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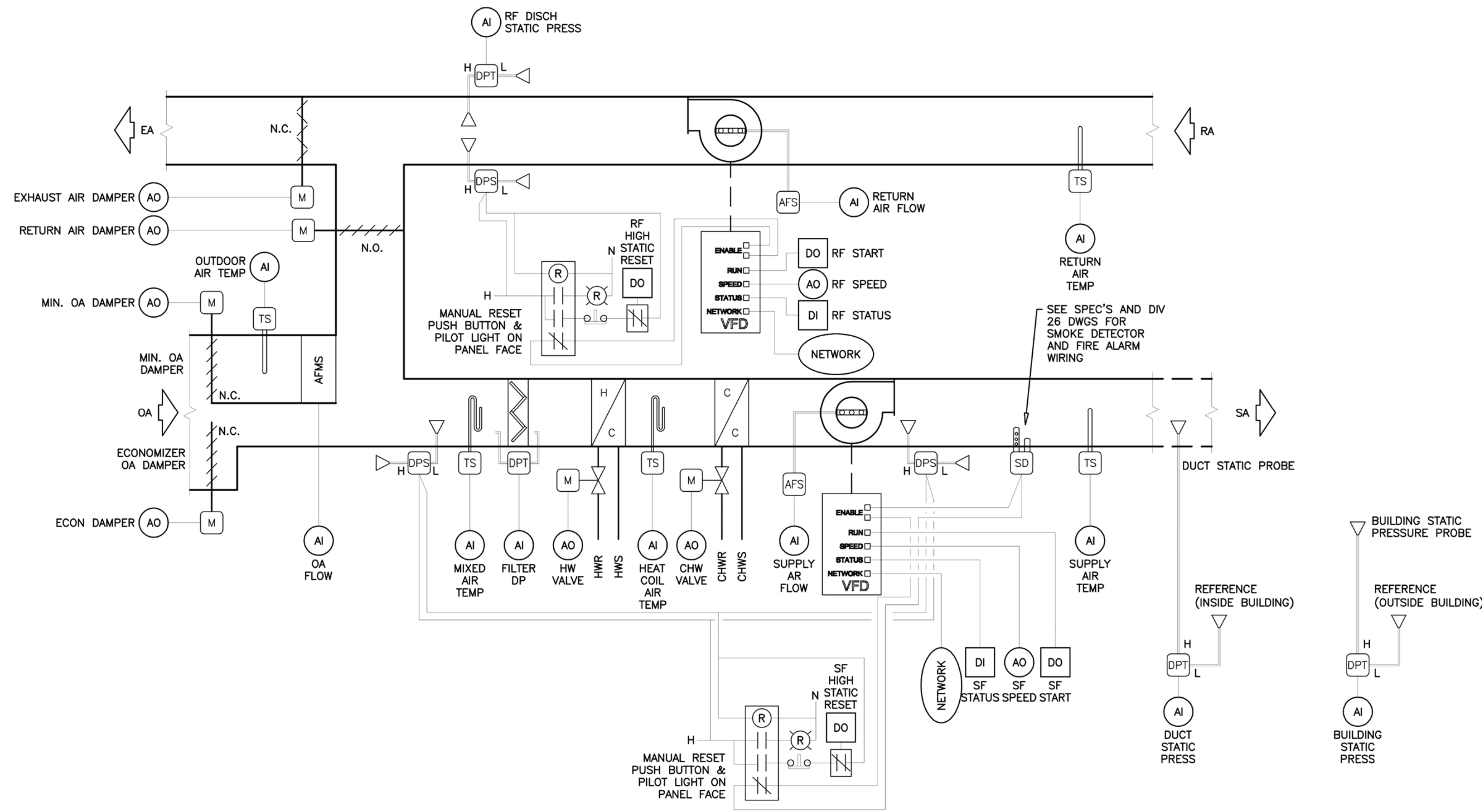
Easy Button
And
AHU Configurator/Shaft Placement

ASHRAE 36:



A-9 Multiple zone VAV air handling unit with return fan and OA measurement station.

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ASHRAE 36-2021 Additions:

Need to add ERU, Pumped preheat coil (freeze protection), and humidifier.

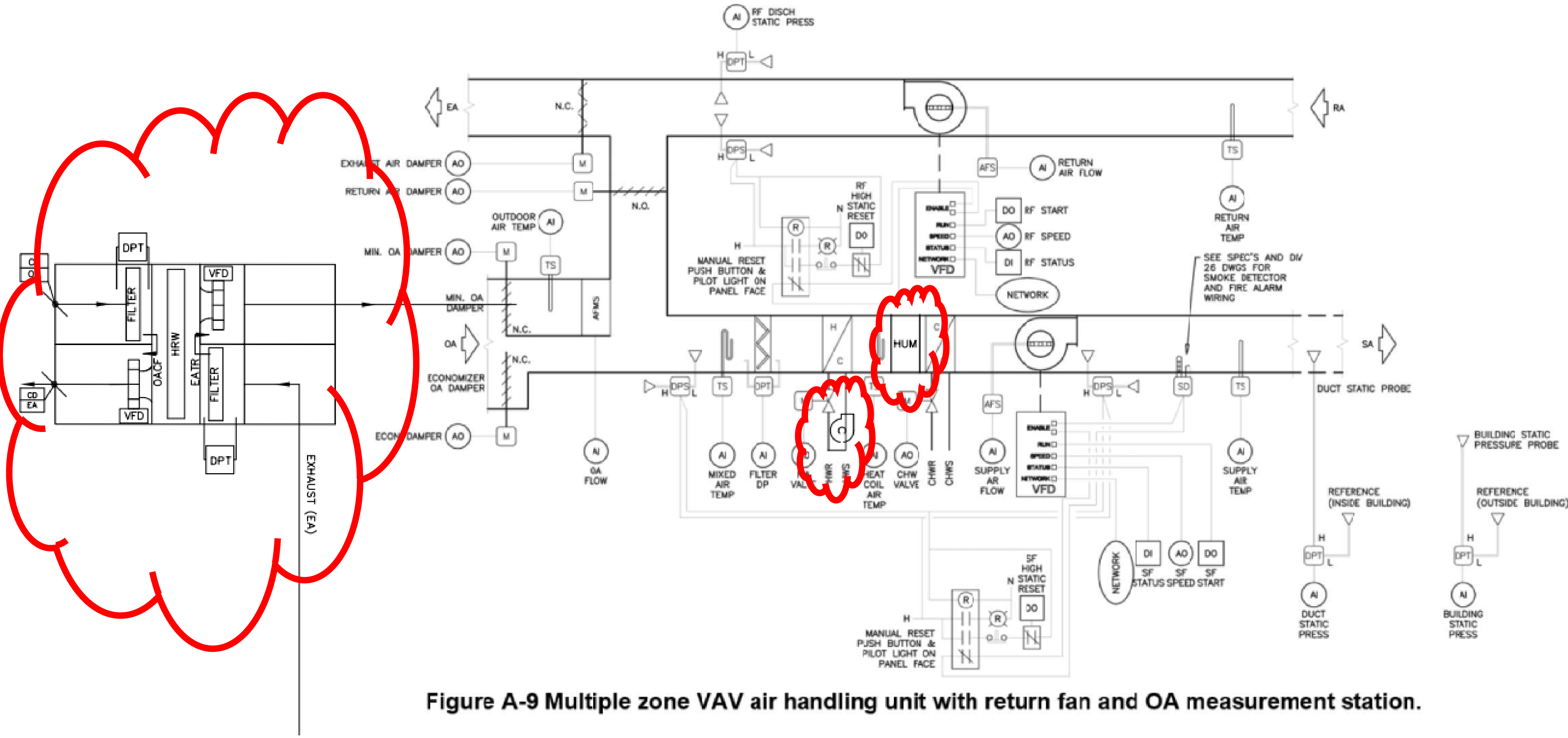

Figure A-9 Multiple zone VAV air handling unit with return fan and OA measurement station.

ASHRAE 36-2021 Additions:

If an ERU is not required, an exhaust fan is required (if applicable).

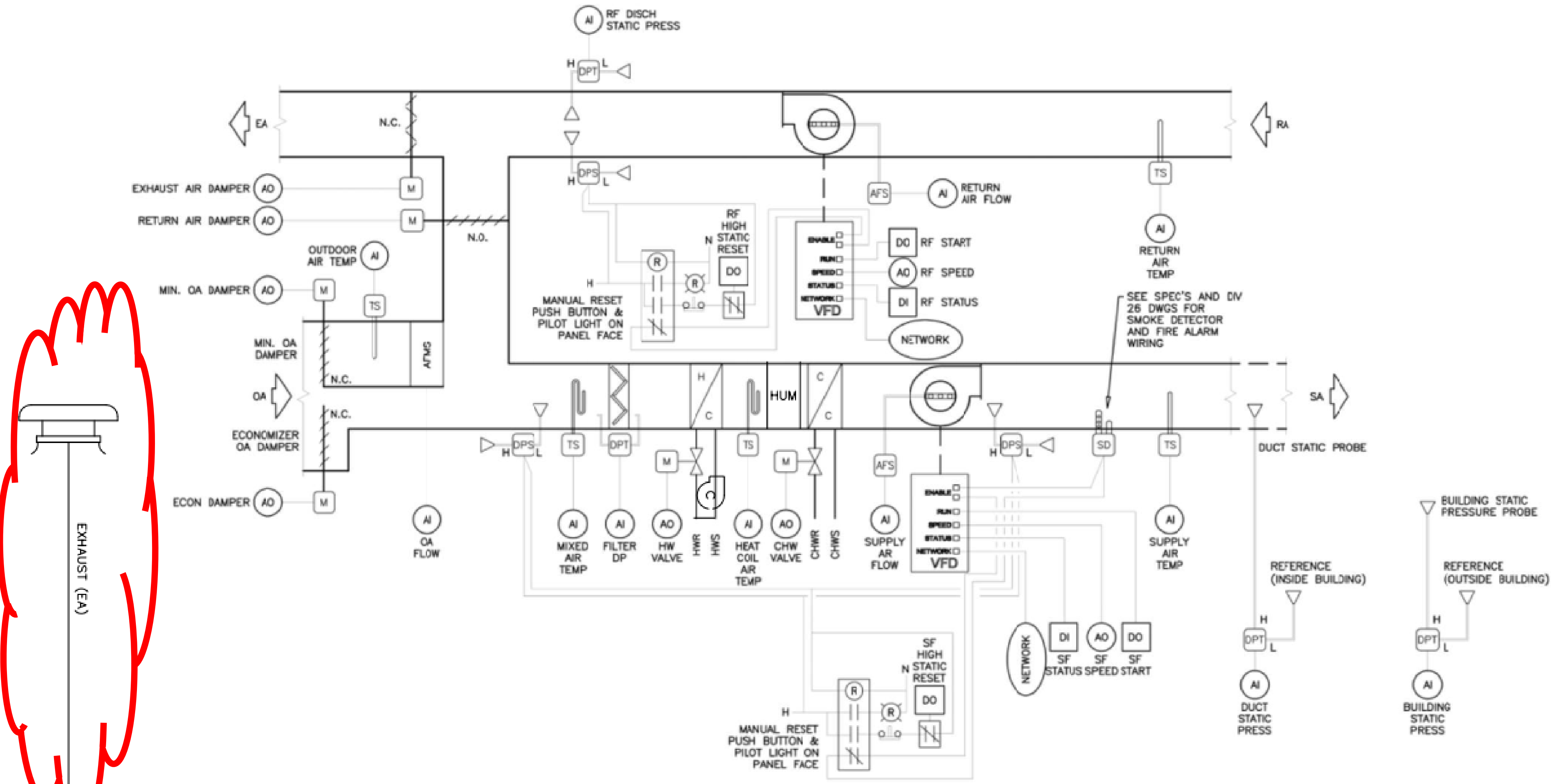

Figure A-9 Multiple zone VAV air handling unit with return fan and OA measurement station.

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Why this multizone VAV?:

VAV with reheat systems are the most economical and efficient systems for most buildings.

- Steven Taylor https://tayloreng.egnyte.com/dl/5XWanNURh0/ASHRAE_Journal_-_VAVR_vs_ACBDOAS.pdf_

IECC Requires Economizer in most situations.

C403.5 Economizers (Prescriptive)

Economizers shall comply with Sections C403.5.1 through C403.5.5.

An air or water economizer shall be provided for the following cooling systems:

1. Chilled water systems with a total cooling capacity, less cooling capacity provided with air economizers, as specified in Table C403.5(1).
2. Individual fan systems with cooling capacity greater than or equal to 54,000 Btu/h (15.8 kW) in buildings having other than a *Group R* occupancy.

   The total supply capacity of all fan cooling units not provided with economizers shall not exceed 20 percent of the total supply capacity of all fan cooling units in the building or 300,000 Btu/h (88 kW), whichever is greater.

No fans, filters, or motors above occupied zones:

Easier maintenance, less noise.

ASHRAE 15 – no refrigerant piping routed through occupied spaces

Revit projects tend to skew larger. Many smaller projects that are better suited to VRF/unitary are still done in CAD.

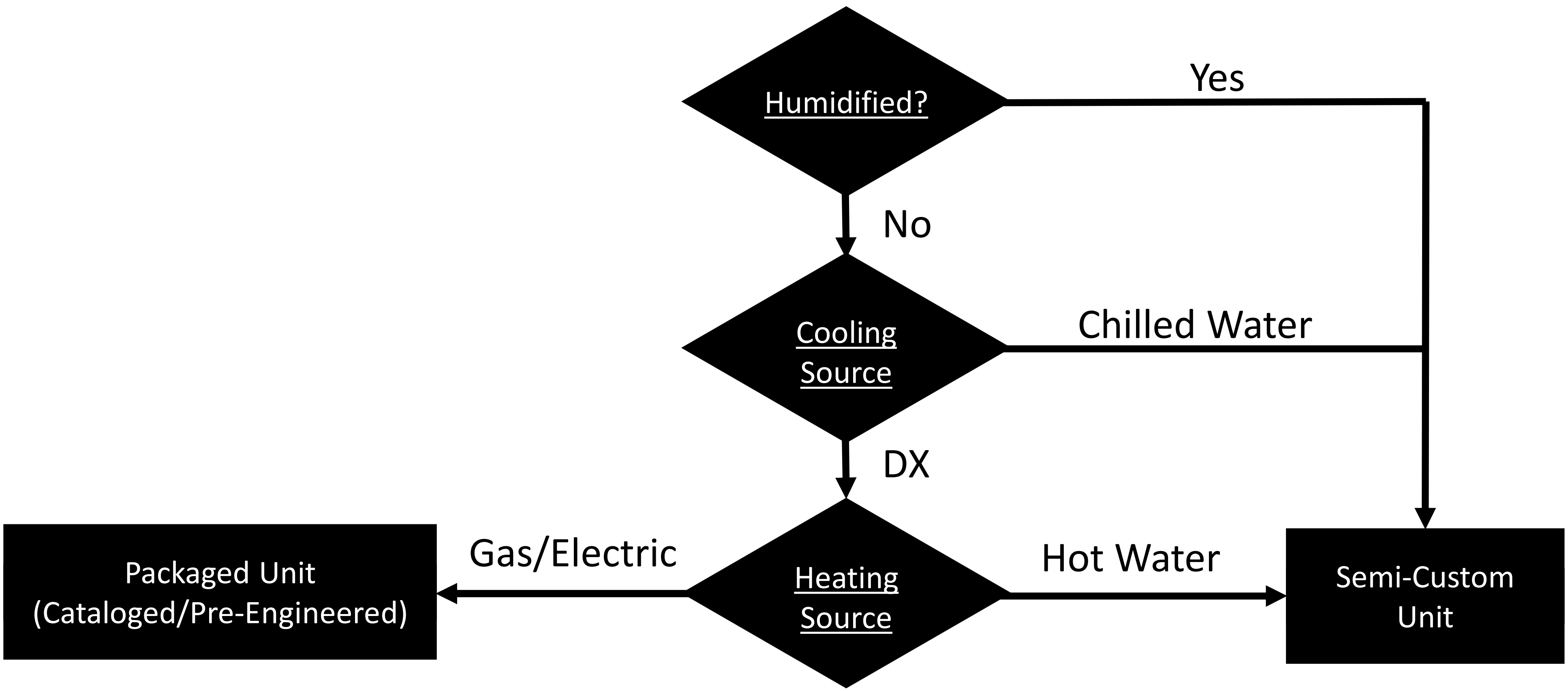
AHU/RTU Model and Component Selection:

Rules of thumb are adjustable and will eventually be replaced with a building-specific energy model.

Cooling Source Selection
Project Load : Cooling Source
0-100 Tons : DX
100-500 Tons : Chilled Water (Air Cooled)
> 500 Tons : Chilled Water (Water Cooled)

Heating Source Selection
Project Load : Heating Source
0-1500 MBH : Gas (if available) or Electric
> 1500 MBH : Hot Water (If gas is available)

Humidifier Selection
If climate zone >3A or specialty zone, provide a humidifier



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Diversity Vs. Safety Factor:

Safety Factor in HVAC design is problematic for 2 reasons:

1) There are multiple places to take a safety factor and it increases exponentially through the design.
    a. Example: The intern takes a safety factor on the window U-values, the project engineer takes those loads and adds a safety factor, the principal adds a safety factor to the airflow when talking to the rep, then the engineer in charge of the hydronic system takes a safety factor when sizing a the chillers and now the chillers are 3x the size they need to be

2) In most engineering disciplines, oversized is good or at least doesn't hurt. That is not true in HVAC. Oversized equipment uses excessive energy and has poor performance.
    a. Example: An oversized DX unit will run for a few minutes every hour and then dump humid OA into the space for the rest of the hour.

Safety Factor Example:

10 rooms, 800 CFM/room, 8000 CFM AHU (No diversity)

10 rooms, 1000 CFM/room (25% factor of safety), 8000 CFM AHU (20% diversity).

Same AHU size, one is hard to balance, and one is not.

The Ripple Easy Button takes **No Safety Factor** and **No Diversity**.

ERUs:

Table 6.5.6.1.2-1 Exhaust Air Energy Recovery Requirements for Ventilation Systems Operating Less than 8000 Hours per Year

Climate Zone	% Outdoor Air at Full Design Airflow Rate							
	≥10% and <20%	≥20% and <30%	≥30% and <40%	≥40% and <50%	≥50% and <60%	≥60% and <70%	≥70% and <80%	≥80%
	Design Supply Fan Airflow Rate, cfm							
3B, 3C, 4B, 4C, 5B	NR	NR	NR	NR	NR	NR	NR	NR
0B, 1B, 2B,5C	NR	NR	NR	NR	≥26,000	≥12,000	≥5000	≥4000
6B	≥28,000	≥26,500	≥11,000	≥5500	≥4500	≥3500	≥2500	≥1500
0A, 1A, 2A, 3A, 4A, 5A, 6A	≥26,000	≥16,000	≥5500	≥4500	≥3500	≥2000	≥1000	≥120
7,8	≥4500	≥4000	≥2500	≥1000	≥140	≥120	≥100	≥80

NR—Not required

Exceptions: Where the largest source of air exhausted at a single location at the building exterior is less than 75 percent of the design outdoor air flow rate.

Air does not mix proportional to flow and temperature!

$$Flow_1 * Temp_1 + Flow_2 * Temp_2 \neq Flow_3 * Temp_3$$

Only enthalpy, humidity ratio, and mass flow rate mix proportionally

You must calculate the mixed enthalpy and humidity ratio, then calculate the resultant dry bulb and wet bulb temperature.

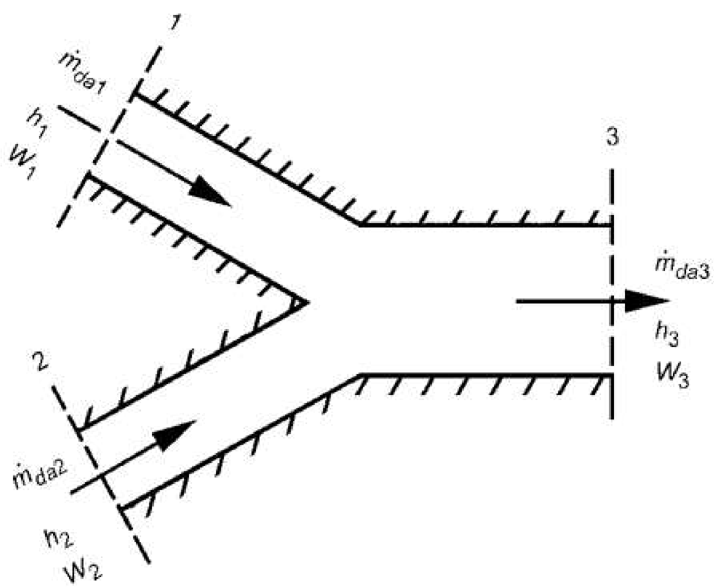


Fig. 6 Adiabatic Mixing of Two Moist Airstreams

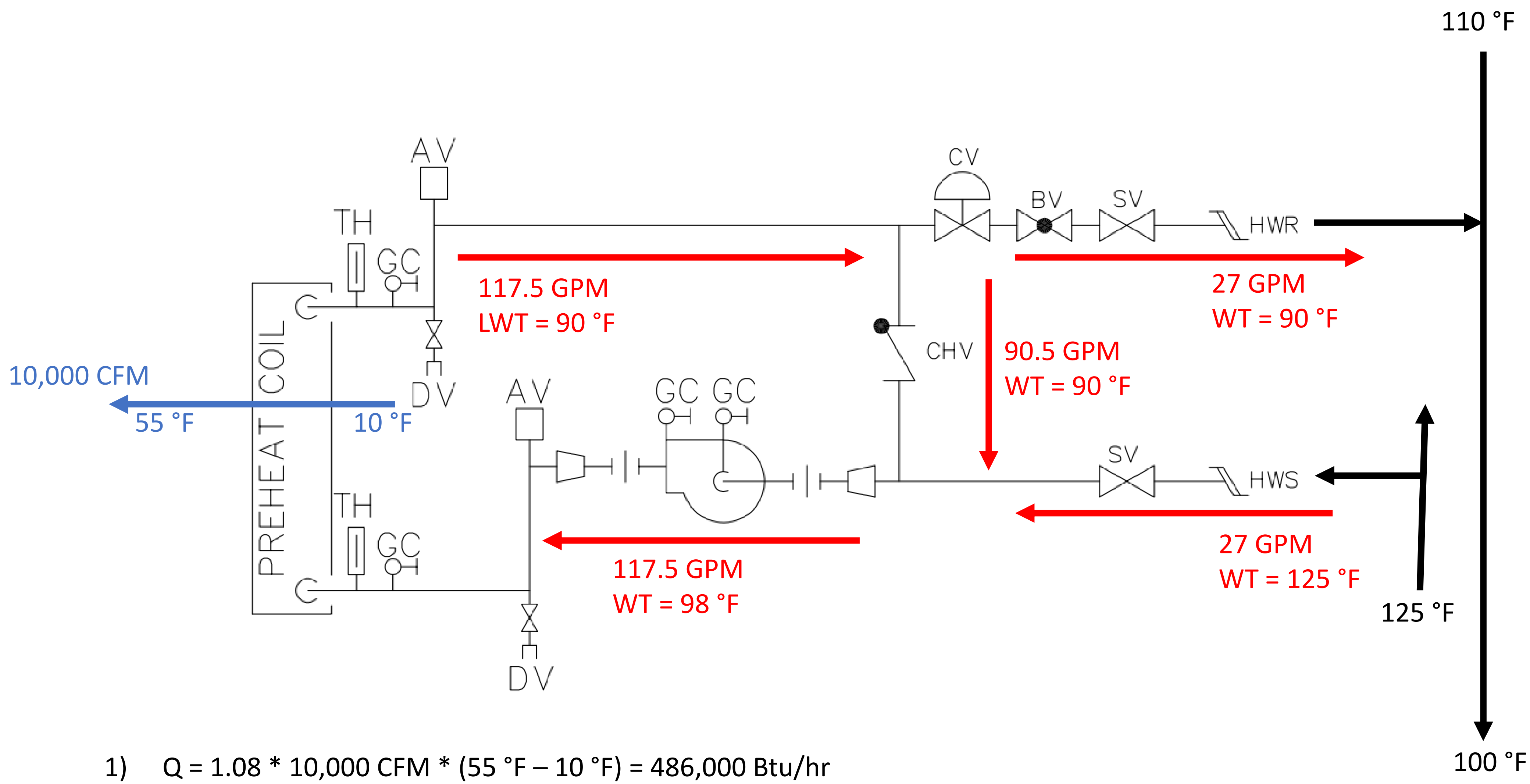
Adiabatic Mixing of Two Moist Airstreams

A common process in air-conditioning systems is the adiabatic mixing of two moist airstreams. Figure 6 schematically shows the problem. Adiabatic mixing is governed by three equations:

$$\dot{m}_{da1} h_1 + \dot{m}_{da2} h_2 = \dot{m}_{da3} h_3$$
$$\dot{m}_{da1} + \dot{m}_{da2} = \dot{m}_{da3}$$
$$\dot{m}_{da1} W_1 + \dot{m}_{da2} W_2 = \dot{m}_{da3} W_3$$

Source: ASHRAE Fundamentals Chapter 1.

Pumped Preheat Coil Selection Example:



1) $Q = 1.08 * 10{,}000\ CFM * (55\ °F – 10\ °F) = 486{,}000\ Btu/hr$
2) 117" x 102" AHU, 64 1/2" ID tubes fed, 117.5 GPM required to maintain 3 fps.
3) Desired return water temperature, $T_R = 100\ °F – 10\ °F = 90\ °F$.
4) Required Inlet Temperature, $T_s = 90\ °F + (486{,}000\ Btu/hr / (500 * 117.5\ GPM)) = 98\ °F$
5) $GPM_{HWSMain} = (117.5\ GPM * (98\ °F – 90\ °F)) / (125\ °F – 90\ °F) = 27\ GPM$

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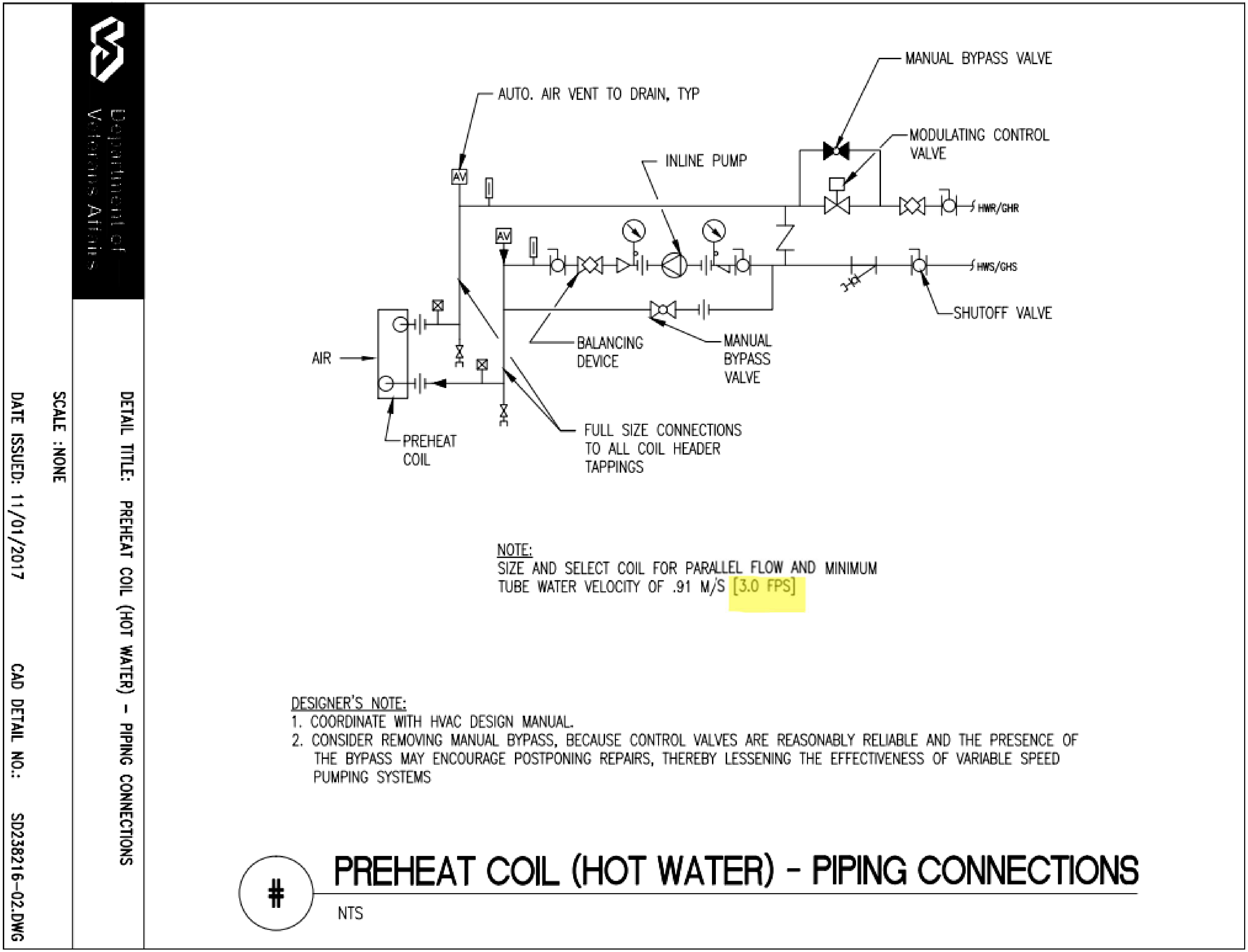
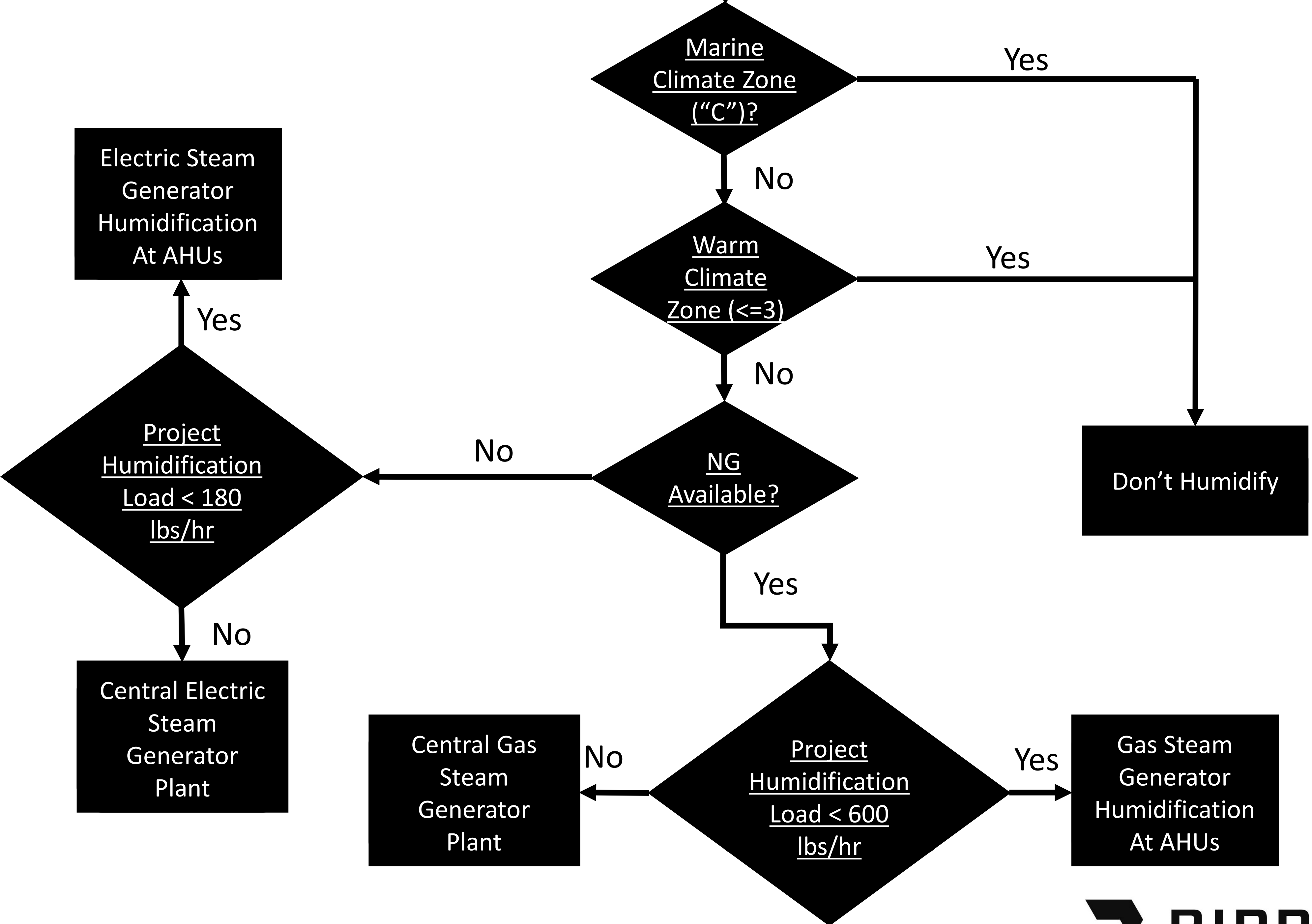
Pumped Preheat Coil (VA Hospital Detail):



Humidification Decision Flow Chart:



ERUs:

Max Air Reintrainment, Class 3 can be reclassified to Class 1 if less than 5%

Less than 5% is easy to do, just leave it at that and you'll never have problems.

Filtration:

AirXchange recommendations.

5.13.3.2.2 Recirculation or transfer of Class 2 air to other Class 2 or Class 3 spaces shall be permitted, provided that the other spaces are used for the same or similar purpose or task and involve the same or similar pollutant sources as the Class 2 space.

5.13.3.2.3 Transfer of Class 2 air to toilet rooms shall be permitted.

5.13.3.2.4 Recirculation or transfer of Class 2 air to Class 4 spaces shall be permitted.

5.13.3.2.5 Class 2 air shall not be recirculated or transferred to Class 1 spaces.

Exception to 5.13.3.2.5: When using any energy recovery device, recirculation from leakage, carryover, or transfer from the exhaust side of the energy recovery device is permitted but shall not be counted as outdoor air. Exhaust air transfer ratio of Class 2 air shall not exceed 10% of the outdoor air intake flow at the design static pressure differential as defined in AHRI 1060.

5.13.3.3 Class 3 Air

5.13.3.3.1 Recirculation of Class 3 air within the space of origin shall be permitted.

5.13.3.3.2 Class 3 air shall not be recirculated or transferred to any other space.

Exception to 5.13.3.3.2: When using any energy recovery device, recirculation from leakage, carryover, or transfer from the exhaust side of the energy recovery device is permitted but shall not be counted as outdoor air. Exhaust air transfer ratio of Class 3 air shall not exceed 5% of the outdoor air intake flow at the design static pressure differential as defined in AHRI 1060.

Filtration:

62.1:

6.1.4.1 Particulate Matter Smaller than 10 Micrometers (PM10). In buildings located in an area where the national standard or guideline for PM10 is exceeded, particle filters or air-cleaning devices shall be provided to clean the outdoor air at any location prior to its introduction to occupied spaces. Particulate matter filters or air cleaners shall have either

a. a MERV of not less than 8 where rated in accordance with ASHRAE Standard 52.2 or
b. the minimum efficiency within ISO ePM10 where rated in accordance with ISO 16890.

Informative Note: See Informative Appendix E for resources regarding selected PM10 national standards and guidelines.

6.1.4.2 Particulate Matter Smaller than 2.5 Micrometers (PM2.5). In buildings located in an area where the national standard or guideline for PM2.5 is exceeded, particle filters or air-cleaning devices shall be provided to clean the outdoor air at any location prior to its introduction to occupied spaces. Particulate matter filters or air cleaners shall have either

a. a MERV of not less than 11 where rated in accordance with ASHRAE Standard 52.2 or
b. the minimum efficiency within ISO ePM2.5 where rated in accordance with ISO 16890.

Preheat Coils:

AHU hot water preheat coil selection can be tricky. Typically, the main goal of a coil is heat transfer, but the main goal of a preheat coil is not to freeze.

Given the low entering air temperatures compared to the high entering water temperatures, the coil may need very little flow to maintain discharge air temperature. This low flow causes poor mixing in the tubes, which can lead to localized freezing which will then stop all flow and cause the freeze stat to trip and cause an unscheduled outage.

The solution is typically achieved by either adding glycol to the heating water system or by pumping the coil (and sometimes both).

Adding glycol to the system reduces the specific heat of the heat transfer fluid and therefore requires larger coils, more pumping power, larger piping, and results in a more costly and complicated system to maintain.

Pumping a coil reduces the delta-T across the preheat coil and ensures the water in the tubes is well-mixed. The Veteran's Affair's hospital design guide (which is a great reference for all HVAC design) recommends at least 3 fps of water velocity in the tube.

Typically discharge air temperature is controlled by modulating the flow of water, but if you must keep the flow of water constant to maintain space temperature, the next best option is to modulate the inlet water temperature by mixing in some return water.

Pumped Preheat Coil Selection Procedure:

1) Calculated the airside load required.

$$Q = 1.08 * CFM * (T_o - T_i)$$

2) Given the coil height, determine the water flow (GPM_{Coil}) required for 3 fps. ½" tubes are typically spaced 1.25" on-center, 5/8" tubes are typically spaced 1.5" on-center. Your favorite coil manufacturer cut sheets can help.

3) Set your return water temperature, T_R. AHU preheat coils have many different options for rows and fpi, where terminal units don't. This allows the preheat coils to produce lower return water temperatures and help "blend down" the return water temperatures from the terminal units. Typically pick this as 10 °F – 25 °F lower than the desired return water temperature to the heating system so that it can help condensing boiler efficiency and dedicated heat recovery chiller/heat pump lift.

4) Calculate the entering water temperature, Ts, required to meet the airside load.

$$T_S = T_R + (Q \ / 500 * GPM_{Coil})$$

5) Calculate the Bypass flow and HWS main flow necessary to achieve the entering water temperature, T_S. You have two equations:

$$GPM_{Coil} = GPM_{HWSMain} + GPM_{Bypass}$$

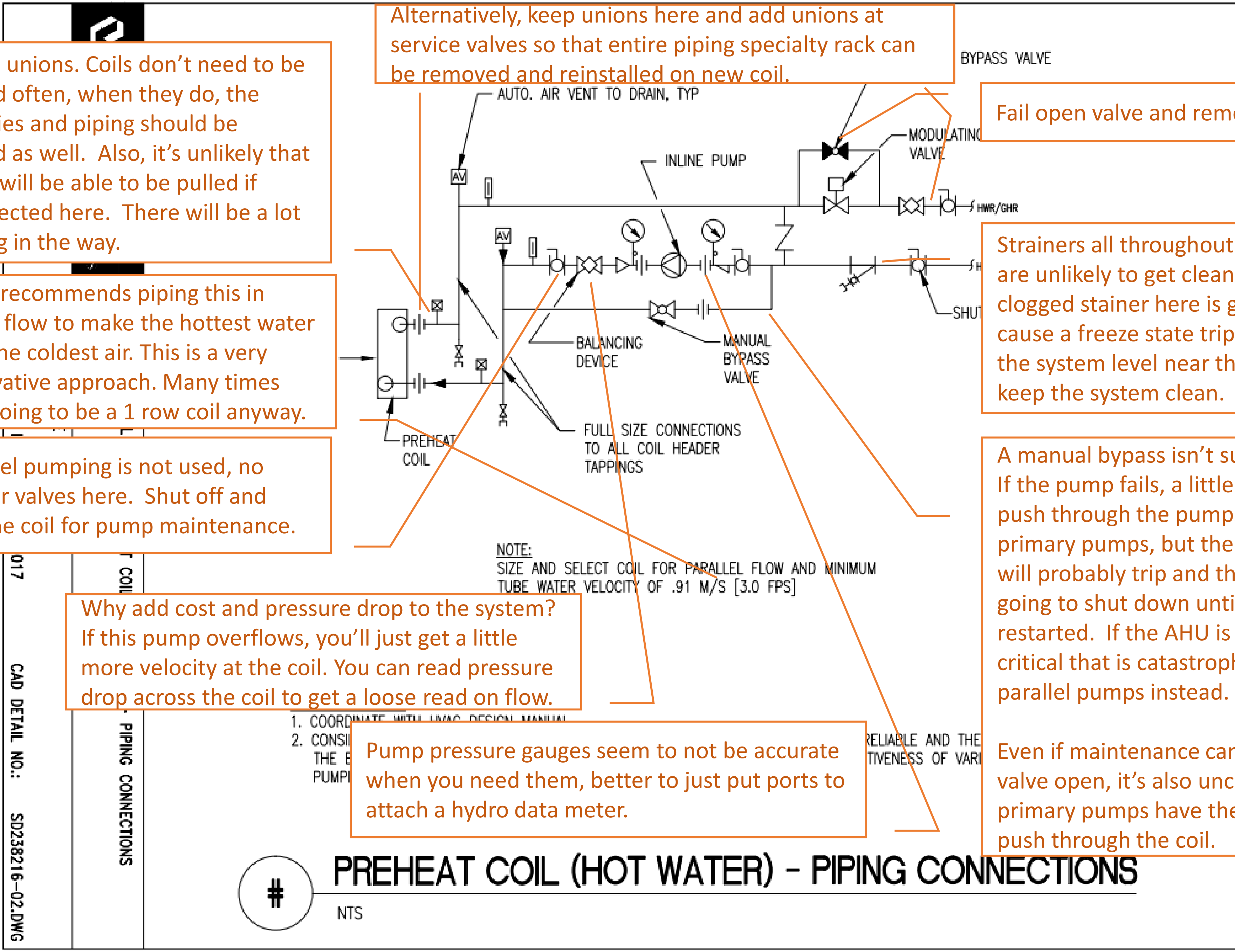
$$GPM_{Coil} * T_S = GPM_{HWSMain} * T_{HWSMain} + GPM_{Bypass} * T_R$$

$$GPM_{HWSMain} = \frac{GPM_{Coil} * (T_S - T_R)}{(T_{HWSMain} - T_R)}$$

6) Select the lowest coil rows and fpi that will achieve the dedicated capacity.

7) Select coil pump for the pressure drop of the bypass system, select the system pumps to overcome the service valves, balance valve, and control valve.

VA Pumped Preheat Coil Mark Up:



Remove unions. Coils don't need to be replaced often, when they do, the specialties and piping should be replaced as well. Also, it's unlikely that the coil will be able to be pulled if disconnected here. There will be a lot of piping in the way.

Alternatively, keep unions here and add unions at service valves so that entire piping specialty rack can be removed and reinstalled on new coil.

Fail open valve and remove bypass.

The VA recommends piping this in parallel flow to make the hottest water touch the coldest air. This is a very conservative approach. Many times this is going to be a 1 row coil anyway.

Strainers all throughout a building are unlikely to get cleaned and a clogged stainer here is going to cause a freeze state trip. Strain at the system level near the pump to keep the system clean.

If parallel pumping is not used, no need for valves here. Shut off and drain the coil for pump maintenance.

A manual bypass isn't super helpful. If the pump fails, a little water will push through the pumps from the primary pumps, but the freeze stat will probably trip and the AHU is going to shut down until manually restarted. If the AHU is mission critical that is catastrophic, put in parallel pumps instead.

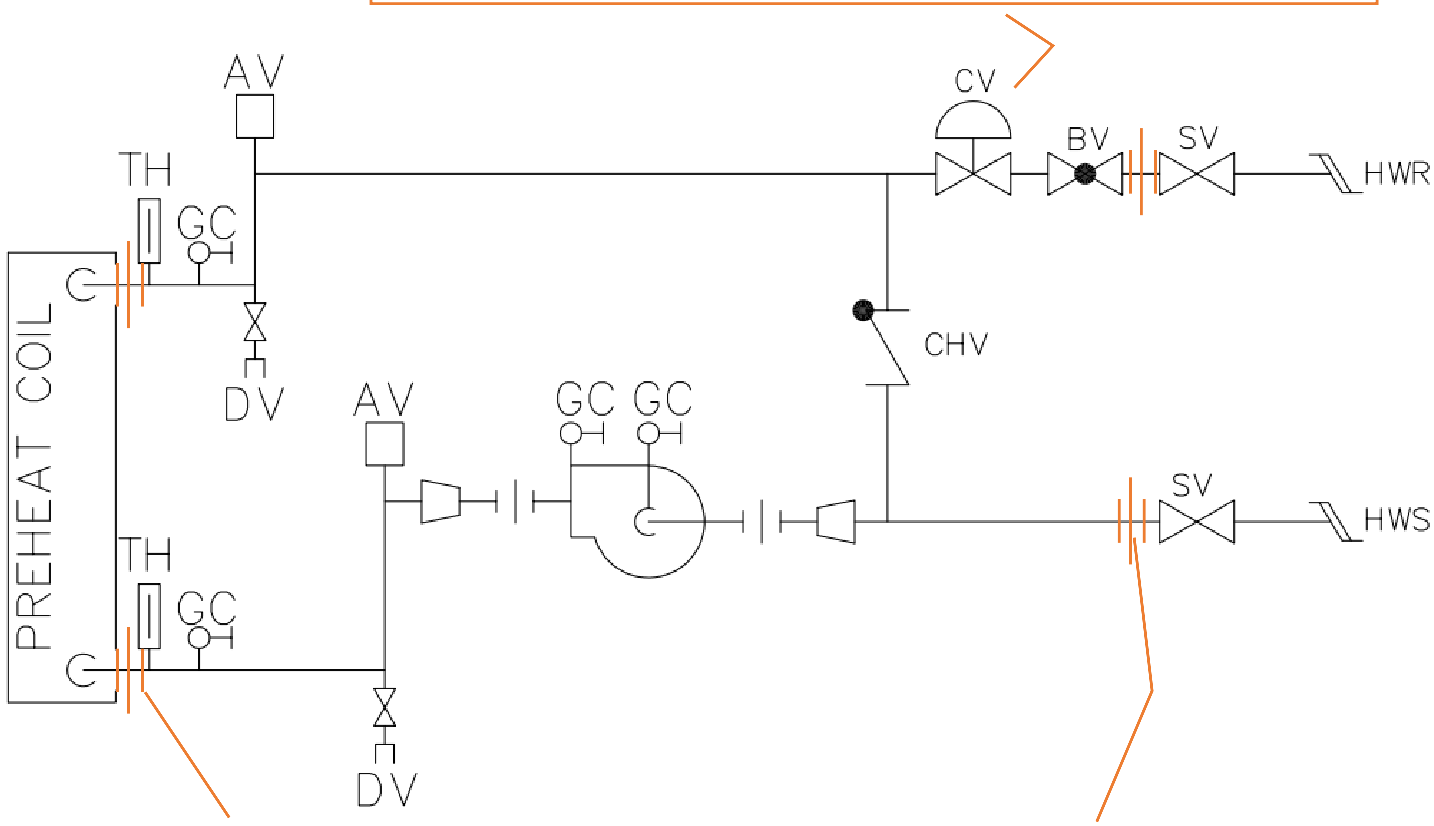
Why add cost and pressure drop to the system? If this pump overflows, you'll just get a little more velocity at the coil. You can read pressure drop across the coil to get a loose read on flow.

Pump pressure gauges seem to not be accurate when you need them, better to just put ports to attach a hydro data meter.

Even if maintenance can get this valve open, it's also unclear if the primary pumps have the capacity to push through the coil.

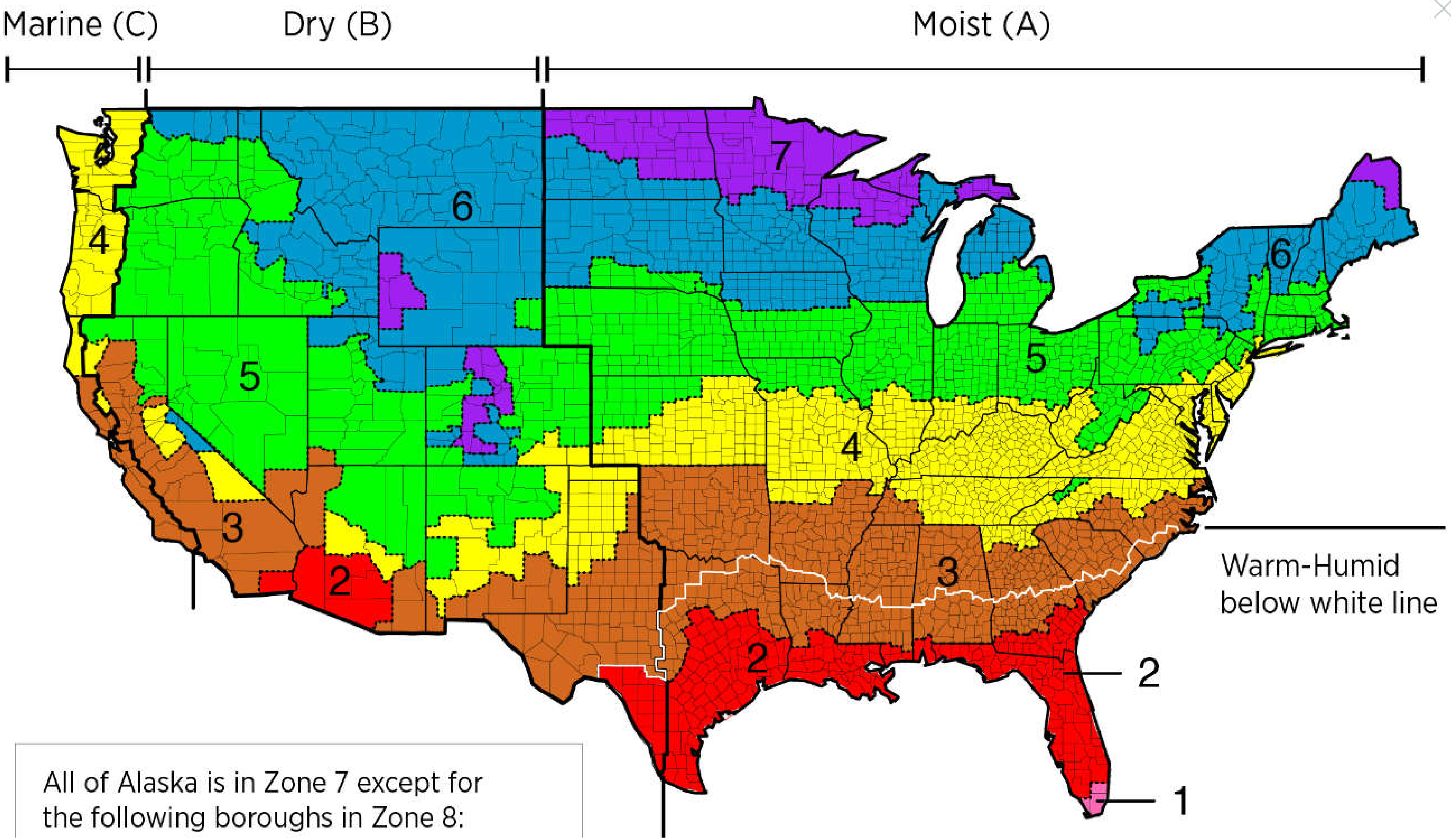
Pumped Preheat Coil (Simplified):

Control Valve modulates to maintain discharge air temperature setpoint (55 °F). If the discharge air temperature falls below the setpoint this valve modulates open to recirculate less water, allow more HWS header water to enter the system and increase the coil entering water temperature. If the discharge air temperature goes above the setpoint, this valve modulates closed to recirculate more water, dropping the inlet temperature.



Adding unions at the coil inlet and shutoff valve may be worthwhile on copper and welded systems. Not necessary on flanged and Vic fittings.

IECC Climate Zones For Reference:



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Humidity:

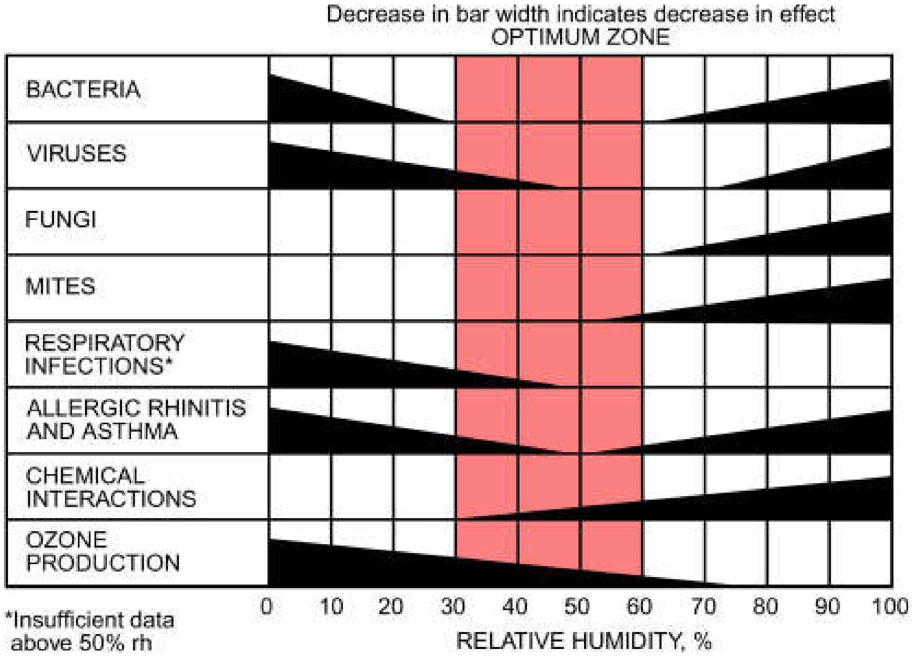


Fig. 1 Optimum Humidity Range for Human Comfort and Health
(Adapted from Sterling et al. 1985)

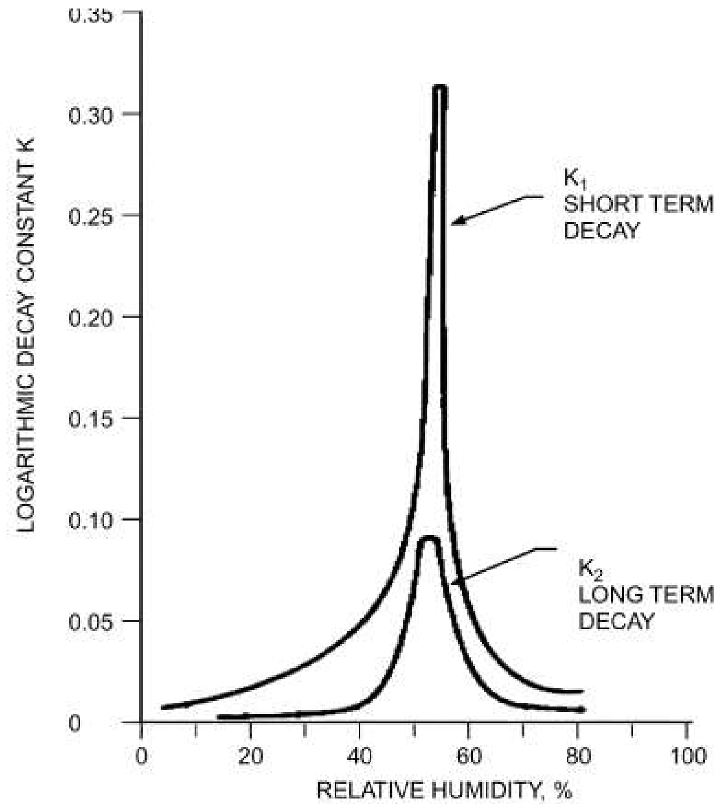


Fig. 2 Mortality of *Pneumococcus* Bacterium
Maximum mortality for airborne *Pneumococci* comes when relative humidity is held at 55% rh. [Adapted from Brundrett (1990), *Criteria for Moisture Control*. Copyright Elsevier © 1990.]

https://www.ashrae.org/file%20library/technical%20resources/covid-19/i-p_s16_ch22humidifiers.pdf

Humidity:

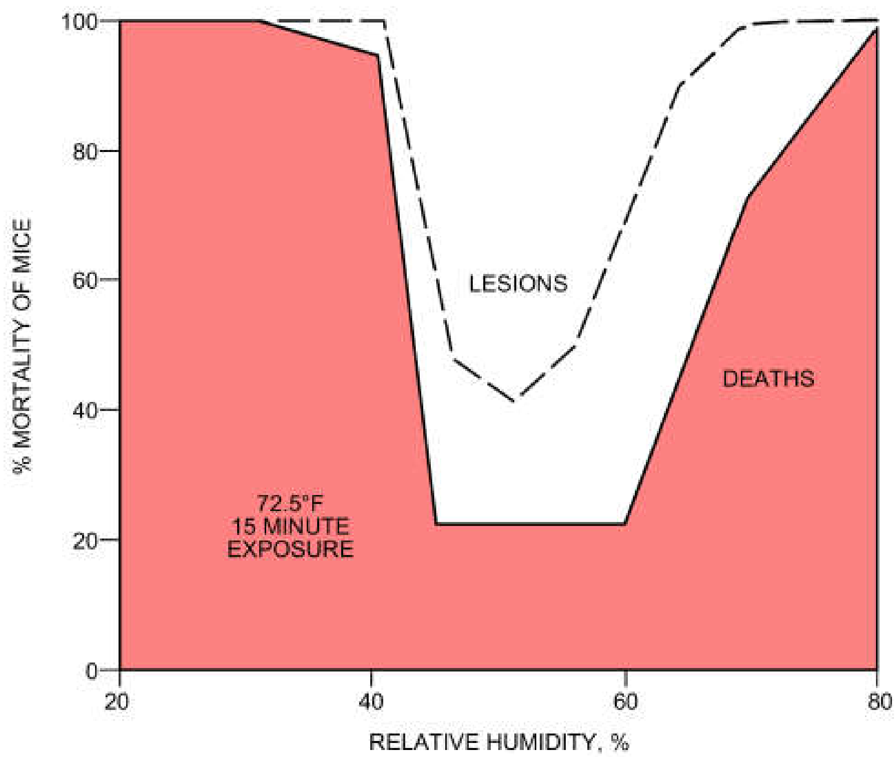


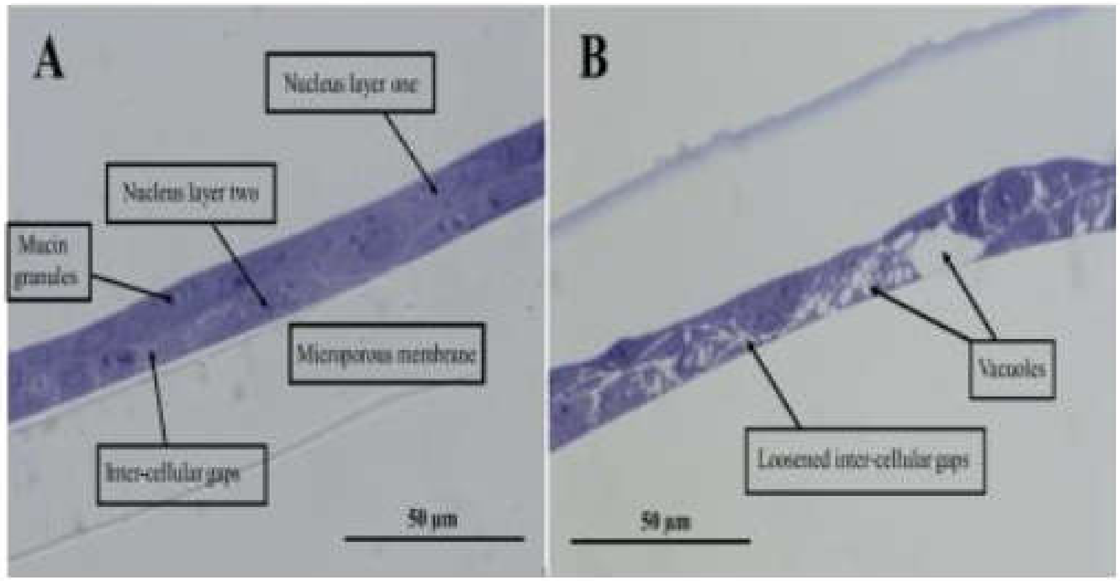
Fig. 3 Mortality in Mice Exposed to Aerosolized Influenza
Note that numbers of deaths and lung lesions were minimized when humidity was held between 40 and 60% rh.
[Adapted from Brundrett (1990), *Criteria for Moisture Control*. Copyright Elsevier © 1990.]

https://www.ashrae.org/file%20library/technical%20resources/covid-19/i-p_s16_ch22humidifiers.pdf

Humidity:

Low RH increases damage to upper airways from particles and gases

Low RH exacerbates harm from inhaled ozone (O_3), and nitrogen dioxide (NO_2) induced loosening of inter-cellular junctions



RH 50% RH 20%

Dr. Huttunen, E. Finland Univ.

Stephanie Taylor, M.D., M. arch. "Quantifying the Impact of IAQ on Occupant Health" 2023 ASHRAE Annual Conference.

Humidity:

Correlation coefficients of pollutant interactions that impact health

	Temp	RH < 40%	RH > 60%	NO_2	O_3	SO_2	PM_1	$PM_{2.5}$	PM_{10}	CO_2	TVOCs	CO
Temp												
RH < 40%	0.20											
RH > 60%	0.10	N/A										
NO_2	0.31	0.19	0.00									
O_3	0.24	0.24	0.00	0.63								
SO_2	0.36	0.31	0.00	0.30	0.28							
PM_1	0.12	0.20	-0.17	0.20	0.30	0.01						
$PM_{2.5}$	0.12	0.15	-0.15	0.19	0.30	0.01	0.19					
PM_{10}	0.10	0.16	0.16	0.18	0.28	0.01	0.94	0.27				
CO_2	0.13	0.19	0.04	0.22	0.21	0.21	0.06	0.07	0.05			
TVOCs	0.60	0.19	0.20	0.31	0.29	0.20	0.37	0.38	0.19	0.11		
CO	0.13	0.17	0.00	0.29	0.22	0.20	0.45	0.12	0.00	0.10	0.32	

"+" = increased harm, "0" = no interaction, "-" protective effect

in development

Stephanie Taylor, M.D., M. arch. “Quantifying the Impact of IAQ on Occupant Health” 2023 ASHRAE Annual Conference.

IECC Fan Power Limitations:

C403.8.1 Allowable fan horsepower (Mandatory).

Each HVAC system having a total fan system motor nameplate horsepower exceeding 5 hp (3.7 kW) at fan system design conditions shall not exceed the allowable *fan system motor nameplate hp* (Option 1) or *fan system bhp* (Option 2) shown in Table C403.8.1(1). This includes supply fans, exhaust fans, return/relief fans, and fan-powered terminal units associated with systems providing heating or cooling capability. Single-zone variable air volume systems shall comply with the constant volume fan power limitation.

Exceptions:

1. Hospital, vivarium and laboratory systems that utilize flow control devices on exhaust or return to maintain space pressure relationships necessary for occupant health and safety or environmental control shall be permitted to use variable volume fan power limitation.
2. Individual exhaust fans with motor nameplate horsepower of 1 hp (0.746 kW) or less are exempt from the allowable fan horsepower requirement.

TABLE C403.8.1(1) FAN POWER LIMITATION

	LIMIT	CONSTANT VOLUME	VARIABLE VOLUME
Option 1: Fan system motor nameplate hp	Allowable nameplate motor hp	$hp \le CFM_S \times 0.0011$	$hp \le CFM_S \times 0.0015$
Option 2: Fan system bhp	Allowable fan system bhp	$bhp \le CFM_S \times 0.00094 + A$	$bhp \le CFM_S \times 0.0013 + A$

For SI: 1 bhp = 735.5 W, 1 hp = 745.5 W, 1 cfm = 0.4719 L/s.

where:

CFM_S = The maximum design supply airflow rate to conditioned spaces served by the system in cubic feet per minute.
hp = The maximum combined motor nameplate horsepower.
bhp = The maximum combined fan brake horsepower.
A = Sum of $[PD \times CFM_D / 4131]$.

where:

PD = Each applicable pressure drop adjustment from Table C403.8.1(2) in. w.c.
CFM_D = The design airflow through each applicable device from Table C403.8.1(2) in cubic feet per minute.

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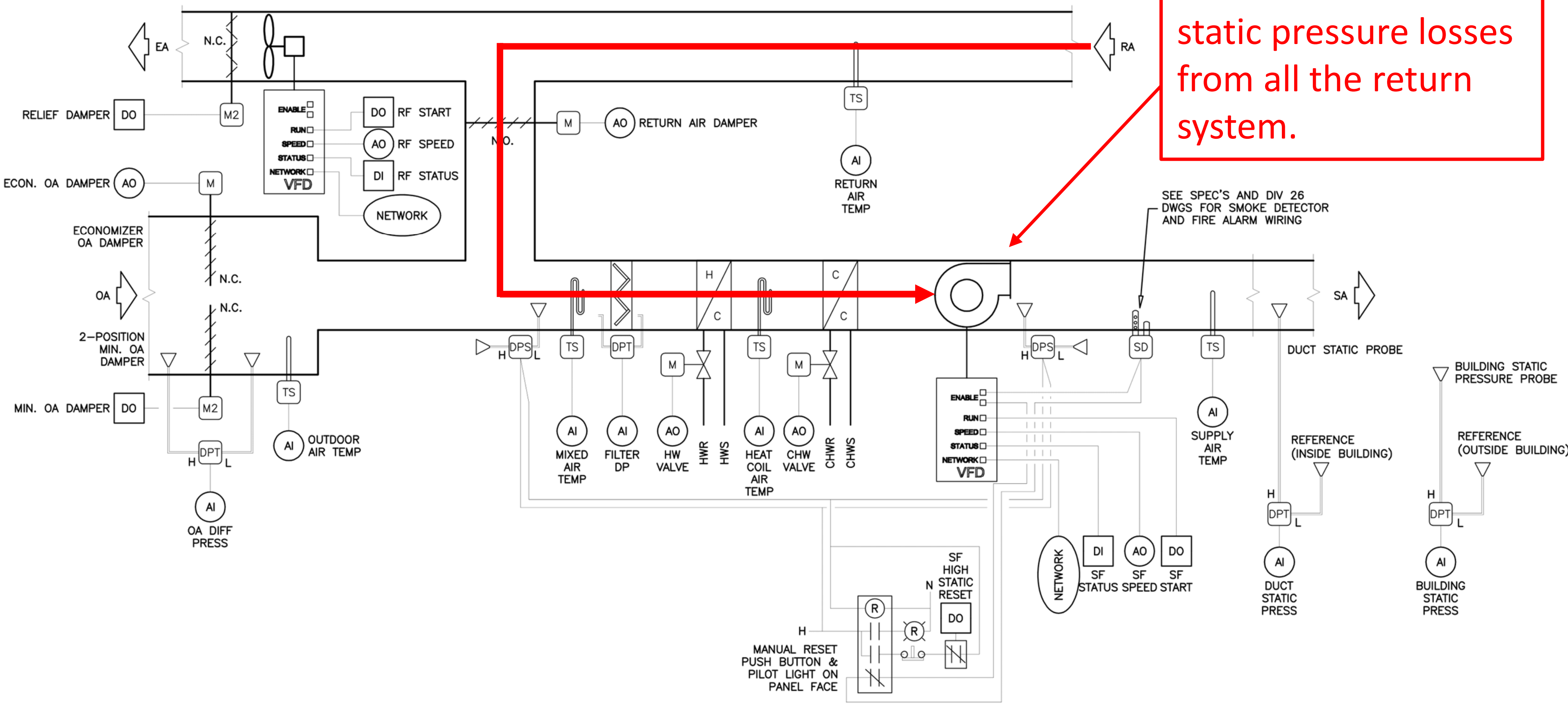
Relief Fan:

Supply fan must be sized to include the static pressure losses from all the return system.



A-10 Multiple zone VAV air handling unit with relief fan and OA measurement station.

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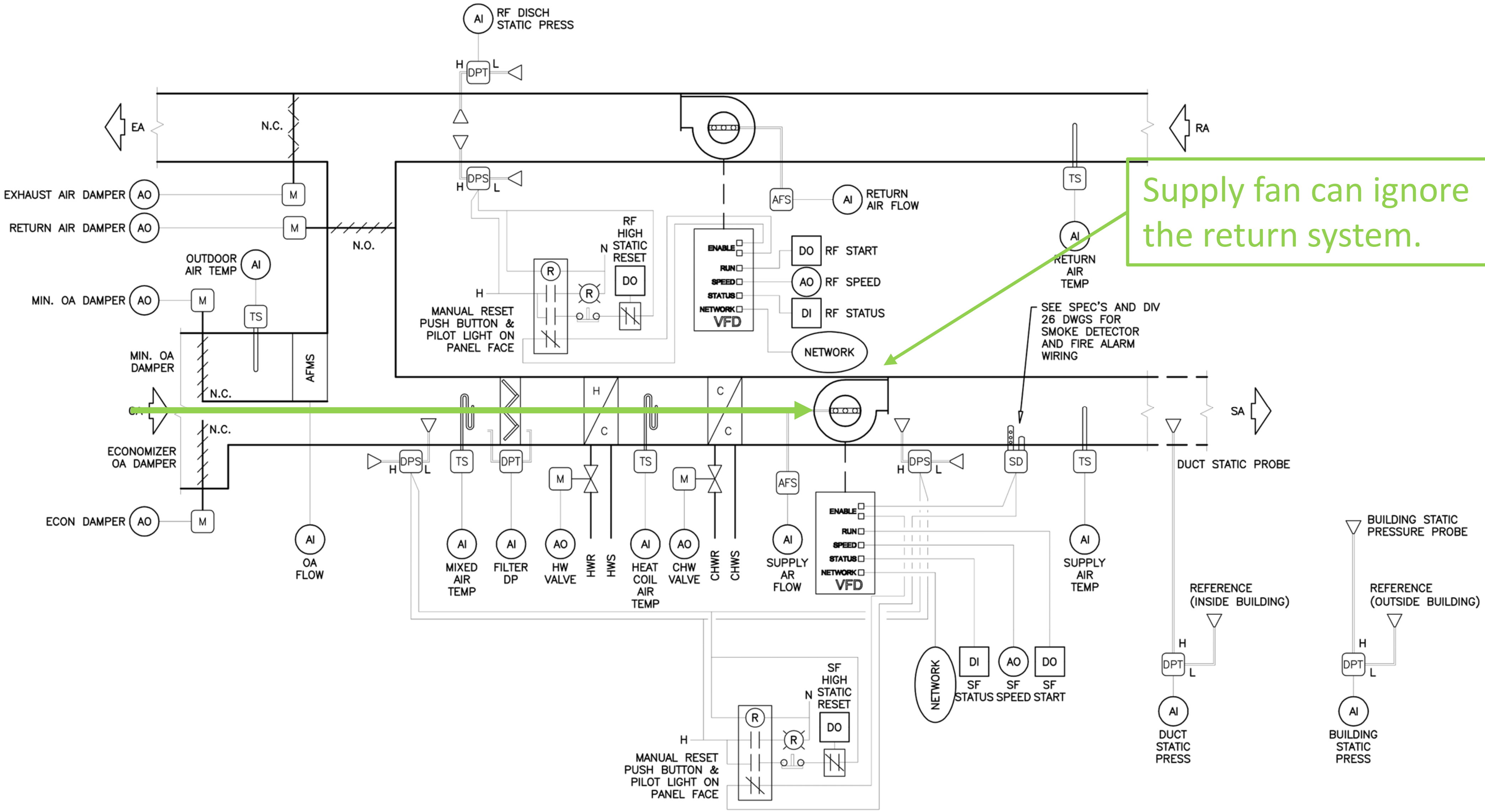
Return Fan:



A-9 Multiple zone VAV air handling unit with return fan and OA measurement station.

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