This paper examines the perennial problem of evaporator superheat control. While standard mechanical devices can operate effectively under design conditions, many behave poorly as conditions vary or under transient operation, resulting in degraded system performance, such as thermostatic expansion valve (TEV) hunting. Technological advances enabling electronic control help alleviate these problems by allowing more sophisticated control approaches to regulating superheat—for example, electronic expansion valves (EEVs). However, poorly tuned EEVs can still exhibit undesirable behavior, and frequent valve adjustments raise concerns about device longevity. In this work, we propose a cascaded control approach, which regulates evaporator pressure and superheat and is achieved with a feedback control device that uses a hybrid of mechanical and electronic feedback. Analysis of the fundamental dynamic behavior of evaporator superheat motivates this approach, while experimental evaluation of two separate systems demonstrates the efficacy of the approach as compared to standard control devices, such as TEV and EEV.

INTRODUCTION

Approximately 40% of all U.S. energy is used for heating, ventilation, and air conditioning (HVAC) (USDOE 2006), and air conditioning remains the largest source of peak electrical demand. Improving the efficiency of these systems has the potential for significant economic and environmental impact but requires refining individual component designs and ensuring effective control during transient operation. This paper considers the fundamental problem of evaporator superheat control in a basic vapor compression cycle. These cycles are used for a majority of household, automotive, and industrial air conditioning and refrigeration applications.

The ideal vapor compression cycle consists of four processes: (1) isentropic compression in a compressor, (2) isobaric heat rejection in a condenser, (3) isenthalpic expansion in an expansion valve, and (4) isobaric heat absorption in an evaporator. This process and the associated four components are shown below in Figures 1 and 2. Because the heat-transfer coefficient between the refrigerant and the evaporator walls is much higher for liquid refrigerant than for vaporized refrigerant, the two-phase portion of the evaporator provides virtually all of the cooling capacity of the system. Therefore, to maximize capacity of this system for a cooling application, the portion of the evaporator with two-phase flow needs to be maximized. However, to ensure safe and reliable operation of the compressor, the fluid entering the compressor must be completely vaporized. Systems with an accumulator at the evaporator exit can ensure safe operation of the compressor while maximizing the evaporator’s performance. For systems
without an accumulator, a generally acceptable compromise is for the fluid to be 5°C (9°F) above the saturation temperature. This is generally referred to as 5°C (9°F) of superheat.

The 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 tube, where the refrigerant flows through a reduction in diameter. Alternative passive control devices include a thermostatic expansion valve (TEV) or a pressure regulating device also known as an automatic expansion valve (AEV). Actively controlled valves include electronic expansion valves (EEV), where the valve opening is controlled by a stepper motor, and solenoid driven proportional control valves.

The literature examining superheat control largely consists of examining TEV- or EEV-controlled evaporators. The principal problem associated with TEV-controlled evaporators is valve hunting, a condition characterized by oscillations in the length of two-phase flow in the evaporator and, thus, oscillations in the amount of superheated vapor at the evaporator exit. This condition was first qualitatively documented in 1963 by Zahn (1963). Wedekind and
Stoecker (1966) published data demonstrating this phenomenon, and Najork (1973) attempted to give optimal parameter settings to minimize hunting. Broersen and van der Jagt (1980) used a mean void fraction model of the evaporator to show that valve hunting is caused by the interaction between the TEV and evaporator dynamics (Broersen 1980), and three years later Broersen and ten Napel (1983) presented an identified model of the TEV’s sensing bulb dynamics. Gruhle and Isermann (1985) used a discretized model of the evaporator to show similar results and contrasted the TEV with the proportional-integral (PI) controlled valve. Other authors also develop TEV and evaporator models for dynamic analysis but provide no validation (James and James 1987; Ibrahim 2001). Although recent research continues to analyze and explain the instabilities of TEV-controlled evaporators (Zhijiu et al. 2002), this condition is usually resolved by adjusting the valve parameters, resulting in decreased system performance.

Although comparatively fewer publications examine EEV-controlled evaporators, these are appearing with increasing frequency. For example, Outtagarts et al. (1997) use a simplified identified model for the evaporator and then compare proportional–integral–derivative (PID) and optimal control algorithms. Similarly, Finn and Doyle (2000) design a PID controller based on a simplified identified model for the evaporator and compare PID control to the TEV. While the EEV offers the capability for more sophisticated control strategies, this approach requires frequent valve adjustments that may shorten the lifetime of the device. Moreover, failure of either electronic or mechanical portions of the EEV renders the system incapable of metering the refrigerant flow and could lead to compressor failure. Design of the EEV control algorithms is further complicated by the inherent nonlinear dynamics, leading many researchers to suggest that scheduling of controller gains is necessary for effective operation (Finn and Doyle 2000; Outtagarts et al. 1997).

This paper seeks to overcome these difficulties using a device 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, including greater anticipated longevity due to significantly decreased electronic actuation. Preliminary tests also indicate that this approach partially compensates for system nonlinearities and, with some modifications, this capability could be improved further. This is also deemed a significant improvement, as the current literature consistently emphasizes the need to schedule controller gains as the operating condition varies. 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 remainder of this paper is organized as follows: a background section gives an overview of common expansion valve technologies and discusses the impact of various transient disturbances on evaporator superheat; an alternative expansion valve device is then proposed, wherein a hybrid of mechanical and electronic feedback devices are used to implement a cascaded control approach, regulating evaporator pressure, and superheat; experimental evaluation of the hybrid expansion valve (HEV) demonstrates the efficacy of the device as compared to commercially standard AEV, TEV, and EEV technologies; in conclusion, the contributions of this paper are summarized.

BACKGROUND

Expansion Valve Technologies

Fixed Expansion Valve. Capillary tubes or fixed “short-tube” orifice valves are examples of the simplest type of expansion device. A fixed length of capillary tubing with a small diameter, or a short tube with a fixed orifice, is the means used to induce the pressure drop for the expansion
Because there are no moving parts, the device results in a relatively inflexible rate of refrigerant flow. Proper system operation requires using the correct amount of refrigerant charge. However, even with proper charge, these devices may at times starve or flood the evaporator; no mechanical or electronic feedback control is used.

**Automatic Expansion Valve (AEV).** The AEV is better described as a constant-pressure expansion valve (ASHRAE 2009). The valve consists of a diaphragm that opens/closes a valve mechanism based on a force balance of the evaporator pressure, atmospheric pressure, and an externally adjustable spring force. These valves are most often used for systems with little variation in the cooling load and where constant evaporator temperature (i.e., evaporator pressure) is acceptable or required. However, because the valve meters refrigerant to regulate evaporator pressure, not superheat, the AEV has a tendency to starve the evaporator during high load conditions (inefficient operation) and flood the evaporator during low load conditions (potentially damaging the compressor). Figure 3 shows a diagram of the AEV’s construction.

**Thermostatic Expansion Valve (TEV).** The TEV features a refrigerant-filled sensing bulb placed at the outlet of the evaporator, as shown in Figure 4. The pressure in this bulb is the refrigerant saturation pressure corresponding to the evaporator outlet temperature. The valve opening of the TEV is determined by a force balance of the evaporator inlet pressure, the pressure in the sensing bulb, and an adjustable spring force. This mechanism results in a mechanical feedback loop that controls the mass flow rate of refrigerant in an effort to maintain a constant pressure difference, or superheat. The use of this valve is widespread despite the well-known problem of valve hunting, where the valve opening continually oscillates due to the interaction of the valve and evaporator dynamics.

**The Electronic Expansion Valve (EEV).** Typical EEVs use a stepper motor to open/close the valve (Figure 5). The stepper motor requires external control electronics, and is notably more complex and expensive than the passive flow control devices presented previously, but does allow for external active control of the refrigerant flow rate. Variations in the mode of actuation exist; for example, some models for EEVs are based on solenoid actuators. These devices typically use PID-type control algorithms to regulate superheat.

![Figure 3. Diagram of AEV construction.](image-url)
Transient and Steady-State Disturbances to Superheat Regulation

Vapor compression systems are used in a wide variety of settings: vehicle air conditioning, supermarket refrigeration, building climate control, etc. In each case, the system is consistently and, in some cases continuously, subjected to transient disturbances. Although the scenarios can be extremely different, the effects of these induced transients from the perspective of superheat regulation are similar. For the purposes of this paper, these disturbances are classified as either internal or external.

**Internal Disturbances.** Transients that directly and immediately alter the flow rate or thermodynamic states of the primary fluid (refrigerant) are classified as internal disturbances. For simple vapor compression systems, compressors switching ON/OFF or varying in speed/displacement are the common instances of this class of disturbances. However, in more complex
systems (e.g., multiple evaporator systems), the effects of solenoid valves, discharge valves, head pressure control valves, etc., are also included in this group. Internal disturbances affect evaporator superheat most strongly and most quickly by altering the saturation pressure. Although the evaporator outlet temperature does vary in response to an internal disturbance, this effect is secondary and generally occurs on a markedly slower time scale.

For example, consider a step decrease in compressor speed (Figure 6) while the EEV’s position is held constant with a constant control signal. Note that the evaporator pressure responds quickly, raising the saturation temperature by almost 4°C (7.2°F). Although the outlet temperature also changes, the response is noticeably slower and less pronounced—a change of less than 1°C (1.8°F). This step response, and that shown in Figure 7, was performed upon a small water chiller (approximately 1.25 kW [0.33 ton] cooling capacity) described later in the “Experimental Evaluation.”

![Figure 6](image1.png)

**Figure 6.** Evaporator temperature response to step decrease in compressor speed in (a) I-P units and (b) SI units. The step change in compressor speed occurs at 100 s and is denoted with a dashed line.
**External Disturbances.** Transient changes external to the evaporator include variations in the flow rate of secondary fluid (e.g., change in fan/pump speed), the temperature of the secondary fluid (e.g., change in cabin/chamber temperature), etc. Some of these transients evolve on a slow time scale, such as the gradual temperature changes of the conditioned space. Other transients can be abrupt, such as ON/OFF fan operation during start up and shut down. However, because these disturbances only affect the refrigerant pressures/temperatures/flow rates indirectly through mechanisms of convection and conduction, their response is still much slower and less severe than that of internal disturbances.

The superheat can be altered by changing either the evaporator saturation pressure or the outlet temperature. For example, consider a ramped change in flow rate of the secondary fluid across the evaporator (i.e., change in fan/pump speed) shown in Figure 7 (again the EEV is held

![Figure 7](image_url)
constant). This increase in secondary flow rate causes an increase in evaporator heat transfer (cooling), which results in an increase in the evaporator outlet refrigerant temperature.

Note that while the outlet temperature increases, the saturation temperature remains virtually constant, since the evaporator pressure is not affected by this change. The result of the external disturbance is a slow increase in superheat but not evaporator pressure. These results suggest that if a fast controller can be implemented to regulate evaporator pressure and reject internal disturbances, then a secondary, slower controller may be used to regulate evaporator superheat in the presence of external disturbances. This insight is at the heart of the proposed control approach.

HYBRID APPROACH TO SUPERHEAT CONTROL

Operating Principles

Current refrigerant expansion valves utilize inexpensive passive mechanical devices (e.g., thermostatic expansion valve) or more expensive actively controlled devices (e.g., electronic expansion valves) for regulating superheat. While these are generally effective in maintaining constant superheat level despite slowly changing external disturbances, they are less effective in the presence of fast internal disturbances that strongly affect saturation pressure and, thus, superheat. While a pressure-regulating expansion device, such as an automatic expansion valve, would minimize the effects of internal disturbances, it would fail to regulate superheat levels over a wide range of conditions or in the presence of external disturbances.

The proposed solution to this dilemma is a device with two control mechanisms, as shown in Figure 8: The inner control mechanism regulates evaporator pressure, while the outer mechanism adjusts the desired pressure to regulate evaporator superheat. Improved transient regulation is then achieved due to (1) the use of two measured quantities, which addresses the two sources of superheat variation separately, and (2) the collocation of sensor/actuator for the pressure control loop, a common best-practice for feedback control design.

Embodiment

In its simplest embodiment, this valve could be viewed as a variation of the standard AEV, which uses a metal diaphragm to meter refrigerant flow to regulate the evaporator pressure. A spring can be used to manually apply an external force on the diaphragm so that technicians can adjust the desired pressure during installation. By augmenting the spring with an electronic motor with linear actuation, a device is created with both passive mechanical regulation and

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**Figure 8.** Schematic of proposed valve control scheme.
active electronic regulation. In passive operation, the diaphragm position is affected by the pressure of the fluid flowing past the valve. In active operation, an electronic motor is used to apply an external force to the diaphragm, altering the valve position until the desired pressure balance has been achieved. This device is referred to hereinafter as the hybrid express valve (HEV), a reference to the hybrid of mechanical (passive) and electronic (active) actuating capabilities. Figure 9 shows a diagram of the HEV’s construction.

Advantages

This proposed construction and hybrid approach offers several advantages over traditional direct control of superheat. As with standard AEVs, the valve’s response to pressure fluctuations is virtually instantaneous, resulting in tight regulation of pressure in the presence of internal transient disturbances. Since the AEV is a pressure regulating mechanical device, its diaphragm adjusts immediately to keep the evaporator pressure constant, and a change in the stepper motor is not required. This implies that a slow controller with minimal control action is acceptable for superheat control, since the fast dynamics associated with internal disturbances are regulated by the mechanical diaphragm.

Figure 9. Hybrid expansion valve design.
Additionally, external disturbances that the mechanical device cannot reject are typically slow in nature, and a nonaggressive approach to superheat control is sufficient. The control architecture thus takes advantage of a richer information stream—pressure and superheat rather than superheat alone—to minimize the control effort required by the electronic actuator and increase the longevity of the expansion device. In the following section, experimental results are shown that compare the HEV to a traditional expansion-valve technologies and clearly demonstrate the advantages of the proposed approach. These advantages are summarized as follows:

HEV vs. TEV: The fast regulation of pressure virtually eliminates the risk of valve hunting, a problem that plagues TEV-controlled systems and results in longer start-up times and inefficient operation.

HEV vs. AEV: The addition of the external electronic actuation overcomes the fundamental problem of starving or flooding the evaporator during off-design conditions.

HEV vs. EEV: The HEV requires significantly less control effort due to the properties of the mechanical pressure control and, hence, enjoys a longer motor service life than the traditional EEV. Moreover, in the event of electronic motor/control failure, the passive mechanical device continues to operate, providing a nominal level of regulation.

EXPERIMENTAL EVALUATION

Initial experiments to verify the proposed control scheme were conducted on two separate experimental systems. The first system is a three-ton residential air-conditioning system with a TEV-controlled evaporator. This system has a two-stage compressor, and variable-speed fans. The second system is a custom built, multiple-evaporator small-scale water chiller. This experimental setup features half-ton EEV-controlled evaporators, variable-speed compressor, and variable water flow. Note that, for the work detailed in this paper, only one evaporator is used. Both systems are equipped with a complete set of sensors measuring pressures, temperatures, and mass flow rates at the various points of the thermodynamic cycle. Additionally, both systems are set up such that different expansion devices can be used for comparison on the same evaporator. Figure 10 shows a schematic typical of both systems and displaying the sensors and actuators used in the research presented in this paper.

The governing philosophy of the following tests is to provide a control performance comparison of the various superheat control mechanisms. For each comparison, different expansion devices were tested with the same protocol on the same test system. Tracking step changes in the superheat setpoint provides a comparison of the control effort required to achieve the same result; the controllers were tuned so that similar settling times were achieved for each method. Tests were also run to investigate the disturbance rejection performance of each method for both step changes in the compressor speed (internal disturbance) and step changes in the flow rate of the secondary fluid—i.e., water/air (external disturbances).

Residential System—HEV vs. TEV and AEV

The residential system used for these experiments is a three-ton unit with a variable-speed evaporator fan and a two-stage compressor. Due to the size of the system, the dynamics are considerably slower than those of the small water chiller mentioned previously. The multichannel evaporators used in these systems are precisely tuned with the expansion valve to ensure equal distribution of refrigerant among the channels. Since the nature of this experimental work requires modification of the fluid flow path, some maldistribution of refrigerant in the evaporator...
is expected. As a result of this, the superheat setpoints are set to a relatively high value (10°C or 18°F) in order to ensure safe operation.

Startup

A set of tests was run to display the system pull-down characteristics of the different valves (Figure 11). The HEV is able to bring superheat down to the setpoint of 10°C (18°F) within four minutes, as is the AEV without compensation. The TEV never settles into a setpoint but after 200 seconds simply hunts for the remainder of the test run. This test shows that the HEV performs pull-down and brings the system to a steady-state superheat setpoint. This suggests that the HEV can perform better in startup conditions than the more traditional purely mechanical expansion devices.

Note that, for all data plots, “normalized control effort” means the control signal divided by the actuator maximum—i.e., a normalized control effort of 1 is 100% of the actuator range. For the HEV, only stepper motor effort is considered; the action of the metal diaphragm in opening and closing the valve is not included. The limits upon the stepper motor travel are dictated by the spring; 0% corresponds to an uncompressed spring, and 100% corresponds to a completely compressed spring. For the EEV, the control effort is the valve position; 0% corresponds to a completely closed valve, and 100% corresponds to a completely open valve.

Fan Speed Change

In the test run for Figure 12, the evaporator fan speed was stepped from 45% to 60% of full flow, which introduced an external disturbance into the system. The HEV brings the superheat back to the required 10°C (18°F). The AEV simply allows superheat to increase, while keeping evaporator pressure constant. This reflects the HEV’s ability to regulate superheat in off-design conditions.
conditions, an ability lacking in the AEV. The TEV settles into a new setpoint, although it does hunt less in the higher flow condition.

**Compressor Stage Change**

For the next set of tests, the compressor was switched from stage one (low compression) to stage two (high compression). Figure 13 shows the results of this test. The HEV brings the superheat back to the setpoint of 10°C (18°F), while the AEV settles at a higher superheat (approximately 15°C [27°F]). The TEV actually loses superheat temporarily then returns to hunting, although at a slightly higher bias than before. Again, the ability of the HEV to regulate superheat under different conditions offers superior controllability in various conditions, which the AEV and TEV cannot match due to the lack of electronic feedback.

**Water-Chiller System—HEV vs. EEV**

The next set of tests was performed on the 1.75 kW (0.5 ton) water-chiller system. This system is intended to be a flexible bench-top experimental system and is therefore much smaller than a typical chiller and has a much faster response time. Additionally, this is a true variable refrigerant flow (VRF) system, since the compressor is variable speed. Thus, more or less refrigerant can be delivered as necessary. Internal disturbances are much more varied for a variable speed compressor, since any number of off-design conditions can be encountered. This provides an opportunity to compare HEV performance to traditional devices on a faster responding system with more frequent transient disturbances. In this more demanding setting, the advantages of the HEV are shown even more clearly, particularly compared with state-of-the-art EEV control.

The first test involves tracking step changes in the superheat setpoint. This test provides a means of tuning the controllers for two different control strategies: a traditional EEV-controlled evaporator (PID control algorithm) and an HEV-controlled evaporator using the proposed hybrid approach. By tuning the associated PID loops to achieve comparable performance, we ensure that the performance differences observed in subsequent tests are due to inherent capabilities of the actuators and are not the result of a particularly well/badly tuned control algorithm.

Figure 14 shows the results from this experiment: the superheat, normalized control effort, and evaporator pressure from the EEV test are on the left side, and the same information from the HEV is seen on the right. Both expansion valves exhibit similar step tracking response; as expected, the EEV requires more actuation changes to achieve this result. As before, the control effort is normalized to the maximum of the respective actuator. Note that the pressures from both tests have the same steady-state values.

In order to quantify the performance of the two controllers in subsequent tests, the following characterizations are used. First, the root mean square error (RMSE) gives a measure of the deviation from the setpoint over the test and is calculated as follows

\[
RMSE = \sqrt{\frac{1}{n} \sum_{i=1}^{n} (SH_i - SH_{set})^2}
\]  

(1)

where \(SH_i\) is the superheat at sample instant \(i\), \(SH_{set}\) is the superheat setpoint, and \(n\) is the total number of sample points. Next, the maximum absolute error (MAE) is calculated to give the maximum deviation from setpoint over all sample points:

\[
MAE = \max |SH_i - SH_{set}|
\]  

(2)
Figure 11. Compressor startup/system pull-down with (a) the HEV, (b) the TEV, and (d) the AEV. This figure also shows (c) the HEV control effort. The compressor was turned on at $t = 0$ s. (A) is in I-P units and (B) is in SI units.
Figure 12. Superheat disturbance rejection with (a) the HEV, (b) the TEV, and (d) the AEV. This figure also shows (c) the HEV control effort. The fan is stepped at 100 s for all tests. (A) is in I-P units and (B) is in SI units.
Figure 13. Compressor stage changes: (a) superheat response for HEV, (b) superheat response for TEV, (c) HEV control effort, and (d) superheat response for AEV. The time of compressor switches is denoted by dashed vertical lines in each plot. (A) is in I-P units, and (B) is in SI units.
Figure 14. Superheat setpoint step change comparison: superheat and setpoint for (a) EEV and (b) HEV; normalized control effort for (c) EEV and (d) HEV; and evaporator pressure for (e) EEV and (f) HEV. (A) is in I-P units, and (B) is in SI units.
Finally, the root mean square of the control effort (RMSU) gives a measure of the actuator effort required of the stepper motors in the EEV and HEV:

\[
RMSU = \sqrt{\frac{1}{n} \sum_{i=1}^{n} (u_i - u_{i-1})^2}
\]

For the next test, superheat is set to a constant value of 8°C (14.4°F), and a series of step changes in the compressor speed is implemented to test internal disturbance rejection of the control techniques. The two control paradigms are identical to those tested in the first experiment. The results are shown in Figure 15 and Table 1.

As is expected, the HEV achieves better disturbance rejection (RMSE and MAE improvements of 63.7% and 77.6%, respectively) with significantly less control effort than the EEV with PID control (69.2% improvement in RMSU), because the mechanical feedback inherent to the HEV keeps the evaporator pressure constant regardless of internal disturbances, with the pressure setpoint changing slowly to keep superheat constant. In other words, since the metal diaphragm performs the task of regulating evaporator pressure, a significant portion of the disturbance rejection is performed by the HEV’s mechanical feedback mechanism, relieving the stepper motor of the need to perform all of the work. Recall that the two controllers are tuned such that they have similar setpoint tracking characteristics; thus, the HEV is able to perform superheat control much more effectively and with much less stepper motor effort than an equivalent EEV/PID combination.

The third comparison test involves changing the water flow rate (external disturbance). Figure 16 and Table 2 show the results of this test. Although the improvement in disturbance rejection is more modest than in the previous test, the improvement in RMSU is still significant. The slow speed of the disturbance ensures that both controllers are able to regulate superheat effectively, but, by slowly changing the pressure setpoint, the HEV is able to achieve comparable results with significantly less effort.

The next test performed subjected both control paradigms to a periodic disturbance; the water flow and compressor speed were stepped in a cycle with a period of 100 s. Again, the HEV performed better than the EEV; the results are shown in Figure 17 and Table 3.

The final test performed involved subjecting the EEV and HEV to a series of random step inputs from the compressor and water flow; while this is perhaps not a realistic situation, it creates a worst-case test for the two control paradigms. The EEV performs noticeably worse than the EEV; indeed, superheat is even lost temporarily. The results of the test are shown in Figure 18 and Table 4.

**CONCLUSION**

The expansion valves that find the most widespread use are passive devices, which operate effectively only at the design conditions. Systems that operate over a wide range of operating conditions (e.g., military vehicles, residential HVAC) or that experience significant transient conditions (e.g., automotive, computing) require more intelligence in the metering control strategies and, thus, generally require external or active feedback control strategies. Although electronically controlled valves are currently available, these valves directly control valve position electronically. Thus any adjustment in valve position requires electronic action, and frequent use of such actuators can lead to failure.

The proposed valve would permit passive regulation of refrigerant flow during normal steady-state operating conditions, with the added capability of active control during abnormal or
Figure 15. Internal disturbance rejection comparison: superheat for (a) EEV and (b) HEV; normalized compressor speed for (c) EEV and (d) HEV; and normalized control effort for (e) EEV and (f) HEV. (A) is in I-P units, and (B) is in SI units.
Figure 16. External disturbance rejection comparison: superheat for (a) EEV and (b) HEV; normalized water flow for (c) EEV and (d) HEV; and normalized control effort for (e) EEV and (f) HEV. (A) is in I-P units, and (B) is in SI units.
Figure 17. Periodic disturbance rejection comparison: superheat for (a) EEV and (b) HEV; normalized water flow for (c) EEV and (d) HEV; and normalized control effort for (e) EEV and (f) HEV. (A) is in I-P units and (B) is in SI units.
Figure 18. Random disturbance rejection comparison: superheat for (a) EEV and (b) HEV; normalized water flow for (c) EEV and (d) HEV; and normalized control effort for (e) EEV and (f) HEV. (A) is in I-P units, and (B) is in SI units.
transient operating conditions. In particular, the proposed valve is shown to (1) provide transient performance superior to typical mechanical valves during start up or transient disturbance conditions, (2) be more robust to large changes in operating conditions than are typical mechanical valves, (3) require less actuator effort than purely electronic valves, resulting in longer installed use before failure, and (4) more robust to motor failure than purely electronic valves, since the mechanical components can continue to provide basic flow regulation despite electronic valve failure. The proposed valve has the potential for widespread use throughout the air-conditioning, refrigeration, and heat-pump industry as an alternative to other actively controlled valves, while preserving sufficient flexibility to allow for passive flow control operation.

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