WHEN YOU NEED TO BE SURE

CyFlex® Knowledge Article

Combustion Air Handler Status

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Temperature Control

Steady State - Heating

- Steady state control is very good as long as the supply differential pressure is well behaved – typically ± 0.2 F vs standard of ± 1.8 F

Transient - Heating

- Any significant change in air flow will cause a disruption in temperature control
  - The weighted relief damper does not maintain pressure at the air handler exit
  - Change in pressure at the air handler exit causes the pressure differential across the fan to change, which results in a change in air flow rate.
  - A change in air flow rate causes temperatures downstream of the heating and cooling coils to change. System has large thermal inertia and is incapable of rapid response.

Maximum Performance - Heating

- System was not capable of delivering 104 F target temperature with ambient 10 F.
  - Supply differential pressure becomes unstable when second pump is needed.
  - Design flow of process hot water nearly, but not quite achieved.
  - System would be capable at higher ambient or if hot water supply problems are corrected.

Cooling Performance

- Largely unknown.
  - Testing limited by ability to simulate hot ambient temperature and high humidity.

Humidity Control

Steady State – Steam Addition

- Humidity control stability in a given combustion air duct can be very good if the controls are tuned for that particular air flow rate. Standard is ± 0.5 C. Tuning across the range of flows is difficult due to pure delays between steam injection and humidity sensing.
    - Delay at idle is typically 18 seconds. Delay at rated is 4 seconds.
    - Steam pressure oscillates between boiler high and low setpoints.
- Humidity control between ducts is often very poor.
    - Mixing length is very limited between injection point and duct entrances.
    - Air and steam flow is affected by asymmetry in the placement of the weighted relief damper.
    - Any combination of conditions leading to need for increased mixing length will adversely affect the distribution of humidity – these include high steam injection rate, high air flow rate, low air temperature.

Transient - Steam Addition

- Any significant change in air flow will cause a disruption in humidity control.
    - The weighted relief damper does not maintain pressure at the air handler exit
    - Change in pressure at the air handler exit causes the pressure differential across the fan to change, which results in a change in air flow rate.
    - A change in air flow rate requires a change in steam injection rate. The system response is very good, but the feedback is delayed due to transport times, so the system has to be somewhat detuned.

Maximum Performance - Steam Addition

- System should be capable of 95 F dewpoint. This could not be demonstrated. The steam system pressure crashed when the valves were opened wide.
- Performance just prior to system pressure loss indicates that the system should be capable if the steam pressure can be maintained. In fact, the steam valves appear to be oversized. The full range of the smaller 1/3 valve is seldom needed in normal operation.

Cooling Performance

- Largely unknown.
    - Testing limited by ability to simulate hot ambient temperature and high humidity.

Summary

- Weighted relief damper performance is a significant issue for both temperature and humidity control.
- Lack of steam mixing length and asymmetric flow are significant problems that affect our ability to control humidity distribution between combustion air ducts.
- Testing has been somewhat limited by poor control of the process hot water delivery system.

$$q = UA\Delta T_{lm} \qquad \text{fb.1}$$

afad

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{ln(\Delta n_2/\Delta\Delta_1)} = \frac{\Delta T_1 - \Delta T_2}{ln(\Delta n_1/\Delta\Delta_2)} \qquad \text{fb.2}$$

For a counterflow heat exchanger

$$\begin{aligned} \Delta T_1 &\equiv T_{h,i} - T_{c,o} \\ \Delta T_2 &\equiv T_{h,o} - T_{c,i} \end{aligned} \qquad \text{fb.3}$$

fadsf

$$q = \dot{m}_h c_{p,h}\left(T_{h,i} - T_{h,o}\right) = C_h\left(T_{h,i} - T_{h,o}\right) \qquad \text{fb.4}$$

fdafd

$$q = \dot{m}_c c_{p,c}\left(T_{c,o} - T_{c,i}\right) = C_c\left(T_{c,o} - T_{c,i}\right) \qquad \text{fb.5}$$

fadfa

$$\varepsilon = \frac{q}{q_{\max}} = \frac{C_c\left(T_{c,o} - T_{c,i}\right)}{C_{\min}\left(T_{h,i} - T_{c,i}\right)} = \frac{C_h\left(T_{h,i} - T_{h,o}\right)}{C_{\min}\left(T_{h,i} - T_{c,i}\right)} \qquad \text{fb.6}$$

afadf

$$if\ C_c < C_h\ then\ \varepsilon = \frac{\left(T_{c,o} - T_{c,i}\right)}{\left(T_{h,i} - T_{c,i}\right)}\ else\ \varepsilon = \frac{\left(T_{h,i} - T_{h,o}\right)}{\left(T_{h,i} - T_{c,i}\right)} \qquad \text{fb.7}$$

fsadfadf

$$NTU \equiv \frac{UA}{C_{\min}} \qquad \text{fb.8}$$

dASD

$$UA\,\Delta T_{lm} = C_{min}\,\Delta T_{max} \Rightarrow NTU = \frac{UA}{C_{min}} = \frac{\Delta T_{max}}{\Delta T_{lm}} \qquad \text{fb.9}$$

fadfd

$$q = C_{min}\,\Delta T_{max} = C_{max}\,\Delta T_{min} \Rightarrow C_r \equiv \frac{C_{min}}{C_{max}} = \frac{\Delta T_{min}}{\Delta T_{max}} \qquad \text{fb.10}$$

F193_3_8_06_FS_Steel_Bal_Data_DP.pdf

$$D.P. = \left(\frac{GPM \times 17.3}{FF}\right)^2 \qquad \text{fb.11}$$

250H, FF=311.4, 400H, FF=709

```
cah_ccl_vf     gal/min     4     -     MED              OFF
1.00
{ 709 / 17.3 * @sqrt( cah_phc_w_dP[in_h2o] ) }

cah_ccl_cv     none        4     -     MED              OFF
1.00
{ cah_ccl_vf[gal/min] * @sqrt( 1 / cah_ccl_sup_dP[psi] ) }

cah_ccl_mf     lb/min      4     -     MED              OFF
1.00
cah_ccl_vf * 1[g/cm3]

cah_ccl_qo     btu/hr      4     -     MED              OFF
1.00
cah_ccl_mf * 4.1813[j/(g_deg_k)] * ( cah_ccl_w_otT - cah_ccl_supT )

cah_cclL_vf    gal/min     4     -     MED              OFF
1.00
{ 709 / 17.3 * @sqrt( cah_ccl_w_dP[in_h2o] ) }

cah_cclL_mf    lb/min      4     -     MED              OFF
1.00
cah_cclL_vf * 1[g/cm3]

cah_cclU_vf    gal/min     4     -     MED              OFF
1.00
{ 709 / 17.3 * @sqrt( cah_rhc_w_dP[in_h2o] ) }

cah_cclU_mf    lb/min      4     -     MED              OFF
1.00
cah_cclU_vf * 1[g/cm3]
```

```
cah_ccl_qc     btu/hr      4     -     MED            OFF
1.00
( cah_cclL_mf + cah_cclL_mf ) * 4.1813[j/(g_deg_k)] *
\
( cah_ccl_w_otT - cah_ccl_w_inT )
```

Taco Accu-Flo 2 and 2.5 inch circuit setter flow balancing valves.

https://www.tacocomfort.com/documents/FileLibrary/402-051.pdf
https://www.tacocomfort.com/documents/FileLibrary/401-048.pdf
https://www.tacocomfort.com/documents/FileLibrary/401-047.pdf
https://www.tacocomfort.com/documents/FileLibrary/400-2.4.pdf

$$D.P. = \left(\frac{GPM}{FF}\right)^2 \qquad \text{fb.12}$$

For a 2-inch circuit setter flow balancing valve, FF = 9

For a 2.5-inch circuit setter flow balancing valve, FF = 15

$$\begin{aligned} &cah\_rhc\_otT \rightarrow cah\_ccl\_otT \\ &ch\_ccl\_w\_otT \leftarrow cah\_ccl\_w\_inT \end{aligned} \qquad \text{fb.13}$$