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# Development of an Energy Impact Model for RTU Economizer Faults

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## ABSTRACT

Stuck outdoor-air dampers can lead to significant energy waste when undetected for extended periods of time. This is especially true for rooftop air-conditioners (RTUs) where preventative maintenance may not be frequent or is only reserved for emergencies. Automated fault detection and diagnosis (AFDD) tools for outdoor-air dampers and economizers have been proposed to reduce the effort and cost of maintenance and are even required by some new building standards California Energy Commission (2012). While qualitatively, the effects of stuck outdoor-air damper faults are understood, less has been written about impacts these faults have on cooling cycle performance and actual energy usage over time.

An investigation of the effects of improper outdoor-air fraction caused by a stuck outdoor-air damper with respect to cooling capacity, cycle efficiency, equipment run-time, and energy usage is presented. Fault impact ratios for equipment performance are derived and demonstrated with laboratory data. In addition, a methodology for fault impact evaluation is presented and example results are shown for several outdoor-air damper positions and ambient conditions. The resulting fault impact ratios and evaluation method could be embedded into an existing AFDD tool in order to aid in optimal maintenance scheduling.

## 1. INTRODUCTION

Automated fault detection and diagnosis (AFDD) tools have been applied to air-conditioning equipment in order to identify and isolate problems that cause the equipment to perform less efficiently or fail to maintain comfortable conditions for occupants Katipamula and Brambley (2005a,b); Hjortland (2014). Outdoor-air economizers (OAE) of rooftop air-conditioners are a good application for AFDD. Since inspection and maintenance of RTUs tends to be infrequent, as much as 30% additional energy is estimated to be wasted annually due to performance degradations Brambley et al. (1998). The occurrence of RTU OAE faults has been shown to be frequent by field equipment surveys Jacobs (2003); Cowan (2004). Because of this, the need for effective AFDD tools is understood. Evidence of this can also be found in the OAE AFDD requirement of 2013 California Title 24 energy efficiency standard California Energy Commission (2012).

While the importance of early detection and diagnosis of RTU cycle and ventilation faults is understood, there has been less study on fault impact evaluation. After fault detection and diagnosis is performed the subsequent step in the general FDD methodology is to determine a recommended action to deal with the fault. For this to be successful, past and potential future impact should be estimated. This is important since the benefit to fixing some faults may be less than the cost to perform the necessary service.

Past work on fault impact evaluation has been primarily focused on refrigerant-side equipment faults. Rossi and Braun developed an optimal maintenance scheduling methodology for evaporator and condenser fouling service Rossi and Braun (1996) using capacity and efficiency degradation models. Li and Braun developed an *Economic Performance Degradation Index*, EPDI, that characterized the combined effects of several refrigerant-side faults on efficiency, capacity, and sensible heat ratio (SHR) Li and Braun (2007b). The economic impact of ventilation faults has been less studied. This paper aims to develop a methodology that can be applied within an AFDD tool to quantify the combined performance impact of a stuck outdoor-air damper fault on cooling capacity, efficiency, run-time, and energy

**Table 1: Test conditions used to characterize stuck damper fault impacts on RTU performance.**

| | $T_{oa}$ [°C] | $\varphi_{oa}$ [%] | $T_{ra}$ [°C] | $\varphi_{ra}$ [%] | $\gamma_{oad}$ [%] |
|---|---|---|---|---|---|
| Condition A | 31.50 | 0.40 | 26.00 | 0.50 | 10, 30, 50, 70 |
| Condition B | 31.50 | 0.50 | 26.00 | 0.50 | 20, 40, 60, 80 |
| Condition C | 37.78 | 0.50 | 26.00 | 0.50 | 0, 33, 50, 67, 100 |

usage.

In order to validate the methodology, a 14.07 kW (4-ton) RTU with integrated OAE was installed and tested within a pair of psychrometric chambers. The chambers were controlled to simulate a range of outdoor-air temperature and humidity conditions with a common indoor condition. The outdoor-air damper actuator control was overridden in order to simulate different stuck positions. A suite of sensors was used to measure system performance including compressor power and cooling capacity. The test conditions are summarized in Table 1.

The remainder of this work describes and develops an energy impact ratio for stuck outdoor-air faults. The impacts on the RTU refrigeration cycle is described in terms of cooling capacity and efficiency. Following this, the ventilation impact is considered. The energy impact ratio is then explored along with the run-time impact. Finally, a RTU fault evaluation methodology is described and example results are provided for several combinations of outdoor-air conditions and damper positions.

## 2. STUCK DAMPER FAULT CYCLE IMPACT

When an outdoor-air damper becomes stuck open, a larger fraction of outdoor-air, OAF, is allowed to enter the RTU mixing box. Because of this, the mixed-air temperature, $T_{ma}$, is impacted. This impact can be evaluated using Equation (1),

$$T_{ma} = \mathrm{OAF}\,(T_{oa} - T_{ra}) + T_{ra} \tag{1}$$

where $T_{ra}$ is the return-air temperature, $T_{oa}$ is the outdoor-air temperature, and

$$\mathrm{OAF} = \frac{h_{ma} - h_{ra}}{h_{oa} - h_{ra}} = \frac{T_{ma} - T_{ra}}{T_{oa} - T_{ra}} = \frac{\omega_{ma} - \omega_{ra}}{\omega_{oa} - \omega_{ra}}. \tag{2}$$

When $T_{ma}$ is increased due to a greater fraction of warmer outdoor-air, the evaporator saturation temperature increases. This impact is shown for experimental data in Figure 1 for two different outdoor-air conditions. Figure 1 also shows the stuck damper has a minimal impact on the condensing temperature since the impact on the compressor discharge pressure is relatively small. Ultimately, the required compression work is decreased slightly since the pressure ratio is decreased.

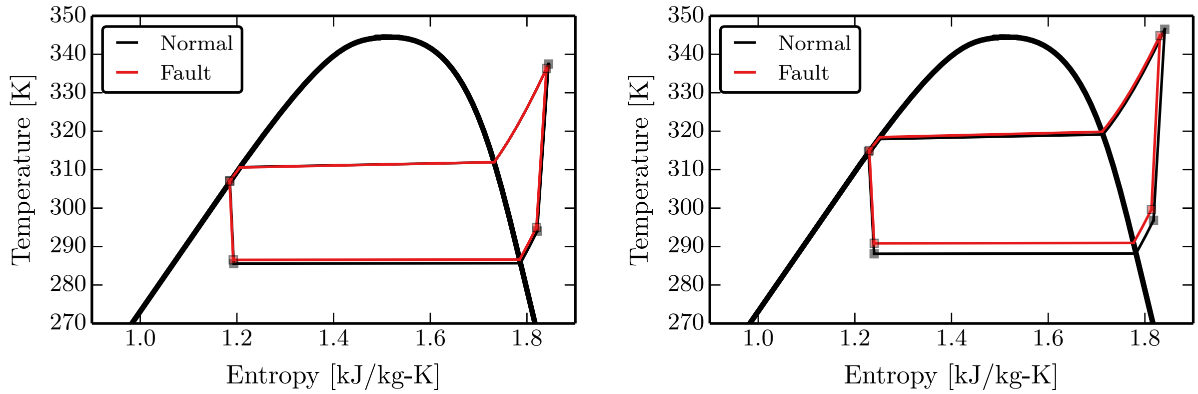
The RTU cooling capacity is increased when $T_{ma}$ is increased. Therefore, when the outdoor-air condition is warmer and/or more humid than the return-air, the capacity of the RTU increases for greater OAF. In order to characterize this impact, a capacity impact ratio, $r_{cool}$, is defined

$$r_{cool} = \frac{\dot{Q}_{cool,actual}}{\dot{Q}_{cool,normal}}. \tag{3}$$

where $\dot{Q}_{cool,actual}$ is the actual cooling capacity and $\dot{Q}_{cool,normal}$ is the normal cooling capacity. In Equation (3), $\dot{Q}_{cool,actual}$ is evaluated using observed measurements, while $\dot{Q}_{cool,normal}$ must be estimated using a model. Using the laboratory data, the capacity was calculated using measurements of refrigerant mass flow rate, $\dot{m}_{ref}$, and the entering and exiting evaporator refrigerant enthalpy, $h_{eri}$ and $h_{ero}$ respectively,

$$\dot{Q}_{cool,actual} = \dot{m}_{ref}\,(h_{ero} - h_{eri})\,. \tag{4}$$

The normal cooling capacity, $\dot{Q}_{cool,normal}$, was determined for each data point using the ASHRAE Toolkit model by adjusting the mixed-air temperature so that the expected OAF was achieved Brandemuehl and Gabel (1994). In the




**(a)** $T_{\mathbf{oa}} = 31.50\ ^\circ$**C**, $\varphi_{\mathbf{oa}} = 40$**%** **(b)** $T_{\mathbf{oa}} = 37.78\ ^\circ$**C**, $\varphi_{\mathbf{oa}} = 50$**%**

**Figure 1: Effect of a stuck open outdoor-air damper on RTU evaporator saturation temperature two different outdoor-air conditions with a constant return-air condition (**$T_{\mathbf{ra}} = 26.00\ ^\circ$**C**, $\varphi_{\mathbf{ra}} = 50$**%)**

model, the total cooling capacity of the system is determined by correcting the rated capacity, $\dot{Q}_{cool,rated}$ for different operating temperatures and flow rates,

$$\dot{Q}_{cool} = f_{\dot{V}} f_T \dot{Q}_{cool,rated} \tag{5}$$

where $f_{\dot{V}}$ is a volumetric flow rate correction factor,

$$f_{\dot{V}} = c_0 + c_1 \frac{\dot{V}}{\dot{V}_{rated}} \tag{6}$$

and $f_T$ is a temperature correction factor,

$$f_T = c_0 + c_1 B_{ei} + c_2 B_{ei}^2 + c_3 T_{ci} + c_4 T_{ci}^2 + c_5 B_{ei} T_{ci}. \tag{7}$$

The sensible capacity is determined using the bypass factor method. In order to ensure that the sensible capacity is not greater than the total capacity, the procedure iterates on the evaporator inlet wet bulb temperature until a dry coil is achieved (SHR = 1). The capacity impact for each test case is shown in Figure 2a. As expected, the capacity increases with increasing outdoor-air fractions (due to the greater evaporator-air inlet enthalpy) and with increasing outdoor-air enthalpy.

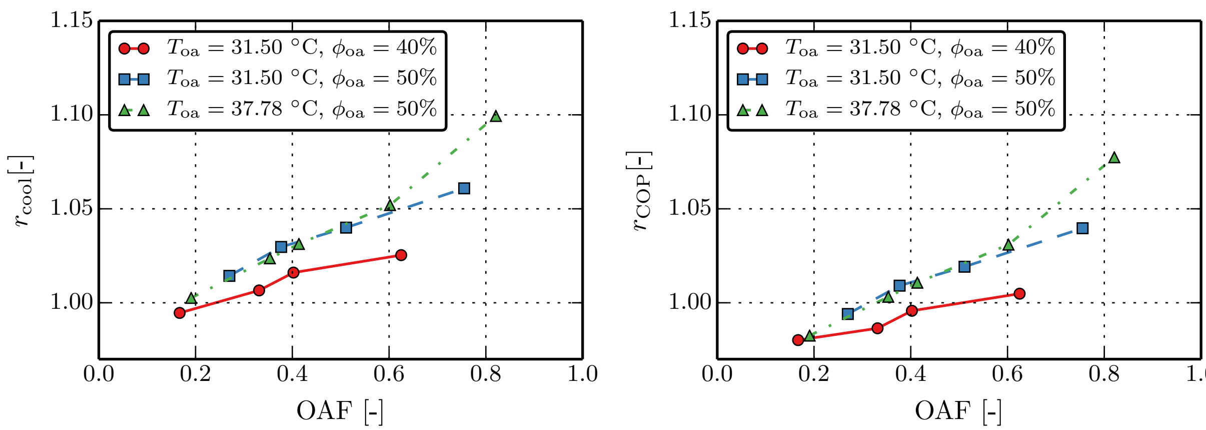
Using a similar convention, an efficiency degradation ratio, $r_{\mathrm{COP}}$, is defined as the ratio of actual cycle efficiency, $\mathrm{COP}_{\mathrm{actual}}$, to the expected cycle efficiency, $\mathrm{COP}_{\mathrm{normal}}$,

$$r_{\mathrm{COP}} = \frac{\mathrm{COP}_{\mathrm{actual}}}{\mathrm{COP}_{\mathrm{normal}}}. \tag{8}$$

The coefficient of performance, $\mathrm{COP} = \dot{Q}_{\mathrm{cool}}/\dot{W}_{\mathrm{comp}}$, was determined using the calculated cooling capacity and a compressor power measurement. The normal efficiency was determined by adjusting $T_{\mathrm{ma}}$ in order to achieved the expected OAF and evaluating the ASHRAE Toolkit Model. Similar trends in efficiency as the impact of cooling capacity are shown in Figure 2b.

## 3. STUCK DAMPER FAULT VENTILATION IMPACT

The outdoor-air damper and economizer have two primary purposes: provide the minimum fresh-air requirement to the conditioned space and reduce mechanical cooling when outdoor-air conditions are favorable. Many commercial building standards specify minimum fresh-air requirements to maintain indoor-air quality (IAQ) American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (2010, 2013). This is typically achieved by limiting the range of the outdoor-air damper actuator that allows at least the required fresh-air to enter the RTU. The




**(a) Capacity Impact** **(b) Efficiency Impact**

**Figure 2: Effect of a stuck open outdoor-air damper on RTU cooling capacity and efficiency under different outdoor-air conditions and damper positions. The impact impact of the fault increases with increasing OAF.**

economizer controller controls the outdoor-air damper to a position that minimizes the energy usage of the RTU for different outdoor-air conditions. This is accomplished using a high-limit control strategy that monitors the outdoor-air temperature (or enthalpy). The damper position is then modulated to the proper position that minimizes mechanical cooling Taylor and Cheng (2010).

A typical fault that occurs in RTU economizers is a stuck outdoor-air damper. There are multiple causes of this fault (burnt out actuator, broken linkages, damper physically blocked) but the effect of the fault is mostly the same. When the outdoor-air damper becomes stuck, it is unable to modulate and remains in its position until it is fixed. When the damper becomes stuck at its minimum position, mechanical cooling or heating is not effected significantly. However, opportunities to provide ``free-cooling" are diminished. Conversely, if the damper becomes stuck in an open position, cooling and heating are affected significantly due to the larger ventilation load. This impact can be visualized in Figure 3. The mixed-air conditions moves up along the mixing line due to the increased OAF caused by the fault. Warmer and more humid air then enters the evaporator, creating a larger load on the RTU. The outdoor-air condition may also change due to a return-air recirculation effect Hjortland (2014). This effect will be neglected, however may be significant in some applications.

The ventilation portion of the RTU cooling load, or the capacity required to condition the outdoor-air, can be calculated using Equation (10),

$$\dot{Q}_{\text{vent}} = \dot{m}_{\text{air}} \text{OAF} \left(h_{\text{oa}} - h_{\text{ra}}\right) \tag{9}$$

where $\dot{m}_{\text{air}}$ is the mass flow rate of the supply-air. Combining Equation (2) and Equation (10), $\dot{Q}_{\text{vent}}$ can be rewritten in terms of the $h_{\text{ma}} - h_{\text{ra}}$ difference

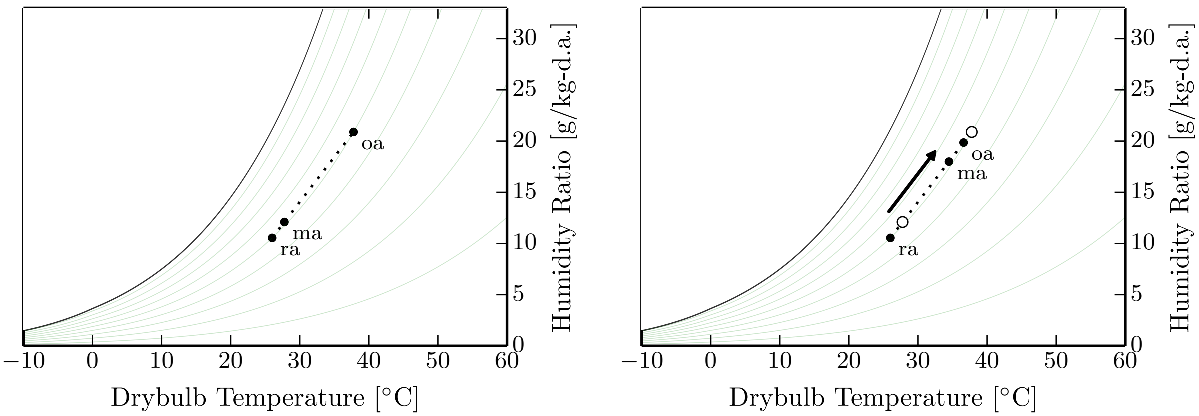
$$\dot{Q}_{\text{vent}} = \dot{m}_{\text{air}} \left(h_{\text{ma}} - h_{\text{ra}}\right). \tag{10}$$

The magnitude of the ventilation load can be related to the cooling capacity using the ventilation load fraction,

$$x_{\text{vent}} = \frac{\dot{Q}_{\text{vent}}}{\dot{Q}_{\text{cool}}}. \tag{11}$$

A stuck damper has a direct impact on the ventilation load since it controls the amount of fresh, outdoor-air that enters the RTU mixing box. When the damper is stuck open under warm and/or humid conditions, the ventilation load increases. In order to quantify this impact, the ventilation load impact ratio is defined,

$$r_{\text{vent}} = \frac{\dot{Q}_{\text{vent,actual}}}{\dot{Q}_{\text{vent,normal}}}. \tag{12}$$




**(a) Normal Ventilation** **(b) Faulty Ventilation**

**Figure 3: When conditions are warm and humid, the damper should be controlled to allow only the minimum outdoor-air requirement. When the damper is stuck open, the proportion of outdoor-air is increased causing the mixed-air enthalpy to increase.**

If it is assumed that the supply-air mass flow rate is not impacted by the stuck damper fault, Equation (12) can be written in terms of the mixed-air and return-air air enthalpy difference,

$$r_{\text{vent}} = \frac{\left(h_{\text{ma}} - h_{\text{ra}}\right)_{\text{actual}}}{\left(h_{\text{ma}} - h_{\text{ra}}\right)_{\text{normal}}}. \tag{13}$$

As was shown in Figure 3, the major impact of the stuck outdoor-air damper fault is the impact on the mixed-air air condition.

## 4. STUCK DAMPER FAULT ENERGY IMPACT

The impact of a stuck damper fault on capacity, efficiency, and ventilation load are important, but they do not provide a measure of the economic impact of the fault when taken by themselves. In order to estimate the economic penalty of the fault, the impact on energy consumption provides a more direct evaluation. Neglecting the indoor and outdoor fan power, the energy consumption required to meet a space load is equal to the product of the compressor power, $\dot{W}_{\text{comp}}$, and the run-time required to condition the space, $\Delta t_{\text{load}}$,

$$W_{\text{elec}} = \dot{W}_{\text{comp}} \Delta t_{\text{load}}. \tag{14}$$

The damper fault has no impact on the outdoor fan power consumption and an insignificant impact on indoor fan power. A power consumption measurement is typically expensive compared to the cost of an RTU so it is not typically installed. Instead of measuring power directly, Equation (14) can be rewritten in terms of the cooling capacity and coefficient of performance of the RTU,

$$W_{\text{elec}} = \frac{\dot{Q}_{\text{cool}}}{\text{COP}} \Delta t_{\text{load}} \tag{15}$$

where $\dot{Q}_{\text{cool}}$ and COP could be measured using virtual sensors Li and Braun (2007a); Kim and Braun (2012). In a similar manner as was done previously, the energy consumption impact ratio, $r_{\text{elec}}$, can be defined as,

$$\begin{aligned} r_{\text{elec}} &= \frac{W_{\text{elec,actual}}}{W_{\text{elec,normal}}} \\ &= \frac{\left(\dot{Q}_{\text{cool}}/\text{COP}\right)_{\text{actual}}}{\left(\dot{Q}_{\text{cool}}/\text{COP}\right)_{\text{normal}}} \\ &= \frac{r_{\text{cool}}}{r_{\text{COP}}} r_{\Delta\text{load}} \end{aligned} \tag{16}$$

where $r_{\Delta\text{load}}$ is the run-time impact ratio.

The run-time requirement is related to the total cooling load, $Q_{\text{load}}$, and the cooling capacity of the RTU,

$$\Delta t_{\text{load}} = \frac{Q_{\text{load}}}{\dot{Q}_{\text{cool}}}. \tag{17}$$

The total cooling load can be divided into an internal space load component, $Q_{\text{space}}$, and a ventilation load component,

$$\Delta t_{\text{load}} = \frac{Q_{\text{space}}}{\dot{Q}_{\text{cool}}} + \frac{Q_{\text{vent}}}{\dot{Q}_{\text{cool}}}. \tag{18}$$

Thus the total run-time requirement, $\Delta t_{\text{load}}$, can be expressed as the sum of the space load and ventilation load run-time requirements,

$$\Delta t_{\text{load}} = \Delta t_{\text{space}} + \Delta t_{\text{vent}} \tag{19}$$

where

$$\Delta t_{\text{space}} = \frac{Q_{\text{space}}}{\dot{Q}_{\text{cool}}} \tag{20}$$

and

$$\Delta t_{\text{vent}} = \frac{Q_{\text{vent}}}{\dot{Q}_{\text{cool}}}. \tag{21}$$

The impact of the stuck damper fault on run-time is of interest. In order quantify this, the space load run-time impact ratio, $r_{\Delta\text{space}}$ is defined,

$$r_{\Delta\text{space}} = \frac{\Delta t_{\text{space,actual}}}{\Delta t_{\text{space,normal}}} \tag{22}$$

where $\Delta t_{\text{space,actual}}$ is the run-time required when the damper is stuck open and $\Delta t_{\text{space,normal}}$ is the run-time requirement when the damper is working properly. Combining Equations (20) and (22) yields,

$$\begin{aligned} r_{\Delta\text{space}} &= \frac{\left(Q_{\text{space}}/\dot{Q}_{\text{cool}}\right)_{\text{actual}}}{\left(Q_{\text{space}}/\dot{Q}_{\text{cool}}\right)_{\text{normal}}} \\ &= \frac{r_{\text{space}}}{r_{\text{cool}}} \end{aligned} \tag{23}$$

where $r_{\text{space}} = Q_{\text{space,actual}}/Q_{\text{space,normal}}$ is the impact of the fault on the internal space load. The internal load is not affected by the stuck damper (or any other equipment fault), thus $r_{\text{space}} = 1$. Furthermore, the impact of the fault on the time required to cool the internal space load is only a function of the impact of the fault on cooling capacity.

By a similar method, the impact of the stuck damper on the time required to cool the ventilation load can be derived. The ventilation load run-time impact is defined by,

$$r_{\Delta\text{vent}} = \frac{\Delta t_{\text{vent,actual}}}{\Delta t_{\text{vent,normal}}}. \tag{24}$$

Combining Equations (21) and (24), $r_{\Delta\text{vent}}$ can be rewritten in terms of the ventilation load impact ratio, $r_{\text{vent}}$, and the cooling capacity impact ratio, $r_{\text{cool}}$,

$$\begin{aligned} r_{\Delta\text{vent}} &= \frac{\left(Q_{\text{vent}}/\dot{Q}_{\text{cool}}\right)_{\text{actual}}}{\left(Q_{\text{vent}}/\dot{Q}_{\text{cool}}\right)_{\text{normal}}} \\ &= \frac{r_{\text{vent}}}{r_{\text{cool}}}. \end{aligned} \tag{25}$$

The impact of the stuck damper fault on the time required to cool the ventilation load is directly proportional to the impact on ventilation load, as expected.

From Equation (19), the actual run-time required to cool the total load is the sum of the two run-time components,

$$\Delta t_{\text{load,actual}} = \Delta t_{\text{space,actual}} + \Delta t_{\text{vent,actual}}. \tag{26}$$

The actual total run-time requirement, $\Delta t_{\text{load,actual}}$, can be rewritten in terms of the run-time impact ratios by substituting Equations (22) and (24),

$$\Delta t_{\text{load,actual}} = r_{\Delta\text{space}} \Delta t_{\text{space,normal}} + r_{\Delta\text{vent}} \Delta t_{\text{vent,normal}}. \tag{27}$$

The total load can be written as the sum of the internal space load and ventilation load components. Thus, substituting Equation (11) yields an expression in terms of the impact ratios and the normal and actual run-times.

$$\Delta t_{\text{load,actual}} = \left[ \left(1 - x_{\text{vent,normal}}\right) r_{\Delta\text{space}} + x_{\text{vent,normal}} r_{\Delta\text{vent}} \right] \Delta t_{\text{load,normal}}. \tag{28}$$

Rearranging Equation (28) yields an expression for the total run-time impact ratio that relates the actual total run-time requirement to the total run-time requirement in the absence of the fault (assuming $r_{\text{space}} = 1$),

$$\begin{aligned} \frac{\Delta t_{\text{load,actual}}}{\Delta t_{\text{load,normal}}} &= \left[ \left(1 - x_{\text{vent,normal}}\right) r_{\Delta\text{space}} + x_{\text{vent,normal}} r_{\Delta\text{vent}} \right] \\ r_{\Delta\text{load}} &= \left[ \left(1 - x_{\text{vent,normal}}\right) \frac{r_{\text{space}}}{r_{\text{cool}}} + x_{\text{vent,normal}} \frac{r_{\text{vent}}}{r_{\text{cool}}} \right] \\ &= \frac{1}{r_{\text{cool}}} \left[ \left(1 - x_{\text{vent,normal}}\right) + x_{\text{vent,normal}} r_{\text{vent}} \right]. \end{aligned} \tag{29}$$

Equation (29) shows that the equipment runs longer when a stuck damper fault is present since the ventilation load impact, $r_{\text{vent}}$, increases. The increase in ventilation is scaled by the fraction of the capacity that is used normally for the ventilation load, $x_{\text{vent,normal}}$. Thus, if the normal ventilation load is small compared to the capacity of the unit, the fault must increase the ventilation load significantly before the fault has an impact on run-time. Run-time impact is inversely proportional to the cooling capacity impact. Since the system capacity increases when $h_{\text{oa}} > h_{\text{ra}}$, $r_{\Delta\text{load}}$ is not 1 : 1 with $r_{\text{vent}}$.

Combining Equations (16) and (29), $r_{\text{elec}}$ can be rewritten in terms of only the efficiency impact ratio, $r_{\text{COP}}$, normal ventilation load fraction, $x_{\text{vent,normal}}$, and the ventilation load impact ratio, $r_{\text{vent}}$,

$$r_{\text{elec}} = \frac{1}{r_{\text{COP}}} \left[ \left(1 - x_{\text{vent,normal}}\right) + x_{\text{vent,normal}} r_{\text{vent}} \right]. \tag{30}$$

The actual energy impact caused by the fault, can be calculated using $r_{\text{elec}}$,

$$\begin{aligned} W_{\text{elec,fault}} &= \frac{W_{\text{elec,actual}}}{r_{\text{elec}} - 1} \\ &= \frac{\dot{W}_{\text{comp,actual}} \Delta t_{\text{load,actual}}}{r_{\text{elec}} - 1} \end{aligned} \tag{31}$$

where $\dot{W}_{\text{comp,actual}}$ is a measurement of the actual power consumption (measured directly or virtually) over a given run-time. Equation (31) is only valid under a combination of outdoor-air and return-air conditions. This means it should be reevaluated when ambient conditions change significantly. Over an extended period of time, $W_{\text{elec,fault}}$ can be summed for an cumulative energy impact estimation.

## 5. FAULT EVALUATION AND RESULTS

In the previous sections, several fault impact ratios were defined that have the following general form,

$$\text{Fault Impact Ratio} = \frac{\text{Actual/Observed Performance}}{\text{Normal/Expected Performance}}.$$

In order to evaluate each fault impact ratio ($r_{\text{cool}}$, $r_{\text{COP}}$, $r_{\text{vent}}$, etc.) a measurement or estimation of the actual performance must be made with an estimation of the normal performance. When the system operates with a fault, the observed performance represents the faulty condition. In order to estimate the expected performance, an accurate normal performance model is needed.

The process of determining the normal performance using observed system measurements is demonstrated in Figure 4 for a stuck outdoor-air damper fault. There are three steps involved in this process,

1. determine normal mixed-air condition using an outdoor-air fraction model,
2. determine normal supply-air condition using total capacity and sensible heat ratio, SHR, models, and
3. adjust return-air condition to maintain latent space load.

In order to perform this procedure, the expected outdoor-air fraction and supply-air mass flow rate must be specified. Additionally, models for the total cooling capacity and sensible heat ratio must be known. Lastly, sufficient measurements to evaluate the humidity ratio and enthalpy of the return-air, outdoor-air, and supply-air conditions must be available, along with a measurement of mixed-air temperature.

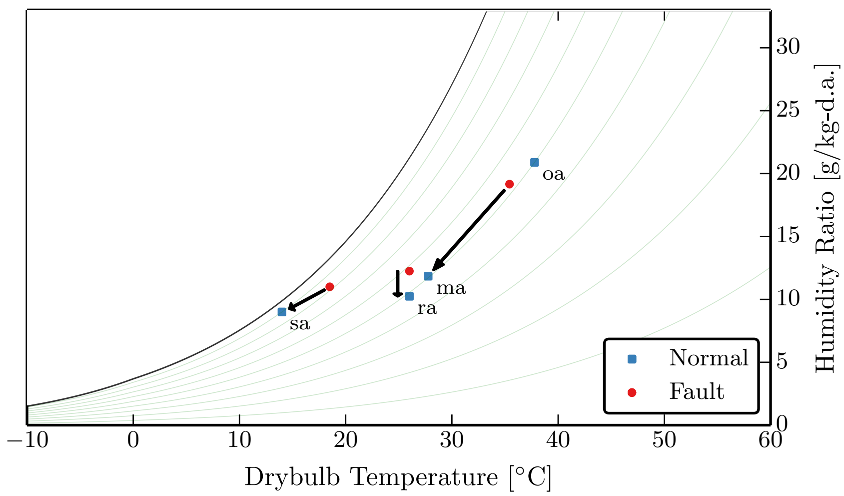



**Figure 4: Characterization of normal performance using observed, faulty performance.**

In order to determine the normal mixed-air condition, first $T_{ra}$ and $T_{oa}$ are measured. Next the expected outdoor-air fraction, $\mathrm{OAF}_{normal}$ is determined, using a model specified *a-priori*. For this paper, an OAF model based on the damper actuator control signal, trained and validated using laboratory data was used Hjortland (2014). Then, the normal mixed-air temperature is determined using Equation (1). The humidity ratio of the return-air and outdoor-air are determined using the respective temperature and relative humidity measurements using humid-air property relations. Equation (2) is then applied to determine $\omega_{ma}$ and fix the normal mixed-air state.

Once the normal mixed-air condition has been determined, the capacity model can be evaluated in order to determine the expected supply-air condition. The supply-air enthalpy can be determined using an energy balance on the evaporator coil,

$$h_{sa} = h_{ma} - \frac{\dot{Q}_{cool,normal}}{\dot{m}_{air}} \tag{32}$$

where $\dot{m}_{air}$ is determined using a virtual sensor Hjortland (2014). The expected supply-air temperature, $T_{sa}$, can be determined using the expected sensible cooling capacity of the RTU,

$$T_{sa} = T_{ma} - \frac{\mathrm{SHR}_{normal}\dot{Q}_{cool,normal}}{\dot{m}_{air}c_{p,air}} \tag{33}$$

where $c_{p,air}$ is the specific heat of air. In the current example, the ASHRAE Toolkit model was used to determine $\dot{Q}_{cool,normal}$ and $\mathrm{SHR}_{normal}$ Brandemuehl and Gabel (1994).

The last consideration is to adjust the return-air condition so that the internal space latent load remains unchanged. In a typical RTU application, the RTU is controlled by a thermostat monitoring the space temperature. Because of this, $T_{ra}$ should be unaffected by the stuck damper (provided the equipment has enough capacity to overcome the additional ventilation load). The humidity of the space is not typically controlled, so $\omega_{ra}$ floats to maintain the latent space load. Because the humidity in the space is able to float, $\omega_{ra}$ must be adjusted to maintain the latent load in the space.

The previous procedure was applied to the laboratory data collected for the 14.07 kW (4-ton) RTU. The energy impact ratio, $r_{elec}$, was evaluated for each case with the results plotted in Figure 5. The results show that damper faults with large outdoor-air fraction impacts have the greatest energy impact. This is due to the additional ventilation load that must be cooled by the RTU in order to maintain the space conditions. The results also illustrate that the impact of the stuck damper increases as the outdoor-air becomes warmer and more humid, as expected.

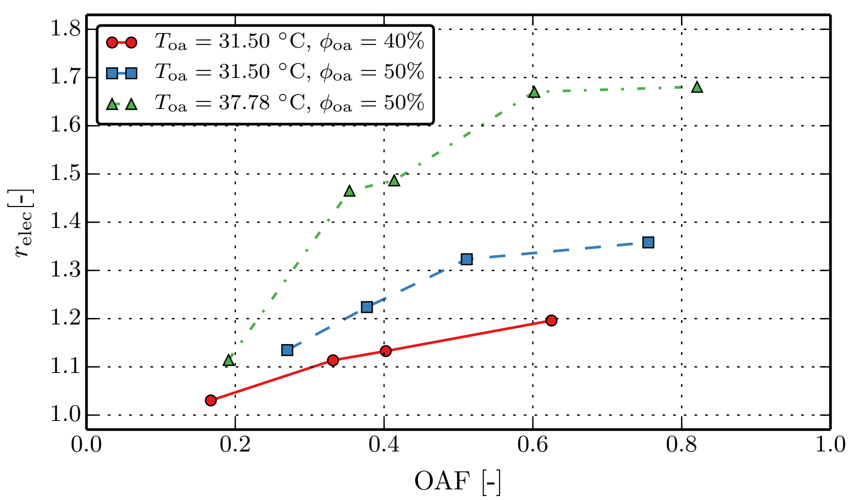



**Figure 5: Energy impact ratio for different outdoor-air damper faults under different outdoor-air conditions.**

## 6. CONCLUSIONS

Fault impact evaluation is an important component for automated fault detection and diagnosis tools in order to minimize operating costs and maintenance costs over the life of the equipment. A methodology to determine the energy impact of a stuck outdoor-air damper fault on the cooling and ventilation performance of a rooftop air-conditioner (RTU) has been proposed. The methodology combines the impacts of the fault on cooling capacity, cycle efficiency, ventilation load, and equipment run-time to yield a physical model for the relative increase in energy consumption of the faulty equipment over a normally operating system. The methodology can be extended to determine the economic penalty of the fault on operating costs and equipment costs using existing methodologies Li and Braun (2007b).

A fault evaluation methodology was also described where an estimation of normal system performance can be derived from measurements made from equipment that currently has a stuck outdoor-air damper fault. Using simple models to characterize the outdoor-air fraction and total and sensible cooling capacities, thermodynamic state points are estimated for a fault-free system. Using laboratory data collected from a 14.07 kW (4-ton) RTU, the evaluation methodology and energy impact model were demonstrated. The fault evaluation methodology does have a extensive sensor requirement. Further work to identify and eliminate potentially unneeded sensors would reduce the implementation costs and improve the fault evaluation method considerably.

Only stuck outdoor-air damper faults existing during warm and humid operating conditions were considered. While these faults are indeed significant, stuck damper faults during cold outdoor-air conditions are also important. Furthermore, only dampers stuck open were addressed in this work. Potential missed ``free-cooling" could also occur if the damper is stuck closed. Work to extend the fault impact model for all economizer modes of operation should be pursued.

Lastly, RTU AFDD tools need to be able to diagnose multiple simultaneous fault conditions. The fault evaluation methods must then also be able to estimate the energy and economic impact of simultaneous faults accurately. Continued work in this area is needed in order to achieve optimal maintenance scheduling.

## NOMENCLATURE

| **Symbols** | | **Subscripts** | |
|---|---|---|---|
| $B$ | wet bulb temperature [°C] | actual | actual, or observed performance |
| $c_{p,\mathrm{air}}$ | specific heat of air [kJ/kg-K] | *ci* | condenser inlet air |
| $h$ | enthalpy [kJ/kg-K] | cool | cooling capacity |
| OAF | outdoor-air fraction [-] | *ei* | evaporator inlet air |
| $Q$ | cooling load [kJ] | elec | electricity |
| $\dot{Q}$ | heat transfer per unit time [kW] | load | total load |
| $\dot{m}$ | mass flow rate [kg/s] | ma | mixed-air |
| $r$ | fault degradation/impact ratio [-] | normal | normal, or expected performance |
| $T$ | temperature [°C] | oa | outdoor-air |
| $W$ | energy consumption [kJ] | *oad* | outdoor-air damper |
| $\dot{W}$ | power consumption [kW] | ra | return-air |
| $x_{\mathrm{vent}}$ | ventilation load fraction [-] | sa | supply-air |
| $\gamma$ | normalized control signal [-] | space | internal space load |
| $\Delta t$ | run-time [s] | vent | ventilation |
| $\varphi$ | relative humidity [-] | | |
| $\omega$ | humidity ratio [kg/kg.d.a] | | |

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