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Mingyu Wang, John L. Pawlak and Charles A. Archibald
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Thermal Systems, Delphi Corporation

ABSTRACT

The majority of today's Automatic Climate Control (ACC) systems are based on linear analog controller technology which was developed in the mid of last century. In cognizance of the state of the ACC technology, we developed a new ACC system based on the concepts of power balance and the separation of transient and steady state control. Production implementations have demonstrated that the new ACC system technology improves overall control performance, and reduces program development time and the associated cost.

INTRODUCTION

The first vehicle Automatic Climate Control (ACC) system was introduced in the 1964 Cadillac after several years of joint development activities among General Motors Engineering Staff and several General Motors divisions including Delco Radio Division (now Delphi Electronics and Safety), Cadillac Motor Car Division and Harrison Radiator (now Delphi Thermal). The system was designed to meet the following three performance requirements [1]:

1. A comfortable passenger compartment climate should be maintained under all weather and engine operating conditions.
2. The driver should be relieved of adjusting controls. He should be able to select his comfort level once and have it automatically repeated during every usage without any attention to controls. It is, of course, realized that changes in body metabolism and dress will at times make a different in-car temperature desirable. In this case only a slight adjustment should be required.
3. The desired in-car temperature should be attained rapidly.

Even though the 1964 Cadillac ACC systems used analog controllers with vacuum actuators, the technology paradigms espoused by the revolutionary ACC system design are still followed today. These include the sensor set selection (ambient temperature thermistor, in-car air temperature thermistor, discharge air temperature thermistor), the principle of air-mixing through the use of a temperature door in the HVAC module to achieve any desired discharge temperature, the concept of solar compensation, the linear combination of the sensor inputs (ambient temperature, in-car temperature, and driver set temperature through a potentiometer, hereon referred to as Linear Load Equation) to drive the actuation mechanism, etc.

With the maturation of the micro-controller devices, the ACC system made the transition from linear analog amplifiers and vacuum actuators to micro-controllers and electrical motor actuators. This made it possible to implement complex control strategies to achieve comfort and safety objectives. New capabilities such as cold and hot purging, snow ingestion control, among others, were introduced into automatic climate control.

Significant as it was, the linear-analog based control, including its digital incarnation, seems to be showing its age. An Illinois Technical Institute administered usage study [2] indicated that on average, customers overrode the automatic climate control systems and operated in the manual mode for 66% of the driving time. Allison-Fisher Advanced Automotive Features Study shows that automatic climate control is the least desired HVAC feature (vs. standard AC, instant heat, left-right individual control, air filtration, etc.) across all vehicle segments [3].

It is also known that the linear-analog based control takes great amount of resources for acceptable calibration in each vehicle program execution. The calibration activities normally start with the prototype build, and it is not unusual for the calibration activities to extend beyond the launch of a vehicle into the market place.

Against this backdrop, the Delphi Thermal's automatic climate control team embarked on the development of an entirely new ACC system that incorporates the latest thinking in the industry. The development objectives include: (1) improved transient and steady state control performance; (2) increased ACC system tunability; (3) reduced calibration effort; (4) simplified ACC software specification; and (5) streamlined software development process.

After three years of extensive effort, it can be said that these development objectives have been largely met: The steady state performance of the ACC system is improved by relying on the proper balance of the passenger compartment thermal load and the HVAC air conditioning capacity. This approach recognizes the inherent non-linearity of typical blower and temperature maps and shuns the traditional linear control concept; The transient ACC performance is improved by the introduction of the tunability and transient-steady state separation concepts; It's been demonstrated that with the new ACC algorithm, a prototype vehicle with acceptable performance level can be calibrated within four to six weeks after the low level motor actuation software are delivered and accepted from the suppliers; Through the use of the modeling software MatLab, the entire ACC high-level algorithm is represented in a multi-layered graphical model, with more details added to each layer in the sequence. Last but not least, the micro-controller executable code can be generated directly using the auto-code function of the MatLab software from the graphical model. Modifications to the model can be performed at the graphical level and quickly implemented on the microprocessor through auto-code generation. This greatly simplifies the software development process.

The present paper covers three areas of the ACC algorithm development: (1) overall algorithm architecture, (2) software development process, and (3) system calibration of vehicle.

STEADY STATE CONTROL ALGORITHM DESIGN

The operation of an Automatic Climate Control system can be divided into the transient and the steady state phases of operation. The objective of the transient phase control is to take a soaked passenger compartment quickly away from the initial condition of discomfort to a more rider friendly condition. Once there, the control objective is to maintain the comfort condition as the ambient temperature or solar loading go through various changes. The latter part of the ACC operation is commonly characterized as the steady state operation.

BASIC CONTROL EQUATIONS

The basic energy balance of the passenger compartment at steady state can be expressed by Equation (1) as provided by the principle of energy conservation,

$$\dot{m}c_p(T_d - T_s) = UA(T_s - T_a) + UA_{dp}(T_s - T_{dp}) - \dot{Q}_{slr} \qquad (1)$$

Specifically, the right side of Equation (1) represents the thermal load of the passenger compartment on the HVAC system, while the left side represents the cooling or heating power supplied by the HVAC system. Given the ambient temperature T_a, solar load $\dot{Q}_{slr}$ and the vehicle passenger compartment internal mass (aka deep mass) temperature T_{dp}, the amount of HVAC power as represented by the left side of Equation (1) must be supplied in order to be able to maintain the desired comfort temperature level of T_s (set temperature).

It can be seen that the HVAC power is the product of the HVAC discharge airflow rate and its enthalpy change experienced in the passenger compartment, as is given in Equation (2),

$$PWR_{hvac} = \dot{m}c_p(T_d - T_s) \qquad (2)$$

where $\dot{m}$ is the discharge airflow rate, c_p is the specific heat of air, T_d the discharge temperature and PWR_{hvac} the power extracted from the HVAC system.

The thermal load from the passenger compartment is composed of three terms. The first term represents the thermal load attributed to the ambient. It is caused by the convective heat transfer that occurs at the exterior of the passenger compartment. It is proportional to the temperature difference between the passenger compartment interior and the ambient air, and inversely proportional to the thermal resistance of the passenger compartment enclosure. The heat transfer resistance is hereby represented by its inverse, the thermal conductance of the passenger compartment enclosure, *UA*.

The second term, $UA_{dp}(T_s - T_{dp})$, represents the amount of additional thermal load from the passenger compartment interior mass such as seats and floor mass, together referred to as the passenger compartment "deep mass". When accounted for, the precision of the ACC control can be substantially improved. To calculate this thermal load, a representative deep mass temperature T_{dp} must be measured or modeled. The interior surface heat transfer conductance UA_{dp} can be determined experimentally.

The third term in the thermal load equation is the net solar radiation into the passenger compartment. The solar heating, $\dot{Q}_{slr}$, can be measured through the use of solar sensors.

If we account for all the thermal load of the passenger compartment by,

$$PWR_{req} = UA(T_s - T_a) + UA_{dp}(T_s - T_{dp}) - \dot{Q}_{slr} \qquad (3)$$

where the subscript "*req*" stands for "required" power due to the thermal load, the energy balance of the passenger compartment can be simplified to:

$$PWR_{req} = PWR_{hvac} \qquad (4)$$

DETERMINATION OF CONTROL PARAMETERS

The energy balance as outlined above must be translated into control parameters such as the HVAC discharge temperature and the discharge airflow rate for the server mechanism to actuate the proper flow control and temperature control devices in the HVAC module. This is accomplished through a real time HVAC power curve calculation and the reverse power lookup.

The real time HVAC power curve calculation begins with the definition of the discharge temperature and blower speed maps. This is illustrated in the top part of Fig. 1, where blower speed is translated to mass flow rate of the discharging air. The abscissa is referred to as the Power Index, and can be approximately correlated with the ambient air temperature. The Power Index has a range of 0~255, corresponding to a temperature range of about –40 °C and 65 °C, with the high temperature corresponding to low Power Index number and low temperature corresponding to high Power Index number. Apparently, at high ambient temperatures, high blower speeds and low discharge temperatures are required, while at very low ambient temperatures, high blower speeds and high discharge temperatures are required. At moderate ambient temperatures, blower speed can be reduced. The maps can be refined as the vehicle development process evolves.

With the blower speed map and the discharge temperature map defined, one can calculate an HVAC Power Map given a particular passenger compartment set temperature by using Equation (2). This is illustrated in the lower part of Fig. 1. Since this map is dependent on the passenger compartment set temperature, which is adjustable by design, it needs to be recalculated each time the set temperature is changed. Fig. 1 shows three different curves corresponding to three different set temperatures.

The HVAC power map is represented in a microprocessor as a two dimensional table with the HVAC power as the independent variable and the HVAC Power Index as the dependent variable. When a thermal load from the passenger compartment is known, the HVAC Power Index can be directly looked up from the table and subsequently used with the blower speed map and the discharge temperature map to determine the HVAC system control parameters. This is referred to as the "reverse power look up" process.

The next step in the determination of the HVAC control parameter is the calculation of the thermal load from the passenger compartment. This can be easily calculated by using Equation (3) if the ambient thermal conductance is properly mapped as a function of the ambient temperature, vehicle speed and possibly the internal air speed. Lacking a good theoretical model, such a map can be generated through tunnel testing.

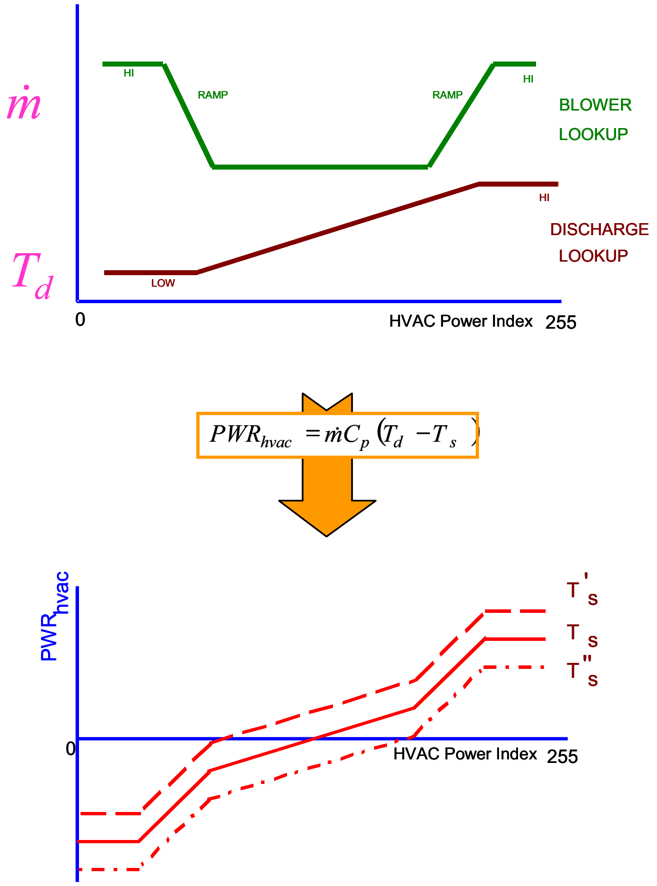

Fig. 1 Steady State Control Process

In order to be able to assess the thermal load contribution from the ambient, the functional relationship is simplified by considering the conductance a pure function of the ambient temperature. Also, instead of trying to determine the ambient thermal conductance as a standalone parameter, the ambient thermal load (heat transfer rate) was determined. The ambient thermal load then can be represented as PWR_{amb}. A look-up table for PWR_{amb} can be established through tunnel testing.

The PWR_{amb} thus determined is a function of the ambient in the following form:

$$PWR_{amb} = F(T_a) = \overline{F}(24 - T_a) \tag{5}$$

It is important to recognize that PWR_{amb} is in fact a function of the temperature difference between the set temperature of 24 °C and the ambient temperature. Use can be made of it to simplify the control equation when the passenger compartment set temperature is different from the standard condition of 24 °C.

With the thermal load determined and the HVAC power map established, we can now easily implement the control equations into a control algorithm by using the "reverse power lookup" process. As is shown in the lower part of Fig. 1, we first determine the HVAC Power Index by using the PWR_{req} given by Equation (3). Thereafter, the Power Index is applied to the blower speed and discharge temperature maps to obtain the control target parameters.

HANDLING OF PASSENGER COMPARTMENT SET TEMPERATURE CHANGE

The control algorithm outlined above deals with the set temperature change with simplicity and effectiveness. On the thermal load side, the impact of the set temperature change resulting from the deep mass term can be explicitly calculated by Equation (3). The impact on the ambient thermal load PWR_{amb} can be straight-forwardly accounted for by first calculating the temperature difference between the set temperature and the ambient temperature,

$$\delta T = T_s - T_a \tag{6}$$

and subsequently using Equation (7) to calculate the ambient thermal load:

$$PWR_{amb} = \overline{F}(\delta T) \tag{7}$$

On the HVAC power supply side, the set temperature change is effected through the real time power map update. As a result, no special provision is required to allow set temperature change.

SEPARATION OF TRANSIENT AND STEADY STATE CONTROL

A key design constraint of the Linear Load Equation based transient ACC control is the entanglement of the transient with the steady state control. In fact, the steady state control is the foundation of the transient operation. One cannot change the steady state control map without affecting the transient operation. Additionally, the path from the initial transient to the final steady state is restrictively defined by the blower curve and the discharge temperature curve. The transient process has to follow these two curves, which by no means are the optimal path to the steady state. Little "control" is given to the controls engineer whose responsibility is to fine-tune the ACC control characteristics for optimized comfort.

In the present ACC control system, a new transient control methodology is defined to allow the separation of the transient and steady state operation. It offers flexible tuning parameters that allow the controls engineer to fine-tune the control characteristics of the transient control process.

TRANSIENT INDICATOR

To effect efficient and accurate control of the transient process, an indicator is needed to identify the progress of the transient control. The following non-dimensional parameter based on the cabin breath temperature has been defined and proven to be a good transient indicator:

$$\theta = \frac{T_i - T_s}{T_{i,0} - T_s} \tag{8}$$

where T_i is the cabin temperature, $T_{i,0}$ is the initial cabin temperature after soak and T_s is the passenger compartment set temperature. At time zero when the transient cooling or heating process is just started, θ has the value of 1.0, while at the end of the transient process when the cabin temperature has attained the preset temperature of T_s, θ takes the value of 0. Therefore, the value of θ itself is an indicator of how far away the passenger compartment is from the steady state control condition.

STEADY STATE CONTROL PARAMETERS

Given the ambient temperature, solar load condition (I_{slr}), and the passenger compartment control temperature target T_s, the steady state HVAC control parameters such as the blower speed, discharge temperature, air distribution mode, etc, can be determined by using the aforementioned reverse power lookup:

$$PWM_\infty = f(T_a, I_{slr}, T_s) \tag{9.1}$$

$$T_{dis,\infty} = g(T_a, I_{slr}, T_s) \tag{9.2}$$

TRANSIENT CONTROL HVAC POWER VARIATION

The steady state control parameters are designed to maintain comfort after the passenger compartment has achieved the preset control temperature. The amount of HVAC power provided to the passenger compartment is inadequate to take the passenger compartment from the initially soaked high or low temperature to the target temperature within a reasonable amount time such that it meets the OEM's specification.

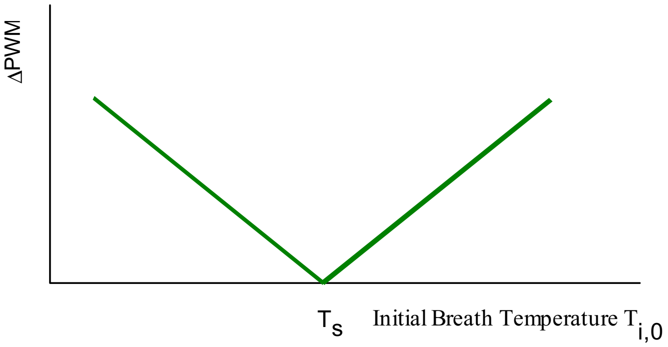

Fig. 2 Initial Blower Speed Variation

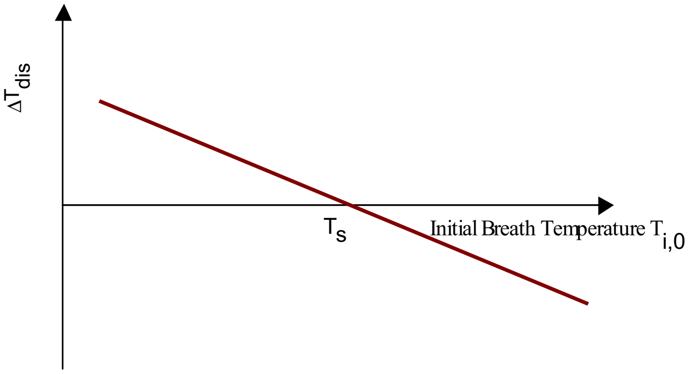

Fig. 3 Initial Discharge Temperature Variation

In order to achieve comfort within a reasonable amount of time, additional HVAC power is needed to accelerate the air conditioning process. In the beginning of the transient control, a significant amount of HVAC power above the steady state is used. As the passenger compartment temperature approaches the preset temperature, the amount of additional HVAC power is brought down to zero, such that only the steady state HVAC power is applied. The sliding of the HVAC power from the initial power down to the steady state power can be accomplished through proper interpolation between the initial power state and the final steady state.

Specifically, the initial HVAC power enhancement can be represented by blower speed and discharge temperature variations from the steady state:

$$\Delta PWM_0 = F(T_{i,0}) \tag{10.1}$$

$$\Delta T_{dis,0} = G(T_{i,0}) \tag{10.2}$$

As is seen in Equations (10.1) and (10.2), the initial blower speed and discharge temperature variations are functions of the initial soaked passenger compartment temperature.

Figures 2 and 3 show a representative set of curves that can be used to establish the initial HVAC power supply. The specific shapes of the curves need to be determined in the ACC development process by giving due considerations to the requirements of accelerated transient control and riders' tolerance for noise, vibration and harshness.

The initial blower speed and discharge temperature is now represented as:

$$PWM_0 = PWM_\infty + \Delta PWM_0 \tag{11.1}$$

$$T_{dis,0} = T_{dis,\infty} + \Delta T_{dis,0} \tag{11.2}$$

We need to bear in mind that this pair of control parameter settings provide the HVAC power for the beginning of the cool-down or warm-up after soak. As the passenger compartment temperature approaches the set temperature, the HVAC power needs to be gradually scaled back to the steady state HVAC power. This can be done by using an interpolation parameter with a range of 0 to1 that corresponds to the progress of the transient process. Designating this parameter as α, which has the value of 1 at the beginning of the transient process and a value of zero at the end, the HVAC control parameter at any time during the transient control can be given by the following interpolation,

$$PWM = \alpha PWM_0 + (1-\alpha) PWM_\infty \tag{12.1}$$

$$T_{dis} = \alpha T_{dis,0} + (1-\alpha) T_{dis,\infty} \tag{12.2}$$

or simplified as,

$$PWM = PWM_\infty + \alpha \cdot \Delta PWM_0 \tag{13.1}$$

$$T_{dis} = T_{dis,\infty} + \alpha \cdot \Delta T_{dis,0} \qquad (13.2)$$

It is to be noted that Equations (12.1) and (12.2) can be used directly as a transient control scheme if the initial blower speed and discharge temperature are specified according to the initial soaked passenger compartment temperature.

This interpolation parameter, or blending function, can be associated with the non-dimensional passenger compartment temperature θ defined earlier to perform the basic function of the HVAC control parameter interpolation. However, such a simple association provides scant opportunity to the controls engineer for fine-tuning the transient control process.

BLENDING FUNCTION

Instead, the blending function α is designed to be a nonlinear function of θ. There are two basic requirements for such a function. First of all, the function needs to be a monotonic function of θ to ensure one-to-one mapping of θ to α. Secondly, the end values of such a function must coincide with that of θ, i.e.,

$$\alpha = 0 \text{ at } \theta = 0 \qquad (14.1)$$

$$\alpha = 1 \text{ at } \theta = 1 \qquad (14.2)$$

There are numerous functions that meet these simple requirements. One such function is given as,

$$\alpha = \theta^n \qquad (15)$$

As is shown in Fig. 4, this blending function allows one degree of calibration control for the transient process. The controls engineer can choose different exponents to impose different paths to the steady state. As θ changes from 1 to 0, an exponent that is less than one allows a high value of α for most of the control process, allowing high HVAC power to be used during the initial transient process. On the other hand, an exponent higher than one allows the HVAC power to drop very quickly to the steady state level.

A slightly more complex blending function can be formed that offers two degrees of calibration control. The controls engineer will have two parameters to manipulate the path to steady state. In addition to the exponent as a calibration parameter, a “threshold” value is defined for θ. For θ greater than the threshold value, the blower speed and discharge temperature are maintained at a high level until the θ drops below it. The following pseudo-code describes the new blending function:

```
Function $\alpha(\theta, \theta_{cr}, n)$
If ( $\theta \geq \theta_{cr}$ ) then
    $\alpha = 1$
Else
    $\alpha = \left(\frac{\theta}{\theta_{cr}}\right)^n$
End if
End Function
```

In the pseudo-code, θ_{cr} represents the threshold value (or critical value) of the non-dimensional temperature θ.

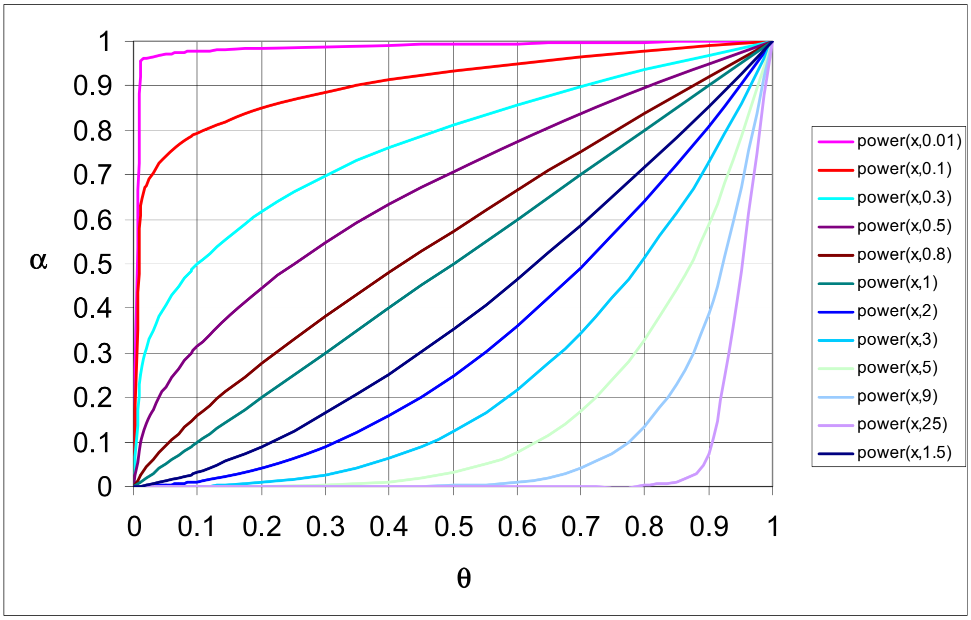

Fig. 4 Simple Blending Function

SOFTWARE DEVELOPMENT

In the development of the Delphi ACC system, sound software engineering principles and the latest code development methods were embraced to achieve efficiency and accuracy. The software is partitioned into logical building blocks that provide for modular coding structure, easy system maintenance, portability, and testability. Graphical modeling tools are used to produce executable specification that eliminates ambiguity and enhances upfront engineering. Auto code generation is employed to translate graphical system models into C code that can be directly compiled and burned into a microprocessor.

An ACC software strategy was developed based on the best practices of the PC software industry. As is well known, the PC software industry uses a multi-layered approach to achieve separation of hardware and software development. Low-level software or BIOS (Basic Input Output System) code are supplied by the hardware suppliers to command actions from a particular piece of hardware such as a printer or a CD-ROM drive. The BIOS has an "*Application Programming Interface*" (API), which defines how high level applications communicate with the BIOS to have mundane items such as writing to hard disks handled. Beyond the BIOS, the Windows Operating System provides additional API definitions, which act as middlemen between the BIOS and the application. At the application level, software developers can develop sophisticated computer programs to meet the end user's need by simply adhering to the stipulations of the API.

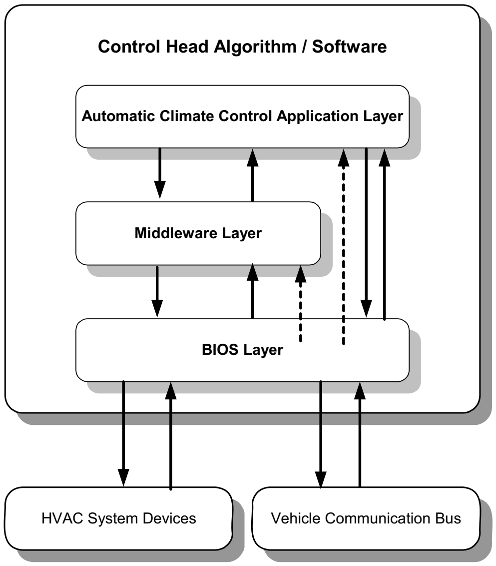

Fig. 5 Software Layer Model

Similar approach was used in the development of the Delphi ACC system. Fig. 5 shows that the ACC system is partitioned into three main sections. The "Automatic Climate Control Application Layer" contains all the logic that determines the system targets to control to in order to achieve the customer's desired cabin temperature. This layer was developed using MatLab, Simulink, Stateflow and MathWorks auto code tools. The "Middle Ware Layer" determines how to make adjustments to the system control devices such that the desired cabin temperature is achieved. The BIOS layer functions include driving system actuators, reading sensors, vehicle bus communications, task scheduling, driving displays and indicators, reading pushbuttons and knobs, power moding, and data conversion to meet API definitions.

SYSTEM DEVELOPMENT AND VALIDATION

WIND TUNNEL DEVELOPMENT

A climatic wind tunnel is used to determine the steady-state HVAC power necessary to offset the effect of the temperature difference between the inside of the cabin and the ambient, and the effect of the cabin thermal mass on the total HVAC power. The climatic tunnel can also be used to provide preliminary calibration for transient control variables. The minimum tunnel requirements to perform this testing are –20 to +40 °C temperature range and a wind speed of at least 100 KPH. The ideal climatic tunnel is one capable of –40 to +50 °C temperature range with a wind speed of 160 KPH.

Once the Ambient Power table is determined, initial development of transient control can be started. A system control test is performed by first soaking the vehicle at rest to a specified ambient temperature, and subsequently running the vehicle with the ACC turned on as in normal system operation. This testing exercises the transient portion of ACC control algorithm and allows the engineer to adjust the magnitude of the initial blower level and discharge temperature as well as the duration of the initial level and decay rate to the steady-state operating point. Although it is not absolutely required, it is recommended that the calibration or system engineer ride in the vehicle to subjectively balance the trade-off between the rate of warm-up or cool-down and system noise. System control tests should be performed over a wide range of ambient temperatures, with special attention given to the use of the adjustable transient features provided by this system. These features are the mass flow and discharge temperature deltas, dwells, and exponents that allow the system development engineer to tune the system's transient behavior.

ROAD DEVELOPMENT

Testing performed in the climatic tunnel provides calibration for steady-state and transient conditions without solar load. Road testing is used to develop solar load calibration and to fine-tune the calibrations obtained in the climatic tunnel.

Some solar testing may be done in the tunnel with solar simulation panels. However due to the difficulty in simulating low solar altitude angles, cloud cover, and the exact solar spectrum, solar calibration is best done on the road.

Road testing also provides an opportunity to fine-tune the transient behavior of the vehicle. Soaking the car outside in all climatic conditions allows the engineer to evaluate the trade-offs between time-to-comfort, noise, and overall system harshness. Since the ACC algorithm allows adjustment to transient control parameters based on cabin soak conditions, it is possible to make adjustments at a specific condition with no effect on other soak conditions. As an example, it is possible to change the initial level of the blower at a 50 °C soak without changing the initial levels at 45 or 55 °C. This flexibility permits the calibration engineer to develop the system that best suits customers' comfort preferences.

SYSTEM PERFORMANCE

The ACC control algorithm described herein has undergone rigorous software engineering development and has been implemented in several production vehicle platforms. In the following, several sets of data from an early development vehicle are shown to illustrate the performance characteristics of the present ACC control algorithm.

Fig. 6 shows a set of control data for a vehicle tested every 10 °C between –30 to +50 °C. The vehicle was soaked uniformly at each ambient and was then operated for one hour at 80 kph following the soak period. Examining Fig. 6, it can be seen that the system did a commendable job of meeting the target set temperature over a very wide range of ambient temperatures.

In addition to the one hour run, it is important that the system can maintain control for an extended drive. Fig. 7 illustrates two warm-up tests where the vehicle was operated for two hours. The ambient condition of –5 °C was chosen to demonstrate control capability in heater mode on a typical winter day when concern for system drift is present. The second ambient condition of 15 °C was chosen because it is a temperature where the system will start in heater mode and then transition through Bi-Level mode during the two-hour run. Both tests show excellent system stability.

Another aspect of automatic climate control is being able to change the set point. All systems are capable of controlling cooler or warmer with a set point change, but due to the HVAC power control strategy and transient operation, this system can accurately change control to a new operating point.

Fig. 8 illustrates a one-hour warm-up with a 24 °C set point. After achieving the target, the set point was changed to 26 °C at 60 minutes. The system responded and met the new target. At 75 minutes, the set point was changed by 4 to 22 °C. Again, despite the larger delta, the system converged on the new setting. This allows the development engineer excellent control in developing the vehicle's climate control characteristics. Often, it is desired to change the control point, without changing set temperature, based on solar load, ambient temperature, or length of soak. Since this system is very capable of meeting the target control point it is possible to have a dynamic control point the changes based on sun load (e.g. cooler set with high sun load) or other OEM defined parameter.

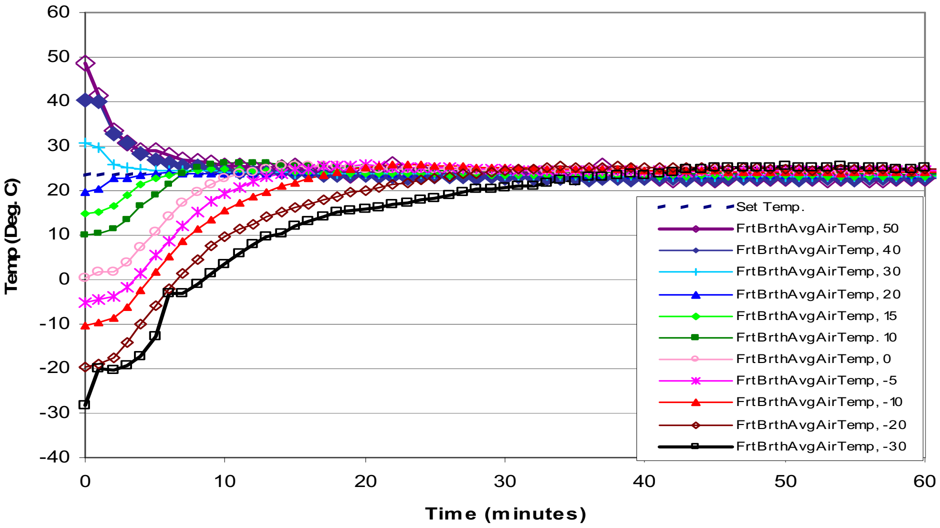

Fig. 6 Control Test Summary

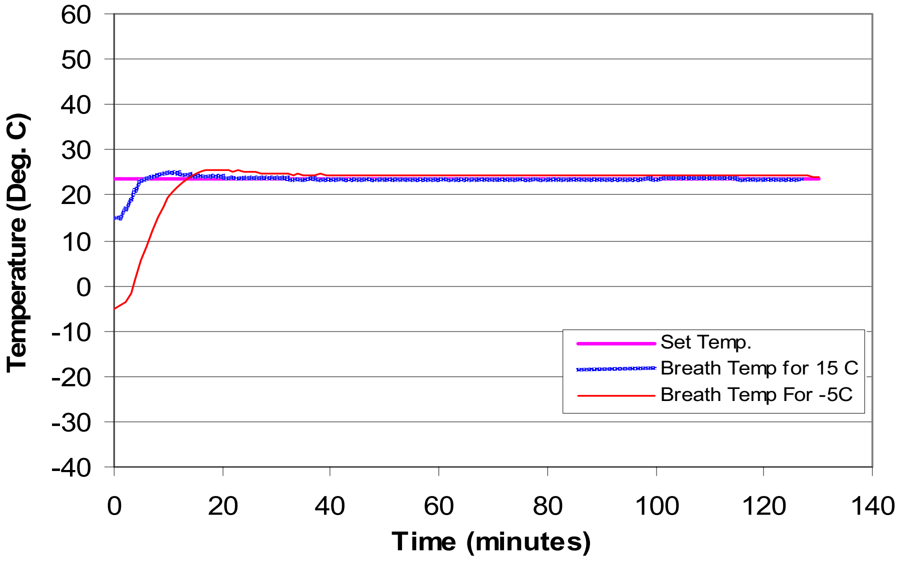

Fig. 7 System Stability Validation During Extended Drives

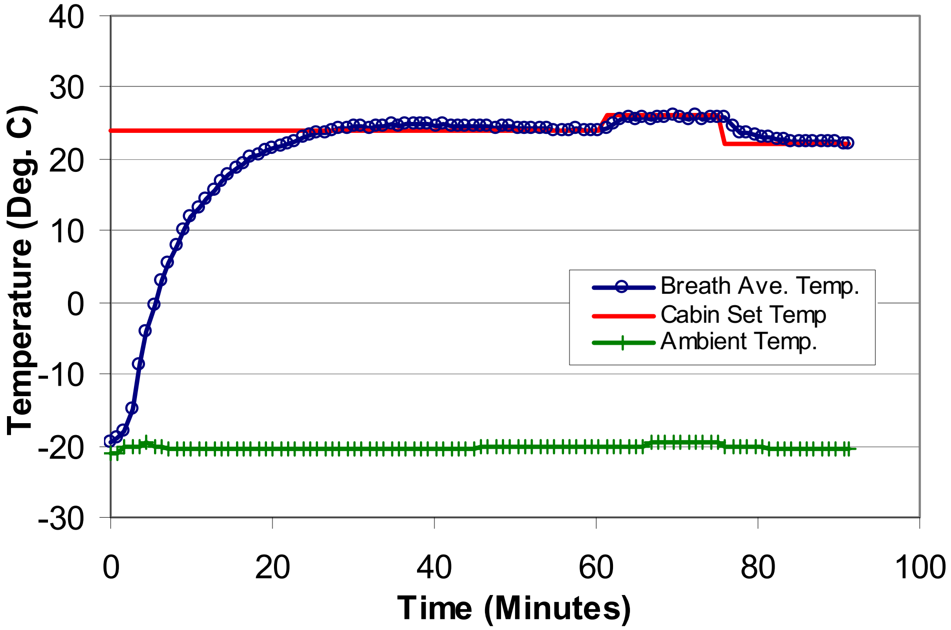

Fig. 8 System Control Capability with Set Temperature Change

CONCLUSIONS

The present ACC system achieves improved performance through the introduction of several new control concepts:

1. Separation of transient and steady state control. This allows the two control processes to be separately optimized without negatively affecting each other.
2. Tunable blending function and dwell time control. Combined, the tunable blending function and the dwell time control provide the controls engineer the mechanism to adjust the path to steady state for optimal comfort.
3. Power balance based steady state control. This makes it possible to have more precise solar and ambient compensation such that tighter cabin temperature control is achieved.

4. Passenger compartment thermal inertia. Explicit accounting for the passenger compartment internal thermal inertia effect in the thermal load calculation reduces reliance on the in-car temperature sensor and improves control accuracy.

Through the development of the present ACC system, an efficient software development process is established. This process makes use of the state-of-the-art software and hardware tools as well as software engineering concepts to achieve efficiency and quality in software development. This proves to bring great advantages to not only the advanced development of the ACC system but also to its implementation in production vehicle programs.

The present patented [4, 5, 6, 7] ACC system has been implemented in several North American and Asia-Pacific vehicles over the last three-year period. Production implementation demonstrated that the new control system met and exceeded the performance of the existing control software. More importantly, as we gained experience with production implementation, the development cycle became significantly reduced.

As a platform, this ACC system interfaces easily with other technologies being introduced to the market place. Automatic windshield defogging system has been demonstrated through proper software and hardware integration. Math based cabin model has been developed to work as a non-aspirated incar sensor by taking advantage of the cabin energy balance, achieving enhanced performance at reduced cost. The tunability of the present system enables comfort-based adaptive system concepts to be designed. Lastly, this system integrates easily with Energy Efficient AC system technology.

NOMENCLATURE

c_p	Specific heat of air
HVAC	Heating, Ventilation and Air Conditioning
I_{slr}	Solar intensity
$\dot{m}$	Discharge air flow rate
n	Exponent in interpolation function α
PWM_0	Pulse Width Modulation for HVAC blower at ignition, representing blower level
PWM_∞	Pulse Width Modulation for HVAC blower at steady state
PWR_{hvac}	Cooling or heating power delivered by HVAC system
PWR_{req}	Cooling or heating power required (thermal load of cabin)
PWR_{amb}	Thermal load directly related to ambient air temperature
$\dot{Q}_{slr}$	Solar heating power
T	Temperature
T_a	Temperature of ambient air
T_d	Discharge air temperature
$T_{dis,0}$	Discharge air temperature at ignition
$T_{dis,\infty}$	Steady state discharge air temperature
T_{dp}	Temperature of cabin internal mass (deep mass)
T_i	Incar temperature (cabin breath level temperature)
$T_{i,0}$	Incar soak temperature at ignition (deep mass)
T_s	Set temperature
UA	Thermal conductance of cabin enclosure
UA_{dp}	Thermal conductance over internal mass (deep mass)

Greek Symbols

α	Interpolation weighting function between initial and steady state control parameters
$\Delta T_{dis,0}$	Offset to the steady state discharge air temperature at ignition
ΔPWM_0	Offset to steady state blower at ignition
θ	Normalized incar temperature as a transient progress indicator
θ_{cr}	Critical value (threshold value) for ramping down blower and discharge temperature

Subscripts

0	Initial condition at ignition
a, amb	Ambient
d, dis	Discharge
dp	Deep mass (internal mass)
hvac	Heating, ventilation and air conditioning
i	In-car, within cabin
req	Required (HVAC Power)

s	Set (temperature)
slr	Solar
∞	Steady state

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ABOUT THE AUTHORS

Mingyu Wang is a Staff Research Engineer with 12 years of experience in the area of advanced PTC and HVAC systems development. He holds a Ph.D. in Mechanical Engineering from Duke University and is a member of SAE. He can be reached at (716) 439-2493 or mingyu.wang@delphi.com

John L. Pawlak is the Manager of Advanced HVAC Development with 17 years of experience in the area of advanced HVAC systems, Automatic Climate Control and HVAC module sub-system design and development. He earned a BSME from the Rochester Institute of Technology and a MSME from SUNY at Buffalo. John is a member of SAE. He can be reached at (716) 439-2248 or john.l.pawlak @delphi.com.

Charles A. Archibald is the Manager for Electronics Integration with over 21 years of experience working on electronic PTC and HVAC systems. This has included Electric Vehicles, Electronic Controlled Compressors and Automatic Climate Control Systems. He graduated from SUNY at Buffalo with a BSEE in 1982 and an MSEE in 1985. Charles is a member of SAE and IEEE. He can be reached at (716) 439-3257, or charles.a.archibald @delphi.com

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