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Evaporator Superheat Control With One Temperature Sensor Using Qualitative System Knowledge

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Abstract— **This paper proposes a novel method for superheat control using only a single temperature sensor at the outlet of the evaporator, while eliminating the need for a pressure sensor. An inner loop controls the outlet temperature and an outer control loop provides a reference set point, which is based on estimation of the evaporation pressure and suitable reference logic. The pressure is approximated as being linear and proportional to the opening degree of the expansion valve. This gain and the reference logic is based on calculation of the variance in the outlet temperature, which have shown to increase at low superheat. The parameters in the proposed controller structure can automatically be chosen based on two open loop tests. Results from tests on two different refrigeration systems indicate that the proposed controller can control the evaporator superheat to a low level giving close to optimal filling of the evaporator, with only one temperature sensor. No a priori model knowledge was used and it is anticipated that the method is applicable on a wide variety of refrigeration systems.**

I. INTRODUCTION

Refrigeration systems normally operate by continuous vaporization and compression of refrigerant. This process is maintained by a valve, an evaporator, a compressor and a condenser, and this setup remains to a considerable extent the same in most refrigeration systems. The details of the vapor compression type refrigeration process are not given here, but can be found in e.g. [1].

Refrigeration systems are typically controlled by decentralized control loops and evaporator superheat is controlled in one of these loops. Superheat control can be achieved by regulating the opening degree (OD) of the expansion valve. Superheating of the refrigerant beyond the evaporation temperature is important, since no superheat means that two-phase refrigerant will enter the compressor and increase the power comsumption and wear. This means that the flow through the valve must be kept a level, where all the refrigerant is evaporated before it reaches the compressor. At the same time, it is important to have as much two-phase refrigerant in the evaporator as possible, to increase the heat transfer and thus optimize the refrigeration process. So a key variable, which greatly effects the efficiency of a refrigeration system, is the superheat, which again is an indirect measure of the filling of the evaporator.

The heating, ventilating and air conditioning (HVAC) industry commonly use some variant of proportional-integral (PI) feedback control [2]. These controllers have traditionally been tuned by refrigeration and control specialists, due to the complexity and nonlinearity of the refrigeration process and the large number of different refrigeration system designs available. The problem is that the human operator often copies parameter values from any previous system in the hope that the new refrigeration system will work with these settings. However, each system is associated with different optimal working point conditions, sensor/actuator configurations and cooling demands. Furthermore, the tuning process can be time consuming and there is a risk of system damage, if the operator is not cautious. It is therefore desirable to automate the tuning process of controllers for refrigeration systems and/or implement adaptive algorithms.

Automatic tuning of PI/PID controllers have been treated in many books, see e.g. [3] and [4]. The relay method is used in [5] to obtain the ultimate frequency and gain, which is used to find PID controller parameters based on model knowledge. These parameters are compared with Zeigler-Nichols tuned parameters and model based gain scheduling is additionally employed to cope with the operating point dependent system gain. In [6], auto-tuners for PI/PID control of HVAC systems are designed based on a combination of relay and step tests. The auto-tuners show better performance than manual tuning and standard relay auto-tuning.

The response from valve OD to superheat is in general very nonlinear, making controller tuning difficult. The need for gain scheduling in [5] is eliminated in [7], by transferring the superheat to a referred variable. In both papers a cascaded control setup is utilized, where a flow meter is used to control the refrigerant mass flow in an inner loop. However, most refrigeration systems does not have such a sensor and [8] instead proposes a cascaded control, where evaporator pressure measurements are used in an inner loop to reduce the nonlinearities. Backstepping can also be used to design a nonlinear controller, as done in [9]. This controller can be made almost independent of the cooling capacity and therefore does not require any gain scheduling. Another possibility is to control the superheat with the compressor and the cooling capacity with the valve. In [10], backstepping is again used to derive a nonlinear controller. However, extensive model knowledge is required in both cases and some model parameters are only partly known and vary with the operating conditions, thus requiring adaptive methods for finding these parameters. These have been pursued in [10].

All the controllers mentioned so far require at least a temperature sensor and a pressure and/or a flow meter to control the filling of the evaporator. In this paper, we will present a novel control method capable of controlling the filling with only one temperature sensor placed at the outlet of the evaporator. This will make it easier to install and buy

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superheat controllers based on electronic valves. The method utilizes the fact that the variance of the outlet temperature increases when the evaporator is close to overflowing and this gives a fix point, where the gain, in a simple linear model relating the valve OD to the pressure, can be identified. The estimated pressure can then be converted into evaporation temperature and thus a reference for a simple PI controller for the outlet temperature. Furthermore, the reference is slowly decreased until the fix point is reached and then stepped back. This makes it possible to adaptively correct the gain in the linear model each time the fix point is reached and ensures that the system is continuously operated close to where the evaporator is fully filled (low superheat). In other words, qualitative system knowledge is used to identify when the filling of the evaporator is suitable and it has been shown in tests that the method works on two completely different refrigeration systems. Additionally, only two open loop tests are required to set the control parameters and these tests can be performed in an automated fashion. Another benefit of the proposed controller is that no a priori model knowledge is required, which is often the case when e.g. gain scheduling and nonlinear control design methods are used.

The structure of this paper is as follows. The two test refrigeration systems are first presented in Section II. Then, calculation of variance of the outlut temperature is shown in Section III, followed by a presentation of the control strategy in Section IV. Then, an adaptive pressure estimator is derived in V and the startup procedure is shown in Section VI. Finally, test results are presented in Section VII and conclusions are drawn in Section VIII.

II. System Description

The proposed superheat control method in this paper is designed for unknown vapor compression type refrigeration systems, where no a priori model knowledge is assumed. The method should work on a wide variety of setups and two different types of refrigeration systems have therefore been used for test. The first system is an air conditioning system and the second is a refrigeration system with a water tank and heater as load on the evaporator. Simplified drawings of these systems are shown in Fig. 1.

The air conditioning system in Fig. 1(a) has a four channel finned-tube evaporator and a Danfoss Ecoflow™valve. It is possible to control the OD of the valve and the distribution of flow into the individual pipes, however, the distribution is kept constant in this setup. Furthermore, it is possible to control the frequency of both the evaporator and condenser fans, and also the frequency of the fans between the cold room, the hot room and the outside. The compressor frequency is also controllable and sensors measure temperature and pressure at the indicated places.

The refrigeration system in Fig. 1(b) has an evaporator with water on the secondary side, which is connected to a water tank with controllable heater and pump. It is possible to control the OD of the electronic expansion valve and the frequency of the condenser fan. The compressor frequency is again controllable and sensors measure temperature and pressure at the indicated places. Both systems are monitored and controlled using the XPC toolbox for Simulink.

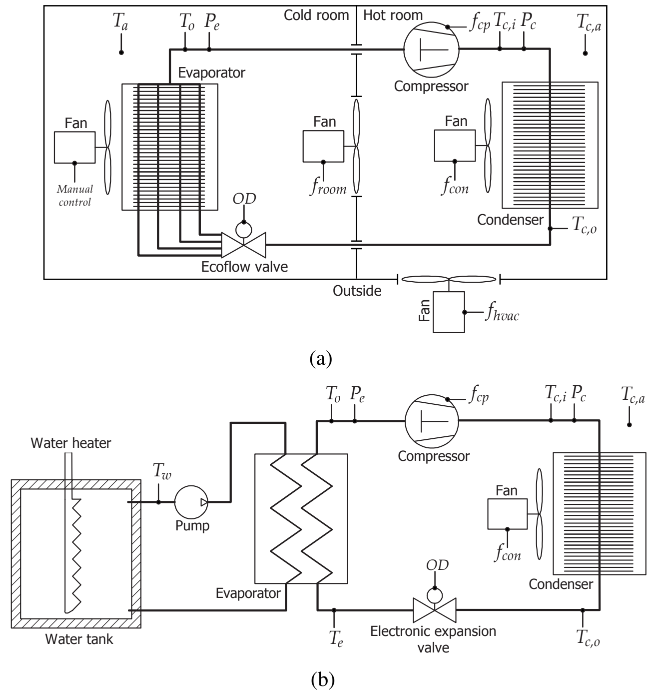

Fig. 1. Simplified drawings of the two available test systems. T, P and f are indicators for temperature sensors, pressure sensors and frequency control, respectively. Only T_o and OD are used for the superheat control and the other sensors are used for verification purposes. System (a) is an air conditioning system and system (b) is a refrigeration system with water on the secondary side of the evaporator.

Evaporator superheat T_{sh} is defined as the outlet temperature T_o minus the evaporation temperature T_e (evaporator saturation temperature). The evaporation temperature is normally measured indirectly by measuring the evaporation pressure P_e. We propose a control method, which does not require an direct or indirect measurement of T_e, but only the T_o measurement. Instead, qualitative system knowledge is used to calculate the variance on T_o to estimate P_e, which is further discussed in Section III. This makes this controller easier to install and buy, compared to other superheat controllers using electronic expansion valves, since we save a pressure sensor.

In the following it is assumed that the condenser pressure is controlled separately and that the compressor is running at constant frequency, which means that any change is considered as a disturbance.

III. Variance Calculation

An open loop test has been performed on each of the test systems, where the OD signal was increased slowly while outlet temperature T_o measurements were saved. By calculating the sample variance as

$$\sigma^2 = \frac{1}{n}\sum_{i=1}^{n}(x_i - \bar{x})^2,\ \bar{x} = \frac{1}{n}\sum_{i=1}^{n} x_i, \qquad (1)$$

where σ^2 is the sample variance using n samples, x_i is the i'th sample and $\bar{x}$ is the sample mean, then it is possible to get an estimate of the variance in the outlet temperature. Fig. 2 and 3 shows the test results using a five minute sample window on the air conditioning system and the refrigeration system, respectively. The system response is clearly different

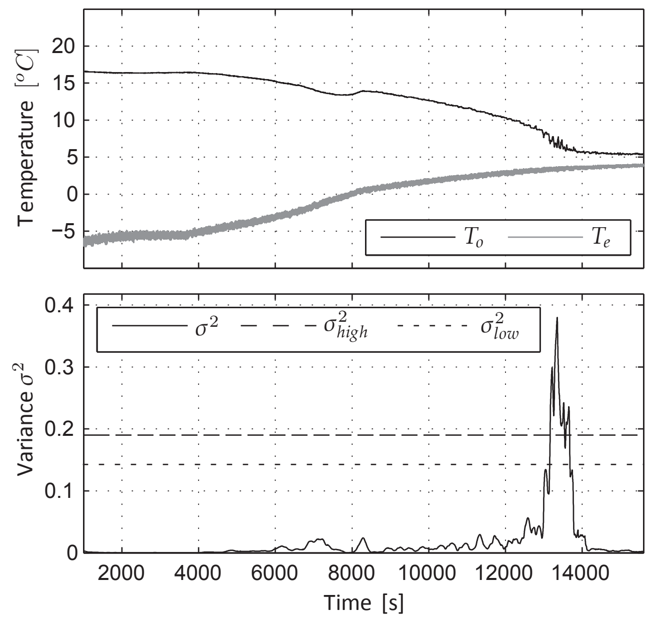

Fig. 2. Evaporator outlet temperature and variance during an OD sweep on the air conditioning system. Variance thresholds are also indicated.

between the two systems, however, the tests indicate, in both cases, that the variance increases considerably at low superheat and then decreases again when the evaporator is flooded. This increase in variance can be used to identify when the evaporator is nearly flooded and provides an alternative way of controlling the filling of the evaporator compared to conventional control.

IV. CONTROL STRATEGY

The control strategy is illustrated in Fig. 4. A simple PI feedback control is used in an inner loop to control the evaporator outlet temperature T_o and an outer loop provides the temperature controller with a suitable reference set point.

The Logic block in Fig. 4 controls the superheat reference, which is implemented so that it continuously decreases in temperature until the variance has increased to a predetermined variance level σ^2_{high}. Then it is stepped back and the cycle is repeated, so that the superheat is constantly kept at a low level despite a change in system load. A waiting period is introduced during startup, which prevents the reference from decreasing until the system has calmed down and the variance is below the hysteresis bound σ^2_{low}, see Fig. 2 for a definition of the variance levels. Furthermore, a step back in reference can only be made if the system has calmed down since last step, since a step will cause a temporary increase in variance. A larger step back in reference is taken if the system has not calmed down since the last step and the reference has decreased to its level from before the previous

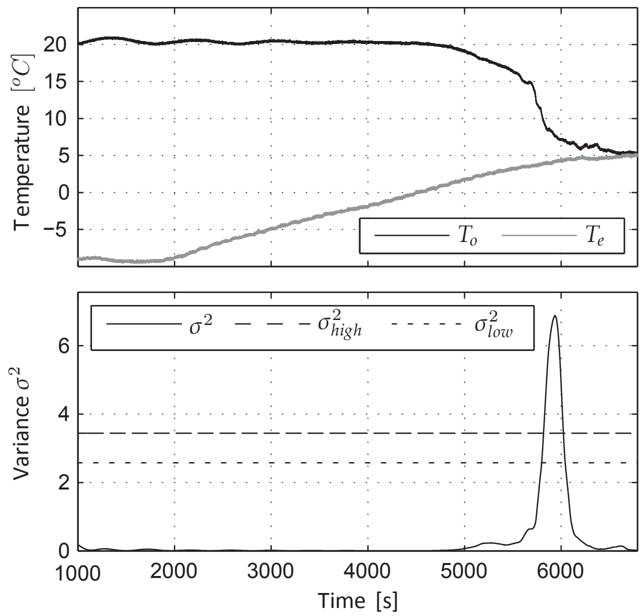

Fig. 3. Evaporator outlet temperature and variance during an OD sweep on the refrigeration system. Variance thresholds are also indicated.

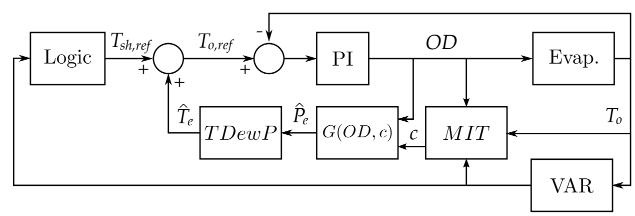

Fig. 4. Control structure for control of the evaporator outlet temperature. A suitable reference is found based on adaptive estimation of the evaporation temperature and superheat reference logic.

step. For further safety, the reference is also stepped back if the superheat reference goes below 1 degree.

The reference to the inner loop $T_{o,ref}$ is made by adding the superheat reference $T_{sh,ref}$ with an estimated evaporation temperature $\hat{T}_e$. The estimated evaporation temperature is based on an estimate of the evaporator pressure, which in steady state can be approximated as being proportional to the OD signal. The gain c from OD to $\hat{P}_e$ is adapted using the MIT rule (see e.g. [4]) and updated each time the reference logic brings the evaporator to a state where it is nearly flooded, which can be identified by an increase in variance. It is important to note that no pressure sensor is used in this setup.

The startup procedure should be made so that the control can start automatically and work on a wide variety of refrigeration systems. In Section VI it is explained how the controller can be tuned based on two open loop tests.

V. PRESSURE ESTIMATOR DESIGN AND ADAPTION

The fundamental concept of conservation of mass in physics (refrigerant is neither added nor removed from the system), implies that the mass flow rate $\dot{m}_v$ through a tube is constant (assuming incompressibility) and equal to the product of the density ρ, velocity v and cross-sectional area A,

$$\dot{m}_v = \rho v A. \tag{2}$$

If assuming laminar, inviscid and incompressible refrigerant mass flow rate through the expansion valve, then Bernoulli's equation furthermore states that

$$\frac{1}{2}v^2 + gz + \frac{P}{\rho} = k, \tag{3}$$

where g is the gravitational constant, z is the elevation, P is the pressure and k is a constant, which does not change across the valve. Combining (2) and (3), while isolating for the valve mass flow $\dot{m}_v$, gives

$$\frac{1}{2}\left(\frac{\dot{m}_v}{A_c\rho_l}\right)^2 + gz + \frac{P_c}{\rho_l} = \frac{1}{2}\left(\frac{\dot{m}_v}{A_e\rho_l}\right)^2 + gz + \frac{P_e}{\rho_l}$$

$$\dot{m}_v = \sqrt{P_c - P_e}\sqrt{\rho_l}C_v, \tag{4}$$

where P_c and P_e are the pressures in the condenser and the evaporator, ρ_l is the density of the liquid refrigerant, A_c and A_e are the cross-sectional area before and after the valve, and C_v is a collection of constants. Equation (4) is consistent with the result in e.g. [11] for a fully open expansion valve and C_v is also called the orifice coefficient. A valve with variable opening degree OD is added to (4). The valve OD is in most refrigeration systems linear going from zero (closed) to one (fully open),

$$\dot{m}_v = OD\sqrt{P_c - P_e}\sqrt{\rho_l}C_v. \tag{5}$$

In steady state, the mass flow through the valve $\dot{m}_v$ must be equal to the mass flow through the compressor $\dot{m}_c$, which can be calculated as the product between the compressor frequency f_{cp}, the compressor inlet volume V_{cp} and the density of the gaseous refrigerant ρ_g,

$$\dot{m}_v = \dot{m}_c = f_{cp}V_{cp}\rho_g. \tag{6}$$

The mass flow $\dot{m}_c$ is essentially the product between a constant and the evaporator pressure P_e, when the system is in steady state (P_e is proportional to ρ_g). However, this is only true if the compressor speed is held constant. Equation (5) can also be simplified if assuming that the fluctuations in the square root of the pressure difference is negligible small and that the density of the refrigerant is constant. Combining (5) and (6) with simplifications, gives a steady state equation for the evaporator pressure P_e with variable input control signal OD,

$$P_e = cOD, \tag{7}$$

where c is a further collection of constants. A first order filter is now introduced, since the outer loop has to be slower than the inner loop for stability. This can be handled by choosing the time constant τ appropriately,

$$G(s) = \frac{P_e(s)}{OD(s)} = \frac{c}{\tau s + 1}. \tag{8}$$

The gain c in the simplified expression is very dependent on the operating point and on the characteristics of the given refrigeration system. Therefore, an adaptive update of the constant c is introduced, in order to better estimate the pressure. By continuously calculating the variance of the outlet temperature, while slowly increasing the OD signal, it is possible to detect the point when the evaporator is close to being fully flooded. This was also discussed in Section III and the point is used as a fix point to find a good estimate of the gain c^* in the fix point, by using (7), since OD is known along with the pressure at the fix point P_e^*. The pressure is not measured directly but can be calculated based on the measured evaporator outlet temperature T_o^* and a predetermined offset temperature T_{off} as

$$P_e^* = PDewT(T_o^* - T_{off}), \tag{9}$$

where the refrigeration equation software package RefEqns by Morten Juel Skovrup has been used, however, there are many other software packages that can do the conversion. Fig. 5 shows a plot of the evaporator pressure P_e, while OD is gradually increased from 0.28 to 0.80 in open loop on the air conditioning system shown in Fig. 1(a). The dot marks the

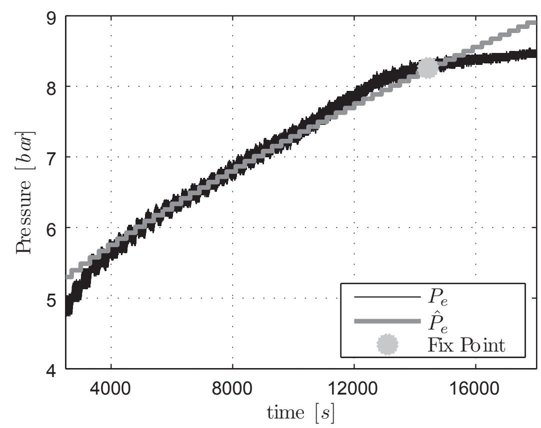

Fig. 5. Measured evaporator pressure during an OD sweep and resulting linear estimated pressure based on the gain c, found in the marked fix point.

identified fix point, where the evaporator is nearly flooded. The estimated linear pressure $\hat{P}_e$ based on the estimated gain c is also shown in the figure. Note that OD has been replaced by $OD^{0.5}$ on the air conditioning system, to better account for valve nonlinearities. However, this is not necessary when the gain c is continuously adapted.

It is undesirable to change the value of the gain c instantly in closed loop, since this could result in unstable behavior. The MIT rule is therefore used to adapt the gain c slowly and it is defined as (see e.g. [4]):

$$J = \frac{1}{2}e^2 \tag{10}$$

$$\frac{d\theta}{dt} = -\gamma\frac{\partial J}{\partial\theta} = -\gamma e\frac{\partial e}{\partial\theta}, \tag{11}$$

where J is an objective function to be minimized, e is the error, θ is the adjustable parameter to be adapted and γ is the adaption gain. The MIT rule can be interpreted as a gradient method for minimizing the error and in the case of adapting the gain c we have

$$\theta = c \tag{12}$$

$$e_c = c - c^* \tag{13}$$

$$\frac{dc}{dt} = -\gamma e_c, \tag{14}$$

since the partial derivative of e_c is equal to 1. The gain c^* is the gain obtained at the last fix point and the gain c is the current gain. Only the adaption gain γ has to be chosen. In general a small γ means slow convergence and a large γ means fast convergence and possibly instability. However, it is hard to say in general how γ influences time variant systems. In the tests on the refrigeration systems γ has been chosen small and thus conservatively. Another possibility would be to use the normalized MIT rule, which would lead to less sensitivity towards signal levels or one could use Lyapunov stability theory to adapt the gain c, and most likely obtain faster adaption and stability guarantees.

VI. STARTUP PROCEDURE

All parameters in the controller can be determined based on two open loop tests. The OD signal is increased slowly in the first test, while the temperature T_o is measured and its variance is calculated. The test is stopped when the variance plot shows a clear peak and has decreased to a low level again. The result on each of the systems is presented in Fig. 2 and 3. The variance levels σ^2_{low} and σ^2_{high} are set to

$$\sigma^2_{high} = \frac{1}{2} max(\sigma^2) \tag{15}$$

$$\sigma^2_{low} = \frac{3}{4}\sigma^2_{high}, \tag{16}$$

where $max(\sigma^2)$ is the highest variance during the test. These have shown to be reasonable values based on multiple tests on the two different systems introduced in Section II.

A temperature offset T_{off} is required in (9) to determine the gain c and thus the evaporator pressure. This temperature offset accounts for the temperature difference between the outlet temperature T_o^*, when the high variance threshold σ^2_{high} is reached and an estimate of the evaporation temperature. This estimate is set to be the lowest outlet temperature measured during the OD sweep test and gives $T_{off} = T_o^* - min(T_o)$. A start guess of the gain c is then obtainable from (7) and (9).

The second open loop test is a small upward step in OD at low superheat, while T_a or T_w is close to T_e, which is considered as a worst case operating point. This test is used to tune the PI controller based on Ziegler-Nichols tuning with quarter decay ratio, see e.g. [12]. The transfer function of the PI controller is defined as

$$D(s) = k_p\left(1 + \frac{1}{T_I s}\right) \tag{17}$$

$$k_p = \frac{0.9}{RL} \tag{18}$$

$$T_I = \frac{L}{0.3}, \tag{19}$$

where R is the slope of the reaction curve and L is the lag obtained from the step test. The PI controller is tuned at an operating point, where the temperature and refrigerant flow is low, which gives the highest system gain. This gives a conservative controller and ensures that the system is stable at all other operating points. The selected worst case operating point is supported by e.g. [13]. The slope R was measured to be -8.08 and -0.95, for the air conditioning system and refrigeration system, respectively, and the lag L was 23.6 and 27.6. These parameters can also be used to determine a suitable value for the reference decrease rate and the time constant τ, since these measures gives an indication of how fast/slow the system is. During the tests, the reference decrease rate and reference step size was set to 3/1000 and 3, respectively, and τ was set to 30 seconds.

VII. TEST RESULTS

Fig. 6 shows the result from a test of the controller on the air conditioning system. The estimated superheat $\hat{T}_{sh}$ follows the reference well and the reference is slowly decreased and then stepped back each time the variance gets too high, which indicates low superheat. The measured superheat T_{sh}, using a pressure sensor, is shown for comparison and the difference between the estimated and measured superheat gets smaller as the estimate of the gain c is adjusted (γ was set to 0.0005).

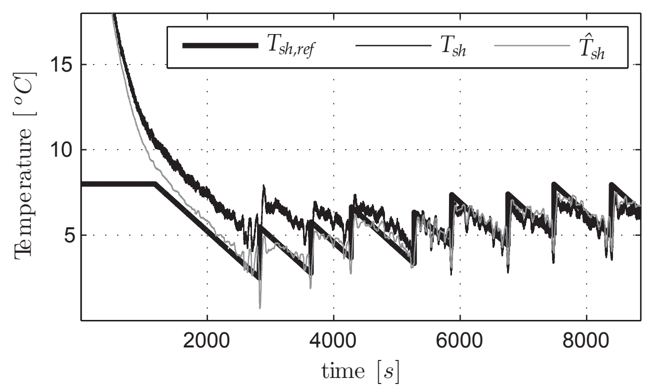

Fig. 6. Closed loop test results on the air conditioning system.

A similar test was conducted on the air conditioning system, where the load was changed by blowing air from the hot room to the cold room. This caused a sudden rise in ambient temperature and thus a change in the load. Fig. 7 shows that this disturbance is handled by the controller.

Fig. 8 finally shows the result from a test of the controller on the refrigeration system. A change in load was also made in this test, by changing the temperature set point in the water tank with the water heater shown in Fig. 1(b).

The estimated superheat follows the reference superheat and is stepped back each time the variance gets too high, as anticipated. However, there is approximately a 5 degree temperature offset between the estimated and measured superheat. This is because the variance starts to increase a little earlier in closed loop, and the temperature offset T_{off} was estimated in open loop. The T_{off} estimate could be improved by allowing a small overflow in closed loop. However, if

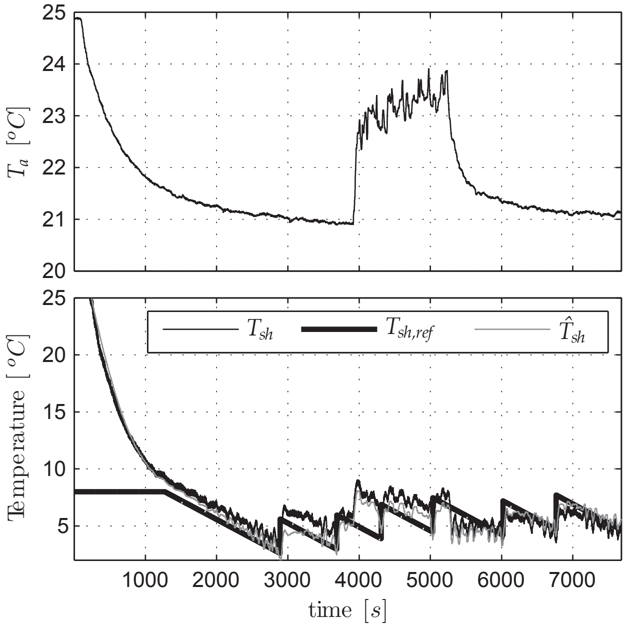

Fig. 7. Closed loop test results on the air conditioning system with a sudden change the in ambient air temperature T_a.

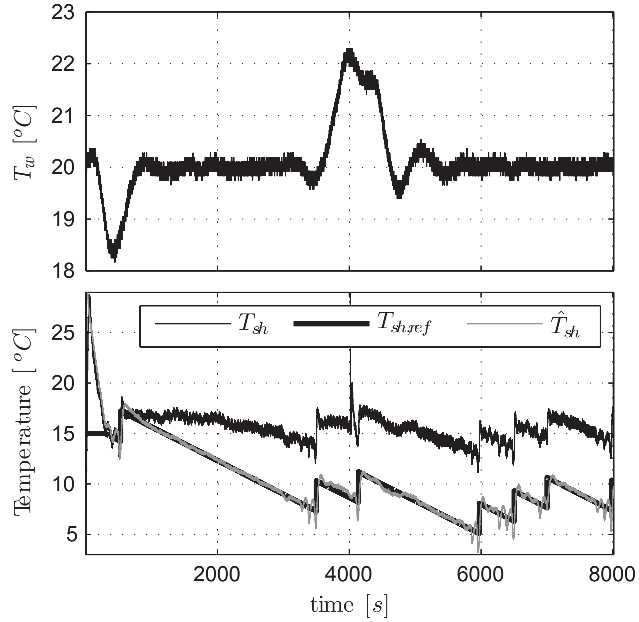

Fig. 8. Closed loop test results on the refrigeration system with a sudden change in the water temperature T_w.

comparing the actual superheat of about 15 degree with Fig. 3, then this superheat corresponds to a working point just before the steep slope, which happens over 2-3 quantizations in OD. Controlling the superheat to a point on the middle of the slope is quite difficult and the result is close to optimal.

The PI controller parameters are chosen conservatively in a situation with low flow and temperature. The controller response time could possibly be improved by limiting the operating range of the system or by adding some kind of gain scheduling. However, the gain scheduling should only be based on the information given by the evaporator outlet temperature measurement. Feed forward, when a step in the reference is made, could also improve the controller.

VIII. Conclusion

Evaporator superheat control is important in order to optimize the heat transfer in refrigeration systems and to prevent compressor wear. The superheat is conventionally obtained by subtracting the evaporation temperature, given by a pressure sensor, from the temperature at the evaporator outlet. In this paper we have shown that the pressure sensor can be saved by looking at the variance in the outlet temperature, which have shown to increase at low superheat. Results from tests on two different refrigeration systems indicate that the proposed controller, using qualitative system knowledge, can control the evaporator superheat to a low level giving close to optimal filling of the evaporator, with only one temperature sensor. No a priori model knowledge was used and it is anticipated that the method is applicable on a wide variety of refrigeration systems.

IX. ACKNOWLEDGMENTS

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