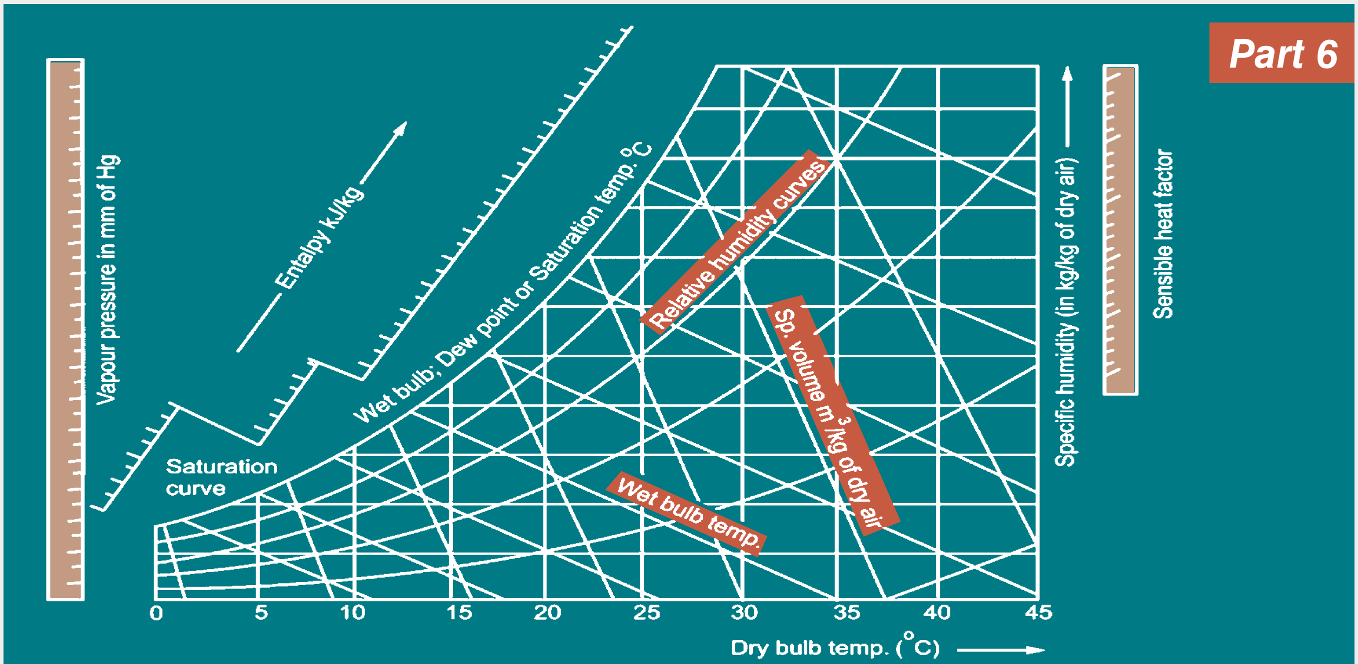

Understanding Psychrometrics

By Ramesh Paranjpey, *Distinguished 50-Year Member, ASHRAE*
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Introduction

In the last part-5, we discussed psychrometric processes in the quarters A, B, C. In this chapter VI, we shall discuss processes in the last quadrant 'D', which is the most important as it covers air conditioning process.

All the processes in the Quarter 'D' deal with cooling and dehumidification. When air that is at a specific dry bulb (Db) and dew point (DP) temperature is cooled below the dew point temperature, a cooling and dehumidification process is obtained.

As soon as the air gets into contact with the cooling coil, which is kept at a temperature that is lower than the air's dew point temperature, the air's dry bulb temperature begins to decrease. The process of cooling continues and at some point, air approaches and reaches the dew point temperature. Because

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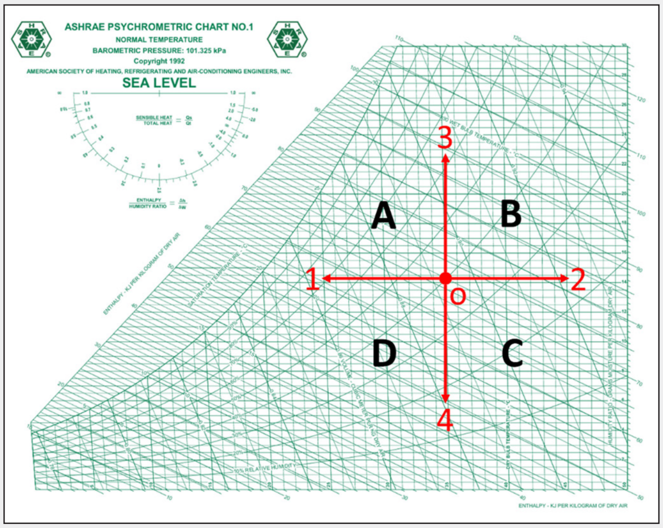

Figure 1: Psychrometric quadrants

About the Authors

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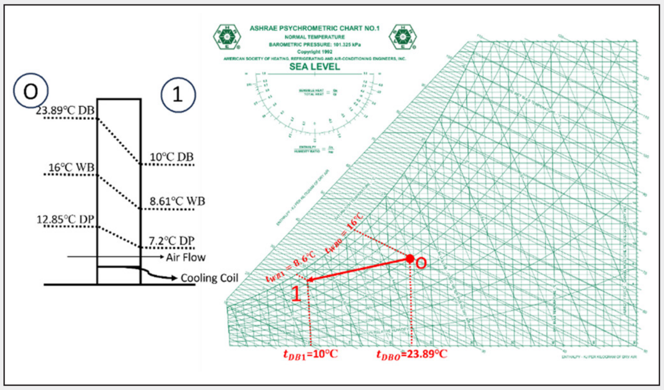


Figure 2: Process over cooling coil

of this, dew will form on the surface of the coil, and the moisture content of the air will decrease, which will cause the humidity level to decrease as well.

During this process of cooling and dehumidifying the air, the dry bulb, the wet bulb, and the dew point the temperature, all will reduce to lower values. Similarly, both the sensible heat and the latent heat of the air will lose some of their intensity, which will ultimately result in a lower overall enthalpy for the air.

The cooling coil in air conditioning systems is cooled by either the refrigerant or the chilled water. On the psychrometric chart, the process of cooling and dehumidification is depicted as a straight angular line. The value of the DB temperature is used as the starting point for the line, which then continues downward and to the left as shown in *Figure 2*.

The process of cooling and dehumidification is basically the air conditioning application. The dry bulb temperature and the moisture content is gradually reduced as the air travels over the cooling coil rows till it reaches ADP as shown in *Figure 3*. If the refrigerant or water temperature in the cooling coil is lower than ADP, the condensation starts as ability of air to hold moisture diminishes and the extra water is then visible in the form of condensation or water starts dripping in the drip tray.

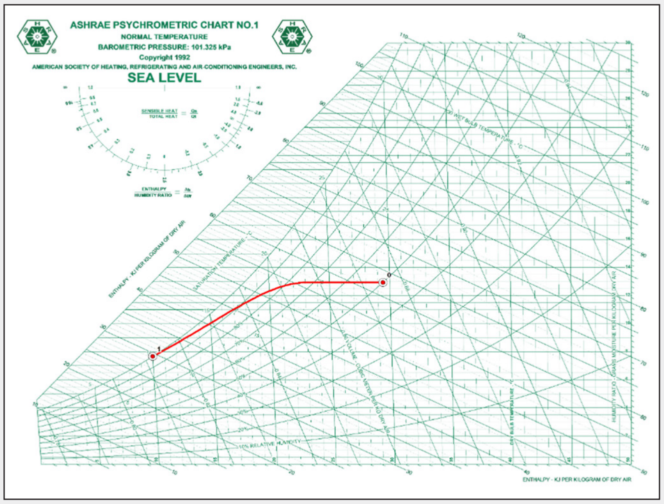


Figure 3: Moist air cooling and dehumidification

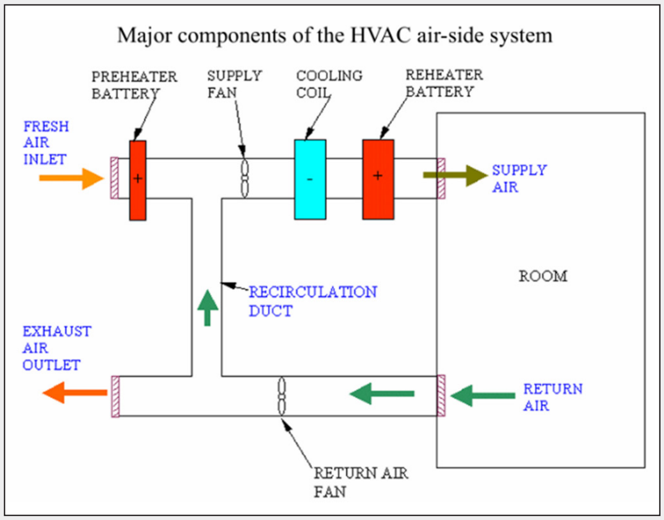


Figure 4: Main components of air side HVAC system

The main function of the psychrometric analysis of an air-conditioning system is to determine the supply air dry bulb and wet bulb temperatures as also the volume flow rates of air to be circulated to achieve required/designed comfort temperature and humidity.

Figure 4 indicates in general, the arrangement of air conditioning system.

Here readers should note that usually Recirculation air quantity = Supply air quantity- Fresh air quantity

HVAC engineers use Psychrometrics to translate the knowledge of heating or cooling loads (which are in kW or TR) into volume flow rates (in m^3/s or CFM) for the air to be circulated into the system.

The cooling load calculation formulae are given below both in FPS as well as in SI units for the ready reference.

1. Sensible Heat in FPS units: -

$$Q(\mathrm{Btu/h}) = m.C_p.\Delta T$$

Where: -

m in lb/h
C_p in Btu/lb. °F
ΔT in °F

$$Q = \frac{\mathrm{cfm} * 60\ \mathrm{min}}{\mathrm{hr}} * \mathrm{sp.\ density} * C_p * \Delta T$$

$$Q = \mathrm{cfm} * 60 * 0.075 * 0.24 * \Delta T$$

$$Q = 1.08 * \mathrm{cfm} * \Delta T$$

If we include allowance for 0.010lb(70gr) of moisture per pound of dry air since it is typical condition of air entering air side of cooling coil/dehumidifying device, then the equation becomes

$$Q = \mathrm{cfm} * 60 * 0.075(0.24 + 0.44(0.010) * 0.24 * \Delta T$$

$$Q = \mathrm{cfm} * 60 * 0.075 * 0.2444 * \Delta T$$

$$Q = 1.1 * \mathrm{cfm} * \Delta T$$

Sp. density in lb./ft³ = 1/sp. volume of air at 70°F & 50% RH = 1/13.5 = 0.075lb/ft³

Specific heat of air = 0.24 Btu/lb.°F

2. Latent Heat: in FPS Units

$$Q = 0.68 * \frac{\Delta\,\text{grains}}{\text{lb}} * \text{cfm}$$

(0.68 is derived as = $60 * 0.07 * \frac{1076}{7000}$ Btu/h)

1 lb. of air is equal to 7000 drains of moisture

Latent heat of vaporization of water is 1076 Btu/lb. at 32°F or 0°C

3. Total heat: in FPS units

$$Q = \text{Sp. density} * \frac{\text{cfm}*60\ \text{min}}{\text{h}} * \Delta H$$

$$Q = 0.075 * 60 * \text{cfm} * \Delta H$$

$$\text{Btu/h} = 4.5 * \text{cfm} * \Delta H$$

Sensible Heat in SI units

1. Sensible heat -

$$Q(W) = m.C_p.\Delta T$$

$$Q(W) = 1.204\left(\frac{kg}{m^3}\right) * \frac{l}{s} * 1.0216\left(\frac{kJ}{kg.K}\right) * \Delta T(K)$$

$$Q = 1.23 * \frac{l}{s} * \Delta T(K)$$

(Air density = 1.204 kg/m³)

(Specific heat = 1.0216 kJ/kg-K)

Latent Heat in SI units

$$Q(W) = 1.204\left(\frac{kg}{m^3}\right) * \frac{l}{s} * 2500\left(\frac{kJ}{kg}\right) * \Delta w\left(\frac{g}{kg_{da}}\right)$$

$$Q(W) = 3.010 * \frac{l}{s} * \Delta w$$

$$Q(W) = 3\ * \frac{l}{s} * \Delta w$$

1 l/s = 0.001 m³/s

Latent heat of vaporization of water = 2500 kJ/kg at 0°C

Total Heat in SI units

$$Q(kW) = 1.204\left(\frac{kg}{m^3}\right) * \frac{m3}{s} * \Delta h\left(\frac{kJ}{kg}\right)$$

$$Q = 1.2 * \frac{l}{s} * \Delta h$$

(Air density = 1.204 kg/m³)

Δh = kJ/kg

$$Q(kW) = 1.204 * \frac{m^3}{s} * \Delta h$$

Outside Air Mixing

Barring applications involving total sensible cooling loads only, in most of the other applications of comfort air conditioning, outside air needs to be introduced in the space for ventilation and needs to be cooled by passing it over the dehumidifying equipment i.e. the cooling coil. The air quantity required for ventilation depends on the number of people occupying the

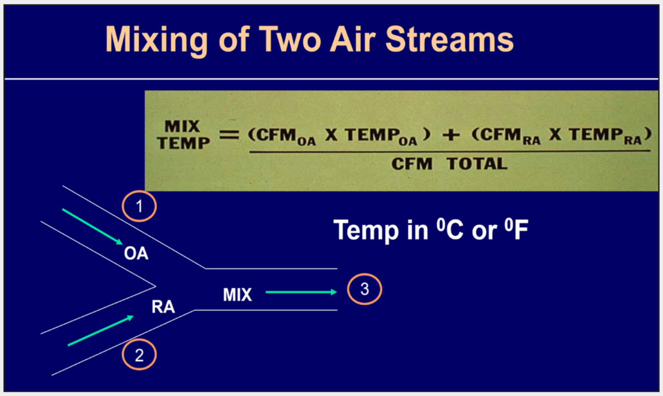


Figure 5: Mixing of outside air and return air

space, and the extent of air contamination on account of food, smoking or any other processes.

Since the outside air's dry bulb temperature and humidity is changing constantly when outside air is mixed with return air, the mixture condition would also have varying conditions.

The final condition of mixed air would depend upon the quantity as well as the temperature and moisture content present in the air at a given time. Mixing outside air with return air (room air) can be plotted on a psychrometric chart as shown in *Figure 6*, by connecting the room conditions with outside conditions by a straight line.

The air mixture conditions would then fall on this straight line and can be determined by knowing the quantities of outside air and return air. The mixture point would be nearer to the larger quantity in the proportion to the air quantities.

For example, if we mix 1500 L/s of outside air at 35°C DBT/28°C WBT (point B) with 4500 L/s of recirculated air at 25°C and 50%

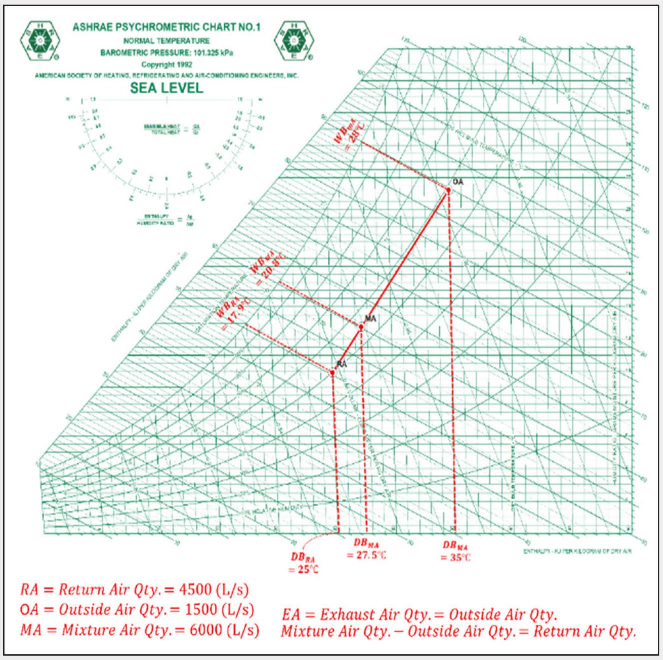


Figure 6: Psychrometric process of mixing of air

R.H. (point A), the mixture point C would be closer to the point A, because of the greater amount of recirculated air. Since the outside air quantity is ¼th the total quantity (6000 L/s) if we divide the straight-line AB in 4 parts, the mixture point would end up at ¼th distance from the recirculated air point. The mixture temperature at C can then be read as 27.5°C and all other conditions of mixture air like relative humidity, wet bulb temperature and other properties can be found easily once the mixture point is located on the chart.

The mixture condition can also be calculated as under.

Use of the psychrometric chart, however, gives fairly accurate values for all practical purposes.

$$T_{mix} = \frac{(1500 * 35) + (4500 * 25)}{6000}$$

$$T_{mix} = 27.5 \text{ Deg C}$$

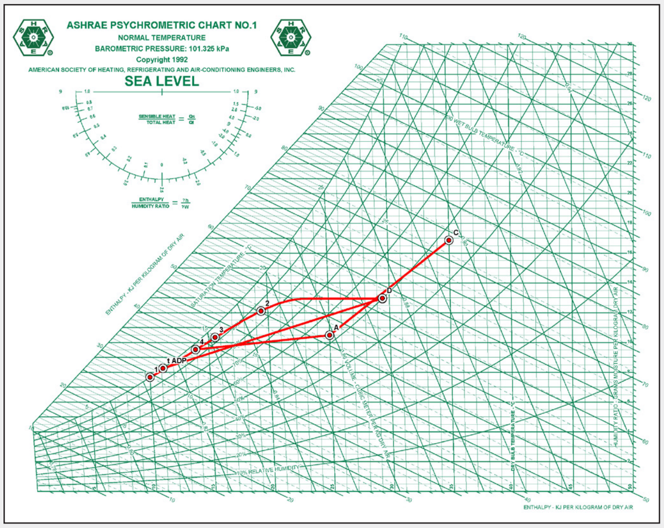


Figure 7: Process of cooling and dehumidification

Figure 7 indicates air travel of mixed air as it travels over the cooling coil, the slope becomes deeper as the number of coil rows increase. This part we will discuss in more details in our next chapter.

Required Air Quantity (RSHF and GSHF)

Significance of RSHF, GSHF, ERSHF

1. Room Sensible Heat Factor (RSHF) indicates condition of air as it moves through air condition hall.
2. Gross Sensible Heat Factor (GSHF) indicates condition of air as it moves through cooling coil.
3. Effective Sensible Heat Factor (ESHF)

To relate BF and ADP of coil to load calculations, ESHF term was developed. ESHF is interwoven with BF and ADP and thus greatly simplifies the calculations of required air quantity and coil selection.

We have now seen how to arrive at conditions of air entering the apparatus or a cooling coil, which is a mixture of outside air and return air. It is now essential to determine the air quantity required to offset, not only the room load comprising of Room Sensible Heat (RSH) plus Room Latent Heat (RLH), but also it should be able to meet total load including load imposed on the apparatus by outdoor air.

We need to, therefore, differentiate between RSHF and GSHF. These factors can be calculated once we calculate room sensible heat (RSH) and room latent heat (RLH) load as also gross sensible heat (GSH) and gross latent heat (GLH) load. Gross total heat is the sum of gross sensible heat (GSH) + gross latent heat (GLH).

$$RSHF = \frac{RSH}{RSH + RLH}$$

$$RSHF = \frac{RSH}{RTH}$$

$$GSHF = \frac{GSH}{GSH + GLH}$$

$$GTH = GSH + GLH$$

$$GSHF = \frac{TSH}{GTH}$$

All these are mapped on the psychrometric chart in *Figure 8*.

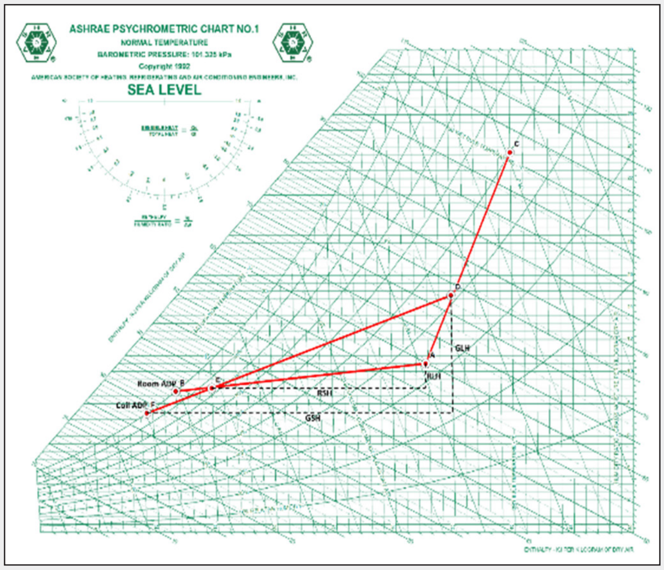


Figure 8: RSH, GSH, RLH, GLH

Once the factors are calculated, we can then draw a line parallel to the line connecting standard condition point of 25°C, 50% R.H. and the sensible factor (RSHF) calculated as above, from room condition 'A' to intersect saturation line at B and from mixture condition 'D' to intersect the saturation line at F. The points at which these lines would intersect the saturation line are termed as room Apparatus Dew Point ADP (B) and coil ADP(F).

It is essential that the supply air condition should always lie on line AB representing RSHF if it is to satisfy room design conditions and, the point E represents supply air condition.

The figure does not show the effect of heat gain due to fan and duct heat gain and duct losses. These also must be considered while establishing the cooling load and for determining supply air quantity. However, to simplify understanding, we have not considered these in this chapter.

The air quantity (L/s room) required to meet room load would then be calculated as:

$$Lps_{room} = \frac{RSH(W)}{1.23 * \Delta T(T_D - T_E)}$$

$$CFM_{room} = \frac{RSH}{1.08 * \Delta T(T_D - T_E)}$$

To arrive at point 'E', one must go through a trial-and-error method since the quantity of air needed to satisfy both room as well as total load must be the same. If the air were to leave fully dehumidified, i.e., at point 'B', the matter would have been simpler, but the coil has its own inefficiency, which is an unknown factor, 7 it is called as coil bypass factor. Another unknown factor is the percentage of outside air to total air, which does not get established, till the amount of air to be handled by the apparatus is arrived at. Normally, the room load and RSHF ratio at full load under a given set of conditions will remain constant.

However, the GSHF ratio, either increases or decreases as the outdoor conditions and the quantities change. It is evident, if the point 'E' is nearer to point 'B' (saturated condition), the quantity of air required to be circulated to meet the room cooling load decreases and as the point 'E' moves away from 'B', on the line BA, the quantity of air required increases.

Bypass Factor

Cooling or heating coil cannot not have 100% efficiency. Since the air is cooled by the cooling coil, the coil efficiency needs to be considered and it leads us to another terminology, called Bypass Factor (BF). Air leaving the cooling coil will not always be saturated and will leave at a condition where some moisture will condense over the coil and thus, both sensible and latent heat removal takes place. It is difficult to calculate mathematically the exact coil leaving conditions of air and hence for simplifying calculations and understanding the subject, it is assumed that a certain percentage of air while passing over the coil is not affected at all and leaves the coil at the same conditions at which it has entered the coil.

This quantity of air to the total quantity circulated is termed as coil bypass factor (BF). It is also assumed that the remaining quantity of air (1- BF) reaches saturation point F and is therefore, called as dehumidified air, (l- BF) factor is also referred as the "contact factor." These are marked in *Figure 9*. (Ref: Fundamentals of Psychrometrics-Carrier-page24).

$$\text{BF (Coil Bypass)Factor} = \frac{T_E - T_F}{T_D - T_F}$$

$$\text{Contact Factor} = \frac{T_D - T_E}{T_D - T_F}$$

$$\text{Contact Factor} = 1 - \text{BF}$$

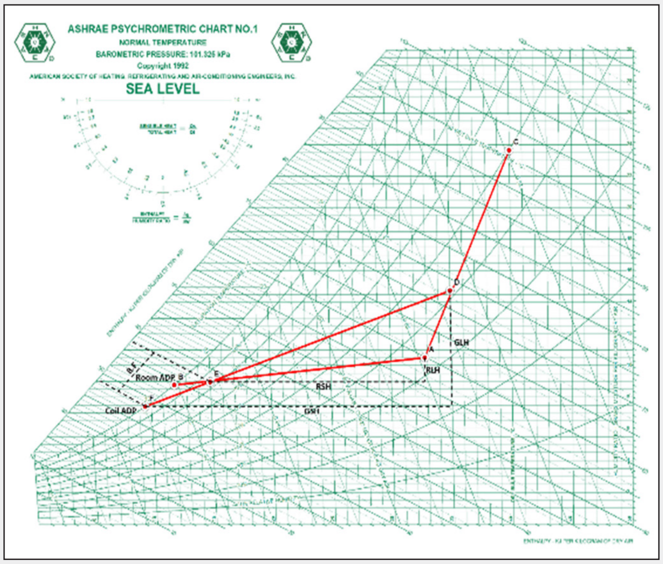


Figure 9: Bypass Factor (marked on extreme left)

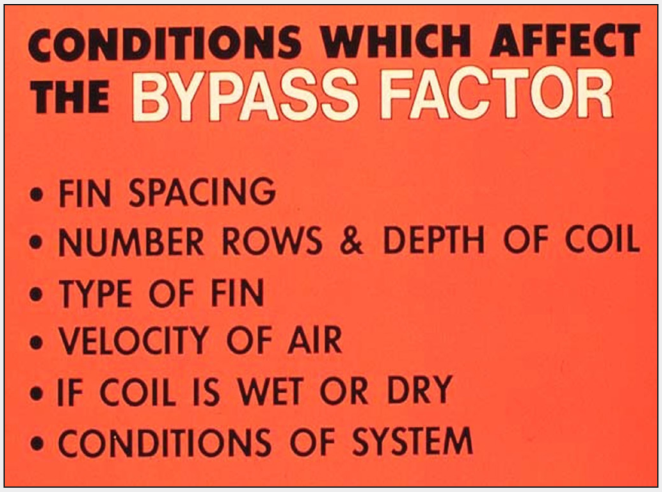


Figure 10: Conditions affecting Bypass Factor

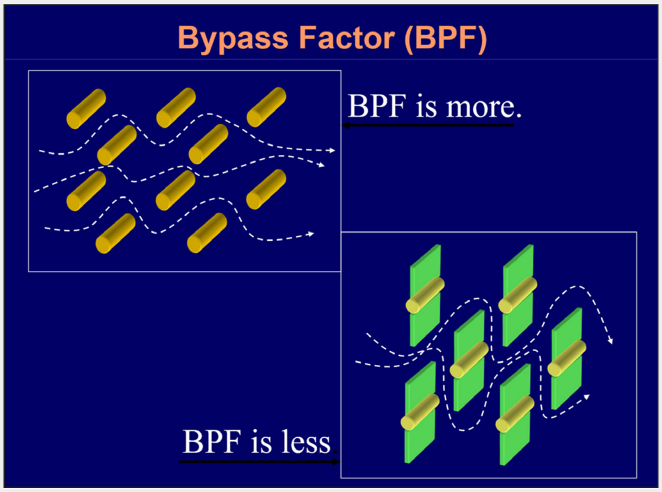
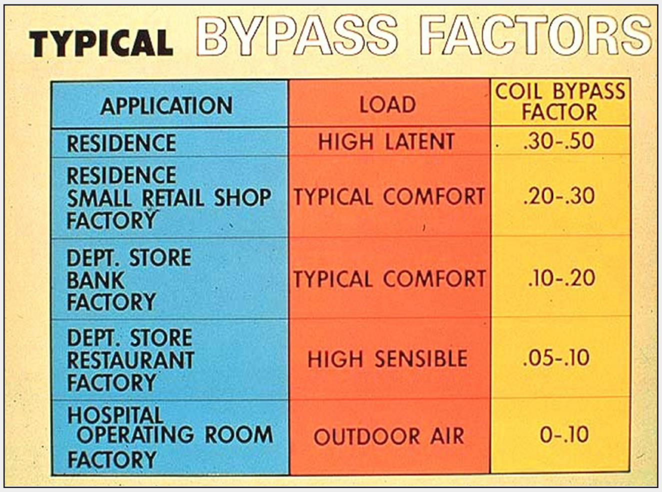


Figure 11: Bypass factor



TYPICAL BYPASS FACTORS

APPLICATION	LOAD	COIL BYPASS FACTOR
RESIDENCE	HIGH LATENT	.30-.50
RESIDENCE SMALL RETAIL SHOP FACTORY	TYPICAL COMFORT	.20-.30
DEPT. STORE BANK FACTORY	TYPICAL COMFORT	.10-.20
DEPT. STORE RESTAURANT FACTORY	HIGH SENSIBLE	.05-.10
HOSPITAL OPERATING ROOM FACTORY	OUTDOOR AIR	0-.10

Figure 12: Typical Bypass Factors

Apparatus Dew Point Temperature

When an air-vapor mixture is cooled at constant pressure, the temperature at which the vapor becomes saturated and moisture begins to condense is called dew point temperature of the air. Since the temperature of saturated vapor is dependent only on absolute pressure, the dew point temperature is the saturation temperature corresponding to the partial pressure of the vapor in the air-vapor mixture. The normal dry bulb temperature of superheated vapor is always greater than dew point temperature. At saturation, the dry bulb, wet bulb and dew point temperatures are equal.

The coil surface temperature varies throughout the surface as air comes in contact with it. For simplicity, the 'effective surface temperature' term has been established. This is the temperature, which would produce the identical air leaving conditions. It can be seen from *Figure 13* that the temperature condition of air drops when it is flowing over the coil and some common reference point, therefore, is needed. This reference point is the effective surface temperature and it helps us to calculate required air quantity

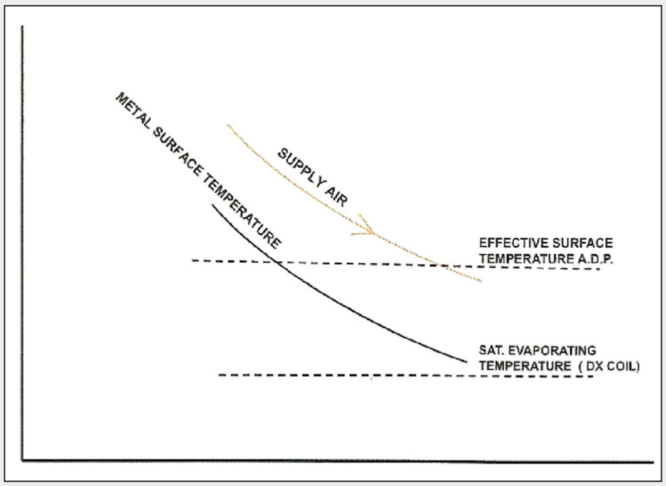


Figure 13: Apparatus Dew Point (ADP)

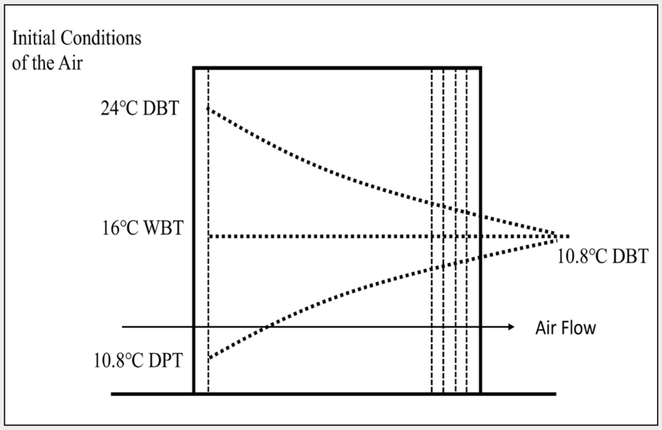


Figure 14: Air processed on a humidifier

and temperature reference while selecting the coil. This is the temperature, at which the GSHF line crosses the saturation line on the psychrometric chart and thus, is also known as Apparatus Dew Point or ADP.

Having understood the terms RSHF, GSHF, BF and ADP, we shall now look at the method used for determining air quantity without resorting to a trial-and-error method.

When unsaturated air is passed through a humidifier, in which the spray water is continuously recirculated, the air will become saturated at the wet bulb temperature of the entering air as represented in *Figure 14*. At saturation the dry bulb, wet bulb and the dew point temperatures are the same.

Effective Sensible Heat Load Factor

The line AB represents the RSHF established by room loads. Point 'C' represents the outside air condition and point 'D' represents the mixture of outside air and return air entering the dehumidifier coil. The slope of the line DME shall be such that its intersection with line AB at M shall divide the line DMe in the ratio of coil bypass factor. In other words, EM/ED is equal to bypass factor.

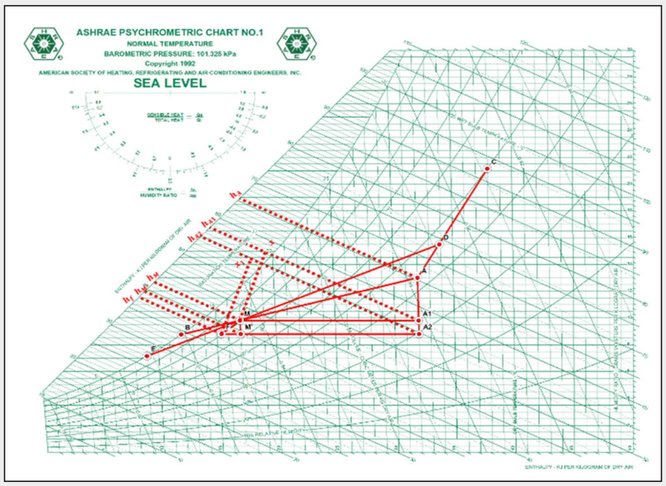


Figure 15: Effective sensible heat load factor

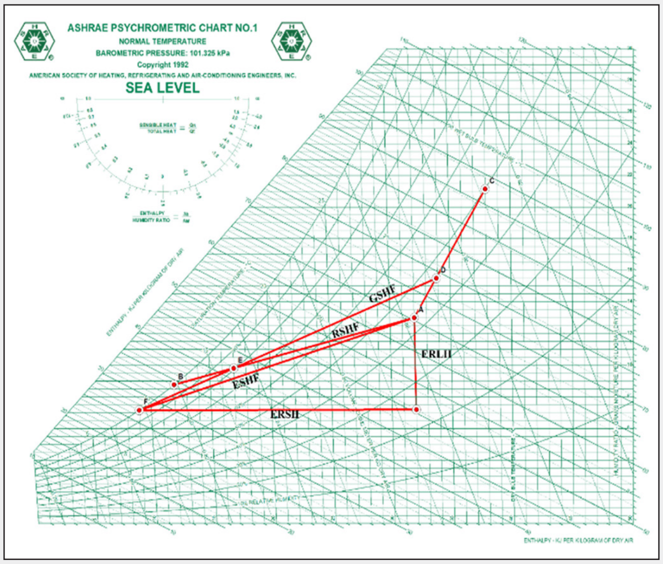


Figure 16: Effective SHF

Line Ae may now be drawn. Also, line MF is drawn parallel to the line AC, which makes triangle EMF similar to triangle EDA. Therefore, the corresponding sides of these equilateral triangles are related in the ratio of bypass factor. It may be noted that since AD represents the OA air load, FM=AD(BF), which is the total load charged to the room due to outside air bypass effects at the coil. It should also be noted that 'FM' represents the sensible heat portion of the bypass load and M'M represents the latent heat portion.

When the various enthalpy lines are drawn as shown in *Figure 15*, it is possible to draw line FX_1 and MX to form triangles M, X, A_1 and F, X_1, A_2 which are similar.

The effective air rise FA2 may be used to determine the dehumidified air quantity m^3 when it is used with the room sensible load which includes the sensible OA by pass load. The same value of dehumidified air will result if the actual air rise MA_1 is used in conjunction with the sensible load only.

Effective Sensible Heat (ESH)considers, room sensible heat, fan motor heat gain, supply duct losses and supply air heat gain and sensible heat due to BF fraction of outside air. It does not include the outside air load imposed on the coil that gets completely dehumidified at dew point i.e., air heat load of (1-BF) quantity.

Similarly, Effective Latent Heat factor would be termed as (ELHF).

$$\mathbf{ESHF} = \frac{\mathbf{ERSH}}{\mathbf{ERSH} + \mathbf{ERLH}} = \frac{\mathbf{ERSH}}{\mathbf{ERTH}}$$

The Effective Sensible Heat Factor (ESHF) is the ratio of Effective Room Sensible Heat (ERSH) to Effective Room Sensible Heat Plus Effective Room Latent Heat (ERLH) or Effective Room Total Heat (ERTH).

The effective sensible heat factor is (ESHF) represented on the psychrometric chart by connecting points F and A as shown in *Figure 16*.

This is a simplified method of establishing required air quantity (L/s), and then it is not necessary to calculate RSHF or GSHF, since the air quantity using ESHF, BE and ADP results in the same air quantity.

$$\text{LPS (dehumidified)} = \frac{\text{ERSH}}{1.23(T_A - T_F) * (1 - \text{BF})}$$

$$\text{CFM (dehumidified)} = \frac{\text{ERSH}}{1.08(T_A - T_F) * (1 - \text{BF})}$$

This calculated dehumidified air quantity, simultaneously offsets both rooms heat load as well as total heat load, including outdoor air loads. The quantity of air, thus, established is called effective since its coil leaving temperature and humidity level is effectively lower than the room conditions required to absorb losses in supply air path, as well as to take care of room load, and the factor is therefore called Effective Sensible Heat Factor (ESHF).

The quantity of air can also be calculated from ERLH or from total heat load ERTH, by using applicable formulas already discussed. In the next part VII-we shall discuss Psychrometrics applied to cooling towers, evaporative condensers and for air cooling coils.