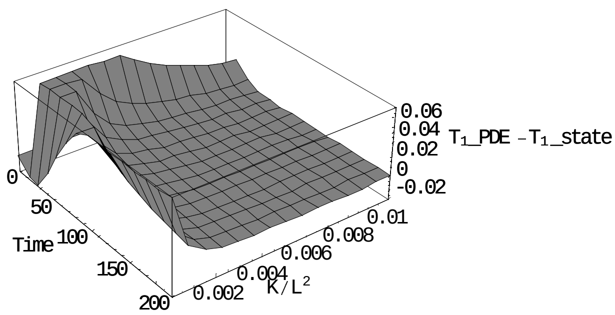

Figure 3.7 shows $T_{PDE}(\frac{5}{6},t) - T_{1,state}(t)$ for different values of κ/L^2.

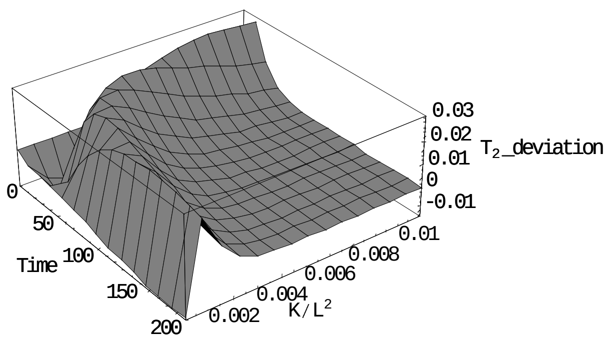

Figure 3.8 shows $T_{PDE}(\frac{1}{2},t) - T_{2.state}(t)$ for different values of κ/L^2.

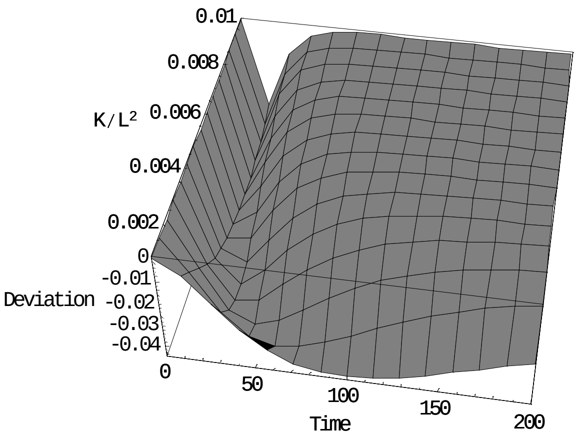

Figure 3.9 shows $T_{PDE}(\frac{5}{6},t) - T_{3.state}(t)$ for different values of κ/L^2.

Calculating the temperature of the side that should remain at T=0 using (3.2) is shown in the above figure. The temperature is first lowered but returns to the value 0. The maximum deviation seems to be approximately 5 % of ΔT. Because the change in temperature is in fact very slow the deviation will be much less.

3.3 Radiation approximations

3.3.1 Air absorption of thermal radiation

Air, mainly consisting of N_2 and O_2 is taken to be transparent to thermal radiation. There is however a small amount of water vapour and carbon dioxide which more easily absorbs radiation but that effect is neglected in this model.

3.3.2 Diffusive radiation

Two types of reflection phenomena may be observed when radiation reaches a surface. If the angle of incidence is equal to the angle of reflection, the reflection is called specular. On the other hand, when an incident beam is distributed uniformly in all directions after reflection, the reflection is called diffuse. No real surface is either specular or diffuse.

In this model all reflection is assumed to be diffuse. That is because it is a good approximation for long wave radiation and introducing a direction in the short wave radiation from the sun will probably have a small effect on the global temperature. Nevertheless the direction is of great importance if one wish to say something about the drivers comfort which is beyond the scope of this work.

3.3.3 View factors

The calculation of radiative exchange between two surfaces requires a quantity that describes the influence of their position and orientation. This is the view factor. The view factor indicates to what extent one surface can be "seen" by another, or more exactly what proportion of the radiation from surface 1 falls on surface 2. Making the assumptions that the radiation is diffusive and that the intensity of radiation emitted is constant over the entire surface 1, one can derive the view factor as

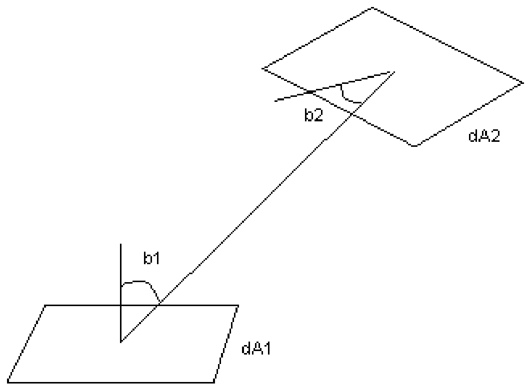

Figure 3.10 shows the radiative exchange between two surface elements.

$$F_{12} = \frac{1}{\pi A_1} \iint_{A_1 A_2} \frac{\cos\beta_1 \cos\beta_2}{r^2} dA_2 dA_1 \qquad (3.30)$$

and

$$F_{21} = \frac{1}{\pi A_2} \iint_{A_1 A_2} \frac{\cos\beta_1 \cos\beta_2}{r^2} dA_2 dA_1 \qquad (3.31)$$

The orientation of the surface elements to the straight line between them is of importance. This is expressed in terms of a cosine function of the two polar angles β_1 and β_2.

The two equations (3.30) and (3.31) provides the important reciprocity rule for view factors.

$$A_1 F_{12} = A_2 F_{21} \qquad (3.32)$$

A further relation can be found when n areas form an enclosure. Because the radiation from a surface j must fall on these n surfaces one has the following condition

$$\sum_{j=1}^{n} F_{ij} = 1 \quad , \; i = 1,2,...,n \qquad (3.33)$$

In the model used n=24. The number of quadruple integrals that need to be calculated is n*n=264. Taking into account that each quadruple integral needs 8 co-ordinates specified one should need to input 2 112 co-ordinates which is time consuming and does not go hand in hand with the aim that the model should be easy to modify.

Because of all the reflections inside the cabin the view factors will not need to be exact and the main effect of the radiation inside the truck will be that it reduces the temperature differences between the interior walls.
I have therefore made the following approximation of how much of the radiation from area i that hits area j.

$$F_{ij} = \frac{A_j}{\sum_k A_k} \qquad (3.34)$$

That is the radiation emitted from A_i that is absorbed by A_j is direct proportional to how large part of the total interior area A_j is.

It can easily be seen that this approximation fulfils the two conditions (3.32) and (3.33).

3.4 Conclusions

To model a wall with a state space model with three states is a good approximation to the heat equation. In steady state both the solutions give the same result and for sudden changes the difference are so small that it can be neglected. As an example for a step the deviation is proportional to the step height ΔT and is of the magnitude $\Delta T \cdot 1.05$. Because the temperature in reality will change slowly the model fault because of the state space approximation is small.

To let the radiation be diffuse is a good approximation because the indoor surfaces are mostly mat. The radiation exchange approximation fulfils the condition that the energy should be conserved and that the temperature differences between the surfaces decreases.

The approximations made in this chapter will not degrade the model in any important way.

4 Main structure of the model

The model is implemented in Simulink, a simulation program developed by Math Works. It has

- 135 different temperature states
- 9 inputs
- 8 outputs
- 6 subroutine levels excluding the build in Simulink blocks
- Global parameters as the dimensions and the material data are set separate Matlab files.

4.1 Inputs and Outputs

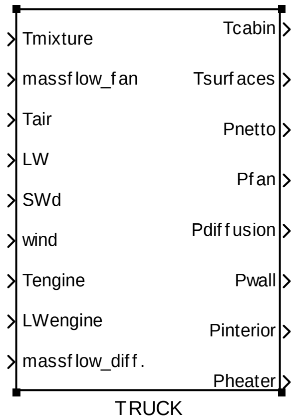

Figure 4.1 The TRUCK block

Inputs

- “Tmixture” is the temperature of the air that is put in by the fan.
- “Massflow_fan” is the amount of mass put in by the fan.
- “Tair” is the temperature of the outside air.
- “LW” is the amount of LW radiation incident on 1 m^2 of the truck each second.
- “SWd” is the amount of SWd radiation incident on 1 m^2 of the truck each second.
- “wind” is the speed of the air that flows around the truck.
- “Tengine” is the temperature of the engine.
- “Lwengine” is the amount of LW radiation incident on the engine tunnel per square metre and second
- “Massflow_diff” is the amount of mass that is exchanged directly between the cabin air and the outside air.

Outputs

- "Tcabin" is the temperature of the cabin air.
- "Tsurface" is the temperatures of the 24 inside walls
- "Pnetto" is the net energy that the cabin air absorbs every second.
- "Pfan" is the effect put in by the fan $P_{fan} = C_p \dot{m}_f (T_m - T_c)$
- "Pdiffusion" is the effect put in by diffusion $P_{diff} = C_p \dot{m}_d (T_{air} - T_c)$
- "Pwall" is the heat flow through the walls including the floor that may be heated by the engine.
- "Pinterior" is the heat-flow from the interior to the cabin air.
- "Pheater" is the effect on the heater $P_h = C_p \dot{m}_f (T_m - T_{air})$

4.2 Presentation of the model

4.2.1 Introduction to the upper level blocks

To be able to go deeper into the model a brief introduction is made to some important blocks. Their internal structure is described in chapter 7

W is a block that models a one material wall that has a material with good conductivity where the temperature can then be assumed to the same everywhere in the wall. The block has one state.

SIS is a block that models a one material wall which is divided into three equally thick layers so it will be possible to model a temperature propagation in a good way when the temperature is non-uniform. It consists of three blocks. Two **Solidsurface** blocks and one **Insidesolid** block. Each block has one state and as a total the SIS block has three states.

SIN is a block that models a one material wall layer that is divided into three equally thick layers. It includes three blocks with one state each, one **Solidsurface** block, one **Insidesolid** block and one **Neighbour** block. The Neighbour block is there to make it possible for the wall layer to interact with another wall layer made by a different material.

NIN models a one material wall layer that is divided into three equally thick layers. The **NIN** block is used when the wall layer has two neighbouring materials layers. It has three states.
The **GIG** block models a glass window. It is built up by two **Glass_surface** blocks and one **Insideglass** block. GIG has three states.

The **isolationAir** block is labelled **A** and has two states. One is the air temperature and the other one is the temperature of the steel beams that is placed in the isolation air area.

In the next level we have combinations of the above blocks.

SINNIS consists of two SIN blocks. It models a wall with two different material layers. The SINNIS block has six states.

A wall with four different material layers should be modelled by inserting two NIN blocks between the two SIN blocks. The **SINNINNINNIS** block is then created. The SINNINNINNIS block has 12 states.

WAW is made up by to W blocks and one isolationAir block. WAW models a double wall with isolation air.

SISASIS consist of two SIS blocks and one isolationAir block. This to be able to model the heat flow through a double wall with isolation air between them. The SISASIS block has eight states

WAW is made up by to W blocks and one isolationAir block. WAW models a double wall with isolation air. WAW has 4 states. WAW is used instead of SISASIS when the walls are good conductors and has a uniform temperature.

WASIS is a combination between a WAW and a SISASIS block and is used when one of the walls has high conductivity and the other has low conductivity. It has 6 states.

Exchanging one of the SIS blocks for a SINNINS block makes a **SISASINNIS** block with 10 states.

4.2.2 Main parts of the model

The main parts of the model are shown below and is built up by the blocks

- Roof,
- Front
- Floor
- Back
- Side Left/Right
- Box
- Cabin

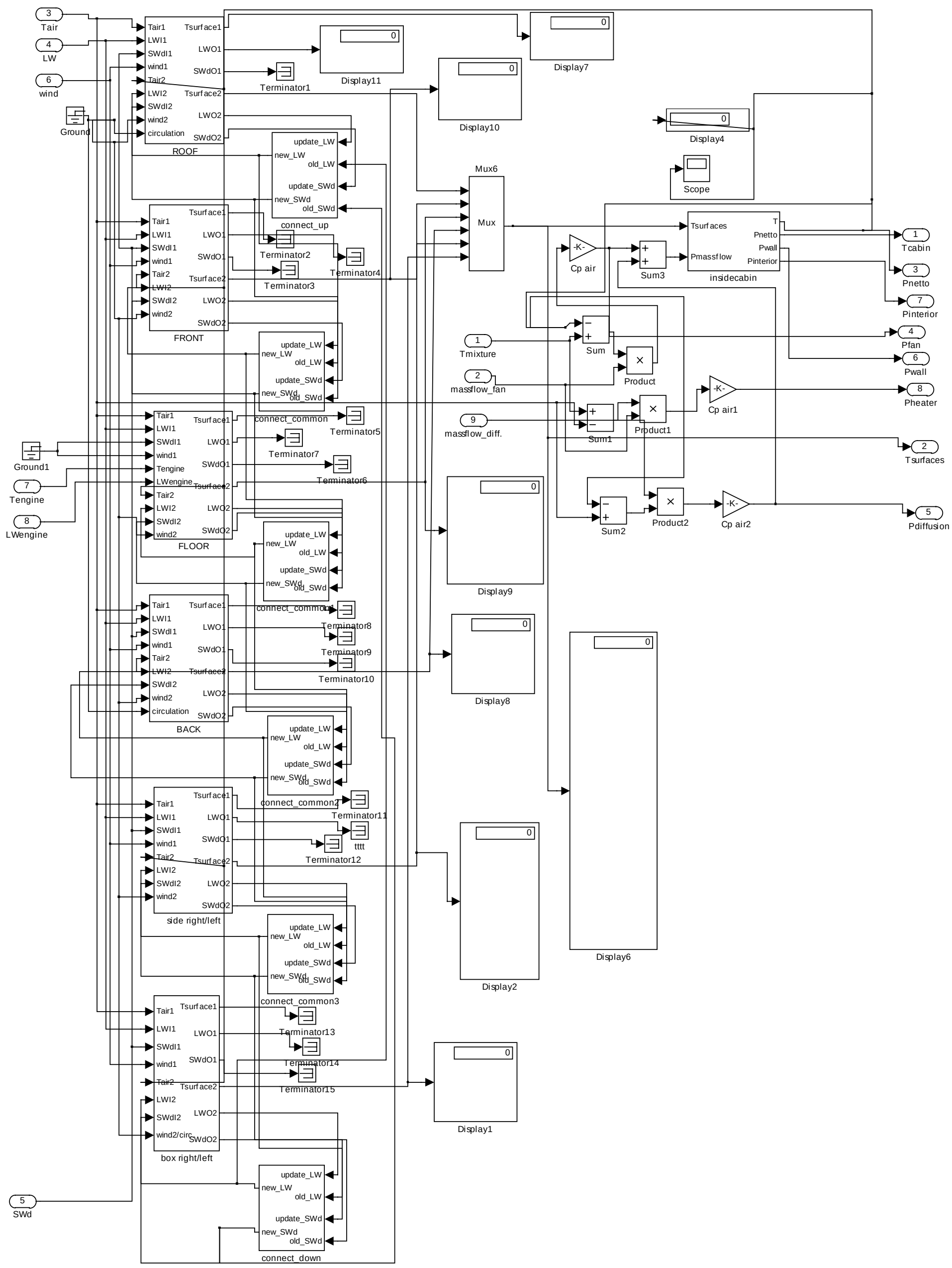

Figure 4.2 Inside the TRUCK block.

4.2.2.1 Roof

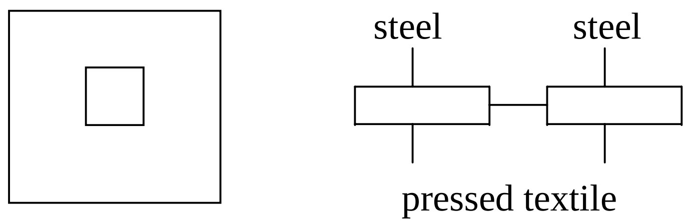

fig 4.3 The roof.

Most of the roof is a double wall with one steel layer and one pressed textile layer with isolation air between them. A SISASIS block models it. There is also a roof window, which is modelled by a GIG block.

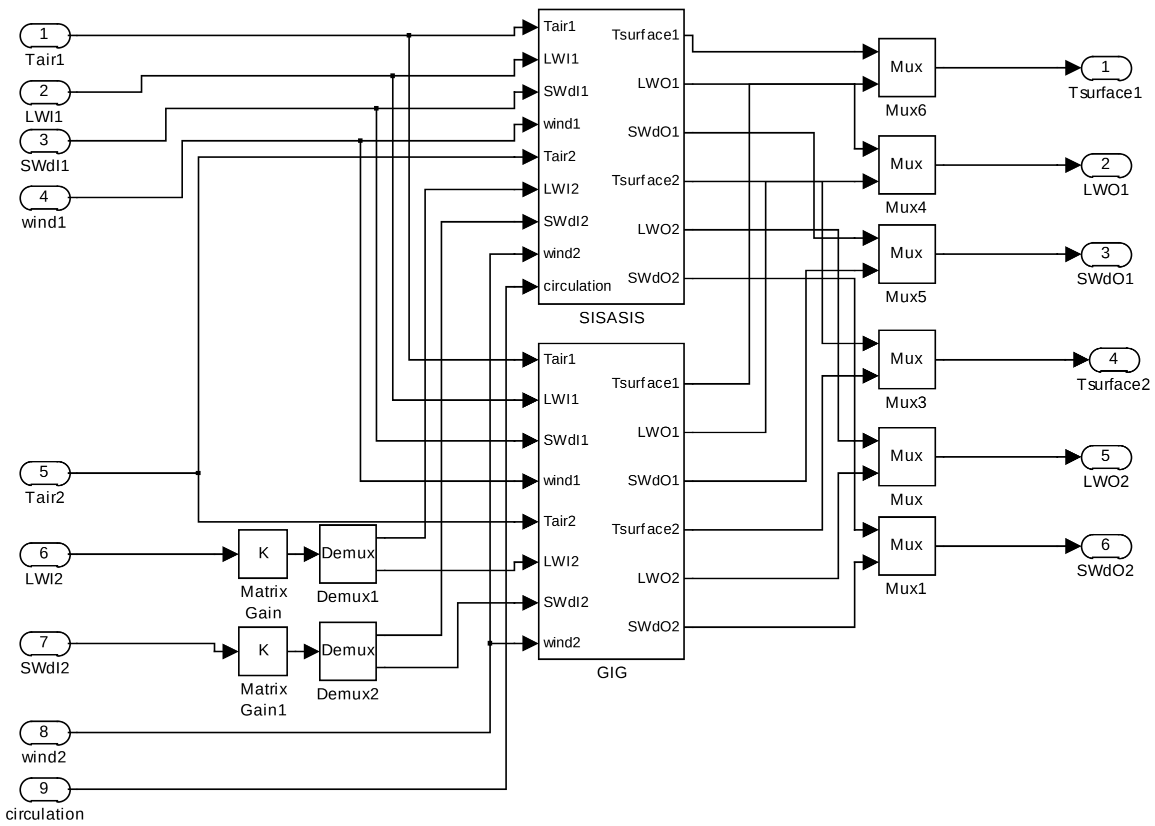

Figure 4.4 Inside the ROOF block

Parameters for the ROOF block.

ROOF

Block	Prompt	Value
SISASIS	Tzero	Tzero
	[area]	[area(1) area(1)]
	[thickness]	[0.001 0.05 0.01]
	[dist. from leading edge]	[1 1]
	F vector 1	area(1)
	F vector 2	1
	up/down/vert	up
	up/down/vert	down
	material properties 1	steel_data
	material properties 2	wood_data (pressed….
GIG	Tzero	Tzero
	area	0.16
	thickness	0.005
	[dist. from leading edge]	[1 1]
	F vector 1	0.16
	Fvector 2	1
	material properties	glass_data
	up/down/vert 1	up
	up/down/vert 2	down

4.2.2.2 Front

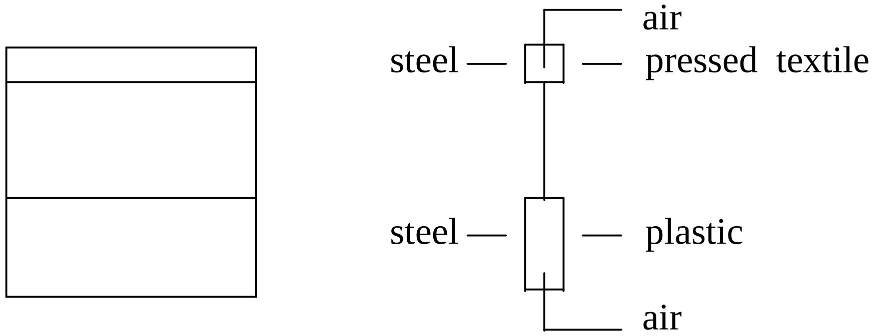

Figure 4.5 The front.

The front has three main parts. The upper part is a double wall with one steel layer and one pressed textile layer with isolation air between them. A SISASIS block models it. In the middle there is a front window modelled by a GIG block. The lower part is a double wall consisting of one steel layer and one plastic layer, with isolation air.

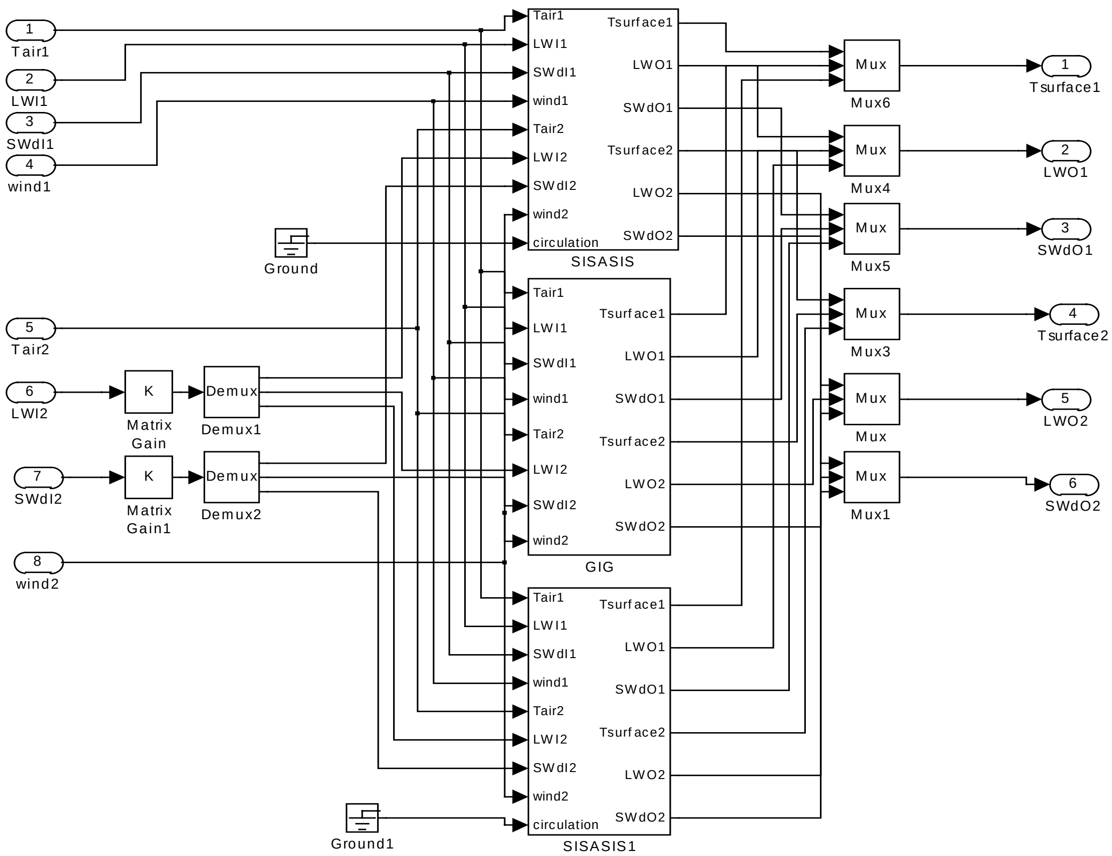

Figure 4.6 Inside the FRONT block

Parameters for the FRONT block

FRONT

Block	Prompt	Value
SISASIS	Tzero	Tzero
	[area]	[area(3) area(3)]
	[thickness]	[0.001 0.05 0.01]
	[dist. from leading edge]	[1 1]
	F vector 1	area(3)
	F vector 2	1
	up/down/vert	vert
	up/down/vert	vert
	material properties 1	steel_data
	material properties 2	wood_data
GIG	Tzero	Tzero
	area	area(4)
	thickness	0.0065
	[dist. from leading edge]	[1 1]
	F vector 1	area(4)
	Fvector 2	1
	material properties	glass_data
	up/down/vert 1	vert
	up/down/vert 2	vert
SISASIS1	Tzero	Tzero
	[area]	[area(5) area(5)]
	[thickness]	[0.001 0.5 0.005]
	[dist. from leading edge]	[1 1]
	F vector 1	area(5)
	F vector 2	1
	up/down/vert	vert
	up/down/vert	vert
	material properties 1	steel_data
	material properties 2	plastic_data

4.2.2.3 Floor

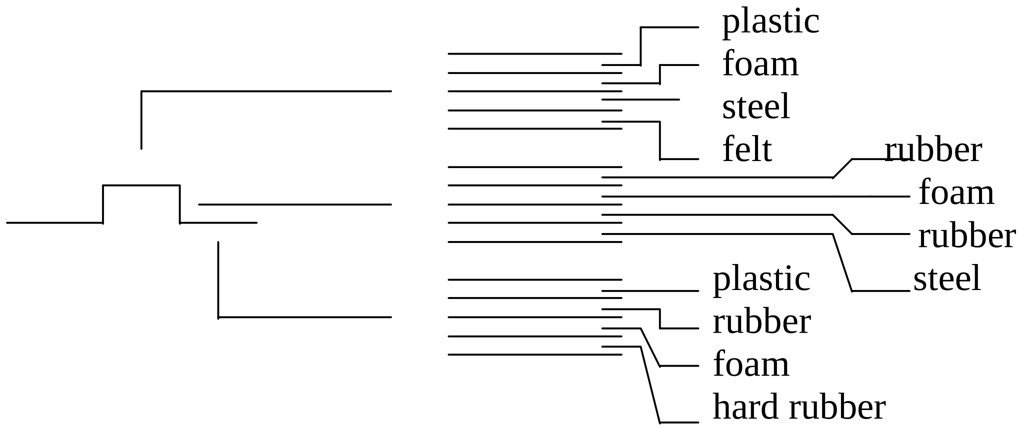

Figure 4.7 The floor structure.

The floor consists of five parts where the approximation made is that the temperatures of the left-hand side equals the right hand side. This results in three different parts. First we have the floor which is a four-layer wall. The materials are from the bottom steel, rubber, foam, rubber.

Secondly we have the vertical walls of the engine tunnel. It is a four-layer wall with the materials felt, steel, foam and rubber.

Third there is the roof of the engine tunnel which is also a four layer wall made up by felt, steel foam and plastic.

The three floor blocks are SINNINNINNIS blocks.

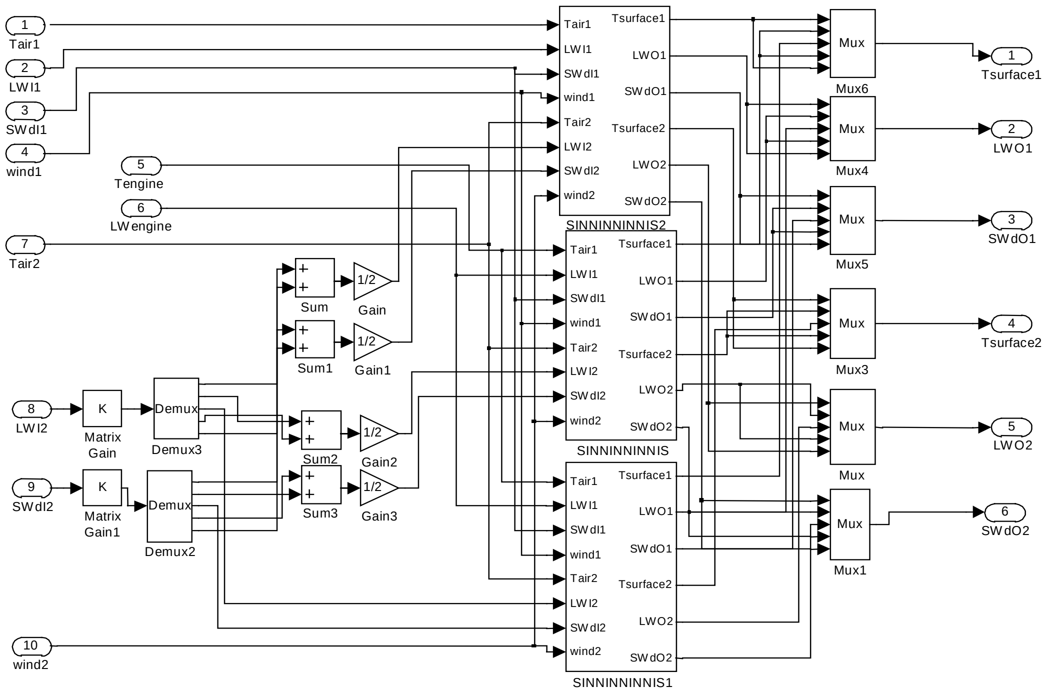

Figure 4.8 Inside the floor block

Parameters for the FLOOR block

Block	Prompt	Value
SIN..NIS2	Tzero	Tzero
	area	0.55
	[thickness]	0.0015 0.004 0.02 0.01]
	[dist. from leading edge]	[1 1]
	F vector 1	0.55
	F vector 2	1
	up/down/vert 1	down
	up/down/vert 2	up
	material properties 1	steel_data
	material properties 2	rubber_data
	material properties 3	foam_data
	material properties 4	rubber_data
SIN..NIS	Tzero	Tzero
	area	0.4
	[thickness]	[0.001 0.001 0.02 0.001]
	[dist. from leading edge]	[1 1]
	F vector 1	0.4
	F vector 2	1
	up/down/vert 1	vert
	up/down/vert 2	vert
	material properties 1	wood_data
	material properties 2	steel_data
	material properties 3	foam_data
	material properties 4	rubber_data
SIN..NIS1	Tzero	Tzero
	area	0.65
	[thickness]	[0.001 0.001 0.02 0.001]
	[dist. from leading edge]	[1 1]
	F vector 1	0.65
	F vector 2	1
	up/down/vert 1	down
	up/down/vert 2	up
	material properties 1	wood_data
	material properties 2	steel_data
	material properties 2	foam_data
	material properties 2	plastic_data

4.2.2.4 Back

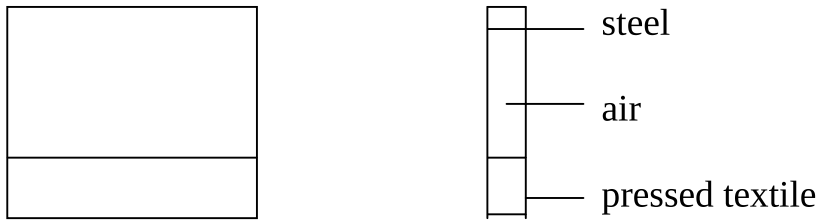

Figure 4.9 The back

The back of the cabin is modelled by two SISASIS blocks. It is a double wall made up by one steel and one pressed textile layer with isolation air between them. It is divided into two parts so that one part is covering the ventilation openings and one part is below the openings.

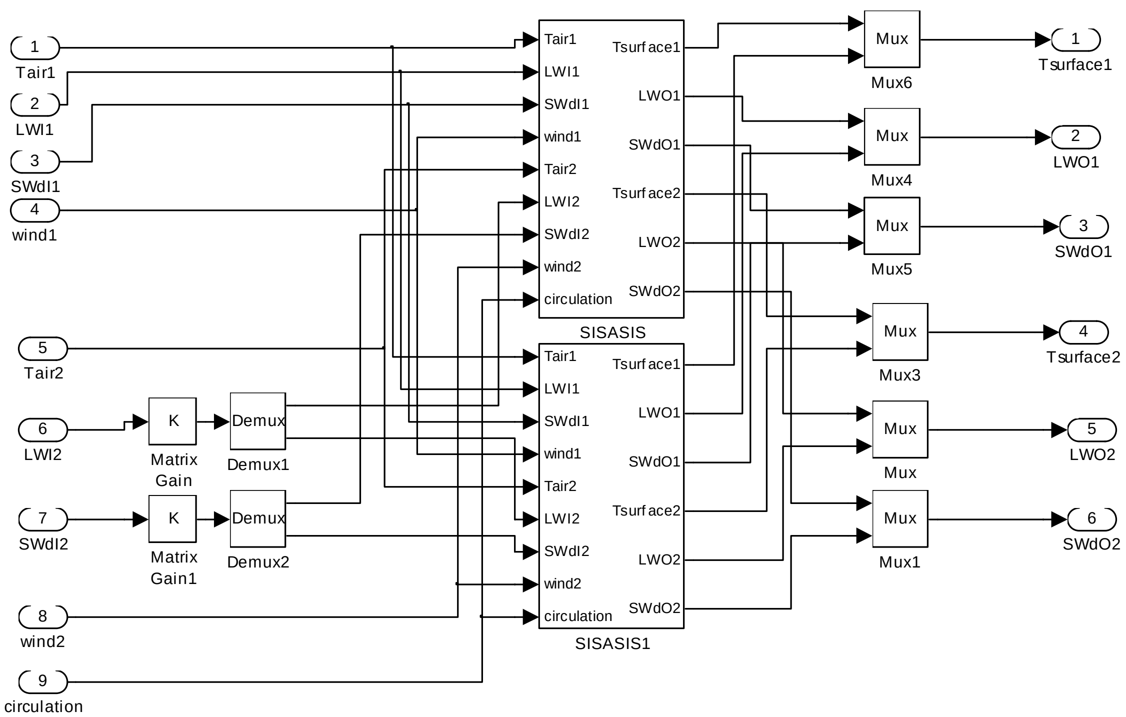

Figure 4.10 Inside the BACK block

Parameters for the BACK block

Block	Prompt	Value
BACK		
SISASIS	Tzero	Tzero
	[area]	[area(11 area(11)]
	[thickness]	[0.001 0.05 0.01]
	[dist. from leading edge]	[1 1]
	F vector 1	area(11)
	F vector 2	1
	up/down/vert	vert
	up/down/vert	vert
	material properties 1	steel_data
	material properties 2	wood_data
SISASIS	Tzero	Tzero
	[area]	[area(12) area(12)]
	[thickness]	[0.001 0.05 0.01]
	[dist. from leading edge]	[1 1]
	F vector 1	area(12)
	F vector 2	1
	up/down/vert	vert
	up/down/vert	vert
	material properties 1	steel_data
	material properties 2	wood_data

4.4.4.5 Side

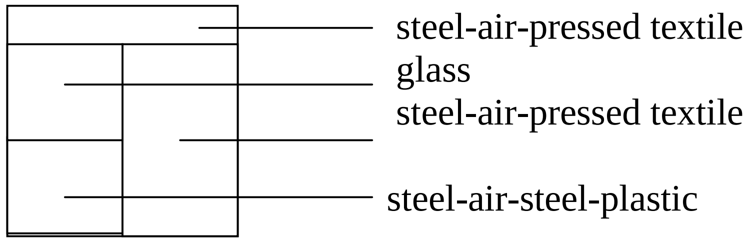

Figure 4.11 The (left) side.

Both sides of the truck are modelled by the same block to reduce the number of states. The truck side is divided into four different areas and these are

- At the top is a SISASIS block, (steel, air and pressed textile) to be able to model a topline addition.
- Then there is the area behind the door, which is a SISASIS block with the same structure as the previous block.
- The door is modelled by a SISASINNIS block with the layers: steel, air, steel and plastic.
- Finally the door window is modelled by a GIG block.

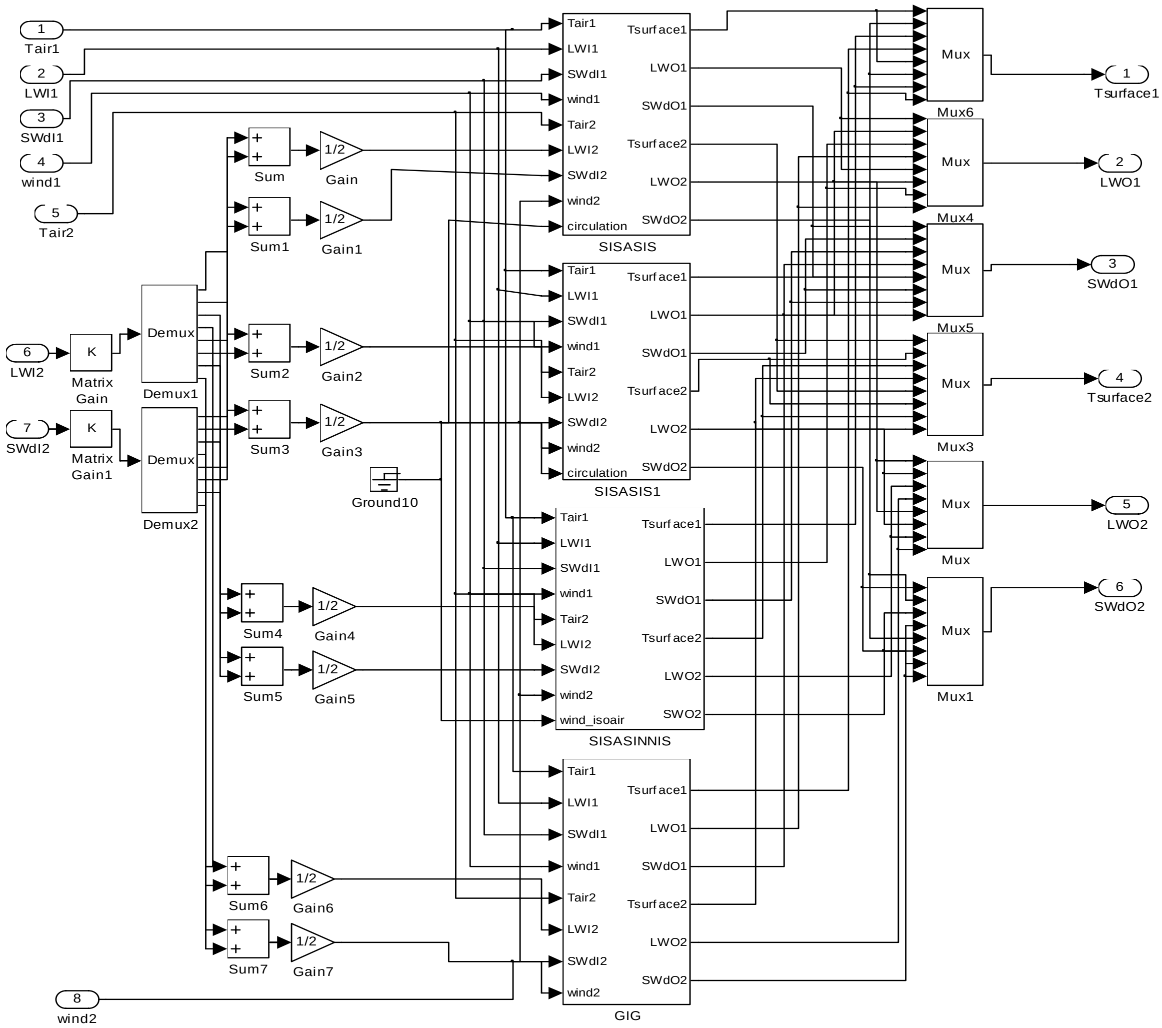

Figure 4.12 Inside the side left/right block

Parameters for the side_left/right block

Block	Prompt	Value	
Side_Left/Right			
SISASIS	Tzero	Tzero	
	[area]	[area(13) area(13)]	
	[thickness]	[0.001 0.05 0.01]	
	[dist. from leading edge]	[1 1]	
	F vector 1	area(13)	
	F vector 2	1	
	up/down/vert	vert	
	up/down/vert	vert	
	material properties 1	steel_data	
	material properties 2	wood_data	
SISASIS1	Tzero	Tzero	
	[area]	[area(14) area(14)]	
	[thickness]	[0.001 0.05 0.01]	
	[dist. from leading edge]	[1 1]	
	F vector 1	area(14)	
	F vector 2	1	
	up/down/vert	vert	
	up/down/vert	vert	
	material properties 1	steel_data	
	material properties 2	wood_data	
SISASINNIS	Tzero	Tzero	global air_data steel_data
	[area]	[area(15) area(15)]	
	[thickness]	[0.001 0.07 0.001 0.001]	
	[dist. from leading edge]	[1 1]	
	F vector 1	area(15)	
	F vector 2	1	
	up/down/vert 1	vert	
	up/down/vert 2	vert	
	material properties 1	steel_data	
	material properties 2	steel_data	
	material properties 3	plastic_data	
GIG	Tzero	Tzero	global vert
	area	area(16)	
	thickness	0.004	
	[dist. from leading edge]	[1 1]	
	F vector 1	area(16)	
	Fvector 2	1	
	material properties mp	glass_data	
	up/down/vert 1	vert	
	up/down/vert 2	vert	

4.2.2.6 Box

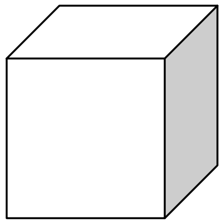

Figure 4.13 A box.

The box block models the storage compartments under the bed. It has six sides just like a box of which two faces the interior of the cabin. The other four is the back, the floor, the engine tunnel and the side.

- The roof part is a SINNIS block modelling a bed with one plastic layer and one plastic foam layer.
- The front part is a plain plastic wall modelled by a SIS block
- The floor part is a SINNINNINNIS block to model the four-layered floor.
- The back part is a SISASIS block.
- The outer side part is a steel SIS block.
- The inner side part is a SINNINNINNIS block.

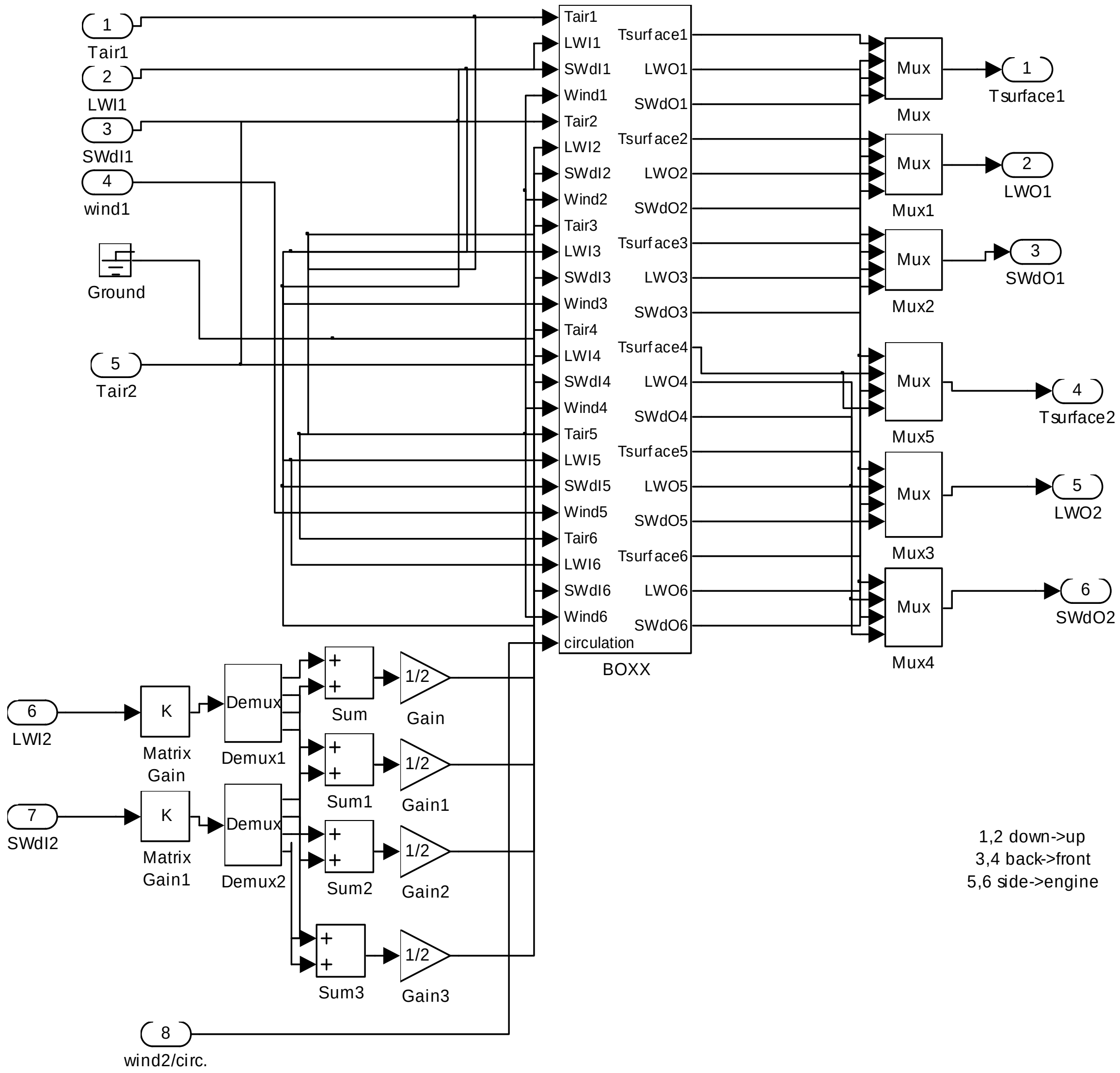

Figure 4.14 Inside the box right /left block

Parameters for the box right/left block

BOX RIGHT/LEFT

Block	Prompt	Value
box right/left	Tzero	Tzero
SINNIS	Tzero	Tzero
	area	area(21)
	[thickness]	[0.001 0.004]
	[dist. from leading edge]	[1 1]
	F vector 1	area(21)
	F vector 2	1
	up/down/vert 1	down
	up/down/vert 2	up
	material properties 1	steel_data
	material properties 2	rubber_data

SINNIS1	Tzero	Tzero	
	area	area(21)	
	[thickness]	[0.004 0.004]	
	[dist. from leading edge]	[1 1]	
	F vector 1	1	
	F vector 2	1	
	up/down/vert 1	down	
	up/down/vert 2	up	
	material properties 1	plastic_data	
	material properties 2	rubber_data	
SISASIS	Tzero	Tzero	
	[area]	[area(22) area(22)]	
	[thickness]	[0.001 0.04 0.001]	
	[dist. from leading edge]	[1 1]	
	F vector 1	area(22)	
	F vector 2	1	
	up/down/vert	vert	
	up/down/vert	vert	
	material properties 1	steel_data	
	material properties 2	wood_data	
SIS	Tzero	Tzero	
	area	area(22)	
	thickness	0.005	
	[dist. from leading edge]	[1 1]	
	F vector 1	1	
	Fvector 2	1	
	material properties mp	plastic_data	
	up/down/vert 1	vert	
	up/down/vert 2	vert	
SIS1	Tzero	Tzero	
	area	0.3	
	thickness	0.005	
	[dist. from leading edge]	[1 1]	
	F vector 1	1	
	Fvector 2	1	
	material properties mp	steel_data	
	up/down/vert 1	down	
	up/down/vert 2	up	
SINNINNINIS	Tzero	Tzero	
	area	0.3	
	[thickness]	[0.001 0.001 0.002 0.002]	
	[dist. from leading edge]	[1 1]	
	F vector 1	0.3	
	F vector 2	1	
	up/down/vert 1	vert	
	up/down/vert 2	vert	
	material properties 1	wood_data	
	material properties 2	steel_data	
	material properties 3	plastic_data	
	material properties 4	plastic_data	
temperature	Tzero	Tzero	
	area	1	
	thickness	1	ÄNDRA
	density	1.293	
	heatcapacity	1000	
convection	area	1	
	warm	vert(1,:)	
	cold	vert(2,:)	
	in solid or air	-1	

4.2.2.7 Cabin

In the Cabin block the temperature of the cabin air is modelled. Input signals are the net heat-flow by mass transport and the 24 interior surface temperatures. The mass flow heat goes directly into the temperature block while the surface temperatures proceed to the convection_cabin block and the Cabin_wind block. The convection cabin block consists of 24 convection blocks. Each convection block calculates the heat exchange with the cabin air for a interior surface. The heat flows are summed and then goes into the temperature block.

Cabin_wind models the heat-flow by forced convection in the same way as Convection_cabin does for the natural convection.

The cabin air also exchanges heat with seats, beds and other equipment. A surface_opaque block represents the interior. It is a plastic plate whose area and thickness can be trimmed until it is a good approximation to the interior equipment.

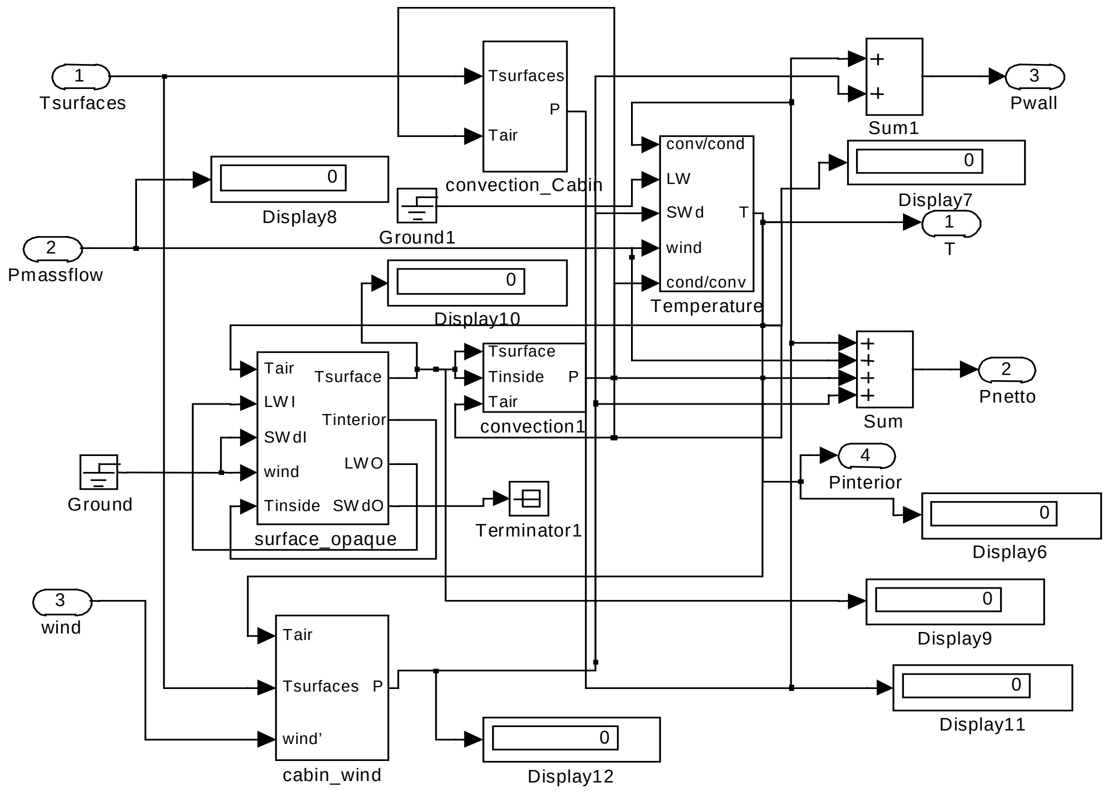

Figure 4.15 Inside the insidecabin block

Parameters for the insidecabin block.

INSIDECABIN

Block	Prompt	Variable	Value	Initialisation commands
insidecabin				
temperature	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	(airvolume)^(2/3)	
	thickness	d	(airvolume)^(1/3)	
	density	ro	1.293	
	heatcapacity	Cp	1000	
convection	area	A	area(i)	global Pr g kinematic_viscosity
	warm	w	vert(1,:)/up(1,:)/down(1,:)	
	cold	c	vert(2,:)/up(2,:)/down(2,:)	
	in solid or air	s	-1	
convection1	area	A	interiorarea	global Pr g kinematic_viscosity
	warm	w	vert(1,:)	
	cold	c	vert(2,:)	
	in solid or air	s	-1	
surface_opaque	Tzero	Tzero	Tzero	
	area	A	interiorarea	
	thickness	d	0.005	
	dist. from leading edge	y	1	
	F vector	F	1	
	material properties	mp	plastic_data	
	warm	w	vert(1,:)	
	cold	c	vert(2,:)	

4. 3 The block library

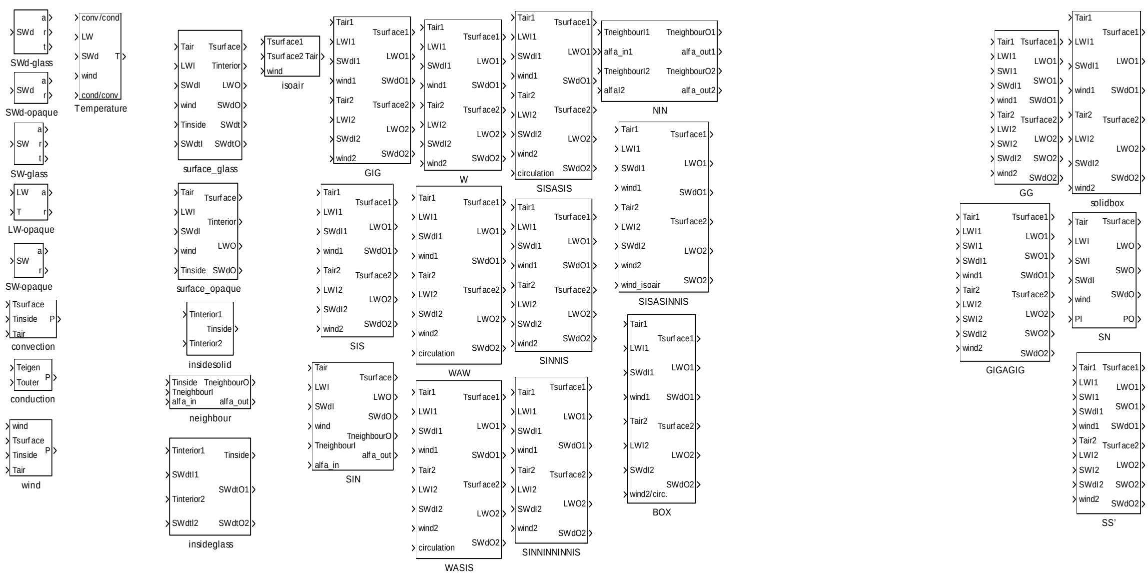

Figure 4.16 The block library

The building blocks are placed inside a library. The only way to change the blocks in the model is to unlock the library and make the changes there. A change in the library block will change all the similar blocks in the model.

5 Simulation and Validation

5.1 Outline of model

The in and out heat-flows from the truck cabin is studied to get a better understanding of how the temperature is influenced by the different inputs and outputs.

$$C_p M \dot{T}_c = \dot{Q}_{wall} + \dot{Q}_{diff} + \dot{Q}_{eng} + \dot{Q}_{fan} + \dot{Q}_{\text{int}} \tag{5.1}$$

The sun radiation is neglected because it was not a part of the experiments.
In steady state

$$\dot{Q}_{wall} = K_w (T_{out} - T_c) = K_w(\Delta T, w) \cdot (T_{out} - T_c) \tag{5.2}$$

$$\dot{Q}_{diff} = C_p \dot{m}_d (T_{out} - T_c) \tag{5.3}$$

$$\dot{Q}_{eng} = K_e (T_e - T_c) = K_w(\Delta T) \cdot (T_e - T_c) \tag{5.4}$$

$$\dot{Q}_{fan} = C_p \dot{m}_f (T_m - T_c) \tag{5.5}$$

$$\dot{Q}_{\text{int}} = K_i (T_i - T_c) \tag{5.7}$$

Where the K values are defined as $K = \alpha \cdot A$ and

- K_w is the K value for all the walls excluding the engine tunnel.
- K_e is the K value for the engine tunnel.
- K_i is the K value for a virtual interior mass.
- T_{out} is the outdoor temperature.
- T_c is the cabin air temperature.
- T_e is the engine temperature.
- T is the temperature of the cabin interior.
- T_m is the temperature of the mixture temperature blown in by the fan.

T_c and T are taken to be states and the other temperatures are taken to be inputs. Arranging the equations on state space form $\dot{T} = AT + Bu$ gives

$$\begin{bmatrix} \dot{T}_c \\ \dot{T}_{\text{int}} \end{bmatrix} = \begin{bmatrix} -(K_w + K_e + C_p \dot{m}_d + C_p \dot{m}_f + K_{\text{int}})/C_p M & K_{\text{int}}/C_p M \\ K_{\text{int}}/C_{pi} M_i & -K_{\text{int}}/C_{pi} M_i \end{bmatrix} \begin{bmatrix} T_c \\ T_{\text{int}} \end{bmatrix} + \begin{bmatrix} C_p \dot{m}_f & K_w + C_p \dot{m}_d & K_e \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} T_m \\ T_{out} \\ T_e \end{bmatrix}$$

(5.8)

Laplace transformed, the transfer-function from the inputs to the cabin air temperature is

$$T_c(s) = \frac{((C_{pi}M_i s + K_i)\cdot((K_w + C_p\dot{m}_d)T_{out} + K_e T_e + C_p\dot{m}_f T_m))}{(C_p Ms + K_w + K_e + K_i + C_p(\dot{m}_f + \dot{m}_d))\cdot(C_{pi}M_i s + K_i) - K_i^2} \quad (5.9)$$

Parameters such as the K values and the mass flows influence, not only the amplitude, but also the "behaviour" of the system. Because they are temperature dependent the system is expected to have a non-linear and complex behaviour. Neither the K values nor the mass flows will change their sign so nothing radical will happen. Nevertheless the system will change in time and actions taken by the regulator when for example changing the fan speed will influence both the step response time and the final temperature. At development of a temperature regulator caution should be taken when it comes to stability.

When the inputs T_{out}, T_e, T_m are constant the final temperature can be calculated according to the "final value theorem" $\lim_{t\to\infty} T_c(t) = \lim_{s\to 0} T_c(s)$ that is $T_c final = T_c(0)$ and using (5.9)

$$T_c final = T_c(0) = \frac{(K_w + C_p\dot{m}_d)\cdot T_a + K_e\cdot T_e + C_p\dot{m}_f\cdot T_m)}{K_w + K_e + C_p(\dot{m}_f + \dot{m}_d)} \quad (5.10)$$

The temperatures of the air and the water where measured both before and after the heat exchanger. The mass flow is calculate with the help of an the energy balance below

$$\dot{Q}_{in} = \dot{Q}_{out} \quad (5.11a)$$

$$Q_{in} = \dot{m}_f \cdot C_{p\_air}\cdot T_a + \dot{m}_w \cdot C_{p\_water}\cdot T_{b\_he} \quad (5.11b)$$

$$Q_{out} = \dot{m}_f \cdot C_{p\_air}\cdot T_m + \dot{m}_w \cdot C_{p\_water}\cdot T_{a\_he} \quad (5.11c)$$

where T_{b_he} and T_{a_he} are the water temperatures before and after the heat exchanger.

Solving for $\dot{m}$ using (5.11abc)

$$\dot{m}_f = \dot{m}_w \cdot \frac{C_{p\_water}}{C_{p\_air}}\cdot\frac{(T_{b\_he} - T_{a\_he})}{(T_m - T_a)} \quad (5.12)$$

The water flow through the heat exchanger is noted with only one decimal. With this exactness $\dot{m}$ can therefore differ with up to 16% from the calculated value.

The main part of this work has focused on getting a good estimation of the heat-flow through the walls. No work has been done to calculate the diffusive mass-flow. The diffusive mass-flow can instead be used to trim the model to get correct steady state values when comparing the simulations to the measurements. This is done by rewriting the equation (5.10) into

$$\dot{m}_d = \frac{K_w(T_a - T_c) + K_e(T_e - T_c) + C_p \dot{m}_f (T_m - T_c)}{C_p(T_c - T_a)} \qquad (5.13)$$

If one thus not believe that there is a significant diffusive mass-flow, the quantity $K_{diffuse} = C_{p\_air} \cdot \dot{m}_d$ can be used to measure the model fault in steady state since $K_{tot} = K_{wall} + K_{diffuse}$

5.2 Simulations and experiments

Available signals, inputs, and outputs from the measurements are T_{out}, T_e, T_m, T_c. The mixture air where heated externally so the engine temperature was the same as the outside temperature.

The measurements, carried out in a cold room, where the temperature was held constant at a temperature of 263 K. Inside the room there was also a big fan that made it possible to make a wind. The fan speed was either on or off. When it was on it gave a wind speed of approximately 20 m/s.

The temperature of the mixture air was altered between 290 K and 320 K.

The third input that was changed was the fan speed. It was altered between level 1 and level 4, which means between 0.06 and 0.17 Kg/s.

Below there are six plots that show first the measured temperature and then the simulated temperature alternately. At the beginning of the curves they are at a steady state and then a sudden change is made in one of the three parameters. The curves then show the cabin air temperature on it’s way to the new steady state.

Measurements and calculations

	Exp 1	Exp 2	Exp 3	Exp 4	Exp 5	Exp 6	Exp 7	Exp 8
Fan speed	H	H	H	L	L	L	L	H
wind	L	H	H	H	H	L	L	L
Fan air temp	H	H	L	L	H	H	L	L

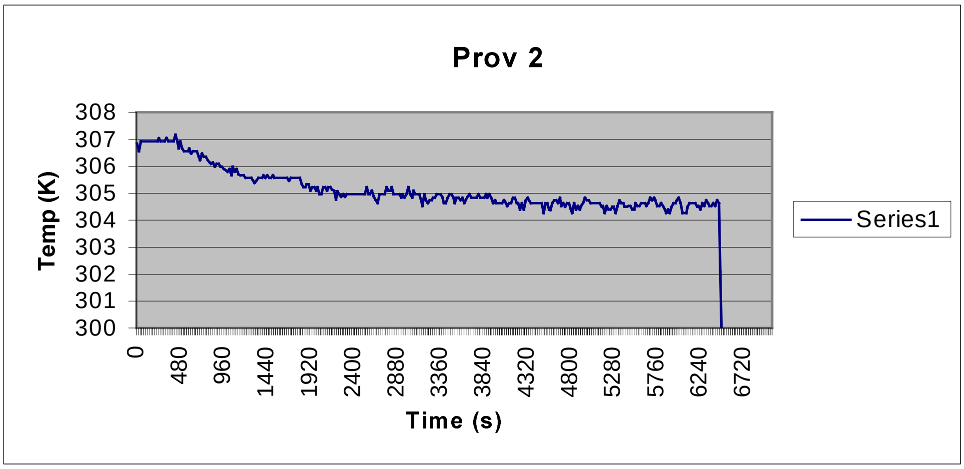

Fig 5.1 Fan speed: H, wind: L -> H, Fan air temp: H (Exp 2) x-axis: time in seconds, y-axis: temperature in Kelvin

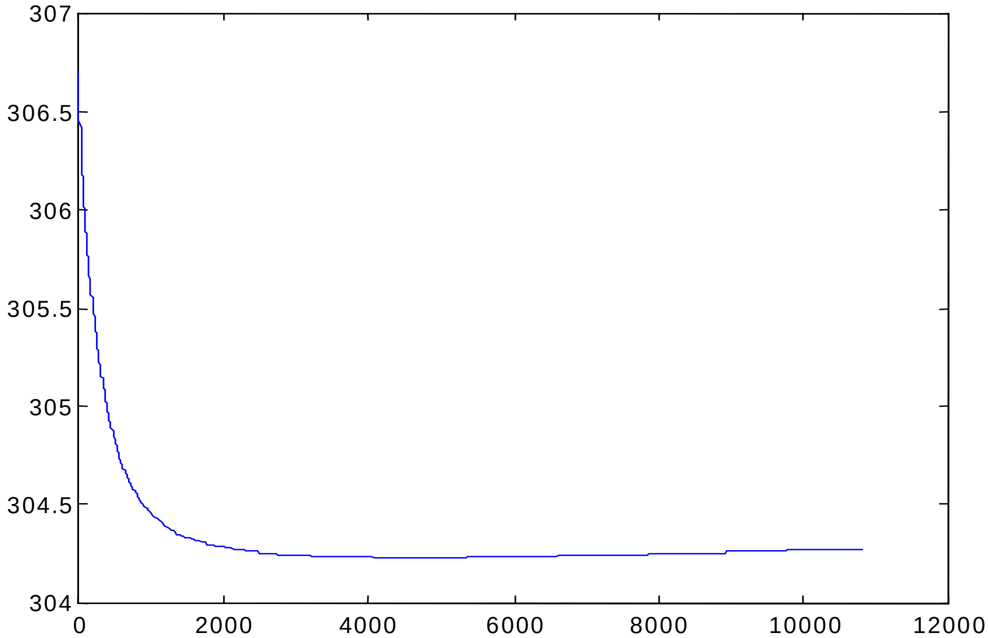

Fig 5.2 Fan speed: H, wind: L -> H, Fan air temp: H (Exp 2,simulated) x-axis: time in seconds, y-axis: temperature in Kelvin

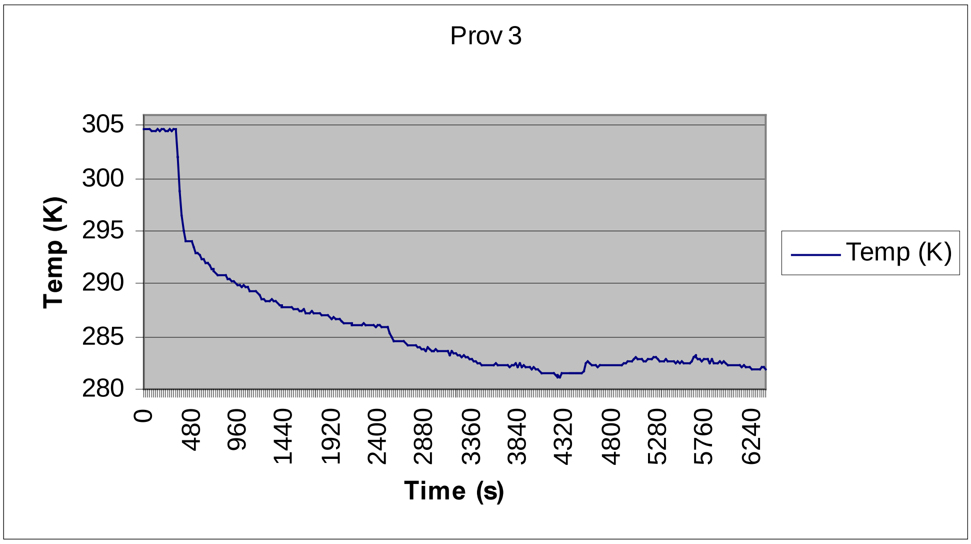

Fig 5.3 Fan speed: H, wind: H, Fan air temp: H->L (Exp 3) x-axis: time in seconds, y-axis: temperature in Kelvin

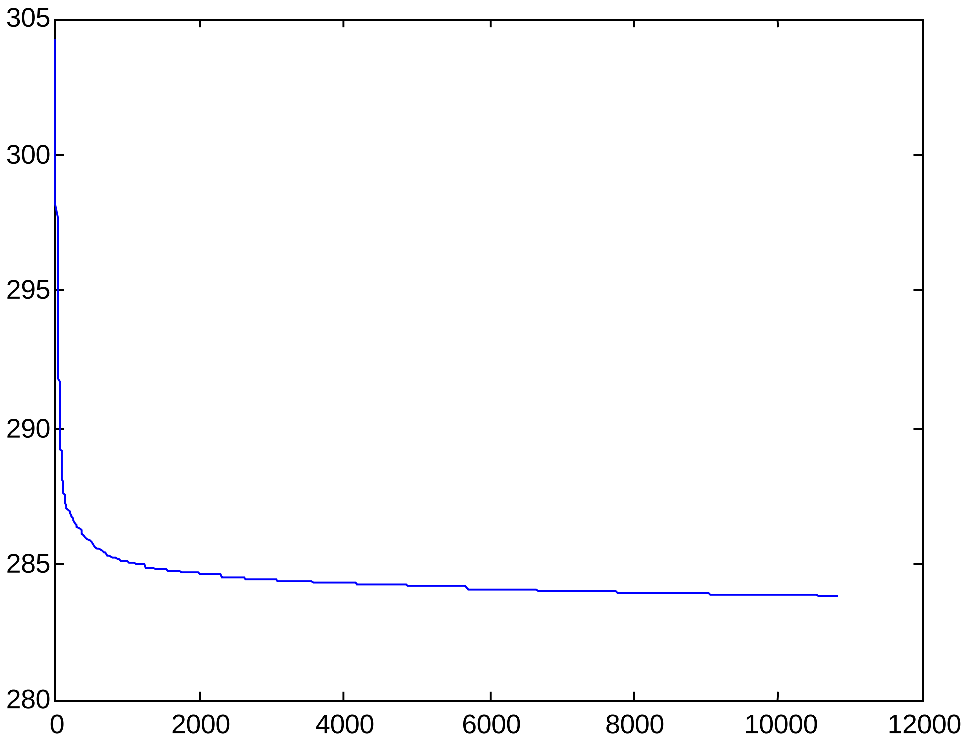

Fig 5.4 Fan speed: H, wind: H, Fan air temp: H->L (Exp 3, simulated) x-axis: time in seconds, y-axis: temperature in Kelvin

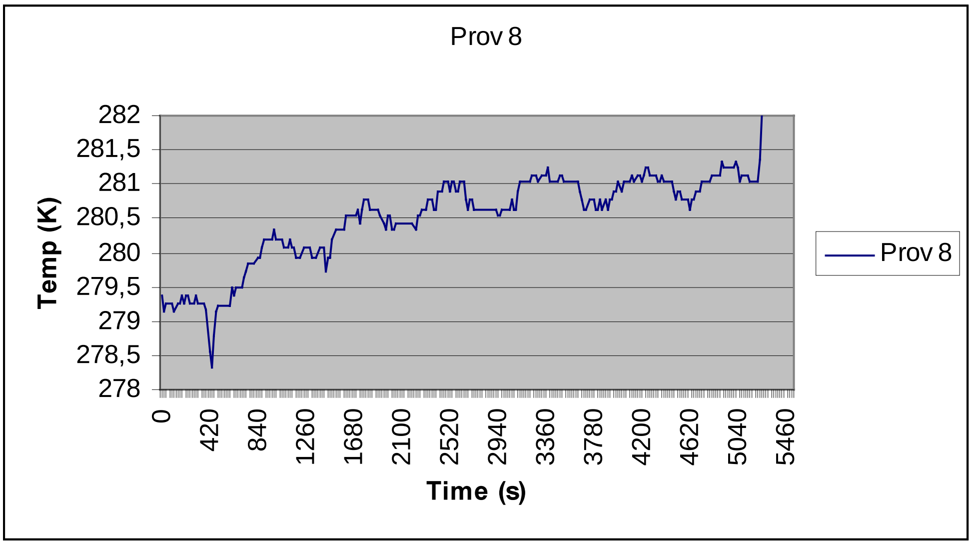

Fig 5.5 Fan speed: L->H, wind: L , Fan air temp: L (Exp 8)   x-axis: time in seconds, y-axis: temperature in Kelvin

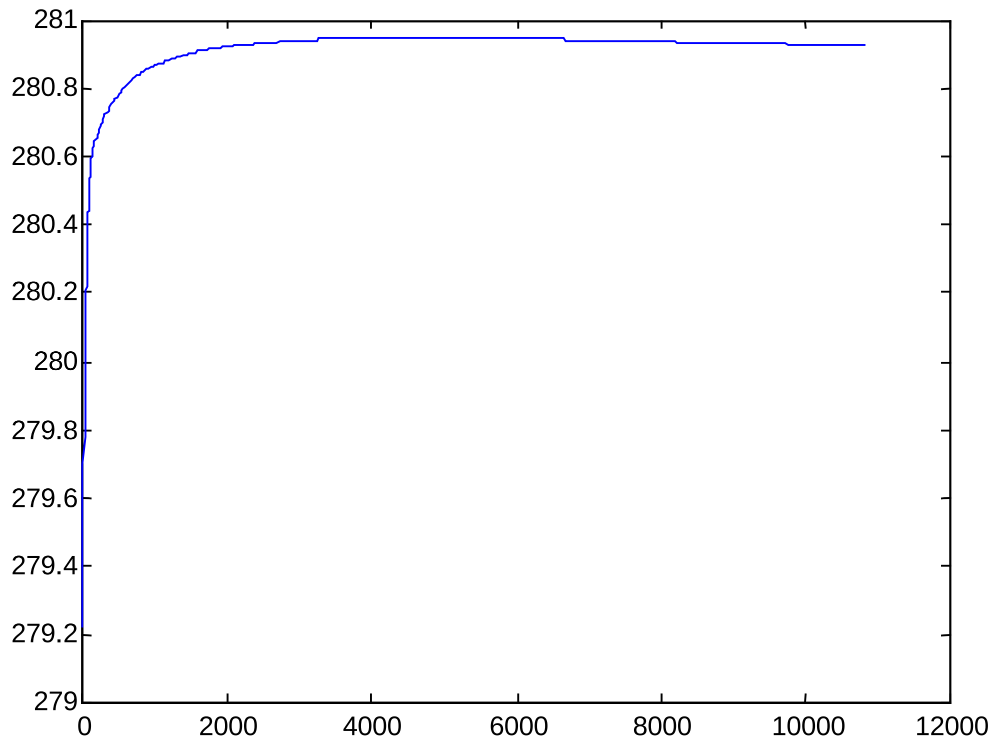

Fig 5.6 Fan speed:L-> H, wind: L , Fan air temp: L (Exp 8, simulated)   x-axis: time in seconds, y-axis: temperature in Kelvin

The measured raise times and the simulated rise time.

Tr measured	6100	2140	2400	3900	3000	2820	4080	2700
Simulated	1250	930	460	140	---	---	2170	450

The rise times and time constants differs with up to a factor 30 between the measured temperature and the simulated temperature for those experiments where the system parameters are constants. This is of cause not acceptable.

5.3 Search for fault

To exclude the possibility that there is something systematically wrong with the large and complex model a much smaller model with only two states is used. As in the experiments $T_e = T_{out}$ and K is the total K value for the cabin calculated through the equations (5.14a,b) fulfilling that in steady state the amount of heat in should equal the amount of heat out.

$$P_{fan} = C_p \dot{m}_f (T_m - T_c) = K(T_c - T_{out}) \qquad (5.14a)$$

Rewritten

$$K = C_p \dot{m}_f \frac{(T_m - T_c)}{(T_c - T_{out})} \qquad (5.14b)$$

The model looks like

$$C_p M \dot{T}_c = K \cdot (T_{out} - T_c) + C_p \dot{m}_f (T_m - T_c) + K_{\text{int}} \cdot (T_{\text{int}} - T_c) \qquad (5.15)$$

$$C_{pi} M_i \dot{T}_{\text{int}} = K_{\text{int}} \cdot (T_c - T_{\text{int}}) \qquad (5.16)$$

On state space form $\dot{T} = AT + Bu$

$$\begin{bmatrix} \dot{T}_c \\ \dot{T}_{\text{int}} \end{bmatrix} = \begin{bmatrix} -(K + C_p \dot{m}_f + K_{\text{int}})/C_p M & K_{\text{int}}/C_p M \\ K_{\text{int}}/C_{pi} M_i & -K_{\text{int}}/C_{pi} M_i \end{bmatrix} \begin{bmatrix} T_c \\ T_{\text{int}} \end{bmatrix} + \begin{bmatrix} C_p \dot{m}_f & K \\ 0 & 0 \end{bmatrix} \begin{bmatrix} T_m \\ T_{out} \end{bmatrix} \qquad (5.17)$$

A step response solution to (5.17) is of the form

$$T(t) = C(1 - e^{\lambda_1 \cdot t}) + D(1 - e^{\lambda_2 \cdot t})$$

where λ_1, λ_2 are the eigenvalues to the A matrix.

The eigenvalues looks like

```
In[26]:= Eigenvalues[A]

Out[26]= {1/(2 CpM CpMi) (-Cpmf CpMi - CpMi K - CpM Kint -
    √(-4 CpM CpMi (Cpmf + K - Kint) Kint + (Cpmf CpMi + CpMi K + CpM Kint)^2)),
  1/(2 CpM CpMi) (-Cpmf CpMi - CpMi K - CpM Kint +
    √(-4 CpM CpMi (Cpmf + K - Kint) Kint + (Cpmf CpMi + CpMi K + CpM Kint)^2))}
```

The eigenvalues have a complicated structure making it difficult to see how changes in the parameters will influence the step response times.

The step response time for one mode, that is the time to reach P % of the final value is

$$t = \frac{\ln(1-P)}{\lambda} \qquad (5.18)$$

Below the measured experiment 5 will be compared to the corresponding simulations with the mini-model. Experiment 5 has been chosen because the system behaves fairly linear for changes in the fan air temperature.

Step response times for the two different modes are shown below for varying values on the parameters.

	K	mf	M	Mi	Kint	eigen 1	eigen 2	P	Time 1	Time 2
					Cp	4000				
Exp 1	51,78	0,1633	7	9	30	-0,03076	-0,00072	0,9	74,8	3219,5
Exp 2	65,54	0,2066	7	9	50	-0,03905	-0,00113	0,9	59,0	2041,2
Exp 3	55,41	0,1747	7	9	50	-0,03310	-0,00108	0,9	69,6	2133,7
Exp 4	21,28	0,0671	7	9	50	-0,01341	-0,00057	0,9	171,7	4027,5
Exp 5	23,31	0,0735	7	9	50	-0,01454	-0,00064	0,9	158,4	3586,3
Exp 6	16,97	0,0535	7	9	50	-0,01106	-0,00037	0,9	208,2	6173,7
Exp 7	14,93	0,0471	7	9	50	-0,00998	-0,00025	0,9	230,8	9384,3
Exp 8	43,02	0,1356	7	9	50	-0,02586	-0,00099	0,9	89,0	2331,6
Exp 5	16,32	0,0514	7	10	5	-0,00967	-0,00012	0,9	238,2	19895,6
Exp 5	23,31	0,0735	7	10	10	-0,01382	-0,00022	0,9	166,6	10282,1
Exp 5	23,31	0,0735	7	10	15	-0,01385	-0,00032	0,9	166,2	7287,4
Exp 5	23,31	0,0735	7	10	20	-0,01390	-0,00039	0,9	165,7	5837,6
Exp 5	23,31	0,0735	7	10	25	-0,01396	-0,00046	0,9	164,9	5013,6
Exp 5	23,31	0,0735	7	10	30	-0,01403	-0,00051	0,9	164,1	4511,2
Exp 5	23,31	0,0735	7	10	35	-0,01412	-0,00055	0,9	163,1	4201,9
Exp 5	23,31	0,0735	7	10	40	-0,01422	-0,00057	0,9	161,9	4024,8
Exp 5	23,31	0,0735	7	10	45	-0,01434	-0,00058	0,9	160,6	3949,6
Exp 5	23,31	0,0735	7	10	50	-0,01446	-0,00058	0,9	159,2	3963,5
Exp 5	23,31	0,0735	7	10	55	-0,01460	-0,00057	0,9	157,7	4065,7
Exp 5	23,31	0,0735	7	10	60	-0,01475	-0,00054	0,9	156,1	4267,4
Exp 5	23,31	0,0735	7	10	65	-0,01492	-0,00050	0,9	154,4	4595,1
Exp 5	23,31	0,0735	7	10	70	-0,01509	-0,00045	0,9	152,6	5101,0
Exp 5	23,31	0,0735	7	10	75	-0,01528	-0,00039	0,9	150,7	5889,4
Exp 5	65,57	0,0735	7	2	40	-0,02149	-0,00328	0,9	107,2	701,1
Exp 5	65,57	0,0735	7	4	40	-0,02056	-0,00172	0,9	112,0	1341,3
Exp 5	65,57	0,0735	7	6	40	-0,02028	-0,00116	0,9	113,5	1984,9
Exp 5	65,57	0,0735	7	8	40	-0,02015	-0,00088	0,9	114,3	2629,3
Exp 5	65,57	0,0735	7	10	40	-0,02007	-0,00070	0,9	114,7	3274,0
Exp 5	65,57	0,0735	7	15	40	-0,01997	-0,00047	0,9	115,3	4886,2
Exp 5	65,57	0,0735	7	30	40	-0,01987	-0,00024	0,9	115,9	9724,0
Exp 5	65,57	0,0735	7	40	40	-0,01984	-0,00018	0,9	116,0	12949,4

Observing that for reasonable values on the parameters one mode will have a step response time between 100- 200 s and the other between 2000- 4000 s

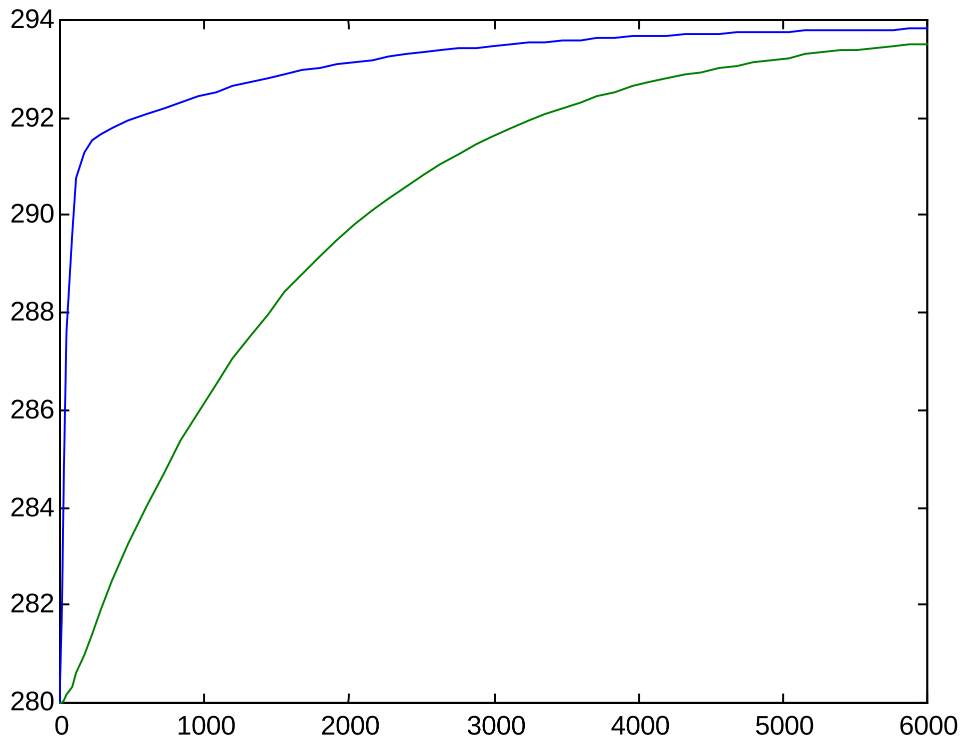

Fig 5.8 Fan speed:L, wind: L , Fan air temp: L->H (Exp 5,simulated) x-axis: time in seconds, y-axis: temperature in Kelvin

The upper curve is the cabin air temperature and the other shows the temperature of the interior. The air temperature is dominated by the fast mode and the interior is dominated by the slow mode.

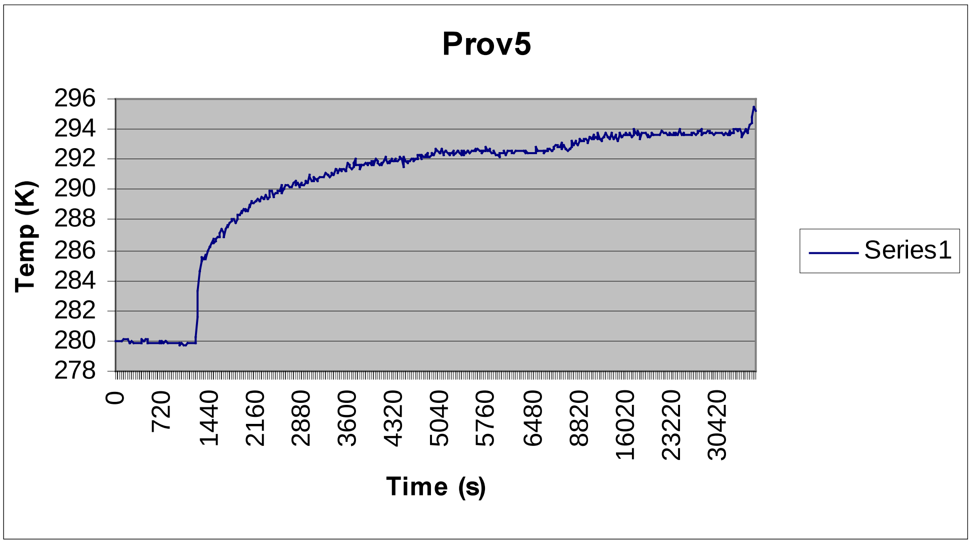

Fig 5.7 Fan speed:L, wind: L , Fan air temp: L->H (Exp 5) x-axis: time in seconds, y-axis: temperature in Kelvin

Step response time 3960 s

The measured temperature is most similar to the interior temperature and is quite different from the simulated air temperature. This indicates that there might be something wrong with the temperature measurements.

5.3.1. Thermoelements

It seems that the measured air temperature is strongly influenced by the temperature of the walls. The flexes of the thermoelements were attached to the walls so it is natural to investigate what effect that will have on the measured temperature.

The thermoelements used consists of to separate wires, though together, that are insulated from the surrounding and each other by a 0,5 mm thick plastic cover. The wires are approximately 1,5 m long and connected to each other at one end and connected to a contact at the other end.

Because of the two metals different ability to diffuse electrons they will, in contact, have different potentials. This potential difference is temperature dependent so the temperature can be calculated from the potential difference.

Assuming that T_c and the wall temperature T_w is constant then the steady state temperature T_{metal} of the wires must fulfil the equation (5.19) so that no net flow of heat is present.

$$K_{am}(T_c - T_{metal}) + K_{apm}(T_c - T_{metal}) + K_{wpm}(T_w - T_{metal}) = 0 \qquad (5.19)$$

where
K_{am} is the K value for the heat flow between the air and the metal.
K_{apm} is the K value for the heat flow between the air and the metal through the plastic layer.
K_{wpm} is the K value for the heat flow between the wall and the metal through the plastic layer.

Calculation of the K values.

A K value is defined as $K = \alpha \cdot A$ and since the area depends on the length of the wire the quantity B = area per metre is used. Below the standard formula $\frac{1}{\alpha_{tot}} = \frac{1}{\alpha_{convection}} + \frac{thickness}{conductivity}$ is used

$$\frac{1}{\alpha_{am}} = \frac{1}{\alpha_{convection}} = \frac{1}{2} \Rightarrow \alpha_{am} = 2 \qquad (5.20)$$

$$\frac{1}{\alpha_{apm}} = \frac{1}{\alpha_{convection}} + \frac{\delta}{\lambda} = \frac{1}{2} + \frac{0.0005}{0.2} \Rightarrow \alpha_{apm} = 1.9 \qquad (5.21)$$

$$\frac{1}{\alpha_{wpm}} = \frac{\delta}{\lambda} = \frac{0.0005}{0.2} \Rightarrow \alpha_{wpm} = 400 \qquad (5.22)$$

Let the flex have a length of 2 meters and let the length that is in contact with the wall be 2 cm and that half the flex area at the connection point is in contact with the wall. Equation () will then look like

$$2 \cdot B \cdot 2 \cdot (T_c - T_{metal}) + 400 \cdot \frac{B}{2} \cdot 0.02 \cdot (T_w - T_{metal}) = 0 \qquad (5.23)$$

solving for T_m yields

$$T_{metal} = \frac{T_c + T_w}{2} \qquad (5.24)$$

This result implies that what in fact is measured is the average temperature of the cabin air and the wall.
A new plot with the average temperature in the middle.

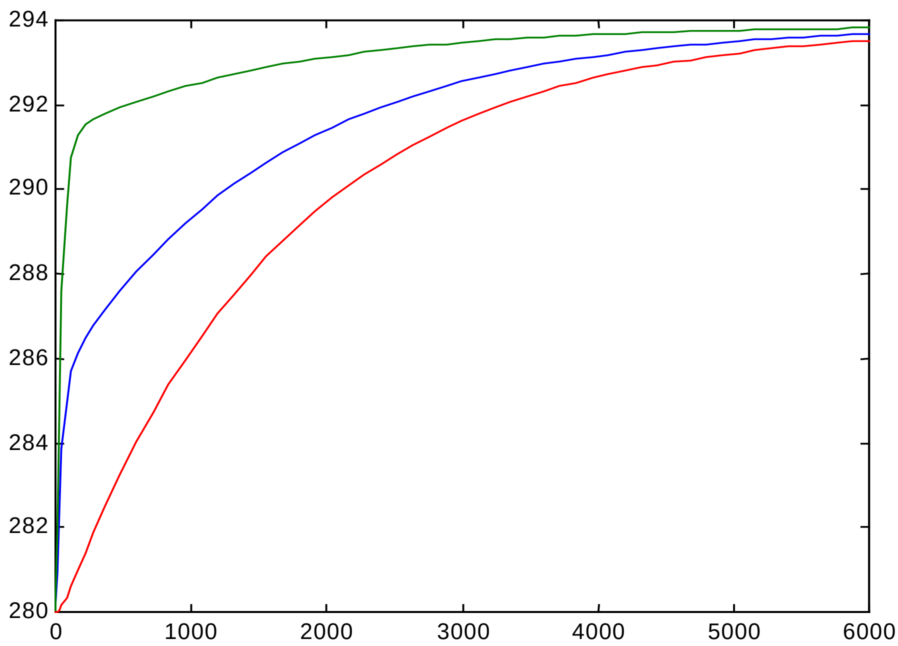

Fig 5.9 Fan speed:L, wind: L , Fan air temp: L->H (Exp 5,simulated) x-axis: time in seconds, y-axis: temperature in Kelvin

Since the mini-model does not include a temperature state for the walls the temperature of the interior is used instead because it will have a raise time of the same magnitude. Doing so, simulating the average temperature and comparing it to experiment 5 in fig 5.7 shows that there is a very good correlation between the measured curve and the simulated average is seen. The conclusion must be that the air temperature is not correctly measured because of disturbances from the attachment point.

5.4 Steady state observations

Most of the figures in the tables below are wrong because of the error in temperature measurement of the cabin air as explained above. It can nevertheless be interesting to study them qualitative.

Measurements and calculations

	Exp 1	Exp 2	Exp 3	Exp 4	Exp 5	Exp 6	Exp 7	Exp 8
fan speed	H	H	H	L	L	L	L	H
wind	L	H	H	H	H	L	L	L
Fan air temp	H	H	L	L	H	H	L	L
Outside temp	263,1	263,2	263	263,2	263,1	262,8	263,1	263,2
Before h.ex	352	352,4	348,9	337,3	344,3	347,6	338,1	346,5
After h.ex	318	320,6	276,2	274	300,5	297,4	278,2	275,6
Inlet air	320,7	319,6	288,9	289,3	320,8	321,2	289,5	288,5
Steering wheel	308,1	305,7	283,3	280,7	294,5	295,3	280,1	281,5
Inner roof	305,3	302,8	282,2	279,4	292,6	293,8	278,9	280,8
Inner wall	307,4	305	282,4	279,7	294,1	294,9	279,1	280,7
Water flow	4	5,3	0,9	0,4	1,4	0,9	0,3	0,7
Air flow	0,1633	0,2066	0,1747	0,0671	0,0735	0,0535	0,0471	0,1356
Average temp	306,9333	304,5	282,6333	279,9333	293,7333	294,6667	279,3667	281

The table shows the different experiments and their final values. In the first group it is shown if the fan speed, the wind or the fan air temperature is either high or low. The second group contains the steady-state temperatures at measuring points and also the water flow through the heat exchanger. The third group shows the calculated airflow through the heat exchanger calculated according to equation (5.12). There is also the average temperature of the three thermoelements measuring the cabin air temperature at different locations.

Pfan	2269,91	3151,11	1105,55	634,56	2008,74	1433,47	481,68	1027,42
Pwall	-1315	-1707	-846	-598,8	-1246	-872,4	-293,3	-396,2
Pheater	9497,33	11769,71	4569,19	1768,18	4282,18	3155,07	1254,91	3465,83
Kwall	29,93	41,13	43,09	35,36	40,54	27,55	17,92	22,01
Ktot	52,11	76,38	55,81	37,99	64,36	44,75	29,66	57,52

These two tables shows the amount of heat that is absorbed from the water, that is P_{heater}, and what amount of heat the air actually gains, that is P_{fan}, when the heat loss due to the outgoing cabin air is accounted fore. As one can see this is not a very efficient way to heat a truck or a car. P_{wall} is the amount of heat passing through the walls. The rest of the energy is lost by the diffusive mass flow. Transforming the heat flows into K values gives the second group of the table. The missing K is $K_{diffuse}$.

Remember, from (5.12a), that $P_{fan} = C_p \dot{m}_f (T_m - T_c)$. Because of the bad measurements T_c is higher then the measured value. This means that P_{fan} is less then noted which result in that K_{tot} and the heat loss by diffusion are less then noted according to equations (5.13) and (5.14b). Probably much less then noted.

	10%	0,1796	0,2273	0,1921	0,0738	0,0808	0,0588	0,0518	0,1492
Air flow		0,1633	0,2066	0,1747	0,0671	0,0735	0,0535	0,0471	0,1356
	-10%	0,146927	0,185955	0,157203	0,060368	0,066132	0,048141	0,042357	0,12207
	10%	307,9069	305,5303	283,0749	280,4977	295,0979	296,0396	279,9548	281,493
	Tc(0)	306,9333	304,5	282,6333	279,9333	293,7333	294,6667	279,3667	281
	-10%	305,7994	303,3069	282,1193	279,2917	292,2158	293,1353	278,7013	280,4324

In these two groups it has been calculated what will happen if there is a 10 percent fault in the water flow measurement which is probably the case because of the few numbers of digits. The first group shows how the airflow will change and the second one what effect that would have one the final temperature. There is a need for better water flow measurements.

5.5 Sources of errors

5.5.1 Diffusive mass flow

The diffusive mass flow $\dot{m}_d$ is unknown and is also temperature dependent. A full treatment of the diffusive mass flow is not given in the undergraduate courses, implying a complex behaviour, and is here approximated with a constant. The heat flow caused by diffusion must be of the same magnitude as the heat flow through the cabin walls if the model should match measurements.

5.5.2 Numerical instability

The numerical instability in the simulations is mainly due to stiffness.
A stiff system is a system where the time constants are different in size.

In this system the cabin temperature changes very slow compared to for example the surface temperature of the truck body. One sign of the systems stiffness is that it does not fully converge. “Pnetto” oscillates near steady state. The oscillation is of the magnitude of 1W or less.
The change in the cabin temperature is then of the magnitude $1 \cdot 10^{-4} [K/s]$, which can be neglected. However the numerical instability gives rise to uncertainties while investigating the stability of the closed loop including the regulator.

To improve the stability, the states that change rapidly could be aggregated so that the new state represents a thicker layer that has a longer time constant.

5.5.3 Measurements

In trying to adapt the model to measurements one must be aware of the fault tolerances in the measurement equipment. Also variations in the environment as the outdoor temperature influences the measurements.

The blow from the air outlet and the overall air movement inside the cabin can inflict the measured temperatures.

5.5.4 Area and height dependence for convection.

The formulas used to calculate the heat flows through convection has a height and area dependence that is not direct proportional to the height and area. The formulas are derived with the presumption that we have one single plate. Dividing the plates into parts will give a small change in the heat flow.

5.5.5 Parameters and dimensions

The parameters and dimensions such as the areas, the thickness, and the material parameters are not very exact. The dimensions have been measured in a sloppy way and some times also the wrong material has been used. For example all the parts having the material properties of wood should instead have the material properties of pressed textile. Also the other material parameters must be checked. For example there are all sorts of rubber and plastic and the model probably does not include the right ones. Radiation parameters as the absorbtivity for LW and SW radiation have sometimes been guessed and especially the parameters for the SW radiation is hard to find and will also depend on the colour of the truck.

6 Results

6.1 Goal

The goal was that the model should make it possible to determine if a temperature regulator, developed and approved for a certain truck cabin model, also could be used in another truck cabins of different sizes considering stability, time to reach the desired temperature and other performances. Therefore the model should be easy to adapt to different truck cabins.

6.2 Result

The model can give a good view of how a real system would behave and is useful in the design of a regulator. The model can also be used for other purposes for example to investigate how additional insulation influences the energy consumption.

The model can easily be adapted to different cabin models regarding size and materials. The model also behaves like expected from a physical point of view. But the model's simulation results do not correlate with the measurements as good as desired. This likely because the measurements are incorrect. Further verifications are needed with better measurements.

With reservation for the insufficient verification because of erroneous temperature measurements the goals has been fulfilled.

7 Low level blocks

7.1 Level 1

Level one contains the blocks temperature, convection, wind, conduction, LW_opaque, SWd_opaque and SWd_glass

7.1.1 temperature

The amount of energy contained in a piece of mass is given by:

$$E = C_p \cdot m \cdot T \tag{7.1}$$

where C_p is the heat capacity and depends on the material.
The energy shall also equal

$$E = E_{heat} + E_0 \tag{7.2}$$

where E_{heat} is the amount of heat absorbed since a point of time t_0 when the energy equalled E_0.
That is

$$E = \int_{t_0}^{t} Q(t)dt + C_p m T_0 \tag{7.3}$$

where Q is the heat flow

(7.1),(7.3) gives

$$C_p m T = \int_{t_0}^{t} Q(t)dt + C_p m T_0 \tag{7.4}$$

dividing with $C_p \cdot m$ on both sides finally makes the expression

$$T = \int_{t_0}^{t} \frac{1}{C_p m} Q(t)dt + T_0 \tag{7.5}$$

This function is calculated in the temperature block

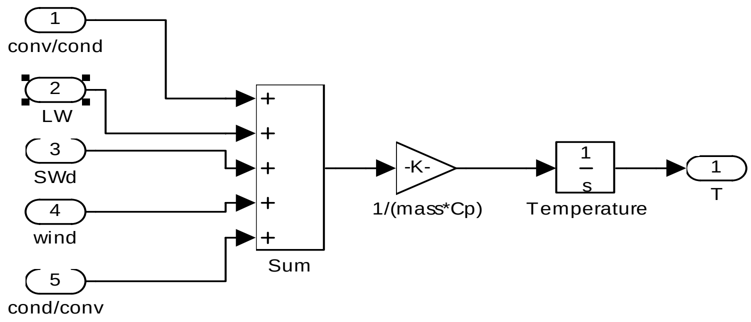

Figure 5.1 Inside the temperature block

Block	Prompt	Variable	Value	Initialisation commands
temperature	Tzero	Tzero		mass=A*d*ro;
	area	A		
	thickness	d		
	density	ro		
	heat capacity	Cp		

Tzero is the initial temperature

7.1.2 convection

The convection block calculates the natural convection according to the formulas given by the equations in Chapter 2.2.

The block estimates the surface boundary temperature and calculates the heat-flow using different formulas depending on if the surface boundary is warmer or colder than the air. The parameters also change with the orientation.

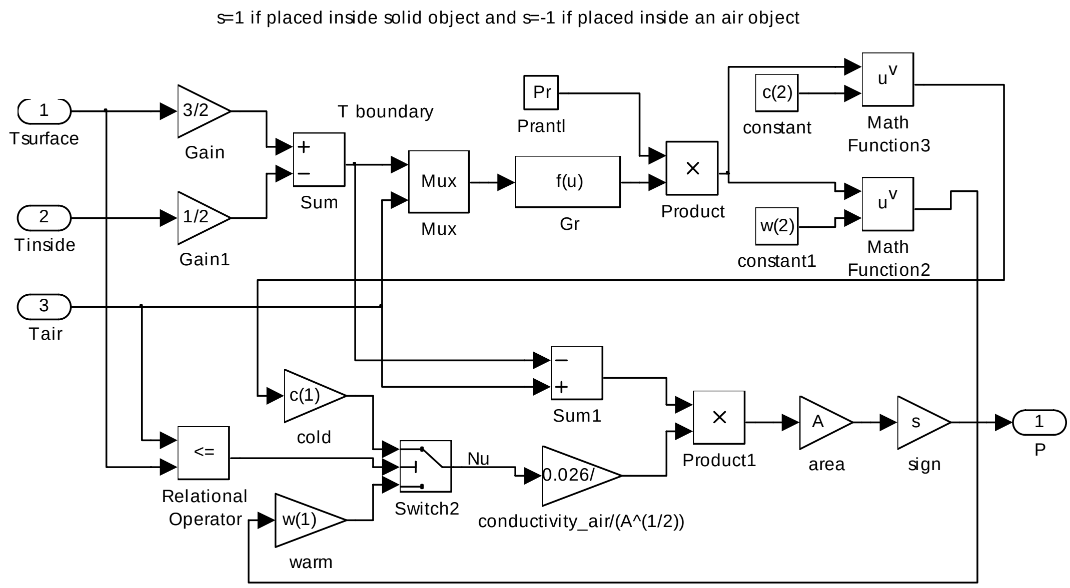

Figure 5.2 Inside the convection block.

Block	Prompt	Variable	Value	Initialisation commands
convection	area	A		global Pr g kinematic_viscosity
	warm	w		
	cold	c		
	in solid or air	s		

w and c must contain vectors of the type [C m]. s should be set to 1 if the block is placed inside a solid and -1 if it is placed inside an air object.

7.1.3 wind

The wind block calculates the forced convection according to the equations (2.33) and (2.34)

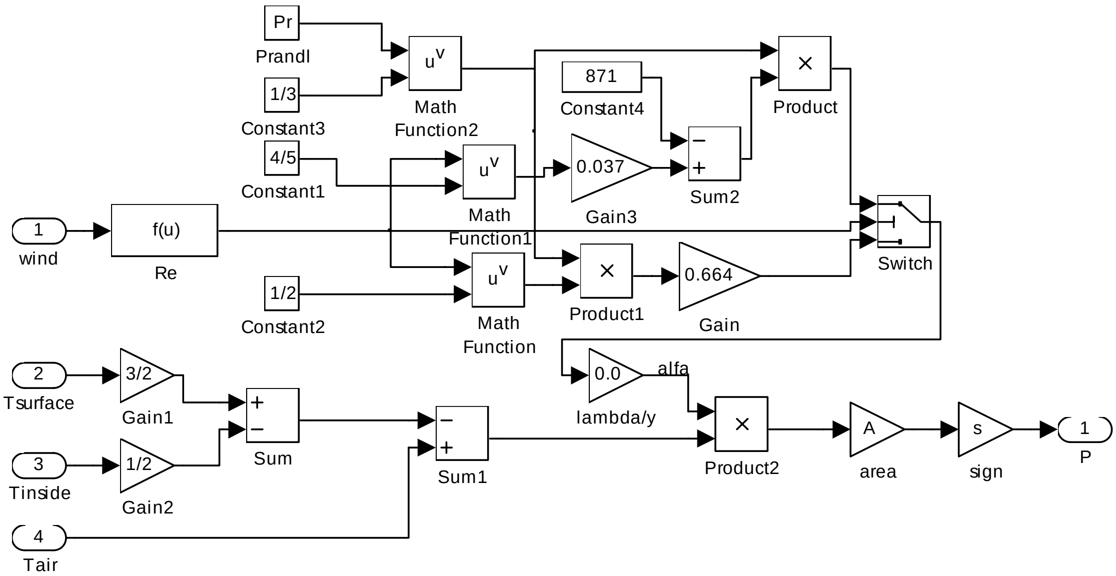

Figure 5.3 Inside the wind block.

Block	Prompt	Variable	Value	Initialization commands
wind	area	A	A	global Kinematic_viscosity
	dist. from leading edge	y	y	
	air or solid	s	1	

s should set to 1 if the block is placed inside a solid and -1 if it is placed inside an air object.

7.1.4 conduction

The conduction block calculates the heat flow between two layers inside a solid. It is assumed that a steady state flow is a valid approximation so the heat flow is given by

$$\dot{Q} = \frac{A \cdot \lambda}{d}(T_1 - T_2) \qquad (7.6)$$

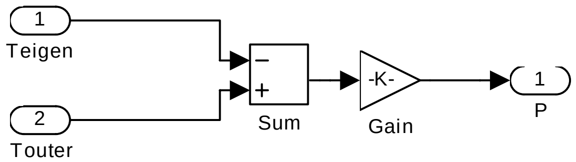

Figure 5.4 Inside the conduction block.

Block	Prompt	Variable	Value	Initialization commands
conduction	area	A		
	thickness of layer	d		
	conductivity	lambda		

7.1.5 LW_opaque

The LW_opaque block calculates the net amount of absorption and reflectance of LW radiation. There is two processes in progress. First there is the absorption and reflectance if incident LW radiation. Second we have the blackbody radiation that is calculated in the blackbody radiation function which looks like

$$\dot{q}_{emission} = \varepsilon\sigma T^4 \qquad (7.7)$$

$\dot{Q}_{emission}$ is then added to the reflected radiation and withdrawn from the absorbed radiation giving the net amount absorption and reflection of LW radiation.
The F vector enables the possibility to use a shape factor F.

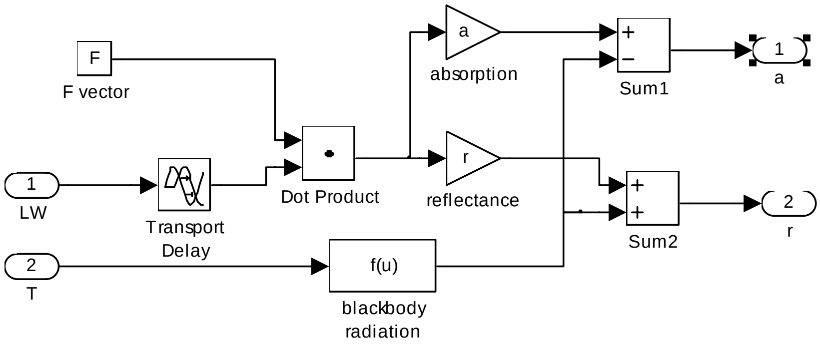

Figure 5.5 Inside the LW_opaque block.

Block	Prompt	Variable	Value	Initialization commands
LW-opaque	area	AA		
	emission coefficient	e		
	absorption coefficient	a		
	reflectance coefficient	r		
	F vector	F		

7.1.6 SWd_opaque

The block SWd_opaque calculates the amount of absorption and reflectance of incident diffusive SW-radiation on an opaque surface. If the radiation originates from two or more sources the F vector can be used to tell how much radiation that hits this particular surface by weighting the incoming radiation vector. The transport delay block works as a virtual state to avoid algebraic loops described in the Simulink manual. It also makes it impossible to linearize the model into a state space model with all the advantages that would bring. It would be a good thing to find another solution.

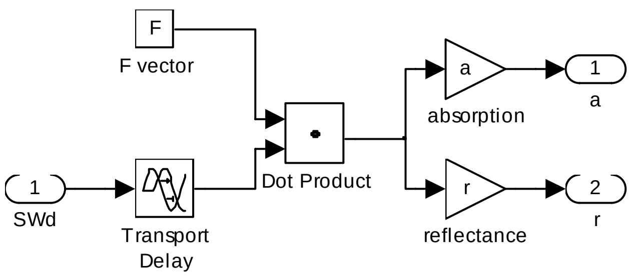

Figure 5.6 Inside the SWd_opaque block.

Block	Prompt	Variable	Value	Initialization commands
SWd_opaque	absorption coefficient	a		global stime
	reflectance coefficient	r		
	F vector	F		

7.1.7 SWd_glass

The block SWd_glass calculates the amount of absorption, reflectance and transmission of incoming diffusive SW-radiation on a glass surface. If the radiation originates from two or more sources the F vector can be used to tell how much radiation that hits this particular surface by weighting the incoming radiation vector.

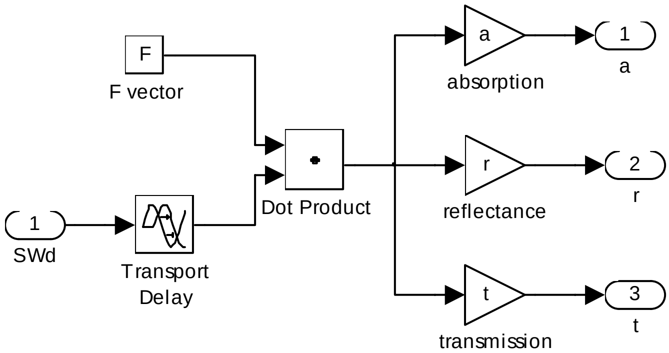

Figure 5.7 Inside the SWd_glass block.

Block	Prompt	Variable	Value	Initialization commands
SWd_glass	absorption coefficient	a		global stime
	reflectance coefficient	r		
	transmission coefficient	t		
	F vector	F		

7.2 Level 2

7.2.1 surface_opaque

The surface_opaque block calculates the heat flows going in and out from the surface. The heat flows are inputs to the temperature block where the temperature is calculated.

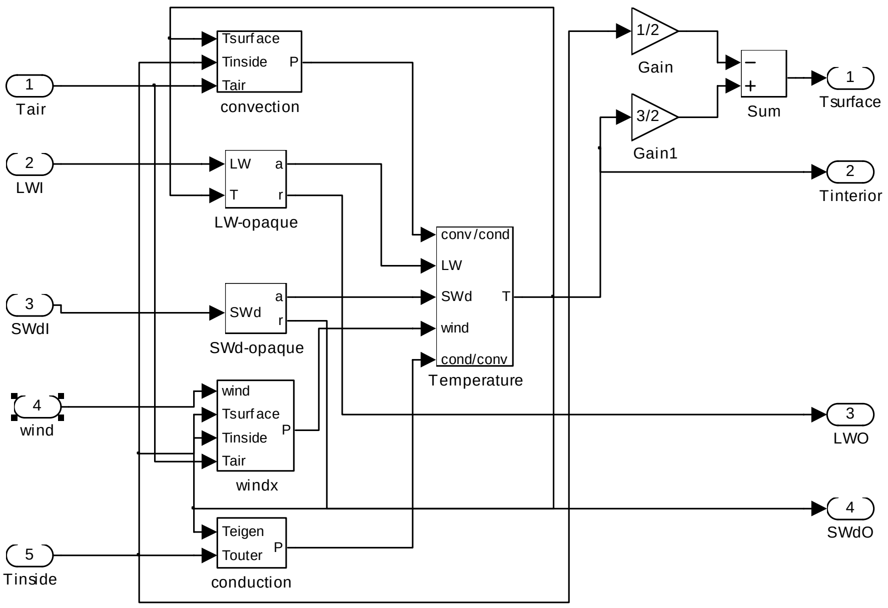

Figure 5.8 Inside the surface_opaque block.

Block	Prompt	Variable	Value	Initialization commands
surface_opaque	Tzero	Tzero		
	area	A		
	thickness	d		
	dist. from leading edge	y		
	F vector	F		
	material properties	mp		
	warm	w		
	cold	c		
temperature	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	A	
	thickness	d	d	
	density	ro	mp(1)	
	heatcapacity	Cp	mp(3)	
convection	area	A	A	global Pr g Kinematic_viscosity
	warm	w	w	
	cold	c	c	
	in solid or air	s	1	

LW-opaque	area	AA	A	global stime
	emission coefficient	e	mp(4)	
	absorption coefficient	a	mp(5)	
	reflectance coefficient	r	mp(6)	
	F vector	F	F	
SWd_opaque	absorption coefficient	a	mp(7)	global stime
	reflectance coefficient	r	mp(8)	
	F vector	F	F	
wind	area	A	A	global Kinematic_viscosity
	dist. from leading edge	y	y	
	air or solid	s	1	
conduction	area	A	A	
	thickness of layer	d	d	
	conductivity	lambda	mp(2)	

7.2.2 surface_glass

Differs from surface_opaque only in the way that the SWd_opaque block is exchanged for a SWd_glass block so that it allows SW radiation to be transmitted. The surface temperature is calculated according to (3.?).

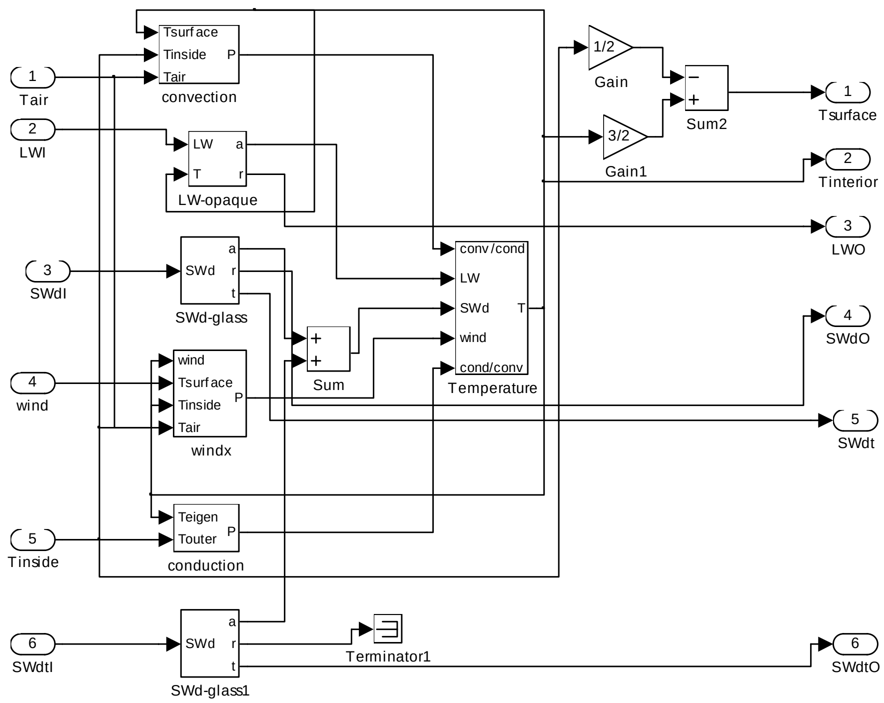

Figure 5.9 Inside the surface_glass block.

SURFACE_GLASS

Block	Prompt	Variable	Value	Initialization commands
surface_glass	Tzero	Tzero		mp(9)=mp(9)^(1/3);
	area	A		mp(7)=mp(7)/(1+mp(9)+mp(9)^2) ;
	thickness	d		
	dist. from leading edge	y		
	F vector	F		
	material propeties	mp		
	warm	w		
	cold	c		
temperature	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	A	
	thickness	d	d	
	density	ro	mp(1)	
	heatcapacity	Cp	mp(3)	
convection	area	A	A	global Pr g kinematic_viscosity
	warm	w	w	
	cold	c	c	
	in solid or air	s	1	
LW-opaque	area	AA	A	
	emission coefficient	e	mp(4)	
	absorption coefficient	a	mp(5)	
	reflectance coefficient	r	mp(6)	
	F vector	F	F	
SWd_glass	absorption coefficient	a	mp(7)	
	reflectance coefficient	r	mp(8)	
	transmission coefficient	t	mp(9)	
	F vector	F	F	
wind	area	A	A	global Kinematic_viscosity
	dist. from leading edge	y	y	
	air or solid	s	1	
SWd_glass1	absorption coefficient	a	mp(7)	
	reflectance coefficient	r	mp(8)	
	transmission coefficient	t	mp(9)	
	F vector	F	F	

7.2.3 insidesolid

Exchanges heat with two other layers of the same material by conduction.

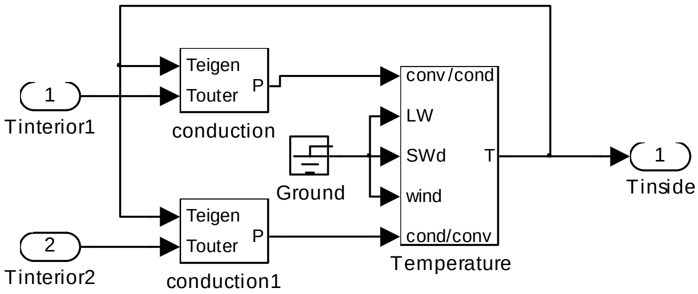

Figure 5.10 Inside the insidesolid block.

INSIDESOLID

Block	Prompt	Variable	Value	Initialization commands
insidesolid	Tzero	Tzero		
	area	A		
	thickness	d		
	material properties	mp		
temperature	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	A	
	thickness	d	d	
	density	ro	mp(1)	
	heatcapacity	Cp	mp(3)	
conduction	area	A	A	
	thickness of layer	d	d	
	conductivity	lambda	mp(2)	
conduction1	area	A	A	
	thickness of layer	d	d	
	conductivity	lambda	mp(2)	

7.2.4 insideglass

The insideglass block exchanges heat with two glass layers by conduction and also absorbs and transmits SW_radiation.

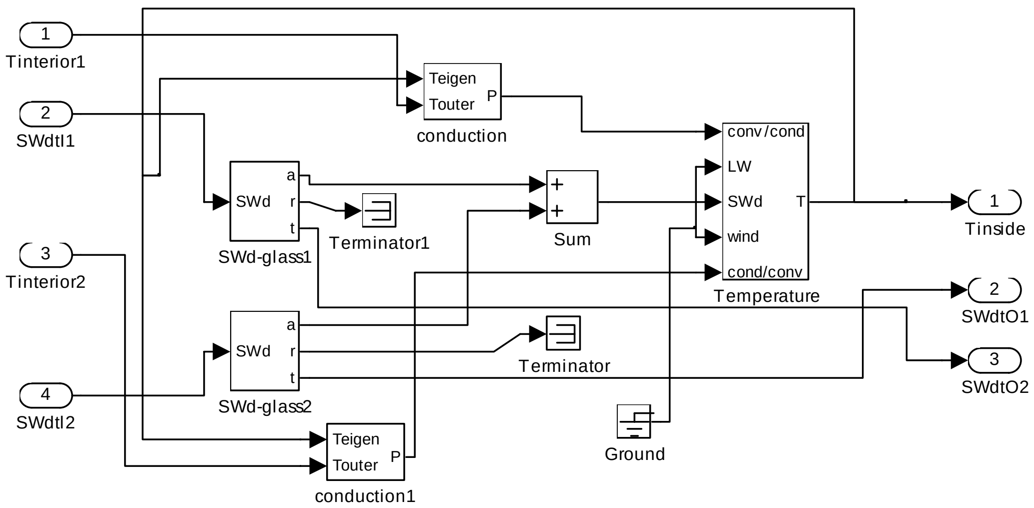

Figure 5.11 Inside the insideglass block.

INSIDEGLASS

Block	Prompt	Variable	Value	Initialization commands
insideglass	Tzero	Tzero		mp(9)=mp(9)^(1/3);
	area	A		mp(7)=mp(7)/(1+mp(9)+mp(9)^2);
	thickness	d		
	material properties	mp		
temperature	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	A	
	thickness	d	d	
	density	ro	mp(1)	
	heatcapacity	Cp	mp(3)	
conduction	area	A	A	
	thickness of layer	d	d	
	conductivity	lambda	mp(2)	
conduction1	area	A	A	
	thickness of layer	d	d	
	conductivity	lambda	mp(2)	
SWd_glass1	absorption coefficient	a	mp(7)	
	reflectance coefficient	r	mp(8)	
	transmission coefficient	t	mp(9)	
	F vector	F	1	
SWd_glass2	absorption coefficient	a	mp(7)	
	reflectance coefficient	r	mp(8)	
	transmission coefficient	t	mp(9)	
	F vector	F	1	

7.2.5 neighbour

The neighbour block interacts with one layer of the same material and one made up by another material. At the boarder between two layers with different material properties we have a change in the temperature slope. To estimate the temperature at the boarder I will look at the heat exchange between the two neighbouring layers. Let the temperature boarder be T_b and the temperature of the layers be T_{n1} and T_{n2}. Then we will have the state equations

$$C_{p1}m_1\dot{T}_{n1} = \alpha_1 \cdot A \cdot (T_b - T_{n1})$$
$$C_{p2}m_2\dot{T}_{n2} = \alpha_2 \cdot A \cdot (T_b - T_{n2})$$
(7.8),(7.9)

Using the fact, that by the energy conservation the heat loss of layer one must equal the heat gain of layer two, we solve for T and obtain

$$T_b = \frac{\alpha_1 T_{n1} + \alpha_2 T_{n2}}{\alpha_1 + \alpha_2} \qquad (7.10)$$

The two neighbour blocks exchange information of their alfa values so that the boarder temperature can be calculated in both blocks. This means that when one which to change material properties of one layer no parameters have to be changed in the neighbour block modelling the other layer.

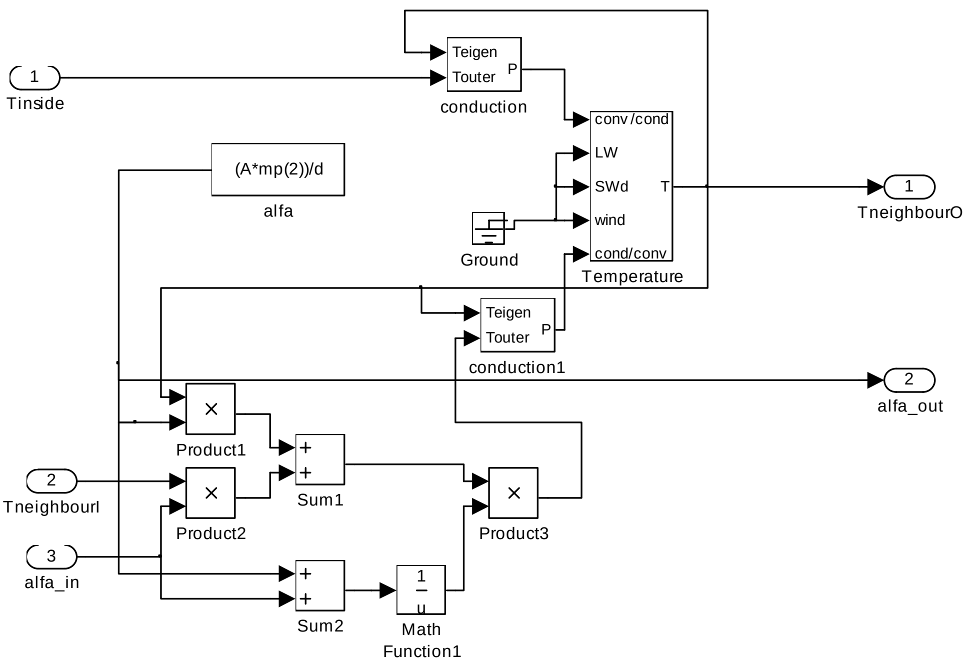

Figure 5.12 Inside the neighbour block.

NEIGHBOUR

Block	Prompt	Variable	Value	Initialization commands
neighbour	Tzero	Tzero		
	area	A		
	thickness	d		
	material properties	mp		
temperature	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	A	
	thickness	d	d	
	density	ro	mp(1)	
	heatcapacity	Cp	mp(3)	
conduction	area	A	A	
	thickness of layer	d	d	
	conductivity	lambda	mp(2)	
conduction1	area	A	A	
	thickness of layer	d	d/2	
	conductivity	lambda	mp(2)	

7.2 Level 3

7.3.1 W

W models a one material wall. It has one state

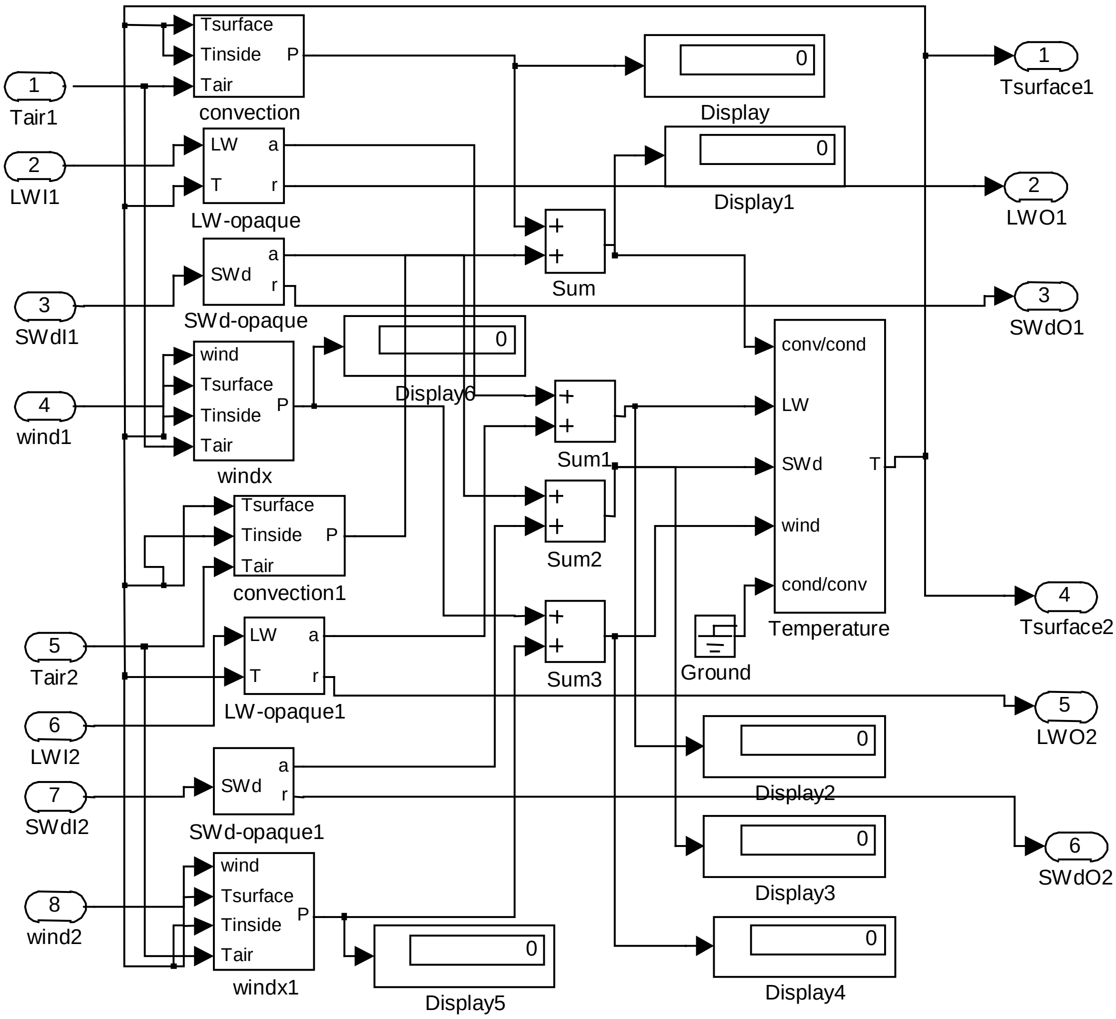

Figure 5.13 Inside the W block.

W

Block	Prompt	Variable	Value	Initialization commands
W	Tzero	Tzero		
	area	A		
	thickness	d		
	[dist. from leading edge]	y		
	F vector 1	F1		
	Fvector 2	F2		
	material properties	mp		
	up/down/vert 1	udv1		
	up/down/vert 2	udv2		

7.3.2 SIS

SIS is a block that models a one material wall that is divided into three equally thick layers. It consists of three blocks. Two Solidsurface blocks and one Insidesolid block. Each block has one state and as a total the SIS block has three states.

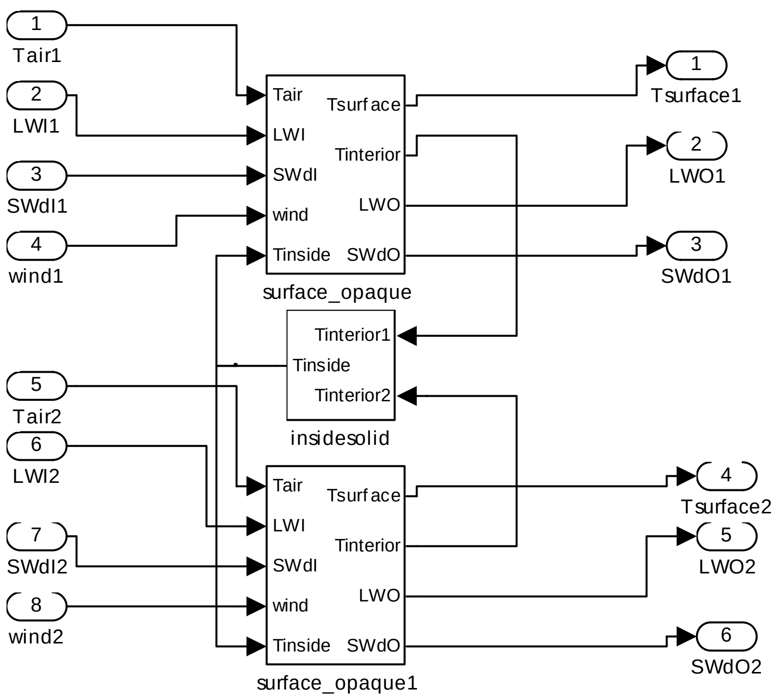

Figure 5.14 Inside the SIS block.

SIS

Block	Prompt	Variable	Value	Initialization commands
SIS	Tzero	Tzero		
	area	A		
	thickness	d		
	[dist. from leading edge]	y		
	F vector 1	F1		
	Fvector 2	F2		
	material properties	mp		
	up/down/vert 1	udv1		
	up/down/vert 2	udv2		
surface_opaque	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	dist. from leading edge	y	y(1)	
	F vector	F	F1	
	material properties	mp	mp	
	warm	w	udv1(1,:)	
	cold	c	udv1(2,:)	
surface_opaque	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	dist. from leading edge	y	y(2)	
	F vector	F	F2	
	material properties	mp	mp	
	warm	w	udv2(1,:)	
	cold	c	udv2(2,:)	
insidesolid	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	material properties	mp	mp	

7.3.3 GIG

A glass window is modelled by the GIG block. It is built up by two Glass_surface blocks and one Insideglass block. GIG has three states.

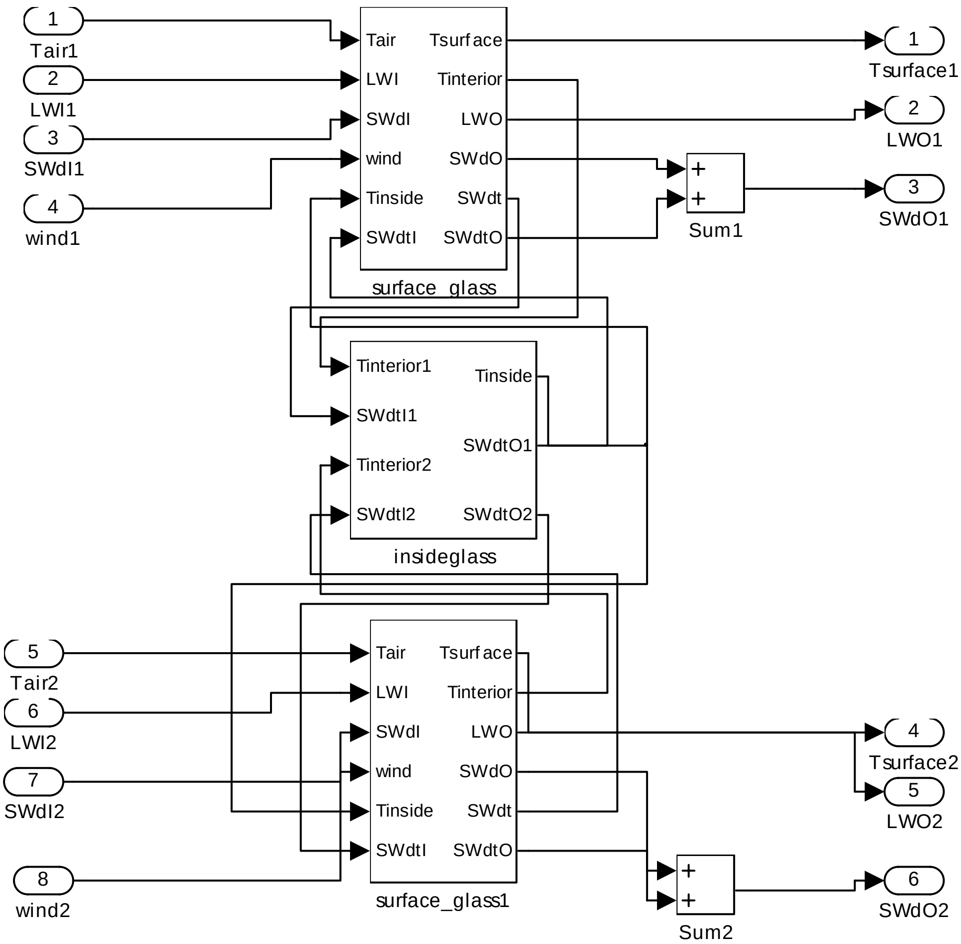

Figure 5.15 Inside the GIG block.

GIG

Block	Prompt	Variable	Value	Initialization commands
GIG	Tzero	Tzero		global vert
	area	A		
	thickness	d		
	[dist. from leading edge]	y		
	F vector 1	F1		
	Fvector 2	F2		
	material properties	mp		
	up/down/vert 1	udv1		
	up/down/vert 2	udv2		
surface_glass	Tzero	Tzero	Tzero	mp(9)=mp(9)^(1/3);
	area	A	A	mp(7)=mp(7)/(1+mp(9)+mp(9)^2);
	thickness	d	d/3	
	dist. from leading edge	y	y(1)	
	F vector	F	F1	
	material propeties	mp	mp	
	warm	w	udv1(1,:)	
	cold	c	udv1(2,:)	
surface_glass1	Tzero	Tzero	Tzero	mp(9)=mp(9)^(1/3);
	area	A	A	mp(7)=mp(7)/(1+mp(9)+mp(9)^2);
	thickness	d	d/3	
	dist. from leading edge	y	y(2)	
	F vector	F	F2	
	material propeties	mp	mp	
	warm	w	udv2(1,:)	
	cold	c	udv2(2,:)	
insideglass	Tzero	Tzero	Tzero	mp(9)=mp(9)^(1/3);
	area	A	A	mp(7)=mp(7)/(1+mp(9)+mp(9)^2);
	thickness	d	d/3	
	material properties	mp	mp	

7.3.4 SIN

SIN is a block that models a one material wall layer, which is, divided into three equally thick layers. It includes three blocks with one state each, one Solidsurface block, one Insidesolid block and one Neighbour block. The Neighbour block has one state and is there to make is possible for the wall layer to interact with another wall layer that is made by a different material

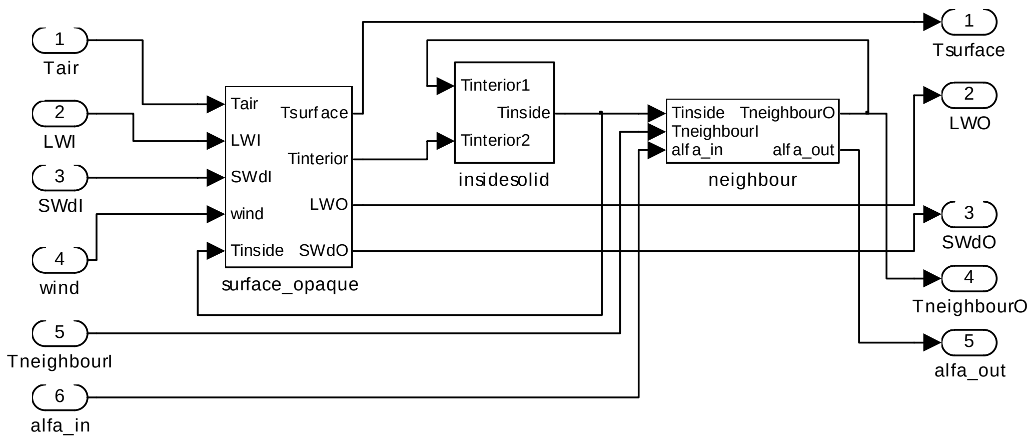

Figure 5.16 Inside the GIG block.

SIN

Block	Prompt	Variable	Value	Initialization commands
SIN	Tzero	Tzero		
	area	A		
	thickness	d		
	dist. from leading edge	y		
	F vector	F		
	material properties	mp		
	up/down/vert	udv		
surface_opaque	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	dist. from leading edge	y	y	
	F vector	F	F	
	material properties	mp	mp	
	warm	w	udv(1,:)	
	cold	c	udv(2,:)	
neighbour	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	material properties	mp	mp	
insidesolid	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	material properties	mp	mp	

7.3.5 NIN

NIN models an one material wall layer which is divided into three equally thick layers. The NIN block is used when the wall layer has two neighbouring layers of other material. It has three states.

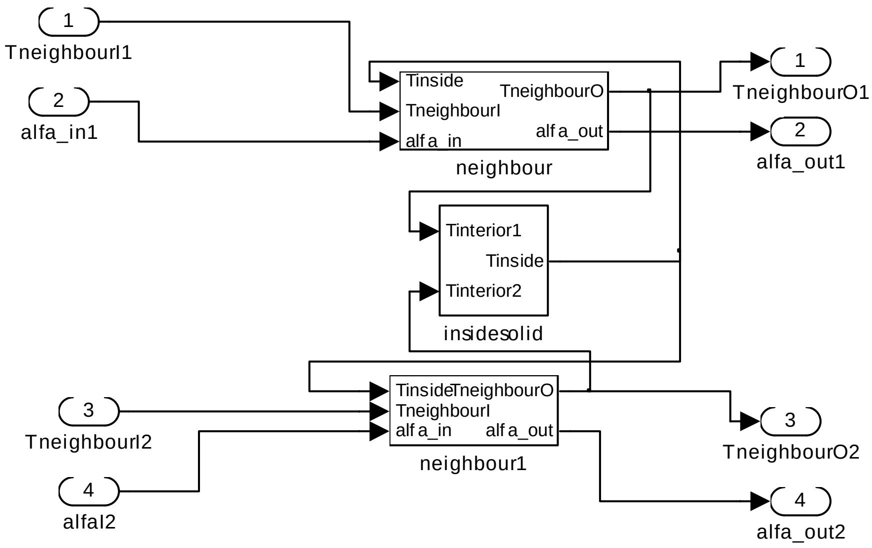

Figure 5.17 Inside the NIN block.

NIN

Block	Prompt	Variable	Value	Initialization commands
NIN	Tzero	Tzero		global steel_data plastic_data
	area	A		global foam_data rubber_data
	thickness	d		
	material properties	mp		
neighbour	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	material properties	mp	mp	
neighbour	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	material properties	mp	mp	
insidesolid	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d/3	
	material properties	mp	mp	

7.3.6 isoair

The isolationAir block is labeled A and has two states. One is the air temperature and the other one is the temperature of the steel balks that is placed in the isolation air area. The wind blocks models the forced convection caused by the movement of the isolation air. The amount of movement is controlled by the wind input.

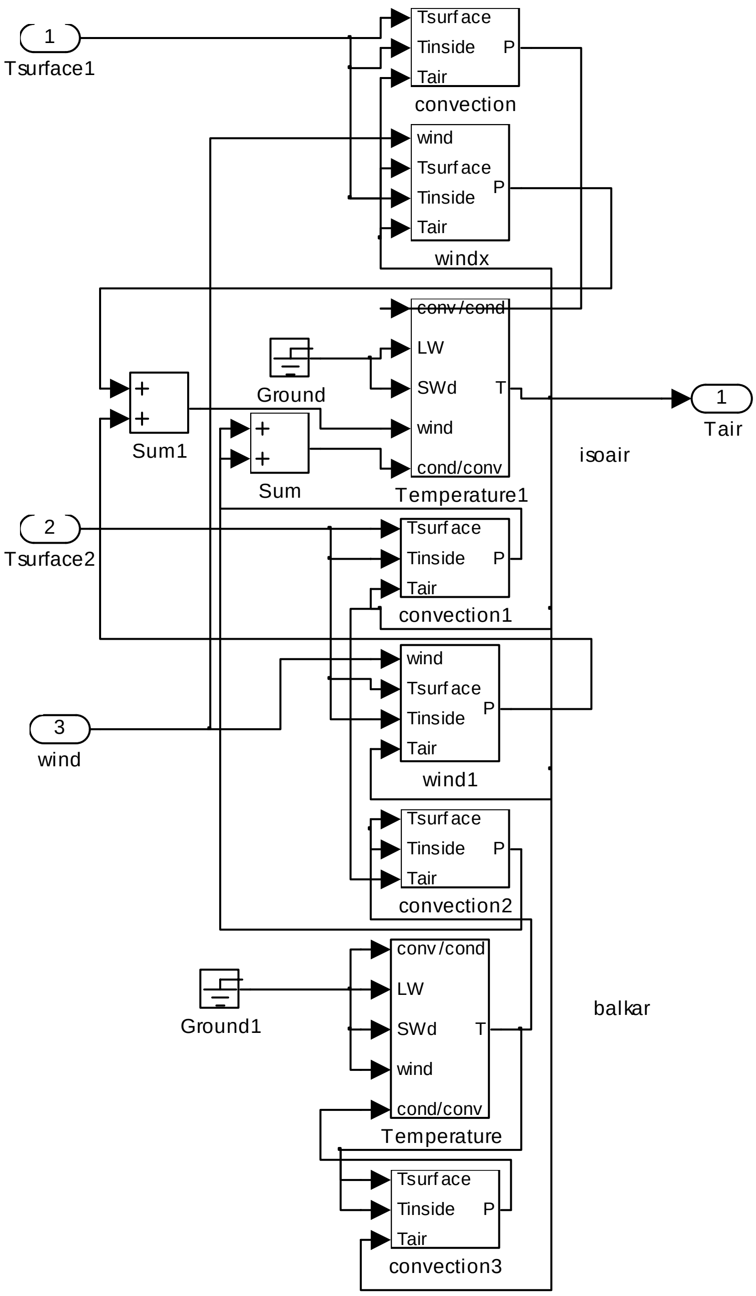

Figure 5.18 Inside the isolationAir block.

ISOAIR

Block	Prompt	Variable	Value	Initialization commands
isoair	Tzero	Tzero		global vert
	[area]	A		
	thickness	d		
	dist. from leading edge	y		
	material properties air	mp1		
	material properties balk	mp2		
	up/down/vert 1	udv1		
	up/down/vert 2	udv2		
temperature	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	A(1)/100	
	thickness	d	d	
	density	ro	mp2(1)	
	heatcapacity	Cp	mp2(3)	
temperature1	Tzero	Tzero	Tzero	mass=A*d*ro;
	area	A	(A(1)+A(2))/2	
	thickness	d	d	
	density	ro	mp1(1)	
	heatcapacity	Cp	mp1(3)	
convection	area	A	A(1)	global Pr g kinematic_viscosity
	warm	w	udv1(1,:)	
	cold	c	udv1(2,:)	
	in solid or air	s	-1	
convection1	area	A	A(2)	global Pr g kinematic_viscosity
	warm	w	udv1(1,:)	
	cold	c	udv1(2,:)	
	in solid or air	s	-1	
convection2	area	A	A(1)/100	global Pr g kinematic_viscosity
	warm	w	vert	
	cold	c	vert	
	in solid or air	s	-1	
convection3	area	A	A(1)/100	global Pr g kinematic_viscosity
	warm	w	vert	
	cold	c	vert	
	in solid or air	s	1	
windx	area	A	A(1)	global Kinematic_viscosity
	dist. from leading edge	y	y	
	air or solid	s	-1	
wind1	area	A	A(2)	global Kinematic_viscosity
	dist. from leading edge	y	y	
	air or solid	s	-1	

7.4 Level 4

7.4.1 WAW

WAW models a double wall with isolation air and has four states. WAW has the same parameters, inputs and outputs as SISASIS se chapter 7.4.3.

7.4.3 WASIS

WASIS models a double wall with isolation air and has six states. WASIS has the same parameters, inputs and outputs as SISASIS se chapter 7.4.3.

7.4.4 SISASIS

SISASIS consist of two SIS blocks and one isolationAir block. It has been made to be able to model the heat flow through a double wall with isolation air between them. The SISASIS block has eight states. The motion of the isolation air is given by the circulation input.

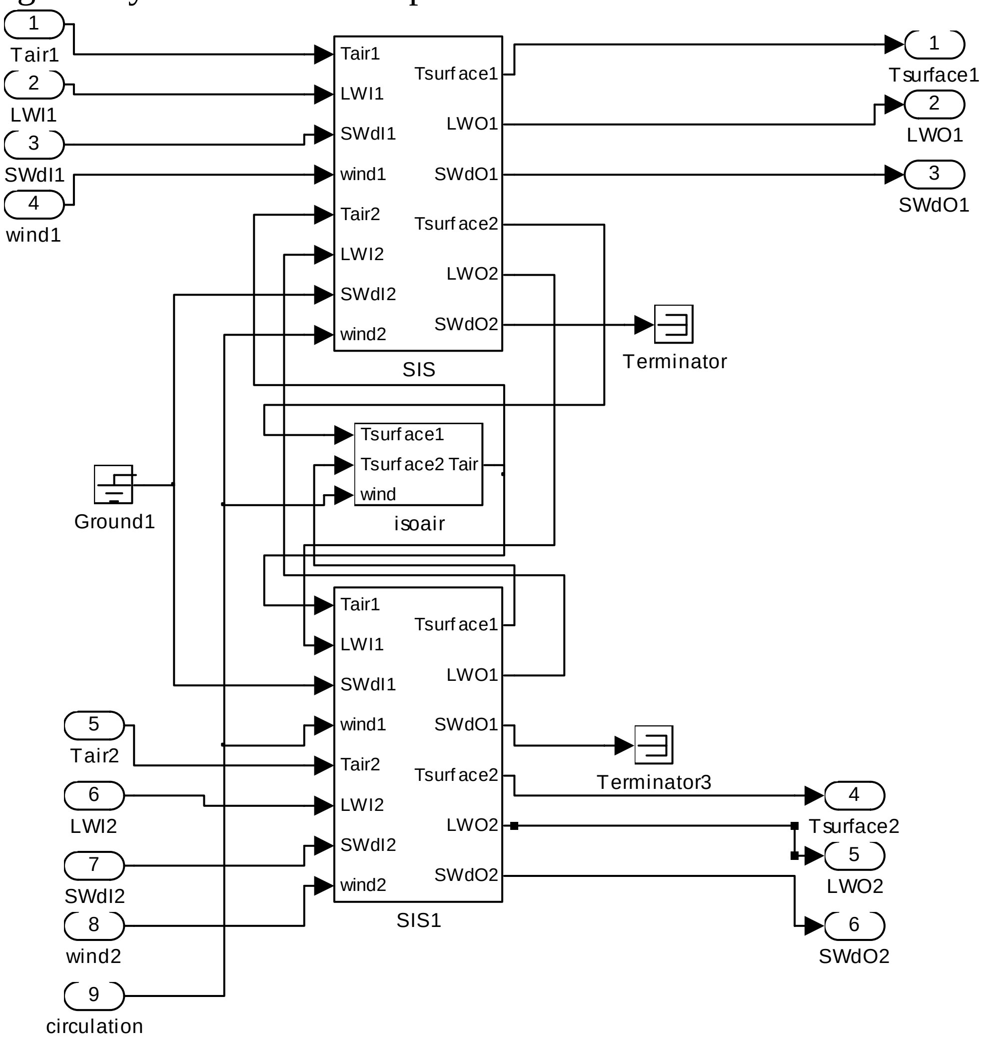

Figure 7.19 Inside the SISASIS block.

SISASIS

Block	Prompt	Variable	Value	Initialization commands	
SISASIS	Tzero	Tzero		global air_data steel_data	
	[area]	A			
	[thickness]	d			
	[dist. from leading edge]	y			
	F vector 1	F1			
	F vector 2	F2			
	up/down/vert	udv1			
	up/down/vert	udv2			
	material properties 1	mp1			
	material properties 2	mp2			
SIS	Tzero	Tzero	Tzero		
	area	A	A(1)		
	thickness	d	d(1)		
	[dist. from leading edge]	y	[y(1) y(1)]		
	F vector 1	F1	F1		
	Fvector 2	F2	1		
	material properties	mp	mp1		
	up/down/vert 1	udv1	udv1		
	up/down/vert 2	udv2	udv2		
SIS	Tzero	Tzero	Tzero		
	area	A	A(2)		
	thickness	d	d(3)		
	[dist. from leading edge]	y	[y(2) y(2)]		
	F vector 1	F1	1		
	Fvector 2	F2	F2		
	material properties	mp	mp2		
	up/down/vert 1	udv1	udv1		
	up/down/vert 2	udv2	udv2		
isoair	Tzero	Tzero	Tzero		global vert
	[area]	A	[A(1) A(2)]		
	thickness	d	d(2)		
	dist. from leading edge	y	1		
	material properties air	mp1	air_data		
	material properties balk	mp2	steel_data		
	up/down/vert 1	udv1	udv2		
	up/down/vert 2	udv2	udv1		

7.4.4 SINNIS

SINNIS is made up by two SIN blocks. It models a wall with two different material layers. The SINNIS block has six states.

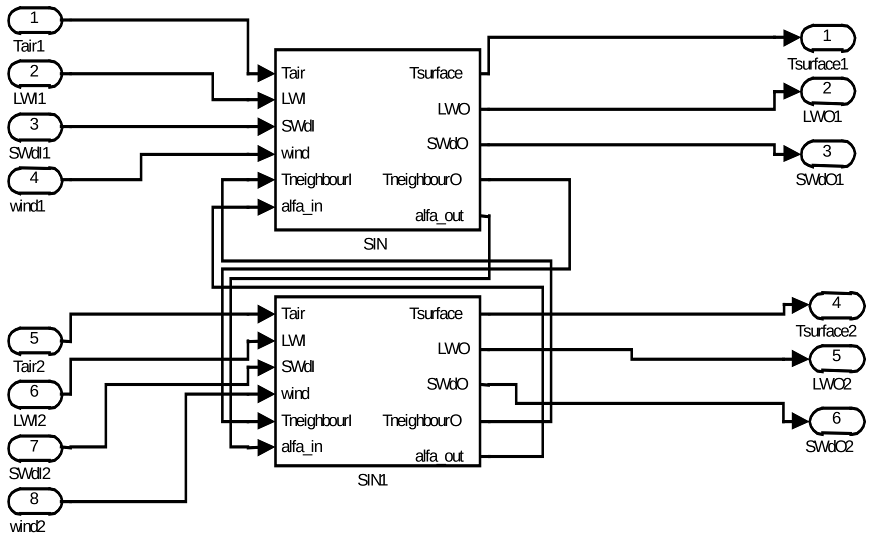

Figure 5.20 Inside the SINNIS block.

SINNIS

Block	Prompt	Variable	Value	Initialization commands
SINNIS	Tzero	Tzero		(global steel_data rubber_data)
	area	A		of no use
	[thickness]	d		
	[dist. from leading edge]	y		
	F vector 1	F1		
	F vector 2	F2		
	up/down/vert 1	udv1		
	up/down/vert 2	udv2		
	material properties 1	mp1		
	material properties 2	mp2		
SIN	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d(1)	
	dist. from leading edge	y	y(1)	
	F vector	F	F1	
	material properties	mp	mp1	
	up/down/vert	udv	udv1	
SIN1	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d(2)	
	dist. from leading edge	y	y(2)	
	F vector	F	F2	
	material properties	mp	mp2	
	up/down/vert	udv	udv2	

7.4.5 SINNINNINNIS

If one wants to model a wall with for example four different material layers one inserts two NIN blocks between the two SIN blocks. The SINNINNINNIS block is then created. The SINNINNINNIS block has 12 states.

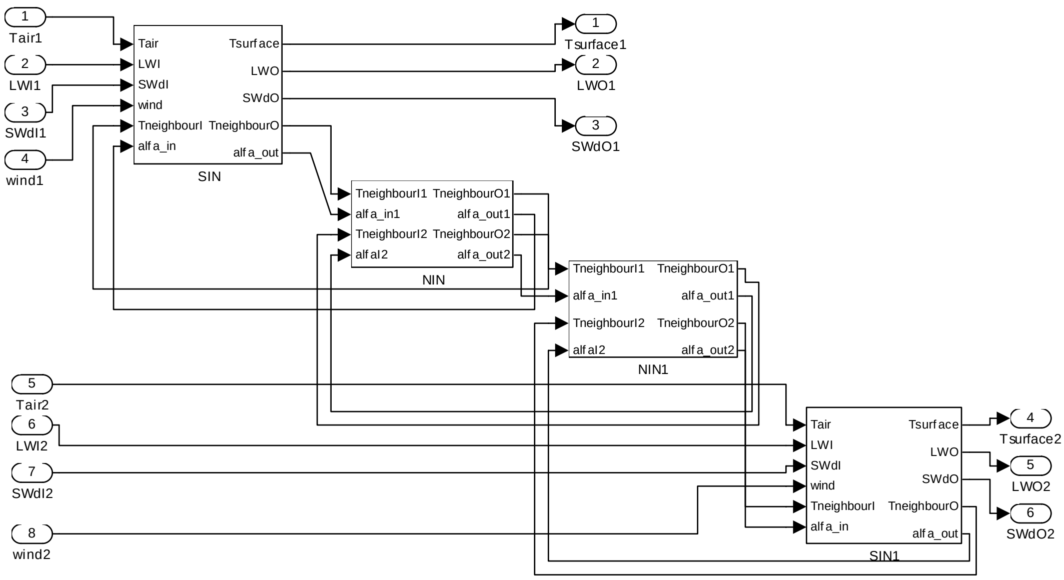

Figure 5.21 Inside the SINNINNINNIS block.

SINNINNINIS

Block	Prompt	Variable	Value	Initialization commands
SINNINNINIS	Tzero	Tzero		global steel_data foam_data
	area	A		global rubber_data plastic_data
	[thickness]	d		
	[dist. from leading edge]	y		
	F vector 1	F1		
	F vector 2	F2		
	up/down/vert 1	udv1		
	up/down/vert 2	udv2		
	material properties 1	mp1		
	material properties 2	mp2		
	material properties 3	mp3		
	material properties 4	mp4		
SIN	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d(1)	
	dist. from leading edge	y	y(1)	
	F vector	F	F1	
	material properties	mp	mp1	
	up/down/vert	udv	udv1	

SIN1	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d(4)	
	dist. from leading edge	y	y(2)	
	F vector	F	F2	
	material properties	mp	mp4	
	up/down/vert	udv	udv2	
NIN	Tzero	Tzero	Tzero	global steel_data plastic_data
	area	A	A	global foam_data rubber_data
	thickness	d	d(2)	
	material properties	mp	mp2	
NIN1	Tzero	Tzero	Tzero	global steel_data plastic_data
	area	A	A	global foam_data rubber_data
	thickness	d	d(3)	
	material properties	mp	mp3	

7.4.6 SISASINNIS

Exchanging one of the SIS blocks for a SINNINS block makes a SISASINNIS block with 10 states.

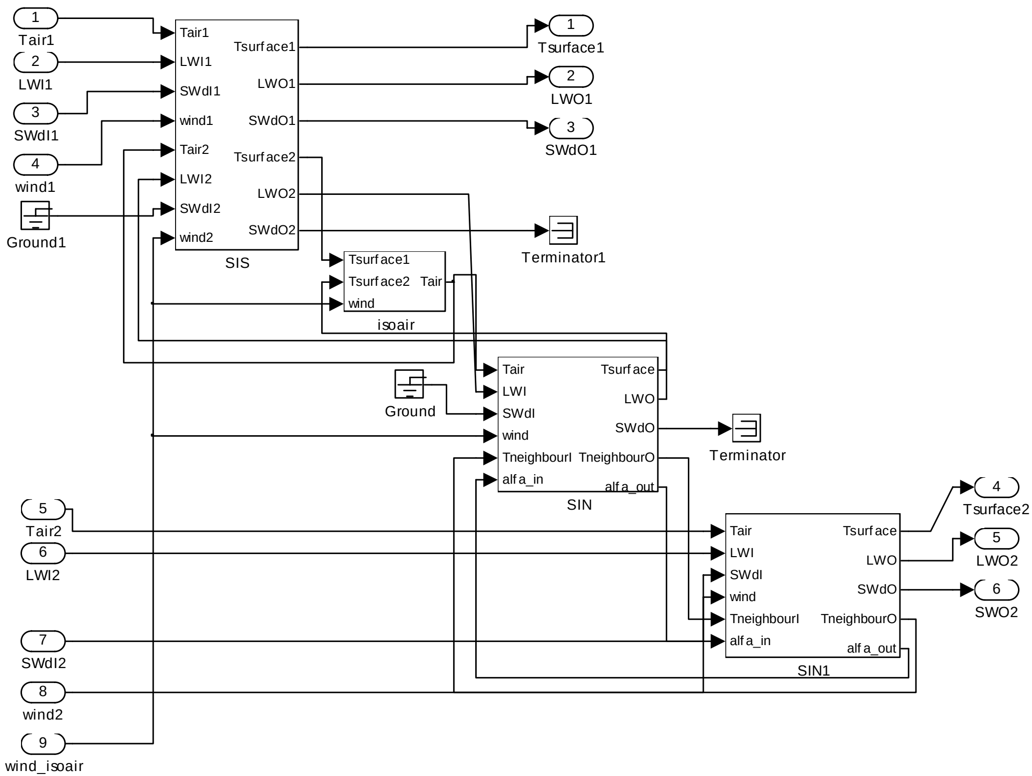

Figure 5.23 Inside the SISASINNIS block.

SISASINNIS

Block	Prompt	Variable	Value	Initialization commands
SISASINNIS	Tzero	Tzero		global air_data steel_data
	[area]	A		
	[thickness]	d		
	[dist. from leading edge]	y		
	F vector 1	F1		
	F vector 2	F2		
	up/down/vert 1	udv1		
	up/down/vert 2	udv2		
	material properties 1	mp1		
	material properties 2	mp2		
	material properties 3	mp3		
SIS	Tzero	Tzero	Tzero	
	area	A	A(1)	
	thickness	d	d(1)	
	[dist. from leading edge]	y	[y(1) 1]	
	F vector 1	F1	F1	
	Fvector 2	F2	1	
	material properties	mp	mp1	
	up/down/vert 1	udv1	udv1	
	up/down/vert 2	udv2	udv2	
SIN	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d(3)	
	dist. from leading edge	y	1	
	F vector	F	1	
	material properties	mp	mp2	
	up/down/vert	udv	udv1	
SIN	Tzero	Tzero	Tzero	
	area	A	A	
	thickness	d	d(4)	
	dist. from leading edge	y	y(2)	
	F vector	F	F2	
	material properties	mp	mp3	
	up/down/vert	udv	udv2	
isoair	Tzero	Tzero	Tzero	global vert
	[area]	A	[A(1) A(2)]	
	thickness	d	d(2)	
	dist. from leading edge	y	1	
	material properties air	mp1	air_data	
	material properties balk	mp2	steel_data	
	up/down/vert 1	udv1	udv2	
	up/down/vert 2	udv2	udv1	

7.5 Matlab files

7.5.1 bodydata.m

```
global Pr thermal_diffusivity kinematic_viscosity
global g glass_data wood_data steel_data foam_data plastic_data air_data
rubber_data
global up down vert td

g=9.81;
thermal_diffusivity=2*10^(-5);
kinematic_viscosity=16.7*10^(-6);
Pr=0.8;

%up=[0.15 1/3;0.27 1/4];
%down=[0.27 1/4;0.15 1/3];
vert=[0.59 1/4;0.59 1/4];
up=[0.59 1/4;0.59 1/4];
down=[0.59 1/4;0.59 1/4];
%vert=[0.10 1/3;0.10 1/3];
%up=[0.10 1/3;0.10 1/3];
%down=[0.10 1/3;0.10 1/3];

%PHYSICS HANDBOOK Steel C 0.85

conductivity_steel=45;
density_steel=7.8*10^3;
heatcapacity_steel=0.46*10^3;
LW_absorption_steel=0.2;
LW_reflectance_steel=0.8;
LW_emission_steel=0.2;
SW_absorption_steel=0.5;
SW_reflectance_steel=0.5;
SWd_absorption_steel=0.5;
SWd_reflectance_steel=0.5;

steel_data=[density_steel conductivity_steel heatcapacity_steel
LW_emission_steel LW_absorption_steel LW_reflectance_steel
SW_absorption_steel SW_reflectance_steel 0.5];

%data for wood

conductivity_wood=0.14;   %
density_wood=0.52*10^3;
heatcapacity_wood=0.4*10^3;
LW_absorption_wood=0.2;
LW_reflectance_wood=0.8;
LW_emission_wood=0.2;
SW_absorption_wood=0.5;
SW_reflectance_wood=0.5;
SWd_absorption_wood=0.5;
SWd_reflectance_wood=0.5;

wood_data=[density_wood conductivity_wood heatcapacity_wood
LW_emission_wood LW_absorption_wood LW_reflectance_wood SW_absorption_wood
SW_reflectance_wood 0.5];
```

```
%isoair data

conductivity_air=0.026;
density_air=1.293;
heatcapacity_air=1.01*10^3;

air_data=[density_air conductivity_air heatcapacity_air LW_emission_steel
LW_absorption_steel LW_reflectance_steel SW_absorption_steel
SW_reflectance_steel 0.5];

% glass window i Bosh
conductivity_glass=0.81;
density_glass=2.5*10^3;
heatcapacity_glass=0.83*10^3;
LW_absorption_glass=0.7;
LW_reflectance_glass=0.3;
LW_emission_glass=0.7;
SW_absorption_glass=0.20; % Pilkington
SW_reflectance_glass=0.13;
SW_transmission_glass=0.67;

glass_data=[density_glass conductivity_glass heatcapacity_glass
LW_emission_glass LW_absorption_glass LW_reflectance_glass
SW_absorption_glass SW_reflectance_glass SW_transmission_glass];


conductivity_rubber=0.14;   % rubber raw
density_rubber=0.92*10^3;  % Bosh rubber raw
heatcapacity_rubber=1.42*10^3; %ebonit
LW_absorption_rubber=0.2;
LW_reflectance_rubber=0.8;
LW_emission_rubber=0.2;
SW_absorption_rubber=0.2;
SW_reflectance_rubber=0.8;
SW_transmission_rubber=0.2;

rubber_data=[density_rubber conductivity_rubber heatcapacity_rubber
LW_emission_rubber LW_absorption_rubber LW_reflectance_rubber
SW_absorption_rubber SW_reflectance_rubber SW_transmission_glass];

%rigid plastic foam

conductivity_foam=0.036;
density_foam=0.015*10^3;
heatcapacity_foam=1.42*10^3; %ebonit
LW_absorption_foam=0.2;
LW_reflectance_foam=0.8;
LW_emission_foam=0.2;
SW_absorption_foam=0.2;
SW_reflectance_foam=0.8;
SW_transmission_foam=0.2;

foam_data=[density_foam conductivity_foam heatcapacity_foam
LW_emission_foam LW_absorption_foam LW_reflectance_foam SW_absorption_foam
SW_reflectance_foam SW_transmission_glass];
```

```
% polyethylene

conductivity_plastic=0.41;
density_plastic=0.94*10^3;
heatcapacity_plastic=2.1*10^3;
LW_absorption_plastic=0.2;
LW_reflectance_plastic=0.8;
LW_emission_plastic=0.2;
SW_absorption_plastic=0.2;
SW_reflectance_plastic=0.8;
SW_transmission_plastic=0.2;

plastic_data=[density_plastic conductivity_plastic heatcapacity_plastic
LW_emission_plastic LW_absorption_plastic LW_reflectance_plastic
SW_absorption_plastic SW_reflectance_plastic SW_transmission_glass];


Tzero=273
```

7.5.2 dimdata.m

```
global dim_roof dim_front dim_floor dim_back dim_right dim_left
global K_roof K_front K_floor K_back K_right K_left
global area interiorarea airvolume

dim_roof=2;
dim_front=3;
dim_floor=3+2;
dim_back=2;
dim_side=4+4;
dim_box=2+2;


interiorarea=7;  % area of extra surface in cabin
airvolume=6.8;   % air volume in the cabin
tjockis=0.064

% the areas of the 24 inner surfaces
area=[4 0.16 0.8 1.6 0.75 0.55 0.4 0.65 0.4 0.55 1.7 0.95 0.5 1 0.5 0.5 0.5
1 0.5 0.5 0.35 0.3 0.35 0.3];
% area(1)= roof
% area(2)= roof window
% area(3)= top of the front
% area(4)= front window
% area(5)= below front window
% area(6)= left hand floor
% area(7)= left hand side of the engine tunnel
% area(8)= top of the engine tunnel
% area(9)= right hand side of the engine tunnel
% area(10)= right hand floor
% area(11)= upper part of back
% area(12)= lower part of back
% area(13)= left side up
% area(14)= left side back
% area(15)= left side door
% area(16)= left side window
% area(17)= right side up
% area(18)= right side back
```

```
% area(19)= right side door
% area(20)= right side window
% area(21)= left box up
% area(22)= left box front
% area(23)= right box up
% area(24)= right box front


% calculation of the view factors

for i=1:24
   K(:,i)=area';
end

for i=1:24
   K(:,i)=(1/(norm(K(:,i),1)))*K(:,i); % normering
end

K_roof=K([1 2],:);
K_front=K([3 4 5],:);
K_floor=K([6 7 8 9 10],:);
K_back=K([11 12],:);
K_side=K([13 14 15 16 17 18 19 20],:);
K_box=K([21 22 23 24],:);
```

8. References

1. Jack Philip Holman, Heat transfer, New York 1997

2. Bo Pierre, Mekanisk värmeteori:fortsättningskurs del 1 och 2, Stockholm 1982

3. Torkel Glad, Lennart Ljung, Reglerteori:flervariabla och olinjära metoder Lund:studentlitteratur 1997

4. Lennart Ljung och Torkel Glad, Modellbygge och simulering, Lund studentlitteratur 1991

5. Charles Kittel, Introduction to solidstate physics, New York:Wiley 1996

6. Per-Erik Lindahl, Mätning av icke-elektriska storheter, Lund, Studentlitteratur 1975