15th Triennial World Congress, Barcelona, Spain

SLIDING MODE CONTROL FOR AUTOMOBILE AIR CONDITIONER

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Abstract: Increasing demand toward driving comfort steers automobile engineers to spend more and more effort on refining the automobile air conditioning system. The system is of high order and operated at the presence of several disturbances. The control algorithm should be able to control the evaporator out temperature with the acceptable control action. In this work, the sliding mode control is adopted for its advantage of dealing with high order, nonlinear system. Besides, several linear and nonlinear observers are proposed such that the controller may be implemented under the constrain of limited sensor set.

Keywords: Automobile industry, Compressors, Automatic control (Closed-loop), Temperature control, Control applications.

1. INTRODUCTION

Due to increasing customer demands toward better driving comfort, the automatic climate (temperature) control becomes one of the most important challenges for automobile engineers. Usually, the compressor of the refrigeration system is connected directly to engine. Therefore, the refrigerant flow rate can only be regulated by properly manipulating the capacity of compressor instead of varying its driving speed. For this electronic variable compressor (EVC) the capacity is indirectly regulated by the pressure difference of crankcase and suction port through the control of valve duty cycle value. The refrigeration system consists of several components such as condenser, orifice tubes, evaporator, refrigerant accumulator and electronic variable compressor. Even more, each of them involves certain kinematics such as the fluid and thermal dynamics. Therefore, there exists a need of advanced control techniques for this complicated system. In this work, the sliding mode control algorithm is considered for its robustness and capability of dealing with high order, nonlinear systems.

2. AUTOMOBILE AIR CONDITIONER MODEL

In this work, an interconnected volume system (IVS) model similar to (I. Kolmanovsky and et al., 1999) is developed based on the report of Kenneth Kelemen (Kenneth Kelemen, 1994). This model provides a simplified representation of refrigerant flow/evaporator temperature dynamics while capturing the dynamics interactions between subsystems which are important for control design. The IVS model comprises electronic variable compressor, condenser, evaporator, accumulator and orifices between them. The corresponding motion equations may be presented in the following two major categories.

2.1 Refrigerant Cycle subsystem

The equations in this category describe the simplified dynamics of temperature and fluid system where phase change dynamics are neglected:

$$\dot{P}_d = \frac{1}{C_1}\left[Q - \frac{P_d - P_{cond}}{R_1}\right] \tag{1}$$

$$\dot{P}_{cond} = \frac{1}{C_2}\left[\frac{P_d - P_{cond}}{R_1} - \frac{P_{cond} - P_{evap}}{R_2}\right] \tag{2}$$

$$\dot{P}_{evap} = \frac{1}{C_3}\left[\frac{P_{cond} - P_{evap}}{R_2} - \frac{P_{evap} - P_S}{R_3}\right] \tag{3}$$

$$\dot{P}_S = \frac{1}{C_4}\left[\frac{P_{evap} - P_S}{R_3} - Q\right] \tag{4}$$

$$\dot{T}_{airout} = \frac{K_6 P_{evap} - T_{airout}}{\tau_4} \tag{5}$$

2.2 Electrical Variable Compressor subsystem:

The equations in this category represent the dynamics of the electronic variable compressor:

$$\dot{P}_{CC} = \frac{-P_{CC} + DC \cdot P_d + (1 - DC)P_S}{\tau} \tag{6}$$

$$\ddot{x} = -\frac{B}{M}\dot{x} - \frac{K}{M}x + \frac{L_2 \cdot 2A_{piston}}{R_1 M}\left(P_S - P_{CC} + P_{fric}\right) + \frac{L_3 A_{piston}}{L_1 M}\left(P_d - P_S + 2P_{CC}\right) \tag{7}$$

$$Q = K_{flow}(N) \cdot x \text{ (Constant compressor speed)} \tag{8}$$

$$P_{fric} = P_o \cdot \mathrm{sgn}(\dot{x}) \tag{9}$$

Please refer to the following Table 1 for the physical meaning of each state variable/parameter:

Table 1. State variables of the system

State Variable	Explanation
P_d	Discharge pressure
P_{cond}	Condenser pressure
P_{evap}	Evaporator pressure
P_S	Suction pressure
P_{CC}	Crank case pressure
P_{fric}	Friction in terms of pressure
P_o	Friction magnitude
x	Compressor stroke
T_{airout}	Evaporator out temperature
DC	Valve Duty Cycle
N	Engine speed
Q	Refrigerant flow rate

Although the complexity of model has been reduced, it is still of high order and several nonlinearites.

3. SLIDING MODE CONTROL DESIGN

As can be observed in the previous section, the system is of high order and several nonlinearities. Based on the hierarchical design concept on block control principle and order of Zero Dynamics, the complexity of the problem may be reduced to subproblems of lower order.

3.1 Big picture of control design

According to the motion equations, the refrigerant flow rate, Q, may be considered as the intermediate control input of the block describing refrigerant cycle subsystem. Next, the flow rate may be regulated by proper VDC control action according to the nonlinear compressor dynamics. The big picture of the control design may be reflected in the following Figure 1. Note that the first derivative of state variable, P_{CC}, depends on the VDC. Based on this fact, we may select desired crank case pressure, $(P_{CC})_{des}$, which may generate desired flow rate, Q, and enforce sliding mode on the surface:

$$S = P_{CC} - (P_{CC})_{des}. \tag{10}$$

The crank case pressure would tend to the desired one after sliding mode is enforced on the surface, $S = 0$ and, as a result, the flow rate tends to desired one after some transient time. The desired refrigerant flow rate would insure proper regulation of control objective: desired evaporator temperature, T_{airout}.

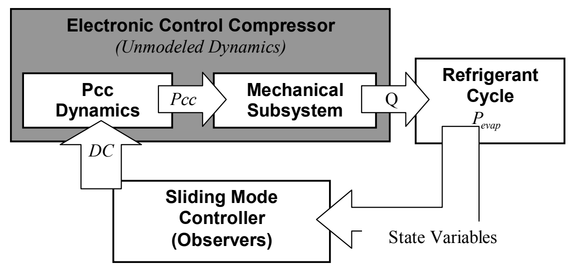

Figure 1. Big picture of control design.

3.2 Control design for Linear Refrigerant Cycle

Represent the system equation of the refrigerant cycle in the matrix format. One may easily find out that the Refrigerant Cycle subsystem is a linear one where the flow rate, Q, serves as a control input and the evaporator temperature, T_{airout}, is the output to be controlled. By introducing the new variable,

$$z = C_1 p_d + C_2 p_{cond} + C_3 p_{evap} + C_4 p_S, \tag{11}$$

the system could be further represented in the following form which contain controllable and uncontrollable part:

$$\begin{bmatrix} \dot{z} \\ \dot{p}_{cond} \\ \dot{p}_{evap} \\ \dot{p}_S \\ \dot{T}_{airout} \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ \frac{1}{C_1 C_2 R_1} & -\left(\frac{1}{C_1 R_1} + \frac{1}{C_2 R_1} + \frac{1}{C_2 R_2}\right) & \left(\frac{1}{C_2 R_2} - \frac{C_3}{C_1 C_2 R_1}\right) & \frac{C_4}{C_1 C_2 R_1} & 0 \\ 0 & \frac{1}{C_3 R_2} & -\left(\frac{1}{C_3 R_2} + \frac{1}{C_3 R_3}\right) & \frac{1}{C_3 R_3} & 0 \\ 0 & 0 & \frac{1}{C_4 R_3} & -\frac{1}{C_4 R_3} & 0 \\ 0 & 0 & \frac{K_6}{\tau_4} & 0 & -\frac{1}{\tau_4} \end{bmatrix} \cdot \begin{bmatrix} z \\ p_{cond} \\ p_{evap} \\ p_S \\ T_{airout} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ -\frac{1}{C_4} \\ 0 \end{bmatrix} \cdot Q \tag{12}$$

Quite fortunately, the uncontrollable part, z, is stable. However, the eigenvalues of the system are: [-0.078 -3.57 -0.35 -307.43 0] which means that one partial motion is much faster than others. This fact may be observed from the Eqn (4). From the system identification results in (Kenneth Kelemen, 1994), the flow resistance R_3 is very small for accumulator. Therefore, it is reasonable to assume $P_{evap} \cong P_S + QR_3$ at any time.

For such a multi-rate system, rather than assigning eigenvalues for each state variable, the idea of block control methodology may be applied: Since the T_{airout} does not effect the dynamics of other states, it may be isolated from the refrigerant cycle.

In other words, we may consider the dynamics of P_d, P_{cond}, P_{evap} and P_S a block and the T_{airout} dynamics another one. These two blocks are marked by dashed-line in Eqn (12). The state, P_{evap}, may serve as the fictitious control input for the T_{airout} dynamics block.

The control design for this multi-rate system may be concluded in the following steps:

1. Since the dynamics of T_{airout} determined by τ_4 are acceptable, one may regulate the temperature to desired one through the fictitious control P_{evap}.

2. The desired P_{evap} may be selected as $P^*_{evap} = T_{des} / K_6$ (the solution of static Eqn (5)) such that $T_{airout} \to T_{des}$ after finite time. The T_{des} is the set-point of evaporator out temperature.

3. The fictitious control for Refrigerant cycle may be selected as

$$x_{des} = \frac{(C_3 + C_4)}{K_{flow}}\left[\Gamma^* + \mathrm{K}^*\left(P_{evap} - P^*_{evap}\right)\right] \qquad (13)$$

where $\Gamma^* = \frac{1}{C_3 + C_4}\left[\frac{P_{cond} - P_{evap}}{R_2}\right]$.

4. The rate of $P_{evap} \to P^*_{evap}$ is determined by K^*.

The proposed desired control for refrigerant cycle subsystem requires less system parameters if compared to traditional eigenvalues placement method. As a result, lower sensitivity to parameter variations is another important benefit using block control methodology in addition to its easy implementation.

3.3 Control design for Nonlinear compressor

To generate desired refrigerant flow rate, Q, it needs desired compressor stroke which is regulated by proper crank case pressure. The desired crank case pressure may be found by solving the static equation of compressor dynamics (6):

$$\left(P_{CC}\right)_{des} = \frac{\left[(2L_2 - L_3)P_S + L_3 P_d + L_2 P_{fric} - \frac{L_1 K_{comp}}{A_{piston}} x_{des}\right]}{2\left(L_2 - L_3\right)} \qquad (14)$$

Since the desired crank case pressure may be evaluated, the control force for compressor (VDC) is implemented in the following form:

$$VDC = \begin{cases} 1 & P_{CC} < \left(P_{CC}\right)_{des} \text{ or } S > 0 \\ 0 & P_{CC} > \left(P_{CC}\right)_{des} \text{ or } S < 0 \end{cases}. \qquad (15)$$

To reduce the chattering due to unmodeled dynamics, the discontinuous control force is then modified as the linear approximation one (J.-J. Slotine et al., 1983):

$$VDC(k \cdot S + bias) = \begin{cases} (k \cdot S + bias) & if \ \left|(k \cdot S + bias)\right| \le \varepsilon \\ 1.0 & if \ (k \cdot S + bias) > \varepsilon \\ 0.0 & if \ (k \cdot S + bias) < -\varepsilon \end{cases}. \qquad (16)$$

where $\varepsilon << 1$ and *bias*=0.5.

With the proper selection of k, the chattering or excessive compressor stroke action may be reduced. Basically, large compressor stroke will increase the engine load which might dramatically hinder the engine performance. Please also refer to the following schematic diagram (Fig. 2) for this approach.

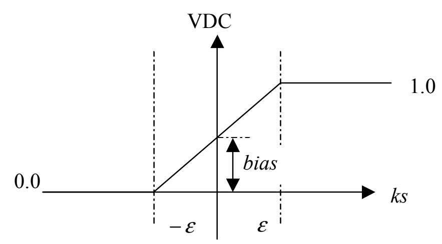

Figure 2. Schematic diagram of linear approximation.

4. OBSERVER DESIGN

To implement the proposed linear and sliding mode controller, the information of P_{evap}, P_d, $\dot{x}_{compressor}$, P_{fric} and P_S is necessary. To predict the states needed for control implementation, two asymptotic and sliding mode observers are designed based on the assumption that P_d, P_S and T_{airout} are available.

4.1 Asymptotic Observer for Refrigerant Cycle

By introducing a new variable $y = C_1 P_d + C_3 P_{evap}$, the unavailable information refrigerant flow, Q, inside the system equation may be excluded. Because P_{evap} may be approximated by $P_S + QR_3$ due to small R_3, the dynamics of the P_{evap} may be governed by

$$\dot{P}_{evap} = \frac{1}{C_3}\left[\frac{P_{cond} - P_{evap}}{R_2} - Q\right].$$

Therefore, the dynamics of newly introduced variable

$$\dot{y} = \frac{P_{cond} - P_{evap}}{R_2} - \frac{P_d - P_{cond}}{R_1}$$

does not depend on control and the information of y may also be approximated by $y = C_1 P_d + C_3 P_S$. To estimate P_{cond}, the following asymptotic observer is designed:

$$\begin{bmatrix} \dot{\hat{P}}_{cond} \\ \dot{\hat{y}} \end{bmatrix} = \begin{bmatrix} -\frac{1}{C_1}\left(\frac{1}{R_1} + \frac{1}{R_2}\right) & \frac{1}{C_2 C_3 R_2} \\ \left(\frac{1}{R_1} + \frac{1}{R_2}\right) & -\frac{1}{C_3 R_2} \end{bmatrix} \begin{bmatrix} \hat{P}_{cond} \\ \hat{y} \end{bmatrix} + L_{observer}(y - \hat{y})$$

The observer gain, $L_{observer}$, may be selected by Ackerman's formula such that the estimation of $\hat{P}_{cond}$ would tend to actual P_{cond} at the desired rate.

4.2 Observer for refrigerant flow rate Q

Since the information of $\hat{P}_{cond}$ can be found, the refrigerant flow rate, Q, may be estimated from the following Sliding Mode Observer:

$$\dot{\hat{P}}_d = \frac{1}{C_1}\left[\frac{\hat{P}_{cond} - P_d}{R_1}\right] - U_{SMO\,I} \cdot sat\left(\frac{P_d - \hat{P}_d}{\varepsilon}\right)$$

where $sat(a) = \begin{cases} a & if\ |a| \le 1 \\ 1 & if\ a > 1 \\ -1 & if\ a < -1 \end{cases}$ and $\varepsilon \ll 1$.

Represent the corresponding error system for the observer:

$$\dot{\overline{P}}_d = \frac{1}{C_1} Q - U_{SMO\,I} \cdot sat\left(\frac{\overline{P}_d}{\varepsilon}\right).$$

The observer gain $U_{SMO\,I}$ is selected high enough to enforce sliding mode on the surface $\overline{P}_d = \left(P_d - \hat{P}_d\right)$. The average value of $U_{SMO\_I} \cdot sat\left(\frac{\overline{P}_d}{\varepsilon}\right)$ would tend to Q/C_1 after sliding mode is enforced. Therefore, the refrigerant flow rate, Q, may be found from

$$\hat{Q} = C_1\left[U_{SMO\_I} \cdot sat\left(\frac{\overline{P}_d}{\varepsilon}\right)\right] \to Q.$$

4.3 Observer for compressor stroke and friction

Obviously, more information is required to implement the sliding mode controller (14) for electronic valve compressor and estimate compressor friction. The observer for P_{CC} and P_{fric} is shown below:

$$\begin{bmatrix} \dot{\hat{x}}_1 \\ \dot{\hat{x}}_2 \\ \dot{\hat{P}}_{CC} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ -\frac{K_{comp}}{M_{comp}} & -\frac{B_{comp}}{M_{comp}} & \frac{-2A_{piston}(L_2 - L_3)}{L_1 M_{comp}} \\ 0 & 0 & \frac{-1}{\tau_1} \end{bmatrix} \begin{bmatrix} \hat{x}_1 \\ \hat{x}_2 \\ \hat{P}_{CC} \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{(2L_2 - L_3)A_{piston}}{L_1 M_{comp}}(P_S) + \frac{L_3 A_{piston}}{L_1 M_{comp}} P_d + \frac{(2L_2 - L_3)A_{piston}}{L_1 M_{comp}} \hat{P}_{fric} \\ \frac{(1 - DC)}{\tau_1} P_S + \frac{DC}{\tau_1} \cdot P_d \end{bmatrix} + L_{ObserverII}(x_1 - \hat{x}_1)$$

where compressor friction may be estimated from

$$\hat{P}_{fric} = -(P^*_{fric})sat(k \cdot \dot{\hat{x}}_1), \quad sat(a) = \begin{cases} a & if\ |a| \le 1 \\ 1 & if\ a > 1 \\ -1 & if\ a < -1 \end{cases}.$$

The coefficient k is selected to approximate the ε zone of friction. P^*_{fric} stands for the magnitude of friction.

4.4 Evaporator pressure (p_{evap}) estimation

Note that the reduction of the fast P_S dynamics will result in steady state estimation error for P_{evap}. As a consequence, it will introduce additional error for linear refrigerant controller (12). Nevertheless, this imperfection may be eliminated by the integral term introduced in Eqn (13) and the additional correction introduced below:

$$\hat{P}_{evap} = \hat{P}_S + \hat{Q} R_3.$$

5. SIMULATION RESULTS

The simulation results may be found in the following Figure 3 and 4. The set point for T_{airout} is 50 $^\circ F$ and the unit of pressure is kPa. Basically the proposed asymptotic/sliding mode observers are able to estimate P_{evap}, P_d, $\dot{x}_{compressor}$, P_{fric} and P_S with acceptable accuracy. Based on that information, the proposed sliding mode controller may deliver qualified set-point tracking performance.

As discussed in section *3.3*, excessive compressor stroke action will undermine engine performance. With the help of linear approximated control input (16), the controller is able to track the desired temperature with acceptable compressor stroke action.

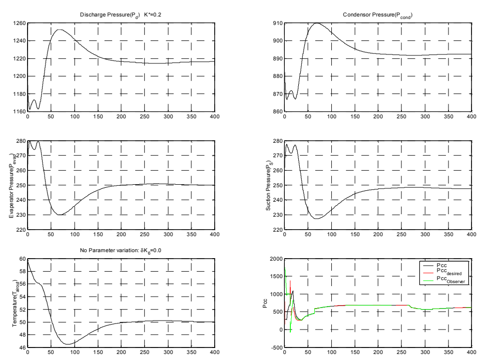

Figure 3. Temperature (T_{airout}) tracking performance.

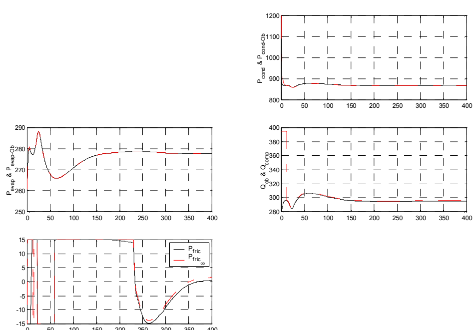

Figure 4. Observer performance.

6. CONCLUSIONS

The sliding mode control for the automobile air conditioner evaporator out temperature regulation is proposed in this work. The controller exhibits its inherent capability to reduce the control design complexity when dealing with high order, nonlinear system. With the help of proposed sliding mode/asymptotic observers, the controller demonstrates adequate evaporator out-temperature tracking performance with acceptable compressor action.

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