Modeling and Analysis of Economy of HVAC System for Automobiles with Vehicle Motion Simulator

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Modeling and Analysis of Economy of HVAC System for Automobiles with Vehicle Motion Simulator

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ABSTRACT

Automobiles are consuming a great quantity of fossil fuel, emitting a great quantity of greenhouse gas CO2. This has caused the global temperature rising problems, leading to the damage of global environment.

Generally, HVAC system (heating, ventilation, and air conditioning) is the single largest auxiliary load on a vehicle. This implies that the HVAC system is the key component among all accessories which causes the fuel economy problem. Therefore, it is important to verify the primary parameter of the components of the HVAC system which affect the fuel consumption. This thesis presents the effect of heat transfer of the HVAC system in order to the fuel consumption of a conventional gasoline engine vehicle operating at various engine speed.

Heat transfer is a physical phenomenon in physics, which refers to the heat energy source transfer phenomenon caused by temperature difference. At the same time, heat transfer is also the most important part in HVAC system. In heat transfer, the change of internal energy is measured by heat. There are three basic forms of heat transfer: heat conduction, heat radiation and heat convection. Where there is a temperature difference inside or between objects, thermal energy must be transferred from high thermal source to low thermal source in one or more of the above three ways.

The energy required by traditional cars is gasoline. When the car is stationary, it can be said that it does not consume fuel. However, when the car is running, not only the control of the throttle will consume fuel, but also the HVAC system in heat exchange due to the low or high external temperature is also necessary. The three basic forms of heat transfer: heat conduction, heat radiation and heat convection will also have an important impact on the heat transfer of automobiles. When the vehicle is running at high speed, the surface (glasses, doors and windows) of the vehicle will have extremely fast air flow, which will bring the heat exchange between the body temperature and the natural environment temperature. With the increase of vehicle speed, the flow rate of air flowing through the vehicle surface will also increase, and the heat transfer coefficient will also change with the surface air flow rate, which will affect the change rate of temperature in the cabin.

In high-speed vehicles, the convective heat transfer between the vehicle surface and the surrounding air is the most important factor affecting the temperature in the cabin and is also the most focus part of this paper. The vehicle speed affects the air flow rate on the vehicle surface, and the air flow rate on the vehicle surface affects the convective

heat transfer coefficient between the vehicle surface and the surrounding air, so that the convective heat transfer on the vehicle surface causes a large amount of heat exchange in a short time.

Simulink 3D Animation™ provides the function that can associate Simulink® model and MATLAB® algorithm with 3D graphic objects in virtual reality scenes. Virtual 3d world animation can be done by changing the position, rotation, scale, and other object properties during desktop or real-time simulation. It can also sense collisions and other events in the virtual world and feed them back to MATLAB and Simulink algorithms. The video from the virtual camera can be streamed to Simulink for further processing. For more understanding for the vehicle behavior during the motion, a virtual reality model has been developed using the Simulink 3D animation toolbox. The structure of the SIMULINK model for the virtual reality environment has been used to realize the motion of the vehicle. A virtual model for the vehicle has been built and the trajectory for the motion of the mass center will be plotted during the simulation. In the virtual reality model, the change in the position coordinates of the center of mass has presented live during the simulation to track the change in the position coordinates.

KEY WORDS: HVAC system, Heat Transfer Coefficient, Convection Heat, Simulink 3D Animation

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Nomenclature

Symbols

A_{Du}	DuBois area [m^2]
c_ρ	specific heat at constant pressure [J/ (kg * K)]
e_i	cabin enthalpies[J]
e_o	cabin enthalpies[J]
H	passenger height[m]
h	heat transfer coefficient [W/(m^2*K)]
h_o	outside convection coefficient[W/(m^2*K)]
h_i	inside convection coefficient[W/(m^2*K)]
$\dot{I}_{Dif}$	the diffuse radiation heat gain per unit area[J]
$\dot{I}_{Dir}$	the direct radiation heat gain per unit area[J]
$\dot{I}_{Ref}$	the reflected radiation heat gain per unit area[J]
k	surface thermal conductivity[°C/m]
$\dot{m}_{Ven}$	ventilation mass flow rate[1/s]
M	passenger metabolic production rate [KJ/s]
Nu_x	Nusselt number
P	air pressure [KPa]
Pr	Prandtl Number
P_s	water saturation pressure [KPa]
$\dot{Q}_{AC}$	thermal load of AC cycle [KJ]
$\dot{Q}_{Amb}$	ambient thermal load [KJ]
$\dot{Q}_{Dif}$	diffuse thermal load [KJ]

$\dot{Q}_{Dir}$	direct thermal load [KJ]
$\dot{Q}_{Eng}$	engine thermal load [KJ]
$\dot{Q}_{Exh}$	exhaust thermal load [KJ]
$\dot{Q}_{Met}$	metabolic thermal load [KJ]
$\dot{Q}_{Ref}$	reflected radiation thermal load [KJ]
$\dot{Q}_{Tot}$	overall thermal load [KJ]
$\dot{Q}_{Ven}$	load generated due to ventilation [KJ]
R	thermal resistance[°C/W]
Re_x	Reynolds number
RPM	engine speed in revolutions per minute[r/min]
S	surface area [m^2]
S_{Eng}	surface area exposed to the engine temperature [m^2]
S_{Exh}	surface area [m^2]
t_c	a pull-down constant which determines the overall pull-down time
T_{comf}	target comfort temperature [°C]
T_{Exh}	surface area temperature [°C]
T_i	average surface temperature [°C]
T_s	average cabin temperature [°C]
u_∞	free-stream velocity [m/s]
U	Reciprocal of R
v	flow velocity[m/s]
V	the vehicle speed in [m/s]
W	passenger weight [kg]

x	distance from leading edge[m]
X	humidity ratio in gram of water per gram of dry air
α	surface absorptivity [KJ/m^2]
θ	angle between the surface normal and the position of sun in the sky
λ	the thickness of the surface element[m]/ Thermal conductivity [$W/(m * K)$]
ϕ	relative humidity
ρ	fluid density [kg/m^2]
μ	fluid viscosity coefficient

Chapter1: Introduction

Increasing standards of living coupled with dwindling supplies of fossil fuels, have forced researchers and engineers to focus on the issue of energy use in buildings. Heating, ventilation and air conditioning (HVAC) systems, which play an important role in ensuring occupant comfort, are among the largest energy consumers in buildings. Performance enhancements to traditional HVAC systems therefore offer an exciting opportunity for significant reductions in energy consumption. Almost 50% of the energy demand is used to support indoor thermal comfort conditions in commercial buildings . Furthermore, as most people spend more than 90% of their time inside, the development of energy-efficient HVAC systems that do not rely on fossil fuels will play a key role in reducing energy consumption [1]. A closer look at world-wide energy consumption by HVAC equipment shows noticeable values: HVAC systems constitute over 50% of the building energy consumed in US. In China, the building energy consumption has been increasing about 10% a year for the past 20 years and has comprised about 20.7% of the total national energy usage. In Europe, around 40% of energy consumption is represented in commercial and residential buildings. In Australia 70% of power consumption in non-residential buildings is used for HVAC systems. In India, air conditioning systems accounts around 32% of electricity consumption of a building. In the subtropical Hong Kong air conditioning and refrigeration systems accounted for 33%. More than 70% of building energy consumption is to support cooling systems in Middle East [2]. However, approximately 80% of the energy usage still comes from fossil fuels. The growing reliance on HVAC systems in residential, commercial and industrial environments has resulted in a huge increase in energy usage, particularly in the summer months. Developing energy efficient HVAC systems is essential, both to protect consumers from surging power costs and to protect the environment from the adverse impacts of greenhouse gas emissions caused by using of energy-inefficient electrical appliances. With rapid changes in science and technology today, there are several methods that can be used to achieve energy-efficient HVAC systems. In order to develop efficient systems, however, a clear understanding of building comfort conditions is necessary. Thermal comfort is all about human satisfaction with their thermal environment. The design and calculation of air conditioning systems to control the thermal environment in a way that also achieves an acceptable standard of air quality inside a building should comply with the ASHRAE standard 55-2004 [3]. According to this standard, thermal comfort conditions are acceptable when 80% of the building's occupants are satisfied. In order to predict appropriate thermal comfort conditions an index called a predicted mean vote (PMV), which indicates mean the thermal sensation vote on a standard scale for a large group of people, is used. PMV is defined by six thermal variables for an indoor environment, subject to human comfort: air temperature, air humidity, air velocity, mean radiant temperature, clothing insulation and human activity. The PMV index predicts the mean value of the votes on the seven-point thermal sensation scales; +3: hot, +2: warm, +1: slightly warm, 0: neutral, −1: slightly cool, −2: cool, −3: cold. According to ISO 7730 standard, the values of PMV between −1 and 1 are the range in which 75% people are satisfied, while between −0.5 and 0.5 is the range in which 90% people will be satisfied [4].

Decreasing the energy consumption of heating, ventilation and air conditioning (HVAC) systems is becoming increasingly important due to rising cost of fossil fuels and environmental concerns. Therefore, finding novel ways to reduce energy consumption in buildings without compromising comfort and indoor air quality is an ongoing research challenge. One proven way of achieving energy efficiency in HVAC systems is to design systems that use novel configurations of existing system components. Each HVAC discipline has specific design requirements and each presents opportunities for energy savings. Energy efficient in HVAC systems can be created by re-configuring traditional systems to make more strategic use of existing system parts. Recent research has demonstrated that a combination of existing air conditioning technologies can offer effective solutions for energy conservation and thermal comfort. This paper investigates and reviews the different technologies approaches and demonstrates their ability to improve the performance of HVAC systems in order to reduce energy consumption. For each strategy, a brief description is first presented and then by reviewing the previous studies, the influence of that method on the HVAC energy saving is investigated [5]. Finally, a study focus on heat transfer specially is carried out.

Besides traditional gasoline engine vehicles, in the wave of electric and intelligent vehicles, electric vehicles have developed rapidly. The sales volume of electric vehicles has increased rapidly in the past few years, but the "anxiety about driving range" and "safety anxiety" are still worried by many consumers. The problem of endurance mileage is particularly obvious in low temperatures. When an electric vehicle is driven at low temperatures, the endurance of the whole vehicle may be reduced by more than 40% due to the use of heating and air conditioning and the decline of battery performance; From the perspective of safety, the thermal management of batteries and power appliances in vehicles is also particularly important. These problems need to be solved by a higher performance vehicle thermal management system. In brief, the vehicle thermal management system mainly includes HVAC, battery thermal management, electrical control of motor and thermal management of high-power electrical appliances.

This solution is still typically adopted in EVs because of the lack of engine waste heat to recycle [6]. In particular, positive temperature coefficient (PTC) heaters are widely used by the industry in EV cabin heating in place of the engine coolant heater core of conventional vehicles [7]; however, these systems present limitations related to high costs and energy consumption [8].

However, in EVs climate control systems, heat pumps seem a more sustainable and efficient solution, so that heat pump (HP) systems are currently stepping into EVs market as a reasonable solution. Several studies have been published demonstrating the feasibility of a reversible heat pump system for electric vehicle air conditioning, pointing to Coefficients of Performance (COP) slightly higher than 2 in heating mode and to electric driving range improvement over the use of PTC by 5–10% at low and mid ambient temperatures [9].

The rest of this paper introduces the development environment, modeling methods, the use of modules, the production of 3D animation, inside code part and the simulation core data.

1.1 Development background

1.1.1About MATLAB™

MATLAB is a commercial mathematical software produced by MathWorks, which is used in data analysis, wireless communication, deep learning, image processing and computer vision, signal processing, quantitative finance and risk management, robots, control systems and other fields. MATLAB is the combination of matrix and laboratory, which means matrix factory (matrix laboratory). The software mainly faces the high-tech computing environment of scientific computing, visualization and interactive programming. It integrates many powerful functions such as numerical analysis, matrix calculation, scientific data visualization, and modeling and simulation of nonlinear dynamic systems into an easy-to-use window environment, providing a comprehensive solution for scientific research, engineering design, and many scientific fields where effective numerical calculation is necessary, And to a large extent, it breaks away from the editing mode of traditional non-interactive programming languages (such as C, Fortran).MATLAB, Mathematica and Maple are known as the three major mathematical software. It is second to none in numerical calculation among mathematical science and technology application software. Row matrix operation, drawing functions and data, implementing algorithms, creating user interfaces, connecting programs in other programming languages, etc. The basic data unit of MATLAB is matrix, and its instruction expression is very similar to the form commonly used in mathematics and engineering. Therefore, it is much simpler to solve problems using MATLAB than using C, FORTRAN and other languages to complete the same thing, and MATLAB also absorbs the advantages of software such as Maple, making it a powerful mathematical software. The new version also adds support for C, FORTRAN, C++, and JAVA [10][11].

1.1.2 About Simulink™

Simulink is a visual simulation tool in MATLAB launched by American Mathworks Company. Simulink is a module diagram environment for multi-domain simulation and model-based design. It supports system design, simulation, automatic code generation and continuous testing and verification of embedded systems. Simulink provides a graphical editor, a customizable module library, and a solver to enable dynamic system modeling and simulation. Simulink is integrated with MATLAB, which can integrate the MATLAB algorithm into the model in Simulink and export the simulation results to MATLAB for further analysis. The application fields of Simulink include automobile, aviation, industrial automation, large-scale modeling, complex logic, physical logic, signal processing [12].

1.1.3 About Simscape™

Simscape enables to rapidly create models of physical systems within the Simulink® environment. With Simscape, you build physical component models based on physical connections that directly integrate with block diagrams and other modeling paradigms. You model systems such as electric motors, bridge rectifiers, hydraulic actuators, and refrigeration systems, by assembling fundamental components into a schematic. Simscape add-on products provide more complex components and analysis capabilities. Simscape helps you develop control systems and test system-level performance. You can create custom component models using the MATLAB based Simscape language, which enables text-based authoring of physical modeling components, domains, and libraries. You can parameterize your models using MATLAB variables and expressions, and design control systems for your physical system in Simulink. To deploy your models to other simulation environments, including hardware-in-the-loop (HIL) systems, Simscape supports C-code generation [13].

1.2 Basic Model

1.2.1 Vehicle HVAC System

This example models moist air flow in a vehicle heating, ventilation, and air conditioning (HVAC) system. The vehicle cabin is represented as a volume of moist air exchanging heat with the external environment. The moist air flows through a recirculation flap, a blower, an evaporator, a blend door, and a heater before returning to the cabin. The recirculation flap selects flow intake from the cabin or from the external environment. The blender door diverts flow around the heater to control the temperature.

The model can be simulated in two modes: predefined system inputs or manual system inputs. For predefined system inputs, the control settings for the HVAC system are in the System Inputs variant subsystem. For manual system inputs, the control settings can be adjusted at run time using the dashboard controls.

The heater core and the evaporator are basic implementations of heat exchangers using the e-NTU method. They are custom components based on the Simscape™ Foundation Moist Air library [14].

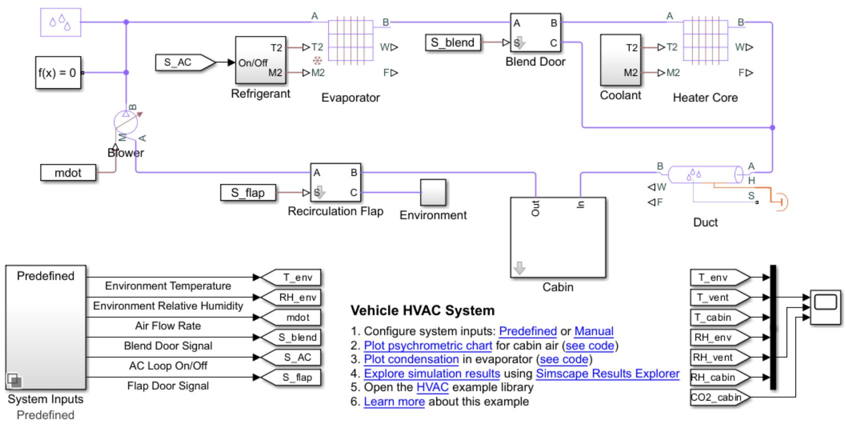

Fig. 1-1 Vehicle HVAC System

1.2.2 Vehicle 3D Dynamic Visualization

The 3d animation shows the benefits of visualization of complex dynamic model in the virtual reality environment. It also shows Simulink® 3D Animation™ 3D off-line animation recording functionality [15].

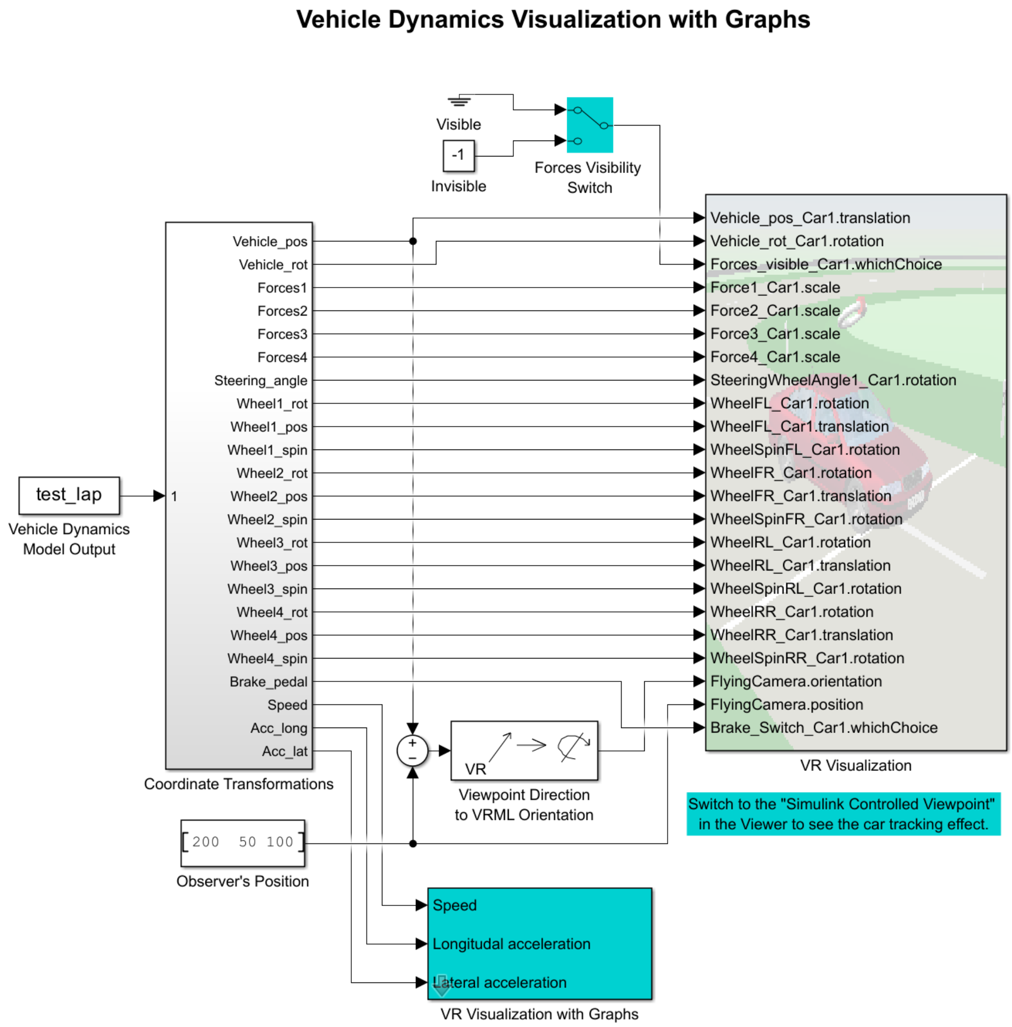

Fig. 1-2 Vehicle dynamic visualization animation block

1.2.3 Convection heat transfer and calculation of heat transfer coefficient

This part first judges the mode of convective heat transfer, and then calculates the convective heat transfer coefficient according to different convective heat transfer modes (laminar flow and turbulence).

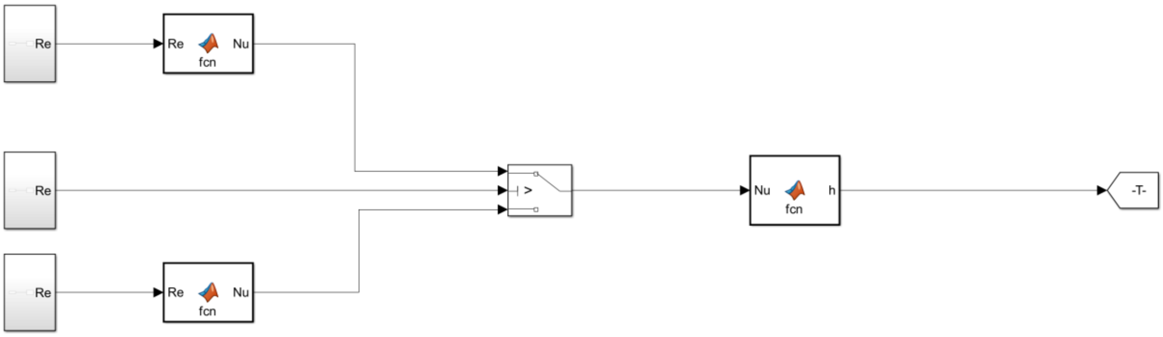

Fig. 1-3 Dynamic heat transfer coefficient calculation block

1.3 Objective of the present thesis

The main objective of this paper is to simulate the change of vehicle speed in the process of driving, to calculate the change of heat transfer rate caused by the difference of heat transfer coefficient of the air flowing on the vehicle surface under different flow conditions, and then affect the temperature regulated by the air conditioning system in the vehicle, and finally save the energy of the vehicle air conditioning system. At the same time, 3d animation is used to simulate the real situation of the car driving on the road, which can more intuitively show the driving state of the vehicle. The air-vehicle convective heat transfer coefficient in the original simulation model is fixed at 20, but the convective heat transfer coefficient of the vehicle cannot be guaranteed to remain unchanged during the high-speed driving of the vehicle. Therefore, by comparing the difference of the convective heat transfer coefficient, the heat exchange of the vehicle during the high-speed driving can be simulated in more detail, so that the power allocation of the HVAC system can be optimized, thus achieving the purpose of saving energy.

For future, the function of this simulation will be more perfect, which will be described in detail later.

1.4 Outline

Chapter 1 briefly introduces the simulation architecture platform, research methods and objectives.

Chapter 2 gives the way of simulating HVAC system and modeling of vehicle air conditioning.

Chapter 3 introduces the structure of simulation in Simulink™, algorithm data and other relevant information in detail then compare the difference between the existing model and the original model.

Chapter 4 simulates various parameters and data of the HVAC simulation model in the vehicle, and visually shows the changes of various data through the line chart then introduces each mathematical function module and the equation parameter variables used in detail

Chapter 5 compares the fixed convection heat transfer coefficient of the original model of the system with the variable convection heat transfer coefficient which is closer to the reality.

Chapter 6 describes the advantages and disadvantages of this simulation model, as well as the aspects that can be improved in the future and add other simulation functions. At the same time, it can also provide thinking direction for the research of doctoral courses.

Chapter2: Overview of HVAC system simulation and comprehensive modeling of vehicle air conditioning

The use of MATLAB, a tool for mathematical programming, is increasing in many fields. Together with its dynamic simulation toolbox Simulink, originally developed for control and automation applications, it has become a powerful tool that is suitable for many applications. In the field of building and HVAC, the number of users of MATLAB/Simulink has also been increasing rapidly in the last years. The tool is suitable for many applications in this field as for example the study of energy consumption, control strategies, hydraulic and air flow studies, IAQ, comfort, sizing problems. More and more studies are being published using MATLAB/Simulink environment for development of specific tools and for simulations of buildings and technical building services. This paper gives a synthesis on the use of MATLAB/Simulink for the improvement of buildings and HVAC systems.

The Heat Balance Method (HBM) is used for estimating the heating and cooling loads encountered in a vehicle cabin. A load estimation model is proposed as a comprehensive standalone model which uses the cabin geometry and material properties as the inputs. The model is implemented in a computer code applicable to arbitrary driving conditions. Using a lumped-body approach for the cabin, the present model can estimate the thermal loads for the simulation period in real-time.

2.1 Modeling approaches for HVAC control

HVAC controllers can be divided into two categories as follows.

Local controllers are low-level controllers that allow HVAC systems to operate properly and to provide adequate services. Local controllers can be further subdivided into two groups:

• Sequencing controllers define the order and conditions associated with switching equipment ON and OFF. Typical sequencing controllers in HVAC systems are chiller sequencing controller, cooling tower sequencing controller, pump sequencing controller, fan sequencing controller, etc.

• Process controllers adjust the control variables to meet the required set point in spite of disturbances and considering the system dynamic characteristics. The typical process controllers used in the HVAC field are P, PI, PID, ON/OFF, step controller, etc.

Supervisory controllers are high level controllers that allow complete consideration of the system level characteristics and interactions among all components and their associated variables. For example, a supervisory controller sets operation modes and sets points for local controllers.

From a modeling point of view, controllers are represented by equations that must be satisfied in every simulation step. The controllers direct the interaction between building and system as well as interactions between components within the system.

The closed-loop local-process control includes a sensor that samples a real-world (measurable) variable. The controller, based on the set point value and measured value, and according to the controller-specific control algorithm, calculates the control signal that feeds the real-world actuator. However, in the simulation tool the user can address variables that cannot be sensed or actuated, as well as apply control algorithms that do not exist. For example, a modeler can directly actuate the heat flux in a model where in reality this could only be done indirectly by changing a valve/damper position.

Furthermore, due to the accessibility of many variables not directly known in the real world, such as the zone cooling/heating load, in simulation the concept of "ideal" (local process) control becomes feasible. An "ideal" local-process controller means that the actuated variable will be adjusted to satisfy the set point requirements for the controlled variable, without specifying the explicit control algorithm and by numerically inverting the (forward) simulation models (from the required output calculate the input needed to satisfy this).

Possibilities to simulate different (advanced) controllers in state-of-the-art BPS tools are limited. Some tools offer pre-defined control strategies (system-based simulation tools), some offer flexibility in specifying only supervisory controllers (EnergyPlus) and some even in specifying local controllers (TRNSYS, ESP-r). The domain-independent environments, such as MATLAB and Dymola, are efficient tools for designing and testing of controllers in a simulation setting but lack the models of all other physical phenomena in buildings.

2.2 Comprehensive modeling of vehicle air conditioning

The Heat Balance Method (HBM) is used for estimating the heating and cooling loads encountered in a vehicle cabin. A load estimation model is proposed as a comprehensive standalone model which uses the cabin geometry and material properties as the inputs. The model is implemented in a computer code applicable to arbitrary driving conditions. Using a lumped-body approach for the cabin, the present model can estimate the thermal loads for the simulation period in real-time. Typical materials and a simplified geometry of a specific hybrid electric vehicle are considered for parametric studies. Two different driving and ambient conditions are simulated to find the contribution and importance of each of the thermal load categories. The Supplemental Federal Test Procedure (SFTP) standard driving cycle is implemented in the simulations for two North American cities and the results are compared. It is concluded that a predictive algorithm can be devised according to the driving conditions, vehicle speed, orientation, and geographical location. By using this model, the pattern of upcoming changes in the comfort level can be predicted in real-time in order to intelligently reduce the overall AC power consumption while maintaining driver thermal comfort.

2.2.1 Model development

The net heat gain by the cabin can be classified under nine different categories. The total load as well as each of these loads can either be positive (heating up the cabin) or negative (cooling down the cabin) and may depend on various driving parameters. In the following, the models developed for each of these load categories are presented and discussed. Some of the correlations used in the present model are based on experiments performed on certain vehicles, which are used here for general validation of the model. New correlations can be readily plugged into the present model can be tailored to any new vehicle, after specifying those correlations for the case. The summation of all the load types will be the instantaneous cabin overall heat load gain. The mathematical formulation of the model can be summarized as below.

$$\dot{Q}_{Tot} = \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} + \dot{Q}_{AC}$$

All the above $\dot{Q}$ values are thermal energies per unit time. $\dot{Q}_{Tot}$ is the net overall thermal load encountered by the cabin. $\dot{Q}_{Met}$ is the metabolic load. $\dot{Q}_{Dir}$, $\dot{Q}_{Dif}$ and $\dot{Q}_{Ref}$ are the direct, diffuse, and reflected radiation loads, respectively. $\dot{Q}_{Amb}$ is the ambient load. $\dot{Q}_{Exh}$ and $\dot{Q}_{Eng}$ are the exhaust and engine load due to the high temperature of the exhaust gases and the engine. Finally, the term $\dot{Q}_{Ven}$ is the load generated due to ventilation, and $\dot{Q}_{AC}$ is the thermal load created by the AC cycle.

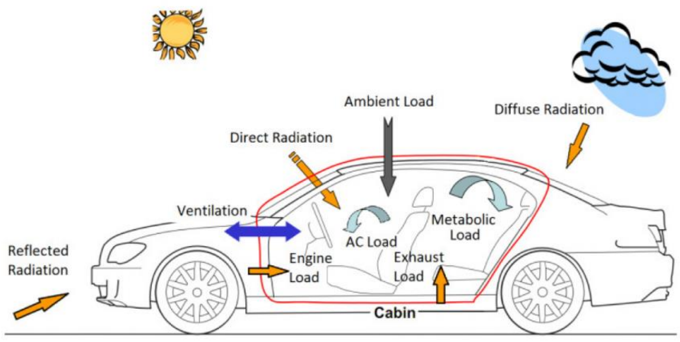

Fig. 2-1 Vehicle heat load balance

2.2.2 Load parts

Metabolic Load

The metabolic activities inside human body constantly create heat and humidity (i.e., perspiration). This heat passes through the body tissues and is finally released to the cabin air. This amount is considered as a heat gain by the cabin air and is called the metabolic load. The metabolic load can be calculated by

$$\dot{Q}_{Met} = \sum_{Passengers} M A_{Du} \qquad \text{(Equation 2-1)}$$

where M is the passenger metabolic heat production rate. It is found from the tabulated values in ISO 8996 [16] based on various criteria such as occupation and activity levels. For a driver and a sitting passenger, the values can be estimated as 85 W/m^2 and 55 W/m^2 , respectively. The DuBois area A_{Du}, which is an estimation of the body surface area as a function of height and weight, is calculated by

$$A_{Du} = 0.202 W^{0.425} H^{0.725} \qquad \text{(Equation 2-2)}$$

where W and H are the passenger weight and height, respectively.

Radiation Load

$$\dot{Q}_{s,Rad} = S\alpha(\dot{I}_{Dir} \cos\theta + \dot{I}_{Dif} + \dot{I}_{Ref})$$

S is the surface area

α is the surface absorptivity

$\dot{I}_{Dir}$ is the direct radiation heat gain per unit area

θ is the angle between the surface normal and the position of sun in the sky

$\dot{I}_{Dif}$ is the diffuse radiation heat gain per unit area

$\dot{I}_{Ref}$ is the reflected radiation heat gain per unit area

Ambient Load

The ambient load is the contribution of the thermal load transferred to the cabin air due to temperature difference between the ambient and cabin air. Exterior convection, conduction through body panels, and interior convection are involved in the total heat transfer between the ambient and the cabin. This equation shows the general form of the ambient load model.

$$\dot{Q}_{Amb} = \sum_{Surfaces} SU(T_s - T_i) \quad \text{(Equation 2-3)}$$

where U is the overall heat transfer coefficient of the surface element. T_s and T_i are the average surface temperature and average cabin temperature, respectively. U has different components consisting of the inside convection, conduction through the surface, and outside convection. It can be written in the form

$$U = \frac{1}{R} \text{ where } R = \frac{1}{h_o} + \frac{\lambda}{k} + \frac{1}{h_i} \quad \text{(Equation 2-4)}$$

where R is the net thermal resistance for a unit surface area. ho and hi are the outside and inside convection coefficients, k is the surface thermal conductivity, and λ is the thickness of the surface element. The thermal conductivity and thickness of the vehicle surface can be measured rather easily. The convection coefficients ho and hi depend on the orientation of the surface and the air velocity. Here, the following estimation is used to estimate the convection heat transfer coefficients as a function of vehicle speed [17]

$$h = 0.6 + 6.64\sqrt{V} \quad \text{(Equation 2-5)}$$

where h is the convection heat transfer coefficient in W/m^2K and V is the vehicle speed in *m/s*. Despite its simplicity, this correlation is applicable in all practical automotive instances. The cabin air is assumed stationary, and the ambient air velocity is considered equal to the vehicle velocity. Numerical simulations can also be used to provide the model with convection coefficients that have higher accuracy and consider the different orientation and position of every surface component. Like the radiation load above, a portion of the ambient load across the body surface is absorbed by the body plate material. The heat gain or loss of each surface element is the difference between the heat gained from the ambient by the surface, and the heat released to the cabin by the surface. Thus, we can write the net absorbed heat as

$$\dot{Q}_{s,Amb} = SU(T_o - T_s) - SU(T_s - T_i) = SU(T_o - 2T_s + T_i) \quad \text{(Equation 2-6)}$$

where T_o , T_i , and T_s is the ambient, cabin, and surface average temperatures, respectively.

Exhaust Load

Conventional and hybrid electric vehicles have an Internal Combustion Engine (ICE) that creates exhaust gases. The Exhaust Gas Temperature (EGT) can reach as high as 1000°C [18]. Because of the high temperature of the exhaust gas, some of its heat can be transferred to the cabin through the cabin floor. Considering S_{Exh} as the area of the bottom surface in contact with the exhaust pipe, the exhaust heat load entering the cabin can be written as

$$\dot{Q}_{Exh} = S_{Exh}U(T_{Exh} - T_i) \quad \text{(Equation 2-7)}$$

where U is the overall heat transfer coefficient of the surface element in contact with the exhaust pipe and it should be calculated by Equation assuming no external convection since the exhaust temperature is measured at the outer side of the bottom surface. S_{Exh} is the surface area exposed to the exhaust pipe temperature and T_{Exh} is the exhaust gas temperature. The temperature of the exhaust gases in Celsius degrees is estimated by

$$T_{Exh} = 0.138RPM - 17 \quad \text{(Equation 2-8)}$$

where RPM is the engine speed in revolutions per minute.

Engine Load

Like the exhaust load above, the high temperature engine of a conventional or hybrid car can also contribute to the thermal gain of the cabin. Equation shows the formulation used for calculating the engine thermal load.

$$\dot{Q}_{Eng} = S_{Eng}U\left(T_{Eng} - T_i\right) \qquad \text{(Equation 2-9)}$$

where U is the overall heat transfer coefficient of the surface element in contact with the engine and S_{Eng} is the surface area exposed to the engine temperature. The overall heat transfer coefficient can be calculated by h above assuming no external convection, since the engine temperature is measured at the outside of the front surface. T_{Eng} is the engine temperature and is estimated in Celsius degrees by

$$T_{Eng} = -2 \times 10^{-6}RPM^2 + 0.0355RPM + 77.5 \qquad \text{(Equation 2-20)}$$

Ventilation Load

Fresh air is allowed to enter the vehicle cabin to maintain the air quality for passengers. As the passengers breathe, the amount of CO2 concentration linearly increases over time. Thus, a minimum flow of fresh air should be supplied into the cabin to maintain the passenger's comfort. Arndt and Sauer reported the minimum fresh air requirements for different numbers of passengers in a typical vehicle. For instance, a minimum of 13% fresh air is needed for a single passenger. On the other hand, showing that for typical vehicles, leakage occurs as a function of the pressure difference between the cabin and the surroundings as well as the vehicle velocity. For a small sedan car at a pressure difference of 10 Pa, a leakage of 0.02 m^3/s was reported. Because of the air conditioning and ventilation, the cabin pressure is normally slightly higher than the ambient. Thus, the ventilation load must take the leakage air flow rate into account. Meanwhile, in the steady-state operation, the built-up pressure is assumed to remain constant. Hence, ambient air is assumed to enter the cabin at the ambient temperature and relative humidity, and the same flow rate is assumed to leave the cabin at the cabin temperature and relative humidity. According to psychrometric calculations, ventilation heat gain consists of both sensible and latent loads. To account for both these terms, assuming a known flow rate of fresh air entering the cabin, the amount of thermal heat gain can be calculated from

$\dot{Q}_{Ven} = \dot{m}_{Ven}(e_o - e_i)$ (Equation 2-31)

where $\dot{m}_{Ven}$ is the ventilation mass flow rate and e_o and e_i are the ambient and cabin enthalpies, respectively. Enthalpies are calculated from

$e = 1006T + (2.501 \times 10^6 + 1770T)X$ (Equation 2-42)

where T is air temperature and X is humidity ratio in gram of water per gram of dry air. Humidity ratio is calculated as a function of relative humidity by

$X = 0.62198 \frac{\Phi P_s}{100P - \Phi P_s}$ (Equation 2-53)

where ϕ is relative humidity, P is air pressure, and P_s is the water saturation pressure at temperature T.

AC Load

The duty of the air conditioning system is to compensate for other thermal loads so that the cabin temperature remains within the acceptable comfort range. In cold weather conditions, positive AC load (heating) is required for the cabin. Inversely, in warm conditions, negative AC load (cooling) is needed for maintaining the comfort conditions. The actual load created by the AC system depends on the system parameters and working conditions. In this work, it is assumed that an AC (or heat pump) cycle is providing the thermal load calculated by

$\dot{Q}_{AC} = -\begin{pmatrix} \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \\ \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} \end{pmatrix} - \frac{(m_a c_a + DTM)(T_i - T_{comf})}{t_c}$ (Equation 2-64)

T_{comf} is the target comfort temperature as described and widely used by ASHARE standards. This is the target bulk cabin temperature which is assumed comfortable at the conditions under consideration. t_c is a pull-down constant which determines the overall pull-down time. Pull-down time is defined as the time required for the cabin temperature to reach the comfort temperature within 1 K. Using above for the air conditioning load, the pull-down constant can be calculated from

$t_c = \frac{t_p}{\ln|T_0 - T_{comf}|}$ (Equation 2-75)

where T_0 is the initial cabin temperature. Of course, the actual AC load depends on the system sizing and design. For a given system, the load may change depending on the compressor and fan speed as well. The actual power consumption of the cycle should be estimated by considering a suitable Coefficient of Performance (COP) for the vapor compression cycle. Equation above is used in this study as a guideline for analyzing the performance of an AC system in a typical vehicle. It shows that analyzing different scenarios with the AC cycle can help efficient sizing and control of the air conditioning cycle [19].

Chapter3: Simulation Model Construction

In this chapter, the method of building the Simulink simulation model, the equations used, and various function parameters are discussed, and the programming code is constructed.

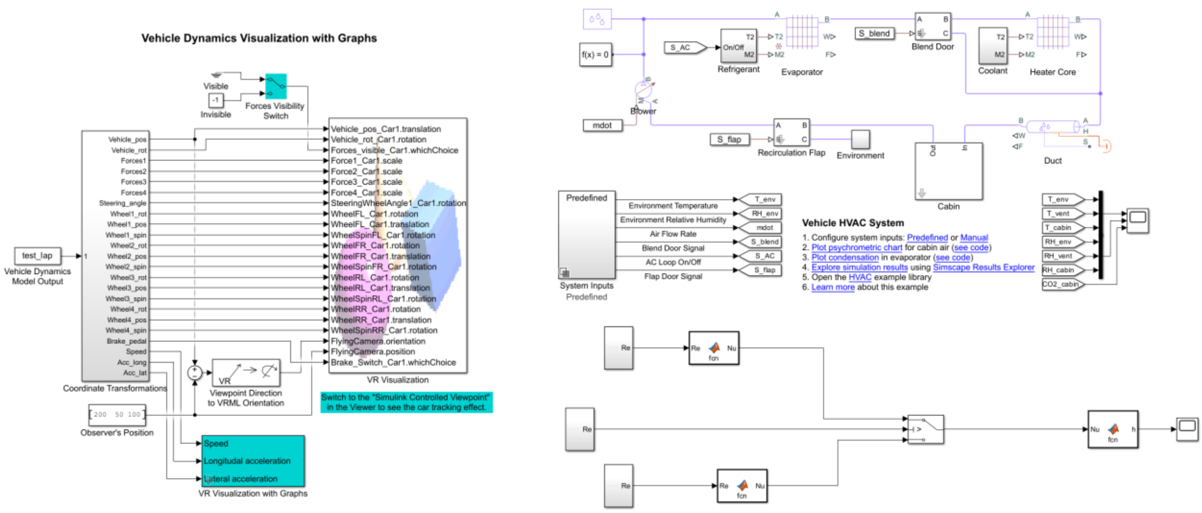

Fig. 3-1 Integrated simulation system

3.1 Vehicle HVAC system

This example models moist air flow in a vehicle heating, ventilation, and air conditioning (HVAC) system. The vehicle cabin is represented as a volume of moist air exchanging heat with the external environment. The moist air flows through a recirculation flap, a blower, an evaporator, a blend door, and a heater before returning to the cabin. The recirculation flap selects flow intake from the cabin or from the external environment. The blender door diverts flow around the heater to control the temperature.

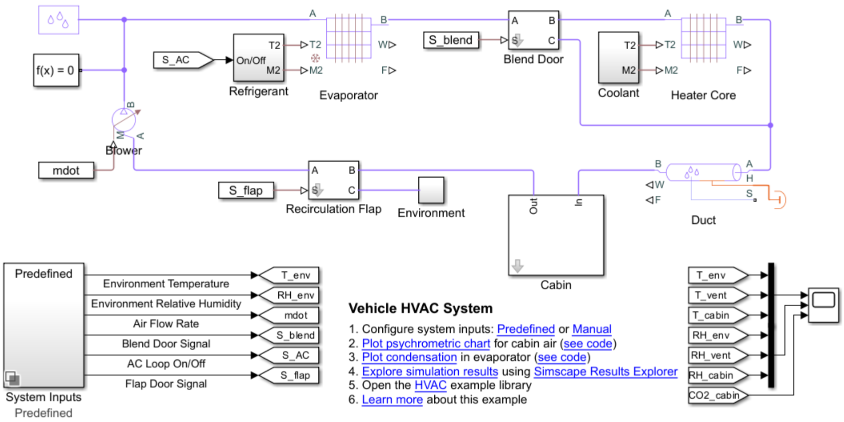

Fig. 3-2 Vehicle HVAC system

The model can be simulated in two modes: predefined system inputs or manual system inputs. For predefined system inputs, the control settings for the HVAC system are in the System Inputs variant subsystem. For manual system inputs, the control settings can be adjusted at run time using the dashboard controls.

The heater core and the evaporator are basic implementations of heat exchangers using the e-NTU method. They are custom components based on the Simscape™ Foundation Moist Air library.

3.1.1 Cabin System

The cabin receives signals such as relative humidity, temperature, carbon dioxide concentration in cabin and ambient temperature, ambient atmospheric pressure, ambient relative humidity, and ambient carbon dioxide concentration.

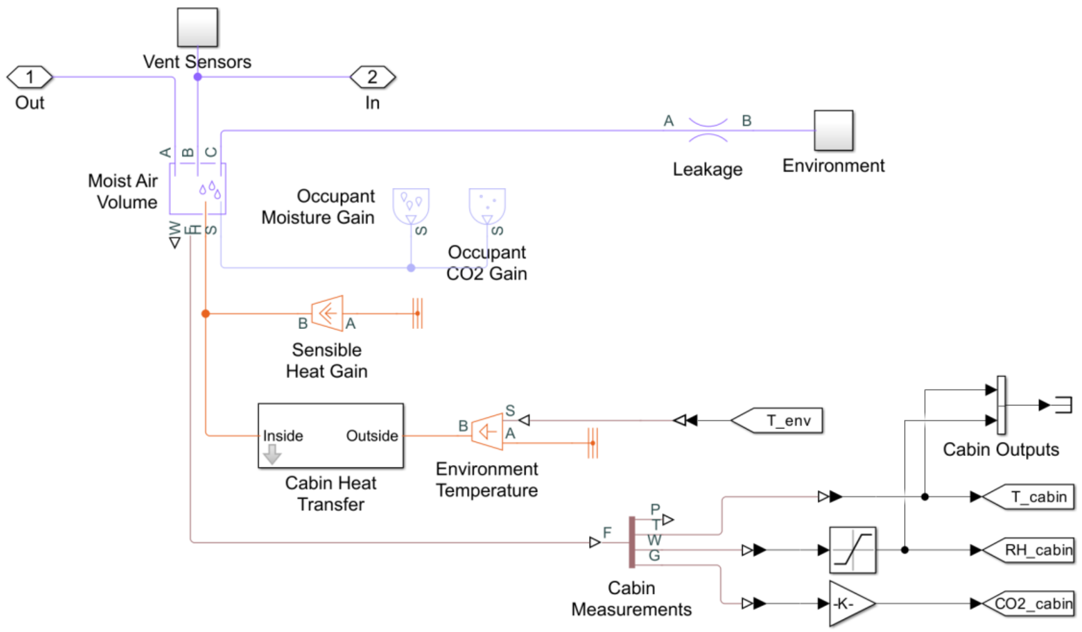

Fig. 3-3 Cabin system

3.1.2 Cabin Heat Transfer System

The heat transfer in the cabin consists of convection heat transfer and conduction heat transfer, ignoring the radiation heat transfer of three basic modes of heat transfer. In order to simplify the simulation model, the radiant heat is integrated into the ambient temperature. The cabin consists of three parts, the roof, the door and the glass. The cabin heat transfer system is the focus of this simulation, especially in the external environment and the ambient load mentioned above, the air flow outside the cabin and the convection heat transfer on the vehicle surface during the gradual increase of the vehicle speed will bring huge heat transfer in a short time. (There are two types of heat transfer modules and the cabin parts that interactive with the flowing air are doors glasses and roof.)

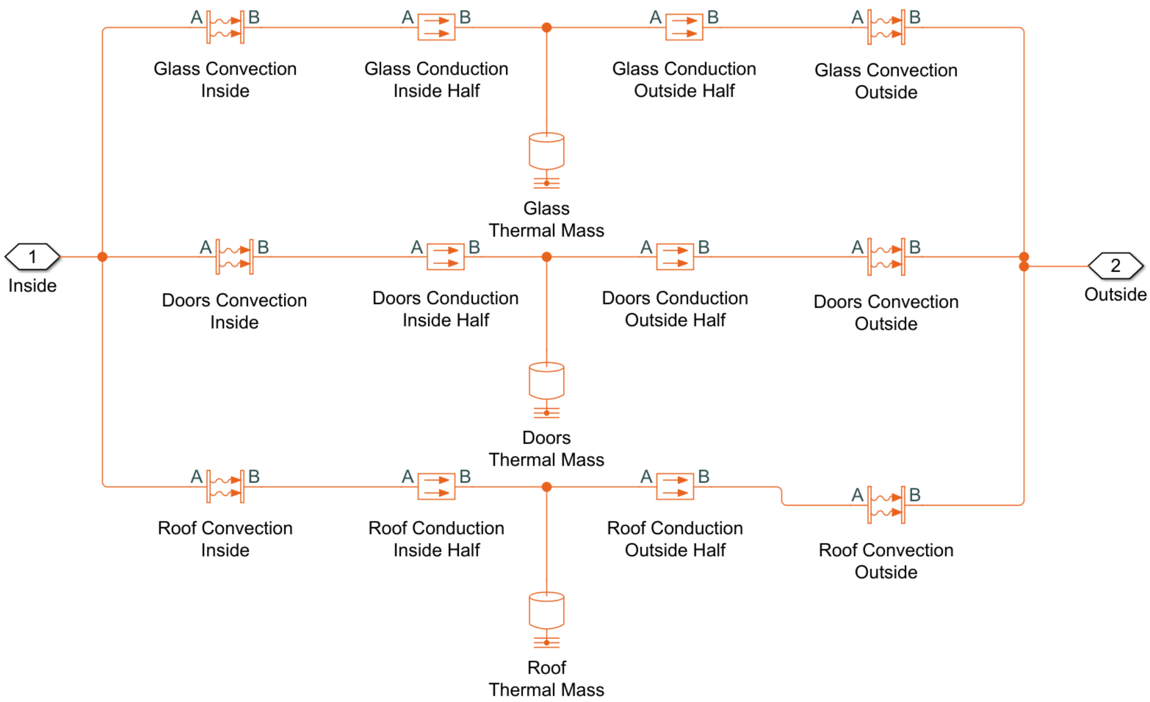

Fig. 3-4 Cabin heat transfer system

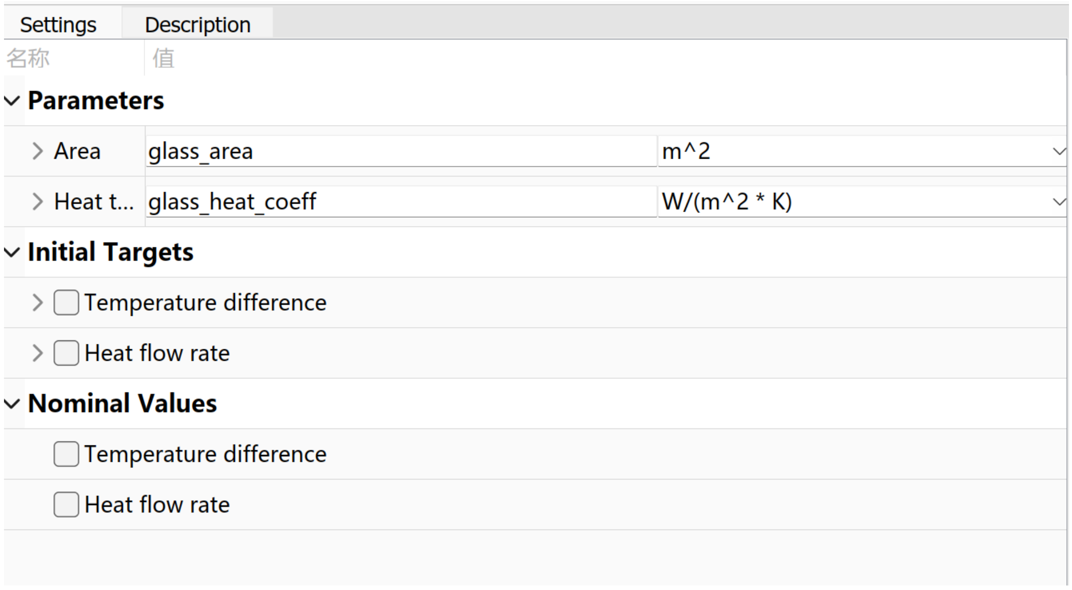

Fig. 3-5 Raw convective heat transfer part

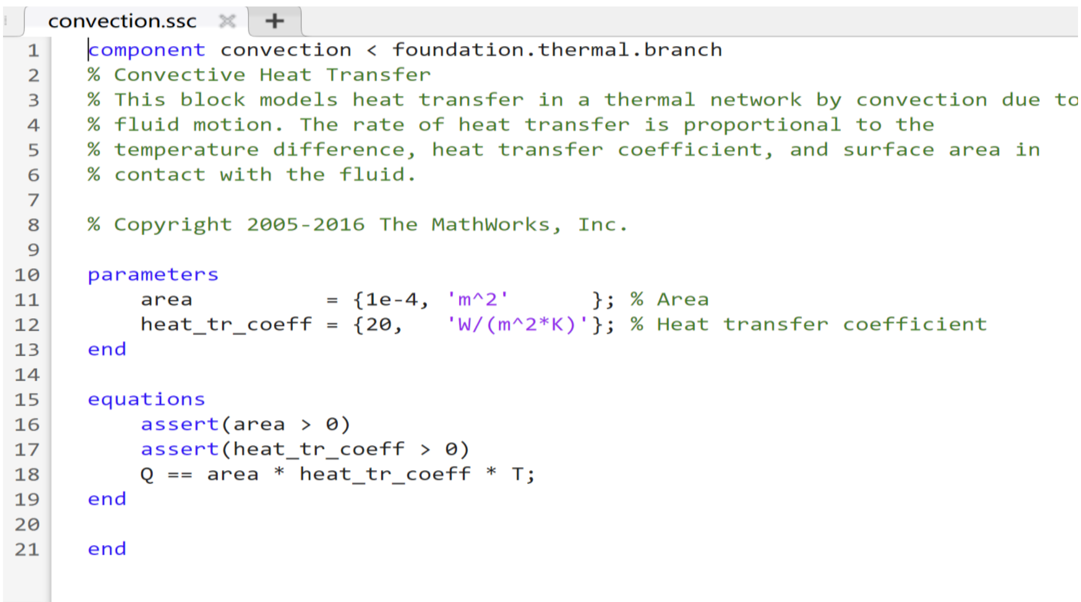

Fig. 3-6 Code in raw convective transfer block

component convection < foundation.thermal.branch

% Convective Heat Transfer

% This block models heat transfer in a thermal network by convection due to

```
% fluid motion. The rate of heat transfer is proportional to the
% temperature difference, heat transfer coefficient, and surface area in
% contact with the fluid.
% Copyright 2005-2016 The MathWorks, Inc.
parameters
    area          = {1e-4, 'm^2'      }; % Area
    heat_tr_coeff = {20,   'W/(m^2*K)'}; % Heat transfer coefficient
end
equations
    assert(area > 0)
    assert(heat_tr_coeff > 0)
    Q == area * heat_tr_coeff * T;
end
end
```

As shown in the figure and the code above, original cabin heat transfer model, in which the convective heat transfer coefficient is a predefined constant number 20. However, due to the increase of vehicle speed, the air flow outside the vehicle and the cabin convective heat transfer coefficient will change dramatically. To this end, improvement is necessary. Introduction of variable convective heat transfer coefficient for the original module and expand it in detail below.

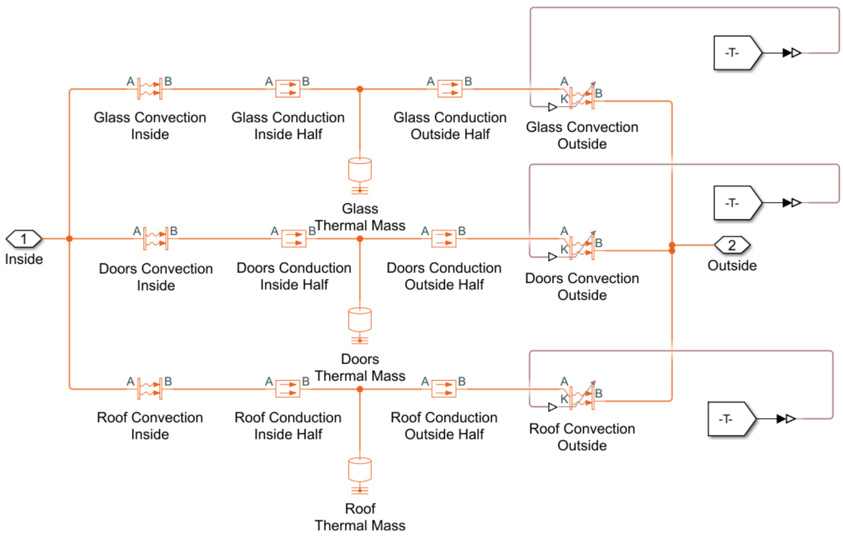

Fig. 3-7 Cabin heat transfer system with dynamic *h*

To calculate convective heat transfer coefficient *h*, an input signal is required. Then the MATLAB function module is used for the calculation of signal and the heat transfer coefficient *h*.

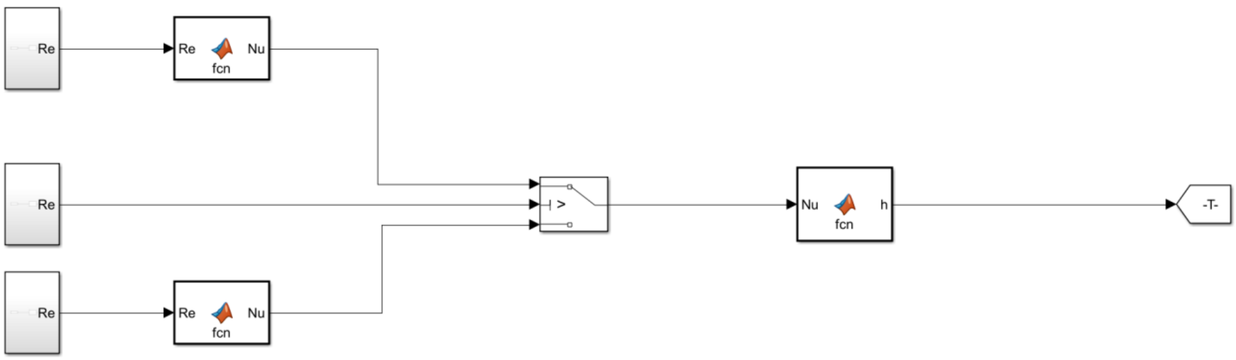

Fig. 3-8 Convective heat transfer coefficient calculation block

The module called Re is a subsystem to calculate Reynolds number. Reynolds number is a dimensionless number that can be used to characterize fluid flow. The function to calculate Reynold number is

$$Re = \frac{\rho v d}{\mu} \qquad \text{(Equation 3-1)}$$

where v、 ρ、 μ is the flow velocity, density and viscosity coefficient of the fluid respectively, and d is a characteristic length.

For more precise calculation will be discuss in Chapter 4.

The modified convective heat transfer module is

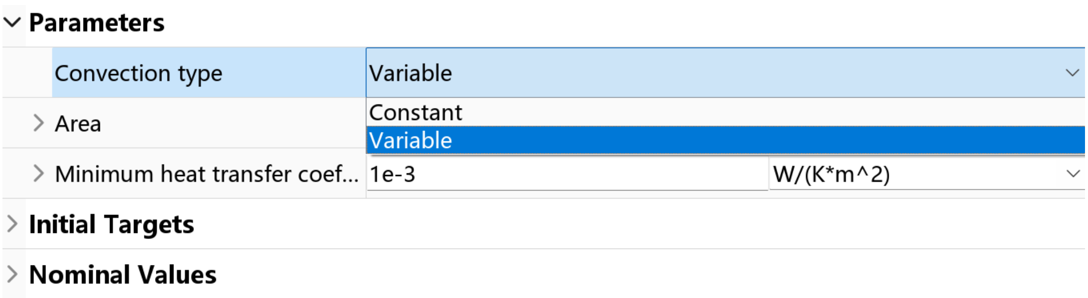

Fig. 3-9 Variable convective heat transfer coefficient block

The parameter of convective heat transfer h is variable. This block models heat transfer in a thermal network by convection due to fluid motion. The rate of heat transfer is proportional to the temperature difference, heat transfer coefficient, and surface area in contact with the fluid. The heat transfer coefficient can be either constant or variable. When the heat transfer coefficient is variable, specify it using the physical input signal at port K. If the heat transfer coefficient is variable, it remains at the Minimum heat transfer coefficient if the physical signal falls below this value.

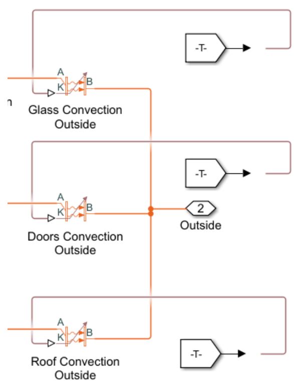

Fig. 3-10 Variable convective heat transfer coefficient signal transaction

Now the three blocks receive signal from calculated data. And the code of convective heat transfer block is

```
inputs (ExternalAccess = none)
    var_heat_tr_coeff = {20,'W/(m^2*K)'}; %K:left
end

parameters
    thermal_type = foundation.enum.constant_variable.constant; % Convection type
    %                         1 - Constant
    %                         2 - Variable

    area         = {1e-4, 'm^2'}; % Area
end
parameters (Access = public, ExternalAccess = none)
    heat_tr_coeff = {20,   'W/(m^2*K)'}; % Heat transfer coefficient
    min_heat_tr_coeff   = {1e-3, 'W/(m^2*K)'}; % Minimum heat transfer coefficient
end

% Parameter and visibility checks
if thermal_type == foundation.enum.constant_variable.constant
    annotations
        heat_tr_coeff : ExternalAccess = modify
    end
    parameters(ExternalAccess = none)
        heat_tr_coeff_ = heat_tr_coeff; % Heat transfer coefficient
```

```
    end
    equations
        assert(heat_tr_coeff > 0)
    end
else %if thermal_type == foundation.enum.constant_variable.variable
    annotations
        var_heat_tr_coeff : ExternalAccess = modify
        min_heat_tr_coeff : ExternalAccess = modify
        Icon = 'convection_variable.svg';
    end
    intermediates(ExternalAccess = none)
        heat_tr_coeff_ = if var_heat_tr_coeff >= min_heat_tr_coeff, var_heat_tr_coeff
else min_heat_tr_coeff end; % Heat transfer coefficient
    end
    equations
        assert(min_heat_tr_coeff > 0)
    end
end
equations
    assert(area > 0)
    assert(heat_tr_coeff_ > 0)
    Q == area * heat_tr_coeff_ * T;
end
end
```

3.1.3 Heat Transfer Coefficient Function System

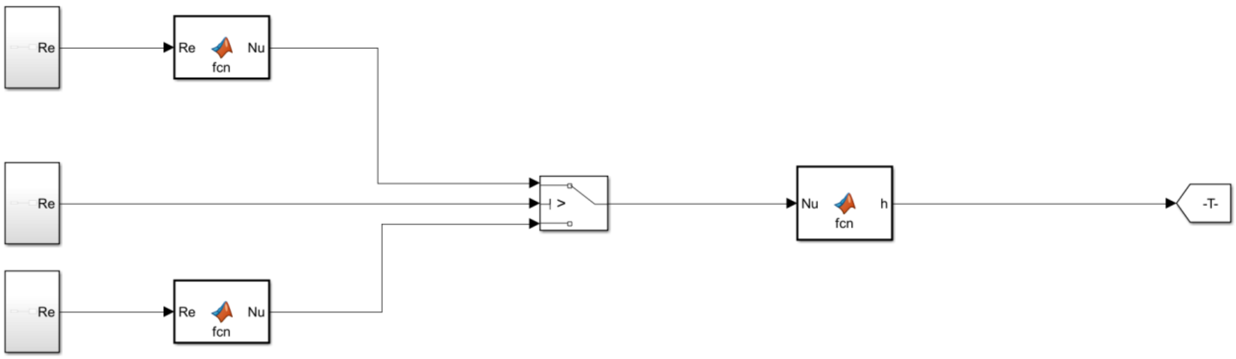

Fig. 3-11 Variable convective heat transfer coefficient calculation

Reynolds number calculation

Re block is a subsystem to calculate Reynolds number to judge the flow state of the fluid from the raw velocity data.

When the Reynolds number is small, the influence of viscous force on the flow field is greater than that of inertia. The disturbance of flow velocity in the flow field will be attenuated by viscous force, and the fluid flow is stable and laminar; On the contrary, if the Reynolds number is large, the impact of inertial convection field is greater than the viscous force, the fluid flow is relatively unstable, and the small change of velocity is easy to develop and strengthen, forming a turbulent flow field with disorder and irregularity.

$Re = 5 \times 10^5$ is a very important number. Initially, the boundary-layer development is 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 becomes turbulent. The turbulent-flow region may be pictured as a random churning action with chunks of fluid moving to and fro in all directions.

The transition from laminar to turbulent flow occurs when

$$\frac{u_\infty x}{v} = \frac{\rho u_\infty x}{\mu} > 5 \times 10^5 \qquad \text{(Equation 3-2)}$$

where u_∞ = free-stream velocity, *m/s*

x = distance from leading edge, m

$v = \frac{\mu}{\rho}$ = kinematic viscosity, m^2/s

This grouping of terms is called the Reynolds number, and is dimensionless if a consistent set of units is used for all the properties:

$$Re_x = \frac{u_\infty x}{v} \qquad \text{(Equation 3-3)}$$

Although the critical Reynolds number for transition on a flat plate is usually taken as 5×10^5 for most analytical purposes, the critical value in a practical situation is strongly dependent on the surface-roughness conditions and the "turbulence level" of the free stream. The normal range for the beginning of transition is between $5 \times 10^5$and 10^6. With very large disturbances present in the flow, transition may begin with Reynolds numbers as low as 5×10^5 , and for flows that are very free from fluctuations, it may not start until $Re = 5 \times 10^6$ or more. The transition process is one that covers a range of Reynolds numbers, with transition being complete and with developed turbulent flow usually observed at Reynolds numbers twice the value at which transition began.

From the input data of the whole model, the max velocity of the vehicle is no more than 40 *m/s*.

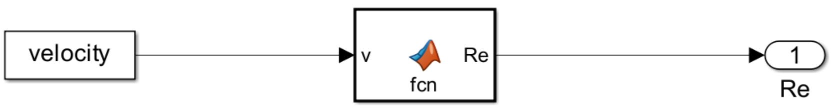

Fig. 3-12 Reynolds number calculation

Velocity is a series of data in data set and from vehicle driving in Simulink 3d animation. The array data is composed of 112 seconds real-time speed from driving to stopping, accurate to microseconds.

Users define function module is also used to calculate Reynolds number. The code is

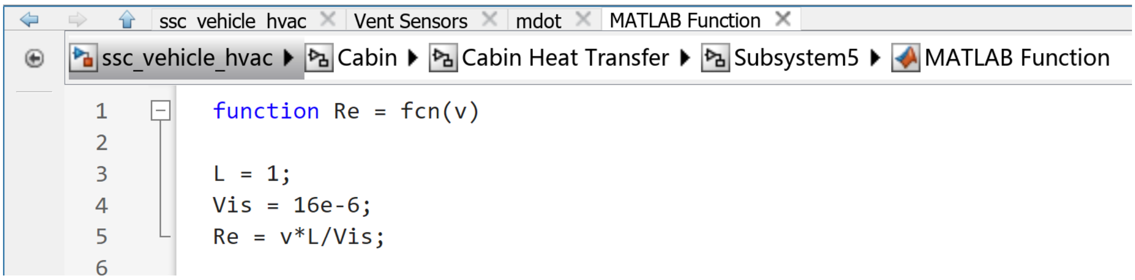

Fig. 3-13 Calculation function of Reynolds number

```
function Re = fcn(v)


L = 1;

Vis = 16e-6;

Re = v*L/Vis;
```

Where in the code, L is surface area of the vehicle is simulated as 1 m^2

Vis is viscosity, at 30 ℃ ambient, the value is $16 \times 10^{-6}\ m^2/s$

And v is a parameter indicates current velocity of the vehicle

Judge the air flow state

After calculating Re in current velocity, air flow state judgment is necessary. There are two air flow state in this simulation: laminar and laminar-turbulent average.

Pr (Prandtl Number) is a dimensionless scalar in fluid mechanics. It reflects the interaction between energy and momentum transfer processes in fluid and plays an important role in thermal calculation. In this case, the ambient temperature is 30 ℃ and Pr is simulated as a constant number 0.7. Normally, the equation of Pr is

$$Pr = \frac{c_\rho \mu}{\lambda} \qquad \text{(Equation 3-4)}$$

Where μ : Dynamic viscosity coefficient (unit: $Pa * s$)

λ : Thermal conductivity (unit: *W/(m * K)*)

c_ρ: specific heat at constant pressure (unit: *J/ (kg * K)*)

In the heat transfer of fluid boundary (surface), Nusselt number (Nu) is the ratio of convective heat and conduction heat across the boundary. In this case, convection includes advection and diffusion. Named after Wilhelm Nusselt, this is a dimensionless quantity. The conduction heat value is measured under the same conditions as the thermal convection value, but there is stagnant fluid during conduction [20].

The Nusselt number is close to 1, that is, the convective heat and the conduction heat are similar, which is the characteristic of "mass flow" or laminar flow. A large Nusselt number indicates that convection is more active, and the Nusselt number of turbulences is usually in the range of 100-1000 [21].

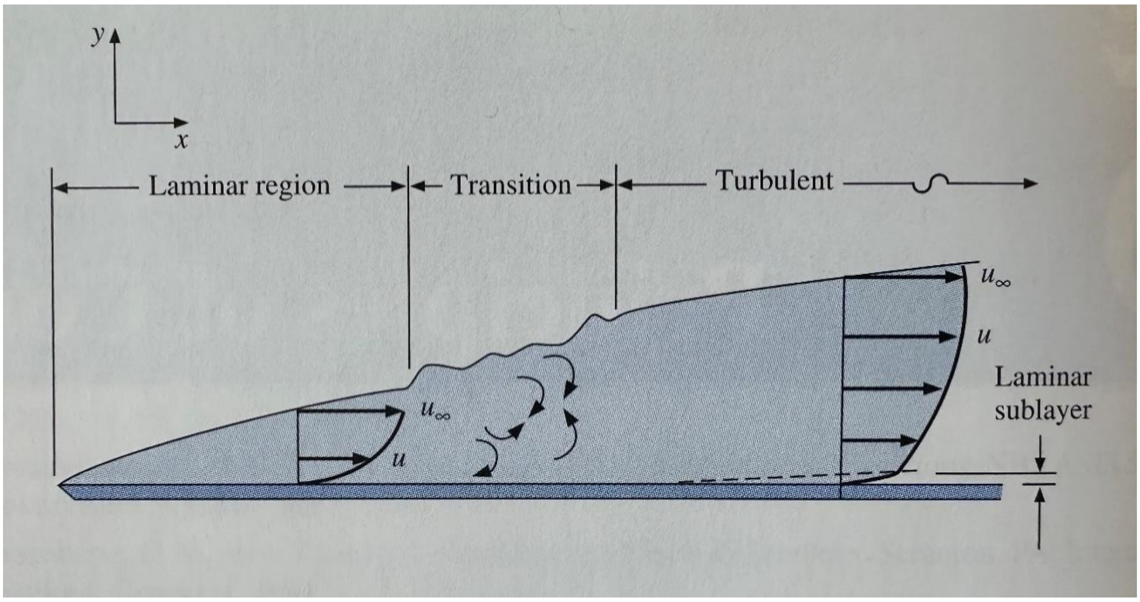

Fig. 3-14 Laminar velocity profile on a flat plate

Table 3-1 Flow regime equation

Flow regime	Restrictions	Equation
Laminar, local	$T_\omega = \text{const}, Re_x < 5 \times 10^5$, $0.6 < Pr < 50$	$Nu_x = 0.332 Pr^{\frac{1}{3}} Re_x^{\frac{1}{2}}$
Laminar-turbulent, average	$T_\omega = \text{const}, Re_x < 10^7$, $Re_{crit} = 5 \times 10^5$	$\overline{Nu_L} = Pr^{\frac{1}{3}}(0.037 Re_L^{0.8} - 871)$

From the table above, when $Re_x < 5 \times 10^5$ the air flow state is laminar, and the function is

$$Nu_x = 0.332 Pr^{\frac{1}{3}} Re_x^{\frac{1}{2}} \quad \text{(Equation 3-5)}$$

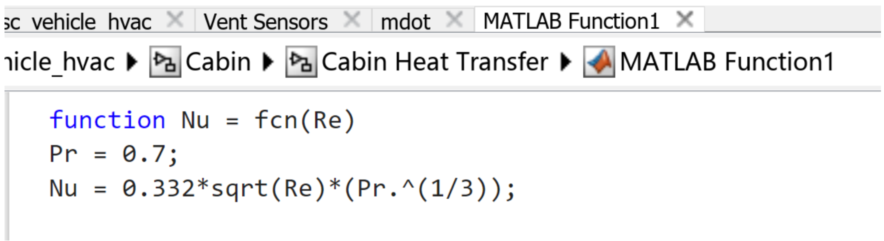

Fig. 3-15 Calculation function of Nusselt number in laminar

The code is

```
function Nu = fcn(Re)
Pr = 0.7;
Nu = 0.332*sqrt(Re)*(Pr.^(1/3));
```

When $5 \times 10^5 < Re_x < 10^7$ the air flow state is laminar-turbulent average,

and the function is

$$\overline{Nu}_L = Pr^{\frac{1}{3}}(0.037 Re_L^{0.8} - 871) \quad \text{(Equation 3-6)}$$

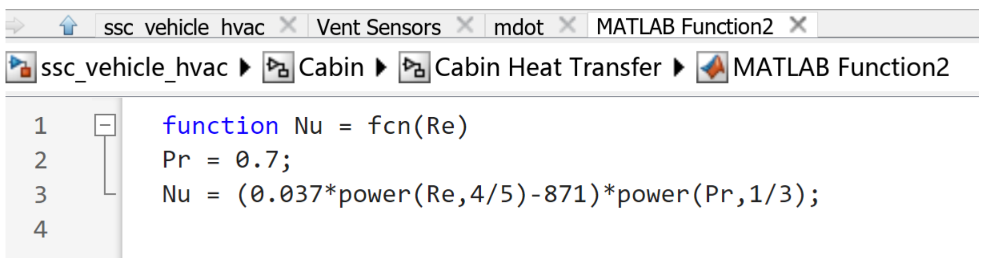

Fig. 3-16 Calculation function of Nusselt number in laminar-turbulent, average

The code is

```
function Nu = fcn(Re)

Pr = 0.7;

Nu = (0.037*power(Re,4/5)-871)*power(Pr,1/3);
```

The next step in this simulation model is a switch block. The Switch block passes through the first input, or the third input signal based on the value of the second input. The first and third inputs are data input. The second input is a control input. Specify the condition under which the block passes the first input by using the criteria for passing first input and threshold parameters.

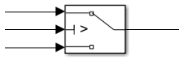

Fig. 3-17 Switch block

This block is used to judge which signal will get through. The threshold is 5×10^5.

When $Re_x < 5 \times 10^5$ *Nu* will be calculated as laminar.

When $5 \times 10^5 < Re_x < 10^7$ *Nu* will be calculated as laminar-turbulent average.

Then convective heat transfer coefficient will be calculated from *Nu*.

```
function h =fcn(Nu)


L = 1;
x = 2.67e-2;

h = Nu*x/L;
```

Fig. 3-18 Calculation function of convective heat transfer coefficient from Nusselt number

```
function h =fcn(Nu)

L = 1;
x = 2.67e-2;

h = Nu*x/L;
```

Where L is the air flow over surface area of the vehicle is simulated as 1 m^2

X is the air heat transfer coefficient, when ambient temperature is 30 °C, the value is 2.67×10^{-2} (unit: $W/(m * K)$)

After calculating the convective heat transfer coefficient h from Nusselt number, the signal is redirected to the three convection blocks (roof, glass and door).

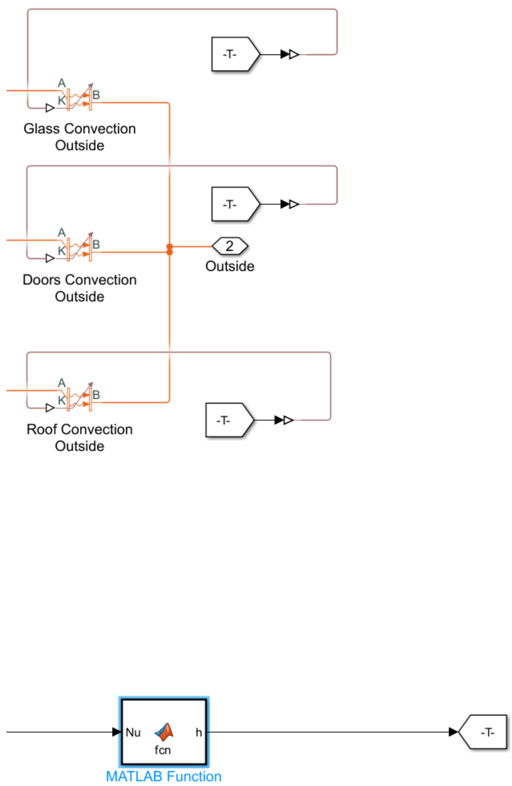

Fig. 3-19 Convective heat transfer coefficient signal redirection

The convection block now is

Fig. 3-20 Variable convective heat transfer coefficient block

The rate of heat transfer is proportional to the temperature difference, heat transfer coefficient, and surface area in contact with the fluid. The heat transfer coefficient can be either constant or variable. When the heat transfer coefficient is variable, specify it using the physical input signal at port K. If the heat transfer coefficient is variable, it remains at the Minimum heat transfer coefficient if the physical signal falls below this value.

```
inputs (ExternalAccess = none)
    var_heat_tr_coeff = {20,'W/(m^2*K)'}; %K:left
end

parameters
    thermal_type = foundation.enum.constant_variable.constant; % Convection type
    %                                1 - Constant
    %                                2 - Variable

    area          = {1e-4, 'm^2'}; % Area
end

parameters (Access = public, ExternalAccess = none)
    heat_tr_coeff = {20,   'W/(m^2*K)'}; % Heat transfer coefficient
    min_heat_tr_coeff   = {1e-3, 'W/(m^2*K)'}; % Minimum heat transfer coefficient
end

% Parameter and visibility checks
if thermal_type == foundation.enum.constant_variable.constant
    annotations
        heat_tr_coeff : ExternalAccess = modify
    end
    parameters(ExternalAccess = none)
        heat_tr_coeff_ = heat_tr_coeff; % Heat transfer coefficient
    end
    equations
        assert(heat_tr_coeff > 0)
    end
else %if thermal_type == foundation.enum.constant_variable.variable
    annotations
        var_heat_tr_coeff : ExternalAccess = modify
        min_heat_tr_coeff : ExternalAccess = modify
        Icon = 'convection_variable.svg';
    end
    intermediates(ExternalAccess = none)
        heat_tr_coeff_ = if var_heat_tr_coeff >= min_heat_tr_coeff, var_heat_tr_coeff else min_heat_tr_coeff end; % Heat transfer coefficient
    end
    equations
        assert(min_heat_tr_coeff > 0)
    end
end

equations
    assert(area > 0)
    assert(heat_tr_coeff_ > 0)
    Q == area * heat_tr_coeff_ * T;
end

end
```

Fig. 3-21 Code of variable convective heat transfer coefficient block

The code is

```
inputs (ExternalAccess = none)
    var_heat_tr_coeff = {20,'W/(m^2*K)'}; %K:left
end


parameters
    thermal_type = foundation.enum.constant_variable.constant; % Convection type
    %                           1 - Constant
```

```
  %                          2 - Variable

  area         = {1e-4, 'm^2'}; % Area
end

parameters (Access = public, ExternalAccess = none)
  heat_tr_coeff = {20,   'W/(m^2*K)'}; % Heat transfer coefficient
  min_heat_tr_coeff   = {1e-3, 'W/(m^2*K)'}; % Minimum heat transfer coefficient
end

% Parameter and visibility checks
if thermal_type == foundation.enum.constant_variable.constant
  annotations
    heat_tr_coeff : ExternalAccess = modify
  end
  parameters(ExternalAccess = none)
    heat_tr_coeff_ = heat_tr_coeff; % Heat transfer coefficient
  end
  equations
    assert(heat_tr_coeff > 0)
  end
else %if thermal_type == foundation.enum.constant_variable.variable
  annotations
```

```
        var_heat_tr_coeff : ExternalAccess = modify
        min_heat_tr_coeff : ExternalAccess = modify
        Icon = 'convection_variable.svg';
    end
    intermediates(ExternalAccess = none)
        heat_tr_coeff_ = if var_heat_tr_coeff >= min_heat_tr_coeff, var_heat_tr_coeff
else min_heat_tr_coeff end; % Heat transfer coefficient
    end
    equations
        assert(min_heat_tr_coeff > 0)
    end
end


equations
    assert(area > 0)
    assert(heat_tr_coeff_ > 0)
    Q == area * heat_tr_coeff_ * T;
end
end
```

And now the convective heat transfer is not a constant anymore, it’s a dynamic signal from current velocity.

3.2 Simulink 3d animation- vehicle dynamics visualization with graphs

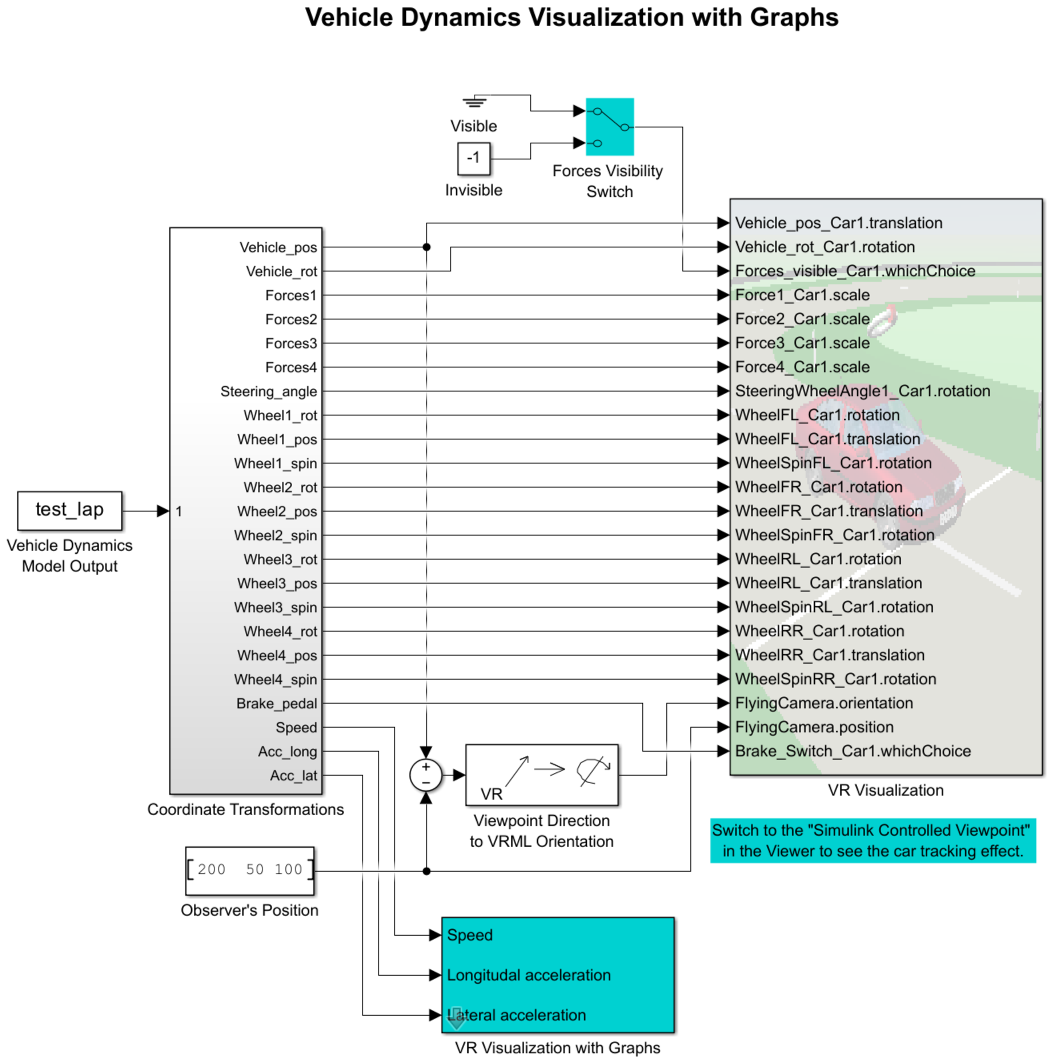

Fig. 3-22 Vehicle dynamics visualization part

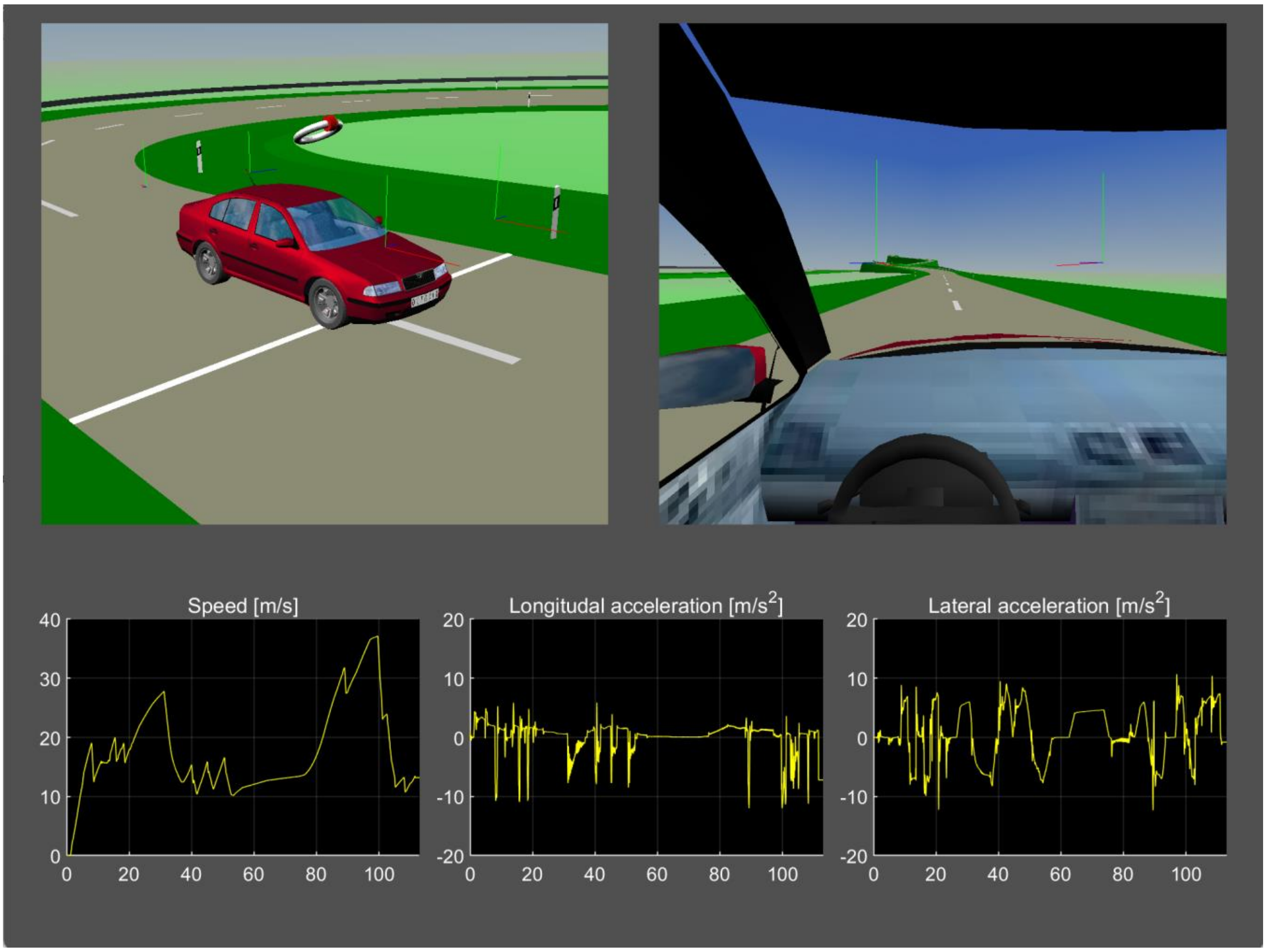

Fig. 3-23 Car diving animation from predefined data

The pre-computed simulation data, representing one lap of a vehicle on a testing circuit, is converted from dynamic model signal structure into the form that can be sent to the virtual reality scene. This conversion includes splitting the combined signals into signals governing individual VRML objects properties and coordinate systems transformations. The data is then sent to the virtual reality scene using VR Sink block with sample rate 25 Hz.

The dynamic model provides not only signals that influence visual properties of objects (positions and rotations), but also forces and other similar quantities, that can be visualized as well. In this example, wheel forces are visualized using VRML triads scaled dynamically according to current force values. Force triads can be switched off by the Forces Visibility Switch.

When working with virtual reality models, there is a common requirement to track certain moving objects with the camera. Usually, the direction from the camera to the object is easily available (both positions are known, so direction from the camera to the object is defined as the difference between the two positions). Because in VRML the viewpoint orientation is defined in the form of 4-element [axis angle] VRML rotation, camera direction must be converted into this format before sending to virtual scene. Simulink 3D Animation provides the Viewpoint Direction to VRML Orientation block to perform this conversion. In this example, switch to the "Simulink Controlled Viewpoint" in the viewer to see the camera tracking effect.

Simulink 3D Animation allows two methods of recording animation files - 2D animation (AVI files) and 3D animation (VRML files). Here we describe how a 3D VRML animation file can be created and further used. In the internal viewer Recording menu select the Capture and Recording Parameters... option.

In the Capture and Recording Parameters dialog, select the Record to VRML checkbox. Leave the File: editbox at its default value. Select the Scheduled Recording mode and specify the time period for which you want to record the offline animation time. Recording the animation during the entire simulation time defined for this model, you may select any time interval between the Simulink model Start time and Stop time. For details on recording parameters please refer to the product documentation.

Once define these animation recording parameters, animation file is created automatically when start the simulation. Unlike with creating the 2D animation files, the internal viewer figure can be minimized during simulation, which significantly speeds up the simulation. At the end of simulation run, the created VRML animation file octavia_scene_anim_1.x3d remains in the working directory for later use.

Chapter4: Simulation results

Convection heat transfer coefficient is an important part in vehicle HVAC system, but for driving vehicles constant convection heat transfer coefficient is unprecise.

4.1 Default parameter

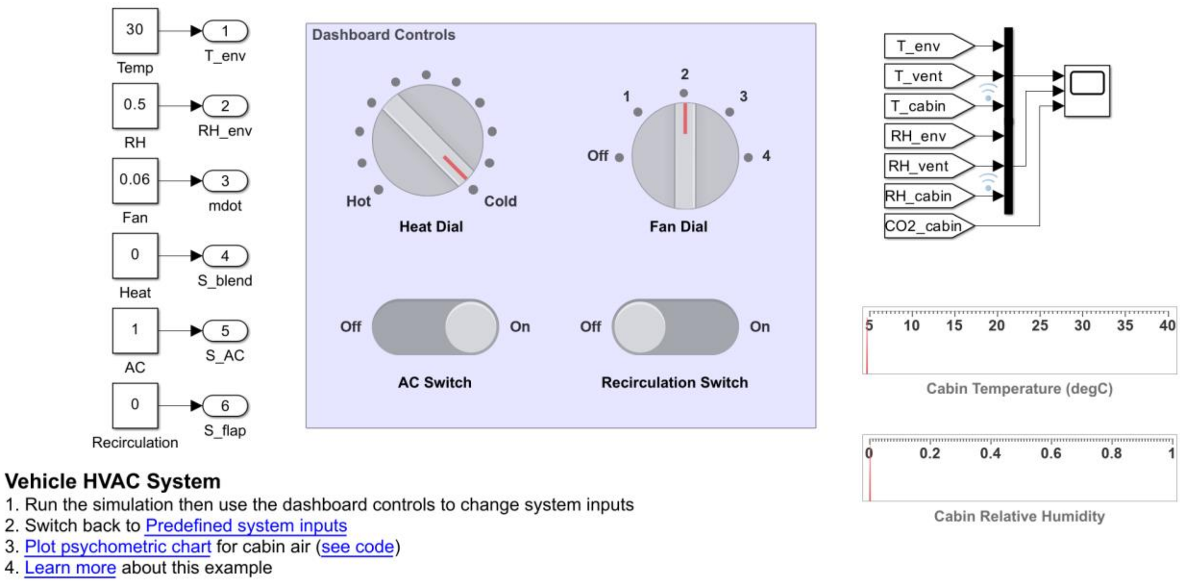

Fig. 4-1 Input parameter panel of vehicle HVAC system

In this model there several default parameters, environment(ambient) temperature 30 °C, ambient relative humidity is 0.5, air flow velocity of air conditioning is 0.06 kg/s, blend door signal is no heat core, air conditioning is on and recirculation in vehicle is off.

Table 4-1 Default parameter

Block	Content	Default Value
Temp	ambient temperature	30 °C
RH	ambient relative humidity	0.5
Fan	air flow velocity of air conditioning	0.06 kg/s
Heat	heat dial to control vent temperature	no heat core
AC	air condition	On
Recirculation	circulation in vehicle	Off

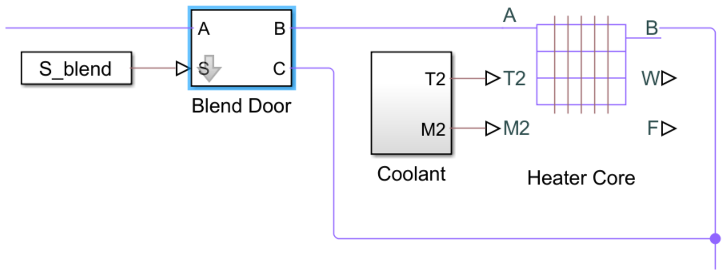

Fig. 4-2 Blend door block

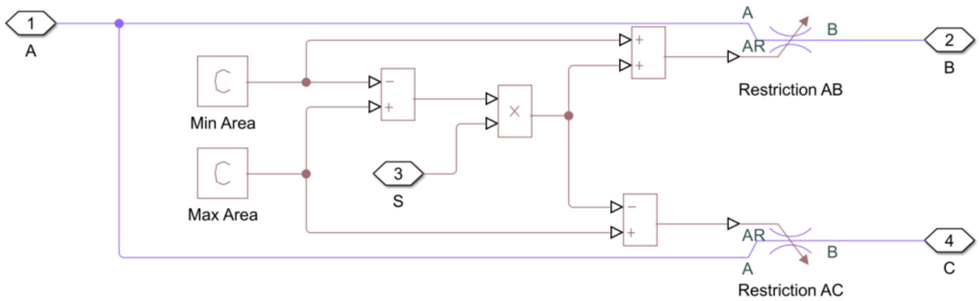

Fig. 4-3 Blend door block details

In blend door block, this subsystem models a restriction that controls flow between path AB and path AC in a moist air network. The relative restriction area is set by the physical signal port S. A value of 1 opens path AB and closes path AC. A value of 0 closes path AB and opens path AC.

4.2 Results of default vehicle HVAC model

4.2.1 Basic information

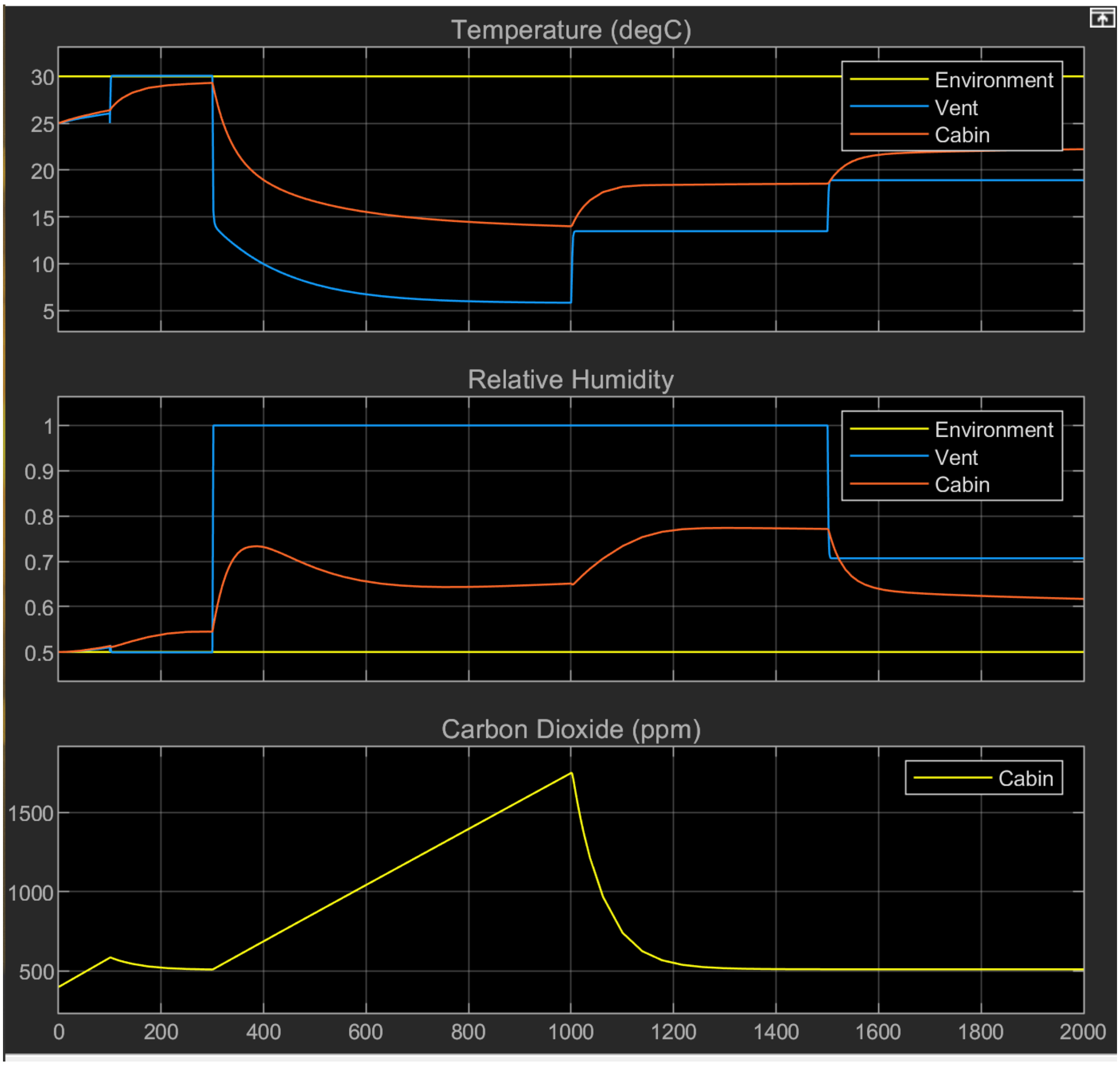

Fig. 4-4 Temperature, relative humidity and carbon dioxide graph

4.2.2 Psychrometric Chart

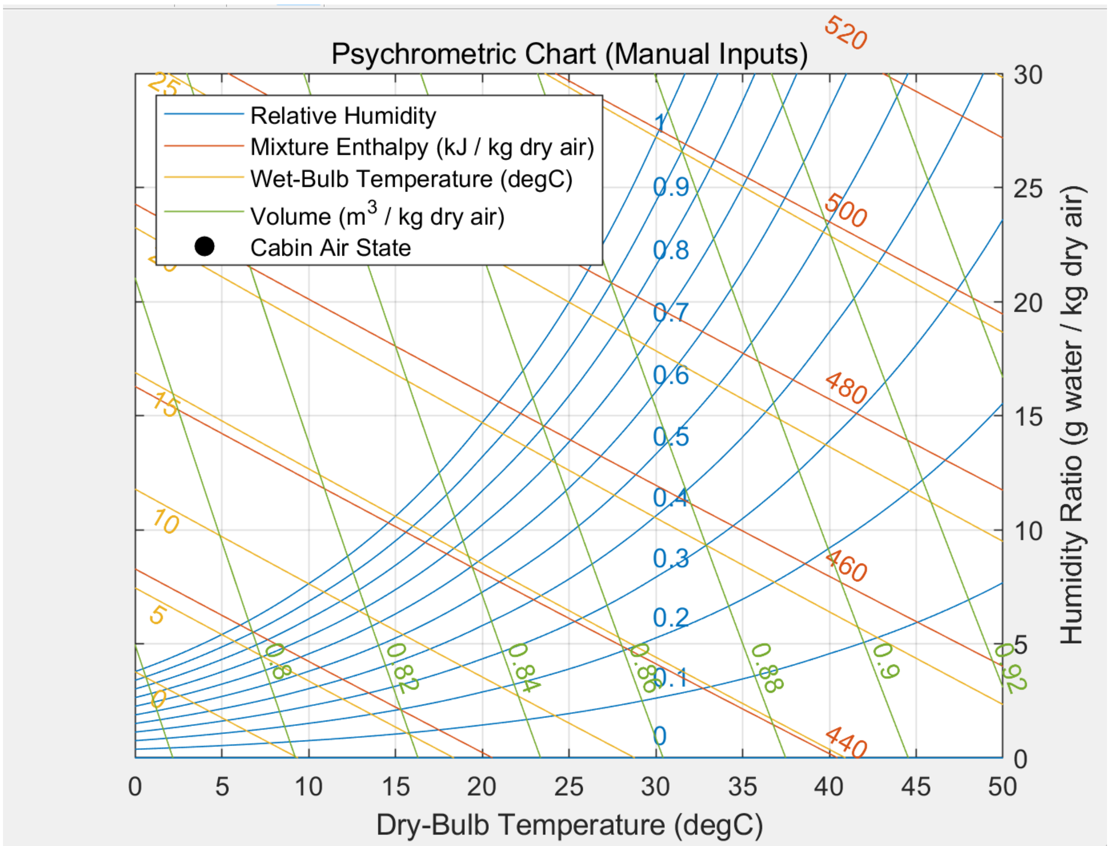

Fig. 4-5 Psychrometric chart with predefined parameters

4.2.3 Using other parameters

Table 4-2 Manually input parameters

Block	Content	Default Value
Temp	ambient temperature	10 °C
RH	ambient relative humidity	0.5
Fan	air flow velocity of air conditioning	0.08 kg/s
Heat	heat dial to control vent temperature	0.9
AC	air condition	On
Recirculation	circulation in vehicle	On

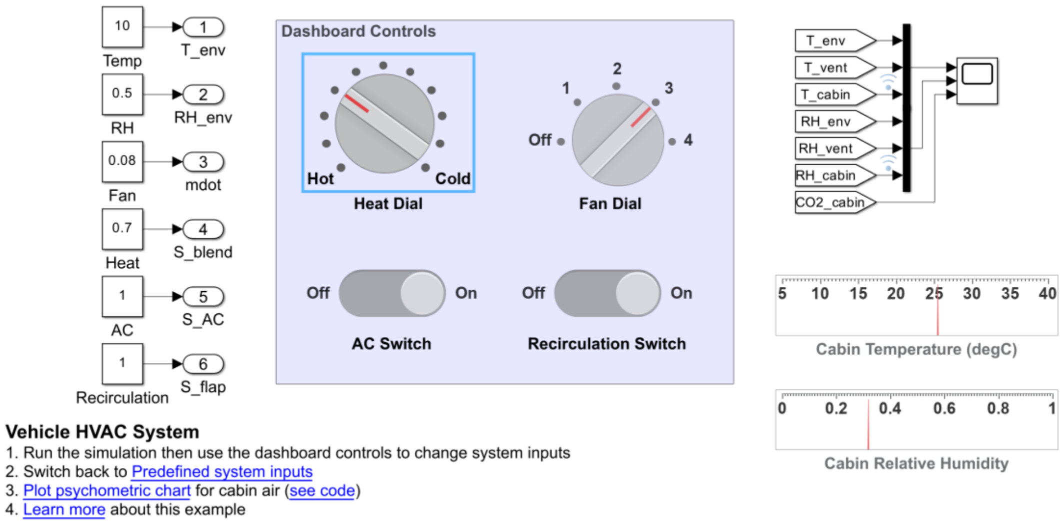

Fig. 4-6 Manually parameters of vehicle HVAC system

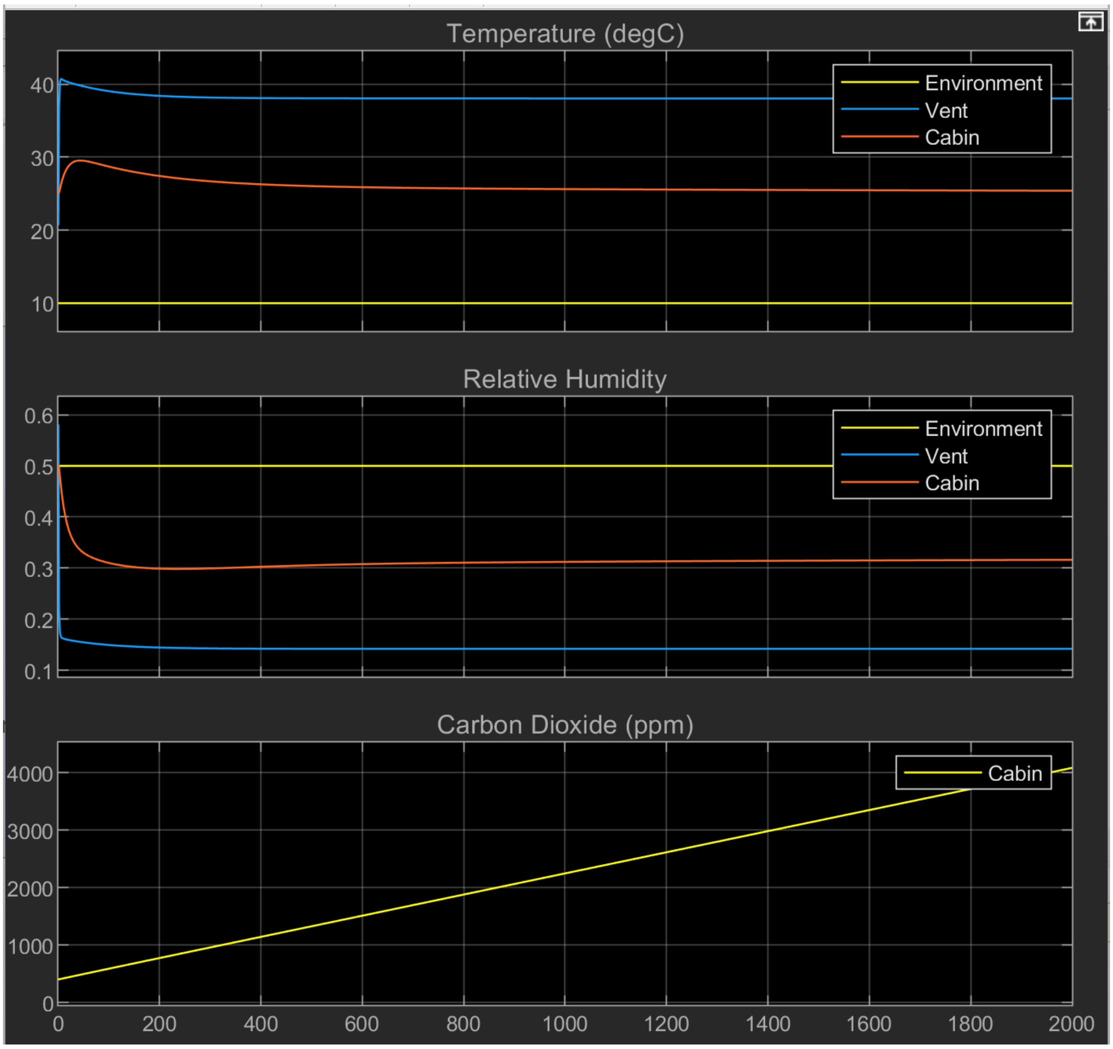

Fig. 4-7 Simulation results with manually parameters

4.3 Input dynamic convective heat transfer signal

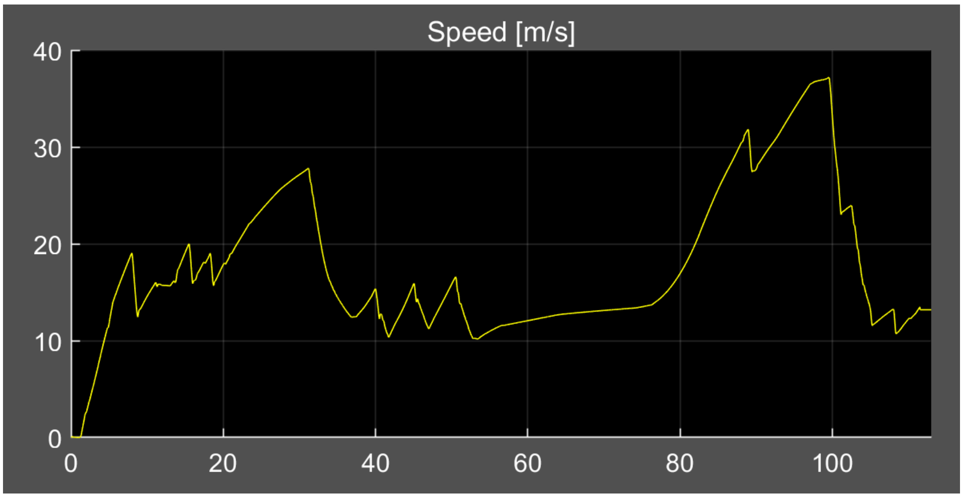

Fig. 4-8 Predefined input velocity data graph

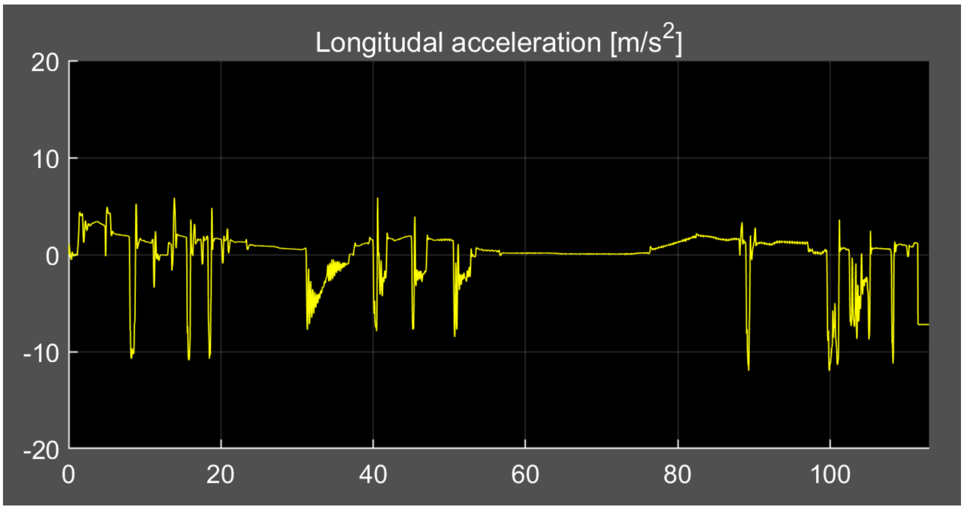

Fig. 4-9 Predefined input longitudal acceleration data graph

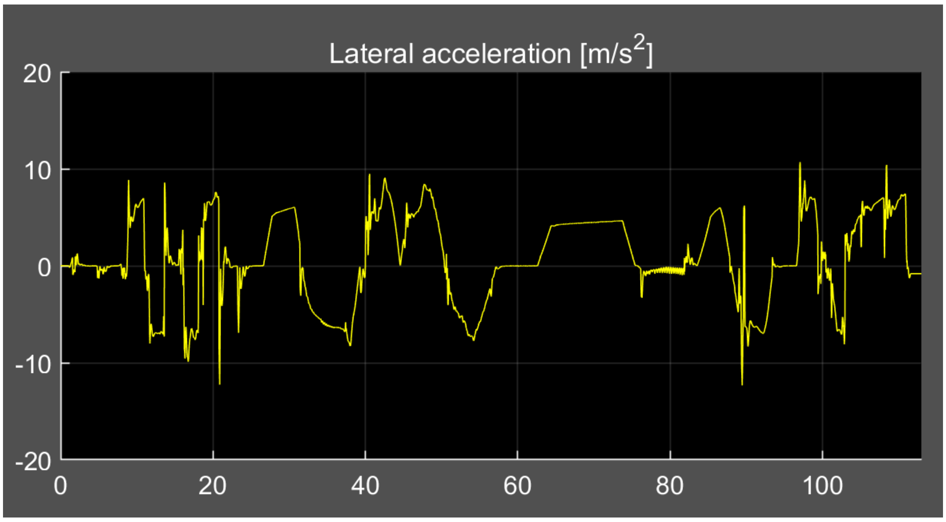

Fig. 4-10 Predefined input lateral data graph

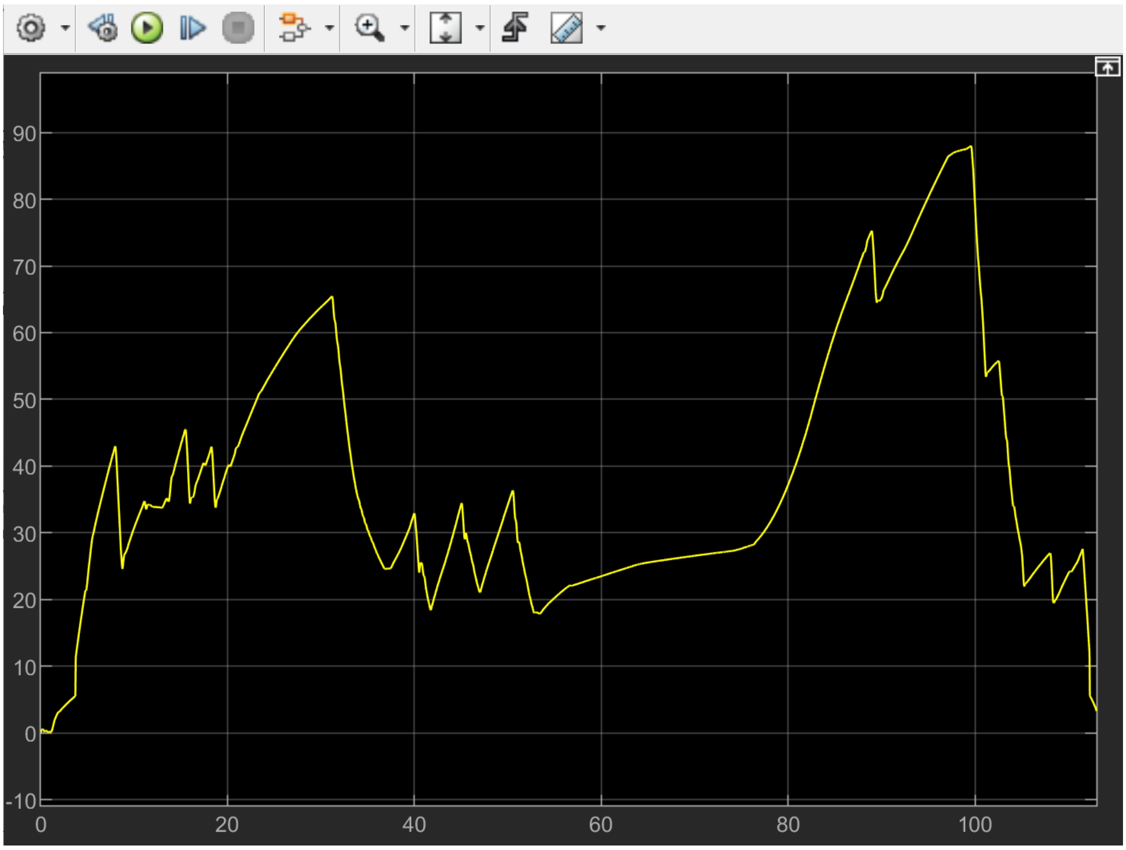

Fig. 4-11 Calculated convective heat transfer coefficient

Chapter5: Chapter 5 Results comparison

About the result comparison in convective heat transfer, the air flow rate (velocity of the vehicle) and convective heat transfer *h* must be taken into consideration. For raw *h* is a constant value 20, but now it's a variable up to the current velocity of vehicle.

5.1 Result graphs

For the constant *h* =20 is

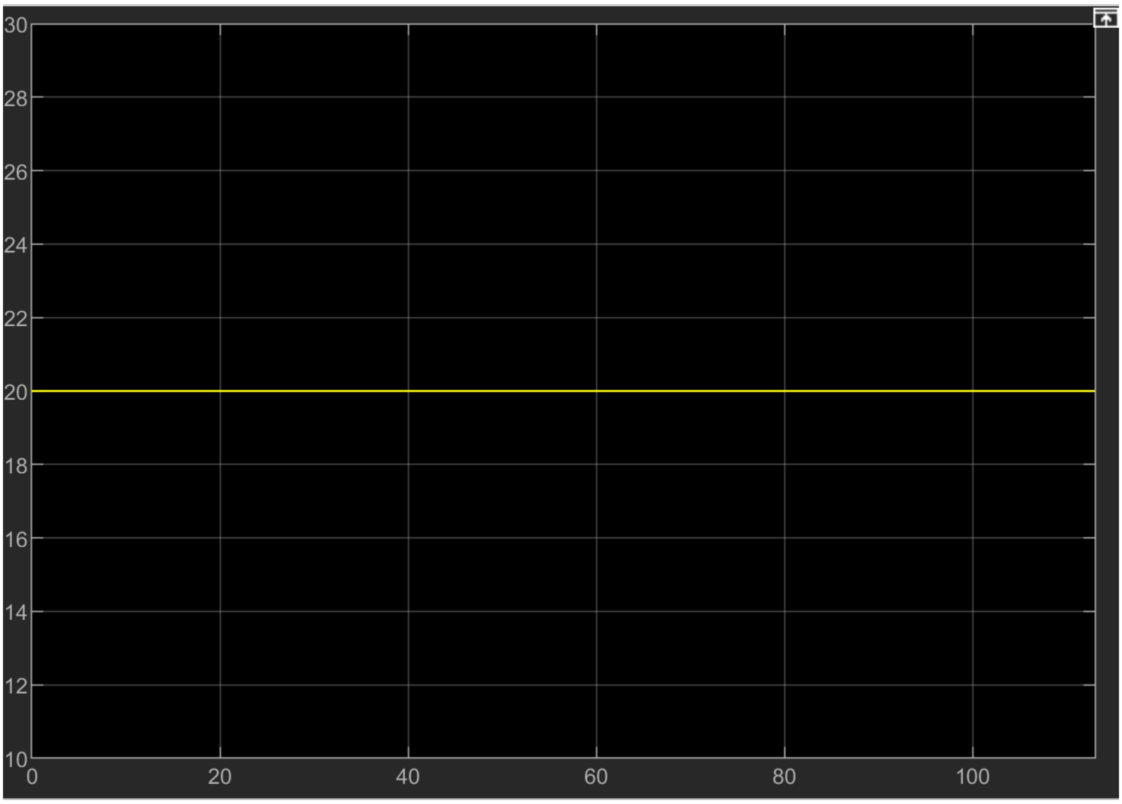

Fig. 5-1 Constant convective heat transfer coefficient *h*=20

For the signal of calculation from velocity of the vehicle, h is

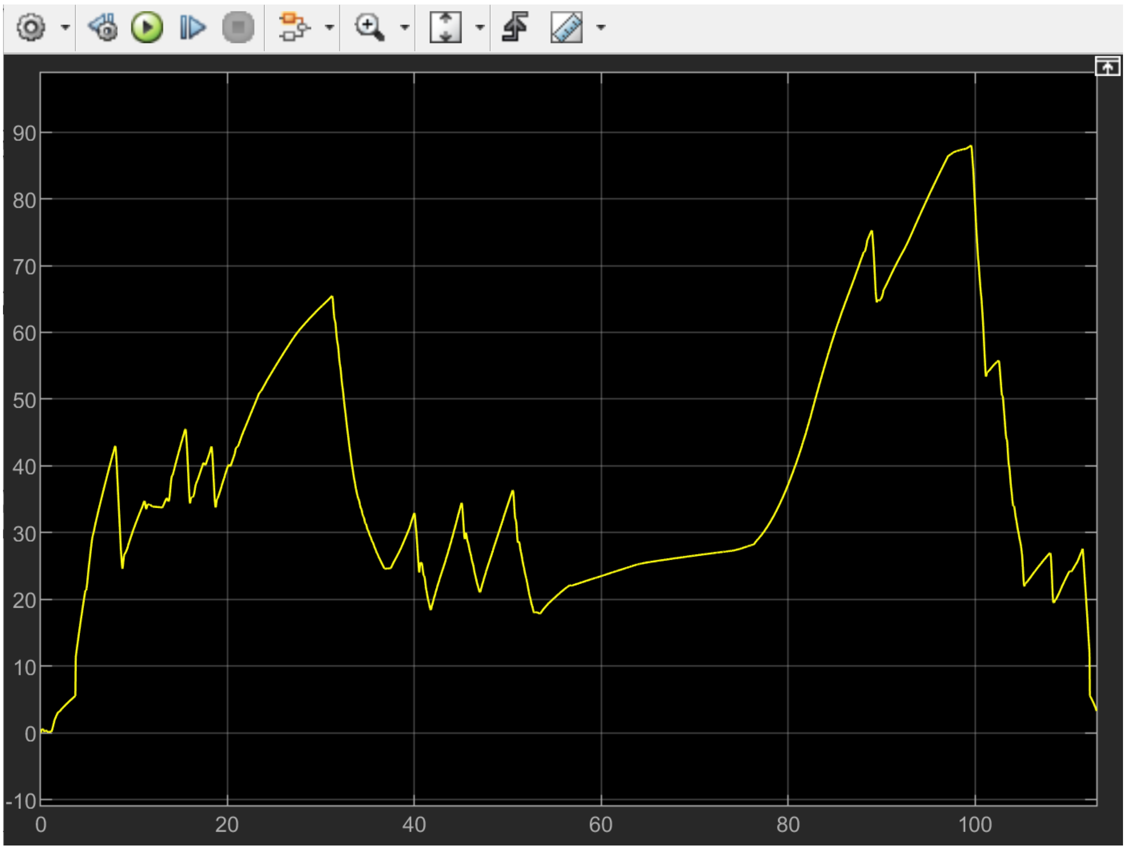

Fig. 5-2 Calculated variable h from predefined input velocity data(112s)

Therefore, the temperature, humidity and carbon dioxide graph of constant *h=20* is

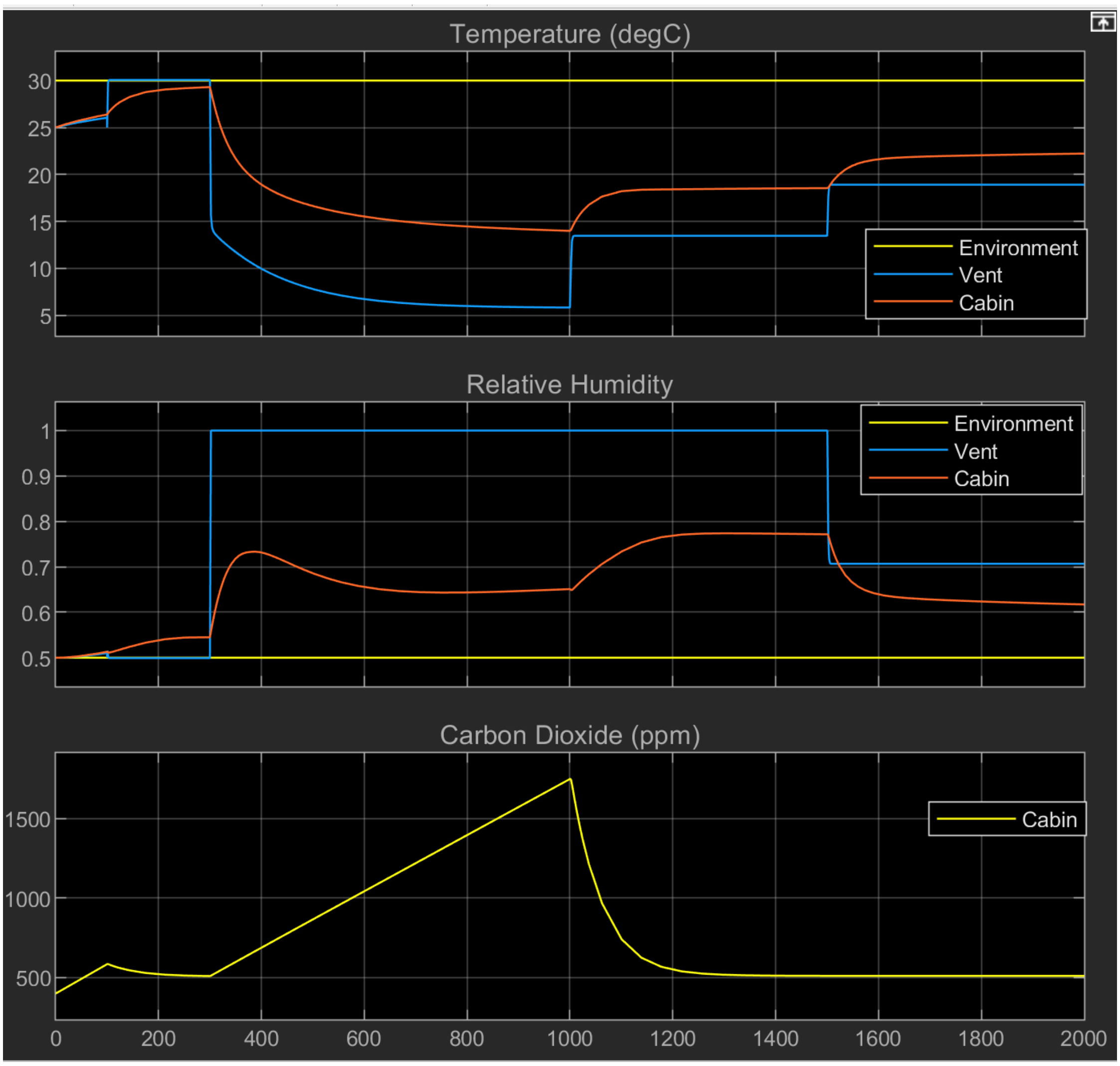

Fig. 5-3 Simulation results with constant heat transfer coefficient h=20(default simulation time 2000s)

Above result is based on predefined parameter, with the simulation time of 2000s.

For comparison, showing more detail graph, simulation results with no changed parameter and simulation time within 112s is showing below.

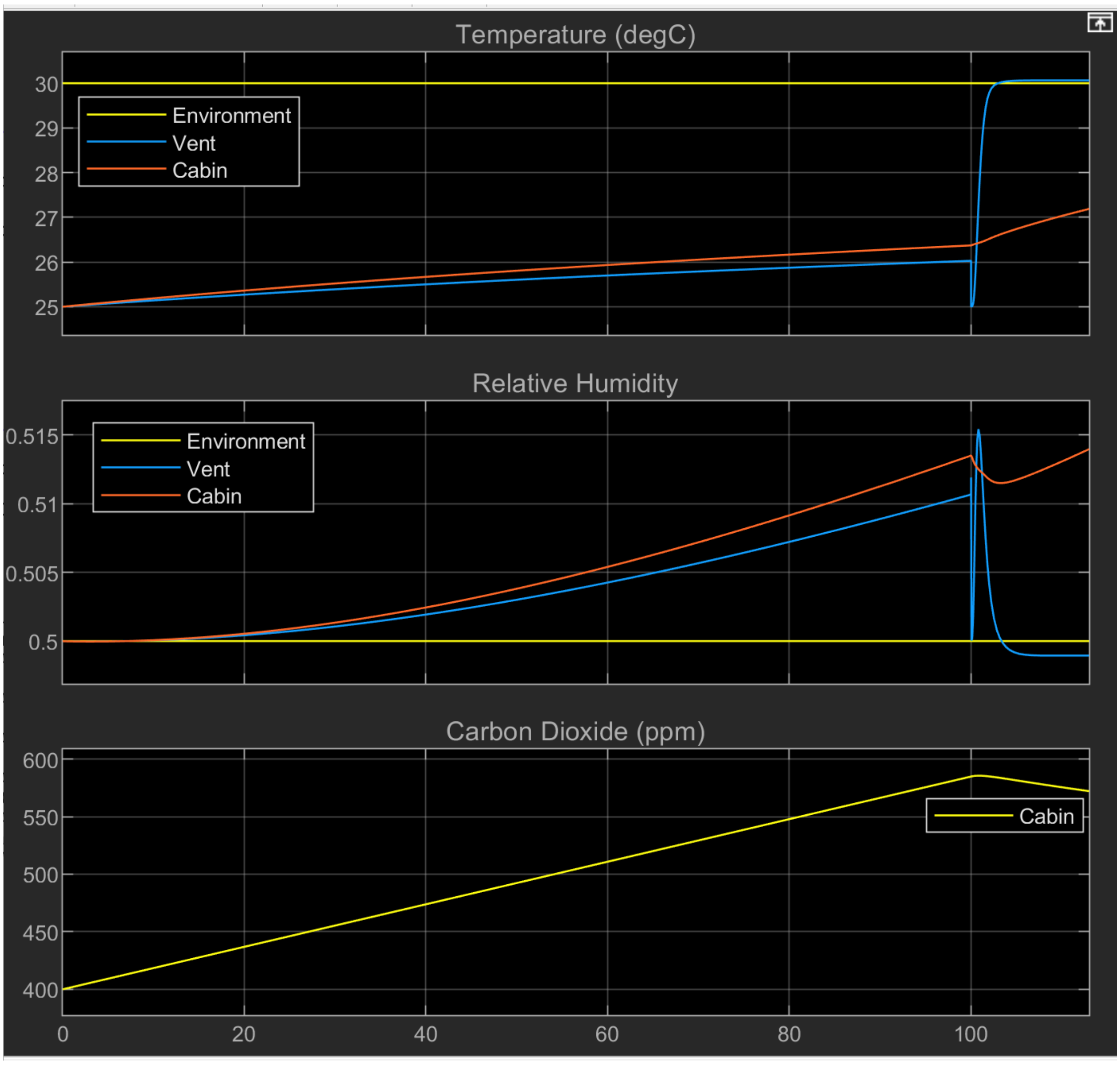

Fig. 5-4 Simulation results with constant convective heat transfer coefficient *h*=20(within 112s)

In contrast, the temperature, humidity and carbon dioxide graph of the dynamic *h* is

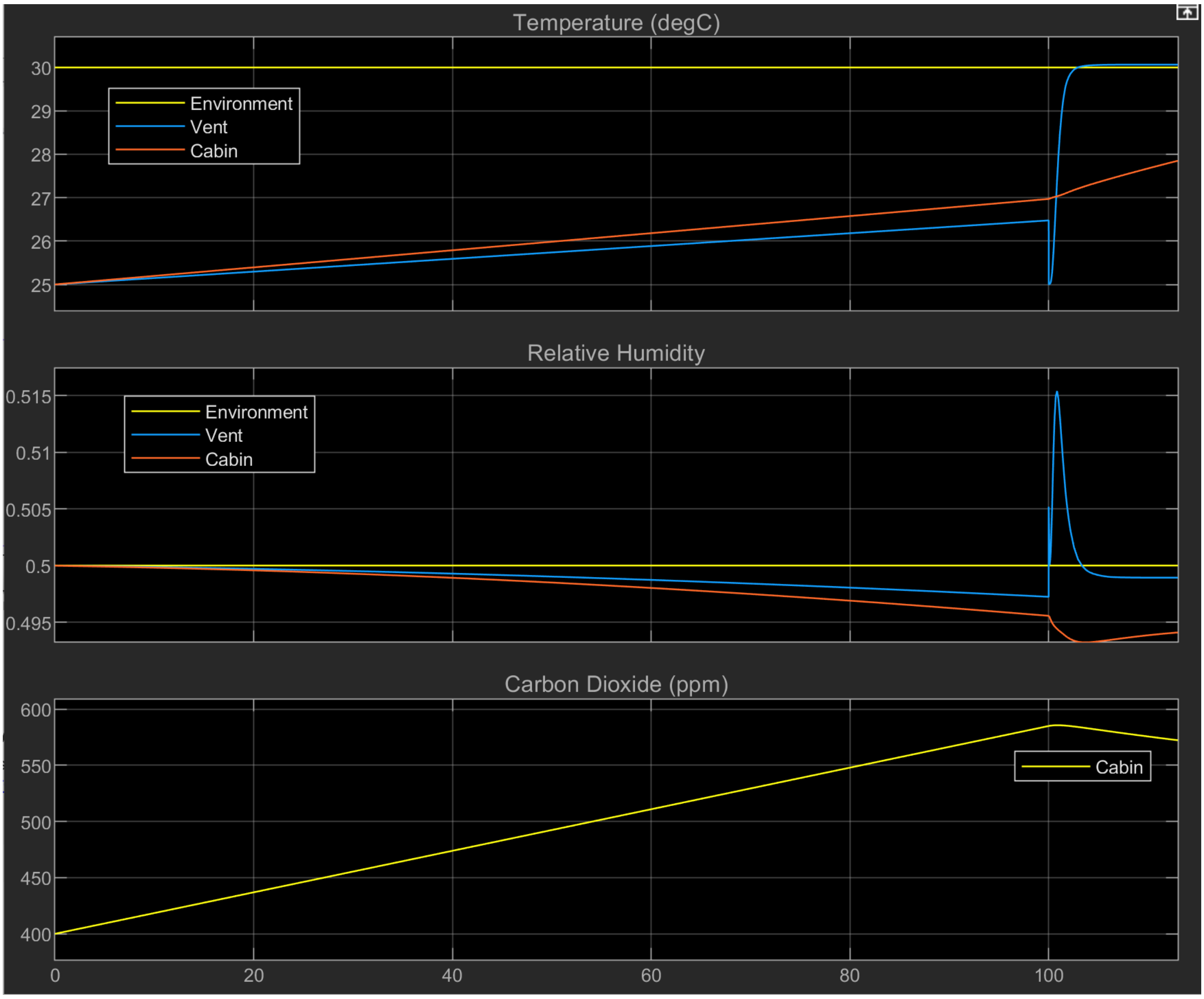

Fig. 5-5 Simulation results with variable convective heat transfer coefficient h (within 112s)

5.2 Discussion from results

Although the input data of velocity is no more than 112s, the difference among the graph is visible. At the situation of default parameters (environment temperature 30 ℃, ambient relative humidity is 0.5, air flow velocity of air conditioning is 0.06 kg/s, blend door signal is no heat core, air conditioning is on and recirculation in vehicle is off), at the point about 100s, the CO2 amount(ppm) is not clearly, but graph of vent and cabin temperature change as long as vent and cabin relative humidity change.

These two major turning points have changed. With dynamic h, at 100s, the cabin temperature is about 27 ℃ whereas with h=20 the temperature at 100s is no more than 26.5 ℃. Also, the vent temperature is different, with dynamic h, at 100s, the vent temperature is about 26.5-25 ℃ whereas with h=20 the temperature at 100s is no more than 26-25 ℃. About relative humidity in two graphs, is also different. With dynamic h, at about 100s, the relative humidity minimum value is less than 0.5 whereas with h=20 the temperature currently is about 0.5. As well, the relative humidity with h=20 shows an overall growth trend, whereas the relative humidity with dynamic h shows an overall decline trend and no more than 0.5.

Though the comparison, temperature increase at that point because of increment of h, the heat amount becomes larger. But the reason that relative humidity trend is getting down is unclear for now.

Chapter6: Chapter 6 Conclusion and future plan

From discussion and simulation above, here comes the conclusion.

6.1 Advantages and disadvantages of this systems

3d animation makes the virtual world more vivid, helping researchers to view full sight of experiments and makes it much easier to understand what is undergoing. But in this model, the input data is too much little to get full results. The predefined data is only up to this vehicle for one circle with 112 seconds. To get more precise heat transfer coefficient, more input data is need.

Vehicle HVAC system provides many modules to simulate for air conditioning in vehicles. But some parameters in this simulation may not be actual, more work is need to complete the simulation.

6.2 Conclusion with economy

In this study, the influence of the convective heat transfer coefficient on the heat exchange and HVAC system during the vehicle movement is analyzed in detail. With the optimization of HVAC system and the improvement of mileage performance, the modeling links between them will enable vehicle designers, policy makers and consumers to understand the interaction between vehicle technology, use conditions, climate, and energy use. For example, considering that regional power companies must plan to incorporate the traction and HVAC loads of electric vehicles into their load growth scenarios, consumers must understand that the performance of their vehicles is sensitive to climate, and vehicle designers should understand that improving the energy consumption of HVAC systems will improve the effectiveness of electric vehicles to meet energy consumption and sustainable development goals.

6.3 Future plan

First, about this system, the result and data are imperfect. To make an improvement, more driving data is need, therefore, I’m going to collect more vehicles driving data, such as vehicles environment temperature, relative humidity, air flow rate through vehicles surface (in this paper, the air velocity outside the vehicle is idealized and set to be the same as the vehicle speed.) and materials of vehicles. Also, I could do more experiments for the research of the relationship among velocity, air flow rate and heat transfer coefficient of driving vehicles.

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Chapter8: Acknowledgement

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Also, I would like to show my gratefulness to Associate Professor Kyaw Thu, for the first 3 months in Japan, I work as a teaching assistant for Associate Professor Kyaw Thu, to do the experiments in ITO campus. Thus, I am familiar with these thermal equipments.

Then, I would like to show my greatest gratefulness to my parents. When I was wandering in the charming metropolis of Fukuoka, I was once lost in the endless lights of the night. A person was too small to find the trace of home anywhere; I used to be intoxicated in an elegant and spacious university. A strong desire to be bold and self-sufficient at that time made me reluctant to board the return flight. But most importantly, after this trip, I began to appreciate and appreciate.

We should be grateful to our parents. I wrote a long letter to my dear parents after I came back. The first time I wrote about my own tears and my parents' tears, I still vaguely remember my original words: "I don't know what I can do. When you haven't set foot on this land, you can do the exploration for your future as you wish. I save every penny in Fukuoka, because I really know that there is too much support behind you."

I never believe that the craze for studying abroad comes from those experts who go abroad with full prizes. This long-standing tradition cannot make education a new industry with rapid development. Studying abroad can become a trend. Many students like me have promoted it and popularized it in the form of self-paying (or semi-self-paying). Family financial support should never be a kind of ostentatious capital but should be the beginning of people at our age to learn to appreciate and cherish.

I don't think any parents who are able to choose to send their children to study abroad

have ever thought of letting their children live outside by themselves. Pain is due to the nature of being a parent, but this has virtually made it natural for us to squander our family. We seem to be getting used to what our parents and families should pay for themselves and support this choice, but we ignore the fact that we have grown up. Why do we squander our parents' hard-earned money? I think many people have heard of the negative evaluation of contemporary international students and feel helpless. In short, it is impossible to go back to the era of reading by washing dishes and dishes. The change of this climate is a cancer of the industry of studying abroad. Unfortunately, most people haven't realized it yet. But this does not affect our gratitude. After all, it is not us who have changed this situation, but our respectable parents!

We should cherish great opportunities. In this competitive society, no matter when and where, I want to say that we really need to be satisfied when we have the opportunity to study abroad. This kind of contentment should be like the eyes of the girl of the Hope Project and should be like the back of those elders who are too hungry to read after the resumption of the college entrance examination. When most young talents just begin to enjoy the sweet fruit of higher education, we can take the first step to try new teaching concepts and experience new life. Although this step does not necessarily bring life-long advantages, it is undoubtedly leading.

I don't think that it is the idea of a few people to study abroad, but it is not the vision that every family and every student can achieve. Behind many envious peers, and behind many hard work and struggle, there is their helplessness to life. Today, when we emphasize harmony and fairness, and when higher education is far from universal, we have such resources and opportunities, which cannot be said to be an "unfair" possession. We have worked hard, and luckily walked to the front of the opportunity with the support of our parents and families. If we take advantage of resources and do not know how to cherish and work hard, such educational deformity will inevitably cause criticism.

There is no right to give. We should be grateful for studying abroad.

Last but not least, thanks Kyushu University so much for giving me this opportunity to study my master courses in this charming university. And for the next three years doctoral courses in Chikushi campus, I will work harder to do research and become a doctor who has contributed to our TECS laboratory and Kyushu University.