A Model-Based Feed-Forward Controller Scheme for Accurate Chilled Water Temperature Control of Inlet Guide Vane Centrifugal Chillers

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Capacity control in most commercial centrifugal chillers is achieved by inlet guide vanes which are activated by the leaving chilled water temperature sensor. Due to mechanical reasons, vane control is done discretely, and not continuously. The control module compares the value provided by the temperature sensor to pre-set control band values, and if those bands are exceeded, it sends a signal to the vane control motor to adjust the vane position by one step which could be upwards or downwards. The advantage of this type of discrete control method is its simplicity. Normally, the accuracy in the outlet chilled water temperature is of the order of 0.5 °C, which is acceptable for normal cooling plants such as used in office buildings. However, there are applications such as in pharmaceutical processes, mechanics labs, or instances in chemical processes where more accurate control is required (sometimes as low as 0.05 °C). This paper proposes a simple method to achieve such tight control without any hardware modifications. The basis of this method is a transient physical inverse model of the refrigerant boiling process in the evaporator, in conjunction with a feed-forward control scheme. The model parameters need to be identified from monitored data since they are chiller-specific. This paper describes the model, and applies it to one-minute monitored data from an actual chiller plant of 1580 kW (450 Tons). It is demonstrated that for this specific chiller such a control scheme has the potential to improve control accuracy by about 28% as compared to the traditional control method. [DOI: 10.1115/1.1775225]

Background

Large commercial chillers (say, over 500 kW) normally provide control accuracy of chilled water temperature of about ±0.5°C. Small commercial chillers with ON-OFF controls are even less accurate. In some applications, such as in the semiconductor industry, chemical engineering industry, food industry, and pharmaceutical industry, highly stable chilled water temperature is required, some of them requiring control accuracy as tight as ±0.05°C. How to achieve such high accuracy has been the objective of a certain number of prior studies.

Chua et al. [1] discussed reducing the fluctuation in the absorption chiller's chilled water outlet temperature by utilizing waste heat. Similar research has also been done by other researchers [2–5]. Due to the unique nature of an absorption chiller, the results from these studies cannot be applied to a centrifugal chiller. Research on optimal on-off switching control of small refrigeration units was conducted by some research groups; for example, Refs. [6–8]. These studies have contributed to improving the control accuracy of small refrigeration units with on-off switching control, but are not applicable to large centrifugal chillers with step capacity control. Other studies [9,10] proposed a continuous capacity control strategy to improve the accuracy of chilled water temperature, and successfully applied this technique to screw chillers. Instead of using traditional step control, they developed a new control module which has the capability of controlling chiller continuously. It is possible to adopt this technique to large centrifugal chillers, but cost is a major issue.

Efforts at reducing chilled water temperature's fluctuation were also made by installing additional equipment to a central chiller plant, such as using a cold storage [11–13]. The disadvantage of this method is its cost and extensive installation space. Arima and Hara [14] provided another kind of solution to improve central chiller plant's stability: fuzzy logic control. Their research was exploratory and conceptual, and much more development is needed in order to apply their result and methodology extensively.

Literature on accurate chilled water temperature control to specific applications can also be found, say, in food processing [15,16]. In these two studies, issues on how to achieve stable plant operation and how the surrounding temperature's fluctuation affects food were discussed.

Objectives

In centrifugal chillers, the chilled liquid temperature sensor is usually placed in thermal contact with the leaving chilled water. Capacity control in large chillers is usually achieved by inlet guide vane control. The vane position does not change continuously. Whenever the temperature sensor receives a signal, the temperature module will compare that value to the control band limits. If that value is greater than the upper control band limits, the control module will send a signal to the vane control motor to close the vane position to the next step. The advantage of this control method is its simplicity. Normally, the accuracy of this control method applied to centrifugal chiller is about 0.5°C, which is acceptable for applications such as office building, or shopping centers. However, there are applications where much

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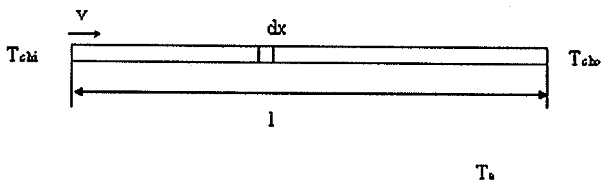

Fig. 1 Simplified sketch of a flooded evaporator with chilled water flowing inside the tube

more accurate control is needed (sometimes as low as 0.05°C). To achieve this objective, we shall propose developing a transient physical model to describe the refrigerant boiling process in the evaporator, and demonstrate how it can be used to provide finer control of the leaving chilled water temperature. Finally, this model will be evaluated based on 1-minute transient data from a 1580 kW chiller located at Drexel University Korman (DUK) center being monitored for this purpose.

The issue under investigation is how to achieve closer control of chilled water temperature using the existing inlet guide vane control module. Since the module receives the temperature signal from the leaving chilled water temperature, the control response always has a delay factor. When entering chilled water temperature changes slowly, this delay factor is not important. In this situation, the control accuracy is mainly determined by the step size of vane position adjustment. The finer the step, the more accurate the control. However, when entering chiller water temperature changes more abruptly, this delay factor becomes critical. Under this condition, the control accuracy is determined by the delay factor. No matter how fine the vane position step, the control accuracy cannot be improved as long as the delay factor exists. Under this limitation, the logical approach towards better control is to adopt a strategy whereby not only the temperature of the leaving chilled water is used but the temperature sensor at the entering chilled water as well.

The difficulty is that no direct physical model exists, which couples entering chilled water temperature to vane position to achieve better control. Developing such a physical model coupled with mathematical models for optimum temperature control is necessary for such a control strategy. The transient chiller data from the DUK chiller will be used for developing and evaluating the physical model and the control strategy.

Transient Model for the Evaporator

To perform feed forward control, a dynamic relationship between chilled water inlet temperature and chilled water outlet temperature has to be determined. In this section, we shall develop the governing equations, along with their initial condition and boundary conditions, which are then solved analytically.

The evaporator can be simplified as a tube immersed in a pool of refrigerant as shown in Fig. 1. The evaporator temperature of the bulk refrigerant far away from the tube is T_e. To balance the heat flow of the small element dx, the following expression has to be satisfied:

$$C_p \rho \pi r^2 dx dT + C_p \rho \pi r^2 v\, dt dT = -h(T - T_e) 2\pi r dx dt \quad (1)$$

where C_p is the heat capacity of chilled water, ρ is the density of chilled water, T is the temperature of the element of chilled water, t is the time, h is the overall heat transfer coefficent between the chilled water and the refrigerant, r is the radius of the tube, and v is the velocity of chilled water.

The first term is the total energy change of the small chilled water element dx; the second term is the net heat flow in and out of the small element; the third term is the overall heat transfer rate between the small element and the refrigerant.

Equation (1) can be rewritten as,

$$\frac{\partial T}{\partial t} + v\frac{\partial T}{\partial x} = -\frac{2h}{C_p \rho r}(T - T_e) \quad (2)$$

This is a first order partial differential equation with two independent variables. To solve this equation, one initial condition and one boundary condition are needed. Please note that T_e is assumed to be constant, which implies that the above equations are only valid when there is no control. For the initial condition, we assume,

$$T|_{t=0} = g^*(x) \quad (3)$$

We use the chilled water inlet temperature, assumed to be any arbitrary periodic function, as our boundary condition,

$$T|_{x=0} = f^*(t) \quad (4)$$

The analytical solution of this equation is [17]

$$T(x,t) - T_e = \sum_{k=0}^{\infty} \left\{ a_k \cos\left[\frac{k\pi}{l}\left(t - \frac{x}{v}\right)\right] + b_k \sin\left[\frac{k\pi}{l}\left(t - \frac{x}{v}\right)\right]\right\} \cdot e^{-(2h/C_p r \rho v\, x)} \quad (5)$$

Thus, the chilled water outlet temperature is given by:

$$T_{cho} - T_e = \sum_{k=0}^{\infty} \left\{ a_k \cos\left[\frac{k\pi}{l}\left(t - \frac{l}{v}\right)\right] + b_k \sin\left[\frac{k\pi}{l}\left(t - \frac{l}{v}\right)\right]\right\} \cdot e^{-2hl/C_p r \rho v} \quad (6)$$

and the expression for the chilled water inlet temperature is,

$$T_{chi} - T_e = \sum_{k=0}^{\infty} \left[a_k \cos\left(\frac{k\pi}{l} t\right) + b_k \sin\left(\frac{k\pi}{l} t\right)\right] \quad (7)$$

It is noted that the net temperature drop of chilled water is only determined by the amount of heat transferred to the refrigerant. In the next section, we shall verify whether the experimental results are consistent with this approximation.

Transient Physical Model Calibration

According to Eqs. (6) and (7), the relations for chilled water inlet temperature and outlet temperature are given by,

$$T_{chi} - T_e = f(t)$$

$$T_{cho} - T_e = f\left(t - \frac{l}{v}\right) \cdot e^{-2hl/C_p r \rho v} \quad (8)$$

where $f(t)$ is any arbitrary periodic function. Let us consider a period when there is no control operation, i.e. when T_e is constant. To validate the above equations, we should carefully choose appropriate monitored data. Let

$$\Delta t = \frac{l}{v} \quad (9)$$

Equation (8) is written as,

$$T_{chi} - T_e = f(t)$$

$$T_{cho} - T_e = f(t - \Delta t) \cdot e^{-2hl/C_p r \rho v} \quad (10)$$

A field operated centrifugal chiller (DUK chiller) has been extensively instrumented as part of this research and is fully described by Jia [17]. To validate Eq. (10), we first calibrate the constants, i.e., determine the values of (Δt) and $(2hl/C_p r \rho v)$. Three sets of data collected at different times, under different load conditions and with constant T_e are selected. After calibration, we shall validate the calibrated parameters by using an additional set of monitored data collected under operating conditions different from those of the initial data sets.

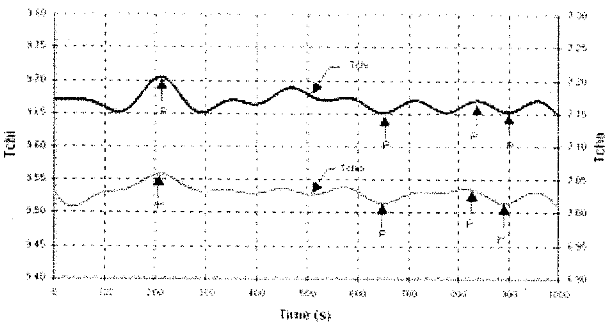

Fig. 2 Interpolated values of T_{chi} and T_{cho} in (°C) over time (interpolated to 1 s time intervals)

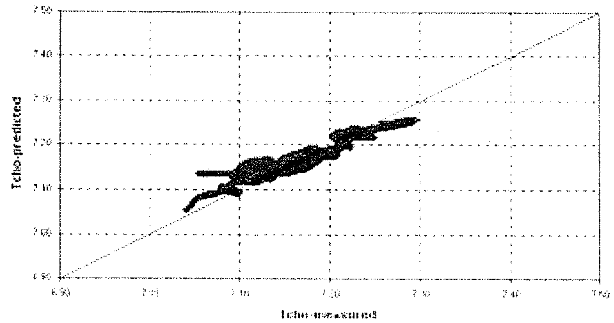

Fig. 3 Comparison of measured and model predicted values of T_{cho} (°C) based on 12,000 sets of data

Since $f(t)$ is any aribitrary periodic function, it must have a peak value at some time. Similarly, $f(t-\Delta t)$ is also a periodic function. By determining the time interval between the peaks, we can calculate

$$\Delta t = t_{peak(chi-e)} - t_{peak(cho-e)} \tag{11}$$

By comparing the magnitude of the peak values, we can determine

$$\frac{(T_{chi}-T_e)_{peak}}{(T_{cho}-T_e)_{peak}} = e^{2hl/C_p r\rho v} \tag{12}$$

i.e.

$$\frac{2hl}{C_p r\rho v} = \ln\left(\frac{(T_{chi}-T_e)_{peak}}{(T_{cho}-T_e)_{peak}}\right) \tag{13}$$

For the DUK chiller, our initial estimate of Δt was around 15–20 seconds. However, our data sets were collected at one-minute intervals, and so, it is impossible to use the monitored data directly to find the time difference between peaks. To determine this, we have to interpolate the monitored data and obtain a finer time resolution data set, say every second. Since T_{chi} is a periodical function, it is possible to achieve this objective by Fourier based interpolation. Figure 2 shows the interpolated data at one second time intervals.

In Fig. 2, however, it is straightforward to determine the time difference where two temperatures reach the peak, as indicated by the arrows. However, peaks picked in Fig. 2 are not good examples since the wave crest and trough are too close to the sensor accuracy. To obtain more accurate results, the wave crest and trough should be far enough so as to minimize the effect of measurement uncertainty.

Table 1 summarizes our calibration results. The calibrated results for Δt by three groups of data were 14, 18 and 15 seconds, while the results for $(2hl/C_p r\rho v)$ were 0.54, 0.55, and 0.53. We use the average value of 16 seconds for Δt and 0.54 for $(2hl/C_p r\rho v)$ as the final values obtained from calibration. From the above analysis

$$\frac{f_{(T_{chi}-T_e)}(t)}{f_{(T_{cho}-T_e)}(t-16)} = e^{0.54} \tag{14}$$

To validate our conclusion, we need to verify this against the interpolated chiller data. This is easily done by shifting the $(T_{chi} - T_e)$ series data by 16 seconds, then multiplying by $e^{-0.54}$. The new data series should equal to $(T_{cho}-T_e)$. Figure 3 shows the comparison of predicted and measured series data of $(T_{cho}-T_e)$ based on 12000 sets of data. The standard deviation of the error is 0.09°C which is low enough for us to have confidence in our calibrated model.

Control Strategy Proposed

The block diagram, depicted in Fig. 4, shows that the traditional control system is a typical single-input and single-output system with a feedback loop. It is well known that high-accuracy motion can be achieved with closed-loop control using position sensors. Such a feedback configuration, however, is inappropriate under fast action mode, since the speed of the feedback action is bounded by the delays in the feedback loop [18]. For the guide vanes to change the value of T_e so as to maintain reasonable accuracy, the guide vane must be directly commanded by a function associated with the input signal. This will be demonstrated below.

Experimental data indicate that every vane position corresponds to about a change of 0.33°C in T_e. In other words, whenever vane position is opened or closed by one step, T_e will increase or decrease by 0.33°C. Figure 5 illustrates how T_{cho} changes with T_{chi} under traditional feedback control.

The standard deviation of T_{cho} under the present feed-back control operation is 0.18°C, which is lower than the value of 0.25°C (design control accuracy) often quoted by chiller manufacturers. This is due to the fact that T_{chi} changes are relatively small in the data. The control band of ±0.3°C means that if the variation in T_{cho} is greater or lower than control target temperature by 0.3°C, signals will be sent to guide vane motor to adjust guide position accordingly. As we discussed, the advantage of this control strat-

Table 1 Results of Calibrating the Transient Physical Chiller Model

	1		2		3		
	Peak Time (s)	Peak Value (°C)	Peak Time (s)	Peak Value (°C)	Peak Time (s)	Peak Value (°C)	Average
T_{chi}	8781	9.7744	5932	9.4931	8943	9.8783	
T_{cho}	8767	7.2236	5914	7.1671	8928	7.474	
T_e		3.6652		4.0200		4.0222	
Δt (s)	14		18		15		16
$2hL/C_p\rho vr$		0.5404		0.5533		0.5285	0.54

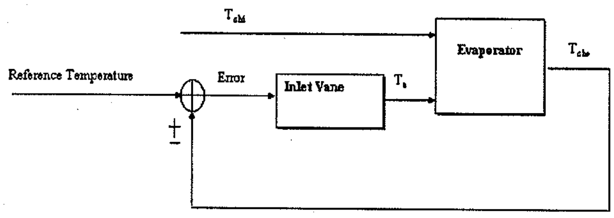

Fig. 4 Control diagram of the traditional chiller feedback control strategy

egy is its simplicity and ease, and hence was adopted by the chiller industry. However, when input values of T_{chi} changed quickly, such as at $t = 1100$ (s), 2600 (s) and 4000 (s) in Fig. 5, the accuracy of this control configuration becomes unacceptable, because the speed of the feedback action is bounded by the delays in the feedback loop. One may think that narrowing the control band may improve control accuracy. If this is true, we may achieve highly accurate control without changing control configuration. By varying control band limits while keeping control step and control set up unchanged, different control accuracies were generated with the same data input sequence of T_{chi} (Fig. 6). It reveals that the best accuracy is achieved when the control band approaches the control step. Beyond that, chiller outlet temperature is not as well controlled. It is easy to understand why control accuracy is getting worse as the control band becomes wider. Here, it is interesting to observe that the control accuracy becomes worse when control band is too narrow (well below the one control step). This can be explained by the fact that the control module always over compensates because of the feedback delay and discrete change of T_e. This effect is demonstrated in Fig. 7.

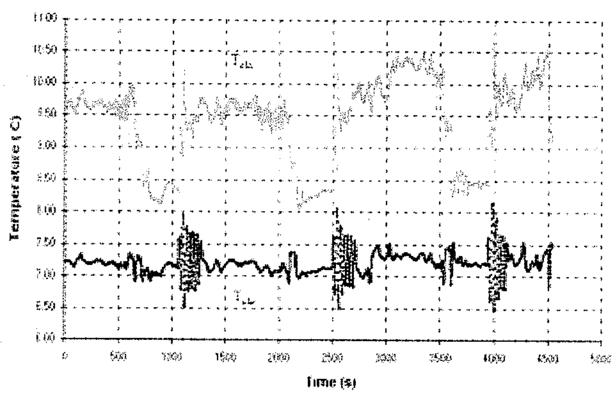

Fig. 5 Variation of T_{chi} (upper plot) and T_{cho} (lower plot) with time under the traditional control strategy with control bands set at ±0.3°C and a control step of 0.33°C

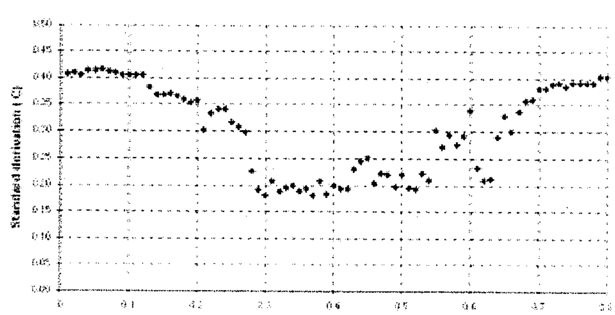

Fig. 6 Variation of control accuracy with control band setting under the traditional control strategy

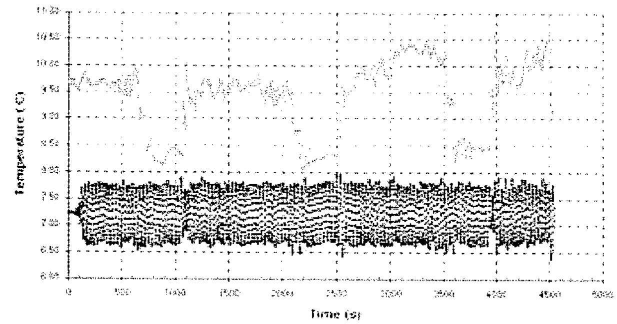

Fig. 7 Variation of T_{chi} (upper plot) and T_{cho} (lower plot) with time under the traditional control strategy with control bands set at ±0.01°C and a control step 0.33°C

We conclude that the accuracy of the traditional chiller control configuration is bounded by its characteristics. To further improve control accuracy, we need to adopt a different control strategy. One such strategy is called command feed-forward. The command feed forward strategy can substantially improve the accuracy of the output response to the command. However, it comes with its own limitation. In order to apply this technique, one needs to know (i.e., model) the system properly, which in our case is the evaporator. Otherwise, the advantage of feed forward is decreased drastically [18].

Based on the above discussions, it is clear that for a given chiller we can calculate T_{cho} by knowing T_{chi}. Thus, it is possible to adopt the feed forward instead of the feed back control scheme to control the guide vanes. In the feed forward control configuration (see Fig. 8), the chilled water temperature sensor is placed in thermal contact with the incoming chilled water. The vane position does not change continuously. Whenever the temperature sensor receives a signal, the temperature module will calculate the corresponding value of T_{cho} and compare that value to the control band. If that value is greater than the upper control band value, the control module will calculate the value of T_e and then send a signal to the vane control motor to adjust the vane position directly. The advantages of this control strategy over the traditional one are: (i) the control has practically no time delay since T_{cho} is pre-controlled; and (ii) since the vane position is calculated, there is no problem of over compensation.

Using the same input data set of T_{chi}, and the same values for control step and control band as those used to generate Fig. 5, new control results were calculated by physical transient model based on the proposed feed-forward control scheme. These are plotted in Fig. 9. The standard deviation of T_{cho} is 0.13°C, which is less that the 0.18°C found by the traditional feed-back control mechanism. There is thus, a decrease of 28% in the standard deviation of T_{cho}

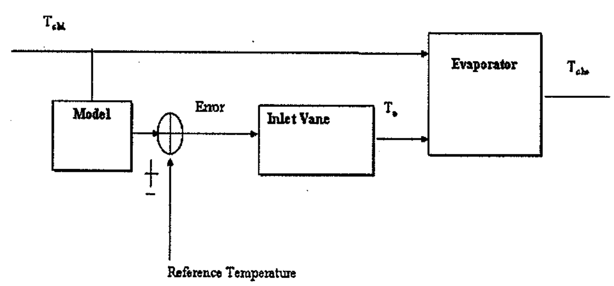

Fig. 8 The control diagram of the proposed chiller feed-forward control strategy

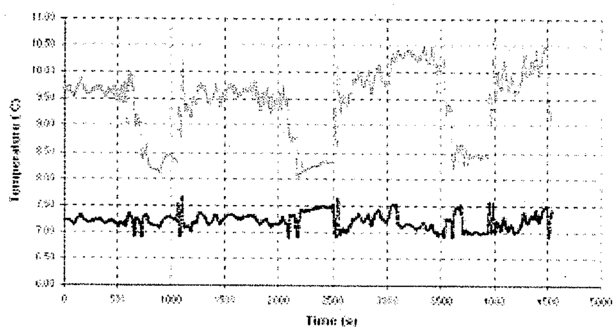

Fig. 9 Variation of T_{chi} (upper plot) and T_{cho} (lower plot) with time under the proposed feed-forward control strategy for the control band set at ±0.3°C and a control step 0.33°C

for this particular data sequence. Since there is no over compensation in this control method, it is predictable that control accuracy should increase as control band decreases. Figure 10 shows the variation of control accuracy with control band setting. In an extreme case, this control strategy permits a control band which equals to zero. In such a case, the standard deviation is 0.03°C, as shown in Fig. 11. It is pointed out that a control band of zero is an extreme and may not be achievable in practice since this would require very fine increments of the inlet guide vanes.

Conclusion

In order to achieve highly accurate chilled water temperature control, a new control strategy, called the command feed-forward control, has been proposed based on a physical transient model for the evaporator. This physical transient model is a partial differential equation which can be solved analytically with specified boundary and initial conditions. Good agreement between model predicted data and measured data were noted using a data set of 12,000 values of one second data generated from our monitored DUK chiller. Based on the proposed physical transient model, computer simulations of this new control strategy versus traditional control method were performed. It was found that the proposed feed-forward control strategy was superior to the traditional feed-back control mechanism currently used in chillers in two respects. It can, theoretically, permit a control band of zero to be attained (provided the inlet guide vanes movement can be made continuous). Of more practical relevance is the fact that the proposed scheme can control the chiller outlet water temperature much more accurately even with the existing inlet guide vane control mechanism. For the particular DUK data set analyzed, our proposed feed-forward control scheme improved control accuracy by 28% as compared to the traditional feedback arrangement. Future studies should include the case of variable control, i.e., under conditions of varying T_e. Finally, the usefulness of a combination of feedforward and feedback control stretegy may also be worth exploring in the future.

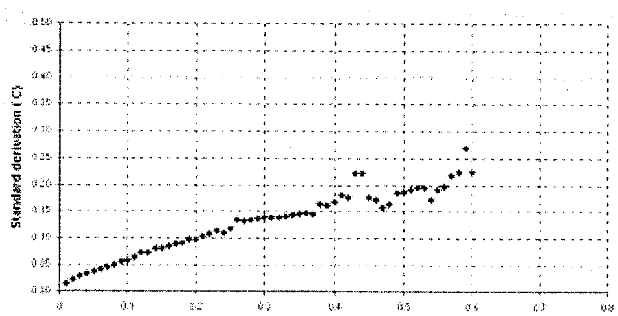

Fig. 10 Variation of control accuracy with control band under proposed feed-forward control strategy

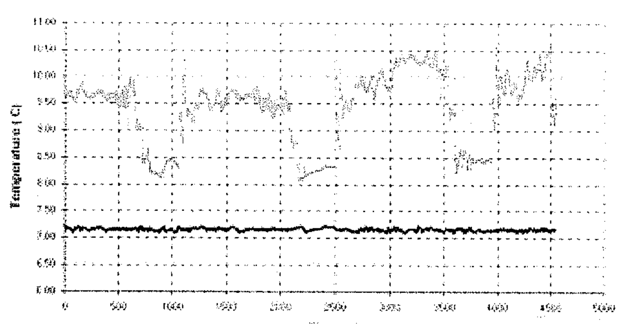

Fig. 11 Variation of T_{chi} (upper plot) and T_{cho} (lower plot) with time under the proposed feed-forward control strategy for the control band set at 0°C and a control step 0.33°C

Nomenclature

a = Constant
b = Constant
C_p = Specific heat at constant pressure, kJ/kg.K
d = Tube diameter of heat exchanger, m
h = Overall heat transfer coefficient, kw/m^2.K
l = Length of tube, m
m = Mass flow-rate, kg/s
r = Radius of tube, m
T = Temperature, °C
$T_{e,s}$ = Average temperature of tube surface, °C
η = Efficiency
ρ = Density, kg/m^3
μ = Viscosity, kg/s.m

Subscripts

chi = Chilled water inlet
cho = Chilled water outlet
e = Evaporator side refrigerant

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