Empirical modeling of the impacts of faults on water-cooled chiller power consumption for use in building simulation programs

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Abstract

Empirical models of four chiller faults that can be applied within existing building models to study overall impacts are developed in this paper. The faults include overcharging, excess oil, non-condensables in refrigerant and water-side condenser fouling. A single generalized model structured was developed for these faults that forces predicted fault impacts to be zero with no fault and increase with increasing fault level. The models were trained and tested using available laboratory data for a water-cooled centrifugal chiller where all four faults were artificially introduced. The fault model behavior was studied and then they were integrated in hospital models from DOE commercial reference building models (Deru et al. 2011) and simulations were performed in different climates. The simulation results showed maximum increases of building electricity consumption, electricity peak demand and water consumption of the hospitals due to faults of 4.7%, 7.8% and 1.8% respectively. The fault impacts were found to be more severe in hotter and more humid climates.

1. Introduction

The evolution of low-cost sensors and control systems for buildings has led to the recent development and evaluation of methods for automated fault detection and diagnostics (FDD) applied to chillers. The FDD algorithms are designed to automatically identify faults that deteriorate equipment performance. For instance, Castro (2000) demonstrated the use of clustering techniques to implement FDD algorithms for chiller faults on measurement data. Reddy (2007) compared the performance of four different FDD algorithms for chillers using data from a water-cooled centrifugal chiller that was tested under various faulty conditions (Comstock and Braun 1999a, b). Zhou et al. (2009) developed a model-based FDD algorithm for chillers based on a semi-empirical chiller model from Ma et al. (2008). Han et al. (2011) developed a machine-learning-based FDD algorithm using data from Comstock and Braun (1999b). Zhao et al. (2012) developed a virtual condenser fouling sensor for chillers.

Chiller models are very useful in the development and evaluation of FDD algorithms. Reddy et al. (2003) summarized four different methods to model chillers: thermodynamic and heat transfer based models, linear empirical models such as the chiller model in the DOE-2.1 building model (DOE 1981), physical models and artificial neural network models. As an example, a thermodynamic and heat transfer based model was developed by Ma et al. (2008) to be used in a building simulation program for the development of FDD algorithms for chillers. Another example is a physical model of a chiller and chiller faults developed by McIntosh et al. (2000) to study chiller performance degradation due to faults. Although not included in the four methods described by Reddy et al. (2003), a Kalman-filter based chiller model was used in Navarro-Esbrí et al. (2006) to create a chiller fault detection algorithm. Furthermore, machine-learning based FDD algorithms such as Han et al. (2011) typically use machine-learning algorithms to create a model of faulted chillers to be used in the chiller FDD.

DOE-2 (DOE-2 2014) and EnergyPlus (EnergyPlus 2015) are commonly used building simulation programs that employ the linear empirical chiller model originally developed for DOE-2.1 (DOE 1981). This model uses a user-defined evaporator water outlet set point temperature, chiller cooling load requirement and condenser water inlet temperature to calculate chiller part-load ratio and power consumption using multiple empirical equations. TRNSYS (TRNSYS 2012) is another building simulation program that employs a water-cooled chiller model that relates the same inputs and outputs with an alternative approach based on a thermodynamic model. For any of these simulation programs, it is relatively easy to implement fault models that act on the outputs of the chiller models. For example, EnergyPlus allows the use of Scripts programmed using a language called Erl to modify chiller model outputs.

Although Zhou et al. (2009) demonstrated that a thermodynamic and heat transfer based model could be modified to simulate fault impacts in building models, it is much more convenient to modify

the inputs and outputs from existing chiller models within a simulation program than to replace the underlying model. One simple example of this approach is Magoulès et al. (2013) who reduced the reference COP of an empirical chiller model by a constant value before running the building simulation. Although this is meant to model fault impacts, it is unrealistic because the impacts of faults on chiller COP change with different environmental conditions as shown in Comstock (1999a). Another example is Cho et al. (2014) where simulated fault impacts on cooling capacity and power consumption for split air conditioners were treated with multipliers applied to the non-faulted cooling capacity and power consumption. These multipliers could be applied to the outputs of a standard model within the building simulation library and thus do not require recoding or replacement of the air conditioner model. The use of this fault impact modeling approach in building simulation was demonstrated by Domanski et al. (2015) for a residential building.

In the current paper, a fault modeling approach similar to work of Cho et al. (2014) for split air conditions is developed for chillers in order to enable fault impact studies for medium to large commercial buildings using an existing widely used building simulation package. The fault models were developed based on the chiller fault data collected by Comstock and Braun (1999b) and predict impacts on chiller power consumption of four different types of faults: overcharging, excessive oil, non-condensable in refrigerant flow and water-side condenser fouling. The models are based on a similar mathematical structure as Cho et al. (2014) and can be applied directly to the outputs from existing chiller models. The training and validation procedures along with the application of the models using an existing building simulation tool are also discussed in this paper.

2. Review of faults

Overcharging of refrigerant beyond that specified by the manufacturer can occur during service and leads to increased compressor discharge pressure of the chiller compared to normal charge. This leads to an increase in the compressor power consumption.

An excess oil condition can also occur as a result of improper service and can cause an increase in viscous losses within the compressor. More refrigerant may also dissolve in the oil, and the effective mass of refrigerant circulating in the chiller may be reduced leading a loss in efficiency.

The non-condensable in refrigerant fault occurs as a result of service when the chiller was not well evacuated prior to charging of refrigerant. As a result, the chiller operates with an air-refrigerant mixture instead of a pure refrigerant. Since air cannot be condensed in the refrigerant liquid during chiller operation, the air is typically trapped in the superheated vapor region downstream of the compressor. The accumulation of air increases the compressor discharge pressure and hence the compressor power consumption.

Water-side condenser fouling occurs due to build up of scale on tube services but was simulated by Comstock and Braun (1999a, b) through blocking of individual heat exchanger tubes. Water-side condenser fouling increases the water flow resistance and reduces the water flow rate. The fouling may also reduce the heat transfer area of the condenser. The reduction of water flow and heat transfer area reduce the heat transfer rate of the condenser and hence the efficiency of the chiller.

Comstock and Braun (1999a, b) also recorded data for a water flow reduction fault, refrigerant leakage and defective expansion valve. Water flow reduction faults can be modeled using existing chiller models by simply reducing water flow rate and therefore was not considered in this paper. The defective expansion valve data do not contain fault levels and cannot be modeled in the same way as other faults. Its low refrigerant data did not show a definite increase of chiller power consumption with

the charge reduction as shown in Figure 1 for a range of other conditions, and therefore was not modeled in this paper. Although performance was not sensitive to charge over the range considered for this water-cooled chiller, this does not suggest that charge effects are not important at lower charge levels for this chiller type or for other chillers within the charge range considered.

Figure 1 Change of chiller power consumption with decreasing amount of refrigerant under a wide range of conditions (Evaporator water outlet temperature from 277.35K to 283.9K, condenser water inlet temperature from 289.25K to 303.36K, evaporator water inlet temperature from 279.4K to 289.5K, evaporator water flow rate at $49\text{m}^3\text{h}^{-1}$ and condenser water flow rate at $61\text{m}^3\text{h}^{-1}$)

3. Experimental setup

Since the details of the experiments were given in Comstock and Braun (1999a), only a review of the experimental procedure is provided in this paper. According to Comstock and Braun (1999a), the experiments were conducted with a 316.5-kW water-cooled centrifugal chiller. The chiller compressor

capacity was controlled to maintain a specified evaporator water outlet temperature, but a range of different set points were considered in the experiments. The chiller was tested under a variety of faults at different environmental conditions and fault levels as shown in Table 1 and Table 2 with a data acquisition frequency at 0.1Hz.

Table 1 Environmental conditions in the chiller tests

Evaporator water inlet temperature [°C]	6.3 – 16.2
Evaporator water outlet temperature [°C]	4.0 – 11.6
Condenser water inlet temperature [°C]	17.8 – 30.0
Evaporator water flow rate [m ³ h ⁻¹]	48.6 – 50.0
Condenser water flow rate [m ³ h ⁻¹]	60.1 – 61.9

Table 2 Fault levels being tested

Fault	Fault levels	Definition of fault level in Comstock and Braun (1999a)	Testing method
Refrigerant overcharge	0% to 40%	Percentage of mass of refrigerant above the manufacturer recommendation	Adding more refrigerant to the refrigerant circuit than manufacturer recommendation
Excess oil	0% to 68%	Percentage of the mass of oil in the chiller above the manufacturer recommendation	Filling the refrigeration system with more oil than manufacturer recommended oil level
Non-condensable in refrigerant	0% to 5%	Ratio of the volume of non-condensable in the chiller at atmospheric pressure and temperature to the volume of the refrigerant circuit	Adding laboratory-grade nitrogen into the refrigerant circuit before charging refrigerant into the system
Condenser fouling	0% to 45%	Proportion of water tubes blocked in the condenser	Blocking water flow paths in the condenser

Volumetric water flow rates were measured using vortex flow meters with accuracies of $\pm 1\%$ full scale for the water flows with the full scale measurement of evaporator water flow sensor being $50.2 \text{ m}^3 \text{ h}^{-1}$ and the full scale measurement of condenser water flow sensor being $62.7 \text{ m}^3 \text{ h}^{-1}$. At the chiller heat exchanger inlets and outlets and the evaporator water volumetric flow meter, temperatures were

measured using resistance temperature detectors (RTD) with an accuracy of $\pm 0.03\text{K}$. Chiller power consumption was measured using a watt transducer with an accuracy of $\pm 1.5\%$ full scale (120kW). The inlet and outlet water temperatures for the evaporator and the evaporator water flow rates were used to calculate the steady-state evaporator heat transfer rates as described in Appendix A. A simplified schematic of the experimental setup is shown in Figure 2.

Figure 2 Simplified schematic of the experimental setup in Comstock and Braun (1999a)

In Comstock and Braun (1999a), the environmental conditions of the fault tests were different from the non-faulted conditions, and the corresponding non-faulted chiller performance for the conditions of the fault cases could not be obtained directly from the experiments. To estimate the corresponding non-faulted chiller power consumption for various fault test conditions, Comstock and Braun (1999a) developed an empirical model as shown in Eqn. (1).

$$\hat{W}_{NF} = a_1 + a_2 T_{evap,out} + a_3 T_{cond,in} + a_4 \dot{Q}_{evap} + a_5 T_{evap,out} \dot{Q}_{evap} + a_6 T_{cond,in} \dot{Q}_{evap} + a_7 \dot{Q}_{evap}^2 \quad (1)$$

Eqn. (1) allows the calculation of fault impact ratios of chiller power consumption for each faulty case by Eqn. (2).

$$FIR_{\dot{W}} = \frac{\dot{W}_F}{\hat{\dot{W}}_{NF}} \quad (2)$$

The fault impact ratio for evaporator heat transfer rate is not needed because the chiller cooling capacity is controlled. Therefore, the evaporator heat transfer rate is not affected by faults unless the maximum chiller capacity is not sufficient to meet the load requirement, which is not considered in this paper.

4. Modeling methodology

The development of the fault modeling approach was guided by the following criteria:

- The model should predict no degradation in chiller performance when the fault level is zero.
- The model inputs and outputs should be independent of the size of the equipment.
- The model should always predict increasing chiller power consumption with increasing fault level.
- The fault model inputs should include the typical chiller model inputs used by common building simulation programs.

The first criterion was satisfied by employing the modeling approach of Cho et al. (2014), where a mathematical expression was chosen that outputs a fault impact ratio of unity when the fault level is zero. The second criterion was satisfied by using a dimensionless input for cooling load to the model and facilitates application of the model to different size chillers than the one tested by Comstock and Braun (1999a). The third criterion forces the model to predict the correct trend of decreasing performance

with increasing fault level. The fourth criterion can be met by creating a fault model with an input-output diagram as shown in Figure 3, where the inputs are evaporator water outlet temperature, condenser water inlet temperature and chiller cooling load.

Figure 3 Input-output diagram of the fault model at design

The resultant mathematical expression to estimate the fault impact ratio for chiller power consumption is shown in Eqn. (3).

$$\widehat{FIR}_W = 1 + F(c_1 + c_2 T_{evap,out} + c_3 T_{cond,in} + c_4 \frac{\dot{Q}_{evap}}{\dot{Q}_{evap,rat}})^2 + c_5 (\frac{\dot{Q}_{evap}}{\dot{Q}_{evap,rat}})F + c_6 F \quad (3)$$

Eqn. (3) is different from the empirical model of Cho et al. (2014) in two ways. The form of Eqn. (3) is such that it always predicts increasing fault impact ratio with increasing fault level while the model of Cho et al. (2014) can theoretically predict a fault impact ratio that decreases with fault level. Also, Eqn. (3) includes the effects of cooling load on fault impact ratio, whereas the Cho et al. (2014) model was developed for systems that don't have continuous cooling capacity and so it doesn't include load effects. Therefore Eqn. (3) is more suitable for modeling chiller faults than the model of Cho et al. (2014).

Eqn. (3) predicts the fault impact ratio for chiller power consumption only in terms of evaporator water outlet temperature, condenser water inlet temperature, cooling rate normalized by the chiller

rated capacity, and fault level. The fault level F in Eqn. (3) follows the definitions of fault levels in Comstock and Braun (1999a) as described in Table 2. The model outputs a value of one when the fault level is zero and its partial derivative with fault level is also always positive with positive values of c_5 and c_6 in Eqn. (3).

Eqn. (3) can only be used to consider one fault at a time. To model the impact of four different faults (overcharging, excess oil, non-condensable in refrigerant and condenser fouling), four different sets of coefficients of Eqn. (3) are needed. The calculation of the coefficients for each fault starts by calculating the fault impact ratios of chiller power consumption in each steady-state fault test by Eqns. (1) and (2). The chiller loads (evaporator heat transfer rates) from the data are normalized by the rated chiller capacity (316.5kW). The ranges of normalized chiller loads associated with the different fault models are listed in Table 3.

Table 3 Range of normalized chiller loads for different fault models

Normalized chiller loads	Overcharging	Excess oil	Non-condensable in refrigerant	Condenser fouling
Minimum	0.277	0.269	0.274	0.268
Maximum	0.932	0.968	0.849	0.987

The coefficients of Eqn. (3) are calculated by minimizing the sum of squared errors of fault impact ratios Eqn. (4) by the Generalized Reduced Gradient (GRG) method (Ladson et al. 1978) to limit c_5 and c_6 in Eqn. (3) to be a positive number.

$$\sum_{i=1}^{N_{\text{Fault}}} (\widehat{FIR}_{\dot{W},i} - FIR_{\dot{W},i})^2 \quad (4)$$

During each minimization run, only data of one type of fault are used, and four minimization runs were needed to calculate the coefficients of equations for all faults. The coefficients estimated for each fault are tabulated in Appendix B.

5. Model validation

In order to directly compare accuracies of the chiller fault models with a similar empirical model for a split air conditioner from Cho et al. (2014), the fit standard errors (FSE) for each fault are calculated by Eqn. (5), following the calculation method in Cho et al. (2014), and are tabulated in Table 4.

$$FSE = \sqrt{\frac{\sum_{i=1}^{N_{Fault}} (\widehat{FIR}_{W,i} - FIR_{W,i})^2}{N_{Fault} - P - 1}} \quad (5)$$

Table 4 Fit standard error of accuracy of Eqn. (3) to estimate fault impact ratio for chiller power consumption

Fault	Overcharging	Excess oil	Non-condensable in refrigerant	Condenser fouling
FSE	0.0154	0.0107	0.0197	0.0119

The FSEs of the models in Table 4 are higher than the average FSE of the fault impact ratio model of outdoor unit power consumption in Cho et al. (2014), which is 0.0067. This is probably because there are a larger number of inputs to the chiller model than for the single-speed split air conditioner in Cho et al. (2014). The split air conditioner in Cho et al. (2014) uses a fixed compressor capacity while the chiller has variable compressor capacity to maintain a water temperature set point. The changing compressor capacity results in a more nonlinear relationship between system performance and fault level than that of a single-speed split air conditioner.

To examine if the accuracy of the fault models is sufficient to be used with building simulation, a parity plot of fault impact ratios of chiller power consumption between the estimation and the training data is presented in Figure 4.

Figure 4 Comparison of estimated fault impact ratios of different faults with the ratios in the training data

Figure 4 shows that all data points lie within ± 0.04 of the fault impact ratios for the training data. Although the estimation of the models are unbiased, the scattering of data points of overcharging and excess oil fault hints at potential overfitting issues and hence erratic changes of predicted fault impact ratios with the model inputs. To examine if the fault impact ratios change appropriately with the inputs, the changes of fault impact ratios with all model inputs are plotted. Figure 5 shows the changes of the predicted fault impact ratios with condenser water inlet temperature when the normalized chiller load is fixed at 0.5 and the evaporator water inlet temperature is fixed at 6.67°C.

Figure 5 Changes of predicted fault impact ratio with fault levels at different condenser water inlet temperature for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling

Figure 5 shows that the fault impact ratios increase with the fault levels, and there is a local minimum of fault impact ratios as condenser water inlet temperature increases from 18°C to 30°C. When the condenser water inlet temperature increases from 18°C to 21°C, the fault impact ratios decrease, while the fault impact ratios increase as the condenser water inlet temperature increases from 24°C to 30°C.

To examine the effect of fault impact ratios with evaporator water outlet temperature, Figure 6 and Figure 7 are plotted to show the changes of fault impact ratios at constant normalized chiller load of 0.5 with fault levels when the condenser water outlet temperature is 18°C and 29.4 °C.

Figure 6 Changes of predicted fault impact ratio with fault levels at different evaporator water outlet temperature for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 18 °C

Figure 7 Changes of predicted fault impact ratio with fault levels at different evaporator water outlet temperature for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 29.4 °C

Figure 6 and Figure 7 show that fault impact ratios change differently with evaporator water outlet temperature at different condenser water inlet temperature. When the condenser water outlet temperature is 18 °C, the fault impact ratios increase with evaporator water outlet temperature; when the condenser water outlet temperature is 29.4°C, the fault impact ratios decrease with evaporator water inlet temperature. However, the fault impact ratios still increase with fault levels monotonically for all faults under different evaporator water outlet temperature in Figure 6 and Figure 7.

Figure 8 and Figure 9 show the changes of fault impact ratios with fault levels at different normalized chiller loads when the condenser water outlet temperatures are 18°C and 29.4°C.

Figure 8 Changes of predicted fault impact ratio with fault levels at different chiller load for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 18 °C

Figure 9 Changes of predicted fault impact ratio with fault levels at different chiller load for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 29.4 °C

Figure 8 and Figure 9 show that the fault impact ratios increase with fault levels under different normalized chiller loads, and the fault impact ratios also increase with normalized chiller load for overcharging and non-condensable faults. However, Figure 8(b) and Figure 9(b) show that the load has negligible effect on fault impact ratios for excess oil. Also, Figure 8(d) shows that the fault impact ratio for condenser fouling reaches a minimum when the normalized load increases from 0.3 to 0.7 for condenser fouling fault when the condenser water inlet temperature is 18°C. This shows that the effect of normalized load on fault impact ratios changes with condenser water inlet temperature and the types of fault.

The trends predicted by the fault impact ratio models are reasonable indicating there are no overfitting issues and the models are suitable for estimating the effects of faults on chiller power consumption.

6. Use of fault models in building simulation

To test the use of water-cooled chiller fault models in building simulation program, a post-1980 period hospital model from the DOE commercial reference building models (Deru et al. 2011) that were created for EnergyPlus was used. Three different climate zones 1A, 2A, 2B were simulated to compare fault impacts for the buildings under different loads and ambient conditions. The definitions of the climate zones and representative cities (Deru et al. 2011; ASRHAE 2013) are tabulated in Table 5.

Table 5 Details of climate zones 1A, 2A and 2B

Climate zone	Description	Representative city
1A	Very hot and wet	Miami, Florida, United States
2A	Hot and wet	Houston, Texas, United States
2B	Hot and dry	Phoenix, Arizona, United States

The hospital model has a total floor area of 22,422m² with 5 stories and a basement. It has a water-cooled electric chiller, a natural gas boiler, 2 variable-air-volume systems and 2 constant-air-volume systems. The required rated capacity of space conditioning equipment was calculated for each climate using the hospital model. Then, the fault models were appended to the hospital model. EnergyPlus version 8.2 was used to perform annual simulations using TMY3 weather data (Wilcox and Marion 2008). For the simulations, the ranges of inputs to the fault models were limited to the conditions in Table 1 and Table 3 so that the models operated within their applicable range. The condenser fouling model was not used for this case study because it would also be necessary to include a model for how the pump flow rate and power consumption change with fouling.

To insert the fault model described in Eqn. (3) within EnergyPlus, Ruby scripts were written with the assistance of the OpenStudio Ruby library (OpenStudio 2015) for insertion of Erl scripts. Erl scripts are similar to Fortran programming statements that are specifically designed to create customized subroutines within EnergyPlus. The fault models were implemented to alter the outputs of the chiller model after each execution of the chiller model during each time step of the building simulation. During each time step, the fault model script uses the inputs from other routines to estimate the fault impact ratio using Eqn. (3), and the estimated fault impact ratio is used to increase the chiller power consumption.

Changes in electricity and water uses due to different fault levels in different climate zones are tabulated in Table 6, Table 7 and Table 8 for this case study.

Table 6 Simulated changes of electricity and water uses of the hospital from its non-faulted operation in climate zone 1A

Fault	Percentage change of chiller electricity use	Percentage change of cooling tower electricity use	Percentage change of building electricity use	Percentage change of building water use	Percentage change of building electricity peak demand
Overcharged by 20%	2.5%	0.0%	0.8%	0.3%	1.5%
Overcharged by 40%	5.1%	0.0%	1.6%	0.6%	3.1%
Excess oil by 30%	1.8%	0.0%	0.6%	0.2%	0.6%
Excess oil by 60%	3.6%	0.0%	1.1%	0.4%	1.3%
Non-condensable in refrigerant fault by 2.5%	7.4%	0.0%	2.3%	0.9%	3.4%
Non-condensable in refrigerant fault by 5%	14.8%	0.0%	4.7%	1.8%	7.0%

Table 7 Simulated changes of electricity and water use of the hospital from its non-faulted operation in climate zone 2A

Fault	Percentage change of chiller electricity use	Percentage change of cooling tower electricity use	Percentage change of building electricity use	Percentage change of building water use	Percentage change of building electricity peak demand
Overcharged by 20%	2.4%	0.0%	0.7%	0.3%	1.8%
Overcharged by 40%	4.9%	0.0%	1.4%	0.6%	3.6%
Excess oil by 30%	1.9%	0.0%	0.5%	0.2%	0.7%
Excess oil by 60%	3.8%	0.0%	1.1%	0.4%	1.5%
Non-condensable in refrigerant fault by 2.5%	7.0%	0.0%	2.0%	0.8%	3.8%
Non-condensable in refrigerant fault by 5%	14.1%	0.0%	4.0%	1.7%	7.8%

Table 8 Simulated changes of electricity and water use of the hospital from its non-faulted operation in climate zone 2B

Fault	Percentage change of chiller electricity use	Percentage change of cooling tower electricity use	Percentage change of building electricity use	Percentage change of building water use	Percentage change of building electricity peak demand
Overcharged by 20%	2.0%	0.0%	0.5%	0.1%	1.4%
Overcharged by 40%	4.1%	0.0%	0.9%	0.3%	2.9%
Excess oil by 30%	2.0%	0.0%	0.4%	0.1%	0.6%
Excess oil by 60%	4.0%	0.0%	0.9%	0.3%	1.2%
Non-condensable in refrigerant fault by 2.5%	6.2%	0.0%	1.4%	0.4%	3.2%
Non-condensable in refrigerant fault by 5%	12.6%	0.0%	2.9%	0.8%	6.5%

The tables show the intuitive trend that a higher fault level leads to a higher fault impact for buildings. In particular, the faults can significantly increase the required electricity consumption of the

chillers to satisfy the same cooling load as the non-faulted case. They also lead to a relatively small increase in the water consumption of the cooling tower because the chiller condenser heat transfer rate is increased with the increased chiller power consumption, leading to higher condenser water temperatures and therefore higher evaporation rates from the cooling tower. There is a very slight increase in the cooling tower fan power consumption presumably because the cooling tower control increases the fan air flow in response to higher condenser water temperatures. The overall peak electricity demand for the hospital increases in all three climates primarily as a result of increased chiller power.

The overall fault impacts are typically greater in hotter and more humid climates. For instance, at a 5% non-condensable fault level, the electricity use of the hospital in climate zone 2B is increased by 1.4%. However, in a more humid climate (climate zone 2A), the percentage increase of electricity use is 2.0%. For the hospital situated in a hotter climate (climate zone 1A) than climate zone 2A, the percentage increase is 4.7%. The only exception is the electricity peak demand under a hotter climate. For instance, the 20% overcharging fault increases the electricity peak demand in climate zone 2A by 1.8%, but the same fault only increases the electricity peak demand by 1.5% in a hotter climate (climate zone 1A). Similar comparisons of electricity peak demand between climate zone 1A and 2A can be made for the other faults.

The results also show that building electricity use and electricity peak demand respond differently to different faults. For instance, in Table 8, a 40% overcharging fault level led to a 0.9% increase in building electrical use and a 2.9% increase in peak demand. In contrast, a 60% excess oil fault level led to a similar 0.9% increase in electrical usage but a lower demand impact of only 1.2%. Similar behavior is observed for the overcharging and excess oil fault results in Table 6 and Table 7.

7. Conclusions

This paper presented a general empirical modeling approach for characterizing the impacts on power consumption of four faults applied to water-cooled centrifugal chillers: overcharging, excess oil, non-condensable in refrigerant and water-side condenser fouling. The models can be used to adjust the standard outputs of existing chiller models and therefore are easily implemented in existing simulation programs. The empirical fault models were trained and validated using laboratory chiller test data available from Comstock (1999a, b). A case study was carried out using a hospital model that was simulated in three different climates. The results demonstrate the different sensitivities of building energy use, water use, and peak demand to different faults and fault levels. Fault sensitivities are greater in hotter and more humid climates. Furthermore, energy use and demand have different sensitivities to different faults.

The fault impact model was developed for water-cooled chillers using data over a limited range of conditions as defined in Table 1, Table 2 and Table 3. Therefore, it may not be applicable to other chiller types or outside the range of conditions for which it was developed. It should also be noted that the model only modifies the power consumption of the chiller and not the maximum cooling capacity. Therefore, it is not useful for estimating the loss of comfort that might occur when the cooling capacity is not sufficient to meet the cooling load requirements. Finally, the form of the model requires inputs of evaporator water outlet temperature, a condenser water inlet temperature and a chiller cooling load. If alternative inputs were used, then the form of the correlations would need to be modified.

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Nomenclature

a_i	the i^{th} coefficient of the empirical model for non-faulted chiller power consumption [unit varies]
c_i	the i^{th} coefficient of the empirical model for fault impact ratio of the chiller power consumption [unit varies]
F	fault level [dimensionless]
FIR_W	fault impact ratio defined as the ratio of the measured power consumption at a given fault level to the output of the empirical model for non-faulted chiller power consumption at the same operating condition [dimensionless]
$\widehat{FIR}_W$	predicted fault impact ratio from the empirical models for different faults [dimensionless]
h_l	enthalpy of saturated water [kJ kg^{-1}]
N_{fault}	number of cases for each fault [dimensionless]
P	number of empirical coefficients in Eqn. (3)
$\dot{Q}_{\text{evap}}$	evaporator heat transfer rate or chiller load [kW]
$\dot{Q}_{\text{evap, rat}}$	rated evaporator heat transfer rate [kW]
ρ	density of water [kg m^{-3}]
t	time [s]
$T_{\text{cond, in}}$	temperature at condenser water inlet [K]
$T_{\text{evap, flow}}$	temperature at evaporator water flow meter [K]
$T_{\text{evap, in}}$	temperature at evaporator water inlet [K]
$T_{\text{evap, out}}$	temperature at evaporator water outlet [K]
$\dot{V}_{\text{evap, flow}}$	evaporator water flow rate [$\text{m}^3 \text{s}^{-1}$]
$\dot{W}_F$	chiller power consumption from faulty experiments [kW]
$\widehat{W}_{NF}$	predicted chiller power consumption from empirical model of non-faulted chiller power consumption [kW]

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Appendix A: Acquiring chiller steady state performance

In Comstock and Braun (1999a), the evaporator heat transfer rates were calculated from water volumetric flow rate, inlet temperature, and outlet water temperatures assuming constant density and specific heat of water with temperature. Steady state data points were established based on duration of time under given operating conditions rather than using a quantitative definition of steady state such as used in the steady state detector of Li and Braun (2007). To consider the changes of water properties with temperature and to use a more rigorous definition of steady state performance in this paper, the steady state performance of the chiller was calculated with a different approach from the original one in Comstock and Braun (1999a).

The steady state definition was defined by the rates of change of temperature values and water volumetric flow rates in the data. The original raw data contained data with a 10s sampling interval. A 15-minute data acquisition period in the raw data file is identified as a steady state data point if its average rates of changes of temperature values were smaller than 0.05K/minute and its average rates of changes of water volumetric flow rates were smaller than 0.1%/minute for 15 minutes. A 15-minute period was chosen because the minimum length of steady state operation was documented as 15 minutes in Comstock and Braun (1999a). To avoid having data points with similar testing conditions, the start and end time between data points must be longer than 15 minutes. The resultant number of data points for each fault is shown in Table 9.

Table 9 Number of steady state data points for each fault

Fault	Number of steady state data points	Number of steady state data points in Comstock and Braun (1999b)
Non-faulted operation	28	54
Overcharging	41	108
Excess oil	48	107
Non-condensable in refrigerant	38	108
Condenser fouling	54	135

Each 15-minute steady-state window within the raw data contains measurements of power consumption, temperature and water flow rates at 90 time instants. The average value for each steady-state data point was obtained by averaging the measurements of power consumption, temperature and water flow rate at all time instants within the steady-state window. To calculate the evaporator heat transfer rate at each data point, the instantaneous evaporator heat transfer rates at each time instant t in the window were calculated based on Eqn. (6), where enthalpy and density values are calculated based on Wagner et al. (2002).

$$\dot{Q}_{evap}(t) = \rho(T_{evap,flow}(t))\dot{V}_{evap,flow}(t)(h_l(T_{evap,in}(t)) - h_l(T_{evap,out}(t))) \quad (6)$$

After calculating the instantaneous evaporator heat transfer rates, the evaporator heat transfer rate for a steady state data point was obtained by averaging the instantaneous evaporator heat transfer rates within the steady-state window.

The steady state performance in the non-faulted cases was used to train the coefficients in Eqn. (1) using linear regression as tabulated in Appendix B, and the steady state performance in the faulty cases was used to calculate the fault impact ratios for each data point by Eqn. (2).

Appendix B: Values of empirical coefficients

This appendix lists the coefficients of empirical equations involved in this paper.

Table 10 Values of coefficients in Eqn. (1)

Coefficients	a_1 [kW]	a_2 [kW/K]	a_3 [kW/K]	a_4 [--]	a_5 [/K]	a_6 [/K]	a_7 [/kW]
Values	35.68	-0.3766	0.3256	-1.291	0.0001256	-0.0005340	0.005188

Table 11 Values of coefficients in Eqn. (3) for different faults

Coefficients	Overcharging	Excess oil	Non-condensable in refrigerant	Condenser fouling
c_1 [--]	-8.854	-6.513	-18.86	-4.278
c_2 [/K]	-0.01228	-0.008258	-0.1033	-0.02031
c_3 [/K]	0.04126	0.02976	0.1616	0.03281
c_4 [--]	0.2838	0.007701	0.3135	0.6404
c_5 [--]	0.1807	0	3.150	0.05028
c_6 [--]	0	0.04893	0.8737	0

Table 1 Environmental conditions in the chiller tests

Evaporator water inlet temperature [°C]	6.3 – 16.2
Evaporator water outlet temperature [°C]	4.0 – 11.6
Condenser water inlet temperature [°C]	17.8 – 30.0
Evaporator water flow rate [m ³ h ⁻¹]	48.6 – 50.0
Condenser water flow rate [m ³ h ⁻¹]	60.1 – 61.9

Table 2 Fault levels being tested

Fault	Fault levels	Definition of fault level in Comstock and Braun (1999a)	Testing method
Refrigerant overcharge	0% to 40%	Percentage of mass of refrigerant above the manufacturer recommendation	Adding more refrigerant to the refrigerant circuit than manufacturer recommendation
Excess oil	0% to 68%	Percentage of the mass of oil in the chiller above the manufacturer recommendation	Filling the refrigeration system with more oil than manufacturer recommended oil level
Non-condensable in refrigerant	0% to 5%	Ratio of the volume of non-condensable in the chiller at atmospheric pressure and temperature to the volume of the refrigerant circuit	Adding laboratory-grade nitrogen into the refrigerant circuit before charging refrigerant into the system
Condenser fouling	0% to 45%	Proportion of water tubes blocked in the condenser	Blocking water flow paths in the condenser

Table 3 Range of normalized chiller loads for different fault models

Normalized chiller loads	Overcharging	Excess oil	Non-condensable in refrigerant	Condenser fouling
Minimum	0.277	0.269	0.274	0.268
Maximum	0.932	0.968	0.849	0.987

Table 4 Fit standard error of accuracy of Eqn. (3) to estimate fault impact ratio for chiller power consumption

Fault	Overcharging	Excess oil	Non-condensable in refrigerant	Condenser fouling
FSE	0.0154	0.0107	0.0197	0.0119

Table 5 Details of climate zones 1A, 2A and 2B

Climate zone	Description	Representative city
1A	Very hot and wet	Miami, Florida, United States
2A	Hot and wet	Houston, Texas, United States
2B	Hot and dry	Phoenix, Arizona, United States

Table 6 Simulated changes of electricity and water uses of the hospital from its non-faulted operation in climate zone 1A

Fault	Percentage change of chiller electricity use	Percentage change of cooling tower electricity use	Percentage change of building electricity use	Percentage change of building water use	Percentage change of building electricity peak demand
Overcharged by 20%	2.5%	0.0%	0.8%	0.3%	1.5%
Overcharged by 40%	5.1%	0.0%	1.6%	0.6%	3.1%
Excess oil by 30%	1.8%	0.0%	0.6%	0.2%	0.6%
Excess oil by 60%	3.6%	0.0%	1.1%	0.4%	1.3%
Non-condensable in refrigerant fault by 2.5%	7.4%	0.0%	2.3%	0.9%	3.4%
Non-condensable in refrigerant fault by 5%	14.8%	0.0%	4.7%	1.8%	7.0%

Table 7 Simulated changes of electricity and water use of the hospital from its non-faulted operation in climate zone 2A

Fault	Percentage change of chiller electricity use	Percentage change of cooling tower electricity use	Percentage change of building electricity use	Percentage change of building water use	Percentage change of building electricity peak demand
Overcharged by 20%	2.4%	0.0%	0.7%	0.3%	1.8%
Overcharged by 40%	4.9%	0.0%	1.4%	0.6%	3.6%
Excess oil by 30%	1.9%	0.0%	0.5%	0.2%	0.7%
Excess oil by 60%	3.8%	0.0%	1.1%	0.4%	1.5%
Non-condensable in refrigerant fault by 2.5%	7.0%	0.0%	2.0%	0.8%	3.8%
Non-condensable in refrigerant fault by 5%	14.1%	0.0%	4.0%	1.7%	7.8%

Table 8 Simulated changes of electricity and water use of the hospital from its non-faulted operation in climate zone 2B

Fault	Percentage change of chiller electricity use	Percentage change of cooling tower electricity use	Percentage change of building electricity use	Percentage change of building water use	Percentage change of building electricity peak demand
Overcharged by 20%	2.0%	0.0%	0.5%	0.1%	1.4%
Overcharged by 40%	4.1%	0.0%	0.9%	0.3%	2.9%
Excess oil by 30%	2.0%	0.0%	0.4%	0.1%	0.6%
Excess oil by 60%	4.0%	0.0%	0.9%	0.3%	1.2%
Non-condensable in refrigerant fault by 2.5%	6.2%	0.0%	1.4%	0.4%	3.2%
Non-condensable in refrigerant fault by 5%	12.6%	0.0%	2.9%	0.8%	6.5%

Table 9 Number of steady state data points for each fault

Fault	Number of steady state data points	Number of steady state data points in Comstock and Braun (1999b)
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c_4 [--]	0.2838	0.007701	0.3135	0.6404
c_5 [--]	0.1807	0	3.150	0.05028
c_6 [--]	0	0.04893	0.8737	0

Figure(s)

Figure 1 Change of chiller power consumption with decreasing amount of refrigerant under a wide range of conditions (Evaporator water outlet temperature from 277.35K to 283.9K, condenser water inlet temperature from 289.25K to 303.36K, evaporator water inlet temperature from 279.4K to 289.5K, evaporator water flow rate at $49\text{m}^3\text{ h}^{-1}$ and condenser water flow rate at $61\text{m}^3\text{ h}^{-1}$)

Figure 2 Simplified schematic of the experimental setup in Comstock and Braun (1999a)

Figure 3 Input-output diagram of the fault model at design

Figure 4 Comparison of estimated fault impact ratios of different faults with the ratios in the training data

Figure 5 Changes of predicted fault impact ratio with fault levels at different condenser water inlet temperature for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling

Figure 6 Changes of predicted fault impact ratio with fault levels at different evaporator water outlet temperature for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 18 °C

Figure 7 Changes of predicted fault impact ratio with fault levels at different evaporator water outlet temperature for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 29.4 °C

Figure 8 Changes of predicted fault impact ratio with fault levels at different chiller load for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 18 °C

Figure 9 Changes of predicted fault impact ratio with fault levels at different chiller load for (a) overcharging, (b) excess oil, (c) non-condensable in refrigerant and (d) condenser fouling when the condenser water inlet temperature is 29.4 °C