WHEN YOU NEED TO BE SURE

CyFlex® Knowledge Article

Combustion Air Handler – Water Side Flow Balancing

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Balance Valves

Each of the water side coil loops is equipped with three balance valves. There is a balance valve in the pipe leading to each individual coil designated on the marked-up version of drawing SEP-TCU-TI145-TC as BV-HH*CAU0?-1 and -2 where * indicates the associated test cell and ? is 1 for the preheat coil, 2 for the reheat coil and 3 for the cooling coil. -1 and -2 indicate the coil number. There is also a balance valve on the exit of each water loop downstream of the modulating valves designated as BV-HH*CAU0?-E.

Each balance valve should have an integral venturi with two pressure taps – one upstream and one in the venturi throat. The approximate flow through the balance valve can be calculated based on the differential pressure measured between the taps. Care must be taken to properly size the venturis so the differential pressure is in a measurable range for the flows of interest. The balance valves for the individual coils see a relatively high and nearly constant flow rate. To assure that the exit balance valve is to be useful across most of the flow range, manufacturers recommend differential pressures in the range from 25 to 100 inches water column.

The purpose of the balance valves for the individual coils is to make sure each coil has the same amount of water flowing through it. The coil found to have the highest flow should be restricted just enough to equal the flow through the other coil. There is no benefit to restricting the flow to match the coil design target since this would only load the pump and reduce the effectiveness of the coil. The coil flow can be monitored periodically to verify that the coils have not fouled on the water side.

The balance valves at the exit of each loop are designed to balance the hot and chilled water flows between all the air handlers. This process would have to be carried out with a consistent pressure differential between the supply and return. The central boiler/chiller and pump system should be designed to deliver at least the design flow and temperature. It is also possible to use the exit balance valves to measure the flow from the loop as a function of valve command and supply to return differential pressure to characterize the valve.

Taken together, the calculated flows through the coils and at the exit, along with the measured temperature differentials, can be used to do a heat transfer analysis for the water side of each loop. This balance could potentially be useful if feedforward control is ever required.

Flow Design Inc. FlowSet balance valves were used in the combustion air handler for HH9. Flow Design provides the following formula for differential pressure, $D.P.$, in inches of water column as a function of the flow rate, GPM, in gallons per minute and a flow factor for each size of venturi, FF.

$$D.P. = \left(\frac{GPM * 17.3}{FF} \right)^2 \qquad \text{bv.1}$$

Rearranging to solve for the flow rate gives

$$GPM = \frac{FF\sqrt{D.P.}}{17.3} \qquad \text{bv.2}$$

Table 1 on page 2 below shows some of the characteristics of the balance valves used on the combustion air handler for HH9.

Table 1

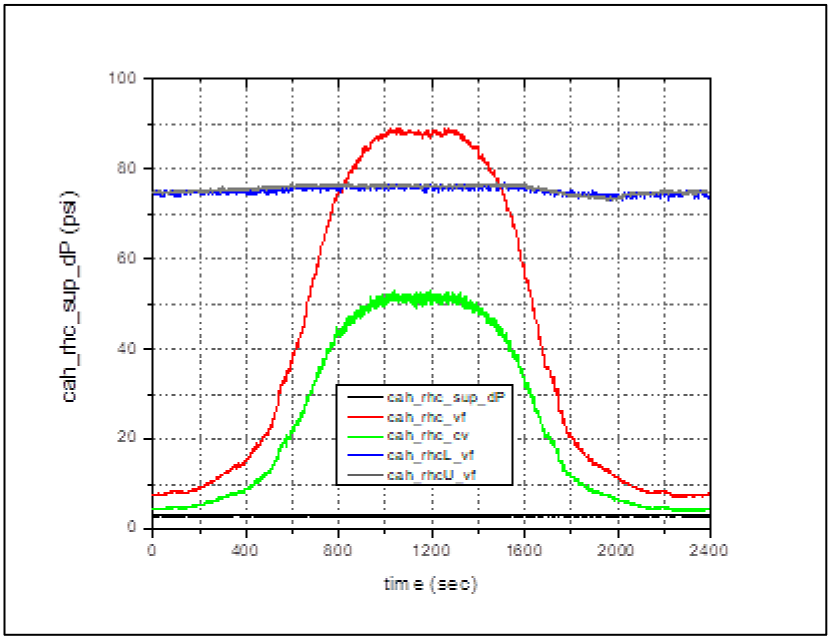
HH9	SEP-TCU-TI145-TC	Flow Design Inc	FF	Design Flow Rate	dP at Design
	Designation	Model	[magic]	[gpm]	[in_h2o]
Preheat Coils	BV-HH*CAU01-1,-2	AF250H	311.4	50	7.7
Preheat Exit	BV-HH*CAU01-E	AF250H	311.4	100	30.9
Cooling Coils	BV-HH*CAU03-1,-2	AF400H	709	130	10.1
Cooling Exit	BV-HH*CAU03-E	AF400H	709	260	40.2
Reheat Coils	BV-HH*CAU02-1,-2	AF250H	311.4	44	6.0
Reheat Exit	BV-HH*CAU02-E	AF250H	311.4	88	23.9

These balance valves were apparently not sized with the manufacturer's 25-inch water column minimum in mind or the option to use the exit balance valves for flow measurement.

A number of tests were run on each of the water loops to balance the flow through each coil and to check the coil and exit flows against design targets. The plot below shows the result of a test where the modulating valve at the exit of the reheat loop was slowly ramped from 0% open to 100% and then back to 0%.

The exit flow, cah_rhc_vf, exactly matched the 88 gpm design target, but the supply differential press, cah_rhc_sup_dP, was only 3 psi versus a design target of 15 psi. This gives us confidence that the piping does not impose an undue restriction but exposes the system to the possibility of commanding excess flow if the design differential pressure is provided. This is also an indication that the tuning of the controls will be affected by the supply differential pressure.

The individual coils see a flow of approximately 76 gpm each. This is significantly in excess of the 44 gpm design target. This is actually a desirable characteristic since higher flow rates enhance heat transfer and provide margin for coil fouling on the water side.



It has been suggested that if the modulating valve at the exit of the loop is wide open, that the exit flow will be equal to the sum of the flows to the coils. This is clearly not the case. Three separate balance valves are required to fully balance and characterize the system.

The measured exit flow rate can be used to back-calculate the *Cv* of the modulating valve based on the assumption that the differential pressure across the valve is the same as the supply differential pressure, *sup_dP* in inches of water column, using

$$Cv = \frac{\sqrt{sup\_dP}}{GPM} \qquad \text{bv.3}$$

For the reheat coil, we see good agreement between the calculated value of 51 at 100% command vs the manufacturers specified value of 53. In other tests, we were not as fortunate, so additional tests will be required before we will be able to predict maximum flow for a given pressure differential.

Similar tests were performed for the cooling and reheat coil loops. Plots and data for these tests are captured in water_flow.opj. Tabulated results are shown in *Table 2* on page 4

Table 2

HH9	Coil 1 Flow	Coil 2 Flow	Design Flow	Max Exit Flow	Meas Cv	Design Cv	Supply dP	Design dP
	[gpm]	[gpm]	[gpm]	[gpm]	[none]	[none]	[psi]	[psi]
Preheat Loop	71.6	72.2	100	92.1	48.4	75	3.62	15
Cooling Loop	162.1	157.4	260	330.9	59.9	180	30.5	15
Reheat Loop	76.2	75.9	88	88	51.2	53	2.95	15

For both the heating loops we were able to achieve or come close to the design exit flow with a supply differential pressure that was much lower than the design target. We were also able to hit our maximum temperature target with very cold ambient air, so we have confidence that the flow rate will be adequate as long as the central facility can provide the necessary flow and temperature. The cooling coil exit flow was significantly higher than the design target, but the supply differential pressure was also higher than design, so we will have to repeat the test once the differential pressure is brought under control.

All the measured coil flows were higher than the target of half the design exit flow. This is not a problem since it improves water side heat transfer and provides additional margin to combat fouling.

The air handlers for HH10 and HH6 have different balance valves for the coil flows than those used for HH9 and the valves at the exits lack venturis to measure and balance the flow between air handlers. *Table 3* lists characteristics of the installed coil balance valves and suggested balance valves to be installed at the exit for each loop.

Table 3

HH10,6	SEP-TCU-TI145-TC	Taco	Cv	C	Design Flow	dP at Design
	Designation	ACCU-FLO			[gpm]	[in_h2o]
Preheat Coils	BV-HH*CAU01-1,-2	2 inch	62.3	8.946	50	31.2
Preheat Exit	BV-HH*CAU01-E	2 inch	62.3	8.946	100	125.0
Cooling Coils	BV-HH*CAU03-1,-2	2.5 inch	122	15.021	130	74.9
Cooling Exit	BV-HH*CAU03-E	3 inch	212	28.284	260	84.5
Reheat Coils	BV-HH*CAU02-1,-2	2 inch	62.3	8.946	44	24.2
Reheat Exit	BV-HH*CAU02-E	2 inch	62.3	8.946	88	96.8

Taco provides a "sliderule" chart that shows the differential pressure between the taps as a function of water flow:

(https://www.tacocomfort.com/documents/FileLibrary/402-051.pdf)

When asked via their internet help, Taco provided the C values listed in Table 3. For a given pressure differential, *D.P.*, the flow in gallons per minute, *GPM*, is given by

$$GPM = C\sqrt{D.P.} \qquad \text{bv.4}$$

Rearranging to solved for the differential pressure gives

$$D.P. = \left(\frac{GPM}{C}\right)^2 \qquad \text{bv.5}$$

The Cv listed in *Table 3* on page 4 for each valve is for the fully open position. Since we don't have a way to measure the pressure differential across the valve, the supply to return pressure differential can be used as a reasonable approximation as in Equation bv.3.

The thought process behind the sizing for the exit balance valves was to specify a size that would give a reading on the high end of the flow range that was consistent with the expected reading for the existing valves. Since the pressure taps are approximately an inch apart and the valves are mounted vertically, we want the low flow reading to be large compared to the pressure differential due to the tap height difference. The sizes suggested in Table 3 should represent a reasonable compromise.

Pressure Transducers

Referring to P&ID drawing SEP_TCU TI145-TC, we have found that differential pressure transducers PDT-11-03-025-01 thru 03 are not particularly useful and PDT-11-03-025-07 and 14 are redundant since they both measure the supply to return pressure differential on the heating hot water circuit. We have experimented with three pressure transducers with long leads and connectors compatible with the fittings on the venturi pressure taps that allow us to measure the flow through all three balance valves on a given loop simultaneously. Once balancing has been accomplished on all loops, the transducers can be used on each of the exit venturis to measure flow from each loop for heat transfer monitoring purposes.

By considering the various scenarios of proposed balance valve sizing and observed flows, it appears that differential pressure transducers with a range of 5[psi] (138.5[in_h2o]) would suffice. The transducers should be accurate to 0.5% full scale and capable of withstanding 100 psi gage pressure. The analog inputs formerly used by PDT-11-03-025-01 thru 03 can be used for the new transducers.

We have also found that the current Veris differential pressure transducers, PDT-11-03-025-07, -12 and -14, used to measure the supply to return differential pressure do not function properly when the system pressure exceeds 50 psi. They are not true differential transducers, but rather two separate gage pressure transducers which saturate at 50 psi. Since system pressure is often seen to exceed this value, the reported differential pressure becomes zero. They should be replaced with true differential pressure transducers capable of at least 30 psi differential and able to withstand 100 psi gage line pressure. Transducer accuracy of 0.5% full scale should be adequate.