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ScienceDirect
International Journal of Refrigeration 30 (2007) 299–310
REVUE INTERNATIONALE DU FROID
INTERNATIONAL JOURNAL OF
refrigeration
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An overall performance index for characterizing the economic impact of faults in direct expansion cooling equipment

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Received 13 October 2005; received in revised form 28 April 2006; accepted 7 July 2006
Available online 24 October 2006

Abstract

Air conditioning is a non-critical application for fault detection and diagnosis (FDD) where decisions about servicing faults should involve the use of economics. Existing methods for evaluating impacts of faults on equipment performance only consider some individual factors such as the equipment coefficient of performance (COP) or cooling capacity. This paper develops an overall economic performance degradation index (EPDI) for air conditioning equipment that includes the combined effects of degradations in COP, cooling capacity, and sensible heat ratio (SHR). EPDI quantifies the performance degradation caused by faults based on economics so it can be used as part of the decision making process in an overall FDD system. Furthermore, EPDI can be used along with estimates of typical field performance degradations to assess the economic benefits associated with the application of automated FDD. A case study is presented where EPDI was applied to measurements for an existing unit where faults were artificially introduced.

Keywords: Air conditioning; Cooling system; Process; Detection; Fault

Grille globale d'évaluation de la performance utilisée afin de caractériser l'impact économique des pannes survenant dans les systèmes de refroidissement à détente directe

Mots clés : Conditionnement d'air ; Système frigorifique ; Procédé ; Détection ; Anomalie ; COP ; Performance

1. Introduction

Direct expansion cooling equipment using air-to-refrigerant heat exchangers is predominantly used for cooling residential and light commercial buildings, including cooling-only split systems, cooling-only packaged systems, heat pumps and window air conditioners. Unlike larger

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doi:10.1016/j.ijrefrig.2006.07.026

Nomenclature

CompLeak	Compressor valve leakage
CondFoul	Condenser fouling
COP	Coefficient of performance
$\overline{C}_{elec}$	Average electricity price [$\$\,kW^{-1}\,h^{-1}$]
$\overline{C}_{equip}$	Average equipment price [$\$\,kW^{-1}\,h^{-1}$]
C_{hourly}	Hourly rate [$\$\,h^{-1}$]
ΔOC	Cost penalty associated with not servicing faults [$]
ΔT_{evap}	Evaporating temperature decrease due to the fault [°C]
ΔT_{ms}	Temperature difference across evaporator [°C]
ΔT_{ss}	Supply air temperature increase caused by reduced evaporator airflow [°C]
EC	Equipment costs [$\$\,kW^{-1}$]
EPDI	Economic performance degradation index
EvapFoul	Evaporator fouling
FDD	Fault detection and diagnosis
h_{mix}	Mixed air enthalpy [$J\,kg^{-1}$]
ma	Mixed air
LLRestr	Liquid-line restriction
NonCond	Presence of non-condensable gas
OC	Operation cost [$]
Q_{lat}	Latent cooling load [kW h]
Q_{sens}	Sensible cooling load [kW h]
Q_{tot}	Total cooling load [kW h]
Q_v	Cooling loads for the ventilation flow stream [kW h]
Q_z	Cooling loads for the conditioned zone [kW h]
$\dot{Q}_{cap}$	Cooling capacity [kW]
RefHigh	High refrigerant charge
RefLow	Low refrigerant charge
$r_{\Delta cap}$	Capacity degradation ratio
$r_{\Delta COP}$	COP degradation ratio
$r_{\Delta runtime}$	Runtime increase ratio
$r_{\Delta SHR}$	SHR degradation ratio
$r_{\Delta W}$	Power consumption increase ratio
r_{cap}	Capacity ratio
r_{EC}	Normalized equipment cost
r_{COP}	COP ratio
r_{equip}	Ratio of average equipment price for faulty operation to the normal value
r_{OC}	Normalized total operating cost
r_{SHR}	SHR ratio
r_{UC}	Normalized utility cost
$r_{\Delta w}$	Electricity consumption increase ratio
SHR	Sensible heat ratio
T_{evap}	Evaporating temperature [°C]
T_{ma}	Mixed air temperature [°C]
T_{sp}	Coil supply air temperature [°C]
T_{run}	Runtime [h]
T_{sh}	Superheat [°C]
T_{suc}	Suction line temperature [°C]
UC	Utility costs [$\$\,kW^{-1}$]
$\dot{V}_{ea}$	Evaporator air volume flow rate [$m^3\,s^{-1}$]
W	Equipment electrical consumption (including compressors and fans) [kW h]
$\dot{W}$	Power consumption (including compressors and fans) [kW]

Greek letters

α	The angle formed by the iso-enthalpy line and the iso-humidity line [rad]
β	The angle formed by tangent of the saturated line and an iso-humidity line [rad]
γ	Heat-gain ratio
ϕ	Evaporating temperature degradation ratio

chilled-water systems, they are not well maintained and typically operate in harsh environments. They are often affected by faults introduced during initial installation or developed in routine operation.

Fault detection and diagnosis (FDD) aims at early identification and isolation of premature faults that are not severe enough to cause significant performance degradation or equipment failure so that corrective measures can be taken proactively. A growing number of publications related to FDD have appeared in the last decade as documented by Comstock et al. [1] and Li [2]. According to the IEA ANNEX 34 final report edited by Dexter and Pakanen [3], 23 prototype FDD performance monitoring tools and three validation tools have been developed, 30 demonstrations have been taken place in 20 buildings, 26 FDD tools have been tested in real buildings, and four performance monitoring schemes have been jointly evaluated on three documented data sets from real buildings. Since 2001, 39 more papers have appeared [2]. Katipamula and Brambley [4,5] conducted a thorough review on methods for automated FDD and prognostics for building systems. This review provided a framework for categorizing methods, identified their primary strengths and weaknesses, addressed their applications specific to the fields of HVAC&R, and briefly discussed the future of automated diagnostics in buildings.

The primary consequences of faults in HVAC systems are comfort-related, environmental and economic instead of safety-critical. Rossi and Braun [6] developed the four fault evaluation criteria (comfort, economics, environment, and safety) and four fault decision criteria (tolerate, repair ASAP, adapt control, and stop to repair) for HVAC&R equipment. Since the safety criterion most means the safety of the most costly component – the compressor, so it is essentially an economic criterion. Impacts of faults on comfort and environment are relatively straightforward to evaluate, while economic impacts are complicated and evaluation of economic performance degradation is crucial for the development of automated FDD. With respect to economic issues, FDD systems should incorporate economic performance degradation as one criterion to assess fault severity and justify fault service as part of the FDD technique. In addition, economic evaluation of fault impacts also contributes to the assessment of the economic benefits associated with application of FDD.

The definition and evaluation of performance degradations are under-investigated issues for FDD. Previous investigations [7–9] used degradation of energy efficiency rating (EER) or coefficient of performance (COP) as an economic index and cooling capacity degradation as a comfort index for evaluating fault severity and FDD sensitivity. For example, Rossi and Braun [7] developed a near-optimal service scheduling algorithm for the cleaning of heat exchangers in air conditioning equipment and demonstrated that there is a significant opportunity for cost savings associated with optimal scheduling of condenser and evaporator maintenance. This algorithm only considered one economic criterion − EER and two heat exchanger fouling faults. However, degradation of EER or COP is not sufficient for evaluating the overall economic performance degradation and all other faults are also worthy of interest.

The current paper investigates factors that impact economic performance and defines an overall economic performance degradation index (EPDI) for air conditioning equipment. In addition, a case study application for EPDI is presented. The faults considered in the development and application of EPDI are the same as those considered by Li and Braun [10,11] and other investigators. These include faults that degrade compressor flow capacity such as compressor valve leakage (CompLeak), high refrigerant charge (RefHigh), low refrigerant charge including leakage or inadequate charging during service (RefLow), air-side fouling or loss of flow for the condenser (CondFoul) or evaporator (EvapFoul), a liquid-line restriction such as filter/dryer clogging (LLRestr), and presence of a non-condensable gas (NonCond).

2. Economic performance degradation evaluation method

2.1. Factors impacting economic performance

In the following subsections, three factors impacting economic performance are investigated: COP, cooling capacity, and sensible heat ratio (SHR) degradations.

2.1.1. COP degradation

Since EER and COP are equivalent parameters, the following discussion is based on COP. For a given time period (e.g., 1 h), the average COP for air conditioning equipment is

$$\mathrm{COP} = \frac{Q_{\mathrm{tot}}}{W} \tag{1}$$

where Q_{tot} is the equipment cooling load (including loads for the conditioned zone (Q_z) and ventilation flow stream (Q_v)) in kilowatt hours and W is the equipment electrical consumption (including compressors and fans) in kilowatt hours. Therefore, if Q_{tot} for a certain space is fixed, equipment with higher COP would consume less electricity. If any fault tends to degrade COP, it would result in more electrical consumption. A COP ratio (r_{COP}) is defined as the ratio of the actual COP to the COP with the unit operating normally,

$$r_{\mathrm{COP}} = \frac{\mathrm{COP}}{\mathrm{COP}_{\mathrm{normal}}} \tag{2}$$

A degradation ratio for the increase in electrical consumption is defined as

$$r_{\Delta W} = \frac{W - W_{\mathrm{normal}}}{W_{\mathrm{normal}}} = \frac{1}{r_{\mathrm{COP}}}\frac{Q_{\mathrm{tot}}}{Q_{\mathrm{tot,normal}}} - 1 \tag{3}$$

where the subscript ‘normal’ denotes a no-fault condition.

If the equipment cooling load is independent of refrigeration faults as assumed in previous investigations, then

$$Q_{\mathrm{tot}} = Q_{\mathrm{tot,normal}} \tag{4}$$

and

$$r_{\Delta W} = \frac{1}{r_{\mathrm{COP}}} - 1 \tag{5}$$

Therefore, an increase in electrical consumption due to faults is a unique function of r_{COP}, if cooling load was independent of refrigeration faults. However, equipment cooling load is impacted by faults primarily due to their impacts on sensible heat ratio (SHR).

2.1.2. SHR degradation

Sensible cooling loads for AC equipment are insensitive to refrigeration cycle faults unless the faults and weather conditions are severe enough so that the unit’s sensible cooling capacity is no longer sufficient to meet the building load requirement. The insensitivity to faults is because the cooling equipment is typically controlled to maintain space temperatures and the ventilation settings do not change with the refrigeration faults considered in this paper. Thus, the unit operates longer with refrigeration faults to maintain the space conditions but the total sensible cooling provided is essentially the same as for no faults.

On the other hand, moisture removal by the equipment can be strongly coupled to refrigeration cycle faults under any condition where moisture is being condensed on the evaporator. The sensible heat ratio, SHR, is a measure of the moisture removal performance of the AC unit. SHR is the ratio of sensible cooling to total cooling provided by the air conditioning equipment. It characterizes the cooling

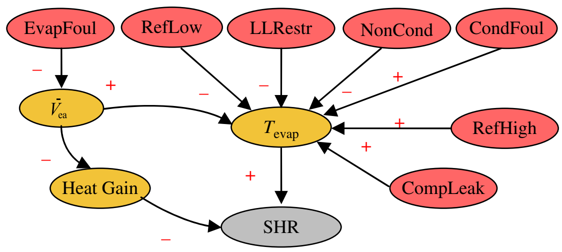

Fig. 1. Faults and SHR.

equipment performance, but it also characterizes the composition of the cooling load.

2.1.2.1. Relationship between SHR and cooling load. For a given air conditioning system using temperature as the control set point, the sensible load is independent of the equipment and its faults as long as the room temperature set point can be maintained. Two air conditioning processes with different SHR and the same sensible load (Q_{sens}) have different latent loads (Q_{lat}) and thus total loads (Q_{tot})

$$\frac{Q_{tot,2}}{Q_{tot,1}} = \frac{SHR_1}{SHR_2} \tag{6}$$

From the above equation, it can be concluded that the total cooling load is inversely proportional to SHR if the sensible load is fixed. That is, the smaller the SHR the bigger the total cooling load demand. So any fault which tends to reduce the equipment SHR will raise the equipment cooling load. The SHR ratio (r_{SHR}) is defined as the ratio of actual SHR to the value for normal operation (SHR_{normal}):

$$r_{SHR} = \frac{SHR}{SHR_{normal}} \tag{7}$$

and

$$\frac{Q_{tot}}{Q_{tot,normal}} = \frac{SHR_{normal}}{SHR} = \frac{1}{r_{SHR}} \tag{8}$$

Substituting Eq. (8) into Eq. (3)

$$r_{\Delta W} = \frac{1}{r_{COP} r_{SHR}} - 1 \tag{9}$$

As indicated by Eq. (9), the electrical consumption degradation ratio is a function of both r_{COP} and r_{SHR}. The following subsection addresses the impact of faults on SHR.

2.1.2.2. Faults and SHR. For a given cooling system, there are three main factors which determine SHR:

(1) *Mixed air status*: if a cooling system operates under wet conditions, the mixed air wet-bulb temperature is a direct driving condition for the cooling system operation, and the higher the mixed air wet-bulb temperature, the lower the SHR. If a cooling system operates under dry conditions, the SHR is a constant one.
(2) *Evaporator airflow rate*: the higher the evaporator airflow rate, the higher the SHR in that higher evaporator airflow rate reduces the time that the air and the evaporator surface are in contact and causes larger portion of the air to bypass the evaporator without processing. For a given setting of the evaporator fan speed, the evaporator airflow rate is approximately fixed for normal operation. However, evaporator and/or filter fouling can have a significant impact on airflow.
(3) *Evaporating temperature (T_{evap})*: the lower the evaporating temperature, the lower the SHR in that lower evaporator temperatures causes higher mass transfer potential. The evaporator temperature is influenced by faults and by the mixed air and outdoor conditions.

Refrigeration cycle faults generally impact evaporation temperature and SHR. However, evaporator fouling has the largest impact on SHR because it causes a reduction in the evaporator airflow rate. Reduced evaporator airflow leads to a lower evaporating temperature, which results in a lower SHR. In addition, reduced evaporator airflow rate causes an increase in duct heat gain, which further reduces SHR. Fig. 1 illustrates how the faults impact SHR. The symbol '−' means that an increase in the input reduces the output, and vice versa. The impact of faults on SHR can be read by means of multiplying all the minus and plus signs from the target fault to SHR. A final negative value after multiplication indicates that the target fault decreases SHR, while a positive value indicates that the target fault increases SHR. For example, from evaporator

Table 1
Impact of faults on T_{evap}, SHR, Q_{tot} and $\dot{Q}_{cap}$ for a TXV system

Faults	Compressor leakage (CompLeak)	Condenser fouling (CondFoul)	Evaporator fouling (EvapFoul)	Liquid-line restriction (LLRestr)	Refrigerant low-charge (RefLow)	Refrigerant high-charge (RefHigh)	Non-condensables (NonCond)
T_{evap}	+++	+	−−−−	−−	−−−−	+	−
SHR	+++	+	−−−−−	−−	−−−−	+	−
Q_{tot}	−−−	−	+++++	+	++++	−	+
$\dot{Q}_{cap}$	−−−−−	−	−−−	−	−−−−	−	−−

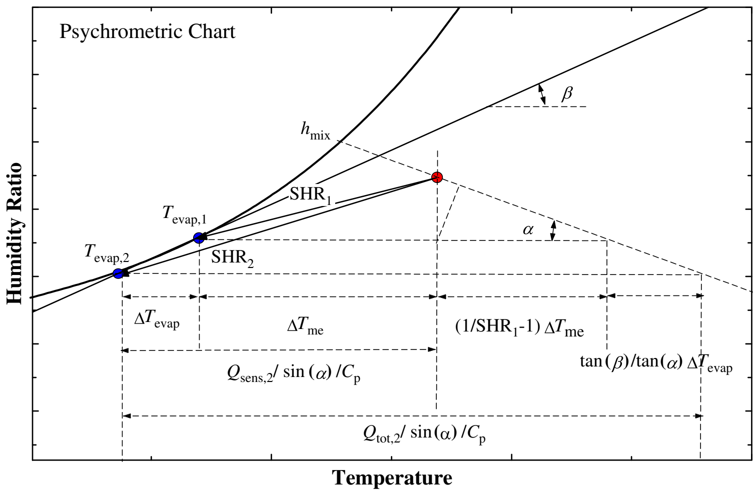

Fig. 2. Relationship between SHR and T_{evap}.

fouling to SHR, there are three minus signs and the final value after multiplication is negative, so evaporator fouling fault decreases SHR.

Table 1 summarizes the impact of the six faults on T_{evap}, SHR, Q_{tot}, and $\dot{Q}_{cap}$ for a system using thermal expansion valve (TXV) as an expansion device. The plus and minus signs indicate the direction and relative magnitude of the changes with respect to different faults. For instance, evaporator fouling results in a significant decrease in SHR due to a lower evaporator temperature. This leads to an increase in the equipment cooling load due to greater moisture removal. However, the sensible cooling and total capacity of the unit decreases. This means that the unit must operate longer to meet the sensible building load.

2.1.2.3. Derivation of impact of T_{evap} on SHR. As discussed, the major impact of faults on SHR is due to changes in evaporating temperature. Fig. 2 illustrates how a decrease in the evaporating temperature impacts SHR for a simple model that assumes that the evaporator air outlet state is on a straight line process between the mixed air (evaporator inlet) state and saturated air at the evaporating temperature (T_{evap}). For this model, it can be shown that

$$\begin{aligned}\mathrm{SHR}_2 &= \frac{Q_{sens,2}}{Q_{tot,2}} = \frac{Q_{sens,2}/\sin(\alpha)/C_p}{Q_{tot,2}/\sin(\alpha)/C_p}\\ &= \frac{\Delta T_{evap}+\Delta T_{me}}{\Delta T_{evap}+\Delta T_{me}+(1/\mathrm{SHR}_1-1)\Delta T_{me}+\tan(\beta)/\tan(\alpha)\Delta T_{evap}}\\ &= \frac{1+\phi}{1/\mathrm{SHR}_1+\left(1+\dfrac{\tan(\beta)}{\tan(\alpha)}\right)\phi} = \frac{1+\phi}{1+\phi\left(1+\dfrac{\tan(\beta)}{\tan(\alpha)}\right)\mathrm{SHR}_1}\mathrm{SHR}_1\end{aligned} \tag{10}$$

where the subscripts '1' and '2' denote conditions for fault-free and faulty performance, respectively, Q_{sens} is the sensible cooling load, α is the angle formed by the iso-enthalpy line and the iso-humidity line (or horizontal axis); β is the angle formed by the tangent of the saturated line at the temperature of $T_{evap,1}$ and an iso-humidity line; C_p is the moisture air specific heat at constant pressure; ΔT_{evap} is the evaporating temperature decrease due to the fault; ΔT_{me} is the difference between mixed air temperature and the evaporating temperature; and $\phi = \Delta T_{evap}/\Delta T_{me}$ is defined as the evaporating temperature degradation ratio.

Since the axes of the psychrometric chart can be constructed in arbitrary scales, coordinates with different axis scales will lead to different slopes of the iso-enthalpy lines and different slopes of the tangent of the saturation line. For a given coordinate frame, $\tan(\alpha)$ is pretty constant with a value of about $0.62c$ (where c is a constant related to a given coordinate frame); $\tan(\beta)$ is a function of $T_{evap,1}$ and varies over a relatively large range ($0.45c$–$0.81c$ corresponding to the evaporating temperature range of 0–10 °C). For example, for normal operation with $T_{evap,1}$ of 7 °C and SHR_1 of 0.75, if the evaporator temperature degradation ratio was $\phi = 0.18$ (=3/17) because of a moderate fault, then SHR_2 (0.69) would be reduced by about 8%.

2.1.2.4. Additional impact of evaporator fouling on SHR. Evaporator fouling reduces both sensible and total cooling capacity due to the decrease in airflow rate. However, evaporator fouling leads to lower evaporating temperatures which result in lower SHR, so the sensible capacity is reduced more than the total capacity. In addition, the net sensible cooling capacity can be further reduced with evaporator fouling because of additional heat gain from the duct and fan motor. This reduction in sensible capacity leads to a further reduction in SHR beyond what would occur in the absence of heat gain.

Fig. 3 shows the impact of heat gain on SHR for a linear process model. In this figure, T_{sp} is the supply air temperature (evaporator air outlet temperature), ΔT_{ss} is the supply air temperature increase caused by reduced evaporator airflow, and ΔT_{ms} is the temperature difference across evaporator without considering heat gain. The latent load is the same for the two processes, and it can be shown that

$$Q_{lat} = \frac{1-SHR_1}{SHR_1} Q_{sens,1} = \frac{1-SHR_2}{SHR_2} Q_{sens,2} \tag{11}$$

$$\frac{SHR_2}{1-SHR_2}\frac{1-SHR_1}{SHR_1} = \frac{Q_{sens,2}}{Q_{sens,1}} = \frac{\Delta T_{ms}-\Delta T_{ss}}{\Delta T_{ms}} = 1-\frac{\Delta T_{ss}}{\Delta T_{ms}} = 1-\gamma \tag{12}$$

and

$$SHR_2 = \frac{(1-\gamma)}{1-\gamma SHR_1} SHR_1 \tag{13}$$

where γ is defined as the heat-gain ratio, $\Delta T_{ss}/\Delta T_{ms}$, which is determined by the airflow rate and supply air temperature for a given system and ambient environment. Since SHR and γ are less than 1, the factor $(1-\gamma)/(1-\gamma SHR_1)$ is always less than 1. For a given SHR_1, the larger the heat-gain ratio, the more SHR decreases. For a given γ, the smaller the SHR_1, the more SHR decreases. For example, for normal operation with an SHR_1 of 0.69, if the heat-gain ratio is 0.15 because of evaporator fouling, then SHR_2 would reduce by about 5% just because of the heat-gain effect.

The overall impacts of evaporator fouling on SHR are illustrated in Fig. 4. SHR_1 and $T_{sp,1}$ are characteristic of process 1 under normal operation, while process 2 has a lower SHR_2 because of lower evaporating temperature under evaporator fouling. If heat gain is not considered, the supply air temperature would be $T_{sp,2}$ (with a lower bypass factor). Although any faults causing lower evaporating temperature would result in certain additional heat gain from the duct, only the effect of evaporator fouling is significant enough to be considered. So the actual supply air temperature would be $T_{sp,3}$ corresponding to a sensible heat ratio of SHR_3.

Although additional heat gain occurs with evaporator fouling, the net effect is that the actual supply air temperature, $T_{sp,3}$, is lower than the normal operation value, $T_{sp,1}$, but not as low as would occur without the additional heat gains, $T_{sp,2}$. For the previous example conditions, the total reduction in SHR would be about 13% and the building cooling load would increase by about 15% for a moderate evaporator fouling fault.

2.1.3. Cooling capacity degradation

Several investigators have documented the impact of faults on cooling capacity for direct expansion cooling equipment. Generally, FDD methods such as the method of Li and Braun [10], can diagnose faults before there has been about a 5–10% loss in capacity. Cooling capacity

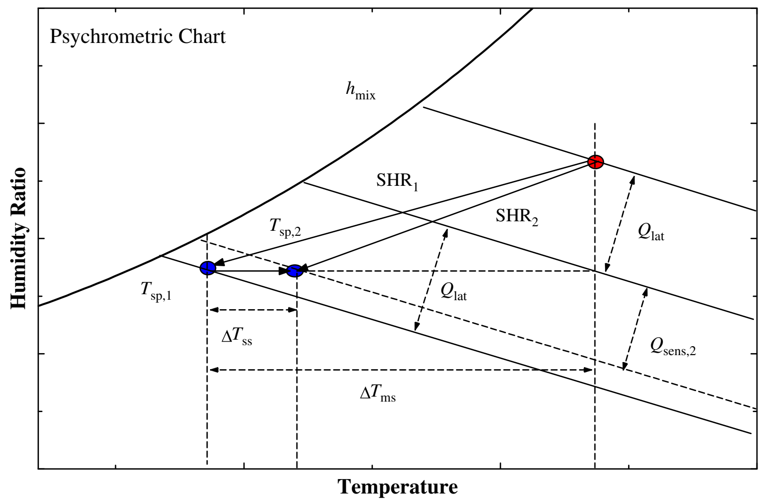

Fig. 3. Degradation in cooling capacity and SHR.

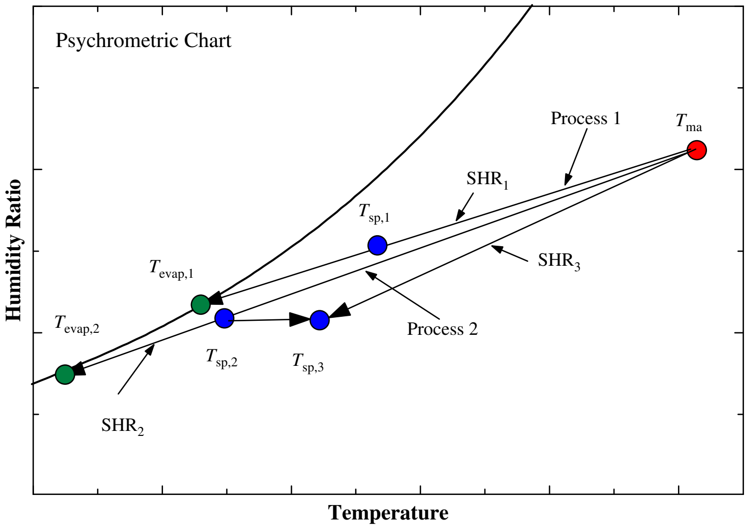

Fig. 4. Overall impacts of evaporator fouling on SHR.

can be estimated during operation using low-cost measurements and a virtual refrigerant mass flow rate sensor. If the compressor is known to operate normally, the refrigerant mass flow rate sensor estimates the refrigerant mass flow rate using a model trained by compressor map data; otherwise, if a compressor fault is identified, the virtual refrigerant mass flow rate sensor can estimate the refrigerant mass flow rate using a compressor energy balance model [11].

2.2. *Development of an economic performance degradation index (*EPDI*)*

As it was discussed earlier, the increase in electrical energy consumption due to faults is a function of r_{COP} and r_{SHR} and is independent of any cooling capacity degradation. However, in addition to utility cost (UC), there is an equipment cost (EC) associated with maintaining the comfort in the conditioned space that is influenced by capacity degradation. Equipment has a certain lifetime and it wears and loses value with increasing runtime. Equipment needs to be maintained regularly because of some certain evolving faults and serviced unexpectedly because of random faults. The longer the equipment runs, the higher the probability that the equipment will develop faults. Overall, EC is proportional to runtime (T_{run}) and any faults that increase T_{run} result in an increase in EC. The combination of a cooling load increase due to reduced SHR and cooling capacity degradation cause an increase in T_{run}. The runtime required to meet a given cooling load is estimated as

$$T_{run} = \frac{Q_{tot}}{\overline{\dot{Q}}_{cap}} \tag{14}$$

and a degradation ratio for the increase in runtime due to a fault is estimated in terms of the degradation ratios for SHR and cooling capacity as follows:

$$r_{\Delta runtime} = \frac{T_{run} - T_{run,normal}}{T_{run,normal}} = \frac{Q_{tot}}{Q_{tot,normal}} \frac{\overline{\dot{Q}}_{cap,normal}}{\overline{\dot{Q}}_{cap}} - 1 = \frac{1}{r_{SHR} r_{cap}} - 1 \tag{15}$$

where $\overline{\dot{Q}}_{cap}$ is the average equipment cooling capacity and r_{cap} is a cooling capacity ratio, $\overline{\dot{Q}}_{cap}/\overline{\dot{Q}}_{cap,normal}$.

Fig. 5 illustrates relationships between costs (utility and energy) and other factors. The parameters with a bar overhead denote averaged quantities. The air conditioning equipment has three types of inputs: driving conditions, fan settings and faults; and five outputs of interest: $\overline{SHR}_{RS}$, $\overline{COP}$, $\overline{\dot{W}}$, $\overline{\dot{Q}}_{cap}$ and $\overline{C}_{equipment}$. The conditioned space and ventilation system produce the cooling load Q_{tot} and the distribution system transports the supply air to the conditioned space.

The utility costs (UC) over a particular runtime T_{run} are estimated as:

$$UC = \overline{\dot{W}}\,\overline{C}_{elec} T_{run} = \frac{\overline{\dot{Q}}_{cap}}{\overline{COP}} \overline{C}_{elec} \frac{Q_{tot}}{\overline{\dot{Q}}_{cap}} = \overline{C}_{elec} \frac{Q_{tot}}{\overline{COP}} \tag{16}$$

where $\overline{\dot{W}}$ is the average unit power consumption, $\overline{C}_{elec}$ ($\$\,kW^{-1}\,h^{-1}$) is the average price for electricity, and $\overline{\dot{Q}}_{cap}$ is the average cooling capacity. The effects of demand costs are not directly considered in this simple utility cost model. However, they could be considered in an approximate manner by choosing an appropriate average cost of electricity that reflects the costs of energy and demand for the site. The periods over which the electricity rates and COP are averaged in this model depend upon the application. For performance monitoring or service decision making, the averaging periods could be small (e.g., 1 h). For evaluating the benefits of FDD, long time periods (e.g., cooling season) with simplified assumptions for operating conditions would be employed.

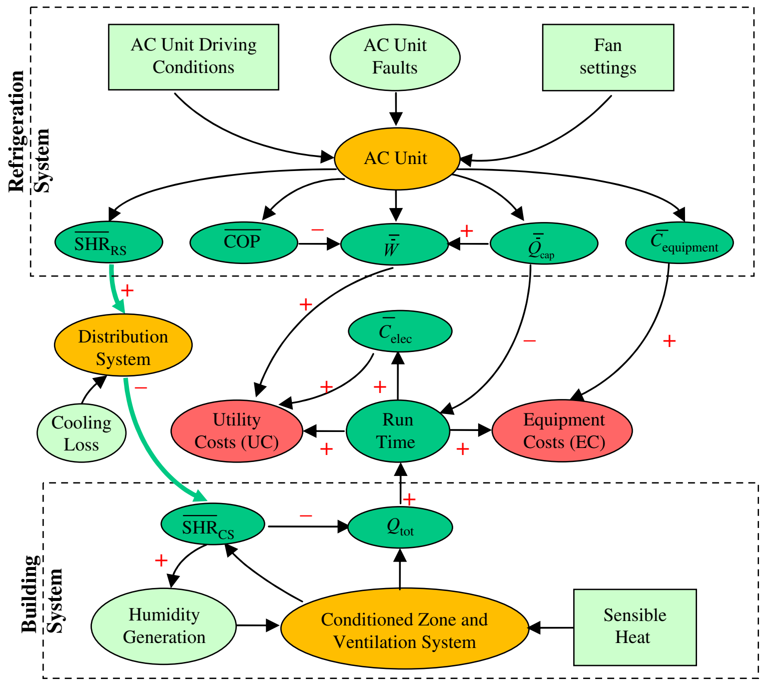

Fig. 5. Factors affecting air conditioning system performance.

Eqs. (2), (7), and (16) can be used to determine a simplified expression for the utility costs normalized relative to the costs for the fault-free case as

$$r_{\mathrm{UC}} = \frac{\mathrm{UC}}{\mathrm{UC}_{\mathrm{normal}}} = \frac{Q_{\mathrm{tot}}/\overline{\mathrm{COP}}}{Q_{\mathrm{tot,normal}}/\overline{\mathrm{COP}}_{\mathrm{normal}}} = \frac{\overline{\mathrm{SHR}}_{\mathrm{normal}}}{\overline{\mathrm{SHR}}} \frac{\overline{\mathrm{COP}}_{\mathrm{normal}}}{\overline{\mathrm{COP}}} = \frac{1}{r_{\mathrm{SHR}} r_{\mathrm{COP}}} \tag{17}$$

The equipment cost model assumes that equipment costs are linear functions of runtime, so that

$$\mathrm{EC} = \overline{C}_{\mathrm{equip}} T_{\mathrm{run}} = \overline{C}_{\mathrm{equip}} \frac{Q_{\mathrm{tot}}}{\overline{\dot{Q}}_{\mathrm{cap}}} \tag{18}$$

where $\overline{C}_{\mathrm{equip}}$ (\$ kW^{-1} h^{-1}) is the average cost per unit of runtime to purchase, install, maintain, and service the equipment.

In addition to increasing runtime, faults could also speed the wear of components and possibly cause abrupt failures (such as compressor and fan motor failures), which could have a significant impact on $\overline{C}_{\mathrm{equip}}$. Although it is difficult to quantify this effect, a factor is incorporated to address this issue. The equipment costs normalized relative to the costs for the fault-free case are determined as

$$r_{\mathrm{EC}} = \frac{\mathrm{EC}}{\mathrm{EC}_{\mathrm{normal}}} = \frac{r_{\mathrm{equip}} Q_{\mathrm{tot}}/\overline{\dot{Q}}_{\mathrm{cap}}}{Q_{\mathrm{tot,normal}}/\overline{\dot{Q}}_{\mathrm{cap,normal}}} = r_{\mathrm{equip}} \frac{\overline{\mathrm{SHR}}_{\mathrm{normal}}}{\overline{\mathrm{SHR}}} \frac{\overline{\dot{Q}}_{\mathrm{cap,normal}}}{\overline{\dot{Q}}_{\mathrm{cap}}} = \frac{r_{\mathrm{equip}}}{r_{\mathrm{SHR}} r_{\mathrm{cap}}} \tag{19}$$

where r_{equip} is the ratio of $\overline{C}_{\mathrm{equip}}$ for faulty operation to the normal value.

Defining $\overline{C}_{\mathrm{utility}} = \overline{C}_{\mathrm{elec}} \overline{\dot{W}}_{\mathrm{normal}}$ (\$ h^{-1}), then the total operating cost (OC) associated with maintaining comfort for the conditioned space, which is the sum of the utility costs (UC) and equipment costs (EC), can be expressed as

$$\mathrm{OC} = \mathrm{UC} + \mathrm{EC} = \left(r_{\mathrm{UC}} \overline{C}_{\mathrm{utility}} + r_{\mathrm{EC}} \overline{C}_{\mathrm{equip}}\right) T_{\mathrm{run,normal}} \tag{20}$$

where it should be noted that $\overline{C}_{\mathrm{equip}}$ is for normal operation.

A normalized total operating cost, r_{OC}, is

$$r_{\mathrm{OC}} = \frac{\mathrm{OC}}{\mathrm{OC}_{\mathrm{normal}}} = \frac{r_{\mathrm{UC}} \overline{C}_{\mathrm{utility}} + r_{\mathrm{EC}} \overline{C}_{\mathrm{equip}}}{\overline{C}_{\mathrm{utility}} + \overline{C}_{\mathrm{equip}}} \tag{21}$$

which can be rewritten as

$$r_{\mathrm{OC}} = r_{\mathrm{UC}} w_{\mathrm{u}} + r_{\mathrm{EC}} (1 - w_{\mathrm{u}}) \tag{22}$$

where

$$w_{\mathrm{u}} = \frac{\overline{C}_{\mathrm{utility}}}{\overline{C}_{\mathrm{utility}} + \overline{C}_{\mathrm{equip}}} \tag{23}$$

The economic performance degradation index, EPDI, is defined as the net increase in the normalized total costs or

$$\mathrm{EPDI} = r_{\mathrm{OC}} - 1 = r_{\mathrm{UC}} w_{\mathrm{u}} + r_{\mathrm{EC}} (1 - w_{\mathrm{u}}) - 1 \tag{24}$$

Alternatively, EPDI can be written as

$$\mathrm{EPDI}=\frac{1}{r_{\mathrm{SHR}}}\left(\frac{1}{r_{\mathrm{COP}}}\frac{\overline{C}_{\mathrm{utility}}}{\overline{C}_{\mathrm{utility}}+\overline{C}_{\mathrm{equip}}}+\frac{1}{r_{\mathrm{cap}}}\frac{r_{\mathrm{equip}}\overline{C}_{\mathrm{equip}}}{\overline{C}_{\mathrm{utility}}+\overline{C}_{\mathrm{equip}}}\right)-1 \tag{25a}$$

If $r_{\Delta\mathrm{COP}}$, $r_{\Delta\mathrm{cap}}$ and $r_{\Delta\mathrm{SHR}}$ are defined as the degradation ratio of COP, $\dot{Q}_{\mathrm{cap}}$, and SHR, respectively: $r_{\Delta\mathrm{COP}}=(\mathrm{COP}_{\mathrm{normal}}-\mathrm{COP})/\mathrm{COP}_{\mathrm{normal}}=1-r_{\mathrm{COP}}$, $r_{\Delta\mathrm{cap}}=(\dot{Q}_{\mathrm{cap,normal}}-\dot{Q}_{\mathrm{cap}})/\dot{Q}_{\mathrm{cap,normal}}=1-r_{\mathrm{cap}}$, and $r_{\Delta\mathrm{SHR}}=(\mathrm{SHR}_{\mathrm{normal}}-\mathrm{SHR})/\mathrm{SHR}_{\mathrm{normal}}=1-r_{\mathrm{SHR}}$, then, Eq. (25a) can be rewritten as:

$$\mathrm{EPDI}=\frac{1}{1-r_{\Delta\mathrm{SHR}}}\left(\frac{1}{1-r_{\Delta\mathrm{COP}}}\frac{\overline{C}_{\mathrm{utility}}}{\overline{C}_{\mathrm{utility}}+\overline{C}_{\mathrm{equip}}}+\frac{1}{1-r_{\Delta\mathrm{cap}}}\frac{r_{\mathrm{equip}}\overline{C}_{\mathrm{equip}}}{\overline{C}_{\mathrm{utility}}+\overline{C}_{\mathrm{equip}}}\right)-1 \tag{25b}$$

EPDI relates performance degradation parameters for air conditioning equipment due to faults to the net increase of normalized total costs associated with maintaining a conditioned space. Faults tend to decrease r_{cap}, r_{COP}, and r_{SHR} but increase r_{equip}, $r_{\Delta\mathrm{cap}}$, $r_{\Delta\mathrm{COP}}$, and $r_{\Delta\mathrm{SHR}}$. More severe the faults lead to higher values of EPDI.

3. Evaluation and application of EPDI

EPDI could be used directly within the context of an FDD system to monitor performance and possibly to evaluate whether service should be performed. For service decisions, it would be necessary to set a threshold for EPDI above which service would be recommended. Alternatively, EPDI could be used to estimate cost savings associated with servicing faults and then tradeoffs between service and operating costs could be evaluated directly. Li [2] present a method for service scheduling that employs EPDI in estimating operating cost savings for service. Evaluation of operating cost savings through servicing is also necessary for evaluating the benefits of automated FDD. Li and Braun [12] present a study that employs EPDI for evaluating FDD benefits.

The cost penalty associated with not servicing faults or conversely the cost savings for fault service (ΔOC) can be determined from EPDI as

$$\Delta\mathrm{OC}=\mathrm{EPDI}\times\mathrm{OC}_{\mathrm{normal}} \tag{26a}$$

or

$$\Delta\mathrm{OC}=\frac{\mathrm{EPDI}}{1+\mathrm{EPDI}}\times\mathrm{OC} \tag{26b}$$

where OC is the cost before service and $\mathrm{OC}_{\mathrm{normal}}$ is the expected cost after service.

In order to evaluate EPDI using Eqs. (25a) and (25b), it is necessary to estimate several factors from on-site measurements and simple models. The degradation factors for cooling capacity, COP, and SHR require both current values from measurements and estimates of normal values for the same operating conditions. Both fan power and compressor power consumptions are included to calculated COP. All the fault-free values for cooling capacity, COP and SHR are determined from normal models correlated using the manufacturers' data. The manufacturers' data are firstly divided into dry-condition data and wet-condition data, and then are correlated separately using the first-order polynomials without cross terms.

Current cooling capacity is determined using low-cost measurements and a virtual sensor for refrigerant flow as documented by Li and Braun [11]. To determine current COP, the rated fan power is assumed and the compressor power consumption is estimated using a virtual sensor as described by Li and Braun [11]. Current SHR can be determined using the measured parameters of the evaporator inlet and outlet air: mixed air dry-bulb temperature and humidity, and supply air dry-bulb temperature and humidity. Typically, supply air humidity is not measured but it can be determined using a virtual sensor as described by Li and Braun [11].

4. Case studies

Li and Braun [10] performed extensive fault testing on a packaged air conditioner to evaluate FDD performance.

Table 2
Method of implementing faults and corresponding fault levels simulated

Faults	Simulation method	Fault level expression	Fault level simulated					
			0	1	2	3	4	5
CompLeak	Partially open a bypass valve between discharge and suction lines	% Refrigerant mass flow rate bypass	0	8	18	33	44	56
CondFoul	Partially block condenser airflow with paper	% Reduction of air volume flow rate	0	3	10	13	16	
EvapFoul	Partially block evaporator airflow with paper	% Reduction of air volume flow rate	0	5	9	16	31	
LLRestr	Partially close the needle valve on the liquid line	% of the pressure drop from high to low sides	0	5	10	13	19	
RefLow	Under-charge the system	% Reduction of charge	0	11	16	21	26	32
RefHigh	Overcharge the system	% Increase of charge	0	11	16	21	26	32

The data from this study are used here in order to demonstrate the impact of faults on EPDI. The installed system is a 18 kW rooftop air conditioner having a rated COP of 3.2. Table 2 tabulates the method of implementation and corresponding levels that were simulated for the six faults considered individually.

In order to estimate EPDI, the following assumptions were employed:

(1) Normal equipment life, $T_{\text{equipment life}}$, is 10 years and 12,000 h of runtime.
(2) The average equipment costs, including capital costs and initial installation, are \$250 kW^{-1}.
(3) $r_{\text{equip}} = 1.0$.
(4) The average maintenance and service costs are \$11.4 per year-kW.
(5) $\overline{C}_{\text{elec}} = \$0.08\ \text{kW}^{-1}\ \text{h}^{-1}$

Assumptions (1)–(4) lead to an equipment cost of \$0.53 h^{-1} of runtime and a utility cost of \$0.44 h^{-1} of runtime for normal operation. Power consumption for fault-free operation and degradation factors for cooling capacity, COP and SHR were calculated at each operating condition as outlined in the previous section.

Fig. 6 plots EPDI and degradations in cooling capacity, COP and SHR with increasing severity of a low refrigerant charge fault. All four indices increase with decreasing refrigerant charge. However, EPDI increases much faster than the other three indices because it incorporates the combined impacts of the other three indices. For these results, the highest value of EPDI was 0.6. This means the operating costs for this fault level are 160% of the costs if the unit were operating at its normal charge level.

Fig. 7 plots EPDIs as a function of fault levels for the six faults implemented individually. For the range of faults considered, compressor leakage, low refrigerant charge and evaporator fouling had a significant impact on operating costs, whereas a condenser fouling fault had a moderate impact and liquid-line restriction and refrigerant overcharge faults had small impacts. However, any general conclusions regarding the importance of individual faults depend on the severity of the fault levels chosen for testing in relation to levels that are typically encountered in the field. It is interesting to note that the EPDI for compressor leakage is insignificant at low fault levels and then increases drastically with increasing fault level. This is because an increase in SHR compensates for degradations in cooling capacity and COP at low fault levels, whereas at higher fault levels SHR saturates at one and degradations in cooling capacity and COP become more significant.

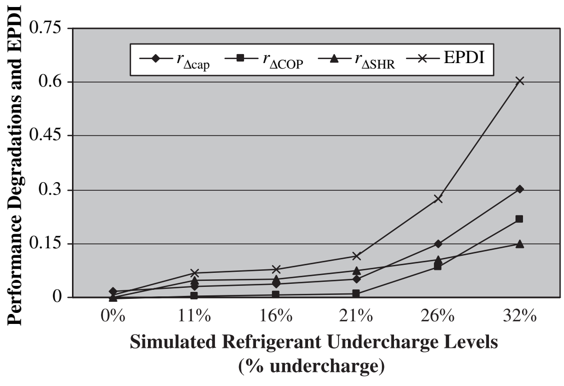

Fig. 6. Performance degradations and EPDI for refrigerant low-charge faults.

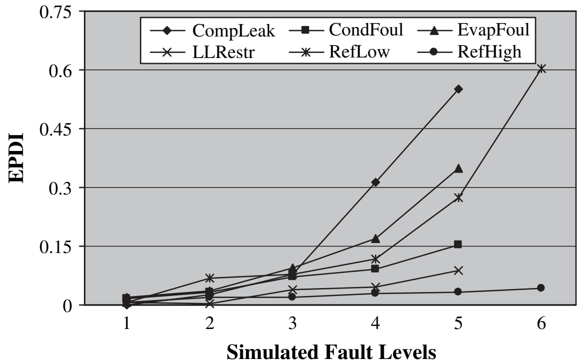

Fig. 7. Impact of fault levels on EPDI for different faults.

In addition, multiple-simultaneous faults were tested by Li and Braun [10]. There are 41 possible combinations of multiple-simultaneous faults among the six types of faults. Table 3 describes the individual fault levels for each combinations along with results for EPDI and degradations for each individual performance index of cooling capacity, COP, and SHR. By comparison, different fault combinations have different impacts on each individual performance index; none of the individual performance indices are coincident with the trend of EPDI; and faults may cause relatively small degradations in cooling capacity or COP but the overall EPDI is significant due to the degradation in SHR. In sum, none of the individual performance indices are adequate for evaluating the economic impacts of faults while EPDI incorporates all three individual performance indices and describes the overall economic performance degradations. For instance, for the test number 6, capacity and COP are only degraded by about 5% and yet the economic impact is 20%. For the test number 33, the combination of liquid-line restriction and refrigerant overcharge has trivial impacts on cooling capacity, COP, SHR and thus EPDI in that the TXV is capable to react to these two faults and compensate their impacts within low-level fault degrees. In addition, the negative values of $r_{\Delta\text{cap}}$, $r_{\Delta\text{COP}}$ and EPDI indicate that the TXV using a proportional control algorithm raises refrigerant mass flow rate due to overshooting; the impacts of liquid-line restriction and refrigerant overcharge on SHR are compensated each other in that they have opposite impacts on SHR.

Table 3
Individual fault levels and degradation indices for multiple-simultaneous fault tests

Test no.	CompLeak (%)	CondFoul (%)	EvapFoul (%)	LLRestr (%)	RefLow (%)	RefHigh (%)	$r_{\Delta cap}$	$r_{\Delta COP}$	$r_{\Delta SHR}$	EPDI
1	27	0	0	0	14	0	0.28	0.19	−0.06	0.27
2	27	11	0	0	14	0	0.31	0.25	−0.09	0.29
3	25	11	12	0	11	0	0.25	0.20	−0.06	0.25
4	25	11	12	12	11	0	0.27	0.22	−0.04	0.30
5	0	11	12	12	11	0	0.09	0.12	0.14	0.31
6	0	0	12	12	11	0	0.05	0.04	0.12	0.20
7	0	0	0	12	14	0	0.05	0.00	0.10	0.15
8	29	0	0	12	14	0	0.30	0.21	−0.06	0.31
9	25	0	12	12	11	0	0.26	0.17	−0.02	0.28
10	25	0	12	0	11	0	0.25	0.17	0.00	0.30
11	0	0	12	0	11	0	0.04	0.01	0.10	0.14
12	0	11	12	0	11	0	0.05	0.09	0.08	0.18
13	0	11	0	0	14	0	0.06	0.07	0.04	0.12
14	0	11	0	12	14	0	0.06	0.06	0.10	0.19
15	29	11	0	12	14	0	0.29	0.23	−0.07	0.30
16	*32*	*11*	*0*	*0*	*0*	*0*	*0.34*	*0.28*	*−0.18*	*0.26*
17	*21*	*11*	*12*	*0*	*0*	*0*	*0.25*	*0.21*	*−0.02*	*0.29*
18	*21*	*11*	*12*	*12*	*0*	*0*	*0.21*	*0.17*	*−0.03*	*0.22*
19	*0*	*11*	*12*	*12*	*0*	*0*	*0.06*	*0.10*	*0.09*	*0.20*
20	*0*	*0*	*12*	*12*	*0*	*0*	*0.01*	*0.00*	*0.08*	*0.09*
21	*19*	*0*	*12*	*12*	*0*	*0*	*0.21*	*0.14*	*−0.02*	*0.21*
22	*32*	*0*	*0*	*12*	*0*	*0*	*0.33*	*0.24*	*−0.15*	*0.25*
23	*0*	*11*	*0*	*12*	*0*	*0*	*−0.03*	*0.04*	*0.02*	*0.02*
24	*32*	*11*	*0*	*12*	*0*	*0*	*0.28*	*0.25*	*−0.15*	*0.21*
25	*0*	*11*	*12*	*0*	*0*	*0*	*0.06*	*0.10*	*0.06*	*0.16*
26	*19*	*0*	*12*	*0*	*0*	*0*	*0.20*	*0.13*	*−0.05*	*0.16*
27	33	0	0	0	0	21	0.30	0.23	−0.16	0.20
28	32	11	0	0	0	21	0.28	0.24	−0.17	0.18
29	35	11	16	0	0	21	0.39	0.35	−0.09	0.50
30	35	11	16	12	0	21	0.36	0.33	−0.09	0.44
31	0	11	16	12	0	21	0.08	0.15	0.08	0.24
32	0	0	16	12	0	21	0.07	0.08	0.09	0.20
33	0	0	0	12	0	21	−0.03	−0.01	0.00	−0.02
34	32	0	0	12	0	21	0.32	0.25	−0.13	0.27
35	35	0	16	12	0	21	0.38	0.31	−0.06	0.48
36	35	0	16	0	0	21	0.38	0.31	−0.07	0.49
37	0	0	16	0	0	21	0.07	0.08	0.08	0.18
38	0	11	16	0	0	21	0.08	0.15	0.07	0.23
39	0	11	0	0	0	21	0.03	0.10	−0.01	0.06
40	0	11	0	12	0	21	0.03	0.11	0.01	0.09
41	32	11	0	12	0	21	0.34	0.31	−0.16	0.31

5. Conclusions and discussions

This paper quantified the factors that influence the impact of faults on operating costs for air conditioning equipment. Degradation in equipment COP influences economics because of increased energy input required to meet a specific cooling requirement. Cooling capacity degradation influences the amount of time that the unit needs to run in order to meet the cooling load and thereby influences economics through reduced equipment life. A decrease in sensible heat ratio increases operating costs due to higher latent loads for the same sensible load. An overall economic performance degradation index, termed EPDI, was defined that considers all three of these effects. Case studies were performed to demonstrate the impact of faults on EPDI. In general, EPDI has a much greater dependence on faults than COP, which is most often proposed as an index for monitoring system performance. EPDI could be used for system performance monitoring and for decision making as part of an FDD system. For example, Li [2] present a method for service scheduling that employs this index, and Li and Braun [12] utilize EPDI to estimate the benefits of automated FDD. To calculate EPDI, it is essential to evaluate the fault impacts on each individual performance index of cooling capacity, COP, and SHR, which may be a possible barrier to the implementation of EPDI. The difficulty is how Li and Braun [11] described and validated practical methods in which normal operation values of each individual performance index can be predicted using fault-free models and current operation values can be estimated using virtual sensors. All these fault-free models and virtual sensors are based on low-cost measurements such as temperatures and pressures and trained by readily available manufacturers' data such as compressor map data.

Acknowledgement

This work was supported by the California Energy Commission (CEC) with additional cost-sharing provided by Honeywell, Inc. and Field Diagnostic Services, Inc.

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