DA09 Section 8

APPLIED PSYCHROMETRICS
8. Applied Psychrometrics

8.1. Section introduction

The preceding Sections contain the practical data to properly evaluate the heating and cooling loads. They also recommend outdoor air quantities for ventilation purposes.

This Section describes practical psychrometrics as applied to plant selection. The 'Carrier Simplified' Method is described in this section. Other psychrometric methods such as the TRANE coil curve and iterative methods also exist. The section is divided into three parts:

1. **Description of terms, processes and factors** – as encountered in normal air conditioning applications.

2. **Air conditioning plant** – factors affecting common processes and the effect these factors have on selection of air conditioning equipment.

3. **Psychrometrics of partial load control** – the effect of partial load on equipment selection and on the common processes.

Psychrometrics is the science involving thermodynamic properties of moist air and the effect of atmospheric moisture on materials and human comfort. As it applies to this Section, the definition must be broadened to include the method of controlling the thermal properties of moist air.

To help recognise terms, factors and processes described in this Section, a brief definition of psychrometrics is offered at this point, along with an illustration and definition of terms appearing on a standard psychrometric chart (Fig. 35).

The dry-bulb, wet-bulb, and dewpoint temperatures and the relative humidity are so inter-related that, if two properties are known, all other properties shown may be determined. When air is saturated, dry-bulb, wet-bulb, and dewpoint temperatures are all equal.
8.2. Air conditioning processes

Fig. 36 shows a typical air conditioning process traced on a psychrometric chart. Outdoor air (2)* is mixed with return air from the room (1) and enters the plant (3). Air flows through the conditioning plant (3-4) and is supplied through the space (4). The air supplied to the space moves along line (4-1) as it picks up room loads, and the cycle is repeated. Normally most of the air supplied to the space by the air conditioning system is returned to the conditioning plant. There, it is mixed with outdoor air required for ventilation. The mixture then passes through the plant where heat and moisture are added or removed, as required, to maintain the desired conditions.

The selection of proper equipment to accomplish this conditioning and to control the thermodynamic properties of the air depends upon a variety of elements. However, only those which affect the psychrometric properties of air will be discussed in this Section. These elements are: room sensible heat factor (RSHF)†, grand sensible heat factor (GSHF), effective surface temperature ($t_{ES}$), bypass factor (BF), and effective sensible heat factor (ESHF).

8.3. Description of terms, processes and factors

8.3.1. Common Psychrometric terms

**Dry-bulb Temperature** – The temperature of air as registered by an ordinary thermometer.

**Wet-bulb Temperature** – The temperature registered by a thermometer whose bulb is covered by a wetted wick and exposed to a current of rapidly moving air.

**Dewpoint Temperature** – The temperature at which condensation of moisture begins when the air is cooled.

**Relative Humidity** – Ratio of actual water vapour pressure of the air to the saturated water vapour pressure of the air at the same temperature.

**Specific Humidity or Moisture Content** – The weight of water vapour in grams of moisture per kilogram of dry air.

**Enthalpy** – A thermal property indicating the quantity of heat in the air above an arbitrary datum, in kilojoules per kilogram of dry air. The datum for dry air is 0°C and, for the moisture content, 0°C water.

**Enthalpy Deviation** – Enthalpy indicated above, for any given condition, is the enthalpy of saturation. It should be corrected by the enthalpy deviation due to the air not being in a saturated state. Enthalpy

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* One italic number in parentheses represents a point, and two italic numbers in parentheses represent a line, plotted on the accompanying psychrometric chart examples.

† Refer to the end of this section for a description of all abbreviations and symbols used in this section.
deviation is in kilojoules per kilogram of dry air. Enthalpy deviation is applied where extreme accuracy is required; however, on normal air conditioning estimates it is omitted.

**Specific Volume** – The cubic metre of the mixture per kilogram of dry air.

**Sensible Heat Factor** – The ratio of sensible to total heat.

**Alignment Circle** – Located at 24°C DB and 50% RH and used in conjunction with the sensible heat factor to plot the various air conditioning process lines.

**Kilograms of Dry Air** – The basis for all psychrometric calculations. Remains constant during all psychrometric processes.

### 8.3.2. Sensible heat factor (SHF)

The thermal properties of air can be separated into latent and sensible heat. The term *sensible heat factor* is the ratio of sensible to total heat, where total heat is the sum of sensible and latent heat. This ratio may be expressed as:

\[
SHF = \frac{SH}{SH + LH} = \frac{SH}{TH}
\]

Equation 9.1

Where:

- \(SH\) = sensible heat
- \(LH\) = latent heat
- \(TH\) = total heat

### 8.3.3. Room sensible heat factor (RSHF)

The *room sensible heat factor* is the ratio of room sensible heat to the summation of room sensible heat and room latent heat. This ratio is expressed in the following formula:

\[
RSHF = \frac{RSH}{RSH + RLH} = \frac{RSH}{RTH}
\]

Equation 9.2

The supply air to a conditioned space must have the capacity to offset simultaneously both the room sensible and room latent heat loads. The room and the supply air conditions to the space may be plotted on the standard psychrometric chart and these points connected with a straight line \((1-2)\), Fig. 37. This line represents the psychrometric process of the supply air within the conditioned space and is called the room sensible heat factor line.

The slope of the RSHF line illustrates the ratio of sensible to latent heat loads within the space and is illustrated in Fig. 37 by \(\Delta h_s\) (sensible heat) and \(\Delta h_l\) (latent heat). Thus if adequate air is supplied to offset these room loads, the room requirements will be satisfied, provided both the dry- and wet-bulb temperatures of the supply air fall on this line.
The room sensible heat factor line can also be drawn on the psychrometric chart without knowing the condition of supply air. The following procedure illustrates how to plot this line, using the calculated RSHF, the room design conditions, the sensible heat factor scale in the upper right hand corner of the psychrometric chart, and the alignment circle at 24°C dry-bulb and 50% relative humidity:

1. Draw a base line through the alignment circle and the calculated RSHF shown on the sensible heat factor scale in the upper right corner of psychrometric chart (1-2), Fig. 38.

2. Draw the actual room sensible heat factor line through the room design conditions parallel to the base line in Step 1 (3-4), Fig. 38. As shown, this line may be drawn to the saturation line on the psychrometric chart.

8.3.4. Grand sensible heat factor (GSHF)

The grand sensible heat factor is the ratio of total sensible heat to the grand total heat load that the conditioning plant must handle, including the outdoor heat loads. This ratio is determined from the following equation:

$$\text{GSHF} = \frac{TSH}{TLH + TSH} = \frac{TSH}{GTH}$$

Equation 9.3
Air passing through the conditioning plant increases or decreases in temperature and/or moisture content. The amount of rise or fall is determined by the total sensible and latent heat loads that the conditioning plant must handle. The condition of the air entering the plant (mixture condition of outdoor and return room air) and the condition of the air leaving the plant may be plotted on the psychrometric chart and connected by a straight line (1-2), Fig. 39. This line represents the psychrometric process of the air as it passes through the conditioning plant, and is referred to as the grand sensible heat factor line.

The slope of the GSHF line represents the ratio of sensible and latent heat that the plant must handle. This is illustrated in Fig. 39 by $\Delta h_s$ (sensible heat) and $\Delta h_l$ (latent heat).

The grand sensible heat factor line can be plotted on the psychrometric chart without knowing the condition of supply air, in much the same manner as the RSHF line. Fig. 40, Step 1 (1-2) and Step 2 (3-4) show the procedure, using the calculated GSHF, the mixture condition of air to the plant, the sensible heat factor scale, and the alignment circle on the psychrometric chart. The resulting GSHF line is plotted through the mixture conditions of the air to the plant.

### 8.3.5. Adjusted Room Sensible, Latent and Total Heat

The terminology of 'adjusted' room sensible heat, 'adjusted' room latent heat and 'adjusted' room total heat is introduced in this method.

The description 'effective' has traditionally been used to identify the heat quantities associated with this method for psychrometric calculations. Using this method, the value for dehumidified air quantity is given by the equation:

$$\text{Air Quantity} = \frac{\text{Effective Room Sensible Heat}}{1.2 \times (1 - \text{Bypass factor}) \times (\text{Room temp} - \text{App. dewpoint})}$$ \text{ Equation 9.4}

Where:

Effective room sensible heat = room sensible heat + supply duct sensible heat + portion of outdoor air bypassed through the apparatus.

This equation is not strictly correct because the by-pass factor should be applied to the GTH line. Manually this can only be done by a trial and error process on the psychrometric chart. With the aid of a computer the exact equation is:

$$\text{Air Quantity} = \frac{(\text{Adjusted Room Sensible Heat})}{1.2 \times (\text{Room temp} - \text{Leaving apparatus temp})}$$ \text{ Equation 9.5}

Where:

Adjusted Room Sensible Heat = Room Sensible Heat + Supply Duct Sensible Heat Gains
This formula can be employed to iterate to the precise answer.

Where the exact equation, i.e. equation 9.5 is used the description 'adjusted' room sensible heat is used to identify the numerator of the equation. Similarly 'adjusted' room latent heat is the sum of room latent heat and supply duct latent heat gains. 'Adjusted' room total heat is the sum of adjusted room sensible and latent heat.

8.3.6. Required air quantity

The air quantity required to simultaneously offset the room sensible and latent loads and the air quantity required through the plant to handle the total sensible and latent loads may be calculated, using the conditions on their respective RSHF and GSHF lines. For a particular application, when both the RSHF and GSHF ratio lines are plotted on the psychrometric chart, the intersection of the two lines (1) Fig. 41, represents the condition of the supply air to the space. It is also the condition of the air leaving the plant.

This neglects fan and duct heat gain, duct leakage losses, etc. In actual practice these heat gains and losses are taken into account in estimating the cooling load. Section 7 gives the necessary data for evaluating these supplementary loads. Therefore, the temperature of the air leaving the plant is not necessarily equal to the temperature of the air supplied to the space as indicated in Fig. 41.

Fig. 42 illustrates what actually happens when these supplementary loads are considered in plotting the RSHF and GSHF lines.

Point (1) is the condition of air leaving the plant and point (2) is the condition of the supply air to the space. Line (1-2) represents the temperature rise of the air stream resulting from supply fan and heat gain to the supply air duct. Line (3-4) represents the temperature rise of the air stream resulting from heat gain to the return air duct and return air fan.

The air quantity required to satisfy the room load may be calculated from the following equation:

\[
\frac{t}{S_{SA}} = \frac{RSH}{1.20(t_{RM} - t_{SA})}
\]

Equation 9.6
The air quantity required through the conditioning plant to satisfy the total air conditioning load (including the supplementary loads) is calculated from the following equation:

$$\frac{\ell}{s_{DA}} = \frac{TSH}{1.20(t_M - t_{LDB})}$$  \hspace{1cm} \text{Equation 9.7}

The required air quantity supplied to the space is equal to the air quantity required through the plant, neglecting leakage losses. The above equation contains the term $t_M$ which is the mixture condition of air entering the plant. With the exception of an all outdoor air application, the term $t_M$ can only be determined by trial and error.

One possible procedure to determine the mixture temperature and the air quantities is outlined below. This procedure illustrates one method of plant selection and is presented to show you how cumbersome and time consuming it may be.

1. Assume a rise ($t_{RM} - t_{SA}$) in the supply air to the space, and calculate the supply air quantity ($L/s_{SA}$) to the space.

2. Use this air quantity and the estimated heat gains from return air duct and return air fan to calculate the temperature rise due to these heat gains. This will establish temperature $t_4$ (point (4), Fig. 42).

3. From the supply air quantity, the outdoor design temperature and temperature $t_4$ calculate the temperature of the mixture condition $t_M$ (Equation 2, Psychrometric Formulae at the end of this Section, substituting $t_4$ for $t_{RM}$).

4. Substitute the supply air quantity and mixture condition of the air in the formula for air quantity through the plant ($L/s_{SA}$) and determine the leaving condition of the air from the conditioning plant ($t_{LDB}$).
5. The rise between the leaving condition from the plant and supply air condition to the space \((t_{SA} - t_{LDB})\) must be able to handle the loads due to supply duct heat gain and supply fan heat. If they cannot, a new rise in supply air is assumed and the trial-and-error procedure repeated.

Normally this difference in supply air temperature and the condition of the air leaving the plant \((t_{SA} - t_{LDB})\) and the temperature rise due to return air duct and return air fan \((t_{4} - t_{RM})\) are not more than a few °C. To simplify the discussion on the interrelationship of RSHF and GSHF, the supplementary loads have been neglected in the various discussions, formulae and problems in the remainder of this Section. It is emphasized, however, that these supplementary loads must be recognised when estimating the cooling and heating loads. These loads are taken into account on the air conditioning load estimate in Section 2, and are evaluated in Section 8.

The RSHF ratio will be constant (at full load) under a specified set of conditions; however, the GSHF ratio may increase or decrease as the outdoor air quantity and mixture conditions are varied for design purposes. As the GSHF ratio changes, the supply air condition to the space varies along the RSHF line (Fig. 41).

The difference in temperature between the room and the air supply to the room determines the air quantity required to satisfy the room sensible and room latent loads. As this temperature difference increases (supplying colder air, since the room conditions are fixed), the required air quantity to the space decreases. This temperature difference can increase up to a limit where the RSHF line crosses the saturation line on the psychrometric chart, Fig. 41, assuming, of course, that the available conditioning equipment is able to take the air to 100% saturation. Since this is impossible, the condition of the air normally falls on the RSHF line close to the saturation line. How close to the saturation line depends on the physical operating characteristics and the efficiency of the conditioning equipment.

In determining the required air quantity, when neglecting the supplementary loads, the supply air temperature is assumed to equal the condition of the air leaving the plant \((t_{SA} = t_{LDB})\). This is illustrated in Fig. 41. The calculation for the required air quantity still remains a trial-and-error procedure, since the mixture temperature of the air \((t_{Ma})\) entering the plant is dependent on the required air quantity. The same procedure previously described for determining the air quantity is used. Assume a supply air rise and calculate the supply air quantity and the mixture temperature to the conditioning plant. Substitute the supply air quantity and mixture temperature in the equation for determining the air quantity through the plant and calculate the leaving condition of the air from the plant. This temperature must equal the supply air temperature; if it does not, a new supply air rise is assumed and the procedure repeated.

Determining the required air quantity by either method previously described is a tedious process, since it involves a trial-and-error procedure, and in actual practice accounting for the supplementary loads in determining the supply air, mixture and leaving air temperatures.

This procedure has been simplified, by relating all the conditioning loads to the physical performance of the conditioning equipment, and then including this equipment performance in the actual calculation of the load.

This relationship is generally recognised as a psychrometric correlation of loads to equipment performance. The correlation is accomplished by calculation the ‘effective surface temperature’, ‘bypass
factor’, and ‘effective sensible heat factor’. These will permit a simplified calculation of supply air quantity.

### 8.3.7. Effective surface temperature ($t_{ES}$)

The surface temperature of the conditioning equipment varies throughout the surface of the plant as the air comes in contact with it. However, the effective surface temperature can be considered to be the uniform surface temperature which would produce the same leaving air conditions as the non-uniform surface temperature that actually occurs when the plant is in operation. This is more clearly understood by illustrating the heat transfer effect between air and the cooling (or heating) medium. *Fig. 43* illustrates this process and is applicable to a chilled water cooling medium with the supply air counterflow in relation to the chilled water.

![Diagram of effective surface temperature](image)

*Fig. 43*

The relationship shown in *Fig. 43* may also be illustrated for heating, direct expansion cooling and for air flowing parallel to the cooling or heating medium. The direction, slope and position of the lines change, but the theory is identical.

Since conditioning the air through the plant reduces to the basic principle of heat transfer between the heating or cooling media of the conditioning plant and the air through that plant, there must be a common reference point. This point is the effective surface temperature of the plant. The two heat transfers are relatively independent of each other, but are quantitatively equal when referred to the effective temperature.

Therefore, to obtain the most economical plant selection, the effective surface temperature is used in calculating the required air quantity and in selecting the plant.

For applications involving cooling and dehumidification, the effective surface temperature is at the point where the GSHF line crosses the saturation line on the psychrometric chart (*Fig. 39*). As such, this effective surface temperature is considered to be the dewpoint of the plant, and hence the term apparatus dewpoint (ADP) has come into common usage for cooling and dehumidifying processes.

Since cooling and dehumidification is one of the most common applications for central station plant, the ‘Air Conditioning Load Estimate’ form, *Fig. 47*, is designed around the term apparatus dewpoint (ADP).
The term is used exclusively in this Section when referring to cooling and dehumidifying applications. The psychrometrics of air can be applied equally well to other types of heat transfer applications such as sensible heating, evaporative cooling, sensible cooling etc., but for these applications the effective surface temperature will not necessarily fall on the saturation line.

**8.3.8. Bypass factor (BF)**

Bypass factor is a function of the physical and operating characteristics of the conditioning plant and, as such, represents that portion of the air which is considered to pass through the conditioning plant completely unaltered.

The physical and operating characteristics affecting the bypass factor are as follows:

1. A decreasing amount of available plant heat transfer surface results in an increase in bypass factor, i.e. less rows of coil, less coil surface, less/wider fins or wider spacing of coil tubes.
2. A decrease in the velocity of air through the conditioning plant results in a decrease in bypass factor, i.e. more time for the air to contact the heat transfer surfaces.

Decreasing or increasing the amount of heat transfer surface has a greater effect on bypass factor than varying the velocity of air through the plant.

There is a psychrometric relationship of bypass factor to GSHF and RSHF. Under specified room conditions, outdoor design conditions and quantity of outdoor air, RSHF and GSHF are fixed. The position of RSHF is also fixed, but the relative position of GSHF may vary as the supply air quantity and supply air condition change.

To properly maintain room design conditions the air must be supplied to the space at some point along the RSHF line. Therefore, as the bypass factor varies, the relative position of GSHF to RSHF changes, as shown by the dotted line in Fig. 44. As the position of GSHF changes, the entering and leaving air conditions at the plant, the required air quantity, bypass factor and the apparatus dewpoint also change.

The effect of varying the bypass factor on the conditioning equipment is as follows:

1. Small bypass factor –
   
   (a) Higher ADP – DX equipment selected for higher refrigerant temperature and chilled water equipment would be selected for less or higher temperature chilled water. Possibly smaller refrigeration machine.

   (b) Less air – smaller fan and fan motor.

   (c) More heat transfer surface – more rows of coil or more coil surface available.

   (d) Smaller piping if less chilled water is used.

2. Larger bypass factor –
(a) Lower ADP – Lower refrigerant temperature to select DX equipment, and more water or lower temperature for chilled water equipment. Possibly larger refrigeration machine.

(b) More air – larger fan and fan motor.

(c) Less heat transfer surface – less rows of coil or less coil surface available.

(d) Larger piping if more chilled water is used.

It is, therefore, an economic balance of first cost and operating cost in selecting the proper bypass factor for a particular application. Table 56 lists suggested bypass factors for various applications and is a guide for the engineer to proper bypass factor selection for use in load calculations.

Tables have also been prepared to illustrate the various configurations of heat transfer surfaces and the resulting bypass factor for different air velocities. Table 55 lists bypass factors for various coil surfaces. Spray washer equipment is normally rated in terms of saturation efficiency which is the complement of bypass factor (1–BF). Table 57 is a guide to representative saturation efficiencies for various spray arrangements.

As previously indicated, the entering and leaving air conditions at the conditioning plant and the apparatus dewpoint are related psychrometrically to the bypass factor. Although it is recognised that bypass factor is not a true straight line function, it can be accurately evaluated mathematically from the following equations:

\[
BF = \frac{t_{LDB} - t_{ADP}}{t_{EDB} - t_{ADP}} = \frac{h_{LA} - h_{ADP}}{h_{EA} - h_{ADP}} = \frac{w_{LA} - w_{ADP}}{w_{EA} - w_{ADP}} \quad \text{Equation 9.8}
\]

And

\[
1 - BF = \frac{t_{LDB} - t_{LDB}}{t_{EDB} - t_{ADP}} = \frac{h_{EA} - h_{LA}}{h_{LA} - h_{ADP}} = \frac{w_{LA} - w_{EA}}{w_{EA} - w_{ADP}} \quad \text{Equation 9.9}
\]
Note: The quantity \((1 - BF)\) is frequently called contact factor and is that portion of the air leaving the plant at the ADP.

### 8.3.9. Effective sensible heat factor (ESHF)

To relate bypass factor and apparatus dewpoint to the load calculation, the *effective sensible heat factor* term was developed. ESHF is interwoven with BF and ADP, and thus greatly simplifies the calculation of air quantity and plant selection.

The effective sensible heat factor is the ratio of effective room sensible heat to the sum of the effective room sensible and latent heats. Effective room sensible heat is composed of room sensible heat (see RSHF) plus that portion of the outdoor air sensible load which is considered as being bypassed, unaltered through the conditioning plant. The effective room latent heat is composed of the room latent heat (see RSHF) plus that portion of the outdoor air latent heat load which is considered as being bypassed, unaltered, through the conditioning plant. This ratio is expressed in the following formula:

\[
\text{ESHF} = \frac{\text{ERSH}}{\text{ERSH} + \text{ERLH}} = \frac{\text{ERSH}}{\text{ERTH}} \quad \text{Equation 9.10}
\]

The bypassed outdoor air loads that are included in the calculation of ESHF are, in effect, loads imposed on the conditioned space in exactly the same manner as the infiltration load. The infiltration load comes through the doors and windows; the bypassed outdoor air load is supplied to the space through the air distribution system.

Plotting RSHF and GSHF on the psychrometric chart defines the ADP and BF as explained previously. Drawing a straight line between the ADP and the room design conditions \((1–2)\), Fig. 45 represents the ESHF ratio (The ESHF line intersects the saturation line at the apparatus dewpoint and not elsewhere for the following reason: by having added the bypassed sensible and latent heat loads of the outdoor air to the room loads, the non-bypassed air now leaves a 100% efficient plant, i.e. at it’s dewpoint). The interrelationship of RSHF and GSHF to BF, ADP and ESHF is graphically illustrated in Fig. 45.

The effective sensible heat factor line may also be drawn on the psychrometric chart without initially knowing the ADP. The procedure is identical to the one previously described for RSHF. The calculated ESHF, however, is plotted through the room design conditions to the saturation line \((1–2)\), Fig. 46, thus indicating the ADP.
8.3.10. Air quantity using ESHF, ADP AND BF

A simplified approach for determining the required air quantity is to use the psychrometric correlation of effective sensible heat factor, apparatus dewpoint and bypass factor. Previously in this Section, the interrelationship of ESHF, BF and ADP was shown with GSHF and RSHF. These two factors need not be calculated to determine the required air quantity, since the use of ESHF, BF and ADP results in the same air quantity.

The formula for calculating air quantity, using BF and $t_{ADP}$ is:

$$\frac{\ell}{s_{DA}} = \frac{ERSH}{1.20(t_{RM} - t_{ADP})(1-BF)}$$

Equation 9.11

Where

*ESHF is used to determine $t_{ADP}$*

This air quantity simultaneously offsets the room sensible and room latent loads, and also handles the total sensible and latent loads for which the conditioning plant is designed, including the outdoor air loads and the supplementary loads.
8.3.11. Air conditioning load estimate form

The ‘Air Conditioning Load Estimate’ form is designed for cooling and dehumidifying applications, and may be used for psychrometric calculations. Normally only ESHF, BF and ADP are required to determine air quantity and to select the plant. But for those instances when it is desirable to know RSHF and GSHF, this form is designed so that these factors may also be calculated. Fig. 47, in conjunction with the following items, explains how each factor is calculated. (The circled numbers correspond to numbers in Fig. 47.)

1. \[ RSHF = \frac{RSH}{RSH + RLH} = \frac{1}{(1) + (2)} \]

2. \[ GSHF = \frac{TSH}{GTH} = \frac{(3) + (4)}{(5)} \]
3. \[ ESHF = \frac{ERSH}{ERSH + ERLH} = \frac{ERSH}{ERTH} \]

\[ (8) = \frac{(3)}{(3) + (6)} = \frac{(3)}{(7)} \]

4. ADP located where ESHF crosses the saturation line. ESHF (8) and room conditions (9) give ADP (10).

5. BF (11) used in the outdoor air calculations is obtained from the equipment performance table or charts. Typical bypass factors for different surfaces and for various applications are given in Table 56. These are to guide the engineer and may be used in the outdoor air calculation when the actual equipment performance tables are not readily available.

6. \[ \frac{\ell}{s_{DA}} = \frac{ERSH}{1.20(t_{RM} - t_{ADP})(1 - BF)} \]

\[ (13) = \frac{(3)}{1.20((9) - (10))(1 - (11))} \]

Once the dehumidified air quantity is calculated, the conditioning plant may be selected. The usual procedure is to use the grand total heat (5), dehumidified air quantity (13), and the apparatus dewpoint (10), to select the plant.

Since guides are available, the bypass factor of the plant selected is usually in close agreement with the originally assumed bypass factor. If, because of some peculiarity in loading of a particular application, there is a wide divergence in bypass factor, that portion of the load estimate form involving bypass factor should be adjusted accordingly.

7. Outlet temperature difference – Fig. 47 shows a calculation for determining the temperature difference between room design dry-bulb and supply air dry-bulb to the room. Frequently a maximum temperature difference is established for the application involved. If the outlet temperature difference calculation is larger than desired, the total air quantity in the system is increased by bypassing air around the conditioning plant. This temperature difference calculation is:

\[ \text{Outlet temp. diff.} = \frac{RSH}{1.20 \times \frac{\ell}{s_{DA}}} = \frac{(1)}{1.20 \times (13)} \]

8. Total air quantity when outlet temperature difference is greater than desired – The calculation for the total supply air quantity for a desired temperature difference (between room and outlet) is:

\[ \frac{\ell}{s_{SA}} = \frac{RSH}{1.20 \times \Delta t} = \frac{(1)}{1.20 \times \Delta t} \]
The amount of air that must be bypassed around the conditioning plant to maintain this desired temperature difference ($\Delta t$) is the difference between $L/SA$ and $L/DA$.\(^1\)

Entering and leaving conditions at the plant – Often it is desired to specify the selected conditioning plant in terms of entering and leaving air conditions at the plant. Once the plant has been selected from ESHF, ADP, BF and GTH, the entering and leaving air conditions are easily determined. The calculations for the entering and leaving dry-bulb temperatures at the plant are illustrated in Fig. 47.

The entering dry-bulb calculation contains the term ‘$L/s^\dagger$’. This air quantity ‘$L/s^\dagger$’ depends on whether a mixture of outdoor and return air or return air only is bypassed around the conditioning plant.

The total supply air quantity $L/SA$ (14) is used for ‘$L/s^\dagger$’ when bypassing a mixture of outdoor and return air. Fig. 48 is a schematic sketch of a system bypassing a mixture of outdoor and return air.

![Fig. 48](image)

When bypassing a mixture of return air only or when there is no need for a bypass around the plant, use the $L/DA$ (13) for the value of ‘$L/s^\dagger$’. Fig. 49 is a schematic sketch of a system bypassing room return air only.

The entering and leaving wet-bulb temperatures at the plant are determined on the standard psychrometric chart, once the entering and leaving dry-bulb temperatures are calculated. The procedure for determining the wet-bulb temperatures at the plant is illustrated in Fig. 50 and described in the following items:

1 Note that bypassing a mixture of outdoor air and return air is equivalent to using a coil with a larger bypass factor.

2 ‘$L/s^\dagger$’ is a symbol appearing in the equation next to (17) in Fig. 47.
a) Draw a straight line connecting room design conditions and outdoor design conditions

b) The point at which entering dry-bulb crosses the line plotted in Step (a) defines the entering conditions of the plant. The entering wet-bulb is read on the psychrometric chart.

c) Draw a straight line from the ADP (10) to the entering mixture conditions at the plant (Step (b)). (This line defines the GSHF line of the plant.)

d) The point at which the leaving dry-bulb crosses the line drawn in Step (c) defines the leaving conditions of the plant. Read the leaving wet-bulb from the plant at this point. (This point defines the intersection of the RSHF and GSHF as described previously)

8.4. **Air conditioning plant characteristic psychrometrics**

The following section describes the characteristic psychrometric performance of air conditioning equipment.

Drafting note: The air conditioning system types discussed are out of date and there are many new system options available from various chilled water systems to VRF DX systems.

Coils, sprays and desiccant dehumidifiers are the three basic types of heat transfer equipment required for air conditioning applications. These components may be used singly or in combination to control the psychrometric properties of the air passing through them.

The selection of this equipment is normally determined by the requirements of the specific application. The components must be selected and integrated to result in a practical system; that is, one having the most economical owning and operating cost.
An economical system requires the optimum combination of air conditioning components. It also requires an air distribution system that provides good air distribution within the conditioned space, using a practical rise between supply air and room air temperatures and adequate total air circulation.

Since the only known items are the load in the space and the conditions to be maintained within the space, the selection of various components is based on these items. Normally, performance requirements are established and then equipment is selected to meet the requirements.

8.5. Coil characteristics

In the operation of coils, air is drawn or forced over a series of tubes through which chilled water, brine, volatile refrigerant, hot water or steam is flowing. As the air passes over the surface of the coil, it is cooled, cooled and dehumidified, or heated, depending on the temperature of the media flowing through the tubes. The media in turn is heated or cooled in the process.

8.5.1. Coil bypass factor

The amount of coil surface not only affects the heat transfer but also the bypass factor of the coil. The bypass factor, as previously explained, is the measure of air side performance. Consequently, it is a function of the type and amount of coil surface and the time available for contact as the air passes through the coil. Table 55 gives approximate bypass factors for various finned coil surfaces and air velocities.

Table 55 contains bypass factors for a wide range of coils. This range is offered to provide sufficient latitude in selecting coils for the most economical system. Table 56 lists some of the more common applications with representative coil bypass factors. This table is intended only as a guide for the design engineer.

<table>
<thead>
<tr>
<th>Depth of Coils (rows)</th>
<th>Without Sprays</th>
<th>With Sprays</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Series 8*</td>
<td>Series 14*</td>
</tr>
<tr>
<td>Velocity (m/s)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.5 – 3.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.42 – 0.55</td>
<td>0.22 – 0.38</td>
</tr>
<tr>
<td>3</td>
<td>0.27 – 0.40</td>
<td>0.10 – 0.23</td>
</tr>
<tr>
<td>4</td>
<td>0.15 – 0.28</td>
<td>0.05 – 0.14</td>
</tr>
<tr>
<td>5</td>
<td>0.10 – 0.22</td>
<td>0.03 – 0.09</td>
</tr>
<tr>
<td>6</td>
<td>0.06 – 0.15</td>
<td>0.01 – 0.05</td>
</tr>
<tr>
<td>8</td>
<td>0.02 – 0.08</td>
<td>0.00 – 0.02</td>
</tr>
</tbody>
</table>
These bypass factors apply to coils with 16mm O.D. tubes and spaced on approximately 32mm centres. The values are approximate. Bypass factors for coils with plate fins, or for combinations other than those shown, should be obtained from the coil manufacturer.

### 1.1.1.2 TABLE 56
Typical Bypass Factors
FOR VARIOUS APPLICATIONS

<table>
<thead>
<tr>
<th>Coil Bypass Factor</th>
<th>Type of Application</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.30 to 0.50</td>
<td>A small total load or a load that is somewhat larger with a low sensible heat factor (high latent load)</td>
<td>Residence</td>
</tr>
<tr>
<td>0.20 to 0.30</td>
<td>Typical comfort application with a relatively small total load or a low sensible heat factor with a somewhat larger load.</td>
<td>Residence, Small Retail Shop, Factory</td>
</tr>
<tr>
<td>0.10 to 0.20</td>
<td>Typical comfort application</td>
<td>Dept. Store, Bank, Factory</td>
</tr>
<tr>
<td>0.05 to 0.10</td>
<td>Applications with high internal sensible loads or requiring a large amount of outdoor air for ventilation.</td>
<td>Office Block, Dept. Store, Restaurant, Factory</td>
</tr>
<tr>
<td>0 to 0.10</td>
<td>All outdoor air applications</td>
<td>Hospital Operating Room, Factory</td>
</tr>
</tbody>
</table>

### 8.5.2. Coil processes

Coils are capable of heating or cooling air at a constant moisture content, or simultaneously cooling and dehumidifying the air. They are used to control dry-bulb temperatures and maximum relative humidity at peak load conditions. Since coils alone cannot raise the moisture content of the air, a water spray on the coil surface must be added if humidification is required. If this spray water is recirculated, it will not materially affect the psychrometric process when the air is being cooled and dehumidified.

*Fig. 51 illustrates the various processes that can be accomplished by using coils.*
1.1.1.3 Sensible Cooling
The first process, illustrated by line (1–2), represents a sensible cooling application in which the heat is removed from the air at a constant moisture content.

1.1.1.4 Cooling and Dehumidification
Line (1–3) represents a cooling and dehumidification process in which there is a simultaneous removal of heat and moisture from the air.

For practical considerations, line (1–3) has been plotted as a straight line. It is, in effect, a line that starts at a point (1) and curves toward the saturation line below point (3). This is indicated by line (1–5). Both lines assume perfect contact between air and coil (i.e. BF = 0).

1.1.1.5 Sensible Heating
Sensible heating is illustrated by line (1–4); heat is added to the air at a constant moisture content.

To better understand these processes and their variations, a description of each with illustrated examples is presented in the following clauses: (Refer to the end of this Section for definition of symbols and abbreviations.)

8.5.3. Cooling and Dehumidification
Cooling and dehumidification is the simultaneous removal of the heat and moisture from the air, line (1–3), Fig. 51. Cooling and dehumidification occurs when the ESHF and GSHF are less than 1.0. The ESHF for these applications can vary from 0.95, where the load is predominantly sensible, to 0.45, where the load is predominantly latent.
The air conditioning load estimate form illustrated in Fig. 47 presents the procedure that is used to determine the ESHF, dehumidified air quantity, and entering and leaving air conditions at the plant. Example 1 illustrates the psychrometrics involved in establishing these values.

1.1.1.5.1.1 Example 1 – Cooling and Dehumidification

Given:

Application – Office Building
Summer design – 35°C DB, 25°C WB
Indoor design - 24°C DB, 50% RH
RSH – 60 000 W
RLH – 15 000 W
Ventilation – 1000 L/s

Find:

1. Outdoor air load (OATH)
2. Grand total heat (GTH)
3. Effective sensible heat factor (ESHF)
4. Apparatus dewpoint temperature ($t_{ADP}$)
5. Dehumidified air quantity (L/s)
6. Entering and leaving conditions at the plant ($t_{EDB}$, $t_{EWB}$, $t_{LDB}$, $t_{LWB}$)

Solution:

1. $OASH = 1.20 \times 1000 \times (35 - 24) = 13200$ W
   $OALH = 3.0 \times 1000 \times (15.9 - 9.4) = 19500$ W
   $OATH = 13200 + 19500 = 32700$ W

2. $TSH = 60000 + 13200 = 73200$ W
   $TLH = 15000 + 19500 = 34500$ W
   $GTH = 73200 + 34500 = 107700$ W

3. Assume a bypass factor of 0.15 from Table 56.

   $ESHF = \frac{60000 + (0.15)(13200)}{60000 + (0.15)(13200) + 15000 + (0.15)(19500)} = 0.776$

4. Determine the apparatus dewpoint from the room design conditions and the ESHF, by plotting on the psychrometric chart. Fig. 52 illustrates the ESHF plotted on the psychrometric chart.

   $t_{ADP} = 10^\circ C$

5. $\ell/s_{DA} = \frac{60000 + (0.15)(13200)}{1.20(24 - 10)(1 - 0.15)} = 4340\ell/s$

6. Assume for this example that the plant selected for 4340 L/s, 10°C ADP and GTH = 107 700, has a bypass factor that is equal, or nearly equal, to the assumed BF = 0.15. Also, assume that it is not necessary to physically bypass air around the plant.
\[ t_{EDB} = \frac{(1000 \times 35) + (3350 \times 24)}{4340} = 26.6^\circ C \text{ DB} \quad (31) \]

Read \( t_{EWB} \) where the \( t_{EDB} \) crosses the straight line plotted between the outdoor and room design conditions on the psychometric chart, Fig. 52.

\[ t_{EWB} = 19.1^\circ C \text{ WB} \]

\[ t_{LDB} = 10 + 0.15(26.5 – 10) = 12.5^\circ C \text{ DB} \quad (32) \]

Determine the \( t_{LWB} \) by drawing a straight line between the ADP and the entering conditions at the plant. (This is the GSHF line.) Where \( t_{LDB} \) intersects this line, read \( t_{LWB} \).

\[ t_{LWB} = 11.6^\circ C \text{ WB} \]

Note:

*Numbers in parentheses at right edge of column refer to equations given under the heading ‘Psychrometric Formulae’, at the end of this Section*

![Psychometric chart](image)

**1.1.5.1.2 Example 2– Cooling and Dehumidification**

Given:

The same office building as in Example 1, with the exception that the internal design conditions are 24°C DB, 65% RH instead of 24°C DB, 50% RH.

Find:

1. Grand Total Heat (GTH)
2. Dehumidified air quantity (L/sDA)
Solution:

1. The only change is in the outdoor air latent heat which is now \( OALH = 3.0 \times 1000 \times (15.9 - 12.2) = 11100 \text{ W} \) instead of 19500 W, a difference of 8400 W. \( \text{GTH} = 107700 - 8400 = 99300 \text{ W} \), a decrease of \( \frac{8400}{107700} \times 100 = 7.8\% \)

2.

\[
\text{ESHF} = \frac{60000 + (0.15)(13200)}{60000 + (0.15)(13200) + 15000 + (0.15)(11100)}
\]

\[
= 0.788
\]

\[
\text{t}_{\text{ADP}} = 15.8^\circ C
\]

\[
\frac{\text{l/s}_{\text{DA}}}{1.20(24 - 15.8)(1 - 0.15)} = 7410 / \text{s}
\]

against 4350 L/s in Example 1, an increase of \( \frac{7420 - 4340}{4340} \times 100 = 70.7\% \)

As the unit cost of refrigeration plant is smaller than that of air handling plant and ductwork, it can be seen that an increase in room design relative humidity is not an economic proposition. This statement holds in spite of the fact that the higher relative humidity results in a higher plant dew point (in this case 15.8°C instead of 10°C) and therefore in a decrease in the size of the refrigeration plant slightly greater than the decrease in grand total heat (in this case greater than 7.8%).

8.5.4. **Cooling and Dehumidification – High Latent Load Application**

On some applications a special situation exists if the ESHF and GSHF lines do not intersect the saturation line when plotted on the psychrometric chart or if the ADP is absurdly low. This may occur where the latent load is high with respect to the total loads (dance halls, etc). In such applications, an appropriate apparatus dewpoint is selected and the air is reheated to the RSHF line. Occasionally, altering the room heat design conditions eliminates the need for reheat, or reduces the quantity of reheat required. Similarly, the utilization of a large air side surface (low bypass factor) coil may eliminate the need for reheat or reduce the required reheat.

Once the ventilation air requirement is determined, and if the supply air quantity is not fixed, the best approach to determining the plant dew point is to assume a maximum allowable temperature difference between the supply air and the room. Then, calculate the supply air conditions to the space. The supply air conditions to the space must fall on the RSHF line to properly offset the sensible and latent loads in the space.

There are four criteria which should be examined, to aid in establishing the supply air requirements to the space. These are:
1. Air movement in the space.

2. Maximum temperature difference between the supply air and the room.

3. The selected ADP to provide an economical refrigeration machine selection.

4. In some cases, the ventilation air quantity required may result in all outdoor air application.

*Example 3* is a laboratory application with a high latent load. In this example, the ESHF intersects the saturation line, but the resulting ADP is too low.

### 1.1.1.5.1.3 Example 3 – Cooling and Dehumidification – High Latent Load

**Given:**

- Application – Laboratory
- Summer design – 32°C DB, 23°C WB
- Indoor design – 24°C DB, 50% RH
- RSH – 35 000 W
- RLH – 19 000 W
- Ventilation – 1200L/sOA
- Temp. diff. between room and supply air, 10°C maximum

**Find:**

1. Outdoor air load (OATH)
2. Effective sensible heat factor (ESHF)
3. Apparatus dewpoint \(t_{ADP}\)
4. Reheat required.
5. Supply air quantity \((L/ssa)\)
6. Entering conditions to coil \((t_{EDB}, t_{EWB}, w_{EA})\)
7. Leaving conditions from coil \((t_{LDL}, w_{SA})\)
8. Supply air condition to the space \((t_{SA}, w_{SA})\)
9. Grand total heat \((GTH)\)

**Solution:**
1. \[ OASH = 1.20 \times 1200 \times (32 - 14) = 11 500 \text{ W} \] (14)

\[ OALH = 3.0 \times 1200 \times (14.0 - 9.4) = 16 500 \text{ W} \] (15)

\[ OATH = 11 500 + 16 500 = 28 000 \text{ W} \] (17)

2. Assume a bypass factor of 0.05 because of high latent load.

\[
ESHF = \frac{35000 + (0.05)(11500)}{35000 + (0.05)(11500) + 19000 + (0.05)(16500)}
\]

\[ = 0.642 \] (26)

When plotted on the psychrometric chart, this ESHF (0.642) intersects the saturation curve at 2°C. With such a low ADP an appropriate apparatus dewpoint should be selected and the air reheated to the RSHF line.

3. For internal design conditions of 24°C DB, 50% RH, an ESHF of 0.74 results in an ADP of 9°C which is a reasonable minimum figure.

4. Determine amount of reheat (W) required to produce an ESHF of 0.74

\[
ESHF (0.74) = \frac{35000 + 0.05(11500) + \text{reheat}}{35000 + 0.05(11500) + \text{reheat} + 19000 + (0.05)16500}
\]

\[ 0.74 = \frac{35575 + \text{reheat}}{55400 + \text{reheat}} \] (25)

\[ \text{reheat} = 20 850 \text{ W} \]

5. Determine cooling coil air quantity \( \ell/\overline{s}_{DA} \)

\[
\ell/\overline{s}_{DA} = \frac{ERSH}{1.20 \times (1 - BF)(t_{RM} - t_{ADP})}
\]

\[ = \frac{35575 + 20850}{1.20(1 - 0.05)(24 - 9.0)} = 3300 \ell/s \]

\( L/s_{DA} \) is also \( L/s_{SA} \) when no air is to be physically by-passed around the cooling coil.

6. \[ t_{EDB} = \frac{(1200 \times 32) + (2100 \times 24)}{3300} \]

\[ = 26.9°C \] (31)

\textbf{Note:}

\textit{Numbers in parentheses at right edge of column refer to equations in the section ‘Psychrometric Formulae’ at the end of this Section.}
Read $t_{EWB}$ where the $t_{EDB}$ crosses the straight line plotted between the outdoor air and room design conditions on the psychrometric chart, Fig. 53

$t_{EWB} = 19.3^\circ C$

The moisture content at the entering conditions to the coil is read from the psychrometric chart.

$w_{EA} = 10.95$ g/kg

7. Determine leaving conditions of air from cooling coil

$t_{LDB} = t_{ADB} + BF (t_{EDB} - t_{ADP})$  \hspace{1cm} (32)

$= 9.0 + 0.05 (26.9 - 9.0)$

$= 9.9^\circ C$

$h_{SA} = h_{ADP} + BF (h_{EA} - h_{ADP})$  \hspace{1cm} (34)

$= 27.2 + 0.05 (55.2 - 27.2)$

$= 28.65$ kJ/kg

$t_{LWB} = 9.65^\circ C$

8. Determine the supply air temperature to space
$t_{SA} = t_{RM} - \frac{RSH}{1.20(i/s_{SA})}$

$= 24 - \frac{(35000)}{1.20(3300)}$

$= 15.16^\circ C$

$t_{SA}$ should also equal $t_{LDB} + \frac{\text{reheat}}{1.20(i/s_{SA})}$

$= 9.9 + \frac{20850}{1.20(3300)}$

$= 15.16^\circ C$

$w_{SA} = 7.3 \text{ g/kg}$

Temp. diff. between room and supply air = $t_{RM} - t_{SA} = 24 - 15.15 = 8.85^\circ C$

which is less than 10°C

9. GTH = 1.19 x 3300 (55.25 – 28.6) = 104 000 W (24)

10. Check on GTH = RSH + RTH + OATH + reheat

$= 35 000 + 19 000 + 28 000 + 21 000$

$= 103 000 \text{ W}$

8.5.5. Cooling and Dehumidification – Using 100% Outdoor Air

In some applications it may be necessary to supply all outdoor air; for example, a hospital operating room or an area that requires large quantities of ventilation air. For such applications, the ventilation or code requirements may be equal to, or more than, the air quantity required to handle the room loads.

Items 1 to 5 inclusive explain the procedure for determining the dehumidified air requirements using the ‘Air Conditioning Load Estimate’ form when all outdoor air is required.

1. Calculate the various loads and determine the apparatus dewpoint and dehumidified air quantity.

2. If the dehumidified air quantity is equal to the outdoor air requirements, the solution is self-evident.

3. If the dehumidified air quantity is less than the outdoor air requirements, a coil with a large bypass factor should be investigated when the difference in air quantities is small. If a large difference exists however, reheat is required. This situation sometimes occurs when the application requires large exhaust air quantities.

4. If the dehumidified air quantity is greater than the outdoor air requirements, substitute $L/S_{DA}$ for $L/S_{OA}$ in the outdoor air load calculations.

5. Use the recalculated outdoor air loads to determine a new apparatus dewpoint and dehumidified air quantity. This new dehumidified air quantity should check reasonably close to the $L/S_{DA}$ in Item 1.
A special situation may arise when the condition explained in Item 4 occurs. This happens when the ESHF, as plotted on the psychrometric chart, does not intersect the saturation line. This situation is handled in a manner similar to that previously described under ‘Cooling and Dehumidification – High Latent Load Application’.

Example 4 illustrates an application where all outdoor air is to be supplied to the space.

1.1.1.5.1.4 Example 4 – Cooling and Dehumidification – 100% Outdoor Air

Given:

Application – Laboratory

Summer design – 35°C DB, 24°C WB

Indoor design – 24°C DB, 55% RH

RSH – 14 500 W

RLH – 3200 W

Ventilation – 750 L/soa

All outdoor air to be supplied to the space.

Find:

1. Outdoor air load (OATH)
2. Effective sensible heat factor (ESHF)
3. Apparatus dewpoint (tADP)
4. Dehumidified air quantity (LsOA)
5. Recalculated outdoor air load (OATH)
6. Recalculated effective sensible heat factor (ESHF)
7. Final apparatus dewpoint temperature (tADP)
8. Recalculated dehumidified air quantity (LsOA)

Solution:

1. OASH = 1.20 x 750 x (35-24) = 9900 W (14)

OALH = 3.0 x 750 x (14.3 – 10.25) = 9100 W (15)

OATH = 9900 + 9100 = 19 000 W (17)
2. Assume a bypass factor of 0.05 from Tables 55 and 56.

\[
ESHF = \frac{14500 + (0.05)(9900)}{14500 + (0.05)(9900) + 3200 + (0.05)(9100)} = 0.805
\]

(26)

3. At the given room design conditions and effective sensible heat factor, \( t_{ADP} = 12.6°C \)

4. \( \dot{V}/S_{DA} = \frac{14500 + (0.05)(9900)}{1.20(1 - 0.05)(24 - 12.6)} = 1150 \dot{V} / s \)

(36)

Since 1150 L/s is larger than the ventilation requirements, and by code all OA is required, the OA loads, the ADP and the dehumidified air quantity must be recalculated using 1150 L/s as the OA requirements.

5. Recalculating outdoor air load

\[
OASH = 1.20 \times 1150 \times (35-24) = 15180 W
\]

(14)

\[
OALH = 3.0 \times 1150 \times (14.3 – 10.25) = 13970 W
\]

(15)

\[
OATH = 15180 + 13970 = 29150 W
\]

(17)

6. \( ESHF = \frac{14500 + (0.05)(15800)}{14500 + (0.05)(15180) + 3200 + (0.05)(13970)} = 0.795
\]

(26)

7. \( t_{ADP} = 12.6°C \)

8. \( \dot{V}/S_{DA} = \frac{14500 + (0.05)(15180)}{1.20(1 - 0.05)(24 - 12.6)} = 1174 \dot{V} / s \)

(36)

This checks reasonably close to the value in Step 4, and recalculation is not necessary.

Note: Numbers in parentheses at right edge of column refer to equations in the section ‘Psychrometric Formulae’ at the end of this Section.

8.5.6. Cooling with Evaporative Humidification

Note: For cooling with humidification at near constant dry-bulb temperature, refer to the heading after next.

Cooling with evaporative humidification may be used at design conditions for applications having relatively high sensible loads and high room relative humidity requirements. Without humidification, excessively high supply air quantities may be required. This not only creates air distribution problems but also is often economically unsound. Excessive supply air quantity requirements can be avoided by introducing moisture into the space to convert sensible heat to latent heat. This is sometimes referred to as a ‘split system’. The moisture is introduced into the space by using water sprays.
When humidification is performed in the space, the room sensible heat load is decreased by an amount equal to the latent heat added, since the process is merely an interchange of heat. The spray pump motor adds sensible heat to the room but the amount is negligible and is usually ignored. A credit to the room sensible heat should be taken in the amount of the latent heat from the added moisture and the latent load introduced into the space should be added to the room latent load.

The introduction of this moisture into the space to reduce the required air quantity decreases the RSHF, ESHF and the apparatus dewpoint. This method of reducing the required air quantity is normally advantageous when designing for high room relative humidities.

The method of determining the amount of moisture necessary to reduce the required air quantity results in a trial and error procedure. The method is outlined in the following steps:

1. Assume an amount of moisture to be added and determine the latent heat available from this moisture. Table 53 gives the maximum moisture that may be added to a space without causing condensation on supply air ducts and equipment.

2. Deduct this assumed latent heat from the original effective room sensible heat and use the difference in the following equation for ERSH to determine $t_{ADP}$.

$$t_{ADP} = t_{RM} - \frac{ERSH}{1.20 \times (1 - BF)/S_{DA}}$$

$L/S_{DA}$ is the reduced air quantity permissible in the air distribution system.

3. The ESHF is obtained from a psychrometric chart, using the apparatus dewpoint (from Step 2) and room design conditions.

4. The new effective room latent load is determined from the following equation:

$$ERLH = ERSH \times \frac{1 - ESHF}{ESHF}$$

The ERSH is from Step 2 and ESHF is from Step 3.

5. Deduct the original ERLH (before adding sprays or humidifier in the space) from the new effective room latent heat in Step 4. The result is equal to the latent heat from the added moisture, and must check with the value assumed in Step 1. If it does not check, assume another value and repeat the procedure.

Example 5 illustrates the procedure for investigating an application where humidification is accomplished within the space to reduce air quantity.

1.1.1.6 Example 5 – Cooling with Evaporative Humidification in the Space

Given:

Application – A high humidity chamber
Summer Design – 35°C DB, 25°C WB

Indoor Design – 21°C DB, 70% RH

RSH – 47 000 W

RLH – 3000 W

RSHF – 0.94

Ventilation – 1900 L/sOA

Find:

A. When space humidification is not used:

1. Outdoor air load (OATH)
2. Grand total heat (GTH)
3. Effective sensible heat factor (ESHF)
4. Apparatus dewpoint ($t_{ADP}$)
5. Dehumidified air quantity ($L/SoA$)
6. Entering and leaving conditions at the plant ($t_{EDB}$, $t_{EWB}$, $t_{LDB}$, $t_{LWB}$)

B. When humidification is used in the space:

1. Assume maximum air quantity and assume an amount of moisture added to the space.
2. New effective room sensible heat (ERSH)
3. New apparatus dewpoint ($t_{ADP}$)
4. New effective sensible heat factor (ESHF)
5. New effective room latent heat (ERLH)
6. Check calculated latent heat from the moisture added with amount assumed in Item 1.
7. Theoretical conditions of the air entering the evaporative humidifier before humidification.
8. Entering and leaving conditions at the plant ($t_{EDB}$, $t_{EWB}$, $t_{LDB}$, $t_{LWB}$)

Solution:

A. When space humidification is not used:
1. \( \text{OASH} = 1.20 \times 1900 \times (35 - 21) = 32\,000 \text{ W} \) \( (14) \)

\( \text{OALH} = 3.0 \times 1900 \times (15.9 - 11) = 27\,900 \text{ W} \) \( (15) \)

\( \text{OATH} = 32\,000 + 27\,900 = 59\,900 \text{ W} \) \( (17) \)

2. \( \text{GTH} = 47\,000 + 3000 + 32\,000 + 27\,900 \)

\( = 109\,900 \text{ W} \)

3. Assume a bypass factor of 0.05 from \( \text{Tables 55 and 56} \).

\[ \text{ESHF} = \frac{47\,000 + (0.05)(32\,000)}{47\,000 + (0.05)(32\,000) + 32\,000 + (0.05)(27\,900)} \]

\[ = 0.92 \] \( (26) \)

4. Plot the ESHF on a psychrometric chart and read the ADP (dotted line in \( \text{Fig. 54} \))

\( t_{\text{ADP}} = 15.0^\circ \text{C} \)

5. \( \frac{t_{\text{SDA}}}{1.20(1 - 0.05)(21 - 15)} = 7100 /\text{s} \) \( (36) \)

6. \( t_{\text{EDB}} = \frac{(1900 \times 35) + (5200 \times 21)}{7100} = 24.7^\circ \text{C DB} \) \( (31) \)

Read \( t_{\text{EWB}} \) where the \( t_{\text{EDB}} \) crosses the straight line plotted between the outdoor and room design conditions on the psychrometric chart (\( \text{Fig. 54} \))

\( t_{\text{EWB}} = 19.0^\circ \text{C WB} \)

\( t_{\text{LDB}} = 15.0 + 0.05(24.7 - 15) = 15.5^\circ \text{C DB} \) \( (32) \)

Determine the \( t_{\text{LWB}} \) by drawing a straight line between the ADP and the entering conditions to the plant (the GSHF line). Where the \( t_{\text{LDB}} \) intersects this line, read the \( t_{\text{LWB}} \) (\( \text{Fig. 54} \))

\( t_{\text{LWB}} = 15.2^\circ \text{C WB} \)

B. When humidification is used in the space:

1. Assume, for the purpose of illustration in this problem, that the maximum air quantity permitted in the air distribution system is 4500 L/s. Assume 0.8 gram of moisture per kilogram of dry air is to be added to convert sensible heat to latent heat. The latent heat is calculated by multiplying the air quantity times the moisture added times the factor 3.0

\[ 0.8 \times 4500 \times 3.0 = 10\,800 \text{ W} \]

2. New \( \text{ERSH} = \text{Original ERSH} - \text{latent heat of added moisture} \)
\[ = [47\,000 + (0.05 \times 32\,000)] - 10\,800 \]
\[ = 37\,800 \text{ W} \]

3. \( t_{\text{ADP}} = 21 - \frac{37800}{1.20(1-0.05)(4500)} = 13.65^\circ \text{C} \) \hspace{1cm} (36)

4. ESHF is read from the psychrometric chart as 0.715 (dotted line in Fig. 55).

5. \( \text{New ERLH} = \text{New ERSH} \times \frac{1-\text{ESHF}}{\text{ESHF}} \)
\[ = 37800 \times \frac{1-0.715}{0.715} \]
\[ = 15100 \text{ W} \]

6. Check for latent heat of added moisture.

Latent heat of added moisture
\[ = \text{New ERLH} - \text{Original ERLH} \]
\[ = 15\,100 - [3000 + (0.05 \times 27\,900)] \]
\[ = 10\,700 \text{ W} \]

This checks reasonably close with the assumed value in Step 1 (10\,800 W)

Note: The trial and error method will reveal the criticality of choosing relative values of air quantity and moisture content.

7. Psychrometrically, it can be assumed that the atomised water from the spray heads in the space absorbs part of the room sensible heat and turns into water vapour at the final room wet-bulb temperature. The theoretical dry-bulb of the air entering the sprays is at the intersection of the room design wet-bulb line and the moisture content of the air entering the sprays. This moisture content is determined by subtracting the moisture added by the room sprays from the room design moisture content.

Moisture content of air entering cooling coil
\[ 10.95 - 0.8 = 10.15 \text{ g/kg} \]

The theoretical dry-bulb is determined from the psychrometric chart as 23°C DB illustrated on Fig. 55.

8. \( t_{\text{EDB}} = \frac{(1900 \times 35) + (2600 \times 21)}{4500} = 26.9^\circ \text{C} \) \hspace{1cm} (31)

Read \( t_{\text{EBW}} \) where the \( t_{\text{EDB}} \) crosses the straight line plotted between the outdoor and room design conditions on the psychrometric chart (Fig. 55)
\( t_{\text{EWB}} = 20.8^\circ \text{C WB} \)

\[ t_{\text{LDB}} = 13.65 + (0.05)(26.9 - 13.65) \]

\[ = 14.3^\circ \text{C DB} \quad (32) \]

Determine \( t_{\text{LWB}} \) by drawing a straight line between ADP and the entering conditions to the plant (GSHF line). Where \( t_{\text{LDB}} \) intersects the line, read the \( t_{\text{LWB}} \) (Fig. 55).

\( t_{\text{LWB}} = 14.1^\circ \text{C WB} \)

The straight line connecting the leaving conditions at the plant with the theoretical condition of the air entering the evaporative humidifier represents the theoretical process line of the air. This theoretical condition of the air entering the humidifier represents what the room conditions are if the humidifier is not operating. The slope of this theoretical process line is the same as RSHF (0.94).

The heavy lines on Fig. 55 illustrate the theoretical air cycle as air passes through the conditioning plant to the evaporative humidifier, then to the room, and finally back to the plant where the return air is mixed with the ventilation air. Actually, if a straight line were drawn from the leaving conditions of the plant (14.3°C DB, 14.1°C WB) to the room design conditions, this line would be the RSHF line and would be the process line for the supply air as it picks up the sensible and latent loads in the space (including the latent heat added by the sprays).
The following two methods of laying out the system are recommended when the humidifier is to be used for both partial load control and reducing the air quantity.

1. Use two humidifiers; one to operate continuously, adding the moisture to reduce the air quantity, and the other to operate intermittently to control humidity. The humidifier used for partial load is sized for the effective room latent load, not including that produced by the other humidifier. If the winter requirements for moisture addition are larger than summer requirements, then the humidifier is selected for these conditions. This method of using two humidifiers gives the best control.

2. Use one humidifier of sufficient capacity to handle the effective room latent heat plus the calculated amount of latent heat from the added moisture required to reduce the air quantity. In Part B, Step 5, the humidifier would be sized for a latent load of 15100 W.

8.5.7. Sensible Cooling

A sensible cooling process is one that removes heat from the air at a constant moisture content, line (1-2), Fig. 51. Sensible cooling occurs when either of the following conditions exist:

1. The GSHF as calculated or plotted on the psychrometric chart is 1.0.

2. The ESHF calculated on the air conditioning load estimate form is equal to 1.0.

In a sensible cooling application, the GSHF equals 1.0. The ESHF and the RSHF may equal 1.0. When only the RSHF equals 1.0, however, it does not necessarily indicate a sensible cooling process because latent load, introduced by outdoor air, can give a GSHF less than 1.0.

The apparatus dewpoint is referred to as the effective surface temperature ($t_{ES}$) in sensible cooling applications. The effective surface temperature must be equal to, or higher than, the dewpoint temperature of the entering air. In most instances, the $t_{ES}$ does not lie on the saturated line and, therefore, will not be the dewpoint of the plant. However, the calculations may still be performed on the air conditioning load estimate form by substituting the term $t_{ES}$ for $t_{ADP}$.

The use of the term $L/s_{DA}$ in a sensible cooling application should not be construed to indicate that dehumidification is occurring. It is used in the ‘Air Conditioning Load Estimate’ form and in Example 6 to determine the air quantity required through the plant to offset conditioning loads.

The leaving air conditions from the coil are dictated by the room design conditions, the load and the required air quantity.

Example 6 illustrates the method of determining the apparatus dewpoint or the effective surface temperature for a sensible cooling application.

### 1.1.1.6.1.1 Example 6 – Sensible Cooling

Given:

Application: Factory

Summer Design: 40°C DB, 21°C WB

Indoor Design: 24°C DB
RSH – 60 000 W
RLH – 15 000 W
Ventilation: 6000 L/s; all outdoor air.

Find:
1. Moisture content of room air ($w_{RM}$)
2. Room wet-bulb temperature ($t_{RM WB}$)
3. RSHF
4. Air leaving conditions ($t_{LDB}$, $t_{LWB}$)
5. Outdoor air sensible heat (OASH)
6. Grand total heat (GTH)
7. Effective surface temperature, assuming BF = 0.05
8. Highest bypass factor which can be used in this instance

Solution

1. $w_{RM} - 7.7 = \frac{15000}{6000 \times 3.0}$ (43)
   
   $w_{RM} = 8.54$ g/kg

2. The room condition is $24^\circ$C DB, 8.54 g/kg. From psychrometric chart read $t_{RM WB} = 16.3^\circ$C Fig. 56

3. $\text{RSHF} = \frac{60000}{60000 + 15000} = 0.80$ (25)

4. $1.20 (24 - t_{LDB}) \times 6000 = 60000$ (35)
   
   $t_{LDB} = 15.65^\circ$C, $t_{LWB} = 12.4^\circ$C (from psychrometric chart)

5. $\text{OASH} = 1.20 \times 6000 (40 - 24) = 115 200$ W (24)

6. $\text{GTH} = 60 000 + 15 000 + 115 200 = 190 200$ W (9)

7. $\text{BF} = 0.05 = \frac{15.65 - t_{ES}}{40 - t_{ES}}$ (28)
   
   $t_{ES} = 14.4^\circ$C
8. The dewpoint temperature of the air entering the equipment is 10°C (from psychrometric chart). This is the lowest possible surface temperature with sensible cooling.

\[ BF_{\text{max}} = \frac{15.65 - 10}{40 - 10} = 0.1885 \]

*Note:*

*Numbers in parentheses at right edge of column refer to equations in the section ‘Psychrometric Formulae’ at the end of this Section.*

**8.5.8. Cooling with humidification at near constant dry-bulb temperature**

When steam is injected or water vapour released into the conditioned space from say electric humidifiers, humidification becomes a near isothermal process. This method of humidifying can be used at design or partial load conditions when the room latent heat load is insufficient to permit the design room relative humidity to be reached. *Example 7* illustrates the method of determining the amount of steam required for a sensible cooling application.

**1.1.1.6.1.2 Example 7 – Sensible Cooling and Near-Isothermal Humidification**

*Given:*

Application: Operating Theatre

Summer design: 40°C DB, 21°C WB

Indoor design: 24°C DB, 60% RH

RSH – 60 000 W

RLH – 15 000 W

Ventilation: 6000 L/s; all outdoor air

*Find:*

1. Room conditions without humidification \((t'_{\text{RM DB}}, t'_{\text{RM WB}})\)
2. Air leaving conditions ($t_{LDB}$, $t_{LWB}$)

3. Effective surface temperature, assuming BF = 0.05

4. Amount of moisture (steam) required to raise room relative humidity to the design figure of 60% RH.

**Solution:**

It will be noted that the answers to the first three points are identical to those given in *Example 6*, i.e.

1. $t'_{RM DB} = 24^\circ C$, $t'_{RM WB} = 16.3^\circ C$

2. $t_{LDB} = 15.65^\circ C$, $t_{LWB} = 12.4^\circ C$

3. $t_{ES} = 14.4^\circ C$

4. Difference between moisture content of room at 60% RH ($w_1$) and that achievable without humidification ($w_2 = 8.54$ g/kg, from *Example 5*): $w_1 = 11.4$ g/kg (from psychrometric chart, *Fig. 57*)

With an air flow of 6000 L/s and air at a specific volume of 0.842 m³/kg, the amount of moisture (steam) required is:

$$\frac{6000}{1000(\ell/m^3)} \times 2.86 \times \frac{1}{0.842} = 20.38 \text{ g/s}$$

This is equivalent to $20.38 \times 2500 \text{ (kJ/kg)} = 51 000 \text{ W}$

(This could have been arrived at by application of formula (23), i.e. TLH = $3.0 \times 6000 \times (11.4 - 8.54) = 51 000\text{ W}$)

If humidification were done by an electric humidifier, its rating would be 51 000 W (neglecting heat losses in the humidifier).

5. The rise in dry-bulb temperature due to steam humidification is usually neglected. In this instance and assuming steam injection at 100°C, the rise would be 0.405°C. Air leaving temperature ($t_{LDB}$) and effective surface temperature ($t_{ES}$) would have to be reduced by that amount.

![Fig. 57](image-url)
8.6. **Desiccant dehumidifiers**

Comment: rewrite to take account of current practice - needs a lot more detail to be of help to designers.

Since 2001 there have been very good technologies that combine the best of refrigeration technology with the best of desiccant dehumidification technology to create a very efficient ventilation dehumidifier. These more efficient technologies should be mentioned in DA09.

Desiccant dehumidifiers are quite different from cooling-coil based dehumidifiers. Instead of cooling the air to condense its moisture, desiccants attract moisture from the air by creating an area of low vapor pressure at the surface of the desiccant. The pressure exerted by the water in the air is higher, so the water molecules move from the air to the desiccant and the air is dehumidified.

Desiccant dehumidifiers contain liquid absorbent or solid adsorbent which are either sprayed directly into, or located in, the path of the air stream. One distinction between desiccants is their reaction to moisture. Some desiccants, called adsorbents, simply collect it like a sponge collects water. Silica gel is an example of a solid adsorbent. Other desiccants called absorbents undergo a chemical or physical change as they collect moisture. Sodium chloride, common table salt, is a hygroscopic salt which collects water vapor by absorption.

Whether the desiccant functions by absorption or adsorption is not usually important to a system designer, but engineers should be aware of the difference between the two terms.

Liquid absorbents change either physically or chemically, or both, during the sorption process. Solid adsorbents do not change during the sorption process.

As moist air comes into contact with either the liquid absorbent or solid adsorbent, moisture is removed from the air by the difference in vapour pressure between the air stream and the sorbent. If the desiccant is cool and dry, its surface vapor pressure is low, and it can attract and remove moisture from the air, which has a high vapor pressure when it is moist. After the desiccant becomes wet and hot, its
surface vapor pressure is high, and it will give off water vapor to the surrounding air. Vapor moves from the air to the desiccant and back again depending on vapor pressure differences.

As the moisture condenses, latent heat of condensation is liberated, causing a rise in the temperature of the air stream and the sorbent material. This process occurs at a wet-bulb temperature that is approximately constant. However, instead of adding moisture to the air as in an evaporative cooling process, the reverse occurs. Heat is added to the air and moisture is removed from the air stream; thus it is a dehumidification and heating process as illustrated in Fig. 62. Line (1-2) is the theoretical process and the dotted line (1-3) approximates what actually happens. Line (1-3) can vary, depending on the type of sorbent used.

8.7. **Psychrometrics of partial load control**

Comment: Brief discussion only of various control strategies - needs to be expanded and updated to include systems such as Chilled Beams, Underfloor Air Distribution, Displacement Ventilation, etc and systems with direct outdoor air injection.

The plant required to maintain proper space conditions is normally selected for peak load operation. Actually, peak load operation occurs but a few times each year and operation is predominantly at partial load conditions. Partial load may be caused by a reduction in sensible or latent loads in the space, or in the outdoor air load. It may also be caused by a reduction in these loads in any combination.

8.7.1. **Partial load analysis**

Since the system operates at partial load most of the time and must maintain conditions commensurate with job requirements, partial load analysis is at least as important as the selection of equipment. Partial load analysis should include a study of resultant room conditions at minimum total load. Usually this will be sufficient. Certain applications, however, should be evaluated at minimum latent load with design sensible load, or minimum sensible load and full latent load. Realistic minimum and maximum loads should be assumed for the particular application so that, psychrometrically, the resulting room conditions are properly analysed.

The six most common methods, used singly or in combination, of controlling space conditions for cooling applications at partial load are the following:

1. Reheat the air supply.
2. Bypass the heat transfer equipment.
3. Control the volume of the supply air.
4. Use on-off control of the air handling equipment.
5. Use on-off control of the refrigeration machine.
6. Control the refrigeration capacity.

The type of control selected for a specific application depends on the nature of the loads, the conditions to be maintained within the space, and available plant facilities.
8.7.2. Reheat control

Reheat control maintains the dry-bulb temperature within the space by replacing any decrease in the sensible loads by an artificial load. As the internal latent load and/or the outdoor latent load decreases, the space relative humidity decreases. If humidity is to be maintained, rehumidifying is required in addition to reheat.

Figure 63 illustrates the psychrometrics of reheat control. The solid lines represent the process at design load, and the broken lines indicate the resulting process at partial load. The RSHF value, plotted from room design conditions to point (2), must be calculated for the minimum practical room sensible load. The room thermostat then controls the temperature of the air leaving the reheat coil along line (1-2). This type of control is applicable for any RSHF ratio that intersects line (1-2).

If the internal latent loads decrease, the resulting room conditions are at point (3), and the new RSHF process line is along line (2-3). However, if humidity is to be maintained within the space, the reduced latent load is compensated by humidifying, thus returning to the design room conditions.

8.7.3. Bypass control

Bypass control maintains the dry-bulb temperature within the space by modulating the amount of air to be cooled, thus varying the supply air temperature to the space. Fig. 64 and 65 illustrates one method of bypass control when bypassing return air only.
Bypass control may also be accomplished by bypassing a mixture of outdoor and return air around the heat transfer equipment. This method of control is inferior to bypassing return air only since it introduces raw unconditioned air into the space, thus allowing an increase in room relative humidity.

A reduction in room sensible load causes the bypass control to reduce the amount of air through the cooling coil. This reduced air quantity results in equipment operation at a lower apparatus dewpoint. Also, the air leaves the cooling coil at a lower temperature so that there is a tendency to adjust for a decrease in sensible load that is proportionately greater than the decrease in latent load.

Bypass control maintains the room dry-bulb temperature but does not prevent the relative humidity from rising above design. With bypass control, therefore, increased relative humidity occurs under conditions of decreasing room sensible load and relatively constant room latent load and outdoor air load.
The heavy lines in Fig. 64 represent the cycle for design conditions. The light lines illustrate the initial of the air when bypass control first begins to function. The new room conditions, mixture conditions and apparatus dewpoint continue to change until the equilibrium point is reached.

Point (2) on Figs. 64 and 65 is the condition of air leaving the cooling coil. This is a result of a smaller bypass factor and lower apparatus dewpoint caused by less air through the cooling equipment and a smaller load on the equipment. Line (2-3-4) represents the new RSHF line caused by the reduced room sensible load. Point (3) falls on the new RSHF line when bypassing return air only.

Bypassing a mixture of outdoor and return air causes the mixture point (3) to fall on the GSHF line, Fig. 64. The air is then supplied to the space along the new RSHF line (not shown in Fig. 64) at a higher moisture content than the air supplied when bypassing return air only. Thus it can be readily observed that humidity control is further hindered with the introduction of unconditioned outdoor air into the space.

8.7.4. Volume control

Volume control of the supply air quantity provides essentially the same type of control that results from bypassing return air around the heat transfer equipment, Fig. 64. However, this type of control may produce problems in air distribution within the space and, therefore, the required air quantity at partial loads should be evaluated for proper air distribution.

8.7.5. On-off control of air-handling equipment

On-off control of air handling equipment (fan coil units) results in a fluctuating room temperature and space relative humidity. During the ‘off’ operation the ventilation air supply is shut off, but chilled water continues to flow through the coils. This method of control is not recommended for high latent load applications, as control of humidity may be lost at reduced room sensible loads.

8.7.6. On-off control of refrigeration equipment

On-off control of refrigeration equipment (large packaged equipment) results in a fluctuating room temperature and space relative humidity. During the ‘off’ operation air is available for ventilation purposes but the coil does not provide cooling. Thus, any outdoor air in the system is introduced into the space unconditioned. Also the condensed moisture that remains on the cooling coil, when the refrigeration equipment is turned off, is re-evaporated in the warm air stream. This is known as re-evaporation. Both of these conditions increase the space latent load, and excessive humidity results. This method of control is not recommended for high latent load applications since control of humidity may be lost at decreased room sensible loads.

8.7.7. Refrigeration capacity control

Refrigeration capacity control may be used on either chilled water or direct expansion refrigeration equipment. Partial load control is accomplished on chilled water equipment by bypassing the chilled water around the air side equipment (fan-coil units) or by shutting down one or more of a multiplicity of chilled water units operating either in series or in parallel. Direct expansion refrigeration equipment is controlled either by unloading the compressor cylinders or by back pressure regulation in the refrigeration suction line.
Refrigeration capacity control is normally used in combination with bypass or reheat control. When used in combination, results are excellent. When used alone, results are not as effective. For example, temperature can be maintained reasonably well, but relative humidity will rise above design at partial load conditions, because the latent load may not reduce in proportion to the sensible load.

### 8.7.8. Partial load control

Generally, reheat control is more expensive but provides the best control of conditions in the space. Bypass control, volume control and refrigeration capacity control provide reasonably good humidity control in average or high sensible heat factor applications, and poor humidity control in low sensible heat factor applications. On-off control usually results in the least desirable method of maintaining space conditions. However, this type of control is frequently used for high sensible heat factor applications with reasonably satisfactory results.

#### 1.1.1.7

#### 1.1.1.8 TABLE 60 – EQUIVALENT EFFECTIVE SENSIBLE HEAT FACTORS FOR VARIOUS ELEVATIONS*

For use with sea level psychrometric chart or tables

<table>
<thead>
<tr>
<th>Effective Sensible Heat Factor from Air Conditioning Load Estimate</th>
<th>Elevation (m) and Barometric pressure (kPa) at Installation</th>
<th>Equivalent Effective Sensible Heat Factor Referred to a Sea Level Psychrometric Chart or Tables</th>
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</thead>
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<tr>
<td></td>
<td>500</td>
<td>1000</td>
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<tr>
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<td>0.53</td>
</tr>
</tbody>
</table>
Values obtained by use of equation:

\[ ESHF_e = \frac{1}{(p_1)(1 - ESHF) + 1} \]
\[ \frac{(p_0)(ESHF)}{p_0} \]

Where:
- \( p_0 \) = barometric pressure at sea level
- \( p_1 \) = barometric pressure at high elevation
- \( ESHF \) = ESHF obtained from air conditioning load estimate
- \( ESHF_e \) = equivalent ESHF referred to a sea level psychrometric chart or Table 60

Notes:
1. The required apparatus dewpoint for the high elevation is determined from the sea level chart or Table 59 by use of the equivalent effective sensible heat factor. The relative humidity and dry-bulb temperature must be used to define the room condition when using this table because the above equation was derived on this basis. The room wet-bulb temperature must not be used because the wet-bulb temperature corresponding to any particular condition, for example, 21°C DB, 40% RH at a high elevation is lower (except for saturation) than that corresponding to the same condition (21°C DB, 40% RH) at sea level. For the same value of room relative humidity and dry-bulb temperature, and the same apparatus dewpoint, there is a greater difference in moisture content between the two conditions at high elevation than at sea level. Therefore, a higher apparatus dewpoint is required at high elevation for a given effective sensible heat factor.

2. Air conditioning load estimate (see Fig. 47). The factors 1.20 and 3.0 on the air conditioning load estimate should be multiplied by the ratio of the barometric pressures \( \frac{P_1}{P_0} \). Using this method, it is assumed that the air quantity (L/s) is measured at actual conditions rather than at standard air conditions. The outdoor room moisture contents, gram per kilogram must also be corrected for high elevations.

3. Reheat – where the equivalent effective sensible heat factor is lower than the asterisked values in Table 59 reheat is required.

8.8. Psychrometric Formulae

Comment: - may need a couple of extra including Adjustable Sensible, latent and total heat

8.8.1. Air mixing equations

(Outdoor and Return Air)

\[ t_M = \frac{(\ell/S_{OA}) \times f_{OA} + (\ell/S_{RA}) \times f_{RM}}{\ell/S_{SA}} \]  
\[ (1) \]
\[ h_M = \frac{(\ell/S_{OA}) \times h_{OA} + (\ell/S_{RA}) \times h_{RM}}{\ell/S_{SA}} \]  
\[ (2) \]
\[ w_M = \frac{(\ell/S_{OA}) \times w_{OA} + (\ell/S_{RA}) \times w_{RM}}{\ell/S_{SA}} \]  
\[ (3) \]

8.8.2. Cooling load equations

ERTH = ERLH + ERSH  
\[ (4) \]
\[ (5) \]
\[ (6) \]
\[ (7) \]
\[ (8) \]
\[ (9) \]
\[ \text{RSH} = 1.20^+ \times \frac{L}{S_{SA}} \times (t_{RM} - t_{SA}) \quad (10) \]
\[ \text{RLH} = 3.0^+ \times \frac{L}{S_{SA}} \times (w_{RM} - w_{SA}) \quad (11) \]
\[ \text{RTH} = 1.19^+ \times \frac{L}{S_{SA}} \times (h_{RM} - h_{SA}) \quad (12) \]
\[ \text{RTH} = \text{RSH} + \text{RLH} \quad (13) \]
\[ \text{OASH} = 1.20 \times \frac{L}{S_{OA}} \times (t_{OA} - t_{RM}) \quad (14) \]
\[ \text{OALH} = 3.0 \times \frac{L}{S_{OA}} \times (w_{OA} - w_{RM}) \quad (15) \]
\[ \text{OATH} = 1.19 \times \frac{L}{S_{OA}} \times (h_{OA} - h_{RM}) \quad (16) \]
\[ \text{OATH} = \text{OASH} + \text{OALH} \quad (17) \]
\[ (\text{BF})(\text{OATH}) = (\text{BF})(\text{OASH}) + (\text{BF})(\text{OALH}) \quad (18) \]
\[ \text{ERSH} = 1.20 \times \frac{L}{S_{DA}^\dagger} \times (t_{RM} - t_{ADP})(1 - \text{BF}) \quad (19) \]
\[ \text{ERLH} = 3.0 \times \frac{L}{S_{DA}^\dagger} \times (w_{RM} - w_{ADP})(1 - \text{BF}) \quad (20) \]
\[ \text{ERTH} = 1.19 \times \frac{L}{S_{DA}^\dagger} \times (h_{RM} - h_{ADP})(1 - \text{BF}) \quad (21) \]
\[ \text{TSH} = 1.20 \times \frac{L}{S_{DA}^\dagger} \times (t_{EDB} - t_{LDB})^{**} \quad (22) \]
\[ \text{TLH} = 3.0 \times \frac{L}{S_{DA}^\dagger} \times (w_{EA} - w_{LA})^{**} \quad (23) \]
\[ \text{GTH} = 1.19 \times \frac{L}{S_{SA}^\dagger} \times (h_{EA} - h_{LA})^{**} \quad (24) \]

### 8.8.3. Sensible heat factor equations

\[ \text{RSHF} = \frac{\text{RSH}}{\text{RSH} + \text{RLH}} = \frac{\text{RSH}}{\text{RTH}} \quad (25) \]
\[ \text{ESHF} = \frac{\text{ERSH}}{\text{ERSH} + \text{ERLH}} = \frac{\text{ERSH}}{\text{ERTH}} \quad (26) \]
\[ \text{GSHF} = \frac{\text{TSH}}{\text{TSH} + \text{TLH}} = \frac{\text{TSH}}{\text{GTH}} \quad (27) \]

* RSH, RLH and GTH are supplementary loads due to duct heat gain, duct leakage loss, fan and pump power gains, etc. To simplify the various examples, these supplementary loads have *not* been used in the calculations. However, in actual practice, these supplementary loads should be used where appropriate. *Section 7* gives the values for the various supplementary loads. *Fig. 1, Section 1*, illustrates the method of accounting for these supplementary loads on the air conditioning load estimate.

† *Item H* of this section gives the derivation of these air constants.

‡ When no air is to be physically bypassed around the conditioning plant, \( \frac{L}{S_{DA}} = \frac{L}{S_{SA}} \).
8.8.4. Bypass factor equations

\[ BF = \frac{t_{LDB} - t_{ADP}}{t_{EDB} - t_{ADP}} \times (1 - BF) \]

\[ BF = \frac{w_{LA} - w_{ADP}}{w_{EA} - w_{ADP}} \times (1 - BF) \]

\[ BF = \frac{h_{LA} - h_{ADP}}{h_{EA} - h_{ADP}} \times (1 - BF) \]

8.8.5. Temperature equations at plant

\[ t_{EDB} = \left( \frac{i/S_{OA} \times t_{OA}}{i/S_{SA}} \right) + \left( \frac{i/S_{RA} \times t_{RM}}{i/S_{SA}} \right) \]

\[ t_{LDB} = t_{ADP} + BF (t_{EDB} - t_{ADP}) \]

\[ t_{EWB} \text{ and } t_{LWB} \text{ correspond to the calculated values of } h_{EA} \text{ and } h_{LA} \text{ on the psychrometric chart.} \]

\[ h_{EA} = \left( \frac{i/S_{OA} \times h_{OA}}{i/S_{SA}} \right) + \left( \frac{i/S_{RA} \times h_{RM}}{i/S_{SA}} \right) \]

\[ h_{LA} = h_{ADP} + BF (h_{EA} - h_{ADP}) \]

8.8.6. Temperature equations for supply air

\[ t_{SA} = t_{RM} - \frac{RSH}{1.20(i/S_{SA})} \]

8.8.7. Air quantity equations

\[ i/S_{DA} = \frac{ERSH}{1.20 \times (1 - BF)(t_{RM} - t_{ADP})} \]

\[ i/S_{DA} = \frac{ERLH}{3.0 \times (1 - BF)(w_{RM} - w_{ADP})} \]

\[ i/S_{DA} = \frac{ERTH}{1.19 \times (1 - BF)(h_{RM} - h_{ADP})} \]

\[ i/S_{DA \parallel} = \frac{TSH}{1.20(t_{EDB} - t_{LDB})} \]

\[ i/S_{DA \parallel} = \frac{TLH}{3.0(w_{EA} - w_{LA})} \]
\[ \frac{l_{/SA}}{L_{/SA}} = \frac{G_{TH}}{1.19(h_{/EA} - h_{/LA})} \]  \hspace{1cm} (41)

\[ \frac{l_{/SA}}{L_{/SA}} = \frac{R_{SH}}{1.20 \times (t_{RM} - t_{/SA})} \]  \hspace{1cm} (42)

\[ \frac{l_{/SA}}{L_{/SA}} = \frac{R_{LH}}{3.0 \times (w_{/RM} - w_{/SA})} \]  \hspace{1cm} (43)

\[ \frac{l_{/SA}}{L_{/SA}} = \frac{R_{TH'}}{1.20 \times (t_{RM} - t_{/SA})} \]  \hspace{1cm} (44)

\[ \frac{l_{/BA}}{L_{/BA}} = \frac{l_{/SA}}{L_{/SA}} - \frac{l_{/DA}}{L_{/DA}} \]  \hspace{1cm} (45)

Note: \( L_{/DA} \) will be less than \( L_{/SA} \) only when air is physically bypassed around the conditioning plant.

\[ \frac{l_{/SA}}{L_{/SA}} = \frac{l_{/OA}}{L_{/OA}} + \frac{l_{/DA}}{L_{/DA}} \]  \hspace{1cm} (46)

** When \( t_{M}, w_{M} \) and \( h_{M} \) are equal to the entering conditions at the cooling plant, they may be substituted for \( t_{/DB}, w_{/EA} \) and \( h_{/EA} \) respectively.

**8.8.8. Derivation of air constants**

\[ 1.20 = 1.02 \times \frac{1}{0.842} = 1.21, \text{say 1.20} \]

Where:

- 1.20 = specific heat of moist air at 21°C DB and 50% RH, \( \text{kJ/(kg DRY AIR)(°C)} \)
- 0.842 = specific volume of moist air at 21°C DB and 50% RH, \( \text{m}^3/\text{kg} \).

\[ 3.0 = \frac{1}{0.842} \times \frac{2500}{1000} = 2.97 \text{say 3.0} \]

Where 0.842 = specific volume of moist air at 21°C DB and 50% RH, \( \text{m}^3/\text{kg} \).

- 2500 = average heat removal required to condense one kilogram of water vapour from the room air, \( \text{kJ/kg} \)
- 1000 = gram per kilogram

\[ 1.19 = \frac{1}{0.842} = 1.19 \]

Where 0.842 = specific volume of moist air at 21°C DB and 50% RH, \( \text{m}^3/\text{kg} \).
### Nomenclature and SI Symbols

Needs checking, needs more non-SI units in Table 61, eg bars and cub.m/hr because these are often quoted by suppliers.

<table>
<thead>
<tr>
<th>NOMENCLATURE</th>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>solar altitude angle</td>
<td>( (\degree) )</td>
</tr>
<tr>
<td>A.C</td>
<td>alternating current</td>
<td>( (A) )</td>
</tr>
<tr>
<td>ADP</td>
<td>apparatus dewpoint</td>
<td>( (\degree) )</td>
</tr>
<tr>
<td>( e )</td>
<td>area, duct area, wall area, floor area, etc.</td>
<td>( (m^2) )</td>
</tr>
<tr>
<td>B</td>
<td>wall solar azimuth angle</td>
<td>( (\degree) )</td>
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<tr>
<td>BF</td>
<td>bypass factor</td>
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</tr>
<tr>
<td>BF (OALH)</td>
<td>bypassed outdoor air latent heat</td>
<td>( (W) )</td>
</tr>
<tr>
<td>BF (OASH)</td>
<td>bypassed outdoor air sensible heat</td>
<td>( (W) )</td>
</tr>
<tr>
<td>BF (OATH)</td>
<td>bypassed outdoor air total heat</td>
<td>( (W) )</td>
</tr>
<tr>
<td>Ch/h</td>
<td>Changes per hour</td>
<td></td>
</tr>
<tr>
<td>( C )</td>
<td>convective heat exchange</td>
<td>( (W/m^2) )</td>
</tr>
<tr>
<td>DB</td>
<td>dry-bulb</td>
<td>( (\degree) )</td>
</tr>
<tr>
<td>DP</td>
<td>dewpoint</td>
<td>( (\degree) )</td>
</tr>
<tr>
<td>( E_{sd} )</td>
<td>non-dimensional clothing insulation factor</td>
<td></td>
</tr>
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<td>ERLH</td>
<td>effective room latent heat load</td>
<td>( (W) )</td>
</tr>
<tr>
<td>ERSH</td>
<td>effective room sensible heat load</td>
<td>( (W) )</td>
</tr>
<tr>
<td>ERTH</td>
<td>effective room total heat load</td>
<td>( (W) )</td>
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<tr>
<td>ESHF</td>
<td>effective sensible heat factor</td>
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<tr>
<td>( ESHF_e )</td>
<td>equivalent effective sensible heat factor referred to a sea level psychrometric chart</td>
<td>( (W/m^2\cdot\degree) )</td>
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<tr>
<td>( f_i )</td>
<td>inside air film or surface conductance</td>
<td>( (W/m^2\cdot\degree) )</td>
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<tr>
<td>GSHF</td>
<td>grand sensible heat factor</td>
<td></td>
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<tr>
<td>GTH</td>
<td>grand total heat</td>
<td>( (W) )</td>
</tr>
<tr>
<td>GTHS</td>
<td>grand total heat supplement</td>
<td>( (W) )</td>
</tr>
<tr>
<td>( g )</td>
<td>glass</td>
<td></td>
</tr>
<tr>
<td>( h )</td>
<td>specific enthalpy</td>
<td>( (kJ/kg) )</td>
</tr>
<tr>
<td>( h_{ADP} )</td>
<td>apparatus dewpoint enthalpy</td>
<td>( (kJ/kg) )</td>
</tr>
<tr>
<td>( h_{EB} )</td>
<td>effective surface temperature enthalpy</td>
<td>( (kJ/kg) )</td>
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<tr>
<td>( h_{EA} )</td>
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<td>outdoor air enthalpy</td>
<td>( (kJ/kg) )</td>
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<tr>
<td>( h_{RA} )</td>
<td>room air enthalpy</td>
<td>( (kJ/kg) )</td>
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<tr>
<td>( h_{SB} )</td>
<td>supply air enthalpy</td>
<td>( (kJ/kg) )</td>
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<tr>
<td>( k )</td>
<td>conductivity</td>
<td>( (W/m\cdot\degree) )</td>
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<tr>
<td>L</td>
<td>duct length</td>
<td>( (m) )</td>
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<td>Lp</td>
<td>wall perimeter</td>
<td>( (m) )</td>
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<tr>
<td>( l_{RA} )</td>
<td>bypassed air quantity around apparatus</td>
<td>( (l) )</td>
</tr>
<tr>
<td>( l_{RA} )</td>
<td>dehumidified air quantity</td>
<td>( (l) )</td>
</tr>
<tr>
<td>( l_{OA} )</td>
<td>outdoor air quantity</td>
<td>( (l) )</td>
</tr>
<tr>
<td>( l_{RA} )</td>
<td>return air quantity</td>
<td>( (l) )</td>
</tr>
<tr>
<td>( l_{RA} )</td>
<td>supply air quantity</td>
<td>( (l) )</td>
</tr>
<tr>
<td>OALH</td>
<td>outdoor air latent heat</td>
<td>( (W) )</td>
</tr>
<tr>
<td>OASH</td>
<td>outdoor air sensible heat</td>
<td>( (W) )</td>
</tr>
<tr>
<td>OATH</td>
<td>outdoor air total heat</td>
<td>( (W) )</td>
</tr>
<tr>
<td>P</td>
<td>rectangular duct perimeter</td>
<td>( (m) )</td>
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<tr>
<td>( f_0 )</td>
<td>barometric pressure at sea level</td>
<td>( (kPa) )</td>
</tr>
<tr>
<td>( f_1 )</td>
<td>barometric pressure at high elevation</td>
<td>( (kPa) )</td>
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<tr>
<td>Q</td>
<td>heat flow rate</td>
<td>( (W) )</td>
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<tr>
<td>( Q' )</td>
<td>solar heat gain to space</td>
<td>( (W/m^2) )</td>
</tr>
<tr>
<td>( q )</td>
<td>combined heat transfer coefficient</td>
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<tr>
<td>( s_e )</td>
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<td>( (W/m^2\cdot\degree) )</td>
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<tr>
<td>( s_r )</td>
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<td>( (W/m^2\cdot\degree) )</td>
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<tr>
<td>Symbol</td>
<td>Description</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
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<tr>
<td>R</td>
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<td>m²·C/W</td>
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<td>RLH</td>
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<td>W</td>
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<td>W</td>
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<tr>
<td>RTH</td>
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<td>r</td>
<td>radiant heat exchange</td>
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<td>SHF</td>
<td>sensible heat factor</td>
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</tr>
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<td>Sat Eff</td>
<td>saturation efficiency of sprays</td>
<td>—</td>
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<tr>
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<td>shading device</td>
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<tr>
<td>t</td>
<td>temperature</td>
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<td>apparatus dewpoint temperature</td>
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</tr>
<tr>
<td>tDP</td>
<td>dewpoint temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tEB</td>
<td>entering air dry-bulb temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tES</td>
<td>effective surface temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tEW</td>
<td>entering water temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tEBW</td>
<td>entering air wet-bulb temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tDB</td>
<td>leaving air dry-bulb temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tDW</td>
<td>leaving water temperature</td>
<td>°C</td>
</tr>
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<td>tWB</td>
<td>leaving air wet-bulb temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tM</td>
<td>mixture of outdoor and return air dry-bulb temperature</td>
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<tr>
<td>tOA</td>
<td>outdoor air dry-bulb temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tSM</td>
<td>room dry-bulb temperature</td>
<td>°C</td>
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<td>tSA</td>
<td>supply air dry-bulb temperature</td>
<td>°C</td>
</tr>
<tr>
<td>ta</td>
<td>ambient temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tB</td>
<td>basement dry-bulb temperature</td>
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<td>ground temperature</td>
<td>°C</td>
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<tr>
<td>tO</td>
<td>operative temperature</td>
<td>°C</td>
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<tr>
<td>T</td>
<td>mean radiant temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tk</td>
<td>skin temperature</td>
<td>°C</td>
</tr>
<tr>
<td>ts</td>
<td>surface temperature</td>
<td>°C</td>
</tr>
<tr>
<td>U</td>
<td>transmission coefficient (U-value)</td>
<td>W/m²·K</td>
</tr>
<tr>
<td>V</td>
<td>air velocity through duct</td>
<td>m/s</td>
</tr>
<tr>
<td>WB</td>
<td>wet-bulb</td>
<td>°C</td>
</tr>
<tr>
<td>w</td>
<td>moisture content or specific humidity</td>
<td>g/kg</td>
</tr>
<tr>
<td>wADP</td>
<td>apparatus dewpoint moisture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>wEB</td>
<td>entering air moisture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>wES</td>
<td>effective surface temperature moisture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>wLA</td>
<td>leaving air moisture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>wM</td>
<td>moisture of outdoor and return air mixture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>wOA</td>
<td>outdoor air moisture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>wSM</td>
<td>room moisture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>wSA</td>
<td>supply air moisture content</td>
<td>g/kg</td>
</tr>
<tr>
<td>x</td>
<td>vertical distance</td>
<td>m</td>
</tr>
<tr>
<td>α</td>
<td>solar absorptivity</td>
<td>—</td>
</tr>
<tr>
<td>γ</td>
<td>air flow rate per unit length</td>
<td>ft/min</td>
</tr>
<tr>
<td>ΔTφ</td>
<td>equivalent temperature difference for latitude, month and time of day desired</td>
<td>°C</td>
</tr>
<tr>
<td>ΔTm</td>
<td>equivalent temperature difference for wall or roof exposed to the sun at 40° South and at desired time of day, corrected if necessary for design conditions</td>
<td>°C</td>
</tr>
<tr>
<td>Δt</td>
<td>equivalent temperature difference for wall or roof exposed to the sun at 40° South and at desired time of day, corrected if necessary for design conditions</td>
<td>°C</td>
</tr>
<tr>
<td>Δp</td>
<td>pressure difference across a window</td>
<td>Pa</td>
</tr>
<tr>
<td>Θ</td>
<td>window infiltration coefficient</td>
<td>—</td>
</tr>
<tr>
<td>λ</td>
<td>perimeter factor</td>
<td>—</td>
</tr>
<tr>
<td>ρ</td>
<td>reflectivity</td>
<td>—</td>
</tr>
<tr>
<td>Σ</td>
<td>“the summation of”</td>
<td>—</td>
</tr>
<tr>
<td>σm</td>
<td>maximum solar heat gain through glass for wall facing or horizontal for roofs, for January at 40° South latitude</td>
<td>—</td>
</tr>
</tbody>
</table>
\[ \sigma \text{ maximum solar heat gain through glass for wall facing or horizontal for roofs, for} \]
\[ \tau \text{ solar transmissibility} \]
\[ (W/m^2) \]

\textbf{SI Symbols}

- m metre (base unit of length)
- kg kilogram (base unit of mass)
- s second (base unit of time)
- A ampere (base unit of electric current)
- K° kelvin (base unit of thermodynamic temperature)
- rad° radian (base unit for plane angle)

\textbf{SI Prefixes for multiples and submultiples}

\[ \mu \text{— micro} = 10^{-6} \]
\[ m \text{— milli} = 10^{-3} \]
\[ k \text{— kilo} = 10^3 \]
\[ M \text{— mega} = 10^6 \]

\textbf{Derived SI Units with Special Names}

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Unit</th>
<th>Symbol</th>
<th>Derivation</th>
</tr>
</thead>
<tbody>
<tr>
<td>electrical potential</td>
<td>volt</td>
<td>V</td>
<td>W/A</td>
</tr>
<tr>
<td>energy</td>
<td>joule</td>
<td>J</td>
<td>Nm</td>
</tr>
<tr>
<td>force</td>
<td>newton</td>
<td>N</td>
<td>kg.m/s²</td>
</tr>
<tr>
<td>frequency</td>
<td>hertz</td>
<td>Hz</td>
<td>s⁻¹</td>
</tr>
<tr>
<td>heat flow</td>
<td>watt</td>
<td>W</td>
<td>J/s</td>
</tr>
<tr>
<td>heat quantity</td>
<td>joule</td>
<td>J</td>
<td>W</td>
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<tr>
<td>power</td>
<td>watt</td>
<td>W</td>
<td>J/s</td>
</tr>
<tr>
<td>pressure</td>
<td>pascal</td>
<td>Pa</td>
<td>N/m²</td>
</tr>
<tr>
<td>temperature</td>
<td>degree Celsius</td>
<td>°C</td>
<td>°C = K - 273.15</td>
</tr>
<tr>
<td>temperature interval</td>
<td>degree Celsius</td>
<td>°C</td>
<td>1 K = 1°C</td>
</tr>
<tr>
<td>work</td>
<td>joule</td>
<td>J</td>
<td>N.m</td>
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</table>

\textbf{Derived SI Units with Complex Names}

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<th>Symbol</th>
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<tr>
<td>acceleration</td>
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<td>m/s²</td>
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<tr>
<td>angular velocity</td>
<td>radian per second</td>
<td>rad/s</td>
</tr>
<tr>
<td>area</td>
<td>square metre</td>
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</tr>
<tr>
<td>caloric value</td>
<td>joule per cubic metre</td>
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</tr>
<tr>
<td>density</td>
<td>kilogram per cubic metre</td>
<td>kg/m³</td>
</tr>
<tr>
<td>intensity of heat flow</td>
<td>watt per square metre</td>
<td>W/m²</td>
</tr>
<tr>
<td>thermal conductance</td>
<td>watt per square metre degree Celsius</td>
<td>W/m²°C</td>
</tr>
<tr>
<td>thermal conductivity</td>
<td>watt per metre degree Celsius</td>
<td>W/m°C</td>
</tr>
<tr>
<td>thermal resistance</td>
<td>square metre degree Celsius per watt</td>
<td>m⁻²°C/W</td>
</tr>
<tr>
<td>thermal resistivity</td>
<td>metre degree Celsius per watt</td>
<td>m⁻²/W</td>
</tr>
<tr>
<td>velocity (speed)</td>
<td>metre per second</td>
<td>m/s</td>
</tr>
<tr>
<td>volume</td>
<td>cubic metre</td>
<td>m³</td>
</tr>
<tr>
<td>volume rate of flow</td>
<td>cubic metre per second</td>
<td>m³/s</td>
</tr>
<tr>
<td>water vapour permeability</td>
<td>microgram metre per newton second</td>
<td>μg.m/N.s</td>
</tr>
<tr>
<td>water vapour permeance</td>
<td>microgram per newton second</td>
<td>μg/N.s</td>
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*This unit is not used in this manual, see Table 61 for preferred non-SI unit.*
<table>
<thead>
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<th>Quantity</th>
<th>SI Unit</th>
<th>Non-SI Unit</th>
<th>Non-SI Symbol</th>
<th>Relationship</th>
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<tbody>
<tr>
<td>Angular measurement</td>
<td>radian</td>
<td>degree minute</td>
<td>(°)</td>
<td>1° = 1.74533 × 10⁻² rad</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(')</td>
<td>1' = 2.80888 × 10⁻⁶ rad</td>
</tr>
<tr>
<td>Angular velocity</td>
<td>radian per second</td>
<td>revolution per second</td>
<td>r/s</td>
<td>1 rev/s = 6.28319 rad/s</td>
</tr>
<tr>
<td>Temperature</td>
<td>kelvin</td>
<td>degree Celsius</td>
<td>°C</td>
<td>1°C = K - 273.15</td>
</tr>
<tr>
<td>Time</td>
<td>second</td>
<td>hour</td>
<td>h</td>
<td>1 h = 3.6 × 10⁻³ s</td>
</tr>
<tr>
<td>Volume</td>
<td>cubic metre</td>
<td>litre</td>
<td>l</td>
<td>1 l = 1 × 10⁻³ m³</td>
</tr>
</tbody>
</table>