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A State Space Approach for the Dynamic Analysis of Automotive Air conditioning System

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ABSTRACT

Automotive Air Conditioning (AAC) system poses unique challenges for designers, to fulfill the customers comfort and efficient operation in wide changing ambient temperature (18 °C to 45 °C) and humidity (30 to 80 R.H). This paper presents the development of a state space dynamic model (SSDM) for analyzing the vehicle air conditioning in both steady and transient mode. This model includes a vehicle cabin, variable displacement capacity (VDC) compressor and an evaporator. An experimental vehicle made up of original components from the air conditioning system of a compact passenger vehicle has been developed in order to check the results from the model. This SSDM has a non linear relationship between output state variables (cabin temperature, cabin humidity) and input manipulated variables (compressor speed, blower fan speed). It confirms the developed model after linearization around the operating point was experimentally valid, to capture the transient change in system parameter and to represent the steady state operation within in the acceptable range. Thus SSDM can be especially useful in developing different control strategies like Multi loop control, Multivariable control and Model Predictive Control etc. for improving passenger comfort, fuel economy, drivability and stalling of engines.

Keywords: AAC- Automotive Air Conditioning, VCRS - Vapor Compression Refrigeration System, TXV- Thermostatic Expansion Valve, SSDM- State Space dynamic model, VDC- Variable Displacement Capacity, PSL – Piston Stroke Length, Evaporator, Condenser.

1. INTRODUCTION

The air conditioning system used in automobile works under the vapor compression refrigeration principle (Shah et al., 2003), (Lou, 2005) and has the following four main components, namely compressor, condenser, thermostatic expansion valve (TXV), evaporator all connected in the sequential closed loop as shown in fig 1. In vehicle real environment, evaporator alone is integrated with the vehicle cabin. Air-Conditioning is one of the most significant sub-systems of a vehicle with respect to comfort, fuel consumption and drivability of a passenger is concern (Benouali et al., 2003), (Johnson, 2002), (Watanabe, 2002). The main aim of the AAC system is to maintain the desired temperature, humidity and air purity inside the vehicle cabin (Park, 2006), thereby keeping the comfortable cabin environment and reduce stalling of engine (Nadamoto and Kubato, 1999).

The first principle based dynamic mathematical model is a key tool to study the system performance and to design successful controller for vehicle air conditioning. So many research works are reported in (Keir et al., 2006), (Rasmussen et al., 2002) and (Qi and Shiming, 2008) on developing dynamic model of ACC system to improve its structure and efficiency. In conventional vehicle, fixed displacement compressor is used, with a belt driven from engine. It continuously cycles on and off to the set point temperature of the user, which creates additional thermodynamic losses (Nadamoto and Kubato, 1999), (Duo et al., 2005). Recently, (Shah et al., 2003) and (Kier et al., 2006) proposed the successful adoption of variable displacement compressor and electronic expansion valve over fixed displacement compressor is that the change in mass flow rate of refrigerant exactly match the user demand of the user. It has the advantage of more comfortable environment inside the vehicle, improved fuel economy and smooth continuous compressor operation.

Several works have been done on steady state and dynamic modeling of an ACC. Ding and Zito(2001), Castro et al.,(1993),Melon et al.,(2002), Rahman et al.,(2003) and Kelemen et al.,(2000) proposed a computer simulation of system components like fixed displacement compressor, evaporator , condenser and mechanical thermostatic expansion valve under steady state and dynamic condition in vehicle air conditioning. Khamsi and Petitjean (2000) proposed a dynamic model based on physical and parametric approach and compared with experimental data of automotive passenger compartment and its air conditioning system. Josef Hager et.al(2001) proposed a simulation tool for ACC in transient condition based on network theory algorithms. Singh et al.,(2000) proposed a adaptive control, based on the dynamic model of HVAC system, and the simulation results shows the closed loop response of the system to changes in operating points, external disturbance, change in system parameters. It shows the adaptive controller is able to adapt to a wide range of operating conditions and is able to maintain the zone temperature and humidity. Elliott and Rasmussen (2009) proposed a Model predictive controller as global controller that generates different set points for the local controllers of a multi evaporator HVAC system. Multi variable control proposed by Qi and shiming (2008) for in door air temperature and humidity in a direct expansion air conditioning system based on separately varying compressor speed and supply fan speed.

Li and Shung-Luen (2009) studied the dynamics of temperature and humidity of air conditioning inside the vehicle air conditioning based on energy balance and mass conservation. But the work does not give details of the cooling and heating load given to the vehicle cabin. Razi and Farokli (2008) proposed a numerical model for fixed displacement compressor based on numerous laboratory test on typical passenger car, dividing vehicle in to two linked modules namely passenger compartment and air conditioning system. Based on the numerical model, neuro predictive controller for temperature and humidity is developed. AAC system is associated with many process variables like compressor speed, piston stroke length variation, air flow rate in blower fan, air flow rate in condenser, mass flow rate of refrigerant, expansion valve opening etc. Also ACC system includes disturbance variables such as solar load, passenger sensible and latent heat load, ambient temperature and humidity, infiltrated ambient fresh air etc. Hence predicting the controlled variables like cabin temperature, cabin humidity, cabin interior mass temperature and evaporator air side temperature and humidity, etc. becomes tedious to compute dynamically as shown in fig 2. There is limited published literature for analyzing the ACC system with wide operating range of above mentioned manipulated, controlled and disturbance variables.

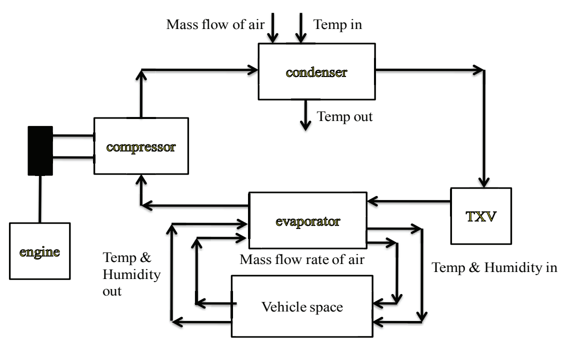

Figure 1. Automotive Air Conditioning System Block Diagram Representation

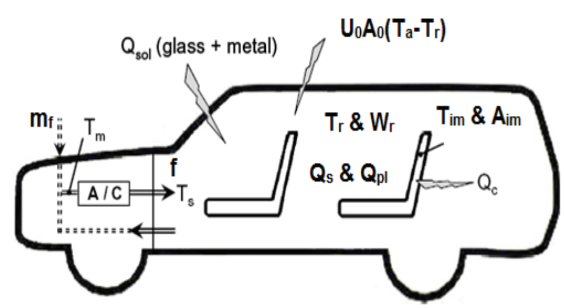

Figure 2. Physical Model of Passengers' Compartment and Thermal Loads

In this paper, the dynamic mathematical model for vehicle cabin, integrated with evaporator and air handling unit is developed. The developed model is converted to State space formulation for analyzing the AAC with wide changing operating range like compressor speed, ambient temperature and humidity, blower fan air flow rate, condenser air flow rate etc. This model show the variation of temperature and moisture in the vehicle cabin , evaporator supply temperature and moisture to the cabin , when the step change in the compressor speed and blower fan air flow rate is given. It is also expected that the SSDM can be useful in designing different control strategies like multi loop, multivariable control and model Predictive Control etc. to optimally control the system for improving passenger comfort, fuel economy, drivability and stalling of engine.

The rest of this paper is organized as follows. Section 2 gives details of the experimental automotive air conditioning system based on vapor compression refrigeration cycle. Section 3, presents the dynamic mathematical model for the experimental AAC. Section 4 presents the state space formulation from the dynamic model developed in section 3. SSDM is verified with experimental data in section 5 and finally conclusions are given in section 6.

2. EXPERIMENTAL VEHICLE AIR CONDITIONING:-DESCRIPTION

The experimental vehicle air conditioning system was mainly composed of two parts i.e a vapor compression refrigeration system (VCRS) namely compressor, condenser, Thermostatic expansion valve and evaporator) and an air distribution system (air side). Its simplified Physical vehicle model is shown fig. 2 as proposed by Gado et al., (2005). The evaporator of VCRS alone was placed inside the supply air duct to work as a evaporator- air cooling coil. The design air face velocity for the AAC cooling coil was 2m/s. The nominal output cooling capacity from the AAC refrigerant plant was 6 KW (1.5 RT). The working fluid of the plant was R134 a, with a total charge of 950 gms. Inside the space there are sensible heat and moisture load generating units. The vehicle experimental AAC system has been fully instrumented. High Precision sensors/ transducer were used for measuring and all operating parameter including temperature and flow rate of both refrigerant, pressure in the AAC unit. All the measurement were automized, so that all the measured data can be recorded for subsequent analysis.

3. DYNAMIC MODELING OF THE EXPERIMENTAL VEHICLE AAC

The vehicle cabin and the dynamics of the two phase flow heat exchangers or evaporator was derived based on the conservation of energy and mass balance principles by using set of simple, coupled ordinary differential equation. Several assumptions must be made in order to simplify these equations in to mathematically traceable form. These assumptions includes

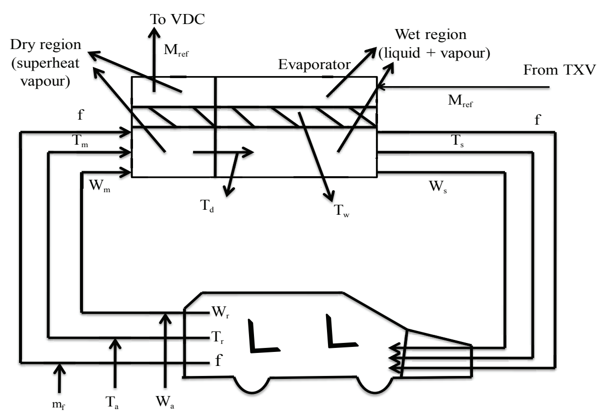

Figure 3. Block diagram of AAC – Evaporator Integrated with Vehicle Cabin

1. The plant is modeled as a first order linear dynamic system without transportation delay;
2. Only cooling load is considered for range between 18 to 45 °C;
3. Condensation and evaporation of refrigerant ends exactly on the liquid/vapor saturation lines;
4. Only lumped parameter is assumed for evaporator model and only saturated vapor is outlet;
5. No pressure drop across the components like evaporator and condenser;
6. Refrigerant oil circulation is neglected for simplicity;
7. According to Amr Gado et.al the coefficient of heat transfer does not change with vehicle speed;
8. Perfect air mixing inside cooling coil and vehicle cabin space;
9. Only two regions on the air side of the cooling coil is considered i.e., dry cooling region and wet cooling region;

10. Negligible thermal losses in air ducts.

The Energy balance equation of vehicle cabin for air sensible heat is given by

$$M_r C_{pr}\frac{dT_r}{dt} + M_{im} C_c \frac{dT_{im}}{dt} = -C_{pe}\rho f(T_m - T_s) + Q_s + Q_{ps} + U_0 A_0 (T_a - T_r) + m_f C_{pa} (T_a - T_r) + K_{spl} f \qquad - (1)$$

where K_{spl} is the heat gain constant for the blower fan . The six terms on the left side of the equation 1 represents the sensible cooling capacity of the evaporator, solar radiation from sun, passenger sensible radiation, solar radiation transmitted to the cabin and metal of the vehicle, heat transfer with the outside ambience, heat load from blower fan respectively. The equation 1 represents the transient variation in the cabin temperature, with change in compressor speed and blower fan speed.

The energy balance equation of vehicle interior cabin mass is

$$M_{im} C_c \frac{dT_{im}}{dt} = -h_{im} A_{im} (T_{im} - T_r) \qquad - (2)$$

The equation 2 represents the transient variation in temperature of the interior mass inside the cabin, with change in compressor speed and blower fan air flow rate.

The temperature and moisture content of the air entering evaporator were T_m and W_m resp. The air temperature decreases along the evaporator wall and was equal to T_d at the end of dry cooling region. The energy balance in the dry- cooling region on the air side of the evaporator is

$$C_p \rho V_{h1} \frac{dT_d}{dt} = C_p \rho f(T_m - T_d) + \propto_1 A_1 \left(T_w - \frac{T_m + T_d}{2}\right) \qquad - (3)$$

In dry cooling region, the evaporator wall temperature is assumed as same and only sensible cooling takes place. Where α_1 is the air side convective heat transfer coefficients for the dry cooling region of evaporator. The equation 3 represents the transient variation in the temperature of dry- cooling region on the air side of the evaporator as shown in fig 3.

The coupled air sensible cooling and dehumidification, in the enthalpy and energy balance form in the wet- cooling region, is given by

$$C_p \rho V_{h2} \frac{dT_s}{dt} + \rho V_{h2} h_{fg} \frac{dW_s}{dt} = C_p \rho f(T_d - T_s) + \rho f h_{fg} (W_m - W_g) + \propto_2 A_2 \left(T_w - \frac{T_d - T_s}{2}\right) \qquad - (4)$$

where α_2 is the air side convective heat transfers coefficients for the wet region evaporator. The equation 4 represents the transient variation in the temperature of the wet cooling region on the air side of the evaporator as shown in fig 3.

The mass flow rate of the variable displacement compressor used in VCRS of an AAC can be calculated by the following equation from Tian et al., (2006)

$$M_{ref} = n \times \frac{\pi}{4} \times D_c^{\,2} S_p \times N_c \times \frac{\eta_v}{\upsilon_s} \qquad - (5)$$

Where, n - Number of cylinder in the compressor, D_c - Diameter of the cylinder in m, S_p - Piston stroke in m,
N_c - Speed of the compressor in rps, η_v - Volumetric efficiency of the compressor in %,
υ_s - Refrigerant specific volume at the suction end of the compressor.

The equation for volumetric efficiency and isentropic efficiency are given in polynomial fit by experimental data provided by Tian et al., (2006),

$$\eta_v = 0.3596 + 1.1072\overline{S_p} - 0.08132\varepsilon + 0.0001175N_c - 0.4025\overline{S_p}^2 - 2.449 \times 10^{-8}{N_c}^2 \quad - (6)$$

$$\eta_i = 0.2402 + 1.4187\overline{S_p} - 0.09698\varepsilon + 0.000123N_c - 0.0585\overline{S_p}^2 - 2.457 \times 10^{-8}{N_c}^2 \quad - (7)$$

Where $\overline{S_p}$ is relative piston stroke length (PSL) or ratio of actual PSL and full PSL.

The dynamic response to change on the air side of the vehicle cabin was much slower than that on the refrigerant side, due to thermal inertia of air and refrigerant. When the air side of the vehicle cabin waited for a long time to fully respond, the refrigerant side was already in its steady state for a quite while. Thus the same refrigerant mass flow rate at both the inlet and the outlet of the evaporator was assumed.

The energy balance equation for the evaporator wall (T_w)

$$(C_p\rho V)_w \frac{dT_w}{dt} = \propto_1 A_1\left(\frac{T_m + T_d}{2} - T_w\right) + \propto_2 A_2\left(\frac{T_d + T_s}{2} - T_w\right) - M_{ref}(h_{r2} - h_{r1}) \quad - (8)$$

The equation 8 represents the transient variation in the temperature of the evaporator wall.

The latent heat energy balance for vehicle cabin air

$$M_r h_{fg}\frac{dW_r}{dt} = -\rho f h_{fg}(W_m - W_s) + M_f h_{fg}(W_a - W_r) + Q_{pl} \quad - (9)$$

Where h_{fg} is the latent heat vaporization of water in kJ/kg. The equation 9 represents the change in the latent heat energy or moisture content of the cabin over time with change in compressor speed and blower fan air flow rate. The relationship between air moisture content and temperature obtained by plotting curve fitting

$$\frac{dW_s}{dt} - \left(\frac{(2\times 0.0198T_s) + 0.085}{1000}\right)\frac{dT_s}{dt} = 0 \quad - (10)$$

The adiabatic mixing temperature (T_m) of outside air and recirculated air prior to cooling and dehumidification by the evaporator in the ACC module as shown in fig 3, is written in mass balance form as

$$\rho f C_{pm} T_m = m_f C_{pa} T_a + (\rho f - m_f) C_{pr} T_r \quad - (11)$$

where, C_{pm} - Specific heat at constant pressure of the mixture air to the evaporator inlet in kJ/kg-°K

The adiabatic mixing humidity (W_m) of outside air and recirculated air prior to cooling and dehumidification by the evaporator in the ACC module as shown in fig , is written in mass balance of water vapor form as

$$\rho f W_m = m_f W_a + (\rho f - m_f) W_r \quad - (12)$$

Equations 1, 2, 3, 4, 8, 9 and 10 all of which were coupled first order differential equations formed the mathematical model of the AAC.

4. STATE SPACE FORMULATION AND SOLUTION

The above differential equations 1, 2, 3, 4, 8, 9 and 10 were represented in state space approach in time domain method as suggested by (S. Menon et. al. 2004 and Qi Qi et. al. 2008) as

$$\dot{X} = D^{-1}.h_1(X,U) + D^{-1}.h_2(Z) \quad - (13)$$

Where the state variables $X = [T_r, T_{im}, T_d, T_s, T_w, W_r, W_s]^T$ and $\dot{X} = \frac{dX}{dt}$, the input variables $U = [N_c, f]^T$ and the disturbance variable $Z = [Q_s+Q_{pl}, Q_{pl}]^T$. h_1 and h_2 are the functions defined as follows:

$$h_1(x,u) = \begin{bmatrix} -C_{pe}\rho f(T_m - T_s) + U_0 A_0 (T_a - T_r) + C_{pa}\rho f(T_a - T_r) + K_{spl} f \\ -h_{im} A_{im} (T_{im} - T_r) \\ C_p \rho f(T_d - T_s) + \propto_1 A_1 \left(T_w - \frac{T_m + T_d}{2}\right) \\ C_p \rho f(T_d - T_s) + \rho f h_{fg} (W_m - W_s) + \propto_2 A_2 \left(T_w - \frac{T_d + T_s}{2}\right) \\ \propto_1 A_1 \left(\frac{T_m + T_d}{2} - T_w\right) + \propto_2 A_2 \left(\frac{T_d + T_s}{2} - T_w\right) - M_{ref}(h_{r2} - h_{r1}) \\ -\rho f h_{fg} (W_m - W_s) + M_f h_{fg} (W_a - W_r) + Q_{pl} \\ 0 \end{bmatrix} \quad \text{- (14.a)}, \quad h_2(z) = \begin{bmatrix} Q_s + Q_{ps} \\ 0 \\ 0 \\ 0 \\ 0 \\ Q_{pl} \\ 0 \end{bmatrix} \quad \text{- (14.b)}$$

$$D = \begin{bmatrix} M_r C_{pr} & M_{im} C_c & 0 & 0 & 0 & 0 & 0 \\ 0 & M_{im} C_c & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & C_p \rho V_{h1} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & C_p \rho V_{h2} & 0 & 0 & \rho V_{h2} h_{fg} \\ 0 & 0 & 0 & 0 & (C_p \rho V)_w & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & M_r h_{fg} & 0 \\ 0 & 0 & 0 & \left(\frac{-2 \times 0.0198 T_s + 0.085}{1000}\right) & 0 & 0 & 1 \end{bmatrix} \quad \text{- (14.c)}$$

The developed SSDM expressed in state space representation in equation 13, was non linear since the relationship state variables and input variables was non linear. AAC was designed to operate in a predetermined set point given that vehicle thermal space cooling load is constant. The controlled system can be represented by a linearized model around certain operating conditions, to regulate the transient deviation of the controlled objectives from the set point.

$$X = x + x^0 \quad \text{- (15.a)}, \qquad U = u + u^0 \quad \text{- (15.b)}$$

Where x^0 and u^0 are the state vector and input vector, both calculated at a steady state operating point, and x and u represent the small deviation from x^0 and u^0 resp. At steady state, the sensible and latent load disturbances are constant. The AAC dynamic deviation at an operating point in linearized form is

$$\dot{X} = \frac{\partial h_1}{\partial X} @(x^0, u^0) + \frac{\partial h_1}{\partial U} @(x^0, u^0) \text{ and } u = A(x^0, u^o)x + B(x^0, u^o)u \quad \text{- (16)}$$

The above equation in linearized form is combined in matrix format. This results in the state vector different equation as

$$\dot{x} = Ax + Bu \quad \text{- (17)}$$

Equation 17 is generally called the state equations, where lower case boldface represents vector and uppercase boldface represents the system matrix.

The output equation of the system can be related to the state variable and the input variable as

$$y = Cx \quad \text{- (18)}$$

where the output variable $y = [\delta T_r, \delta W_r]^T$, the dynamic deviation of cabin air temperature and moisture content from the set point. At the particular operating point, where T_r= 24 °C, W_r = 0.0113 kg/kg dry air, T_s = 13.25 °C and W_s = 0.009 kg/kg dry air and air flow rate, f = 0.115 m^3/s. The linearized model have the eigen value with negative real parts, proves the system is asymptotically stable.

PARAMETERS	NUMERICAL VALUES
M_r	9.16 Kg
M_{im}	200 Kg
C_{pr}, C_c, C_{pe}, C_{pa}, C_{pm}, C_p	1.005 KJ/kg
U_0	4 W/m²K
A_0	30 m²
P	1.2 Kg/m ³
V_{h1}	0.004 m³
V_{h2}	0.016 m³
A_1	0.53 m²
A_2	4.263 m²
h_{fg}	2450 KJ/kg
V	8m³
H_{im}	100 W/m²K
A_{im}	3 m²

Table 1. Numerical parameters

PARAMETERS	MINIMUM	MAXIMUM
α_1	0.025 KW/ m² ºC	0.225 KW/ m² ºC
α_2	0.017 KW/ m² ºC	0.065 KW/ m² ºC
h_{r1}	233.5 kJ/kg	324.5 kJ/kg
h_{r2}	393.8 kJ/kg	401.8 kJ/kg
P_s	2.17bar	3.55bar
P_d	6.5bar	27bar
V_s	0.093	0.057
S_p	9mm	28.7mm
$\bar{S}_p$	0.3	1

Table 2. Operating Condition of the AAC System

5. SSDM VERIFICATION

Vehicle air conditioning experimental data have been compared with linearized dynamic model, for the purpose of model validation. The simulation and experimental results were the open loop response to step change in compressor and blower fan air flow rate, resp. When the system was operating around a steady state condition, step changes were introduced to the manipulated variables such as compressor speed and blower fan air flow rate. The following comparisons were based on the steady state operational condition of around 24 °C vehicle cabin temperature and 11.3 g/kg moisture content. The numerical values of both the system parameters used in the simulation and the operating condition of the AAC are given in Table 1 and 2 respectively. The linearization of the model was also based on this operating condition. The fig 4 - 7 presents the comparisons between the simulation results and vehicle experimental data in the response to a step change in compressor speed from 1200 to 1600 rpm, introduced at 300 s. When the compressor speed increased, the temperature (T_r) and moisture (W_r) content of air inside the cabin decreases due to the increased output cooling capacity of the vehicle air conditioning. As seen in fig. 4 – 7 there exist a good agreement between simulated response and the experimental results.

Similar observation can be obtained from fig. 8 - 11 for other operating parameters such as the temperature (T_s) and moisture (W_s) content of air leaving the evaporator coil to vehicle cabin. A comparison between simulated result and experimental data in response to step change in blower air flow rate from 0.150 m^3/s to 0.120 m^3/s, is introduced at 320 second are illustrated in fig. 8 – 11. When the blower air flow rate was given step reduction, the cabin temperature (T_r) gets slightly increased due to sensible load and the cabin moisture (W_r) content decreases slightly. But, the supply air temperature and moisture to cabin reduces steeply due to reduction in air flow rate as shown in fig. 10 & 11.

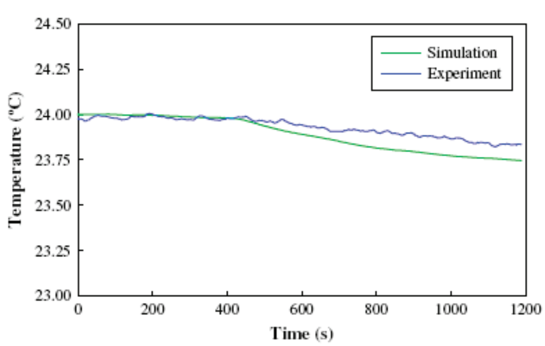

Fig. 4 Response to simulated and measured vehicle cabin temperature (T_r) for step input in compressor speed

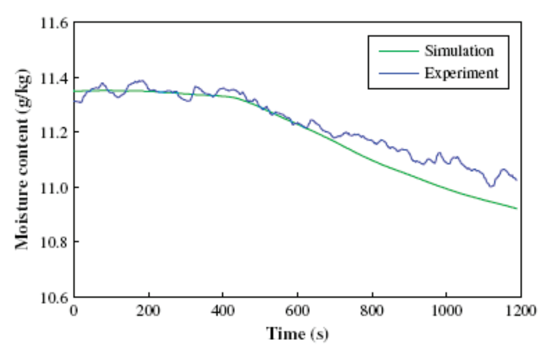

Fig. 5 Response to simulated and measured vehicle cabin moisture (W_r) content for step input in compressor speed

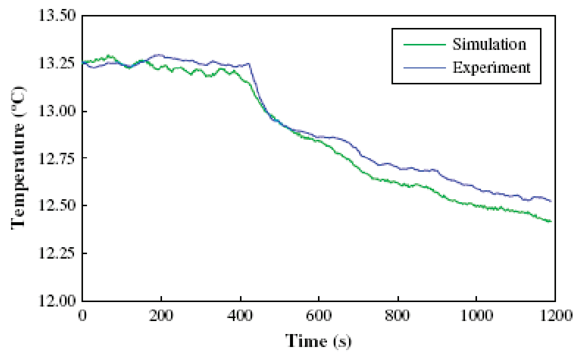

Fig. 6 Response to simulated and measured evaporator supply temperature (T_s) for step input in compressor speed

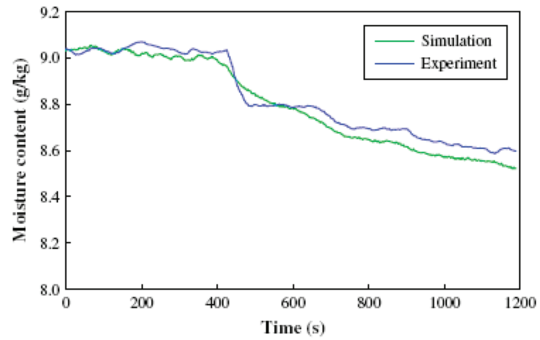

Fig. 7 Response to simulated and measured evaporator supply moisture (W_s) content for step input in compressor speed

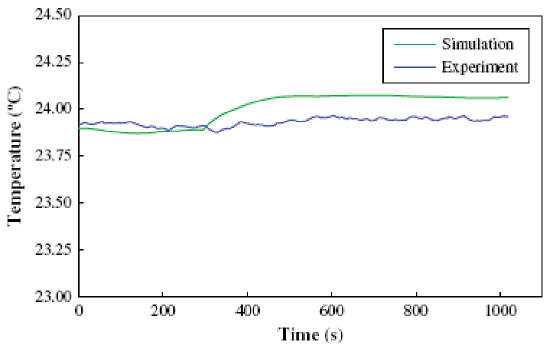

Fig. 8. Response to simulated and measured vehicle cabin temperature (T_r) for step input in blower fan air flow rate

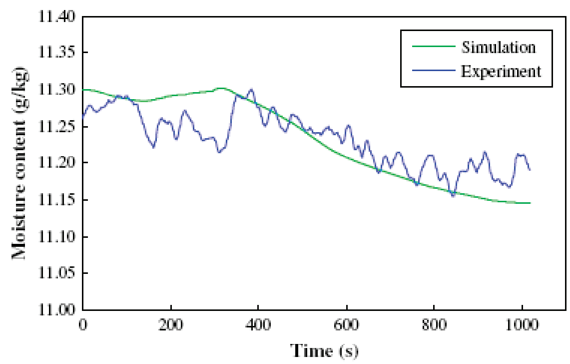

Fig. 9. Response to simulated and measured vehicle cabin moisture (W_r) content for step input in blower fan air flow rate

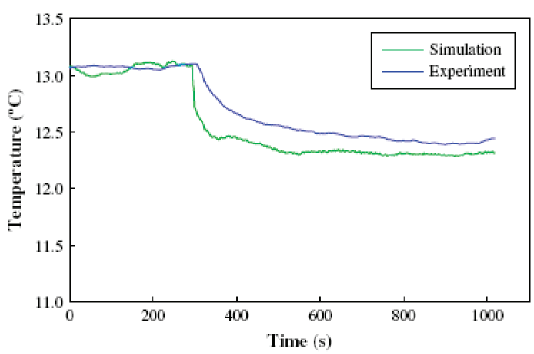

Fig. 10 Response to simulated and measured evaporator supply temperature (T_s) for step input in blower fan air flow rate

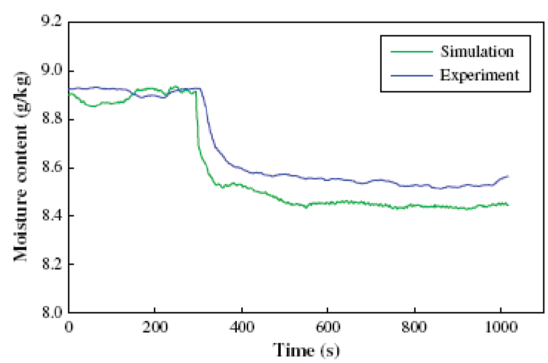

Fig. 11 Response to simulated and measured evaporator supply moisture (W_s) content for step input in blower fan air flow rate

The developed SSDM after linearization about certain operating point was experimentally valid to capture the transient and steady state change of system parameters in time domain with acceptable accuracy as shown in fig. 4 – 11. Since the model used was a simplified one, there exists a difference between the simulated and measured response.

6. CONCLUSIONS

A dynamic model for AAC with vehicle cabin, VDC compressor and evaporator is developed based on conservation of energy and mass balance principle. The model was represented in state space form. It was linearized at its operating point to analyze the system parameters like vehicle cabin temperature and humidity, evaporator supply temperature and humidity for change in compressor speed and blower fan air flow rate. The linearized SSDM was experimentally valid with acceptable accuracy. The SSDM developed can be useful in developing different control scheme like multiloop and multivariable control, MPC, etc. to optimally control the system parameters for improving passenger comfort, fuel economy, drivability and stalling of engines.

NOMENCLATURE

T_r - Temp of the room (or) inside the vehicle cabin in ºC	A_{im} - Area of the interior mass (or) Core in m²
T_{im} - Temp of the interior mass inside the cabin in ºC	C_p - Specific heat of air
T_m - Temp of the mixture air to the evaporator inlet in ºC	ρ - Density of the moisture air in kg/ m³
T_a - Temp of the ambient air in ºC	V_{h1} - Air side volume of the evaporator core in dry-cooling region in m³
T_s - Temp of the supply air to the vehicle cabin in ºC	α_1 - Heat transfer coefficient between air and evaporator wall in dry-cooling region in KW/ m² ºC
T_d - Air temp leaving the dry-cooling region (or) superheated region of the evaporator core cooling coil in ºC	A_1 - Heat transfer area of the evaporator core (or) cooling coil in dry – cooling region in m²
T_w - Evaporator surface (or) Evaporator wall temperature in ºC	V_{h2} - Air side volume of the evaporator core in wet-cooling region in m³
M_r - Mass of the air inside the vehicle cabin in kg	W_m - moisture content of the mixture air to the evaporator inlet in kg/ kg- dry air
M_{im} - Mass of the internal mass inside the cabin in kg	W_s - moisture content of the supply air to the vehicle cabin in kg/ kg- dry air
m_f - Mass flow rate of infiltration and/or ventilation air in kg/s	α_2 - Heat transfer coefficient between air and evaporator wall in wet-cooling region in KW/ m² ºC
C_{pr} - Specific heat at constant pressure at room in kJ/kg K	A_2 - Heat transfer area of the evaporator core (or) cooling coil in wet – cooling region in m²
C_{pe} - Specific heat at constant pressure at evaporator in kJ/kg K	h_{fg} - Latent heat of vaporization of water in kJ/kg
C_c - Specific heat at constant pressure at interior mass (or) core in kJ/kg K	V - Volume of the vehicle cabin in m³
C_{pa}, C_{pm} - Specific heat at constant pressure at ambient and mixing point before evaporator in kJ/kg K	M_{ref} - Mass flow rate of refrigerant into evaporator core in kg/s
Q_s - Heat load of solar radiation in KW	h_{r1} - Enthalpy of refrigerant at the evaporator core (or) cooling coil inlet in kJ/kg
Q_{ps} - Sensible heat load of passenger in KW	h_{r2} - Enthalpy of refrigerant at the evaporator core (or) cooling coil outlet in kJ/kg
U_o - Overall heat transfer co-efficient of vehicle cabin wall in W/ m² K	W_r - Moisture content of the air inside the vehicle cabin in kg/ kg- dry air
A_o - Surface of the vehicle cabin in m²	W_a - Moisture content of the ambient air in kg/ kg- dry air
K_{spl} - Heat gain co-efficient of supply fan in KJ/ m³	Q_{pl} - Latent heat load of passenger in KW
f - Volumetric air flow rate in m³/s	
h_{im} - Convective heat transfer coefficient between interior mass and cabin air in KW/ m² K	

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