Commercial Air Conditioning Systems

Julio O. Zeller

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Chapter 1

Air Conditioning

The American Society of Heating and Ventilating Engineers in 1929 defined comfort air conditioning as "The process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution." The same definition applies today, but there have been certain refinements as the art has progressed.

Comfort air conditioning or climate control, means the maintenance of those atmospheric factors affecting comfort. Specifically, it is the maintenance of the following variables, all within well-defined limits, and by systems which do not contribute an objectionable noise:

- The desired temperature
- An acceptable humidity
- Minimal atmospheric particulates, including pollens and bacterias
- An acceptable odor level
- A uniform air pattern and air motion

Eliminating thermal stress, or "heat stress", results in a more comfortable environment which, whether the task be mental or physical, is conducive to efficient production. By this means, the combination of proper temperature, humidity, air motion and radiant temperature are so balanced that the body can dissipate, at the correct rate, the excess heat generated.

The ambient temperature affects the heat dissipation rate due to conduction. Ambient relative humidity affects heat being dissipated by evaporation from the body. The ambient air motion affects both the moisture dissipation rate and the dry heat dissipated by convection. The surface temperature of the partitions, walls, floors, etc. affects the radiant heat losses from the body. When the outside wall and glass are at a lower temperature than the interior surfaces, a contrast in radiant loss is detectable and a feeling of discomfort is noticed.
1.1 Refrigeration Equipment/Cycle

Vapor-Compression refrigeration systems are the most common systems in use today. In considering the steady-state operation of the vapor-compression system illustrated in Figure 1.1. Shown on this figure are the principal work and heat transfers, which are positive in the direction of the arrows. Kinetic and potential energy changes can be ignored in the following analyses of the components.

1.1.1 Principal Work and Heat Transfers

From Figure 1.1, A slightly superheated vapor is compressed in process 1–2, as it goes through a compressor. The compression process takes the gas from low pressure and temperature to high pressure and temperature. Supreheating ensures that the compressor will not handle any liquid that might wash lubricants from surfaces, cause erosion in an axial-flow compressor, or produce excessive pressure in a reciprocating compressor by interfering with the piston at top-dead-center. Work energy $Q_W$ is required, and is usually provided by an electric motor. The high pressure gas flows through piping to a condenser where the gas condenses (the refrigerant is selected with properties which allow it to condense at the temperature and pressure in the condenser) when heat ($Q_c + Q_w$) is removed by heat transfer into a heat sink, usually water or air. The high-pressure liquid is then carried through a pressure reducing device or expansion valve where the liquid expands to the evaporator pressure while being throttled. Temperature decreases through this process because those molecules with sufficient energy move from the liquid to the vapor phase across the liquid-vapor interface, leaving the lower-energy, cooler molecules behind; the new vapor molecules use up their excess of energy in moving across the interface and thus are cooled as well. The refrigerant exits as a two-phase liquid-vapor mixture. The low pressure liquid-vapor mixture is finally passed through the evaporator. At the lower pressure the liquid tend to fully evaporate, by removing the heat of vaporization ($Q_v$) from its surroundings. The cold vapor is then returned to the compressor to be recycled and reused.
Figure 1.1: Equipment Layout of a Two-phase Mechanical Refrigeration Cycle
1.1.2 Performance of Vapor-Compression Systems

If it is assumed that there are no irreversibilities throughout the whole system, frictional pressure drops are ignored, stray heat transfers, the refrigerant flows at constant pressure through the two heat exchangers, and the compression process is isentropic. Under these assumptions, the ideal vapor-compression refrigeration cycle 1-2,3-4-1 on the \( T - s \) diagram of Figure 1.2. The cycle consists of the following processes:

- **Process 1-2**: Isentropic compression of the refrigerant from state 1 to the condenser pressure at state 2.

- **Process 2-3**: Heat transfer from the refrigerant as it flows at constant pressure through the condenser. The refrigerant exits as a liquid at state 3.

- **Process 3-4**: Throttling process from state 3 to a two-phase liquid-vapor mixture at 4.

- **Process 4-1**: Heat transfer to the refrigerant as it flows at constant pressure through the evaporator to complete the cycle.

The above processes are all reversible, with the exception of *Process 3-4*. Although it has the irreversible process, this system can be considered as ideal. The operating temperatures of the vapor-compression refrigeration cycle are established by the temperature regions as shown in Figure 1.2. The refrigerant temperature in the condenser must be greater than \( T_h \), and the refrigerant temperature in the evaporator must be less than \( T_c \).

The \( T - s \) diagram of Figure 1.2 also illustrates a departure from actual system performance from that of the ideal cycle, with cycle 1-2,3-4-1. This departure is due to the internal irreversibilities which are present during the compression stage *Process 1-2*. The dashed line is used to show the increase in specific entropy that would accompany an adiabatic irreversible compression.

Comparing cycle 1-2,3-4-1 with the corresponding ideal cycle 1-2,3-4-1, we see that the refrigeration capacity is the same for each, but the work input is greater for the case of irreversible compression than in the ideal cycle. Accordingly, the coefficient of performance is of cycle 1-2,3-4-1 is less than that of cycle 1-2,3-4-1. The effect of irreversible compression can be accounted for by using the isentropic compressor efficiency, which is given by:

\[
\eta_c = \frac{(\frac{W_c}{m})_s}{(\frac{W_c}{m})_{is}} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1.1}
\]

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Figure 1.2: T-s diagrams of Vapor-Compression Refrigeration Cycles

Figure 1.3: T-s diagrams for compression of a gas
Figure 1.4: P-h diagrams of Vapor-Compression Refrigeration Cycles

The $P-h$ (pressure-enthalpy) diagram of Figure 1.4 illustrates Refrigerant 12 (most commonly known as Freon 12) acts in a vapor compression cycle. As in the $T-s$ diagram, the most important lines illustrate the saturated liquid and the saturated vapor. The condition of the compressed liquid can be advantageously shown in this type of diagram, between $A$ and $A'$. If the temperature of the liquid is constant there will be no significant change in the enthalpy of the compressed liquid, like there will be when the pressure of the liquid varies. However there is a slight increment in the enthalpy of the compressed liquid as the pressure is increased. Luckily in load calculations these changes are rather small and can be unappreciated.

The pressure-enthalpy diagrams are very important in determining the loads in refrigeration processes; mainly because in these diagrams the changes in enthalpy can be read easily and changes which represent added heat, or the work received (as in adiabatic compression). However, the refrigeration cycle shows only two pressures:

- The pressure involved in the evaporator $Process\ D-A$.

  This pressure is associated with the availing refrigeration.

- The pressure involved in the condenser $Process\ B-C$.  

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This pressure is associated with the phase of dissipating heat from the system.

Both pressures are more easily demonstrated in the pressure-enthalpy diagram.

### 1.1.3 Compressors

Compressors differ from fans in their ability to move gases across a larger pressure difference. While density of the handled fluid remains almost constant in both fans and pumps, it increases substantially across a compressor. The compression process for a gas is shown in Figure 1.3.

Also in order to maintain a determined pressure in the evaporator, the compressor must extract the vapor as soon as it is being formed. If the load in the evaporator is low (the difference in the temperature between the conditioned space and the evaporator is small) only a small amount of refrigerant will have to be evaporated, and the suction from the compressor may cause a reduction in the pressure of the evaporator. This drop in pressure will continue until the difference in temperature between the conditioned space and the evaporator is justly proportional to generate enough vapor to allow for an effective displacement in the piston of the compressor. If in any other part of the cycle, the difference in temperature is great (excessive load), then vapor will be generated extremely fast at a very high temperature in the evaporator, and the compressor may be overcharged. The characteristics in the thermal load transfer should be such that do not provoke the compressor to overcharge.

Form Figure 1.3, the initial conditions are called state 1 and the final conditions are termed state 2. This adiabatic compression process is isentropic resulting in a temperature $T_{2s} > T_1$, with work continuously being done on the gas. In contrast an adiabatic compression has increasing entropy and results in a higher temperature $T_2 > T_{2s}$. However a minimum amount of work is required, considering that it is being done isentropically (at constant entropy). Isentropic compression is impossible to be obtained with the equipment which has already been developed, but with it there is a base for calculating the ideal work for different conditions of operations. Under this basis real work can be estimated. The entropy of a vapor can be obtained from property tables and also from pressure-enthalpy diagrams.

The theoretical work of compression (or isentropic compression) can be found using either the symbols from Figure 1.2 or those from Figure 1.4, and can be found as:

$$W_T = (h_D - h_C)_s$$  \hspace{1cm} (1.2)

$$= (h'_D - h'_C)_s$$  \hspace{1cm} (1.3)

Where:
\[ h_D = \text{the enthalpy at the discharge conditions} \]
\[ h_C = \text{the enthalpy of the vapor which reaches the compressor} \]
assuming that the entropy remains constant during compression.

### 1.1.4 Condenser

Condensers whether, it employs water or air, should ideally eliminate the heat from the refrigerant to change the superheated vapor that is discharged from the compressor, and convert this vapor into a saturated liquid (or subcooled). The heat eliminated in the condenser is the same as the heat absorbed at lower temperatures, such as when in the evaporator, and the heat equivalent to the work put into the vapor in the compressor.

### 1.1.5 Expansion Valve

The liquid refrigerant, after it has left the condenser, it enters the expansion valve. This valve serves the purpose of controlling the flow and the drop in pressure of the refrigerant, the temperature and pressure at the conditions of the condenser and the evaporator. The throttling process is an adiabatic process without any work being produced. However, a certain amount of liquid refrigerant is transformed into a gas. In the Temperature-entropy diagram, the expansion valve function is shown in Process 3-4, and in the Pressure-enthalpy diagram, the expansion valve function is shown by the Process A-B.

### 1.1.6 Evaporator

In the evaporator, the liquid from the expansion valve is changed into vapor, while it absorbs heat from the conditioned space. The heat absorbed appears as an increment in the enthalpy of the refrigerant. The vapor that reaches the can be saturated dry as in C(Figure 1.4) or superheated as in C'(Figure 1.4).
Chapter 2

Psychrometrics

2.1 Psychrometry

Psychrometry consists of the interactions between heat, moisture and air. It is basically the study of moist air. As air temperature rises, its capacity to hold moisture rises also; and warmer air becomes less dense. This makes moisture a very influential factor for heat gain, both for comfort and in calculations. The knowledge of systems consisting of dry air and water vapor is essential for the design and analysis of air conditioning devices, cooling towers, and industrial processes requiring close control of the vapor content in air. Air moisture and heat interactions are rather complex; Fortunately, these interactions can be combined in a single chart (see figure 1). However before explaining the details of how to use the chart, some terms, definitions, and principles used in the study of systems consisting of dry air and water must be introduced.

2.1.1 Moist Air

The term moist air, refers to a mixture of dry air and water vapor in which the dry air is treated as if it were a pure component. However it should be noted that the overall mixture and each mixture component behave as ideal gases at the states under present considerations. Thus:

\[ p = \frac{nRT}{V} = \frac{m \left( \frac{R}{M_a} \right) T}{V} \]  \hspace{1cm} (2.1)

And since the mixture pressure is the sum of the partial pressures of the dry air and water vapor. Thus:

\[ p = p_a + p_v = \frac{n_a RT}{V} + \frac{n_v RT}{V} = \frac{m_a \left( \frac{R}{M_a} \right) T}{V} + \frac{m_v \left( \frac{R}{M_v} \right) T}{V} \]  \hspace{1cm} (2.2)
At the state of water vapor shown in figure 2, fixed by the partial pressure $p_v$ and the mixture temperature $T$, the vapor is superheated. When the partial pressure of the water vapor corresponds to the saturation pressure of water at the mixture temperature, $p_g$, the mixture is said to be saturated. Saturated air is a mixture of dry air and saturated water vapor. The amount of water vapor in dry air generally varies.

![T-v diagram for water vapor in an air-water mixture](image)

Figure 2.1: T-v diagram for water vapor in an air-water mixture

### 2.1.2 Humidity Ratio and Relative Humidity

The composition of a given moist air sample can be indicated by the means of the humidity ratio $\omega$, defined as the ratio of the mass of the water vapor to the mass of dry air, and referred to also as the specific humidity. The humidity ratio can be expressed as:

$$\omega = \frac{m_v}{m_a} = \frac{M_v p_v V}{M_a p_a V} = \frac{M_v p_v}{M_a p_a}$$  \(2.3\)

Also noting that the ratio of molecular weight of water to that of dry air is approximately 0.622, so the expression can then become:

$$\omega = 0.622 \left( \frac{p_v}{p - p_v} \right)$$  \(2.4\)

The composition of a moist air sample can also be described in terms of relative humidity $\phi$, defined as the ratio of mole fraction of water vapor $y_v$ in a given moist air sample to the mole fraction $y_{v, sat}$ in a saturated moist air sample at the same mixture temperature and pressure, or as the ratio of actual water vapor pressure...
in air to the pressure of saturated water vapor in air at the same temperature. Thus

\[
\phi = \left( \frac{y_v}{y_{v,\text{sat}}} \right)_{T,p} = \left( \frac{p_v}{p_g} \right)_{T,p}
\]

(2.5)

The values of U, H, and S for moist air can be found by adding the contribution of each component at the conditions at which the component exists in the mixture.

\[
H = H_a + H_v = m_a h_a + m_v h_v
\]

(2.6)

\[
\frac{H}{m_a} = h_a + \frac{m_v}{m_a} h_v = h_a + \omega h_v
\]

(2.7)

The enthalpies of the dry air and water vapors are evaluated at the mixture temperature.

### 2.1.3 Dew Point

An important aspect for the behavior of moist air is the occurrence of partial condensation of the water vapor when the temperature is lowered. In order to study this, consider a sample of moist air within a closed system, at constant pressure. The property diagram will locate the states of the water vapor.

![T-v diagram for states of water for moist air cooled at constant mixture pressure](image)

Figure 2.2: T-v diagram for states of water for moist air cooled at constant mixture pressure

- **Process 1-d**: Both the system pressure and the composition of the moist air remain constant. The partial pressure of the water vapor would also remain
constant. The water vapor will then cool at constant $p_v$ until it reaches the *dewpoint*.

- *Process d–2*: The system is now cooled below the dewpoint temperature. Some of the water vapor condenses. The vapor now has a partial pressure equal to the saturation pressure $p_{g2}$ corresponding to this temperature.

- *Process 2–3*: The condensate has now become a saturated liquid.

### 2.1.4 Mass and Energy Applications Psychrometrics

Devices accomplishing air-conditioning processes such as heating, cooling, humidification, or dehumidification are analyzed within a control volume basis. An example of this can be given by the system shown below, which has two inlets and only one exit, while maintaining a steady state. Moist air enters at 1 and exits at 2 and a water-only stream enters at 3 (Either a liquid or a vapor). Heat transfer may also occur between the control volume and its surroundings. This would happen at the rate $Q_{cv}$ and depending upon the application it could have either a positive, negative or zero value.

![Figure 2.3: System for Conditioning Moist Air](image)

At steady state, the amounts of dry air and water vapor within the control volume do not vary. Therefore for each component the total incoming and outgoing
mass flow rates remain the same.

\[ m_{a1} = m_{a2}(\text{dry air}) \]
\[ m_{v1} + m_w = m_{v2}(H_2O) \]

Or in terms of humidity ratios

\[ m_w = m_a(\omega_2 - \omega_1) \]

When the water is added \( \omega_2 \) becomes greater than \( \omega_1 \).

If for the Energy Conservation equation, we regard the entering and exiting moist air streams as ideal gas mixtures of dry air and water vapor, and ignore all the kinetic and potential energy effects, we yield this energy balance equation.

\[
\frac{dE}{dt} = Q - W + (m_a h_{a1} + m_{v1} h_{v1}) - (m_a h_{a2} + m_{v2} h_{v2}) + m_w h_w
\]

(2.8)

or

\[
0 = Q - W + m_a[(h_{a1} + \omega_1 h_{v1}) - (h_{a2} + \omega_2 h_{v2}) + m_w h_w
\]

(2.9)

knowing that \((h_a + \omega h_v) = h\) of mixture

\[
0 = Q - W + m_a(h_{a1} - h_{a2}) + \omega_1 h_{v1} - \omega_2 h_{v2} + (\omega_2 - \omega_1) h_w
\]

(2.10)

\(h_{a1} - h_{a2} = c_{pa}(T_1 - T_2)\)

\(h_{v2} & h_{v1}\) — use values of \(h_g\)

In most cases there is no work, except for flow work where matter crosses the boundary. Other simplifying assumptions used when analyzing the class of devices under present consideration.

### 2.1.5 Adiabatic-Saturation

The humidity ratio \(\omega\) of an air water mixture can be determined by knowing: pressure \(p\), the temperature \(T\), and the adiabatic-saturation temperature \(T_{as}\). These quantities are related by the equation:

\[
\omega = \frac{h_a(T_{as} - h_a(T)) + \omega' h_g(T_{as}) - h_f(T_{as})}{h_g(T) - h_f(T_{as})}
\]

(2.11)

where:

\[ h_a(T_{as}) - h_a(T) = c_{pa}(T_{as} - T) \]

and:

\[ \omega' = 0.622 \frac{p_g(T_{as})}{p - p_g(T_{as})} \]

where \(p_g(T_{as})\) is the saturation pressure at the adiabatic saturation temperature and \(p\) is the mixture pressure.
Adiabatic Saturator

An adiabatic saturator is a two inlet, single exit device through which moist air passes. It is assumed that the device operates at steady state, with negligible heat transfer with its surroundings.

When the air-water vapor mixture of humidity ratio $\omega$ enters the saturator at pressure $p$ and Temperature $T$. As the mixture passes through the device it crosses over a pool of water which if the mixture is not saturated, some of the water evaporates. The energy required to evaporate the water, comes from the moist air. At steady state, the mass flow rate of the makeup water is the difference between the exiting and entering vapor flow rate. Therefore:

$$m_{a1} = m_{a3}$$
$$m_{v1} + m_{w2} = m_{v3}$$

Conservation of Energy

$$m_{a1} h_{a1} + m_{v1} h_{v1} + m_{w2} h_{w2} = m_{a3} h_{a3} + m_{v3} h_{v3}$$  \hspace{1cm} (2.12)

$$\omega_1 = \frac{m_{v1}}{m_{a1}}$$
$$\omega_3 = \frac{m_{v3}}{m_{a3}}$$
$$\omega_3 - \omega_1 = \frac{m_{w2}}{m_{a1}}$$
And

$$h_{a1} + \omega_1 h_{v1} + (\omega_3 - \omega_1) h_{w2} = h_{a3} + \omega_3 h_{v3}$$  \hspace{1cm} (2.13)
\[
\omega_1 = \frac{(h_{a3} - h_{a1}) + \omega_3(h_{v3} - h_{w2})}{h_{v1} - h_{w2}} = \frac{c_{pa}(T_3 - T_1) + \omega_3h_{fg3}}{h_g1 - h_f2}
\] (2.14)

Several assumptions accompany these expressions: Each moist air stream is modeled as an ideal gas. There is no work or changes in kinetic and potential energy. Also these relationships are not limited to this type of system or even control volumes. This relationship allows the humidity ratio \( \omega \) to be determined for any moist air mixture for which the pressure \( p \), temperature \( T \), and adiabatic-saturation temperature \( T_{ad} \) are known.

### 2.1.6 Wet-Bulb and Dry-Bulb Temperatures

The water vapor contained in air, is under the superheated condition for any relative humidity less than 100\%. This means that any air with relative humidity less than 100\% can be cooled up to the temperature to which it can be cooled, without causing precipitation of its humidity.

The adiabatic-saturation temperatures can be determined indirectly by finding Dry-Bulb and Wet-Bulb temperatures. Dry-Bulb temperatures are those read from an ordinary thermometer. However when the bulb of a thermometer is covered with a wet wick and is moved across air which is not saturated, the water evaporates in proportion to the capacity that the air can absorb humidity. This makes the indicated temperature less than that of the Dry-Bulb. Finally the equilibrium temperature obtained by the thermometer is known as the Humid-Bulb temperature.

### 2.1.7 Psychrometric Charts

The psychrometric chart is a graphical representation of psychrometric properties. There are many charts available from various equipment manufacturers and other sources. The ASHRAE chart is the most commonly used.

The basic coordinate grid lines of the ASHRAE chart are enthalpy, which slopes up to the left, and the humidity ratio which runs horizontal. Dry-Bulb lines are uniformly spaced and approximately vertical; the slope of the lines changes across the chart. Wet-Bulb lines slope similarly to enthalpy lines, but the slope increases as the temperature increases and no Wet-Bulb line is parallel to an enthalpy line. This is because of the heat added to the mixture by the moisture as it changes from dry to saturated air. Relative humidity lines are curved, with the saturation line defining the upper boundary of the chart.
2.1.8 Psychrometric Factors

SENSIBLE HEAT FACTOR

The thermal properties of air can be separated into latent and sensible heat. Sensible Heat Factor is the ratio of sensible heat to total heat, where total heat is both latent and sensible heat. The ratio is expressed as:

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

- SHF = sensible heat factor
- SH = sensible heat
- LH = latent heat
- TH = total heat

ROOM SENSIBLE HEAT FACTOR (RSHF)

Room Sensible Heat Factor is the ratio of room sensible and room latent heat. This ratio is expressed as:

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

20
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 points (1 - 2) be connected with a straight line.

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 loads within the space \( \Delta h_s(\text{sensible heat}) \) and \( \Delta h_l(\text{latent heat}) \). 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.

**GRAND SENSIBLE HEAT FACTOR (GSHF)**

Grand Sensible Heat Factor is the ratio of the total sensible heat to the grand total heat load that the conditioning apparatus must handle, including the outdoor air heat loads. This ratio is expressed as:

\[
GSHF = \frac{TSH}{TLH + TSH} = \frac{TSH}{GTH}\tag{2.17}
\]
The air which is passing through the conditioning apparatus increases or decreases in temperature and/or moisture content. The amount of rise or fall is determined by the total sensible and latent heat load that the conditioning apparatus must handle. The condition of the air entering the apparatus (mixture condition of outdoor and returning room air) and the condition of the air leaving the apparatus can be plotted on the psychrometric chart and connected by a straight line (1 - 2).

Figure 2.7: GSHF Line Plotted Between Mixture Conditions to Apparatus and Leaving Condition from Apparatus

The (1 - 2) line represents the psychrometric process of the air as it passes through the conditioning apparatus, 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 apparatus must handle. This being illustrated by $\Delta h_s$ (sensible heat) and $\Delta h_l$ (latent heat).
REQUIRED AIR QUANTITY

The air quantity required to offset simultaneously the room sensible and latent loads and the air quantity required through the apparatus 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, represents the condition of the supply air to the space. It is also the condition of the air leaving the apparatus. However, this neglects fan and duct heat gain, duct leakage losses, etc. In actual practice, these heat gains are taken into account in estimating the cooling load. Therefore, the temperature of the air leaving the apparatus is not necessarily equal to the temperature of the air supplied to the space.

The figure below illustrates what actually happens when these supplementary loads are considered in plotting the RSHF and GSHF lines.

Figure 2.8: RSHF and GSHF Line Plotted with Supplementary Load Line

Point (1) is the condition of air leaving the apparatus and point (2) is the condition of supply air to the space. Line (1 - 2) represents the temperature rise of the air stream resulting from fan horsepower and heat gain into duct. The air quantity required to satisfy the room load may be calculated from the following
The required air quantity through the air conditioning apparatus, to satisfy the total air conditioning load (including the supplementary loads) may be calculated from the following equation:

\[ cf_{msa} = \frac{RSH}{1.08(t_{rm} - t_{sa})} \]  

The air quantity required through the apparatus, neglecting leakage losses. The above equations contain the terms:

- \( t_{rm} \) which is the room dry-bulb temperature
- \( t_{sa} \) which is the supply air dry-bulb temperature
- \( cf_{msa} \) which is the supply air quantity
- \( t_m \) which is the mixture of outdoor and return air dry-bulb temperature
- \( t_{lbd} \) which is the leaving dry-bulb temperature
- \( cf_{mda} \) which is the dehumidified air quantity

With the exception of an all outdoor air application, the term \( t_m \) can only be determined by trial and error.

In well designed, tight systems the difference in supply air temperature and the condition of the air leaving the apparatus \( (t_{sa} - t_{lbd}) \) is usually no more than a few degrees.

The difference in temperature between the room and the air supply to the room determines the amount of air required to satisfy the room sensible and 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; if it is assumed that the available conditioning equipment is able to take the air to 100% saturation.
Chapter 3

Air Handling/Transport Systems

3.1 Air Handling Systems

Air-Conditioning, as has been established requires control of air temperature, humidity, cleanliness, and distribution. It then seems natural that an Air Handling Unit (AHU) of some kind is an essential part of air-conditioning systems. The purpose of the AHU, is providing air at a quantity, temperature, and humidity which can offset the sensible and latent heat gains of the space, while maintaining the required temperature and humidity in the space. Suppose for example we take a specific air conditioned room with the following design criterias:

1. Cooling design room condition is 58 °F
2. A load of 120,000 btuh sensible heat
3. A load of 30,000 btuh latent heat
4. Relative Humidity of 50
5. A change in temperature between outside and inside of 20 °F
6. Supply air at 78 °F

The Design room cfm will be:

\[
\text{cfm} = \frac{GSH}{\Delta T \times 1.08} \]

\[
= \frac{120,000}{20 \times 1.08} \]

\[
= 5555 \text{ cfm}
\]

25
The specific humidity $\omega$ may also be calculated:

$$\Delta \omega = \text{Latent heat} \times \frac{1}{\text{cfm}} \times \frac{1}{\nu_a} \times \frac{1}{h_w} \quad (3.2)$$

$$= \frac{30,000 \text{ btu}}{\text{hr}} \times \frac{\text{min}}{\text{hr}} \times \frac{1}{\text{lb}_w}$$

$$= 0.0011 \frac{\text{lb}_w}{\text{lb}_a}$$

The point defined by these two differential values can be plotted on a psychrometric chart. The validity of the point must be verified, based on the cooling capabilities of the coil, and the AHU arrangements. For a draw-through arrangement, that is with the supply fan past the cooling coil, the supply air temperature will be greater than the "coil leaving" temperature, because of the heat added by the fan work. For this example if a 5hp motor is required, the temperature rise will be:

![Psychrometric chart](image)

**Figure 3.1:** Psychrometric chart, for draw-through AC process

$$\Delta T = \frac{P}{\text{hp hr cfm}}$$

$$= 0.0011 \frac{\text{lb}_w}{\text{lb}_a}$$

(3.3)
\[
\begin{align*}
&= 5 \text{ hr cfm °F} \\
&= 2.1 \text{ °F}
\end{align*}
\]

Then a coil leaving condition of about 55.9°F db and 55.5°F wb can be plotted, and this would then become valid. For a blow-through arrangement, the fan work

\[\begin{align*}
\text{Fan Work} &= \\
\Delta T &= \\
\Delta \omega &= \\
\text{Outside (Cooling)} &= \\
\text{Mixed Air} &= \\
\text{Room (Cooling)} &= \\
\text{Enter Cooling Coil} &= \\
\text{Leave Coil (cooling)} &= \\
\text{Room (Cooling)} &= \\
\text{Outside (Cooling)} &= \\
\text{Mixed Air} &= \\
\text{Enter Cooling Coil} &= \\
\text{Leave Coil (cooling)} &= \\
\text{Room (Cooling)} &= \\
\text{Outside (Cooling)} &= \\
\text{Dry Bulb Temperature} &= \\
\text{Humidity Ratio} &= \\
\end{align*}\]

Figure 3.2: Psychrometric chart, for blow-through AC process

causes an increase in the mixed-air temperature before the air goes through the cooling coil, and the process is as in figure 2. For this case, it is necessary to increase the supply of air \(\Delta T\) to 22°F to get a valid “coil leaving” condition. This will reduce the air quantity to 5050 cfm and will require more care in the air distribution system to avoid cold air spillage and drafts. The \(\Delta \omega\) will be greater due to the reduction in cfm. Humidity control is not always required, but some upper limit will be inherent in any refrigeration-type cooling process-chilled water, brine, or direct expansion.
3.1.1 Humidity Control

Any refrigeration-type cooling coil will provide some high limit control of humidity, intentionally or otherwise. Lower humidities may be obtained by means of adsorption type dehumidifiers. Humidity may be added in several ways. Controlled dehumidification always requires the use of extra energy.

Dehumidification by Refrigeration

This process requires a cooling coil utilizing chilled water, brine or refrigerant. The air passing over the coil is cooled to a dew point corresponding to the desired relative humidity and space dry-bulb temperature. For example to maintain a space at a temperature of about 70°F db and 40% relative humidity, a coil DP of about 43°F would be required. To obtain this, it would be necessary to use brine at about 36°F entering. Chilled water cannot safely be produced at this low temperature. Direct expansion could also be a source for dehumidification, as long

Figure 3.3: Psychrometric chart, for dehumidification by refrigeration
as it has an appropriate suction temperature and superheat. Care must be taken so as not to produce coil icing.

**Chemical Dehumidification**

A chemical dehumidifier utilizes an adsorbent, such as silica gel, to remove unwanted moisture from the air. The continuous process is shown in figure Trays of silica gel rotate through the conditioned air stream-where moisture is adsorbed—and then through a hot regeneration air stream where the gel is dried. The adsorption process generates some heat and additional heat is carried over from regeneration so that the process results in dryer, but hotter air as shown in the psychrometric chart. The air must then be recooled.

### 3.1.2 Outside Air Quantity Control

Outside air “economy cycle” control. A “fixed” outside air quantity may be required to satisfy exhaust or ventilation requirements. This is provided as shown by: With
100% outside air is preferable to provide a time delay in the supply-fan motor-starter circuit, so the damper will be at least 50% open before the fan starts. It is possible to control outside air and return air quantities to match a varying exhaust. The space being exhausted is usually controlled at lower pressure than an adjacent space. The real problem here is accurate sensing of the very low pressure differences used, while taking into account wind and stack effects. One of the better methods presently used is to provide an orifice in the wall between two spaces, in which is located a device for measuring velocity (*hot-wire anemometer or fluidic sensor*). Exhaust and make-up air quantities can then be adjusted to maintain a velocity corresponding to the desired pressure difference.
3.1.3 Smoke Control

The use of environmental air-handling systems for primary or supplementary smoke control is neither simple nor economical. There is a lack of good data on the design of adequate smoke-control systems, and code requirements are often vague and conflicting.

3.2 Air Transport Systems

This section describes available central station apparatus equipment and recommends suitable application of the various components.

The air-handling apparatus should be centrally located to obtain a minimum first cost system. In a few instances, however, it may be necessary to locate the apparatus, refrigeration machine, and cooling tower in one area, to achieve optimum system cost. When the three components are grouped in one area, the cost of extra ductwork is offset by the reduced piping cost. In addition, when the complete system becomes large enough to require more than one floor becomes practical. This design is often used in large buildings. The upper floor equipment is used for the lower 20 to 30 floors.

Occasionally a system is designed requiring a grouping of several units in one location, and the use of a single unit in a remote location. This condition should be carefully studied to obtain the optimum coil selection-versus-piping cost for the remotely located unit. Often, the cost of extra coil surface is more than offset by the lower pipe cost for the smaller water quantity resulting when the extra surface coil is used.

Air-conditioning practice includes only five basic AHU arrangements, although there are many variations on these basic concepts. However these are:

1. Single Zone
2. Multizone
3. Double-Duct
4. Variable Volume (VAV)
5. Induction

(see Appendix X)

3.2.1 Outdoor Air Louvers and Screen

Outdoor air louvers minimize the entry of snow and water into equipment. It is impossible to eliminate all the moisture with vertical louvers, and this is usually
not necessary. The screen is added to arrest most foreign materials such as trash and birds. Often the type of screen required and the mesh are specified by codes.

The screen and louver is located sufficiently above the roof to minimize the pickup of roof dust, a minimum of 2.5’ is recommended for most areas. In some locations, doors are added outside the louver for closure during extreme inclemetn weather such as hurricanes.

It is best to locate the outdoor air louver in such a manner that cross contamination from exhaust fan to louver does not occur, specifically toilet and kitchen exhaust. In addition, the outdoor air intake is located to minimize pulling air over a long stretch of roof since this increases the outdoor air load during the summer.

**Louver Dampers**

The louver damper is used for three important functions in the air handling apparatus:

1. to control and mix outdoor and return air
2. to bypass heat transfer equipment
3. to control air quantities handled by the fan

A double acting damper is used where control of air flow is required. This arrangement is superior since the air flow is throttled more or less in proportion to the blade position, whereas the single action type damper tends to divert the air and does little or no throttling until the blades are nearly closed.

Outdoor and return air dampers are located so that good mixing of the two air streams occurs. On installations that operate 24 hours a day and are located in amild climate, the outdoor damper is occasionally omitted. With the fan operating and the damper fully closed, leakage cannot be completely eliminated.

**3.2.2 Air Cleaning Equipment**

The control of air-purity consists of reducing or eliminating unwanted particulate or gaseous matter from the air being supplied to a space. This is a function of the air conditioning system. However, normal applications are concerned with particulate matter only.

Effectively applied air cleaners can materially reduce operating expenses and increase productivity. Specific benefits include:

1. The reduction of building cleaning costs - an item otherwise accounting for as much as 40% of total operating expenses.
2. The reduction of employee absenteeism - a result of the removal of bacteria, viruses and allergens from the air.
3. An increase in employee efficiency
4. An increase in product quality
5. An increase in the life of machinery or equipment.

Contaminants

Air is contaminated in varying degrees by soil, organic matter, spores, viruses, bacteria and allergens, as well as aerosols such as smokes, dusts, fumes and mists. These contaminants may be introduced into the air from outdoors, or they may be returned to the air conditioning apparatus from within the space. The ease and efficiency with which they may be removed depends on the size, shape, specific gravity, concentration and surface characteristics of the particle.

Contaminant characteristics vary widely. Particle diameters range from molecular size up to 5,000 microns. Concentrations as high as 400 grains per 1,000 cubic feet may be encountered. However, air conditioning applications usually involve the removal of particles no smaller than 0.1 micron in diameter and as large as 200 microns. Normal concentrations seldom exceed 4 grains per 100 cubic feet. This specific characteristics of the particles to be removed are determined by the application. Thus, air purity control is a relative concept.

Atmospheric air cleaners (referred to as air filters) are rated in terms of efficiency (arrestance), resistance to air flow, and dust capacity. The three most critical performance factors are the following:

1. The variation of filter resistance with air flow.
2. The variation of filter resistance with the dust load at design air flow.
3. The effect of dust loads at design air flow on the filter efficiency.

Filter resistance increases with air flow (face velocity) or with dust load at design air flow. The efficiency of a particular filter varies not only with dust load but also with the characteristics of the contaminating particles. The capacity of a filter is a measure of its usable life prior to disposal, renewing or cleaning.

Application

The choice of a particular type of air filter for a given application involves the following steps:

1. A determination of the size, concentration and characteristics of contaminants present in both the outdoor air and return air.
2. A decision regarding the size of particles to be removed and the efficiency required for removal.
3. The selection of the filter which will provide most economically the desired efficiency under the prevailing conditions of cost, power costs and annual hours of operation.

Air contamination may be appraised by costly laboratory analysis or by estimation based on past experience and general data. The latter method is preferable in all except highly specialized applications.

The determination of which contaminants to be removed and to what degree, should be based on the requirements of the processes, equipment, materials or occupants within the conditioned space. For example, a greater filtering efficiency would be required for an electronics laboratory than for a bowling alley. However for any application certain contaminants should be removed. These contaminants include abrasive dusts, lint, pollen, concentrations of toxic fumes, if present, and carbon, if in appreciable quantities.

Outdoor air and return air may be separately cleaned with different filter types if the characteristics of the contaminants to be removed are widely different. Regardless of the type of filter selected, the automatic self-cleaning feature renders servicing less dependent on the human element and provides a relatively uniform air resistance and air flow.

3.2.3 Cooling Coils

Cooling coils are used with chilled water, well water or direct expansion for the purpose of pre-cooling, cooling, and dehumidifying or for aftercooling. The resulting velocity through the cooling coil is dictated by the air quantity, coil size, available space, and the coil load. Manufacturer’s data gives recommended maximum air velocities above which water carry-over begins to occur.
Chapter 4

Heat Exchangers

Heat exchangers have the function of transferring heat from one fluid to another. Many types of heat exchangers exist. This is because there are many design considerations involved. Some may even end up in basic conflict. The design considerations are as follow:

1. Resistance to heat transfer should be minimized.
2. Contingencies should be anticipated via safety margins; for example, allowance for fouling during operation.
3. The equipment should be sturdy
4. Costs and material requirements should be kept low.
5. Corrosion should be avoided.
6. Pumping costs should be kept low.
7. Space required should be kept low where applicable.
8. Required weight should be kept low.

Design involves trade-offs among factors not related directly to heat transfer. Meeting the objective of minimized thermal resistance implies thin wall separating fluids. Thin walls may not be compatible with sturdiness. Auxiliary steps may have to be taken, for example, the use of support plates for tubing, to realize sturdiness.

Use of thin walls and the presence of support plates may facilitate corrosion or other types of chemical action, ultimately leading to equipment damage. The formation and growth of a product at the location where tubes pass through a support plate have caused significant difficulty. Potential for corrosion within the
heat exchanger may lead to requirements on treating fluids prior to entry into the heat exchanger. Filters and demineralizers might be used for such purposes.

Impurities entering the heat exchanger may also cause deterioration of heat transfer performance. Material may settle so as to produce coatings that "foul" surfaces, leading to reduced heat transfer coefficients and possibly other problems too.

Impurities can originate within the equipment. The heat exchanger may be part of a complex machine. A fluid may go through several other pieces of equipment in addition to the heat exchanger itself. Contaminants may enter the fluid in one of the other pieces of equipment and cause difficulty in the heat exchanger.

The prospect for deterioration of performance due to fouling implies a desirability or a safety margin in heat exchanger design so that the overall industrial process can function after some time. However, the friction-heat transfer analogy implies that the heat transfer margin may require increased pumping costs, which leads to a conflict with another objective. Heat exchangers differ considerably depending on which objectives are most important.

4.1 Internal Flow

4.1.1 Hydrodynamic Considerations

Flow Conditions

A laminar flow in a circular tube of radius $r_o$ Figure(4.1), where fluid enters the tube with a uniform velocity. We know that when the fluid makes contact with the surface, viscous effects become important and a boundary layer develops with increasing $x$. This development occurs at the expense of a shrinking inviscid flow region and concludes with boundary layer merger at the centerline. Following this merger, viscous effects extend over the entire cross section and the velocity profile no longer changes with increasing $x$, the flow is then said to be fully developed, and the distance from the entrance at which this occurs is termed the hydrodynamic entry length, $x_{fd,h}$. As shown in Figure(4.1), the fully developed velocity profile is parabolic for laminar flow in a circular tube. For turbulent flow, the profile is flatter due to turbulent mixing in the radial direction.

When dealing with internal flows, it is important to be cognizant of the extent of the entry region, which depends on whether the flow is laminar or turbulent. The Reynolds number for flow in a circular tube is defined as:

$$Re_D \equiv \frac{\rho u_m D}{\mu}$$  \hspace{1cm} (4.1)

where $u_m$ is the mean fluid velocity over the tube cross section and $D$ is the diameter. In a fully developed flow, the critical Reynolds number corresponding to
the onset of turbulence is

$$Re_{D,c} \approx 2300$$ (4.2)

although much larger Reynolds number ($Re \approx 10,000$) are needed to achieve fully turbulent conditions. The transition to turbulent conditions. The transition to turbulence is likely to begin in the developing boundary layer of the entrance region.

For laminar flow ($Re_{D} \leq 2300$), the hydrodynamic entry length may be obtained from an expression of the form

$$\left( \frac{x_{fd}, h}{D} \right)_{lam} \approx 0.05 Re_D$$ (4.3)

This expression is based on the presumption that fluid enters the tube from a rounded converging nozzle and hence characterized by a nearly uniform velocity profile at the entrance Figure(4.1). Although there is no satisfactory general expression for the entry length in turbulent flow, we know that it is approximately independent of Reynolds number and that, as a first approximation,

$$10 \leq \left( \frac{x_{fd}, h}{D} \right)_{turb} \leq 60$$ (4.4)

The Mean Velocity

Since the velocity over the cross section varies and there is no well-defined free stream, it becomes necessary to work with a mean velocity $u_m$ when internal flows are taken into account. This velocity is considered such that when multiplied by
the density $\rho$ and the cross sectional area of the tube $A_c$, it provides the rate of mass flow through the tube. Therefore:

$$\dot{m} = \rho u_m A_c$$ (4.5)

For steady incompressible flow in a tube of uniform cross sectional are $m$ and $u_m$ are constants independent of $x$. SO it becomes evident that, for flow in a circular tube ($A_c = \pi D^2/4$), the Reynolds number becomes:

$$Re_D = \frac{4\dot{m}}{\pi D \mu}$$

The mass flow rate may also be expressed as the integral of the mass flux ($\rho u$) over the cross section

$$\dot{m} = \int_{A_c} \rho u(r, x) dA_c$$

it then follows for incompressible flow in a circular tube

$$u_m = \frac{\int_{A_c} \rho u(r, x) dA_c}{\rho A_c} = \frac{2}{r^2} \int_0^{r_m} u(r, x) r dr$$

The above expression can be used to determine $u_m$ at any axial location $x$ by knowing the velocity profile $u(r)$ at that location.

**Velocity Profile in the Fully Developed Region**

The form of the velocity profile can be readily determined for the laminar flow of an incompressible, constant property fluid in the fully developed region of a circular tube. An important feature of the hydrodynamic conditions in the fully developed region is that both the radial velocity component $\nu$ and the gradient of the axial velocity component ($\delta u/\delta x$) are everywhere zero.

The radial dependence of the axial velocity can be obtained by solving the appropriate form of the $x$-momentum equation. This form is determined by first recognizing that, the net momentum flux is everywhere zero in the fully developed region. Hence the momentum conservation requirement reduces to a simple balance between shear and pressure forces in the flow. This force balance can be expressed as:

$$0 = \tau_r (2\pi r dr) - \left\{ \tau_r (2\pi r dr) + \frac{d}{dr}[\tau_r (2\pi r dr)] dr \right\} + p(2\pi r dr) - \left\{ p(2\pi r dr) + \frac{d}{dx}[p(2\pi r dr)] dx \right\}$$

which reduces to:
Figure 4.2: Force balance on a differential element for laminar, fully developed flow in a circular tube

\[-\frac{d}{dr}(r\tau_r) = r \frac{dp}{dx}\]  \hfill (4.6)

With \(y = r_o - r\) Newton’s law of viscosity, assumes the form:

\[\tau_r = -\mu \frac{du}{dr}\]  \hfill (4.7)

\[\frac{\mu}{r} \frac{d}{dr} \left( r \frac{du}{dr} \right) = \frac{dp}{dx}\]  \hfill (4.8)

Since the axial pressure gradient is independent of \(r\), the above equation may be solved by integrating twice:

\[r \frac{du}{dr} = \frac{1}{\mu} \left( \frac{dp}{dx} \right) \frac{r^2}{2} + C_1\]

\[u(r) = \frac{1}{\mu} \left( \frac{dp}{dx} \right) \frac{r^2}{4} + C_1 \ln r + C_2\]

\[u(r) = -\frac{1}{4\mu} \left( \frac{dp}{dx} \right) r_o^2 \left[ 1 - \left( \frac{r}{r_o} \right)^2 \right]\]  \hfill (4.9)

\[u_m = -\frac{r_o^2}{8\mu} \frac{dp}{dx}\]  \hfill (4.10)

\[\frac{u(r)}{u_m} = 2 \left[ 1 - \left( \frac{r}{r_o} \right)^2 \right]\]  \hfill (4.11)

since \(u_m\) can be computed from knowledge of the mass flow rate, the above equation can be used to determine pressure gradient.

4.1.2 Thermal Considerations

If fluid enters the tube in figure (4.3) at a uniform temperature \(T(r, 0)\) that
is less than the surface temperature, convection heat transfer occurs and a thermal boundary layer begins to develop. Moreover, if the tube surface condition is fixed by imposing either a uniform temperature \( (T_s \text{ is constant}) \) or a uniform heat flux \( (q'' \text{ is constant}) \), a thermally fully developed condition is eventually reached. The shape of the fully developed temperature profile \( T(r, x) \) differs according to whether a uniform surface temperature or heat flux is maintained. For both surface conditions, however, the amount by which fluid temperatures exceed the entrance temperature increases with increasing \( x \).

For laminar flow the thermal entry length may be expresses as:

\[
\left( \frac{x_{fd,t}}{D} \right)_{lam} \approx 0.05 Re D Pr
\]  
Formula (4.12)

Figure 4.3: Thermal boundary layer development in a heated circular tube

The Mean Temperature

The absence of a free stream temperature necessitates using a mean temperature. The mean (or bulk) temperature of the fluid at a given cross section is defined in terms of the thermal energy transported by the fluid as it moves past the cross section. The rate at which this transport occurs \( \dot{E}_t \), may be obtained by integrating the product of the mass flux \( (\rho u) \) and the internal energy per unit mass \( (c_v T) \) over the cross section. That is,

\[
\dot{E}_t = \int_{A_c} \rho u c_v T dA_c
\]

mean temperature defined as:

\[
\dot{E}_t = \dot{m} c_v T_m
\]

we obtain

\[
40
\]
\[ T_m = \frac{\int_{A_c} \rho v c_v T dA_c}{\dot{m} c_v} \]  \hspace{1cm} (4.15) \\
\[ T_m = \frac{2}{u_m r_o^2} \int_0^{r_o} u T dr \]  \hspace{1cm} (4.16)

**Note:** when multiplied by the mass flow rate and the specific heat, \( T_m \) provides the rate at which thermal energy is transported with the fluid as it moves along the tube.

**Newton’s Law of Cooling**

The mean temperature \( T_m \) is a convenient reference temperature for internal flows. Newton’s law of cooling may be expressed as:

\[ q''_s = h(T_s - T_m) \]  \hspace{1cm} (4.17)

where:

- \( h \) is the local convection heat transfer coefficient.
- \( T_m \) varies in the flow direction
  - increases with \( x \) if heat transfer is from the surface to the fluid
  - \( dT_m/dx \) is never zero if heat transfer is occurring.

**4.1.3 The Energy Balance**

**General Considerations**

Since the flow in a tube is completely enclosed, an energy balance may be applied to determine how the mean temperature \( T_m(x) \) varies with position along the tube and how the total convection heat transfer \( q_{conv} \) is related to the difference in temperatures at the tube inlet and outlet. Consider the tube flow of figure (4.4). Fluid moves at a constant flow rate \( \dot{m} \), and convection heat transfer occurs at the inner surface. Typically, fluid kinetic and potential energy changes, as well as energy transfer by conduction in the axial direction, are negligible. Therefore if no shaft work is being done by the fluid as it passes through the tube, the only significant effects will be those associated with thermal energy changes and with flow work. Flow work is performed to move the fluid through a control surface and, per unit mass of fluid, may be expressed as the product of the fluid pressure \( p \) and specific volume \( v \) (\( v = 1/\rho \)).

Applying conservation of energy, to the differential control volume of figure (4.4) and recalling the definition of the mean temperature, we obtain:

\[ dq_{conv} + \dot{m}(c_v T_m + pv) + \left[ \dot{m} \frac{d(c_v T_m + pv)}{dx} \right] = 0 \]  \hspace{1cm} (4.18)
or

\[ dq_{\text{conv}} = \dot{m}d(c_v T_m + pv) \]  

The rate of convection heat transfer to the fluid must equal the rate at which the fluid thermal energy increases plus the net rate at which work is done in moving the fluid through the control volume. A good approximation for incompressible liquids, is:

\[ dq_{\text{conv}} = \dot{m}c_p dT_m \]  

In this case \( c_v = c_p \), and since \( v \) is very small, \( d(pv) \) is generally much less than \( d(c_v T_m) \).

A special form of the equation above, relates to conditions for the entire tube. In particular, integrating from the tube inlet \( i \) to the outlet \( o \), it follows that:

\[ q_{\text{conv}} = \dot{m}c_p (T_{m,o} - T_{m,i}) \]  

where \( q_{\text{conv}} \) is the total tube heat transfer rate. It is a general expression that applies irrespective of the nature of the surface thermal or tube flow conditions.

The two previous equations may be cast in a convenient form by expressing the rate of convection heat transfer to the differential element as \( dq_{\text{conv}} = \dot{q}_s P dx \) where \( P \) is the surface perimeter (\( P = \pi D \) for a circular tube). Therefore:

\[ \frac{dT_m}{dx} = \frac{\dot{q}_s}{\dot{m}c_p} = \frac{P}{\dot{m}c_p} h(T_s - T_m) \]  

This expression is an extremely useful result, from which the axial variation of \( T_m \) may be determined. The solution to this equation for \( T_m(x) \) depends on the surface thermal condition. Recall that the two special cases of interest are constant surface heat flux and constant surface temperature. It is common to find one of these conditions existing to a reasonable approximation.
4.2 Heat Exchangers

4.2.1 Heat Exchanger Types

Heat exchangers tend to be classified according to the flow arrangement and type of construction. The simplest heat exchanger is one for which the hot or cold fluids move in either the same or opposite directions in a concentric tube (or double-pipe) construction. In the parallel-flow arrangement of figure (4.5a), the hot and cold fluids enter at the same end, flow in the same direction, and leave at the same end. In the counterflow arrangement of figure (4.5b), the fluids enter at opposite ends, flow in opposite directions, and leave at opposite ends.

![Figure 4.5: Concentric tube heat exchangers. (a) Parallel-flow. (b) Counterflow.](image)

Alternatively, fluids may move in cross flow (perpendicular to each other), through finned or unfinned tubular heat exchangers. The two configurations differ according to whether the fluid moving over the tubes is unmixed or mixed. For unmixed flow, fins prevent the motion in a direction (y) that is transverse to the mainflow direction (x). In this case, the fluid temperature varies with x and y. In contrast, for unfinned tubes (hence mixed), fluid motion in the transverse direction is possible, and temperature variations are primarily in the main-flow direction. The nature of the mixing condition can significantly influence the heat exchanger performance.

4.2.2 The Overall Heat Transfer Coefficient

An essential but uncertain part of any heat exchanger analysis is the determination of the overall heat transfer coefficient. This coefficient is defined in terms of the total thermal resistance to heat transfer between two fluids. The coefficient is determined by accounting for conduction and convection resistances between fluids separated by composite plane and cylindrical walls, respectively. NOTE: such results apply only to clean, unfinned surfaces.
During normal heat exchanger operation, surfaces are often subject to fouling fluid due to impurities, rust formation, or other reactions between the fluid and the wall material. The subsequent deposition of a film or scale on the surface can greatly increase the resistance to heat transfer between the fluids. This effect can be treated by introducing an additional thermal resistance, termed the *fouling factor*, $R_f$. Its value depends on the operating temperature, fluid velocity, and length of service of heat exchanger.

In addition, fins are often added to surfaces exposed to either or both fluids and that, by increasing the surface area, they reduce the resistance to convection heat transfer. Accordingly, with inclusion of surface fouling and fin (extended surface) effects, the overall heat transfer coefficient may be expressed as:

$$
\frac{1}{UA} = \frac{1}{UcAc} = \frac{1}{UhAh} = 
\frac{1}{(\eta_ohA)_c} + R_{fc} + R_w + 
\frac{1}{(\eta_ohA)_h} + R_{fh} = 
\frac{R''_{fc} + 1}{(\eta_ohA)_c} + 
\frac{R''_{fh} + 1}{(\eta_ohA)_h}
$$

where $c$ and $h$ refer to the cold and hot fluids.

*NOTE:* calculation of $UA$ product does not require designation of the cold or hot side however the overall coefficient does depend on whether it is based on the cold or hot side of the surface area.

- $R_w$ is $\frac{L}{kA}$ for a plane wall
  $\frac{\ln(r_o/r_i)}{2\pi Lk}$ for a cylindrical wall
- $\eta_o$ is the *overall surface efficiency* *temperature effectiveness* of a finned surface

It is defined for the hot or cold surface, the heat transfer rate is

$$q = \eta_o h A (T_b - T_{inf})$$

where $T_b$ is the base surface temperature and $A$ is the total surface area, and $\eta_o$ is derived from

$$\eta_o = 1 - \frac{A_f}{A} (1 - \eta_f)$$

- $A_f$ is the entire fin surface area
• $\eta_f$ is the efficiency of a single fin

If a straight or pin fin of length $L$ is used and an adiabatic tip is assumed, then:

$$\eta_f = \frac{\tanh(mL)}{mL} \quad (4.26)$$

where:

$$m = \left(\frac{2h_k}{kt}\right)^{1/2}$$

$t$ is the thickness of the fin

### 4.2.3 Heat Exchanger Analysis: Use of the Log Mean Temperature Difference

To design or predict the performance of a heat exchanger, it is essential to relate the total heat transfer rate to quantities such as the inlet and outlet fluid temperatures, the overall heat transfer coefficient, and the total surface area for heat transfer. Two such relations may readily be obtained by applying overall energy balances to the hot and cold fluids, as shown in figure (4.6). In particular, if $q$ is the total rate of heat transfer between the hot and cold fluids and there is negligible heat transfer between exchanger and surroundings, application of an energy balance gives:

$$q = \dot{m}_h (i_{h,i} - i_{h,o}) \quad (4.27)$$

and

$$q = \dot{m}_c (i_{c,i} - i_{c,o}) \quad (4.28)$$

where:

• $i$ is the fluid enthalpy
• $h$ refers to the hot fluid
• $c$ refers to the cold fluid
• $i$ and $o$ refer to inlet and outlet conditions

If the fluid is under no phase change, and specific heats are assumed, the above expressions become:

$$q = \dot{m}_h c_p,h (T_{h,i} - T_{h,o})$$

and

$$q = \dot{m}_c c_p,c (T_{c,i} - T_{c,o})$$

where the temperatures appearing in the expressions refer to the mean fluid temperatures at the designated locations.
Another useful expression is just relating the total heat transfer rate $q$ to the temperature difference $\Delta T$ between hot and cold fluids, where:

$$q = UA\Delta T_m$$ \hspace{1cm} (4.29)

$\Delta T_m$ is an appropriate mean temperature difference.

**Parallel-Flow Heat Exchanger**

The hot and cold fluid temperature distributions associated with a parallel-flow heat exchanger are shown in figure (4.7). The temperature difference $\Delta T$ is initially large but decays rapidly with increasing $x$, approaching zero asymptotically. **Note** for such an exchanger, the outlet temperature of the cold fluid never exceeds that of the hot fluid.

The form $\Delta T_m$ may be determined by applying an energy balance to differential elements in the hot and cold fluids. Each element is of length $dx$ and heat transfer surface area $dA$, as shown in figure (4.7). The energy balances and the subsequent analysis are subject to the following assumptions.

1. The heat exchanger is insulated from its surroundings, in which case the only heat exchange is between the hot and cold fluids.
2. Axial conduction along the tubes is negligible.
3. Potential and kinetic energy changes are negligible.
4. The fluid specific heats are constant.
5. The overall heat transfer coefficient is constant.

The specifics heat are liable to change due to temperature variations, and the overall heat transfer coefficient may change because of variations in fluid properties and flow conditions. However in many applications these variations are not significant, and it is reasonable to work with average values of $c_{p,c}, c_{p,h}$, and $U$ for the heat exchanger.
Figure 4.7: Temperature Distributions for a parallel flow heat exchanger

- Applying an energy balance to each of the differential elements in figure (4.7) it follows

\[ dq = -\dot{m}_h c_{p,h} dT_h \equiv -C_h dT_h \]  \hspace{1cm} (4.30)

\[ dq = -\dot{m}_c c_{p,c} dT_c \equiv C_c dT_c \]  \hspace{1cm} (4.31)

\[ dq = U \Delta T dA \]  \hspace{1cm} (4.32)

where:

- \( \Delta T = T_h - T_c = \) local temperature difference
- \( d(\Delta T) = dT_h - dT_c \)
- \( d(\Delta T) = -dq \left( \frac{1}{C_h} + \frac{1}{C_c} \right) \)
- \( \frac{\int_1^2 d(\Delta T)}{\Delta T} = -U \left( \frac{1}{C_h} + \frac{1}{C_c} \right) \int_1^2 dA \)
- \( \ln \left( \frac{\Delta T_2}{\Delta T_1} \right) = -UA \left( \frac{1}{C_h} + \frac{1}{C_c} \right) \)  \hspace{1cm} (4.33)

Through substitution,

\[ \ln \left( \frac{\Delta T_2}{\Delta T_1} \right) = -\frac{UA}{q} \left[ (T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o}) \right] \]
Therefore
\[ q = UA \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \]

Accordingly
\[ q = UA \Delta T_{lm} \]

where:
\[ \Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \]

For the parallel-flow heat exchanger
\[
\begin{align*}
\Delta T_1 & \equiv T_{h,1} - T_{c,1} = T_{h,i} - T_{c,i} \\
\Delta T_2 & \equiv T_{h,2} - T_{c,2} = T_{h,o} - T_{c,o}
\end{align*}
\]

The Counterflow Heat Exchanger

The hot and cold fluid temperature distributions associated with a counterflow heat exchanger are shown in figure (4.8). In contrast to the parallel flow exchanger, this configuration provides for heat transfer between the hotter portions of the two fluids at one end, as well as the colder portions at the other. This is why the temperature difference, \( \Delta T = T_h - T_c \), with respect to \( x \) is nowhere as large as it is for the inlet region of the parallel-flow exchanger. Note the outer temperature of the cold fluid may now exceed the outlet temperature of the hot fluid.

The equations derived above apply to any heat exchanger, and hence may be used for the counterflow arrangement.

Note for the same inlet and outlet temperatures, the log mean temperature difference for counterflow exceeds that for parallel flow, \( \Delta T_{lm,CF} > \Delta T_{lm,PF} \). Hence the surface area required to effect a prescribed heat transfer rate \( q \) is smaller for the counterflow than for the parallel-flow arrangement, assuming the same value of \( U \). Also note that \( T_{c,o} \) can exceed \( T_{h,o} \) for counterflow but not for parallel flow.

For the counterflow heat exchanger
\[
\begin{align*}
\Delta T_1 & \equiv T_{h,1} - T_{c,1} = T_{h,i} - T_{c,o} \\
\Delta T_2 & \equiv T_{h,2} - T_{c,2} = T_{h,o} - T_{c,i}
\end{align*}
\]

Special Operating Conditions

There are certain special conditions under which heat exchangers may be operated. Figure (4.9a) shows temperature distributions for a heat exchanger in which the hot fluid has a heat capacity rate, \( C_h \equiv \dot{m}_h c_{p,h} \), which is much larger than that of
the cold fluid, $C_c \equiv \dot{m}_c c_p$. For this case the temperature of the hot fluid remains approximately constant throughout the heat exchanger, while the temperature of the cold fluid increases. The same condition is achieved if the hot fluid is a condensing vapor. Condensation occurs at constant temperature, and for all practical purposes, $C_h \rightarrow \infty$. Conversely, in an evaporator or a boiler figure (4.9b), it is the cold fluid that experiences a change in phase and remains at a nearly uniform temperature ($C_c \rightarrow \infty$). The same effect is achieved without phase change if $C_h \ll C_c$. The third special case figure (4.9c) involves a counterflow heat exchanger for which the heat capacity rates are equal ($C_h = C_c$). The temperature difference $\Delta T$ must then be a constant throughout the exchanger, in which case $\Delta T_1 = \Delta T_2 = \Delta T_{lm}$.

**Multipass and Cross-Flow Heat Exchangers**

Although cross flow conditions are more complicated in multipass and cross flow heat exchangers, the equations derived above may once again be used if the following modification is made to the log mean temperature difference:

$$\Delta T_{lm} = F \Delta T_{lm,CF}$$ (4.36)

That is, the appropriate form of $\Delta T_{lm}$ is obtained by applying a correction factor to the value of $\Delta T_{lm}$ that would be computed under the assumption of counterflow.
Figure 4.9: Special heat exchanger conditions. (a) $C_h \gg C_c$ or a condensing vapor. (b) $C_h \ll C_c$ or an evaporating liquid. (c) $C_h = C_c$ or a counterflow heat exchanger with equivalent fluid heat capacities.

Algebraic expressions for the correction factor $F$ have been developed for various shell-and-tube and cross-flow heat exchanger configurations.
Chapter 5

Cooling Load Principles

In the following chapter, the method of calculating air conditioning cooling loads for sizing cooling equipment for non residential applications will be discussed.

The heat balance approach is a fundamental concept in calculating cooling loads. While generally cumbersome for widespread or routine use, this underlying concept is the basis for the procedures outlined for varying purposes.

The variables affecting cooling load calculations are numerous, often difficult to define precisely, and always intricately interrelated. Many cooling load components vary in magnitude over a wide range during a 24-hr period. Since these cyclic changes in load components are often not in phase with each other, each must be analyzed to establish the resultant maximum cooling load for a building or zone. A zoned system (a system of conditioning equipment serving several independent areas, each with its own temperature control) need recognize no greater total cooling load capacity than the largest hourly summary of simultaneous zone loads throughout a design day; however it should be able to handle the peak cooling load for each zone at its individual peak hour. At certain times of the day during the heating or intermediate seasons, some zones may require heating while others require cooling.

5.1 Heat Flow Rates

In air-conditioning design, four related heat flow rates, each of which varies with time, must be differentiated:

1. Space heat gain
2. Space cooling load
3. Space heat extraction
4. Cooling coil load
5.1.1 Space Heat Gain

This instantaneous rate of heat gain is the rate at which heat enters into and/or is generated within a space at a given instant. Heat gain is classified by:

1. the mode in which it enters the space
   - solar radiation through transparent surfaces
   - heat conduction through exterior walls and roofs
   - heat conduction through interior partitions, ceilings, and floors
   - heat generated within the space by occupants, lights, and appliances
   - energy transfer as a result of ventilation and infiltration of outdoor air
   - miscellaneous heat gains

2. whether it is a sensible or latent gain
   - sensible heat gain is directly added to the conditioned space by conduction, convection, and/or radiation.
   - latent heat gain occurs when moisture is added to the space (e.g., vapor emitted by occupants and equipment). To maintain a constant humidity ratio, water vapor must condense on cooling apparatus at a rate equal to its rate of addition into the space. The amount of energy required to offset the latent heat gain essentially equals the product of the rate of condensation and the latent heat of condensation.

   in selecting cooling apparatus, it is necessary to distinguish between the sensible and latent heat gain. Every apparatus has a maximum sensible heat removal capacity and a maximum latent heat removal capacity for particular operating conditions.

5.1.2 Space Cooling Load

This is the rate at which heat must be removed from the space to maintain a constant space air temperature. The sum of all space instantaneous heat gains at any given time does not necessarily equal the cooling load for the space at that same time. However the space heat gain by radiation is not immediately converted into cooling load. Radiant energy must first be absorbed by the surfaces that enclose the space (walls, floor and ceiling) and the objects in the space (furniture, etc.). As soon as these surfaces and objects become warmer than the space air, some of their heat is transferred to the air in the space by convection. The composite heat storage capacity of these surfaces and objects determines the rate at which their respective surface temperatures increase for a given radiant input, and thus governs the relationship between the radiant portion of the heat gain and its corresponding part of the space cooling load.
5.1.3 Space Heat Extraction Rate

The rate at which heat is removed from the conditioned space equals the space cooling load only to the degree that room air temperature is held constant. In conjunction with intermittent operation of the cooling equipment, the control system characteristics usually permit a minor cyclic variation or swing in room temperature. Therefore, a proper simulation of the control system gives a more realistic value of energy removal over a fixed time period than using the values of the space cooling load.

5.1.4 Cooling Coil Load

The rate at which energy is removed at the cooling coil that serves one or more conditioned spaces equals the sum of the instantaneous space cooling loads (or space heat extraction rate if it is assumed that the space temperature does not vary) for all the spaces served by the coil, plus any external loads. Such external loads include heat gain by the distribution system between the individual spaces and the cooling equipment, and outdoor air heat and moisture introduced into the distribution system through the cooling equipment.

5.2 Heat Gain Calculation Concepts

5.2.1 Heat Gain through Exterior Surfaces

Heat gain through exterior opaque surfaces is derived from the same elements of solar radiation and thermal gradient as that for fenestration areas. It differs primarily as a function of the mass and nature of the wall or roof construction, since those elements affect the rate of conductive heat transfer through the composite assembly to the interior surface.

5.2.2 Sol-Air Temperature

Sol-air temperature is the temperature of the outdoor air that, in the absence of all radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air.

Heat flux into exterior sunlit surfaces. The heat balance at a sunlit surface gives the heat flux into the surface $q/A$ as:

$$q/A = \alpha I_t + h_o (t_o - t_s) - \varepsilon \Delta R$$

(5.1)

where:

- $\alpha$ = absorptance of surface for solar radiation
\[ I_t = \text{total solar radiation incident on surface, Btu/(h ft}^2) \]
\[ h_o = \text{coefficient of heat transfer by long-wave radiation and convection at outer surface, Btu/(h ft}^2 \ {}^\circ F) \]
\[ t_o = \text{outdoor air temperature, } {}^\circ F \]
\[ t_s = \text{surface temperature, } {}^\circ F \]
\[ \varepsilon = \text{hemispherical emittance of surface} \]
\[ \Delta R = \text{difference between long wave radiation incident on surface from sky and surroundings and radiation emitted by blackbody at outdoor air temperature, Btu/(h ft}^2) \]

Assuming the rate of heat transfer can be expressed in terms of the sol-air temperature \( t_e \)
\[
q/A = h_o (t_e - t_s) \quad (5.2)
\]
and rearranging the previous equation:
\[
t_e = t_o + \alpha I_t/h_o - \varepsilon \Delta R/h_o \quad (5.3)
\]

### 5.2.3 Heat Gain through Interior Surfaces

Whenever a conditioned space is adjacent to a space with a different temperature, transfer of heat through the separating physical section must be considered. The heat transfer rate is given by:
\[
q = UA(t_o - t_i) \quad (5.4)
\]
where:
- \( q = \text{heat transfer rate, Btu/h} \)
- \( U = \text{coefficient of overall heat transfer between adjacent and conditioned space, Btu/(h ft}^2 \ {}^\circ F) \)
- \( A = \text{area of separating section concerned, ft}^2 \)
- \( t_o = \text{average air temperature in adjacent space, } {}^\circ F \)
- \( t_i = \text{air temperature in conditioned space, } {}^\circ F \)
Temperature $t_b$ may range widely from that in the conditioned space. The temperature in a kitchen or boiler room, for example, may be as much as 15 to 50 °F above the outdoor air temperature. Actual temperatures in adjoining spaces should be measured when possible. Where nothing is known, except that the adjacent space is of no conventional construction, contains no heat sources, and itself receives no significant solar heat gain, $t_b - t_i$ may be considered the difference between the outdoor air and conditioned space design dry-bulb temperature minus 5 °F. In some cases, the air temperature in the adjacent space will correspond to the outdoor air temperature or higher.

### 5.3 Heat Sources in Conditioned Spaces

#### 5.3.1 People

The table below gives representative rates at which heat and moisture are given off by human beings in different states of activity. Often these sensible and latent heat gains constitute a large fraction of the total load. Even for short-term occupancy, the extra heat and moisture brought in by people may be significant.

The conversion of sensible heat gain from people to space cooling load is affected by the thermal storage characteristics of that space and is thus subject to application of appropriate room transfer functions (RTF). Latent heat gains are considered instantaneous.
<table>
<thead>
<tr>
<th>Activity</th>
<th>Total Heat, Btu/h</th>
<th>Sensible Heat</th>
<th>Latent Heat</th>
<th>Radiant Sensible Heat</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Adjusted M/F°</td>
<td>Btu/h</td>
<td>Btu/h</td>
<td>Low V</td>
</tr>
<tr>
<td>Theatre</td>
<td>330</td>
<td>225</td>
<td>105</td>
<td></td>
</tr>
<tr>
<td>Seated</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Theatre</td>
<td>350</td>
<td>245</td>
<td>105</td>
<td>60</td>
</tr>
<tr>
<td>Seated at Night</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Light Work</td>
<td>400</td>
<td>245</td>
<td>155</td>
<td></td>
</tr>
<tr>
<td>Seated</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Moderate Work</td>
<td>450</td>
<td>250</td>
<td>200</td>
<td></td>
</tr>
<tr>
<td>Standing/</td>
<td>450</td>
<td>250</td>
<td>200</td>
<td>58</td>
</tr>
<tr>
<td>Light Work</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Standing/</td>
<td>500</td>
<td>250</td>
<td>250</td>
<td></td>
</tr>
<tr>
<td>Walking</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sedentary Work</td>
<td>550</td>
<td>275</td>
<td>275</td>
<td></td>
</tr>
<tr>
<td>Light Bench</td>
<td>750</td>
<td>275</td>
<td>475</td>
<td></td>
</tr>
<tr>
<td>Work</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Moderate</td>
<td>850</td>
<td>305</td>
<td>545</td>
<td>49</td>
</tr>
<tr>
<td>Dancing</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Walking</td>
<td>1,000</td>
<td>375</td>
<td>625</td>
<td></td>
</tr>
<tr>
<td>3 mph</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bowling</td>
<td>1,450</td>
<td>580</td>
<td>870</td>
<td></td>
</tr>
<tr>
<td>Heavy Work</td>
<td>1,450</td>
<td>580</td>
<td>870</td>
<td>54</td>
</tr>
<tr>
<td>Heavy Machine</td>
<td>1,600</td>
<td>635</td>
<td>965</td>
<td></td>
</tr>
<tr>
<td>Work</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Athletics</td>
<td>1,800</td>
<td>710</td>
<td>1,090</td>
<td></td>
</tr>
</tbody>
</table>

### 5.3.2 Lighting

Since lighting is often the major space load component, an accurate estimate of the space heat gain it imposes is needed. Calculation of this load component is not straightforward; the rate of heat gain at any given moment can be quite different from the heat equivalent of power supplied instantaneously to those lights.

Only part of the energy from lights is in the form of convective heat, which
is picked up instantaneously by the air-conditioning apparatus. The remaining portion is in the form of radiation, which affects the conditioned space only after having been absorbed and rereleased by walls, floors, furniture, etc. This absorbed energy contributes to space cooling load only after a time lag, with some part of such energy still present and reradiating after the lights have been switched off.

5.3.3 Instantaneous Heat Gain from Lighting

The primary source of heat from lighting comes from light-emitting elements, or lamps, although significant additional heat may be generated from associated appurtenances in the light fixtures that house such lamps. Generally, the instantaneous rate of heat gain from electric lighting may be calculated from:

\[ q_{el} = 3.41 W F_{ul} F_{sa} \]  

(5.5)

where:

- \( q_{el} \) = heat gain, Btu/h
- \( W \) = total light wattage
- \( F_{ul} \) = lighting use factor
- \( F_{sa} \) = lighting special allowance factor
- 3.41 Btu = 1 watt-hour (W h)

The total light wattage is obtained from the ratings of all lamps installed, both for general illumination and for display use.

The special allowance is the ratio of the wattage in use, for the conditions under which the load estimate is being made, to the total installed wattage. For commercial applications such as stores, the use factor would generally be unity.

The special allowance factor is for fluorescent fixtures and/or fixtures that are either ventilated or installed so that only part of their heat goes to the conditioned space. For fluorescent fixtures, the special allowance factor accounts for the ballast losses, and can be as high as 2.19 for 32 W single lamp high-output fixtures on 277 V circuits. Rapid start, 40 W lamp fixtures have special allowance factors that vary from a low of 1.18 for two lamps at 277 V to a high of 1.30 for one lamp at 118 V, with a recommended value of 1.20 for general applications. Industrial fixtures other than fluorescent, such as sodium lamps, may have special allowance factors varying from 1.04 to 1.37, depending on the manufacturer, and should be dealt with individually.

For ventilated or recessed light fixtures, manufacturers' or other data must be sought to establish the fraction of the total wattage that may be expected to enter the conditioned space directly (and subject to time lag effect), versus that which must be picked up by return air or in some other appropriate manner.
5.3.4 Light Heat Components

Cooling load caused by lights recessed into ceiling cavities is made up of two components: one part comes from the light heat directly contributing to the space heat gain, and the other is the light heat released into the above-ceiling cavity, which (if used as a return air plenum) is mostly picked up by the return air that passes over or through the light fixtures. In such a ceiling return air plenum, this second part of the load (sometimes referred to as heat-to-return) never enters the conditioned space. It does, however, add to the overall load and significantly influences the load calculation.

Even though the total cooling load imposed on the cooling coil from these two components remains the same, the larger the fraction of heat output picked up by the return air, the more the space cooling load is reduced. The minimum required airflow rate for the conditioned space is decreased as the space cooling load is reduced. The minimum required airflow rate for the conditioned space is decreased as the space cooling load becomes less. Supply fan power reduces accordingly, which ultimately results in reduced energy consumption for the system, and possibly reduced equipment size as well.

For ordinary design load estimation, the heat gain for each component may simply be calculated as a fraction of the total lighting load by using judgment to estimate heat-to-space and heat-to-return percentages.

5.3.5 Return Air Light Fixtures

Two generic types of return air light fixture are available—those that allow and those that do not allow return air to flow through the lamp chamber. The first type is sometimes called a heat-of-light fixture. The percentage of light heat released through the plenum side of various ventilated fixtures can be obtained from lighting fixture manufacturers. Even unventilated fixtures lose some heat to plenum spaces; however, most of the heat ultimately enters the conditioned space from a dead-air plenum or is picked up by return air via ceiling return air openings. The percentage of heat to return air ranges from 40 to 60% for heat-to-return ventilated fixtures or 15 to 25% for unventilated fixtures.

subsection Plenum Temperatures

As heat from lighting is picked up by the return air, the temperature differential between the ceiling space and the conditioned space causes part of that heat to flow from the ceiling back to the conditioned space. Return air from the conditioned space can be ducted to capture light heat without passing through a ceiling plenum as such, or the ceiling space can be used as a return air plenum, causing the distribution of light heat to be handled in distinctly different ways. Most plenum temperatures do not rise more than 1 to 3°F above space temperature, thus generating only a relatively small thermal gradient for heat transfer through
plenum surfaces but a relatively large percentage reduction in space cooling load.

5.3.6 Energy Balance

Where the ceiling space is used as a return air plenum, an energy balance requires that the heat picked up from the lights into the return air (1) becomes a part of the cooling load to the return air (represented by a temperature rise of the return air as it passes through the ceiling space), (2) is partially transferred back to the conditioned space through the ceiling material below, and/or (3) may be partially “lost” (from the space) through the floor surfaces above the plenum. In a multistory building, the conditioned space frequently gains heat through its floor from a similar plenum below, offsetting the loss just mentioned. The radiant component of heat leaving the ceiling or floor surface of the plenum is normally so small that all such heat transfer is considered convective for calculation purposes.

The figure below shows a schematic diagram of a typical return air plenum.

\[ q_1 = U_c A_c (t_p - t_r) \]  
\[ q_2 = U_f A_f (t_p - t_fa) \]  
\[ q_3 = 1.1Q(t_p - t_r) \]  
\[ q_{lp} - q_1 - q_2 - q_3 = 0 \]  
\[ Q = \frac{q_r + q_1}{1.1(t_r - t_s)} \]

where:

- \( q_1 \) = heat gain from plenum through the ceiling, Btu/h
- \( q_2 \) = heat loss from plenum through floor above, Btu/h
- \( q_3 \) = heat gain “pickup” by return air, Btu/h
- \( Q \) = return airflow, cfm
- \( q_{lp} \) = light heat gain to plenum via return air, Btu/h
- \( q_{lr} \) = light heat gain to space, Btu/h
- \( q_f \) = heat gain from plenum below, through floor, Btu/h
- \( q_w \) = heat gain from exterior wall, Btu/h
- \( q_r \) = space cooling load, Btu/h, including appropriate treatment of \( q_{lr}, q_f, \) and/or \( q_w \)
Figure 5.1: Heat Balance of Typical Ceiling Return Plenum

- $t_p =$ plenum temperature, °F
- $t_r =$ space temperature, °F
- $t_{fa} =$ space temperature of floor above, °F
- $t_s =$ supply air temperature, °F
- $1.1\text{Btu/h} =$ heat capacity for air in (ft$^3$ °F/min)
5.3.7 Power

Instantaneous heat gain from equipment operated by electric motors within a conditioned space is calculated as:

\[ q_{em} = 2545 \left( \frac{P}{E_M} \right) F_{UM} F_{LM} \]  

(5.11)

where:

- \( q_{em} \) = heat equivalent of equipment operation, Btu/h
- \( P \) = motor power rating, horsepower
- \( E_M \) = motor efficiency, as decimal fraction < 1.0
- \( F_{UM} \) = motor use factor, 1.0 or decimal fraction < 1.0
- \( F_{LM} \) = motor load factor, 1.0 or decimal fraction < 1.0
- 2545 Btu/h = 1 hp

The motor use factor may be applied when the motor use is known to be intermittent with significant nonuse during all hours of operation (e.g. overhead door operator). For conventional applications, its value would be 1.0.

The motor load factor is the fraction of the rated load being delivered under the conditions of the cooling load estimate. In the equation above it is assumed that both the motor and the driven equipment are within the conditioned space. If the motor is outside the space or airstream, then:

\[ q_{em} = 2545 P F_{UM} F_{LM} \]  

(5.12)

When the motor is inside the conditioned space or airstream but the driven machine is outside:

\[ q_{em} = 2545 \left( \frac{1.0 - E_M}{E_m} \right) F_{UM} F_{LM} \]  

(5.13)

The above equation also applies to a fan or pump in the conditioned space.

5.3.8 Overloading and Underloading

Heat output of a motor is generally proportional to the motor load, within the overload limits. Because of typically high no-load motor current, fixed losses, and other reasons, \( F_{LM} \) is generally assumed to be unity, and no adjustment should be made for underloading or overloading unless the situation is fixed, can be accurately established, and the reduced load efficiency data can be obtained from the motor manufacturer.
5.3.9 Radiation and Convection

Unless the manufacturer’s technical literature indicates otherwise, the heat gain normally should be equally divided between radiant and convective components for the subsequent cooling load calculations.

5.3.10 Appliances

In a cooling load estimate, heat gain from all appliances-electrical, gas, or steam-should be taken into account. Because of the variety of appliances, applications, schedules, use, and installations, estimates can be very subjective. Often, the only information available about heat gain from equipment is that on its nameplate.

\[ \alpha \beta \gamma \delta \epsilon \zeta \eta \theta \iota \kappa \lambda \mu \nu \xi \pi \varpi \rho \sigma \varsigma \tau \upsilon \phi \varphi \chi \psi \omega \]  

(5.14)
5.4 Infiltration And Ventilation

This chapter incorporates infiltration and ventilation quantities of outdoor air. These outdoor air quantities have different heat content than the air within the conditioned space, which imposes a load on the air conditioning equipment. For infiltration, the load manifests itself directly upon the conditioned space. The ventilation air manifests its load directly on the air conditioning apparatus through which it is taken, and through the space thru apparatus bypass effect.

5.4.1 Infiltration

Infiltration of air and moisture are sources of sizable heat gains or losses for a conditioned space. The quantity of outside air that manages to infiltrate is dependent on the tightness of doors and windows, the building envelope, height of the building, stairwells, elevators, direction and velocity of wind, and the amount of ventilation and exhaust of air. Many of these cannot be evaluated accurately and must be based on assumptions. Infiltration is more generally caused by wind velocity, stack effect or even both:

- 1. - Wind Velocity - The velocity of the wind builds a pressure on the side of the building upon which it hits, and creates a slight vacuum on the opposite side. The outdoor pressure created causes air to infiltrate through crevices and cracks in the construction, windows and doors. This causes a slight build up of pressure inside the building which allows for an equal amount of exfiltration to occur on the side opposite to where the wind hits the building.

- 2. - Difference in Density or Stack Effect - The variations in the temperatures and humidities produce differences in density of air between the inside and the outside of the building. In tall buildings infiltration caused by density differences in the summer and winter occur as follows:
  
  Summer - infiltration at the top and exfiltration at the bottom.
  
  Winter - infiltration at the bottom and exfiltration at the top

Infiltration Thru Windows and Doors, Summer

During the summer most of the infiltration which occurs is caused primarily by the wind velocity creating pressure on the windward side. Stack effect is very close to being a negligible factor because the difference in the density is rather small. Typically in small buildings, air infiltrates thru open doors on the windward side, unless there is enough outdoor air being introduced to offset it. During the summer, infiltration is only calculated for the windward side. Some of the air infiltrating through the windows will exfiltrate through the windows on the opposite side,
while the remaining infiltration air will flow out through the doors, this then offsets some of the infiltration through the doors. To determine net infiltration through the doors, determine the infiltration through the windows on the windward side, multiply this by 0.80 and subtract from the door infiltration.

**Offsetting Infiltration with Outdoor Air, Summer**

Offsetting the infiltration completely by the introduction of outdoor air through the air conditioning apparatus is usually uneconomical, except for buildings with few doors and windows. The air which is introduced from the outside should be able to develop a pressure equal to that created by the wind velocity to offset the infiltration. This pressure would then cause exfiltration through the leeward walls at a rate equal to that of the wind velocity. Take for example a four sided building with equal crack areas on each side.

- With the wind blowing on one side, the amount of air introduced through the apparatus must be a little more than three times the amount that infiltrates.

- With the wind blowing on two ends, the amount of air introduced through the apparatus must be a little more than that which infiltrates.

Offsetting swinging door infiltration is not as difficult considering that air tends to take the path of least resistance, typically an open door. Therefore most of the outdoor air introduced through the apparatus will flow out the door when it is opened.

<table>
<thead>
<tr>
<th>DESCRIPTION</th>
<th>CFM PER SQ FT SASH AREA</th>
<th>Small-30&quot; x 72&quot;</th>
<th>Large-54&quot; x 96&quot;</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>No. W-Strip</td>
<td>W-Strip</td>
<td>Storm Sash</td>
</tr>
<tr>
<td>Average Wood Sash</td>
<td>.43</td>
<td>.26</td>
<td>.22</td>
</tr>
<tr>
<td>Poorly Fitted Wood Sash</td>
<td>1.20</td>
<td>.37</td>
<td>.60</td>
</tr>
<tr>
<td>Metal Sash</td>
<td>.80</td>
<td>.35</td>
<td>.40</td>
</tr>
</tbody>
</table>

### 5.4.2 Ventilation

Introducing Outdoor air for ventilation of conditioned spaces is necessary and required in order to dilute the odors given off by people, smoking and other internal air contaminants.
Ventilation Standards

The amount of ventilation required varies with several factors, such as:

- Total number of people within the conditioned space
- The height of the ceiling on the conditioned space
- The number of people smoking

People give off body odors which require a minimum of 5 cfm per person for a satisfactory dilution; however it is recommended to allow for 7.5 cfm per person. When people smoke additional odors are given off by cigarettes or cigars. To dilute these odors a minimum of 15 to 25 cfm per person is required, and if it is within a special gathering room (cigar lounge), 30 to 50 cfm per person is recommended.

Scheduled Ventilation

The capacity requirements can be reduced at times of peak in order to, minimize the outdoor air load. At times other than peak load, the calculate outdoor air quantity is used. This type of ventilation technique is recommended only for installations which operate for more than 12 hours, or 3 hours longer than there is occupancy, to allow for some time to flush out the buildings when no odors are being generated. It has been found through tests that as long as the flushing period is provided very few people complain about the stuffiness in the conditioned space. The procedure for estimating and controlling scheduled ventilation is:

<table>
<thead>
<tr>
<th>Net Outdoor Air*(cfm)</th>
<th>Door Infiltration(cfm)</th>
<th>Net Outdoor Air*(cfm)</th>
<th>Door Infiltration(cfm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>140</td>
<td>100</td>
<td>1370</td>
<td>1100</td>
</tr>
<tr>
<td>270</td>
<td>200</td>
<td>1480</td>
<td>1200</td>
</tr>
<tr>
<td>410</td>
<td>300</td>
<td>1560</td>
<td>1300</td>
</tr>
<tr>
<td>530</td>
<td>400</td>
<td>1670</td>
<td>1400</td>
</tr>
<tr>
<td>660</td>
<td>500</td>
<td>1760</td>
<td>1500</td>
</tr>
<tr>
<td>790</td>
<td>600</td>
<td>1890</td>
<td>1600</td>
</tr>
<tr>
<td>920</td>
<td>700</td>
<td>2070</td>
<td>1700</td>
</tr>
<tr>
<td>1030</td>
<td>800</td>
<td>2250</td>
<td>1800</td>
</tr>
<tr>
<td>1150</td>
<td>900</td>
<td>2450</td>
<td>1900</td>
</tr>
<tr>
<td>1260</td>
<td>1000</td>
<td>2650</td>
<td>2400</td>
</tr>
</tbody>
</table>

Table 5.1: OFFSETTING SWINGING DOOR INFILTRATION WITH OUTDOOR AIR
• In estimating cooling loads, reduce air quantity at design conditions to a minimum of 40% of the recommended air quantity.

• Use a dry-bulb thermostat following the cooling and dehumidifying apparatus to control the leaving dewpoint such that:

  With the dewpoint at design, the damper motor closes the outdoor air damper to 40% of the design ventilation air quantity.

  As the dew point decreases below design, the outdoor air damper opens to the design setting.
Appendix A

Refrigerants, Brines, and Oils

A.1 Refrigerants

Refrigerants are the working fluids in refrigeration, air-conditioning, and heat pumping systems. They absorb heat from one area, such as an air-conditioned space, and reject it into another such as outdoors, usually through evaporation and condensation, respectively. These phase changes occur both in absorption and mechanical vapor compression systems, but they do not occur in systems operating on a gas cycle using a fluid such as air.

A.1.1 Refrigerant Properties

EPA ~ Under the Clean Air Act Amendments of 1990, the US Environmental Protection Agency is required to evaluate substitutes for ozone depleting substances. Both new and existing alternatives will be reviewed and evaluated through EPA’s Significant New Alternatives Policy (SNAP) program on the basis of a substance’s ozone depletion potential, and global warming potential, flammability, toxicity, exposure potential, and economic and technical feasibility.

Replacement Alternatives

Suva MP39 and Suva MP66 are a mixture of 3 components, HCFC-22, HFC-152A and ACFH-124, which have been created to replace CFC-12 in medium temperatures. The majority of compressors for CFC-12 can adapt to the use of MP39 and MP66 easily and economically. This allows for the equipment already installed to continue in service during their lifetime.

The two new refrigerants demonstrate many properties which have been improved towards the ozone and the environment, in comparison with CFC-12.

The MP mixtures are compatible with the majority of the materials used for the construction of the CFC systems. The composition of each mixture has been
selected to be compared with the efficiencies of CFC-12 in specified applications, in energetic capacity and efficiency. This gives as a result a minimal mechanical interruption when CFC-12 is replaced with MP66 or MP39. Other refrigerants like the HCFC-22 or the HCFC-134A, require an extensive mechanical intervention in the systems when they are replaced, which will at times be expensive.

MP66 and MP39 give the user the chance to change refrigerants without spending much in modifications to the CFC-12 systems.

CFC-134A can also be used in evaporation temperatures lower than $-7^\circ C$ and $20^\circ F$ but can demonstrate less capacity in these temperatures than the CFC-12, to cover the whole range of temperatures of the CFC-12, we also have the MP66 and the MP39.

Refrigerant Selection

MP39 can be used for the same things as CFC-12 in temperatures as low as $-23^\circ C$ or $(-10^\circ F)$ and higher. This is put in practice in:

- Cold Rooms
- Refrigerators
- Drinking Fountains
- Soda Dispensers

MP66 is used for temperatures lower than $-23^\circ C$ or $(-10^\circ F)$, therefore:

- Refrigerated hauling trucks
- Commercial freezers

A.2 Brines

At temperatures above $32^\circ F$, water is the most commonly used heat transfer medium for conveying a refrigeration load to an evaporator. At temperatures below $32^\circ F$, brines are used. They may be:

- An aqueous solution of inorganic salts, i.e. sodium chloride or calcium chloride. For low temperatures, a eutectic mixture may be used.

- An aqueous solution of organic compounds, i.e. alcohols or glycols. Ethanol water, methanol water, ethylene glycol and propylene glycol are examples.

- Chlorinated or fluorinated hydrocarbons and halocarbons.
A solution of any salt in water, or in general any solution, has a certain concentration at which the freezing point is at a minimum. A solution of such a concentration is called a eutectic mixture. The temperature at which it freezes is the eutectic temperature. A solution at any other concentration starts to freeze at a higher temperature.

A.2.1 Brine Selection

The selection of a brine is based upon a consideration of the following factors:

- **Freezing Point**—Brine must be suitable for the lowest operating temperature.

- **Application**—When using an open piping system, the possibility of product contamination by the brine should be checked.

- **Cost**—The initial charge and quantity of make-up required are factors in the determination of costs.

- **Safety**—Piping and system equipment material require a stable and relatively corrosive-free brine.

- **Codes**—Brine must be acceptable by codes ordinances, regulatory agencies and insuror.

The most common brines are aqueous solutions of calcium chloride or sodium chloride. Although both of these brines have the advantage of low cost, they have the disadvantage of being corrosive. To overcome corrosion, an inhibitor may be added to the brine. Sodium dichromate is a satisfactory and economical inhibitor. Sodium hydroxide is added to keep the brine slightly alkaline.

Sodium chloride is cheaper than calcium chloride brine; however it cannot be used below its eutectic point of -6 °F. It is preferred where contact with calcium chloride cannot be tolerated, for example, with unsealed foodstuffs. The use of calcium chloride of commercial grade is not satisfactory below -40 °F.

Load, brine quantity and temperature rise are all related to each other so that, when any two are known, the third may be found by the formula:

\[
\text{Load (tons)} = \frac{\text{gpm} \times \text{temp rise (°F)} \times \text{sp gr} \times C_p}{24} \tag{A.1}
\]

where:

- \(\text{sp gr} = \text{specific gravity of brine}\)
- \(C_p = \text{specific heat of brine (Btu/ lb-° F)}\)
A.2.2 Piping

All materials in the piping system including flange gaskets, valve seats and packing, pump seals and other specialities must be compatible with the brine. Copper tubing (except for the salt brines) and standard steel pipe are suitable for general use. The pump rating and motor horsepower should be based on the particular brine used and the actual operating temperature.

Friction Loss

To determine friction loss in a brine piping system, the engineer should first calculate the loss as if water were being used. A multiplier is then used to convert the calculated loss for the brine system. The multiplier is calculated as such:

$$\text{Multiplier} = \text{sp gr} \times \frac{f_b}{f_w} \quad (A.2)$$

where:

$$\text{sp gr} = \text{specific gravity of brine}$$
$$f_b = \text{friction factor for the brine}$$
$$f_w = \text{friction factor for water at the brine velocity}$$

Friction factor is determined from the Reynolds number. The Reynolds number is:

$$\text{Re} = \frac{7740 \times d \times v \times \text{sp gr}}{\mu'} \quad (A.3)$$

where:

$$d = \text{inside pipe diameter (in.)}$$
$$v = \text{brine velocity (ft/sec)}$$
$$\text{sp gr} = \text{specific gravity of brine} = \frac{\text{lb/cu ft}}{62.5}$$
$$\mu' = \text{viscosity (centipoises)} = \frac{\text{absolute viscosity, lb/(hr)(ft)}}{2.42}$$

Example 1—Friction Loss Multiplier

Given:
A 5/8 in. copper tube coil with a circuit water velocity of 4.29 ft/sec and a pressure drop of 7.5 psi. Mean water temperature = 55° F

Find:
Friction loss multiplier and pressure drop when using ethylene glycol at a mean brine temperature of 92.5° F and 41% solution by weight at the same circuit liquid velocity.

Solution: Referring to *chart 1 Carrier Part*

\[
\frac{\epsilon}{d} = \frac{0.00006}{0.575} = 0.000104 \quad \text{(A.4)}
\]

where:

\[
\epsilon = \text{absolute roughness of drawn tubing} \quad d = \text{inside diameter of 5/8 in. copper tubing}
\]

Referring to *Chart 19 (Carrier Manual)*. The Specific gravity of ethylene glycol at a mean brine temperature of 92.5° F and 41% solution by weight is 1.05. Referring to *Chart 18 (Carrier Manual)*. Viscosity of ethylene glycol at the same conditions equals 2.1 centipoises.

\[
Re = \frac{7740 \times 0.575 \times 4.29 \times 1.05}{2.1} = 9,520 = 9.52 \times 10^{3}
\]

Referring to *Chart 1 (Carrier Manual)*. For a Reynolds number of 9.52 \times 10^{3} and a relative roughness of 0.000104, the chart indicates friction factor \(f_b=0.031\).

Specific gravity of fresh water at a mean temperature of 55° F = 1.00

Referring to *Chart 28 (Carrier Manual)*. Viscosity of fresh water at a mean temperature of 55° F = 1.2 centipoises.

\[
Re(\text{water}) = \frac{7740 \times 0.575 \times 4.29 \times 1.00}{1.2} = 15,900 = 1.59 \times 10^{4}
\]

Referring again to *Chart 1 (Carrier Manual)*. For a Reynolds number of 1.59 \times 10^{4} and a relative roughness of 0.000104, the chart indicates a friction factor \(f_w=0.027\).

Friction Multiplier = sp gr(brine) \times \frac{f_b}{f_w} \quad \text{(A.5)}

\[
= 1.05 \times \frac{0.031}{0.027} = 1.21
\]

Brine friction loss = 1.21 \times 7.5psi = 9.08psi \quad \text{(A.6)}

or

\[
= \frac{9.08 \times 2.31}{1.05} = 20.0 \text{ft brine}
\]
Pump Brake Horsepower

To determine the horsepower required by a pump with brine, the following formula may be used:

\[ \text{bhp} = \frac{\text{gpm} \times \text{total head(ft. brine)} \times \text{sp gr}}{3960 \times \text{eff}} \]  

(A.7)

where:

- \( \text{gpm} = \text{gallons/min. of brine} \)
- \( \text{total head} = \text{total pump head (ft. brine)} \)
- \( \text{sp gr} = \text{specific gravity of brine} \)
- \( \text{eff} = \text{pump efficiency} \)

A.3 Refrigeration Oils

A.3.1 Classification

Oils classified by source fall into three main groups: Animal, Vegetable, and Mineral.

- Animal and Vegetable oils are called fixed oils because they cannot be defined without decomposing.

  They are also unstable and tend to form acids and gums that make them unsuitable for refrigeration purposes.

- There are three major classifications for Mineral oils: Naphthenic based, Paraffin based, and Mixed base.

  Naphthenic oil yields a residue of heavy pitch or asphalt when distilled.
  Paraffin oil yields a paraffin wax when distilled.
  Mixed oil contains both naphthenic and paraffinic oil.

Experience has shown that the naphthenic base oils are more suited for refrigeration work for three main reasons:

- They flow better at low temperatures
- Carbon deposits from these oils are of a soft nature and can easily be removed.
- They deposit less wax at low temperatures
A.3.2 Properties

To meet the requirements of a refrigeration system, a good refrigeration oil should:

- Maintain sufficient body to lubricate at high temperature and yet be fluid enough to flow at low temperature.
- Have a pour point low enough to allow flow at any point in the system.
- Leave no carbon deposits when in contact with hot surfaces encountered in the system during normal operation.
- Deposit no wax when exposed to the lowest temperatures normally encountered in the system.
- Contain little or no corrosive acid.
- Have a high resistance to the flow of electricity.
- Have a high flash and fire point to indicate proper blending.
- Be stable in the presence of oxygen.
- Contain no sulfur compounds.
- Contain no moisture
- Be light in color, to indicate proper refining.

As lubricating oils for refrigeration compressors are a specialty product, they require consideration apart from normal lubricants. The emphasis in this chapter is on oil used in refrigeration. Do not consider the emphasis as applicable to lubricants in general.
Appendix B

Radiation

Radiation, one of the basic mechanisms for energy transfer between different temperature regions, is distinguished from conduction and convection in that it does not depend on an intermediate material between the regions. The radiation energy transfer process is the consequence of energy-carrying electromagnetic waves that are emitted by atoms and molecules due to changes in their energy content. In any case we wish to attribute to radiation the standard wave properties of frequency $\nu$ and wavelength $\lambda$. For radiation propagating in a particular medium, the two properties are related by:

$$\lambda = \frac{c}{\nu} \quad (B.1)$$

where $c$ is the speed of light in the medium. These waves are, in turn, sustained by the internal energy, and therefore the temperature, of the material. Therefore emission of thermal radiation can be associated with thermally excited conditions within the material. The amount and characteristics of radiant energy emitted by a quantity of material depend on the nature of the material, its microscopic arrangement, and its absolute temperature. Although rate of energy emission is independent of the surroundings, the net energy transfer rate depends on the temperatures and spatial relationships of the surface and its surroundings.
B.1 Blackbodies

B.1.1 Blackbody Radiation

A blackbody is an ideal surface having the following properties:

1. A blackbody absorbs all incident radiation, regardless of wavelength and direction.

2. For a prescribed temperature and wavelength, no surface can emit more energy than a blackbody.

3. Although the radiation emitted by a blackbody is a function of wavelength and temperature, it is independent of direction. That is, the blackbody is a diffuse emitter.

The blackbody is the perfect absorber and emitter, and so it serves as a standard against which the radiative properties of actual surfaces may be compared.

Although approximated by some surfaces, it seems imperative to note that no surface has precisely the properties of a blackbody. The closest approximation is achieved by a cavity whose inner surface is at a uniform temperature. If radiation enters the cavity through a small aperture Figure (B.1(a)), it is likely to experience many reflections before reemergence. Therefore it can be assumed that it has been absorbed by the cavity, and so blackbody behavior is very similar. It can then be argued through thermodynamic principles that the radiation leaving the aperture depends only on the surface temperature and is correspondent to blackbody emission Figure (B.1(b)). Since blackbody emission is diffuse, the spectral intensity $I_{\lambda,b}$ of radiation leaving the cavity is independent of direction. Therefore the radiation field in the cavity, tends to be the cumulative effect of emission and reflection from the cavity surface, and thus must have the same form as that which emerges from the aperture, and so a blackbody radiation field must exist within the cavity. Any small surface in the cavity experiences irradiation as in Figure (B.1(c)), for which $G_{\lambda} = E_{\lambda,b}(\lambda, T)$. Regardless of orientation such a surface would be diffusely irradiated. Blackbody will exist within the cavity, regardless of whether the cavity surface is reflective of absorptive.
Diffuse emission
\( I_{\lambda,0} = I_{\lambda,b} \)

Isothermal Surface, \( T \)

(a) (b) (c)

Figure B.1: Characteristics of an isothermal blackbody cavity. (a) Complete absorption. (b) Diffuse emission from an aperture. (c) Diffuse irradiation of interior surfaces.

The Planck Distribution

The spectral distribution was first determined by Planck. It is of the form:

\[
I_{\lambda,b}(\lambda, t) = \frac{2hc^2}{\lambda^5 \left[ e^{hc/\lambda kT} - 1 \right]} \tag{B.2}
\]

Since the blackbody is a diffuse emitter, its spectral emission power is of the form:

\[
E_{\lambda,b}(\lambda, t) = \pi I_{\lambda,b}(\lambda, t) = \frac{2\pi hc^2}{\lambda^5 \left[ e^{hc/\lambda kT} - 1 \right]} \tag{B.3}
\]

where:

- \( h = 6.6256 \times 10^{-34} \) Js (Planck’s Constant)
- \( k = 1.3805 \times 10^{-23} \) J/K (Boltzmann’s Constant)
- \( c_o = 2.998 \times 10^8 \) m/s (Speed of light in vacuum)
- \( T = \text{absolute temperature of the blackbody (K)} \)
- \( \lambda = \text{wavelength} \)
However several special important features should be noted;

1. The emitted radiation varies *continuously* with wavelength.

2. At any wavelength the magnitude of the emitted radiation increases with increasing temperature.

3. The spectral region in which the radiation is concentrated depends on temperature, with *comparatively* more radiation appearing at shorter wavelengths as the temperature increases.

4. A significant fraction of the radiation emitted by the sun, which may be approximated as a blackbody at 5800 K, is in the visible region of the spectrum. In contrast, for $T \leq 800$ K, emission is predominantly in the infrared region of the spectrum and is not visible to the eye.

**Wien’s Displacement Law**

Blackbody spectral distribution has a maximum and that the corresponding wavelength $\lambda_{\text{max}}$ depends on the temperature. The nature of this dependence is expressed as

$$\lambda_{\text{max}}T = C_3$$  \hspace{1cm} (B.4)

where the $C_3$ is a radiation constant ($2897.8 \, \mu mK$).

**The Stefan-Boltzmann Law**

The total emissive power of a blackbody $E_b$ may also be expressed as:

$$E_b = \int_0^\infty \frac{2\pi \hbar c^2}{\lambda^5 \left[ \exp \left( \frac{\hbar c}{\lambda kT} \right) - 1 \right]} d\lambda$$  \hspace{1cm} (B.5)

Performing the integration it can be shown that:

$$E_b = \sigma T^4$$  \hspace{1cm} (B.6)

where the *Stefan-Boltzmann constant/law*, has the numerical value of:

- $\sigma = 5.670 \times 10^{-8} \, \text{W/m}^2\text{K}^4$

It enables the calculation of the amount of radiation emitted in all directions and over all wavelengths simply from knowledge of the temperature of the blackbody.
\section*{B.1.2 Blackbody Radiation Exchange}

Generally, radiation may leave a surface due to both reflection and emission, and upon reaching a second surface, experiencing reflection and absorption. This is however simplified for surfaces that are approximated as blackbodies, since reflection is non-existent for these. Therefore for these bodies, energy only leaves as a result of emission, and all incident radiation is absorbed.

For example, when there is radiation exchange between two black surfaces of arbitrary shape Figure\ref{fig:B.2}. Defining $q_{i \rightarrow j}$ as the rate at which radiation leaves surface $i$ and is intercepted by surface $j$, it follows that:

\begin{equation}
q_{i \rightarrow j} = (A_i J_i) F_{ij}
\end{equation}

or, since the radiosity equals emissive power for a black surface i.e. $(J_i = E_{bi})$, then:

\begin{equation}
q_{i \rightarrow j} = A_i F_{ij} E_{bi}
\end{equation}

\begin{figure}[h]
\centering
\includegraphics[width=0.8\textwidth]{figB2}
\caption{Radiation transfer between two surfaces that may be approximated as blackbodies.}
\end{figure}

It then becomes apparent that net rate at which radiation leaves surface $i$ as a result of interaction with $j$, which is equal to the rate at which $j$ gains radiation due to its interaction with $i$ (the net radiative exchange between two surfaces), can be expressed as:

\begin{align}
q_{ij} &= q_{i \rightarrow j} - q_{j \rightarrow i} \\
q_{ij} &= A_i F_{ij} E_{bi} - A_j F_{ji} E_{bj} \\
q_{ij} &= A_i F_{ij} \sigma (T_i^4 - T_j^4)
\end{align}

\section*{B.1.3 Kirchhoff’s Law}

Taking into consideration a large, isothermal enclosure of surface temperature $T_s$, wherein several small bodies are confined Figure\ref{fig:B.3}. These bodies will be
small in comparison with their enclosure, and will tend to have a negligible influence on the radiation field, which will mostly be due to the cumulative effect of emission and reflection by the enclosure surface (Such a surface as that enclosing these bodies becomes a blackbody cavity). However regardless of orientation, the irradiation experienced by any body in the cavity is diffuse and equal to emission from a blackbody at $T_s$. Also under steady state conditions, thermal equilibrium must exist between the bodies and the enclosure. Therefore:

$$G = E_b(T_s)$$  \hspace{1cm} (B.10)

Also the rate of energy transferred between surfaces must equal zero. So in an energy balance to a control surface about body 1, we yield:

$$\alpha G A - E(T_s) A_b = 0$$  \hspace{1cm} (B.11)

Which becomes

$$\frac{E(T_s)}{\alpha} = E_b(T_s)$$

Hence for any surface in the enclosure,

$$\varepsilon_{\lambda,\theta} = \alpha_{\lambda,\theta}$$

That is the total hemispherical emissivity of the surface is equal to its total hemispherical absorptivity. This derivation can also be repeated for spectral conditions considering that there are no restrictions involved in the spectral directional properties by Kirchhoff’s law.

This relation is known as Kirchoff’s law. A major implication is that, since $\alpha \leq 1$, $E(T_s) \leq E_b(T_s)$. Hence no real surface can have an emissive power exceeding that of a black surface at the same temperature, and the notion of the blackbody as an ideal emitter is confirmed.

### B.2 Surface Absorption, Reflection, and Transmission

In the most general situation the irradiation interacts with a semitransparent medium, such as a layer of water or a glass plate. Figure(B.4) shows for a spectral component of the irradiation, portions of this radiation may be reflected, absorbed, and transmitted. From a radiation balance on the medium, it follows that

$$G_{\lambda} = G_{\lambda,ref} + G_{\lambda,abs} + G_{\lambda,tr}$$  \hspace{1cm} (B.12)
The absorptivity is a property which determines the fraction of irradiation absorbed by a surface. Determination of the property is complicated by the fact that, like emission, it may be characterized by both a directional and spectral dependence. The spectral, directional absorptivity, $\alpha_{\lambda,\theta}(\lambda, \theta, \phi)$, of a surface is defined as the fraction of the spectral intensity incident in the direction of $\theta$ and
\( \phi \) that is absorbed by the surface. Therefore:

\[
\alpha_{\lambda, \theta}(\lambda, \theta, \phi) \equiv \frac{I_{\lambda,i,abs}(\lambda, \theta, \phi)}{I_{\lambda,i}(\lambda, \theta, \phi)}
\]  

(B.13)

This expression neglects any dependence of the absorptivity on the surface temperature. This is because such a dependence is small for most spectral radiative properties.

It is implicit in the foregoing result that surfaces may exhibit selective absorption with respect to wavelength and direction of the incident radiation.

The total, hemispherical absorptivity, \( \alpha \), represents an integrated average over both direction and wavelength.

\[
\alpha_{\lambda}(\lambda) = \frac{\int_0^{2\pi} \int_0^{\pi/2} \alpha_{\lambda, \theta}(\lambda, \theta, \phi) I_{\lambda,i}(\lambda, \theta, \phi) \cos \theta \sin \theta d\theta d\phi}{I_{\lambda,i}(\lambda, \theta, \phi) \cos \theta \sin \theta d\theta d\phi}
\]

\[
\alpha_{\lambda}(\lambda) = 2 \int_0^{\pi/2} \alpha_{\lambda, \theta}(\lambda, \theta, \phi) \cos \theta \sin \theta d\theta d\phi
\]

It is defined as the fraction of the total irradiation absorbed by a surface:

\[
\alpha \equiv \frac{G_{abs}}{G}
\]  

(B.14)

Accordingly \( \alpha \) depends on the spectral distribution of the incident radiation, as well as on its directional distribution and the nature of the absorbing surface. Note that although \( \alpha \) is approximately independent of surface temperature, the same may not be said for the total hemispherical emissivity, \( \varepsilon \).

However since \( \alpha \) depends on the spectral distribution of the irradiation, its value for a surface exposed to solar radiation may differ appreciably from its value for the same surface exposed to longer wavelength radiation originating from a source of lower temperature.

### B.2.2 Reflectivity

The reflectivity is a property that determines the fraction of the incident radiation reflected by a surface. However, its specific definition takes different forms, because its property is bidirectional. That is, in addition to depending on the direction of the incident radiation, it depends on the reflecting radiation. This complication is avoided by taking into account only an integrated average over the hemisphere associated with the reflected radiation and therefore provides no information concerning the directional distribution of this radiation. The spectral directional reflectivity, \( \rho_{\lambda, \theta}(\lambda, \theta, \phi) \), of a surface is defined as the fraction of the
spectral intensity incident in the direction of \( \theta \) and \( \phi \), which is reflected by the surface. Therefore:

\[
\rho_{\lambda,\phi}(\lambda, \theta, \phi) = \frac{I_{\lambda,i,ref}(\lambda, \theta, \phi)}{I_{\lambda,i}(\lambda, \theta, \phi)} \quad (B.15)
\]

\[
\rho_{\lambda}(\lambda) = \frac{G_{\lambda,ref}(\lambda)}{G_{\lambda}(\lambda)} \quad (B.16)
\]

\[
\rho_{\lambda}(\lambda) = \frac{\int_{0}^{2\pi} \int_{0}^{\pi/2} \rho_{\lambda,\phi}(\lambda, \theta, \phi) I_{\lambda,i}(\lambda, \theta, \phi) \cos \theta \sin \theta d\theta d\phi}{I_{\lambda,i}(\lambda, \theta, \phi) \cos \theta \sin \theta d\theta d\phi}
\]

The total, hemispherical reflectivity \( \rho \) is then defined as the fraction of the spectral irradiation that is reflected by the surface, in which case:

\[
\rho = \frac{G_{\text{ref}}}{G} \quad (B.17)
\]

Surfaces may be idealized as diffuse or specular, according to the manner in which they reflect radiation Figure (B.5). Diffuse reflection occurs if, regardless of the direction of the incident radiation, the intensity of the reflected radiation is independent of the reflection angle. Specular reflection in contrast is said to occur when the angle of reflection \( \theta_2 \) equals the angle of incidence \( \theta_1 \). Although no surface is either a perfect diffusive or specular surface, their conditions are more closely approximated by a polished, mirrorlike surface for specular reflection, and by a rough surface for diffuse reflection.

Figure B.5: Diffuse and Specular reflection

### B.2.3 Transmissivity

Although treatment of the response of a semitransparent material to incident radiation becomes a problem, reasonable results are often obtained through the use of hemispherical transmissivities defined as:

\[
\tau_{\lambda} = \frac{G_{\lambda,tr}(\lambda)}{G_{\lambda}(\lambda)} \quad (B.18)
\]
\[
\tau = \frac{G_{tr}}{G}
\]  

(B.19)

### B.2.4 Special Considerations

In conclusion for a semitransparent medium. Using a radiation balance, and for properties averaged over the entire spectrum, it follows that:

\[
\alpha_\lambda + \rho_\lambda + \tau_\lambda = 1
\]

(B.20)

However, if the medium is opaque, there is no transmission, and absorption and reflection are the only surface processes taken into account. Then:

\[
\alpha_\lambda + \rho_\lambda = 1
\]

(B.21)

### B.3 Environmental Radiation

Solar radiation is essential to all life on earth. Through the process of photosynthesis, it satisfies our need for dood, fiber, and fuel. Moreover, through thermal and photovoltaic processes, it has the potential to satisfy much of our demand for space heat, process heat, and electricity.

The sun is a nearly spherical radiation source that is \(1.39 \times 10^9\) m in diameter and is located \(1.50 \times 10^{11}\) m from the earth. With respect to the magnitude and the spectral and directional dependence of the incident solar radiation, it is necessary to distinguish between conditions at the earth's surface and at the outer edge of the earth's atmosphere. For a horizontal surface outside the earth's atmosphere, solar radiation appears as a beam of nearly parallel rays that form an angle \(\theta\), the zenith angle, relative to the surface normal Figure (B.6).

The extraterrestrial solar irradiation \(G_{s,o}\) depends on the geographic latitude, as well as the time of day and year. It is determined by an expression of the form:

\[
G_{s,o} = S_e f \cos \theta
\]

(B.22)

where:

- \(S_e\) is the solar constant

  which is the flux of solar energy incident on a surface normal to the sun's rays with the earth at a mean distance from the sun.

  at this point \(S_e = 1353\) W/m\(^2\)
The quantity $f$ is a small correction factor
it accounts for the eccentricity of the earth’s orbit around the sun
$(0.97 \leq f \leq 1.03)$

$\theta$ is the angle of incidence as shown in figure(B.6)

Long wavelength forms of environmental radiation include emission from the earth’s surface, as well as emission from certain atmospheric constituents. The emissive power associated with the earth’s surface may be computed in the conventional manner. That is:

$$E = \varepsilon \sigma T^4$$  \hspace{1cm} (B.23)

where:

- $\varepsilon$ is the surface emissivity
  Emissivities are generally close to unity
  water for example, is approximately 0.97

- $T$ is the surface temperature
  Temperatures range from 250 to 320 K

- Emission is concentrated on the spectral region
  it is approximately 4 to 40 $\mu$m, peaking at 10 $\mu$m
Atmospheric emission is largely from the CO₂ and H₂O molecules and is concentrated in the spectral regions from 5 to 8 μm and above 13 μm. Although the spectral distribution of atmospheric emission does not correspond to that of a blackbody, its contribution to irradiation of the earth’s surface is estimated using:

\[ G_{atm} = \sigma T_{sky}^4 \]  

(B.24)

where:

- \( T_{sky} \) is the effective sky temperature
  - its values depend on atmospheric conditions
  - it ranges from 230 K under a cold clear sky to 285 under warm cloudy conditions

At night, atmospheric emission is the only source of earth irradiation.

Since solar radiation is concentrated in the short wavelength region of the spectrum and surface emission is at much longer wavelengths, it follows that many surfaces may not be approximated as gray in their response to solar irradiation. Basically solar absorptivity of a surface \( \alpha_S \) may differ from its emissivity \( \varepsilon \).

The table below shows values of \( \alpha_S \) and the emissivity at moderate temperatures for representative surfaces.

<table>
<thead>
<tr>
<th>Surface</th>
<th>( \alpha_S )</th>
<th>( \varepsilon ) (300 K)</th>
<th>( \alpha_S/\varepsilon )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporated aluminum film</td>
<td>0.09</td>
<td>0.03</td>
<td>3.0</td>
</tr>
<tr>
<td>Fused quartz on aluminum film</td>
<td>0.19</td>
<td>0.81</td>
<td>0.24</td>
</tr>
<tr>
<td>White paint on a metallic substrate</td>
<td>0.21</td>
<td>0.96</td>
<td>0.22</td>
</tr>
<tr>
<td>Black paint on a metallic substrate</td>
<td>0.97</td>
<td>0.97</td>
<td>1.0</td>
</tr>
<tr>
<td>Stainless steel, as received, dull</td>
<td>0.50</td>
<td>0.21</td>
<td>2.4</td>
</tr>
<tr>
<td>Red brick</td>
<td>0.63</td>
<td>0.93</td>
<td>0.68</td>
</tr>
<tr>
<td>Snow</td>
<td>0.28</td>
<td>0.97</td>
<td>0.29</td>
</tr>
<tr>
<td>Corn leaf</td>
<td>0.76</td>
<td>0.97</td>
<td>0.78</td>
</tr>
</tbody>
</table>