

Composite Thermal Model for Design of Climate Control System

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ABSTRACT

We propose a composite thermal model of the vehicle passenger compartment that can be used to predict and analyze thermal comfort of the occupants of a vehicle. Physical model is developed using heat flow in and out of the passenger compartment space, comprised of glasses, roof, seats, dashboard, etc. Use of a model under a wide variety of test conditions have shown high sensitivity of compartment air temperature to changes in the outside air temperature, solar heat load, temperature and mass flow of duct outlet air from the climate control system of a vehicle. Use of this model has subsequently reduced empiricism and extensive experimental tests for design and tuning of the automatic climate control system. Simulation of the model allowed several changes to the designs well before the prototype hardware is available. In addition to the reduced vehicle field tests and wind tunnel tests man hours and the cost associated with them, simulation of the model allowed for the greater potential benefits of increased accuracy and optimized heating and cooling of the passenger compartment to be achieved. An outlook is provided for integrating the composite thermal model of the passenger compartment with the human thermal comfort model to achieve greater benefit of optimized energy use by the climate control system and yet provide the thermal comfort to the occupants.

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INTRODUCTION

Several industrial applications of model-based engineering in the development are well known. In fact, the complex algorithms for features such as drive-by-wire and advanced engine management have mandated the use of modeling environment such as MATLAB/Simulink [1]. The graphical modeling tools such as Simulink and Stateflow allow the simulation analysis well before the physical prototype is made available [2]. Modeling tools let engineers develop algorithms in a flexible and collaborative way, allowing them to explore different aspects of design and test various approaches irrespective of the availability of the physical prototype. Nevertheless, testing on prototyping platform and even the production hardware is indispensable - the modeling tools support the need to produce code for prototype hardware as well as the production hardware. Model-based development environments therefore clearly provide a way to expedite the design of complex embedded control system and certainly have the impetus for shorter design cycles, greater levels of flexibility, lower costs and letting engineers focus on the domain of their expertise.

Another widely appreciated aspect of model-based engineering tools is that of letting engineers develop models specific to the several key system functions and test each designed *component-based* model individually, before integrating models to complete the embedded control system. Component-based software allows for defining component performance specifications, in addition to the system performance specifications. Also, the *tuning guides* can be developed component-wise. Component-based development lets system level performance issues to be categorized into component centric fixes, resulting in more cost-effective, less invasive, and faster performance resolutions. Furthermore, the software components are reusable and allows for the simultaneous system development by engineers [3].

Simulation of the complex designs such as the climate control system, using the graphical modeling tools, let engineers investigate many aspects of design including complexity, interactions between several subsystems, numerical aspects such as overflow and saturation, robustness and sensitivity to parameters and inputs - controlled as well as uncontrolled, etc. Simulation tools such as Simulink/Stateflow allow simultaneous simulation of continuous-time (differential) equations, used for

plant models and discrete-time (difference) equations, used for controllers, letting their extensive use in the development of control systems [4]. A thorough simulation of the system, under a wide variety of test conditions, allows for the analysis of a system behavior and prediction of the system performance to its inputs and parameters. However, power of the modeling tools, can be further exploited for achieving larger potential benefits, such as the increased accuracy, optimum use of energy, etc., with the use of a thorough understanding of the system dynamics, such as the heat transfer processes involved in the temperature control of the passenger compartment, than relying on the experimental data-based and experience-based designs, which, lacks in-depth study.

In order to investigate key influencing parameters of the climate control system, the model-based engineering tools - Simulink/Stateflow are used to develop a composite thermal model of the passenger compartment space of a vehicle. The composite thermal model is developed to accurately predict temperature of air inside the passenger compartment. Simulation of a model, under a wide variety of test conditions improved understanding of heat transfer mechanisms involved in the energy balance of a passenger compartment. Simulation results have clearly shown influence of several parameters and input variables - disturbance and manipulated control inputs of the climate control system. The composite model is also used to design a feedback control system for automatic control of climate - temperature and humidity of a vehicle passenger compartment. This approach superseded the earlier method of tuning the climate control system, which for a long time, relied upon a combination of experimental data, empiricism and experience. Simulation of the model along with the designed controller eliminated need for extensive field and wind tunnel tests, and the man hours and cost associated with them. Furthermore, simulations can be carried out under a wide set of conditions, which otherwise certainly would have required several man hours of the field and wind tunnel tests.

COMPOSITE THERMAL MODEL

A physical model is developed using the model-based engineering tools Simulink/Stateflow to simulate as far as is practical, the mechanisms of heat transfer, involved in a passenger compartment of a vehicle. An enclosed, by the walls or shell elements of a vehicle body, space or volume is considered to simulate heat transfer, more closely to the vehicle operating conditions. Sixteen surfaces - six glasses (four side screens, windshield and rear glass) and ten body (four doors, roof, instrument panel/dashboard, floor, rear seats, and parcel shelf) elements - along with their thermal properties are used, as illustrated in Fig. 1.

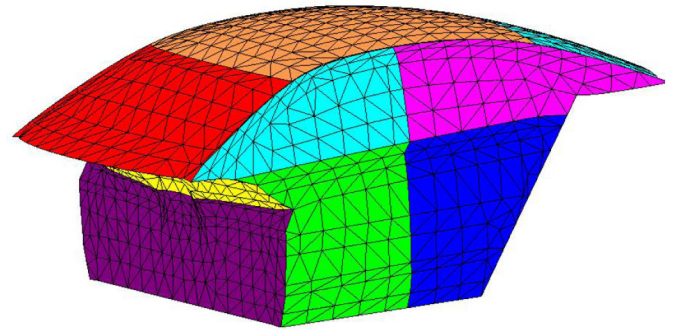


Figure 1. Vehicle body elements/surfaces, enclosing the passenger compartment space, are considered for modeling heat transfer by conduction, convection and radiation.

The passenger compartment is modeled as a *control volume* with the rate of mass flow into the volume is equal to the rate of mass flow out of it, i.e. $\dot{m}_{in} = \dot{m}_{out} = \dot{m}$ which essentially keeps pressure inside the volume constant. Energy balance was obtained for the bulk air inside the passenger compartment under steady operating conditions as follows.

$$\dot{Q} = \dot{m}c_p\Delta T \quad (1)$$

Here $\dot{Q}$ is the rate of net heat transfer into or out of the control volume in J/s or Watts, c_p is the specific heat capacity of air at constant pressure in J/kg °C, and ΔT is the change in temperature in °C of the mass of air inside the control volume. All the three modes of heat transfer i.e., conduction, convection and radiation are modeled to calculate the net heat transfer. Fig. 2 shows several mechanisms of heat flow into or out of the passenger compartment.

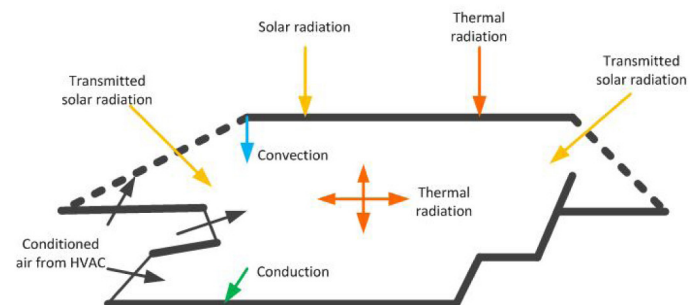


Figure 2. Several mechanisms of heat flow in and out of the control volume.

Heat Transfer by Conduction

Heat transfer from outer to inner surface of the shell elements is by conduction and is given by

$$\dot{Q}_{cond} = \frac{\lambda}{d} \cdot A \cdot (T_{se} - T_{si}) \quad (2)$$

where λ is the thermal conductivity of the shell element in W/m °C, d is the thickness of the shell element in meters, A is the surface area in square meters, and T_{se} and T_{si} are the exterior and interior surface temperatures, respectively.

Heat transfer by conduction also takes place from the rear defogger electrical grid to the interior surface of the rear glass and is given by

$$\dot{Q}_{rear,defog} = \frac{V^2}{R} \quad (3)$$

Here V is the voltage drop across the terminals of rear defogger electrical grid and R is the electrical resistance of the grid obtained as a function of temperature of the grid lines. For a known resistance value at a known temperature and for a given grid material such as silver or ceramic silver compound, R is given by

$$R(T) = R_{ref} (1 + \alpha(T - T_{ref})) \quad (4)$$

where α is the temperature coefficient of resistance of the material. For silver, value of α at 20°C is 0.003819 per °C.

Heat Transfer by Convection

In the presence of bulk fluid motion (for example, due to wind speed, vehicle speed, blower motor speed, etc.), heat transfer by convection also takes place. In fact, fluid motion enhances the heat transfer between the solid surface and fluid [5]. The convection heat transfer between air and surface at temperatures T_{air} and T_s , respectively is given by

$$\dot{Q}_{conv} = hA (T_{air} - T_s) \quad (3)$$

where h is the convection heat transfer coefficient in W/m² °C, and is determined experimentally for both the interior and exterior surfaces. Due to the uniform fluid motion and the bulk fluid velocity heat transfer coefficient for the interior surfaces are observed to be constant. However, due to a dynamic fluid motion on the outside, heat transfer coefficient for the exterior surfaces is not constant, and it was observed to obey the following rule, as shown in Fig. 3.

$$h_{extr} = 2 + 12 \sqrt{v} \quad (6)$$

Here v is the vehicle speed in m/sec.

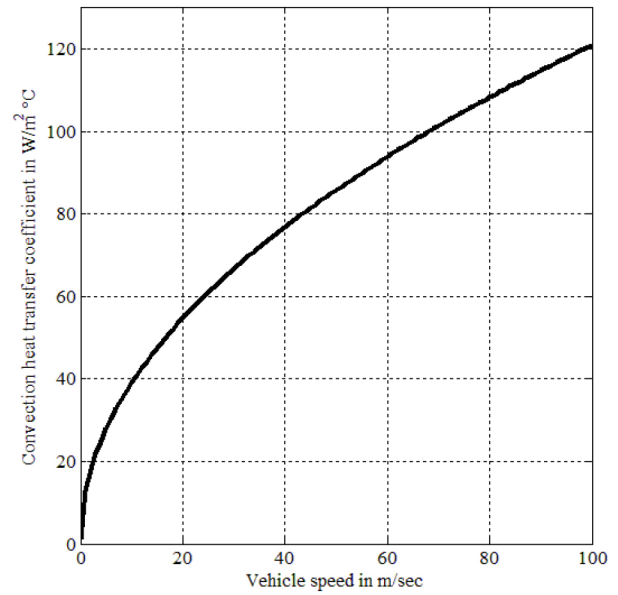


Figure 3. Variation in the convection HTC for the exterior surfaces is shown with respect to the changing vehicle speed.

Heat Transfer by Radiation

Heat transfer by radiation involves absorption of solar radiation - direct beam and diffuse solar radiation as well as exchange of thermal radiation between the shell elements, earth surface and sky. Solar radiation absorbed by a shell element is given by

$$\dot{Q}_{solar} = I_e A \beta \quad (7)$$

where β is the absorptivity of the shell element and I_e is the effective solar intensity, for a given impact angle between the solar rays and the surface normal, and is given by

$$\cos \gamma = \cos(\phi_1 - \phi_2) \cos \theta_1 \cos \theta_2 + \sin \theta_1 \sin \theta_2 \quad (8)$$

where $\theta_{1,2}$ are the elevation angle of the sun radiation and surface normal, respectively and $\phi_{1,2}$ are the azimuth angle of the sun radiation and surface normal, respectively. The effective solar intensity at the inclined surface for a given solar intensity I is given by

$$I_e = I \cos \gamma \quad (9)$$

For transparent (glass) elements, a portion of incident radiation which is not absorbed by the surfaces is transmitted by the surfaces, and is given by

$$\dot{Q}_{trans} = I_e A \tau \quad (10)$$

where τ is the transmissivity of the glass element, which is measured for the normal incident radiation, and therefore it shall be corrected for an angle of incidence of solar radiation, as illustrated in Fig. 4. The transmitted solar radiation to the compartment does not leave the compartment because glasses are not transparent to the long wave (or infrared) radiation; instead they are absorbed at the interior surfaces of the shell elements. Absorption of transmitted solar radiation by the bulk air inside the passenger compartment is found negligible, and is therefore ignored.

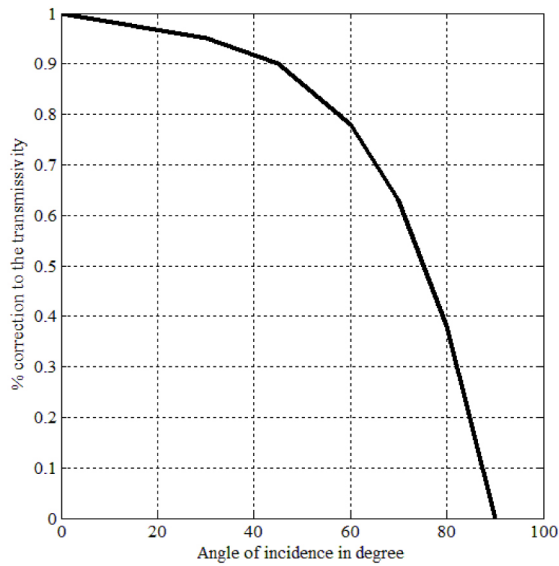


Figure 4. Transmissivity of a glass is shown to change with an angle of incidence of solar rays.

Heat also flows by thermal radiation i.e., the radiation emitted by a surface, between the shell elements, because of their temperature differences. The maximum rate of thermal radiation, emitted from a surface at an absolute temperature T_s is given by Stefan-Boltzmann law as follows.

$$\dot{Q}_{blackbody} = \sigma A T_s^4 \quad (11)$$

where $\sigma = 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4$ is the Stefan-Boltzmann constant. Equation (11) corresponds to a radiation emitted by a black body, whereas, the thermal radiation from a real surface at the same temperature is lesser than that given by equation (11), and is expressed by

$$\dot{Q}_{emit} = \epsilon \sigma A T_s^4 \quad (12)$$

where ϵ is the emissivity of a surface. Heat transfer by radiation between the shell elements also depends on the orientation of the surfaces relative to each other, in addition to their emissivity, absorptivity and temperatures. Geometric quantity to specify the orientation of the two surfaces is given by a view factor, F_{ij} defined as the fraction of radiation leaving surface i that strikes surface j directly [5].

$$\dot{Q}_{emit,ij} = A_i \cdot \epsilon_i \beta_j \cdot \sigma \cdot F_{ij} (T_i^4 - T_j^4). \quad (13)$$

For the sixteen surfaces of a passenger compartment, total 256 view factors are calculated from the CAE model. Following simple rules are useful when calculating view factor matrix.

1. Shell element surfaces are assumed to be flat, i.e., a surface cannot see itself, therefore giving $F_{i,i} = 0$,
2. The reciprocity rule for a pair of view factors F_{ij} and F_{ji} require them to follow a relation $A_i F_{ij} = A_j F_{ji}$, and
3. The conservation of energy principle requires that the radiation leaving a surface be intercepted by the surfaces of the enclosure. Therefore, the sum of the view factors from surface i of an enclosure to all surfaces of the enclosure, including itself shall be unity. This rule is known as the summation rule.

$$\sum_{i=1}^{16} F_{i,j} = 1. \quad (14)$$

Heat transfer by thermal radiation also occurs between the exterior surfaces of the shell elements, the earth surface and sky. Emissivity of the shell element and the earth surface is used in equation (12), to calculate thermal radiation emitted by exterior surface of the shell element and by the earth surface, respectively. The thermal radiation from a clear sky is given by an Ångström equation [6], as follows.

$$\dot{Q}_{emit,clear\ sky} = \sigma T_{air}^4 (0.790 - 0.174 \cdot 10^{-0.041 \cdot p_w}) \quad (15)$$

Here, p_w is the partial pressure of steam given by

$$p_w = \frac{p \cdot x}{0.622 + x}. \quad (16)$$

Atmospheric pressure $p_0 = 1013.25 \text{ hPa}$ is used for the barometric pressure p , and x is the moisture content of air in $\text{kg of water vapor/kg of dry air}$. Clouds are assumed to emit thermal radiation like a blackbody (i.e., emissivity $\epsilon = 1$) at a dew-point temperature (or 100% relative humidity) of the ambient air. Cloud coverage data is used to calculate thermal radiation from the sky that is, by combining thermal radiation from clouds and the clear sky.

The net energy transfer between the exterior surface of a shell element, earth surface and sky as illustrated in Fig. 5, is given by the following expression.

$$\dot{Q}_{thrm,extr} = \dot{Q}_{1,2} + \dot{Q}_{1,3} = A_1 \beta_1 \sigma (\epsilon_2 F_{12} (T_2^4 - T_1^4) + \epsilon_3 F_{13} (T_3^4 - T_1^4)) \quad (17)$$

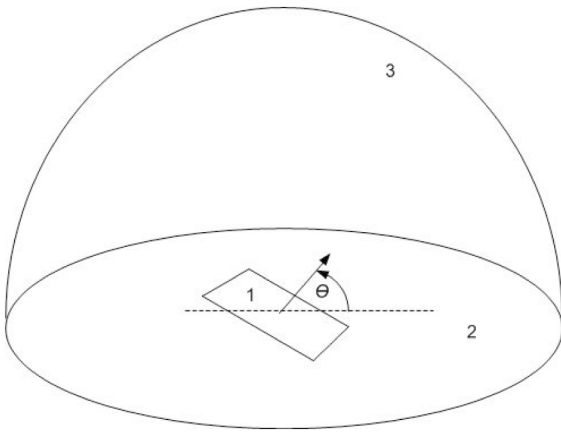


Figure 5. View factor of a flat surface 1 with the earth surface 2 and sky 3. θ is the elevation angle of a normal to the flat surface 1.

For a flat surface, we have $F_{11} = 0$. And, $F_{12} = 0.5 (1 - \sin \theta)$ where θ is the elevation angle of the shell element [6]. From the summation rule, we obtain $F_{13} = 1 - F_{11} - F_{12}$.

Net Heat Transfer and Equation of Temperatures

We combine the above heat transfer expressions to obtain for exterior and interior surface temperatures of the shell elements and the bulk air temperature of the passenger compartment.

$$\dot{Q}_{extr} = \dot{Q}_{thrm,extr} + \dot{Q}_{solar} + \dot{Q}_{conv,T_{air} \rightarrow T_{se}} - \dot{Q}_{cond,T_{se} \rightarrow T_{si}} \quad (18)$$

$$\dot{Q}_{intr} = -\dot{Q}_{thrm,intr} + \dot{Q}_{trans} - \dot{Q}_{conv,T_{si} \rightarrow T_{air}} + \dot{Q}_{cond,T_{se} \rightarrow T_{si}} \quad (19)$$

Surface temperatures are obtained by integrating the net heat transfer for a surface over a period of time, as given below.

$$T_s(t) = \frac{1}{\dot{m} c_p} \int_{t_0}^t \dot{Q}_{net} dt + T_s(t_0). \quad (20)$$

Heat flow to the bulk of an air inside the passenger compartment occurs from the two sources, simultaneously. These are convection heat transfer from the interior surface of the shell elements, and the heat flow from the Heating, Ventilation and Air Conditioning (HVAC) system of a vehicle, as illustrated in Fig. 2. The rate of heat transfer is defined in terms of the difference of enthalpies, for air coming out from the duct outlets at T_{air} , and for the bulk air inside the passenger compartment at T_{bulk} , as follows.

$$\dot{Q}_{air} = \dot{m}(h_{in}(T_{air}) - h_{cabin}(T_{bulk})) \quad (21)$$

where h is the specific enthalpy of moist air at temperature T , and is given by

$$h(T) = c_p^{Air} T + x(r_0 + c_p^{Steam} T) \quad (22)$$

Here c_p^{Air} , and c_p^{Steam} are the specific heat capacities of air and steam, respectively for a constant pressure, x is the moisture content of air in kg of water vapor per kg of dry air, and r_0 is the heat of evaporation of liquid water at 0°C.

Occupants of the vehicle also emit heat at a rate depending on their clothing, level of activity and surface area. Model assumes that the two occupants are present in a vehicle, and each occupant is emitting heat at 50 J/s (or Watts) corresponding to the summer clothing and a normal level of activity.

$$\dot{Q}_{human} = 50 W \quad (23)$$

The net rate of heat transfer to the bulk air is given by

$$\dot{Q}_{net,bulk} = \dot{Q}_{conv,T_{si} \rightarrow T_{air}} + \dot{Q}_{air} + \dot{Q}_{human} \quad (24)$$

Change in the bulk air temperature is calculated using equation (1), and is integrated to obtain the bulk air temperature at time t as follows.

$$T_{bulk}(t) = \frac{1}{\dot{m} c_p} \int_{t_0}^t \dot{Q}_{net,bulk} dt + T_{bulk}(t_0) \quad (25)$$

In equation (20) and (25), the initial temperatures are obtained by weighing the sensor data for the outside air temperature, the in-car temperature, and the duct air temperature. These are usually present in a vehicle with the automatic climate control system. The weighing factors for initialization temperatures for several interior and exterior surfaces and the bulk air temperature are obtained experimentally under a various vehicle operating conditions. The man hours and cost of obtaining initialization temperatures experimentally, is significantly lower than that for developing the climate control system via field and wind tunnel tests.

SIMULATION RESULTS

Simulation results are obtained for the wide variations in several inputs to the composite thermal model of a passenger compartment of a vehicle. Simulation results have shown a high sensitivity of the model to inputs such as the outside air temperature, the air temperature and mass flow at the duct outlets, as shown in Fig. 6, 7, 8, 9. Simulation results for changes in the ambient humidity, vehicle speed, and the distribution of air from duct outlets - vent, floor, and defrost - into the passenger compartment, on contrary have shown relatively less or insignificant sensitivity to the temperature outputs of the composite thermal model.