On reducing evaporator superheat nonlinearity with control architecture

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Abstract

Evaporator superheat control is an important aspect of the operation of refrigeration and air conditioning systems; since the majority of cooling in these systems occurs through evaporation of two-phase refrigerant, the energy efficiency is dramatically improved by reducing the amount of superheat present. However, allowing refrigerant to leave the evaporator without completely vaporizing risks catastrophic damage to the compressor, so superior control is required at low superheat levels. One of the most significant challenges present in this control problem is the presence of significant nonlinearities in the response from the control input, e.g. expansion valve position, to evaporator superheat. This paper reveals how a particular control architecture inherently compensates for both the static and dynamic nonlinearities that dominate the valve-to superheat transient response. Furthermore, the control implementation only requires temperature measurements, which are frequently available in ordinary HVAC systems. Experimental results confirm the reduction of nonlinearities using the proposed approach, and the authors discuss the effect of actuator limitations on the nonlinearity compensation.

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Diminution de la nature non linéaire de la surchauffe de l’évaporateur à l’aide de la conception de la régulation

Mots clés : Système frigorifique ; Évaporateur ; Détente directe ; Procédé ; Régulation ; Surchauffe

1. Introduction

Air conditioning and refrigeration systems comprise a significant portion of national electrical energy usage (e.g. 24% of commercial building electricity) (USDOE, 2008) and are the largest source of peak electrical demand. These systems operate primarily using a vapor compression cycle, where a primary metric for system efficiency is evaporator superheat. Improved superheat control leads not only to higher efficiencies, but also greater system reliability and performance. Superheat control is a difficult problem, largely due to the nonlinearities present in all VCC systems. This difficulty is compounded by the lack of complete measurement of system states in typical systems, and by the lack of a priori knowledge...
of the system components parameters, e.g. valve flow characteristics, which render traditional feedback linearization schemes cumbersome to implement. The primary contribution of this paper is a control architecture that addresses these problems by compensating for system nonlinearities using a cascaded feedback loop, while only requiring measurements typically available for use, namely, refrigerant pressures and temperatures.

The ideal VCC consists of four processes: (1) isentropic compression, (2) isobaric heat rejection and condensation, (3) isenthalpic expansion, and (4) isobaric heat absorption and evaporation. Fig. 1 illustrates this ideal VCC; Fig. 1(a) shows the cycle components, and Fig. 1(b) is a typical pressure–enthalpy (P–h) curve for a refrigeration cycle. Evaporator superheat is defined to be the difference between the refrigerant temperature at the evaporator outlet and the evaporator saturation temperature. Superheat control is a critical control problem for VCC-based systems, both in terms of optimizing system efficiency and preventing component failure. As the fluid passes through the evaporator, it absorbs heat and transitions from a liquid–gas mixture to a saturated vapor, and then further to a superheated vapor. If the refrigerant is allowed to leave the evaporator without completely vaporizing (i.e., no superheat), it will enter the compressor as a two-phase mixture, with the potential of causing catastrophic failure of the compressor.

However, since the majority of the heat transfer occurs during the vaporization process, excessively high superheat results in reduced cooling capacity of the system. Therefore, the portion of two-phase flow in the evaporator should be maximized in order to obtain maximum cooling capacity of the system. In general, an acceptable compromise between efficiency and safety is for the refrigerant at the evaporator exit to be a few degrees above its saturation temperature. Regulating this temperature difference is called superheat control, an important control problem for HVAC&R applications (e.g., Gruhle and Isermann, 1985; Finn and Doyle, 2000).

2. Superheat control

Superheat control is an exercise in fluid flow control. This metering of refrigerant in air conditioning, refrigeration, or heat pump systems is generally achieved by a number of different valve types, which vary in expense and design sophistication. The primary refrigerant metering device is known as the expansion valve, so called because the fluid expands from the liquid phase to a two-phase fluid mixture as the refrigerant travels through the valve, transitioning to a lower pressure. The simplest of expansion valve devices is a capillary tube or orifice where the refrigerant flows through a reduction in diameter. Other mechanical control devices include the thermostatic expansion valve (TEV) or a pressure-regulating device also known as the automatic expansion valve (AEV). Expansion valves that are actively controlled by a computer-based algorithm include the electronic expansion valve (EEV), which is opened and closed by a stepper motor.

The electronic expansion valve (EEV) was a major step forward in superheat control, since it allows the implementation of automatic control paradigms such as

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**Nomenclature**

- AEV: automatic expansion valve
- HVAC&R: heating, ventilation, air conditioning, and refrigeration
- VCC: vapor compression cycle
- TEV: thermostatic expansion valve
- EEV: electronic expansion valve
- HEV: hybrid expansion valve
- PID: proportional-integral-derivative
- $K_F$: proportional gain, inner loop of cascaded architecture
- $P_{SET}$: pressure setpoint for inner controller
- $K_M(v)$: gain from valve position $v$ to refrigerant mass flow
- SH: evaporator superheat
- $P_{evap}$: evaporator pressure
- $Q(s)$: transfer function from $P_{SET}$ to evaporator superheat (inner loop)
- $G(s)$: transfer function from mass flow to evaporator superheat
- $H(s)$: transfer function from mass flow to evaporator pressure
- $v$: expansion valve position
- $m$: refrigerant mass flow
- $K_{FM}$: product of $K_F$ times $K_M(v)$
- $K_U$: gain from HEV position to mechanical pressure setpoint
- $U$: HEV position
- MEMS: micro-electrical-mechanical systems

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![Fig. 1 - (a) Vapor compression cycle components. (b) Vapor compression cycle pressure–enthalpy (P–h) diagram.](image-url)
Proportional-Integral-Derivative (PID). Since electronic controllers can be tuned, problems with mechanical devices such as valve hunting can be overcome. In (Outtagarts et al., 1997) the authors use a simplified identified model for the evaporator, and then compare PID and optimal control algorithms. Similarly in (Finn and Doyle, 2000) the authors design a PID controller based on a simplified identified model for the evaporator, and compare PID control to the TEV.

Design of EEV control algorithms is complicated by the inherent nonlinear dynamics, leading many researchers to suggest that scheduling of controller gains is necessary to effective operation (Finn and Doyle, 2000; Outtagarts et al., 1997). Additionally, the work of Rasmussen et al. (2006) implemented a cascaded control architecture, wherein an inner loop regulated the refrigerant mass flow to a setpoint generated by a linear PID controller seeking to regulate superheat; this approach significantly linearized the response of the evaporator to the controller, although it requires a means to measure refrigerant flow, which is not typically available in refrigeration systems. A method for characterizing and cancelling static nonlinearities (such as that found in EEVs) is presented in (Singhal and Salsbury, 2007). A feedback linearization approach to HVAC control that required a system model was also presented in (He and Asada, 2003). The interested reader can find a survey of linearization through feedback in (Guardabassi and Savaresi, 2001).

In (Elliott et al., 2009) the authors proposed a Hybrid Expansion Valve (HEV) that regulates fluid flow based on both pressure and superheat measurements, resulting in superior transient regulation. The novel combination of mechanical and electronic regulation mechanisms offers several advantages over traditional EEVs.

Fig. 2 – Nonlinearity of response for EEV, showing (a) mass flow as function of valve position (static valve map) and (b) superheat response to step change in valve position for high and low flows.

Fig. 3 – Linearized valve response, showing (a) mass flow as function of valve position (static valve map) and (b) superheat response to step change in valve position for high and low flows. Note the difference in speed of response, even though steady state response is very similar.
advantages including greater anticipated device longevity due to significantly decreased electronic actuation. This approach also partially compensates for system nonlinearities; this is deemed a significant improvement, since the ability to compensate for system nonlinearities could greatly simplify the control design task, and lead to better performance and efficiency over a wider range of operating conditions. The nonlinearity compensation is due to the cascaded control structure of the HEV; similar results can also be replicated with a traditional EEV, which is in general not a linear actuator, as detailed below.

Inspection of refrigerant mass flow as a function of valve position for a typical EEV reveals a nonlinear relationship, as shown in Fig. 2(a). Fig. 2(b) shows superheat responses to EEV step changes for high refrigerant flow and low refrigerant flow conditions. This is caused by the same step change causing different changes in flow depending on the valve position. This nonlinearity in the EEV means that adequate superheat control is much more difficult to achieve for this device, and illustrates why a gain scheduling approach based upon the different flow conditions is frequently used.

Alternatively, the controller can be “de-tuned,” sacrificing performance for stability over all operating regions due to smaller control gains; however, this risks losing superheat during system transients. Using a static inversion mapping between command signal and valve position (Juricic et al., 1986), refrigerant flow was linearized, similar to the work presented in (Singhal and Salsbury, 2007). This resulted in a much smaller difference in step response for the different conditions, although differences in dynamic response are still evident. See Fig. 3.

Characterizing the complete effect of valve nonlinearities on system behavior requires dynamic models of the complex two-phase flow dynamics of the heat exchangers. This has been the subject of significant research (e.g. Bendapudi and Braun, 2002; Bourdouze et al., 1998), and a complete presentation of evaporator dynamics is outside the scope of this paper. However, to facilitate understanding, the resulting models from such an effort are provided here. The governing equations for the evaporator and condenser are based on the conservation of mass and energy. Standard assumptions are applied, such as isobaric conditions in the heat exchangers, isenthalpic expansion through the valve, and adiabatic compression. Using a discretized modeling approach for the evaporator and condenser results in high order nonlinear models that capture the salient transient behavior, and approximates the distributed nature of the heat exchanger parameters. After the application of standard linearization and model reduction techniques, approximate second order transfer function models can be obtained of the following form:

\[ P_1(s) = \frac{T_{int}(s)}{U_{valve}(s)} \frac{k_1(s + a_1)(s + a_2)}{s^2 + 2\xi_0\omega_n s + \omega_n^2} \]

\[ P_2(s) = \frac{P_0(s)}{U_{valve}(s)} = \frac{k_2(s + b_1)(s + b_2)}{s^2 + 2\xi_0\omega_n s + \omega_n^2} \]

(1)

Selecting particular set of operating conditions (ambient temperatures/humidity, external flow rates, compressor speed, etc.) results in the following range of model parameters over the nominal range of the expansion valve:

\[ a_1 \in [-0.39, -0.60], \quad a_2 \in [0.48, 0.33], \]
\[ b_1 \in [0.55, 1.34], \quad b_2 \in [0.20, 0.22], \]
\[ k_1 \in [0.14, 0.02], \quad k_2 \in [0.61, 0.04], \]
\[ \xi_0 \in [1.00, 0.92], \quad \omega_n \in [0.62, 0.92]. \]

(2)

The frequency response for the full order models is depicted in Fig. 4. These models show only moderate variation in zero

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**Fig. 4 – Bode plots of the nonlinear EEV to superheat transfer function for different flow rates. As the flow rate increases, the steady state gain decreases.**

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**Fig. 5 – Cascaded control loops.**
Fig. 6 – Steady state values of transfer functions (a) $H(s)$ – mass flow to pressure and (b) $G(s)$ – mass flow to superheat. Both are shown as functions of refrigerant flow.

location, natural frequency, and damping ratio. However, the steady state gains of the system responses exhibit more than an order of magnitude variations as $P_1(0) = 7.0 \cdot 0.4$ and $P_2(0) = 18 \cdot 1.2$. These results suggest that if the refrigerant flow is a linear function of valve command, the resulting superheat response will be substantially linearized. This may render the use of gain scheduling unnecessary, since robust control techniques can then be employed without unacceptable decreases in control performance. However, since EEVs will not in general be linear in flow characteristics, and since mass flow measurements may not be available for development of static inversion maps, a method for inherently linearizing the flow (and hence the superheat response) is needed that does not require time intensive experimentation or testing. The cascaded control approach proposed herein meets these requirements, and only requires one additional sensor.

3. Cascaded control

The cascaded approach to superheat control consists of two nested control loops, as shown in Fig. 5. The inner, “fast” loop uses a proportional controller with gain $K_F$ that seeks to regulate the evaporator pressure to a setpoint ($P_{\text{SET}}$) generated by an outer, “slow” controller $C(s)$. This pressure setpoint is chosen by the controller $C(s)$ to regulate evaporator superheat to a user-defined setpoint. The relationship between valve position $v$ and mass flow $m$ is treated as a nonlinear gain function $K_M(v)$, and is a characteristic of the actuator used. The transfer functions $G(s)$ and $H(s)$ are the dynamic relationships from mass flow to superheat and pressure, respectively. Note that the only measurements necessary for implementation are refrigerant temperature at the outlet of the evaporator and evaporator pressure; since the refrigerant at the evaporator inlet is two-phase, its temperature can be found from pressure measurements using a lookup table. Alternatively, the inlet temperature can be measured directly.

The fast inner loop block diagram can be reduced to a transfer function $Q(s)$:

$$Q(s) = \frac{SH}{P_{\text{SET}}} = \frac{K_F K_M(v) G(s)}{1 + K_F K_M(v) H(s)}$$

Assuming stability allows invocation of the final value theorem, which gives a value for the steady state gain of $Q$:

$$Q(0) = \frac{SH_{\text{final}}}{P_{\text{SET}}} = \frac{K_F K_M(v) G(0)}{1 + K_F K_M(v) H(0)} = \frac{K(v) G(0)}{1 + K(v) H(0)}$$

Thus, if the steady state gains of $G$ and $H$ vary in the same direction as mass flow changes, e.g., they both decrease with increasing mass flows, then the variation of $Q(0)$ as mass flow changes will be minimized as $K(v)$ becomes larger. Experimental evaluation of the same EEV used earlier gave the following result, shown in Fig. 6.

Clearly, the steady state responses of both $G(s)$ and $H(s)$ decrease with increasing mass flow. From Fig. 6, a set of least squares curve fits was generated to give the relationships between valve position and mass flow, and mass flow to pressure and superheat. Using these functions in Eq. (4) gives the curves seen in Fig. 7. This figure shows the steady state gains for $Q(s)$ ($P_{\text{SET}}$ to SH) and $K_M(v) G(s)$ ($v$ to SH) from their respective inputs. There is a much more significant difference in the plant gain for the more traditional EEV to superheat control paradigm, than for the proposed $P_{\text{SET}}$ to superheat cascaded control architecture. Note that both the EEV and $P_{\text{SET}}$

![Fig. 7 - Steady state superheat gain as a function of refrigerant flow for EEV and cascaded loop.](image-url)
values were scaled for this analysis so that the maximum for each value is 1; this allows an even comparison for the two paradigms.

4. Implementation of architecture

A cascaded control loop can effectively compensate for inherent valve nonlinearities, but hardware limitations play a critical role in the relative success in practice. The first implementation was with the hybrid expansion valve (HEV), a device detailed in (Elliott et al., 2009). This device consists of a mechanically adjustable pressure-regulating valve with an electronically controlled stepper motor adjusting the desired pressure; an electronic controller changes the pressure “setpoint” to regulate evaporator superheat. HEV construction can be seen in Fig. 8; this is the same control architecture as the cascaded loop, but with the fast acting valve serving as a mechanical version of the inner control loop. The pressure “setpoint” $P_{SET}$ is generated by the position of the stepper motor, which is calculated by an electronic controller in order to regulate superheat to a desired value. The diaphragm that regulates pressure in the valve can be modeled as a linear spring, resulting in a proportional controller with fixed gain $K_F$.

The product of the gain and the pressure error is the position of the needle valve; this signal (designated $v$) is not known.

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**Fig. 8** – Hybrid expansion valve (HEV) construction.

**Fig. 9** – Relationship between unknown internal HEV gains and measurable quantities.
The mass flow rate through the needle valve is assumed to be a static, nonlinear function of the valve position, since the valve actuator dynamics are much faster than the evaporator dynamics (i.e., pressure and superheat). Thus the mass flow is a product of the valve position \( v \) and a gain \( K_M(v) \), which is a nonlinear function of \( v \). As before, \( G(s) \) and \( H(s) \) are nonlinear transfer functions from mass flow to superheat and pressure, respectively.

As noted in (Elliott et al., 2009), the first advantage of the HEV is that the stepper motor used for electronic control requires much less actuation than the EEV, since much of the control effort is performed by the valve diaphragm and spring. The second advantage is that the mechanized feedback loop built into the valve mechanism affords an extremely fast response. This gives the HEV superior transient regulation and disturbance rejection compared to traditional control methods. As will be shown later, experimentation with the HEV showed only minimal differences between superheat responses at low and high refrigerant flows, which along with the fast pressure response, suggests that the HEV features very high gains, corresponding to a stiff spring and diaphragm, and therefore a very large internal gain \( K_F \) (see Fig. 5).

While the HEV successfully limits the difference seen between high and low flow step responses, it also requires more hardware to implement than a typical EEV. In addition, the HEV’s diaphragm has the potential for fatigue failure over time. These implementation issues suggest the use of a standard EEV in combination with the cascaded loop. In order to recreate a similar performance using the EEV in a cascaded loop, the internal HEV gains would need to be found. Since there is no way of knowing what the actual valve position \( v \) is in the HEV, the loop gains \( K_T \) and \( K_M(v) \) cannot be obtained independently. Measurement of the refrigerant mass flow is available, however, as is the evaporator pressure and the HEV position. Therefore, if a relationship between stepper motor position and the mechanical pressure setpoint can be found, then the total gain \( K_{FM} \) (product of \( K_T \) and \( K_M(v) \)) can be calculated from measured quantities. This is illustrated in Fig. 9.

From this block diagram, the following relationship can be seen:

\[
\dot{m} = K_{FM} (K_UU - P_{evap})
\]  

(5)

A series of tests was performed for the HEV and cascaded EEV where different positions and mass flows were recorded. Since for all these tests, \( P_{evap}, U \), and \( \dot{m} \) are measured, a least squares regression can be performed on the data to yield the values of \( K_U \) and \( K_{FM} \). The value of \( K_U \) for the HEV was calculated to be 4.46 kPa per % of HEV range; this signifies that if, for example, the HEV is set to 50%, the mechanical pressure setpoint will be 223 kPa.

In the implementation of the EEV as the cascaded loop actuator, a proportional gain \( K_F \) of 0.22 was used. A larger gain was not useable, since the delays and slow rate limit of the EEV rendered the closed loop plant unstable with a higher gain. The values of \( K_{FM} \) for both HEV and cascaded EEV are plotted in Fig. 10, and are shown varying as the pressure setpoint changes, since the needle valve position is not available for measurement in the HEV. The higher gain shown by the HEV means that the HEV will have better response and more nonlinearity compensation than the EEV, but the high gain results in large, fast valve movements.

Since the proportional gain used with the EEV cannot be high enough to give an internal loop gain as high as that of the

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Fig. 10 – Internal gain \( K_{FM} \) for HEV and cascaded control loop with EEV. \( P_{SET} \) is scaled for easy comparison.

Fig. 11 – Superheat step responses at low and high flows for (a) HEV and (b) cascaded control with EEV.
HEV, the EEV in cascaded configuration will not give the same degree of nonlinearity compensation as the HEV; this is illustrated in Fig. 11. The HEV (Fig. 11a) has a steady state difference of approximately 0.5 K between high and low flows; the EEV with cascaded control has a difference of about 2.5 K. This is a significant improvement over the original case shown in Fig. 2, and shows that the cascaded control loop does indeed partially compensate for the nonlinearities of the system; additionally, the system exhibits much faster dynamic changes to the step response.

These data show that the HEV’s significantly higher gains contribute to a much higher degree of nonlinearity compensation than that of the cascaded EEV. However, attempting to recreate the HEV’s performance by using its gains on the EEV-based cascaded loop resulted in an unstable system. This is due to the inherent delay and relatively slow response of the EEV. An actuator that responds slowly is said to have a low bandwidth; this places a physical limitation on the control schemes that can be implemented with the actuator. In this case, the EEV’s bandwidth limits the gains that can be used in the internal proportional controller. Other research efforts implemented a micro-electrical-mechanical (MEMS) based expansion valve with the cascaded control structure (Elliot and Rasmussen, in press). This actuator, which features a much higher bandwidth than the standard EEV, promises to combine the ease of implementation of the standard EEV with the high gains – thus greater nonlinearity compensation and faster response – of the HEV.

5. Conclusions

Superheat control is an important problem in refrigeration and air conditioning systems; since good superheat regulation allows for more efficient operation of these systems, an improvement in this control can have a significant impact on worldwide energy usage. The nonlinearities present in these systems make control difficult, however. Experimental results showed that linearizing the flow rate as a function of valve position will linearize the superheat response; however, since EEVs are generally not linear and mass flow measurements may not always be available, this may not be a viable solution for most applications. This paper proposed a cascaded control algorithm that eliminates most of the nonlinearity of the response between algorithm-generated control input and the superheat output, while only requiring pressure and temperature measurements. This reduces the need for gain scheduling or other advanced control algorithms, and takes advantage of system measurements widely found in HVAC&R applications. The cascaded control approach was implemented using both an EEV and a “hybrid” expansion valve wherein the pressure regulation is handled mechanically. The mechanical pressure regulator functions much more quickly, i.e., it has a very large proportional gain, due to the stiff spring and diaphragm, and thus reduces nonlinearities. Using the EEV enables reduction of nonlinearities, but the same level of performance cannot be obtained due to inherent limitations – low bandwidth – of the actuator. This suggests the use of a faster actuator will yield better results, and enable successful implementation of a digital-only control mechanism.

References


