DETECTING CONDENSER FAULTS IN COMMERCIAL REFRIGERATION SYSTEMS

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Abstract

Fault detection in commercial refrigeration systems is an important topic due to both economic and food safety reasons. If faults can be detected and diagnosed before the system drifts outside the specified operational envelope, service costs can be reduced and in extreme cases the costly discarding of food products can be avoided. In the situations where the operational requirements can be met with a fault present, the system will operate with a higher energy consumption increasing the cost of operation. The objective of this study is to develop a robust method for detecting condenser faults in commercial refrigeration systems under the wide operational conditions that they are exposed to. The approach described uses a non-linear parity equation comparing the heat transfer rates of the air and the refrigerant of the condenser. The paper presents the detection method and discusses the application of the method on experimental results obtained on a Danish supermarket.

1 INTRODUCTION

One of the trends in commercial refrigeration is the establishing of monitoring and service centers that handle temperature and other operational alarms for a large number of supermarkets. The center staff can access the supermarket control system by modem or a network and investigate the cause of alarms. In cases where a repair is necessary, the staff can dispatch service technicians to carry out the repair. By providing the monitoring centers with more specific information about the fault types, they can optimize their procedures for handling the faults and reduce the service cost even further. The objective of this paper is to describe a fault detection scheme that is robust enough to handle the large variations in operating conditions that display cabinets are exposed to. The paper treats the class of faults that causes a reduction of the air flowing through the condenser. The proposed detection scheme uses a non-linear parity equation approach (Gertler, 1993), comparing the heat transfer rate to the refrigerant with the heat transfer rate from the air. If the estimated airflow deviates from the airflow at a no-fault condition, the parity equation indicates a fault. The paper is organized as follows: The refrigeration system considered is described in Section 2 where also the model of the condenser is discussed. The proposed method for fault detection is described in Section 3, and in Section 4 the direct and indirect causes of the faults are discussed. The results obtained when applying the method to data, obtained from a medium size supermarket, are presented and discussed in section 5.

2 REFRIGERATION SYSTEM

The high pressure side of a refrigeration system is illustrated in Figure 1. A group of compressors, that are capacity controlled, compresses the refrigerant from the low pressure side of the system to the high pressure. The compressors are typically controlled to keep the low pressure at a desired level. The condenser receives the hot gas from the compressors, cools it to the condensing temperature and condenses the refrigerant. The refrigerant, now on liquid form, can then be fed to an expansion valve.

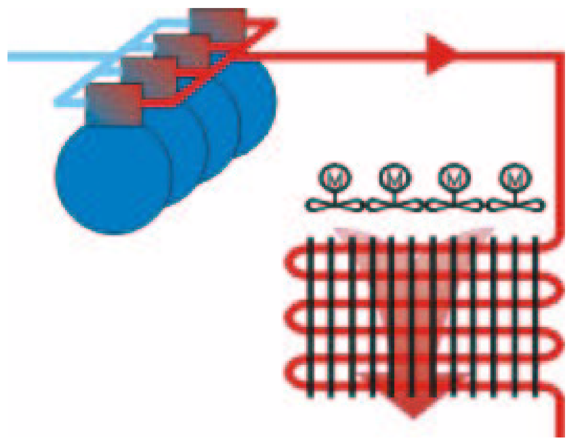

Figure 1: The high pressure side of a commercial refrigeration system.

2.1 Condenser model

A condenser is basically a heat-exchanger, that cools the hot primary side refrigerant with a colder secondary fluid. The condensers are typically constructed with more passes of the refrigerant to obtain a cross flow function. This increases the cooling capabilities of the secondary fluid. An idealized temperature profile is illustrated in Figure 2

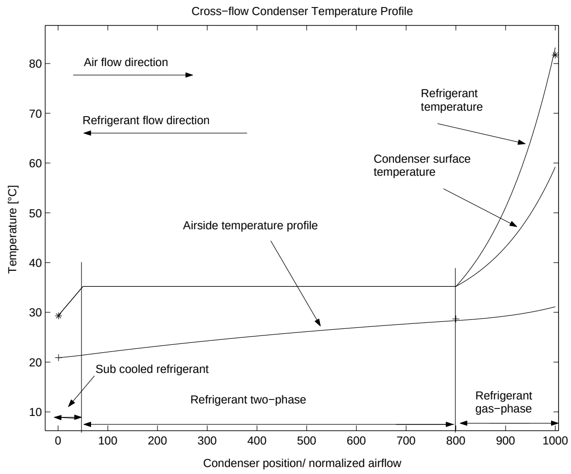

Figure 2: An idealized condenser temperature profile.

The secondary fluid is in the following assumed to be air, but other fluids can also be applied. For this type of evaporators the flow on the air-side is unmixed, whereas the flow on the refrigerant side is normally fully mixed. This means that the measurements of air temperatures are "local". The typical pattern of operation is dynamic due to the cut-in and -out of the compressors. However, to keep the model of the condenser simple the thermal capacity of the condenser material (pipes and fins) is neglected. The model of the condenser can then be divided into two parts: a refrigerant side and an air-side.

2.1.1 Refrigerant side

The heat transfer rate to the refrigerant can be described as

$$\dot{Q}_{ref} = \dot{m}_{ref}(h_{ref,out} - h_{ref,in}), \tag{1}$$

$$h_{ref,in} = g_1(T_{ref,in}, P_{ref,in}), \tag{2}$$

$$h_{ref,out} = g_2(T_{ref,out}, P_{ref,out}), \tag{3}$$

where

$\dot{Q}_{ref}$: Heat transfer rate
$\dot{m}_{ref}$: Refrigerant mass flow rate
$h_{ref,in}$: Refrigerant enthalpy at the condenser inlet
$h_{ref,out}$: Refrigerant enthalpy at the condenser outlet
$T_{ref,in}$: Refrigerant inlet temperature
$P_{ref,in}$: Refrigerant inlet pressure
$T_{ref,out}$: Refrigerant outlet temperature (liquid state)

g_1 and g_2 are specific functions for the particular type of refrigerant. Both in- and outlet temperatures and pressures are measured by the control system.

2.1.2 Air side

Figure 2 illustrates the temperature profile of both cross-flow condenser and the air flow. The air's outlet temperature over the condenser, $T_{air,out}$ is modelled by using an airflow-dependent variable α. The temperature of the condenser surface in the two-phase and the liquid regions, is assumed to be equal to the condensing temperature. Additionally, the surface temperature for the gas region is assumed to be the mean of the gas temperature and the liquid temperature. The outlet temperature is given by the following relation

$$T_{air,out} = g_T(S_d, P_c, \alpha) \tag{4}$$

S_d : Refrigerant temperature at the compressor outlet
P_c : Condensing pressure

g_T denotes a function through which the outlet temperature is computed. The variable α is assumed to be a linear function of the nominal air speed over the condenser, i.e.

$$\alpha = k_\alpha . v_{air,nom}, \tag{5}$$

where k_α is a proper constant. The value of the nominal air speed, $v_{air,nom}$, is dependent on the condenser type and hence known. Knowing the temperature values, the heat transfer rate from the air can be determined in a straight manner. This is shown in the following:

$$\dot{Q}_{air} = \dot{m}_{air}(h_{air,out} - h_{air,in}), \tag{6}$$

$$h_{air,in} = g_3(T_{air,in}, RH_{air,in}), \tag{7}$$

$$h_{air,out} = g_4(T_{air,out}, RH_{air,out}), \tag{8}$$

where

$\dot{Q}_{air}$: Heat transfer rate
$\dot{m}_{air}$: Air mass flow rate
$h_{air,in}$: Air enthalpy at the condenser inlet
$h_{air,out}$: Air enthalpy at the condenser outlet
$T_{air,in}$: Air inlet temperature
$P_{air,in}$: Air inlet pressure
$T_{air,out}$: Air outlet temperature
$P_{air,out}$: Air outlet pressure
RH : Air relative humidity

Similar to the refrigerant case, the functions g_3 and g_4 are specific functions used to compute the air enthalpy at the inlet and the outlet.

The specific enthalpy on the air side is also a function of the air pressure. However, the variations in ambient pressure are usually small and they are therefore not significant for the calculation of specific enthalpy. The relative humidities in the inlet and outlet are not measured, but for a certain cabinet type (multi deck, open top), and temperature level (medium or low) good estimates of both can be made when knowing the relative humidity in the sales area.

3 Structural modeling

Consider the system S as a set of components $\bigcup_{i=1}^{m} C_i$, each imposing one (or several) relations f_i between a set of variables $z_j, j = 1,..,n$ i.e.

$$f_i(z_1,..,z_p) = 0, \qquad 1 < p \leq n \tag{9}$$

f_i can represent any kind of relation (dynamic, static, linear, or non-linear). (These relations are also called constraints as the value of an involved variable can not change independently of the other involved variables ((Cassar *et al.*, 1994), (Declerck and Staroswiecki, 1991)). The system's structural model is represented by the set of relations $\mathscr{F} = \{f_1, f_2, \cdots, f_m\}$ and the set of variables $\mathscr{Z} = \mathscr{K} \cup \mathscr{X} = \{z_1, z_2, \cdots, z_n\}$. $\mathscr{X}$ is the set of unknown variables and $\mathscr{K} = \mathscr{U} \cup \mathscr{Y}$ is the set of known variables i.e. input/reference signals ($\mathscr{U}$), and measured signals ($\mathscr{Y}$).

3.1 Structural model representation

The system's structural model can be represented by a bipartite graph, $G(\mathscr{F}, \mathscr{Z}, \mathscr{A})$ where elements in the set of arcs $\mathscr{A} \subset \mathscr{F} \times \mathscr{Z}$ are defined in a certain way. To specify the elements in the set $\mathscr{A}$ in a useful manner, an additional property that is the calculability property, needs to be taken into considerations.

Calculability property: Let $z_j, j = 1, \cdots, p, \cdots, n$ be variables that are related through a constraint f_i, e.g. $f_i(z_1, \cdots, z_n) = 0$. The variable z_p is calculable if its value can determined through the constraint f_i under the condition that the values of the other variables $z_j, j = 1, \cdots, n, j \neq p$ are known. Taking calculability properties into considerations, the systems structural model is now represented by a bipartite directed graph(Izadi-Zamanabadi and Staroswiecki, 2000): The structure graph of the system is a bipartite directed graph $(\mathscr{F}, \mathscr{Z}, \mathscr{A})$ where the elements in the set of $\mathscr{A} \subset (\mathscr{F} \times \mathscr{Z})$, with $\mathscr{Z} = \mathscr{K} \bigcup \mathscr{X}$, are defined by:

$$\begin{cases} a_{ij} = (f_i, x_j) = 1 & \text{iff } f_i \text{ applies to } x_j, \\ a^*_{ij} = (x_i, f_j) = 1 & \text{iff } x_i \text{ is calculable through } f_j \\ kf_i = (k_i, f_j) = 1 & \text{iff } f_j \text{ applies on known var. } k_i. \\ 0 & \text{Otherwise.} \end{cases}$$

for all $x \in \mathscr{X}$ and $k \in \mathscr{K}$.

3.2 Matching

The main purpose of developing a matching algorithm is to identify the subsystem(s), which contain redundant information. The idea is depicted in figure 3. The algorithm initiates the matching from the known variables. The figure illustrates the idea of making the unknown variable "known" by successively matching them to previously known variables. First, variables x_1 and x_2 are matched to constraints f_1 and f_2 (full line). These variables become "known" as all the other variables that enter f_1 and f_2 are known. Hence, the new set of known variables can be considered as $\mathscr{K}_{new} = \mathscr{K} \cup x_1 \cup x_2$.

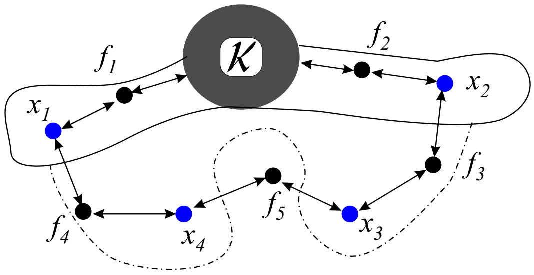

Figure 3: The process of matching.

Next, x_3 and x_4 are matched to f_3 and f_4 correspondingly (dotted line) etc. The procedure is repeated until a stop criteria is met.

4 Fault Detection Approach

A structural model of the refrigeration system is shown in Figure 4. The performed matching on the system is also illustrated by encircling each pair of relation and unknown variable, i.e. $(f, x), f \in \mathscr{F}, x \in \mathscr{X}$. All variables on the left side of the figure are known variables. The system analysis, which results in obtaining a residual expression for fault diagnosis, is described below:

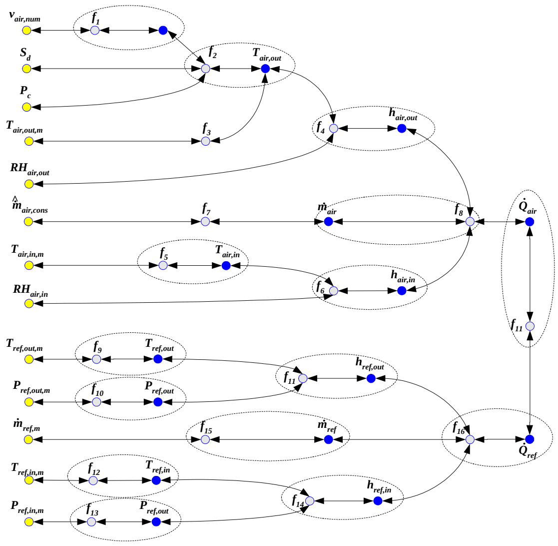

Figure 4: The performed matching on the systems structural model.

Through relation $f_{11}(\dot{Q}_{air}, \dot{Q}_{ref}) = 0$, which represents the fact that $\dot{Q}_{air} = \dot{Q}_{ref}$ under ideal operational conditions, we can calculate the value of $\dot{Q}_{air}$ when $\dot{Q}_{ref}$ is known. $\dot{Q}_{ref}$ is indeed calculable, and its value can be computed via relation f_{16}, which in turn corresponds to Eq. 1. The value of the air mass flow rate, $\dot{m}_{air}$, is calculable through relation f_8, since all other involved variables $(\dot{Q}_{air}, h_{air,out}, h_{air,in})$ can be computed through their corresponding relations. The inherent insight in the system functionality shows the fact that the nominal mass flow of air can be assumed constant for a given type and manufacture of condenser. The nominal value is denoted by $\hat{\dot{m}}_{air}$. This system knowledge is represented by relation f_7. The above-mentioned discussion can be summarized as:

$$\begin{aligned}\hat{\dot{m}}_{air} &= \dot{m}_{air} \\ &= \frac{\dot{Q}_{air}}{h_{air,out} - h_{air,in}} \\ &= \frac{\dot{Q}_{ref}}{h_{air,out} - h_{air,in}} \\ &= \frac{\dot{m}_{ref}(h_{ref,out} - h_{ref,in})}{h_{air,out} - h_{air,in}}\end{aligned}$$

When $\hat{\dot{m}}_{air}$ is known, following residual expression can be defined and used for fault diagnosis purposes.

$$r = \hat{\dot{m}}_{air} - \frac{\dot{m}_{ref}(h_{ref,out} - h_{ref,in})}{h_{air,out} - h_{air,in}} \tag{10}$$

An estimate for the nominal airflow can be obtained by performing a tuning during fault-free operation and then be used in the residual expression. The proposed detection approach consists of a non-linear parity equation, which basically compares the energy flow of the refrigerant in the condenser with the energy flow of the air-side.

A fault resulting in an airflow reduction produces a bias in the fault residual, when the actual airflow deviates from the nominal airflow. Another effect contributing to the bias is that the change of specific enthalpy is increased when the airflow is reduced. Hence, both the first and second term of the air-side expression of Equation 10 contributes to the residual bias when the airflow is reduced. The fault residual does not include the thermal capacities of the condenser. Also the mass-flow of the refrigerant does not include actual start/stop of the compressors but uses the capacity at the sampling time. These simplifications by using steady state models contribute with a high frequency noise component to the residual signal during normal operation. The high frequency noise components can be filtered out of the residual, but the low frequency noise components can present a challenge for the residual evaluation and limit the time-to-detect requirement for fault detection.

5 Experimental Results

The experimental results presented are obtained on medium size supermarket in Sønderborg, Denmark under normal operation. The refrigeration control system is equipped with more temperature sensors than it normally would be, but the presented results are based only on the standard measurements (ambient air temperature, refrigerant inlet and outlet pressures, temperature of the high pressure refrigerant entering the condenser and the actual compressor capacity). A significant amount of evaporator-dirt was observed when the inspected when initiating the experiment. This caused the evaporator fans to operate with a high duty-cycle, but no warnings was issued by the controller and the temperatures were within the specified reference band. After collecting data for a week, the condenser was cleaned. To avoid having to apply artificial dirt, the tuning was performed subsequent to the condenser cleaning. The detection capability was then evaluated on the previous data, when the condenser was dirty. To evaluate the noise on the residual in the no-fault mode, the detection algorithm was run on the following two weeks data. Figure 5 shows the residual after the condenser was cleaned. Figure 6 shows

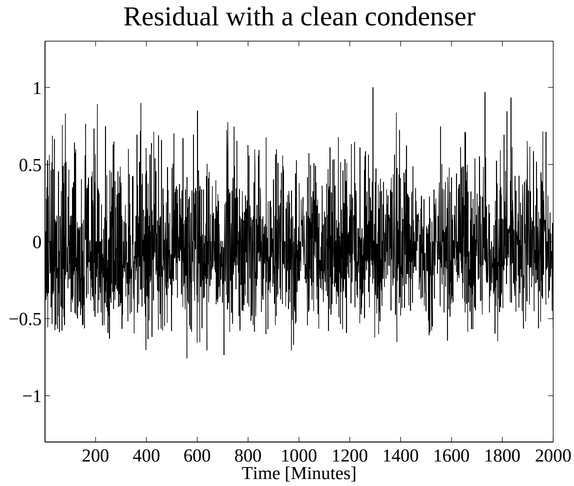

Figure 5: Residual with a clean condenser. The signal has about zero mean value but a significant noise component.

the unfiltered residual, as calculated with Equation 10, beginning while the condenser was dirty. The problem of finding a

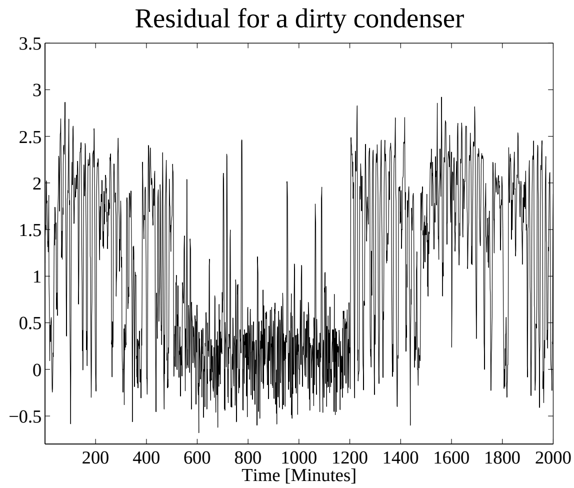

Figure 6: Residual with a dirty condenser. A positive mean value is seen during the experiment.

reasonable threshold is balanced between avoiding false detections and provide sensibility to the dirt buildup. Figure 7 shows the CUSUM (Basseville and Nikiforov, 1993) filtered residual for the non-faulty time after the condenser was cleaned. Notice that the maximum value is normalized to about 1. This gives an indication the obtainable sensitivity. The filtered residual for

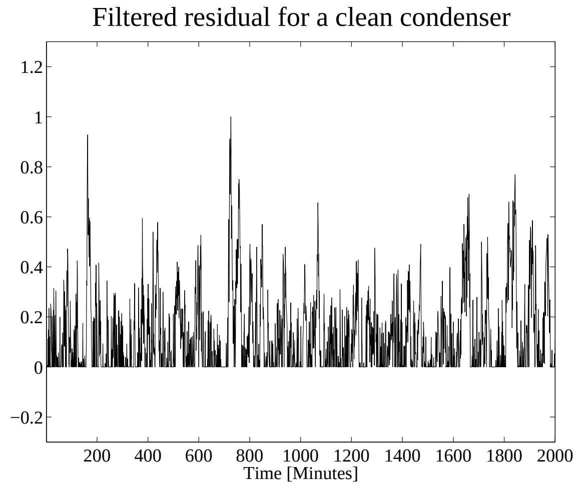

Figure 7: The CUSUM filtered residual of the clean condenser.

the dirty residual, displayed in Figure 8, shows the sensitivity to the fault. Compared with the non-faulty residual in Figure 7, a strong sensitivity is displayed. By applying the CUSUM filter, the residual evaluation problem is reduced to selecting the threshold that gives the optimal compromise between the detection delay and the probability of false detections.

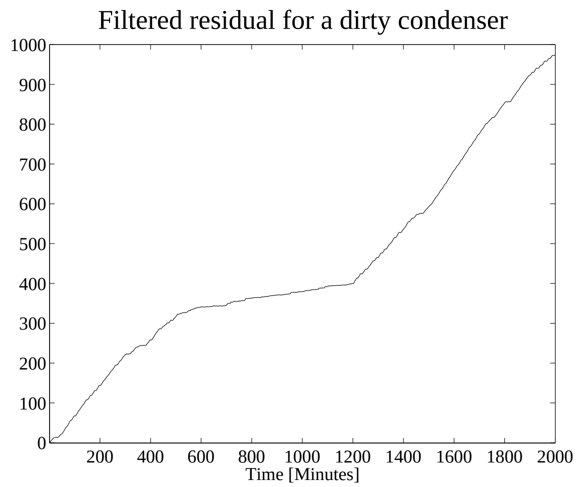

Figure 8: CUSUM filter applied to the residual in Figure 6.

6 CONCLUDING REMARKS

The objective of this paper was to present and evaluate a proposed fault detection scheme for detecting a class of faults that causes a reduction of airflow in a supermarket display cabinet. The proposed detection scheme uses a structural analysis approach to obtain a non-linear parity equation where the heat transfer rate on the air and refrigerant side of the condenser are involved. It was shown that, despite a high frequency noise component in the raw residual, remarkable detection results could be achieved by applying the statistical CUSUM algorithm.

7 ACKNOWLEDGMENTS

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