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A/C Energy Management and Vehicle Cabin Thermal Comfort Control

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Xingda Yan, James Fleming and Roberto Lot

Abstract—**This paper introduces a novel multi-objective controller which regulates A/C system operation in a trade-off between vehicle cabin comfort and fuel consumption for a conventional vehicle with internal combustion engine. The controller has been developed and tested in a simulated environment, where an energy-based model of the A/C system is combined with a thermal dynamic model of the cabin which considers heat transfer to the environment. The control algorithm proposed herein is compared with two widely used control techniques in the industry, respectively the thermostat and PI control, under different driving cycles. This novel method is implementable in real-time, and simulation results show a reduction of up to 2% in A/C system fuel consumption compared to existing methods with similar thermal performance.**

Index Terms—**A/C system modelling, Thermal comfort**

I. INTRODUCTION

For a modern vehicle, the air conditioning (A/C) system is typically the ancillary system with the largest fuel consumption, and has a significant impact on the fuel efficiency of the vehicle [1]. An increment of 5% in A/C system efficiency could potentially save more than 1.32 billion litres of fuel per year in the USA, according to a study by the US National Renewable Energy Laboratory [2].

Generally, the control techniques which are used to control the A/C system can be summarized as: (1) thermostat control, which is the mostly commonly used [3]; (2) PID control; (3) fuzzy logic based control; (4) optimization-based control. Research attention has been mainly placed on improving steady-state performance due to the temperature oscillation around the set-point [4] and reducing the energy consumption and the mechanical component wear due to the frequency of ON/OFF actions [5]. In order to further improve steady-state performance [6], another widely used control technique for the A/C system is PID control. One of the major challenges of controlling A/C systems is the nonlinearity and uncertainty in A/C processes [7] Fuzzy logic based control control techniques have been utilized to deal with these issues. Khayyam et al. [8] described a intelligent energy management system to reduce the energy consumption of A/C, which includes three fuzzy controllers. Frazaneh et al. [9] focuses on thermal comfort temperature and energy. They use Fanger's predicted mean vote as controller feedback, and evaporator cooling capacity is selected as a criterion for energy consumption. Two fuzzy controllers are designed, one with a temperature feedback and the other with the predicted mean value index

feedback. The main disadvantage of these fuzzy logic based control scheme is that a large amount of experimental data is required for model training, which should cover the whole operating range. In recent years, some research effort has been spent on optimization-based control design for the A/C system to achieve a tradeoff between fuel consumption, cabin comfort and system durability [10]. However, the possibility of implementation of these optimization-based controllers is unknown due to the high computation burden of solving optimal control problems in realtime.

Therefore, it is still a open question how to design an A/C system controller for a conventional vehicle which considers both thermal performance and energy efficiency, yet can be implemented at low cost and low effort.

In this paper, a multi-objective control strategy which can make a trade-off between cabin comfort and the fuel consumption is presented for a conventional vehicle with internal combustion engine. Fanger's predicted mean vote (PMV) is used as the main control target, which is explained in section II. In section III, the energy-based model of the A/C system proposed by Zhang et al. [10] is adopted and combined with a thermal dynamic model of the cabin with consideration of the heat transfer with the environment and the powertrain model . In section IV, a multi objective controller is formulated which takes into account both cabin comfort and instantaneous engine efficiency. By incorporating a stored map of engine efficiency, the control law can encourage A/C system work when the system efficiency is high and discourage A/C system work when the system efficiency is low. The performance of the proposed control law is compared with two widely used A/C system control techniques in the industry: thermostat control and PI control. Simulation results in section V indicate that the proposed law can make a compromise between cabin comfort and fuel efficiency. It should be stressed that implementation of the proposed method does not require estimation of the parameters of the complex A/C system model. The implementation of the proposed method is straightforward, requiring only a table lookup and calculation of a single additional term compared to existing PI controllers.

II. THERMAL COMFORT

Thermal comfort in the vehicle is usually evaluated using the air temperature in the cabin and the mean radiant temperature. In accordance with the ASHRAE 55 [11] international standards, *thermal comfort* can be defined as "the condition of mind which expresses satisfaction with the thermal environment", illustrating that it is a cognitive

Xingda Yan, James Fleming and Roberto Lot are with the Mechanical Engineering Department, Faculty of Engineering and the Environment, University of Southampton, Southampton, UK, SO17 1BJ e-mail: X.Yan@soton.ac.uk .

process influenced by various quantities, physical activity, physiological and psychological factors. The two most widely used thermal comfort indexes: Predicted Mean Vote (PMV) and Predicted Percentage of Dissatisfied (PPD), both of which were developed by Fanger [12] are adopted in this paper. For simplicity, the details of the calculation of PMV is omitted here. In general, the PMV is a empirical function of several parameters as follow.

$$PMV = f(P_{met}, I_{cl}, T_c, T_r, T_{cl}, v_a, p_a) \tag{1}$$

where I_{cl} is the clothing insulation, T_r is the mean radiant temperature, T_{cl} is the clothing surface temperature, v_a is the air velocity and p_a is the water vapor partial pressure. The recommended acceptable PMV range for thermal comfort from ASHRAE 55 is between -0.5 and +0.5 for an interior space.

The associated PPD index is a function of the PMV index:

$$PPD = 100 - 95e^{[-(0.3353PMV^4 + 0.2179PMV^2)]} \tag{2}$$

From the concept of thermal comfort, people will hardly notice the temperature change if the thermal comfort is maintained within a neutral zone. In practice, the closed environment is considered as comfortable if the PMV is between -0.5 and $+0.5$ [12]. For convenience, the mean PPD value is used to indicate the mean discomfort in the cabin during the whole simulation. The reason to use PPD instead of PMV is that the PPD is a unsigned value which represents the discomfort caused by hot and cold in the same way, while PMV is a signed value which has a different sign for hot and cold sensations.

III. A/C AND VEHICLE MODELS

This section describes the mathematical model which has been used to develop and test the new A/C control algorithms. In order to take into account both cabin thermal comfort and fuel consumption, three models are combined which characterize the power flow among three different subsystems: the cabin, the A/C system, and the vehicle powertrain. Moreover, a model of driving power losses is included as it has major influence on the power-train efficiency.

In this paper, a typical car air conditioning system with refrigerant R134A is considered, which contains four main components: compressor, condenser, evaporator and expansion valve. According to the thermodynamic property of R134A [13], when the refrigerant pressures p_1 and p_2 in the condenser and evaporator are given, the enthalpy of the four vertices (1,2,3,4) can be calculated under the assumption that $h_3 = h_4$, which are equal to the saturated liquid enthalpy at the condenser pressure p_1, and the refrigerant temperature out of the evaporator is superheated by 10°C.

A. Compressor and Heat Exchangers

The compressor is connected to and disconnected from the crankshaft via a magnetic clutch. The mass flow rate of the refrigerant circulating in the system is determined by the engine speed ω_e. The quasi-static model proposed in [10] is summarised below.

$$\dot{m}_c = \eta_v V_d \rho_1 \omega_e \qquad h_2 = h_1 + h_1 \left(\left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \frac{1}{\eta_s}$$

$$P_c = \dot{m}_c(h_2 - h_1) \quad \tau_c = \frac{P_c}{\omega_e} \qquad COP = \frac{P_c}{\omega_e \tau_c}$$

Here, $\dot{m}_c$ is the mass flow rate of the refrigerant R134A, P_c is power consumption of the compressor, τ_c is the torque demand for the compressor, COP is the coefficient of Performance, ρ_1 is the saturated vapour density under pressure p_1, γ is the specific heat ratio of R134A and V_d, η_v, η_s are the displacement, volumetric efficiency and isentropic efficiency respectively for a specific compressor model.

The energy-based condenser and evaporator model can be expressed in the following form.

$$\dot{p}_1 = \frac{\dot{m}_{a,e}c_a(T_{a,in,e} - T_{a,out,e}) + \dot{m}_c(h_4 - h_1) * \sigma}{d_1(p_1)} \tag{3}$$

$$\dot{p}_2 = \frac{\dot{m}_{a,c}c_a(T_{a,in,c} - T_{a,out,c}) + \dot{m}_c(h_2 - h_3) * \sigma}{d_2(p_2)} \tag{4}$$

where σ is the compressor clutch signal, $\dot{m}_{a,e}$, $\dot{m}_{a,c}$ are the mass flow rates of air cross flowing through the evaporator and condenser respectively, The term d_1 and d_2 are the same function dependent on the refrigerant pressures in the evaporator and condenser with the subscript (e,c) indicated the different heat exchanger. The reader is directed to [10] for further details.

B. Cabin Thermal

The purpose of the A/C is maintaining a comfortable temperature inside the cabin. Hence, a model of the cabin temperature is necessary. For simplicity, the cabin is treated as a control volume with various energy loads. The model can be established based on energy balance as follows,

$$\frac{dT_c}{dt} = \frac{1}{\rho_a V_{ca} c_a} \left(P_{Met} + P_{Rad} + P_{Amb} + P_{AC} \right),$$

where T_c is the cabin temperature, ρ_a is the air density, V_{ca} is the cabin volume, and c_a is the specific heat. Four major energy loads are considered: the metabolic load from the passengers P_{Met}, the solar radiation load P_{Rad} and ambient load caused by heat exchanging with the ambient air outside the cabin P_{Amb}. P_{AC} is the cooling load from the A/C. The four loads may be calculated as,

$$P_{Met} = N \ P_{Ms} \qquad P_{Rad} = A \ S_c \ e^{(-E_c \sec \theta_z)}$$

$$P_{Amb} = h_c A(T_{Amb} - T_c) \quad P_{AC} = \dot{m}_{a,b}c_a(T_{a,out,e} - T_c).$$

where N is the number of passengers, P_{Ms} is the metabolic load for single passenger, which is about 100 W on average, $m_{a,b}$ is the mass flow rate of air out of the blower, T_{Amb} is the ambient temperature, and h_c is the heat transfer coefficient. For the calculation of solar radiation [14], A is the car surface area, S_c is the apparent solar irradiation, E_c is the extinction coefficient and θ is the Zenith angle.
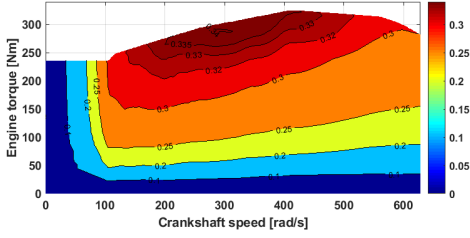
Figure 1: Engine efficiency map for the IC engine

C. Powertrain and Driving Losses

For the powertrain, a traditional passenger car is considered with a internal combustion engine and an automatic transmission. The internal combustion engine model is implemented as a lookup table [15] that maps engine torque and crankshaft speed values to the engine efficiency as shown in Fig. 1. Hence we may consider the model as consisting of a single mathematical function

$$\dot{m}_f = f(\tau_e + \tau_c, \omega_e) \tag{5}$$

where $\dot{m}_f$ is the rate of change of fuel mass in the fuel tank of the vehicle, τ_e is the crankshaft torque in order to power the wheel, ω_e is the angular velocity of the crankshaft. The crankshaft torque τ_e and speed ω_e are obtained from the powertrain model which consists of the transmission model and the vehicle motion model. The relationship of the torque and angular velocity before and after the gearbox is,

$$\tau_w = N_g \tau_e \qquad \omega_e = \omega_w N_g,$$

where τ_w is the torque delivered to the wheel, ω_w is the angular velocity of the wheel and N_g is the gear ratios.

The vehicle is taken as a mass obeying Newton's second law, so that the vehicle motion can be modelled as

$$m\dot{v} = \frac{\tau_w + \tau_h}{r} - \frac{1}{2}\rho A_f C_d v^2 - C_{rr} mg - mg\sin\theta$$

where m is the vehicle mass, v the vehicle speed, F_D the traction force, F_A the aerodynamic resistance force, F_R the rolling friction force, τ_h the mechanical braking torque, r the wheel radius, ρ the air density, A_f the frontal area, C_d the coefficient of drag and C_{rr} the coefficient of rolling resistance, θ is the road gradient.
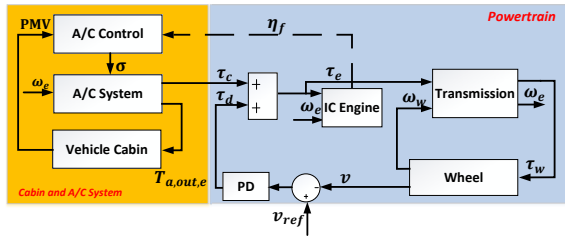

Figure 2: Control diagram of the whole system

IV. CONTROLLER DESIGN

Firstly, the proposed control structure for the simulation, which is shown in Fig. 2, is described. There are two inputs

for the A/C system, which are the switching signal of the compressor clutch σ and the engine speed ω_e. However, the engine speed ω_e is the output of the transmission and depends on the drive cycle, and therefore the switching signal σ is the only control variable. The cabin comfort index PMV is one of the inputs for the A/C controller. In some control algorithms considered below, the engine efficiency η_e is also an input for the controller. The vehicle powertrain is controlled to track a desired vehicle speed profile v_{ref} by a PD controller. The output of the PD controller is the powertrain torque demand τ_d. And the A/C system torque τ_c will be controlled in order to maximise thermal comfort and minimise fuel consumption as discussed in the following sections.

A. Thermostat Controller

The A/C system switches from OFF to ON when the cabin temperature exceeds the desired temperature T_{SET} by more than a threshold T_h, and it switches from ON to OFF when the cabin temperature is lower than the desired temperature T_{SET} by T_h. Hence, the thermostat controller can be defined as,

$$\sigma = \begin{cases} 0 \to 1: & if \quad T_c - T_{SET} > T_h \\ 1 \to 0: & if \quad T_{SET} - T_c > T_h \end{cases} \tag{6}$$

where T_{SET} is the desired cabin temperature in the cabin, and T_h is the temperature difference threshold.

B. PI Controller

The obvious benefit of using thermostatic control is that the rules directly relate to the ON/OFF of the A/C system. However, due to the discrete nature of the switching signal σ, it is troublesome to apply control theory developed for continuous models, such as PI controllers. Hence, the Pulse Width Modulation (PWM) technique, which is widely used in communication technology and power electronics control is adopted here to convert the switching signal to a so-called "duty cycle" D, which represents the fraction of the ON time period of the A/C system for a fixed switching frequency [16].

With the help of the PWM technique, a PI controller can be used to regulate the cabin comfort index PMV to the desired value PMV_{ref}, which should be zero. Hence, the duty cycle can be calculated as

$$D_{PI} = K_p \left(PMV(t) - PMV_{ref} \right)$$
$$+ K_i \int_0^t \left(PMV(s) - PMV_{ref} \right) ds. \tag{7}$$

where K_p and K_i is the coefficient of the PI controller.

C. Multi-objective Controller including Cabin Comfort and Overall Efficiency

In order to make a trade-off between the cabin comfort and fuel consumption, this proposed multi-objective controller can not only regulate the cabin comfort index PMV to the reference PMV_{ref}, but may be modified to adjust the operation of A/C system based on the overall efficiency of the cooling system, which corresponds to the product by the coefficient of performance of the A/C system efficiency COP

and engine efficiency η_e. An extra term $K_e(\eta_e COP - \eta_t)$ is included in the controller: when the overall efficiency $\eta_e COP$ is greater than the threshold η_t, this term is positive and encourages the A/C usage; when the overall efficiency $\eta_e COP$ is less than the threshold η_t, the term is negative and discourages A/C usage. The duty cycle function is defined as follows:

$$D = K_p (PMV(t) - PMV_{ref}) + K_e (\eta_e COP - \eta_t)$$
$$+ K_i \int_0^t (PMV(s) - PMV_{ref}) \, ds. \tag{8}$$

D. Multi-objective Controller including Cabin Comfort and Engine Efficiency

The major drawback of the multi-objective controller (8) defined above is that the calculation of COP instantaneously is not a trivial task, which makes this method difficult to implement in practice. Therefore, in this section, the proposed controller is simplified by only including the engine efficiency η_e as

$$\bar{D} = K_p(PMV(t) - PMV_{ref}) + K_e(\eta_e - \bar{\eta}_t)$$
$$+ K_i \int_0^t (PMV(s) - PMV_{ref}) ds \tag{9}$$

where $\bar{\eta}_t$ is the efficiency threshold for this controller. The engine efficiency η_e can be calculated by a stored map with two inputs: the engine speed ω_e and the crankshaft torque τ_e as shown in Fig. 1. As ω_e and τ_e can be obtained from the vehicle ECU, the controller can be implemented without any additional sensors than the PI controller (7) and it is only necessary to store a look-up table for the engine efficiency map. Moreover, simulations will show that the performance of the proposed controller (9) is very close to that of the more complex multi-objective controller (8).

V. SIMULATION RESULTS

A. A/C System Performance Under Different Control Schemes

In this subsection, the vehicle speed is controlled to track the WLTP driving cycle [17], which includes a combination of urban, rural and highway driving. At the same time, the A/C system is controlled by the four different control strategies: the thermostat controller (6), PI controller (7) which tracks thermal comfort only, the multi-objective controller (8) which takes into account the overall efficiency of the engine and A/C system, and the multi-objective controller (9) which includes only engine efficiency without considering the COP of the A/C. For simplicity, it is assumed that the solar radiation is constant during the simulation as well as the mass flow rate of air out of the A/C system.

1) Fuel Consumption Compared under Similar Cabin Comfort: In this first set of simulations, the four controllers are tuned to have similar thermal performance in order to compare the fuel consumption. The threshold of the thermostat controller T_h is set as $0.5°C$, the PMV reference is null, control gains are set $K_p = 1$, $K_i = 0.05$, $K_e = 1$ and the efficiency threshold in (8) is $\eta_t = 0.8$, while the threshold in (9) has been set to $\bar{\eta}_t = 0.15$ to take into account an average COP of 5.
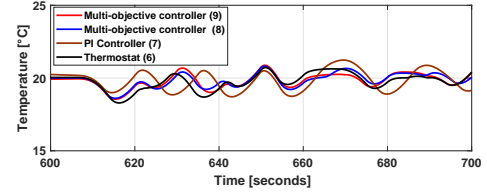

Figure 3: The cabin temperature, PMV, switching signal under different A/C control schemes against the engine speed.

As shown in Fig. 3, the cabin temperature and the PMV are varied in a similar range for the four different controllers. The mean temperature and PMV are same which is $20°C$ and 0. In other words, these four controllers give similar performance in term of cabin comfort. However, the fuel consumption differs as shown in Table I, where the fuel consumption of the IC engine under the different A/C control schemes is presented. It is interesting to note that the two proposed multi-objective controllers (8) and (9) have the same fuel consumption.

Table I: Fuel consumption of different control methods (WLTP)

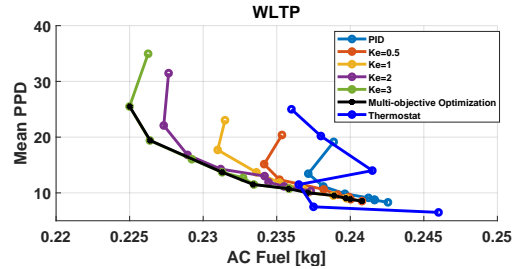| Methods	Fuel Usage	A/C Fuel Usage	A/C Fuel Saving
No A/C	1.344 kg	Baseline	
Thermostat (6)	1.586 kg	0.242 kg	0%
PI controller (7)	1.581 kg	0.237 kg	2.10%
Multi-objective controller (8)	1.576 kg	0.232 kg	4.31%
Multi-objective controller (9)	1.576 kg	0.232 kg	4.31%

Figure 4: The mean PPD against the A/C system fuel consumption when tracking WLTP driving cycle

2) Optimization of the Proposed Multi-objective Controllers: From the concept of thermal comfort, people will hardly notice the temperature change if the thermal comfort is maintained within the neutral zone. In practice, the closed environment is considered as comfortable if the PMV is between -0.5 and $+0.5$ [11]. Hence, it is meaningful to investigate the relationship between the cabin comfort and fuel consumption. For convenience, the mean PPD value is used to indicate the mean discomfort in the cabin during the whole simulation. For the PI controllers (7) and the two proposed multi-objective controllers (8)(9), the temperature and PMV vary in a larger range with the same mean value when decreasing K_p or increasing K_e. For the thermostatic controller, the variation of the temperature and PMV will increase when increasing the temperature threshold T_h.

In order to make a fair comparison of these controllers, multi-objective optimization, which makes a compromise be-

tween cabin comfort and AC fuel consumption, has been carried out. It should be mentioned that as the performance of the two multi-objective controllers is very similar. In the following part, only the results of controller (9) is displayed. And the PI controller has been treated as a special case of the proposed multi-objective controllers with $K_e = 0$. As shown in Fig. 4, each line (except the black line) indicates the relationship between the mean PPD and the A/C system fuel consumption for a fixed K_e and varied K_p. Then the black line is the Pareto frontier after multi-objective optimization. As we can see, for the PI controller (7) and multi-objective controller (9), the A/C system fuel consumption decreases when the mean PPD increases. For the thermostat this relationship is not convex. On the other hand, for a given mean PPD value the A/C fuel usage is always minimised when the A/C system is controlled by the multi-objective controller (9).
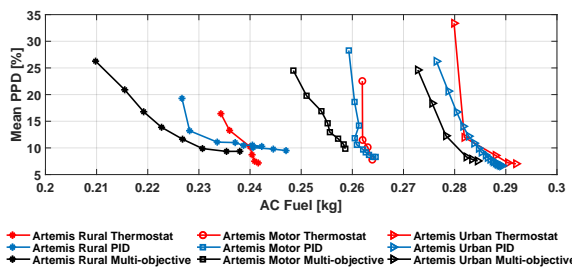

Figure 5: The mean PPD against the A/C system fuel consumption when tracking Artemis cycles

B. A/C System Performance Under Different Driving Cycles

In this subsection, the vehicle speed is controlled to track three different driving cycles [18]: the urban, rural and highway Artemis cycles. A similar simulation procedure has been carried out as in subsection A above. The parameters K_P, K_e, and T_h have been changed in each set of simulations to investigate the relationship between the mean PPD and the A/C system fuel consumption. As shown in Fig. 5, the fuel consumption under the modified PI controller (9) is always the minimum among the three controllers for a given driving cycle. Hence, we can claim that by introducing the term $K_e(\eta_e - \eta_t)$, the controller (9) can give an trade-off between the thermal performance and fuel consumption.

VI. CONCLUSION

In this paper, the cabin thermal comfort and energy management of A/C systems of passenger cars has been studied. In order to simultaneously maintain a high thermal comfort in the cabin and reduce A/C system fuel consumption, a multi-objective controller is proposed, which is formulated as a multi-input, single-output, PI controller. To test the performance of the controller, the A/C system model, cabin thermal model and vehicle model are also presented. Simulations were carried out to compare the proposed controller with two state-of-the-art controllers: thermostat control and traditional PI control under several different driving cycles. It is shown by the simulation results that the fuel consumption of the proposed controller is the minimum among several controllers with similar thermal performance. Despite its enhanced performance, the proposed controller requires only a table lookup of engine efficiency compared to the corresponding PI controller. Importantly, if engine torque and RPM are obtained from the engine ECU, no additional sensors are required compared to existing approaches .

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