The Bayview Marquis Hotel and Marina Ball Room AHU

Developing a Flow Profile from Proxy's

If you are using the electronic version of this document, you can quickly navigate through it by opening up the bookmarks feature for the .pdf version or the navigation pane for the Word version. Note that if you do this, you will see the headings that are tied to the questions associated with this exercise so you can easily jump to them if you want to.

Overview

Baby Step One

In some ways, I have been trying to work you into energy savings calculations in baby steps. For instance, in our initial efforts at projecting cost savings, (specifically, the pump optimization example) the flow rate (and thus the operating point and all related variables) were steady under all operating conditions.

As a result the only load profile impact was related to the impact of climate on the number of hours the plant operated. That meant you could do one kW calculation for each operating option, and then multiply it by the number of hours it would be in effect and have your answer.

Of course, as we saw, the savings potential for the different pump optimization options, both in terms of energy and dollars, varied significantly. But for this first example, the variation was totally due to the climate (number of hours when the pump needed to run) and the cost of energy. In fact, in the climate where the most energy was saved (St. Louis), the fewest dollars were saved because the cost of energy there is so low relative to the other locations we looked at.

Baby Step Two

Our last exercise introduced additional variables into the savings projection. In other words, not only did the savings associated with repairing the hot water coil valve leak vary with the coil entering condition (climate) and utility rates (cost of energy), they also varied with the flow rate in the air handling system. And they rippled out to impact the chilled water system because the chilled water coil had to remove the unnecessary energy introduced by the leaking valve on the preheat coil.

The flow rate in the system varied because the fan ran at a constant speed, which meant that there would an unintended and unnecessary change in flow rate as the filters loaded among other things. Recognizing this in and of itself was recognizing an opportunity to save energy by adding a control process that would maintain a constant flow rate no matter how clean or dirty the filters were.

But fan energy savings potential aside, the flow variation also impacted the potential to save energy on a number of other fronts.

1. For one thing, at a fixed water side leakage rate and temperature rise (meaning with a constant amount of energy injected into the air stream by the hot water system), the air side temperature rise across the coil will vary with the flow rate,
2. If the air side temperature rise varies with the air flow rate, then that means means the load imposed on the chilled water coil will vary with the air flow rate.
3. In addition, for an economizer equipped system, the impact of variations in outdoor temperature filtered by the economizer. In other words, the entering air temperature to the preheat coils (the mixed air temperature) is not the current outdoor air temperature all of the time. As you will recall from our "perfect economizer" discussion:
    a. Up to a point, if everything is working properly (which it was not in the model, thus another opportunity), then if the outdoor air temperature is below the current mixed air temperature/supply air temperature set point, it will hold steady as the outdoor air temperature drops. This is because the economizer dampers will move to add return air and reduce outdoor air as needed to maintain set point.
    b. In a cold climate, at some point, once the system reaches minimum outdoor air, the mixed air temperature will start to drop as the outdoor air temperature drops with the rate of fall being a function of the percentage of outdoor air relative to return air along with the temperature of both air streams.
    c. Going the other way, as the outdoor air temperature rises above the mixed air/supply air set point, the mixed air temperature and outdoor air temperature will be equal.

d. At some point, which is a function of climate and the nature of the load, it will not longer be cost effective to use 100% outdoor air for cooling and, if everything is working correctly, the system will then revert to minimum outdoor air.

e. Once back on minimum outdoor air, increases on outdoor air temperature will still cause the mixed air temperature to increase, but not on a one for one basis. Rather, it will be a function of the percentage of outdoor air relative to return air along with the temperature of both air streams.

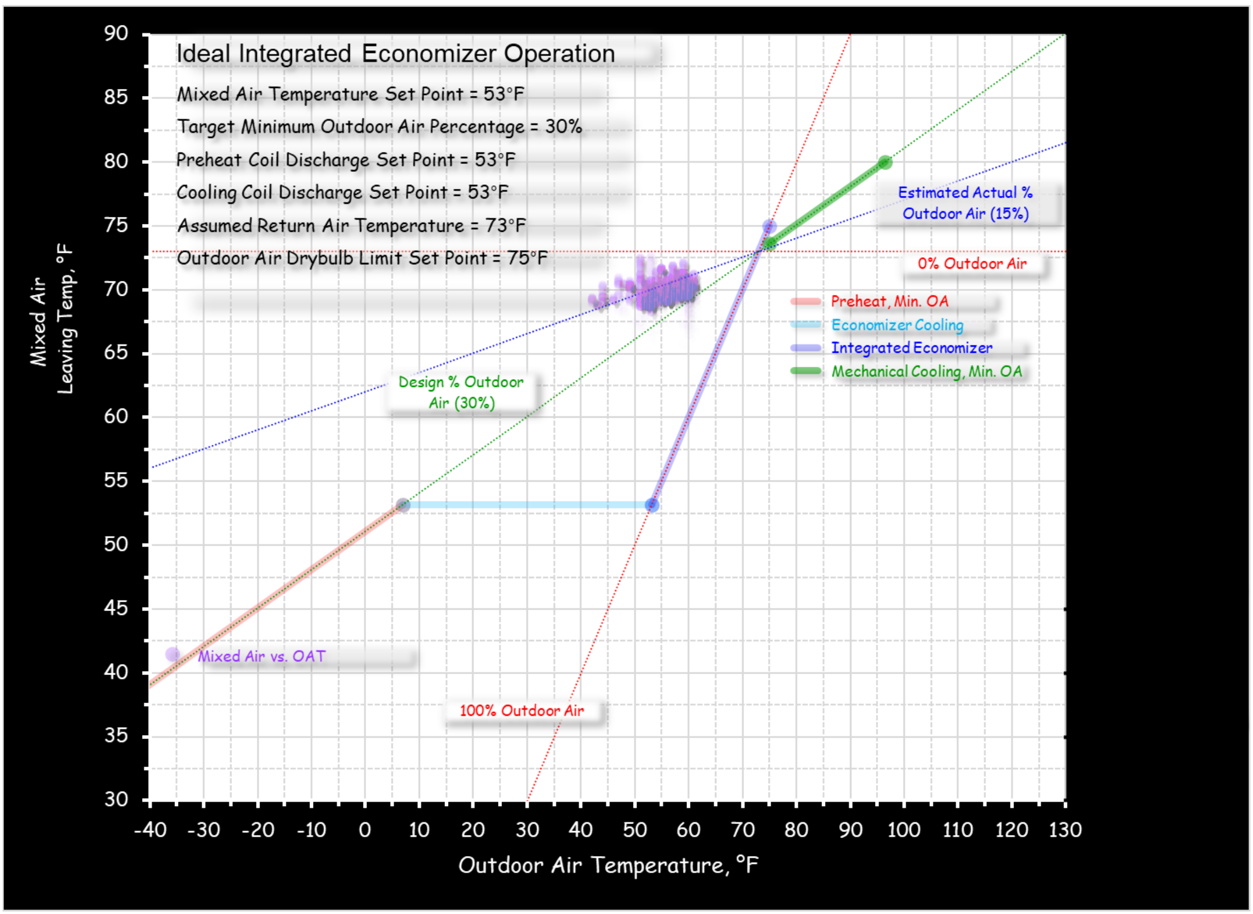

Figure 1 – The Perfect Economizer (see the slides and spreadsheets on the topic from our fist face to face meeting for more information)

4. Even with perfectly sized economizer dampers, the movement of the dampers will change the pressure in the system and cause the flow to vary unless there is a mechanism in place to hold the flow steady. This pressure variation potential is above and beyond the pressure variation potential associate with the filters loading. And it can change very quickly, from minute to minute even,

where as the filter pressure drop will only change very gradually over time for a given air flow rate.

5. The outdoor climate will also impact the efficiency of the chilled water plant, which means that the value of the false load imposed on the chilled water system by the preheat coil will vary.

Bottom line, things can get very complicated, very fast.

But part of the point of the exercise was to help you see that once you recognize an opportunity and understand the energy savings mechanism, frequently, you can justify the cost of multiple improvements based on one, very simple, conservative, estimate of the savings associated with one of them. All of the rest of the benefits simply become "icing on the cake".

For the preheat example, if you extrapolate your field estimate of the preheat coil energy waste occurring as you stand there and observe it by;

1. Coming up with a more reasonable mixed air temperature than you would get from the existing single point sensor by using the mixed air stratification testing technique we have discussed, and
2. Using a conservative air flow rate based on the 350 fpm to 500 fpm face velocity rule of thumb we have discussed, and
3. Using an estimated number of operating hours from the events planning folks,

you would have probably come up with a savings potential in the range of 330,847,200 Btu per year, or about $8,600.

That does not include any of the other savings that could be achieved including the fan energy savings associated with the flow variation introduced into the system by the filters and economizer dampers moving around and the added unnecessary load on the chilled water coil under some operating conditions.

If you start to take some of those variables into account and do an hour by hour estimate with a normalized weather data file, you will probably end up with results similar to the following.

- Preheat energy - 480,678,263 Btu per year or $12,516 per year
- Chilled water energy - 31,626 kWh per year or $5,127 per year

- Fan energy - 12,007 kWh per year or $1,946 per year

So in the range of $19,000 - $20,000 a year in savings if you "sharpen your pencil". And those numbers assume the economizer is working; and in the model, it obviously had issues.

The ROI perspective I often encounter from your Owners is usually in the range of a 2 to 3 year simple payback. I think that means that if you felt you could save $8,600 per year (the ball park preheat coil energy savings) by fixing the preheat coil valve leak, they would consider spending in the range of $17,200 - $25,800 to support the repair.

Given the talent and creativity I observe in the typical Marriott Engineering team, I suspect for that kind of money, you could:

1. Fix the leaking valve, and
2. Add a pressure sensor and control logic to hold the fan speed steady under all operating conditions, and
3. Improve the economizer by:
    a. Improve the damper performance by repairing the blade seals and disabling damper blades to improve linearity (or simply replacing the dampers with better sized and configured dampers), and
    b. Adding a baffle to help promote mixing, and
    c. Replacing the single point sensor with an averaging sensor or two, and
    d. Stringing out the freeze-stat element and maybe adding another one or two so the coil is fully covered,

In doing that, you would not only capture the $8,600 associated with wasted preheat energy, you would also pick another $10,000 - $11,000 in savings from the other things you improved. That makes the recommendation and investment a pretty safe bet. And that is important for your Owner because at the end of the day, they are trying to make money for their investors and delivering a successful project on something like this would give you a pretty good track record in terms of making money for them (and I believe most of you have Marriott stock).

Baby Step Three

The current exercises adds a new “wrinkle”. Specifically, in addition to all of the other variables we have been discussing, it throws in a variable by design; i.e. the variation in flow rate as a result of the design intent of the system.

In other words, the design intent of a variable flow system is that the flow in the system (and thus the pressure required by the system) will vary with the load. For a fixed system, that means that the energy associated with serving the load in terms of distribution pump energy or fan energy will vary as the cube of the flow rate for a fixed system (see Equation 2 and related discussion below).

That is a very powerful relationship but it is not true for many of our systems because they are not fixed systems. In a technical sense, the words “fixe system” mean that the system curve is steady state and never changes; i.e. no dampers or valves move out in the network served by the fan or the pump.

In the real world most of us live in, for a variable flow water system, every time a valve moves, it creates a new system curve. For a variable volume air handling system, each time a terminal unit damper moves, it creates a new system curve. So for most (but not all) of our systems, the fan or pump power required to meet the load will not vary as the cube of the flow rate; rather, it will vary by an exponent that is less that 3 (perfection) but most likely, more than 1.

But the flow vs. fan or pump power relationship still is a powerful relationship, even if the exponent is less than three. But in doing our calculations, we need to be careful to not overstate the benefits. And with exponents above 1, it is pretty easy to do that.

The goal of this exercise is to help you “see” that, and also help you understand how to come up with the flow profile you need for your energy savings calculation in the first place. For those of us blessed with newer systems, frequently a flow meter is included in the controls package. But for those of us working with legacy systems (probably most of us), we do not have that luxury.

And truth be told, the flow meters provided in new systems are often not accurate as the result of installation or calibration issues. So having a way to do a “sanity check” on the data you are getting from them is not a bad thing and the techniques we will use in this exercise are a good way to do that.

Proxy's

The approach we will use depends on a "proxy" for system flow rate. The definition of "proxy" that applies here is:

> *A figure that can be used to represent the value of something in a calculation.*

In other words, the distribution system in the central chilled water plant in the model did not have a flow meter installed in it. But, as Jim Brooks, one of my mentors would have said:

> *Tons are tons.*

That was Jim's way of stating conservation of mass and energy. In other words, the tons produced by the chillers are exactly equal to the tons used by the loads they served. That means that if I can come up with the tons the chillers are producing, I may be able to come up with the flow rate in the distribution loop.

To get more specific, by design, the variable flow primary/secondary arrangement of the chilled water plant in the model we are working with ensures that the flow rate through the chillers will hold steady, no matter what happens out in the distribution system. That means:

- If we do a pump test for the evaporator pumps, we know the flow rate through a chiller under any operating condition, and
- If we log temperature difference across the chiller, by virtue of the water side load equation, we can calculate the tons the chiller is producing for each logged data interval simply by doing the math illustrated in Equation 1, then

If we log temperature difference across the mains leaving the plant,

$$Q_{Btu/Hr} = 500 \times Flow_{gpm} \times \left(t_{Entering,°F} - t_{Leaving,°F}\right)$$

Where:

$Q_{Btu/Hr} =$ Load in Btu/hr

$500 =$ Units conversion constant, good for water between 30 and 200°F

$Flow_{gpm} =$ Flow through the heat exchanger in gallons per minute

$t_{Entering,°F} =$ Temperature entering the heat exchanger in °F

$t_{Leaving,°F} =$ Temperature leaaving the heat exchanger in °F

Equation 1 – Water Side Load

we can solve the water side load equation for flow rate based on the tons we calculate from our logger data and pump test. In other words, the tons produced by the chiller have become a proxy for the flow rate in the distribution system.

Supporting Information

On the class web page, you will find the following information.

Logger Data

There are three sets of logger data for the chilled water plant in the model.

- CH-1 and CH-2 CHW Temps.txt is the data logged for each of the chillers; specifically the entering and leaving chilled water temperature for each machine.
- Evap Pump 1 and 2 Amps.txt is the motor amp data for each of the evaporator pumps.
- Plant CHWS and CHWR.txt is the plan leaving and entering water temperature data.

Weather Data

You will find a file from the ASOS site at the San Diego Airport (SAN ASOS.txt) that covers the data range during which the data loggers were installed on the chillers. The Bayview Marquis Hotel is very near the airport; in fact, you can watch the planes come and go if you have a room on the north east side of the hotel. So the weather data from that location is a good representation of the climate conditions at the hotel location.

Pump Curves

You will find digitized versions of three pump curves.

- Bell and Gossett e1510 6G Curve.xlsx is the existing evaporator pump curve.
- Bell and Gossett e1510 5A Curve.xlsx is the existing distribution pump curve.
- Bell and Gossett 1510 5E Curve.xlsx is a curve for a distribution pump that would be much more efficient than the existing distribution pump

Pump Test Results

For this exercise, assume you did pump tests on the evaporator pumps that yielded the following results.

Pump	Operating Mode	Suction Pressure psig	Discharge Pressure psig
Chiller 1	Throttled	145.0	162.0
Chiller 1	Dead Headed	147.0	168.5
Chiller 2	Throttled	145.0	162.5
Chiller 2	Dead Headed	146.5	168.0

Distribution System Pressure Parameters

For the purposes of the exercise, you can make the following assumptions.

- At design conditions, the plant needs to maintain 70 ft.w.c. across the distribution headers to deliver design flow to the hydraulically most remote location.
- The hydraulically most remote location is the equipment room with the Ball Room air handling unit in it that we have been using in our exercise.
- If there is 30 ft.w.c. of differential pressure available across the mains where the Ball Room AHU connects to them, then the Ball Room AHU cooling coil can achieve design chilled water flow.

Southern California Edison Pump Power Exponents

In the webinar and previously in this exercise instruction document, we discussed the fact that you an not directly use the pump affinity law for as a function of flow (Equation 2) for most of our variable flow systems because the flow variations in them are caused by the movement of valves and dampers; each time a valve or damper moves a new system curve is generated and the affinity laws only apply on a fixed system curve.

Then, in the webinar, I showed you a technique you can use to develop a relationship for pump power as a function of flow based on the system and load pressure drop characteristics, arbitrarily selected flow conditions, and pump curve analysis.

$$Bhp_{New} = Bhp_{Old} \times \left(\frac{Flow_{New}}{Flow_{Old}} \right)^3$$

Where:

Bhp_{New}	=	The brake horse power you want to know in consistent units
Bhp_{Old}	=	The brake horse power you know in consistent units
$Flow_{New}$	=	A flow rate you have selected and want to predict the brake horse power requirement for
$Flow_{Old}$	=	The flow rate associated with the known brake horse power

Note that the relationship can be applied using kW or any other power metric as long as you keep the units consistent. In other words:

$$kW_{New} = kW_{Old} \times \left(\frac{Flow_{New}}{Flow_{Old}} \right)^3$$

Equation 2 - Pump (and Fan) Power vs. Flow Affinity Law a.k.a. "The Cube Rule"

The advantage of the approach I showed you is that it is based on the physics of your system and its equipment and thus, has the potential to provide a fairly accurate result. The over-all accuracy will be a function of the assumptions you make about the pressure drop from the plant to the hydraulically most remote load[1].

But, the approach I showed you also takes a bit of time, effort, and understanding. Because of that, a consortium of California utility companies, led by Southern California Edison, did some research and modeling to see if they could come up with prescriptive exponents that could be used in the pump/fan power affinity law that would provide reasonably accurate results and would allow pump power to be calculated directly from a flow profile by using the affinity law with an exponent other than three.

In other words, they know a result based on the cube rule would not be correct. But they suspected that for a given circumstance, there might be an exponent somewhere between 1 and 3 that could be applied which, while not perfect, would

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[1] The hydraulically most remote load is the load that requires the most differential pressure at the plant to deliver its design flow on the design day. Frequently, but not always, it is also the most physically remote load.

generate a result that was good enough for awarding incentives in utility incentive program that was targeting flow optimization strategies.

As a result of their research, they published a number of exponents and guidelines for their use. That means that if you have data that tells you the flow rate for a system for a given point in time, you can use a modified version of the cube rule (i.e. the cube rule with an exponent that is less than three which is based on the SCE research) to calculate the pump power for that condition.

I actually intended to show you this in the webinar but forgot to do it because I jumped into my slide deck after the slides that covered it to answer a question and then forgot to go back. (I suspect that the other contributing factor was that I had about 12 hours of sleep over the past 3 days, so I was a little "woozy".)

Having said that, in the exercise materials, you will find two files that provide this information.

- Variable Flow Fan and Pump Affinity Law values_SCE_2013.pdf is the Southern California Edison (SCE) publication that documented the exponents and their application.
- Pump Power vs Flow Exponent Tool.xlsx is a tool I built that is based on the SCE publication. The first tab provides a detailed explanation of the concept. The second tab has a tool that allows you to estimate an exponent based on interpolating the SCE data for your particular system. The third tab graphically compares the different exponents with the cube rule.

Adding Trend Lines to Charts and Using VLOOKUP

To do this exercise, you will need to use the Excel function VLOOKUP and know how to add trend lines to the data in a scatter plot. You will find some information on VLOOKUP in these blog posts and information on scatter plots, adding trendlines and doing regressions (including video screen captures of me doing it) in these blog posts.

The Exercise

The goals of the exercise are:

1. Develop a plant load profile from the logger data provide.
2. Develop a plant flow profile from the logger data provided.

3. Assess the load and flow profiles to see if you think there are any issues with them that either indicate an operational problem that needs to be resolved or indicate a problem with the data that needs to be addressed.
4. Use the Excel trendline feature to project the load profile for an entire year and assess it to see if there are issues you would need to address if you wanted to use it for energy savings projections.
5. Using the flow profile you generate and the SCE coefficients, project what it costs to operate the distribution pumps currently based on maintaining a fixed differential pressure at the plant headers.
6. Using the flow profile you generate and the SCE coefficients, project what it would cost to operate the distribution pumps based on maintaining a fixed differential pressure across the mains at the Ball Room AHU equipment room.

Step 1

Using the logger data for the chiller entering and leaving water temperatures and the pump amps along with the pump test results provided above to calculate the tons produced by each chiller for each logged data interval. Then use the logged data for the plant entering and leaving water temperature and the total tons produced by the chillers to calculate the distribution system flow rate for each logged data interval.

Step 2

Add temperature data from the ASOS weather data file to your data set so you have an outdoor air temperature associated with each logged data interval.

Step 3

Plot your results as time series to get your first look at the plant load and flow profile.

Question 1

Do the results seem reasonable and match your expectations?

Question 2

If there are issues, see if you can correct them.

In general terms, you should fairly smooth curves with the load generally, but not totally following the outdoor air temperature. Depending on how you combined the data, you may see some large spikes.

Question 3

If you have them, can you explain why they happened and can you figure out a way to filter them out?

Step 4

Plot your exercise results as a scatter plot of tons as a function of outdoor air temperature (i.e. outdoor air temperature is on the X axis).

Question 4

Does the pattern surprise you or does it make sense? To give you a bit more perspective, here is what the pattern would look like if you had an entire year's worth of data.

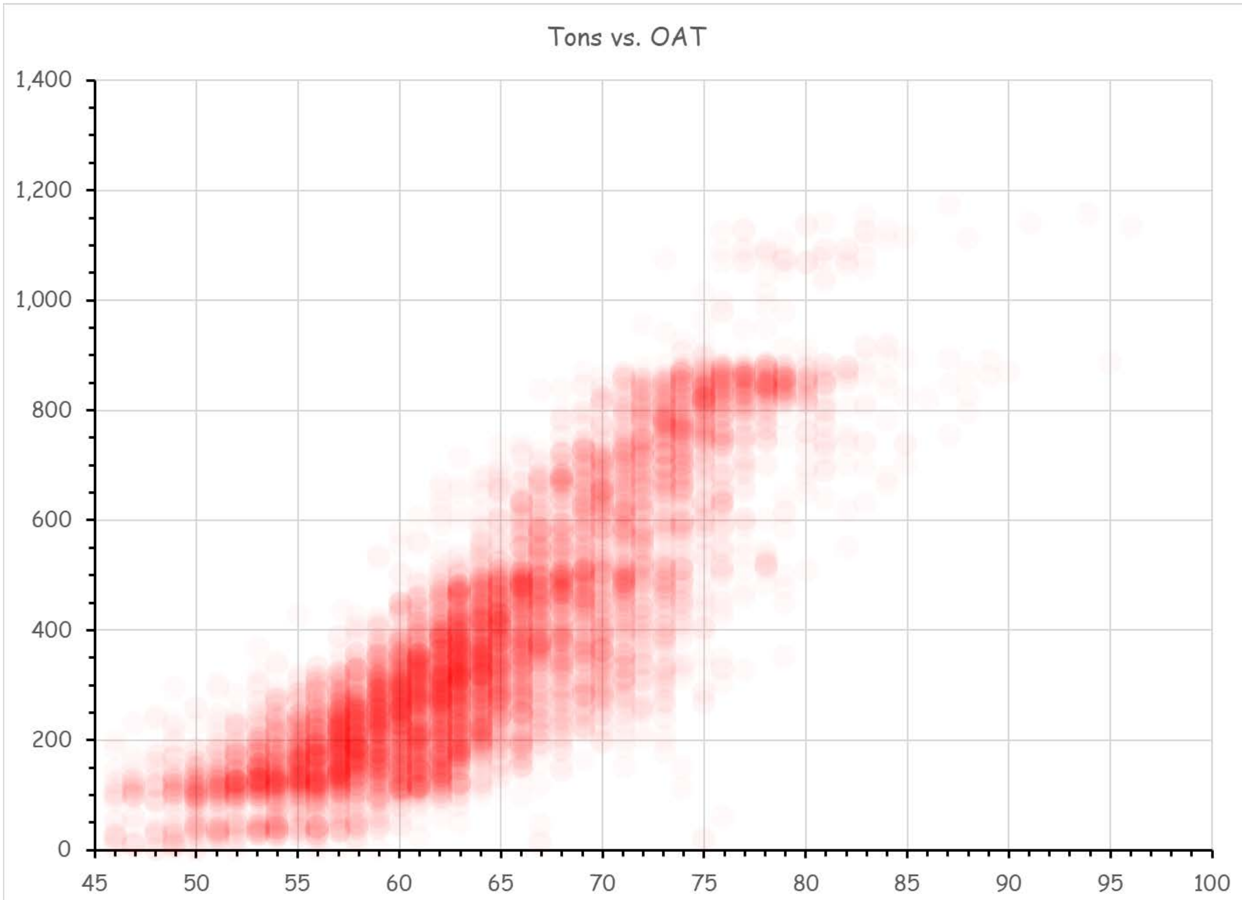

Figure 2 – A Full Year's Worth of Data

You may also want to reference the slides I presented in Webinar 2 - Framing Up Your Project.

Question 5

Do you think the shape of the cloud would be different for a more hot and humid climate and if so, what difference would you expect to see.

Question 6

If all of the systems in the facility were 100% outdoor air systems, how would the shape of the cloud compare to a facility of the same size but with integrated economizer equipped air handling systems only (no 100% outdoor air systems)?

Question 7

What are the causes of the data scatter; in other words, why are there different tonnages associated with a given outdoor air temperature?

Step 5

Using the Excel trendline feature, develop an equation for the relationship between tons and outdoor air temperature. Then add a second series to your scatter plot that projects the tons for an entire year based on the relationship you have developed.

Question 5

Have you identified a weakness in your data set in terms of making an annual energy savings projection from it?

Question 6

If so, how can you address this issue so you can proceed with developing a saving projection?

Step 6

Make a scatter plot of flow vs. outdoor air temperature. Once you have the scatter plot, add the trend line to it and the use that to project the flow vs. outdoor air temperature for the entire year.

Step 7

Make any corrections you feel are necessary to the annual flow profile and then use it and the appropriate SCE coefficient to project the distribution pump operating cost for the entire year using the current approach (maintaining discharge header pressure).

Step 8

Select the appropriate SCE coefficient to reflect controlling the pumps based on the differential pressure across the mains in the Ball Room AHU mechanical room and calculate the cost savings you would achieve by doing this.

Step 9

Make a list of the things that would need to be done to capture these savings.

Question 7

Based on your list, do you think you have identified a viable ROI project?

Question 8

Do you think your project should just focus on adding the differential pressure sensor or do you think you should try to leverage the savings to make a few additional improvements to the chilled water system? If so, what might you consider adding to the project.

Question 9

Do you think you could use a similar technique to develop a cost savings projection for using a remote sensor to control a VAV air handling system?

Question 10

If the VAV air handling unit did not have a flow meter, what could you use as a proxy for flow and what are some of the benefits and limitations of the approach or approaches you come up with?

Question 11

Is there anything different that you would need to do in terms of the control logic if you wanted to use a remote sensor to control an large VAV system fan as compared to using a remote sensor to control a distribution pump in a large variable flow water system?

You should also feel free to try to develop the cost savings using the pump curve based technique I demonstrated in the webinar. But I recognize this will take some time and if you do it with the SCE coefficients, you will have the general idea of how to do the calculation, which is what you need to know in some cases for your personal and group projects.

Having said that, do you see a weakness in the SCE coefficient based approach. There is one and it is the reason I came up with the approach I showed you. It is most obvious if you look at the graphical representation of the SCE coefficients that is on the third tab of the Pump Power vs Flow Exponent Tool.xlsx tool.