CHAPTER II
THEORY

2.1 Theory and working principle of drying

Drying is defined as a process of moisture removal due to simultaneous heat and mass transfer. Heat transfer from the surrounding environment evaporates the surface moisture. The moisture can be either transported to the surface of the product or then evaporated, or evaporated internally at a liquid vapor interface and then transported as vapor to the surface. The transfer of energy depends on the air temperature, air humidity, airflow rate, and exposed area of material and pressure. The physical nature of the material, including temperature, composition, and in particular moisture content, govern the rate of moisture transfer. The dehydration equipment generally utilizes conduction, convection, or radiation to transfer energy from a heat source to the material. The heat is transferred directly form a hot gas or indirectly thorough a metal surface. (Soponronnarit, 1997).The drying is a complex operation involving transient transfer of heat and mass along with several rate processes, such as physical or chemical transformations, which in turn, may cause changes in product quality as well as the mechanisms of heat and mass transfer. Physical changes that may occur include shrinkage, puffing, crystallization and glass transition. Drying occurs by effecting vaporization of the liquid by supplying heat to the wet feedstock. As noted earlier, heat may be supplied by convection (direct dryer), by conduction (contact or indirect dryer), radiation or volumetrically by placing the wet material in a microwave or radio frequency electromagnetic field. Transport of moisture within the solid may occur by any one or more of the following mechanisms of mass transfer.

The drying process cannot be discussed without a thorough understanding of the basic heat and mass transfer concepts. The typical drying cycle consists of three stage: heat the food to the drying temperature, evaporation of the moisture from the solid surface occurring at a rate proportional to the moisture content, and once the critical moisture point is reached, the falling rate associated with drying is set in motion.

2.1.1 Moisture content

The moisture content of solid denotes the quantity of water per unit mass of either wet or dry solid. It is usually on a percentage basis. The moisture on a wet basis (%, M_{wb}) is defined as:

$$M_{wb} = \frac{m_w}{m_w + m_d} x100 \qquad (2.1)$$

The moisture content dry basis (% M_{db}) is defined as

$$M_{db} = \frac{m_w}{m_d} x100 \qquad (2.2)$$

Where m_w is mass of water in product (kg) and m_d is mass of bone dry material (kg).

The moisture content in a product is an indicator of its quality and key to safe storage. Farmers and commercial operators use the wet basis moisture content. While, engineers and scientist employ moisture content on a dry basis because it is easier to use in drying calculation. Conversion from a dry basis (%M_{db}) to a wet basis (%M_{wb}) moisture content, and vice versa, is done as follows:

$$M_{wb} = \frac{100 M_{db}}{100 + M_{db}} \qquad (2.3)$$

$$M_{db} = \frac{100 M_{wb}}{100 - M_{wb}} \qquad (2.4)$$

2.1.2 Drying characteristics

The drying behavior of a product can be characterized by measuring the moisture content loss as a function of time. Menon and Mujumdar (1987) have defined a generalized drying curve that includes an induction period, a constant drying rate period and falling drying rate period. After the induction period, where the food is heated to the drying temperature, the product undergoes a constant drying rate when a film of water is freely available at the drying surface for evaporation into the drying medium. During this period the rate of heat transfer to the solid and the rate of mass transfer to the air will be in equilibrium. The surface maintains a constant temperature equal to the wet-bulb temperature of drying air in the hot air during this

process. The factors, which control the rate of drying during this period, are air temperature, air velocity, total pressure and partial pressure of vapor.

The falling rate-drying period is the indicative of an increased resistance to both heat and mass transfer. This period occur when the free water at surface no longer exists and water to be evaporated comes from within the structure and must be transported to the surface. In this period the surface temperature increases above the wet-bulb temperature and the rate of drying is influenced mainly by the factors, which control the movement of water within the solid and external factors become less important. The drying rate decreases and the moisture content of material slowly reaches the equilibrium moisture content corresponding to the actual relative humidity of the air.

2.1.3 Thermodynamic properties of drying air

Moist air, which is a mixture of dry air and water vapor, almost always is used both to supply the heat for evaporation and to carry away the evaporated moisture from solid. Dry air consists of a number of gases, mainly oxygen and nitrogen plus minor components such as argon, carbon dioxide, and neon. Moist air contains a varying amount of water vapor. Although the mass fraction of water vapor in the air used for drying is always less than one tenth, the presence of water vapor molecules has a profound effect on the drying process. Thus, thermodynamic properties of humid air are required for the design calculations of such dryers. The definitions discussed bellow are given in Soponrounnarit (1997) and Brooker *et al.*(1992).

Three humidity terms are used in the solid drying literature to describe the amount of water vapor held in the drying air: vapor pressure, relative humidity, and humidity ratio. The temperatures of moist air may be referred to as dry-bulb and wet-bulb temperatures. Two additional moist air properties frequently used in solid drying calculatetions are enthalpy and specific volume. These seven moist-air thermodynamic properties are defined in the following paragraphs.

a) Vapor pressure

The water vapor pressure (P_v) is the partial pressure exerted by the water vapor molecules of a solid or liquid in moist air. When air is fully saturated with water vapor, its vapor pressure is called the saturated vapor pressure (P_{vs})

b) Relative humidity

The relative humidity (RH) is the ratio of the mole fraction (or vapor pressure) or water vapor in the air to the mole fraction (or vapor pressure) of the water vapor in saturated air at the same temperature and atmospheric pressure. The relative humidity is expressed as a decimal or percentage.

c) Humidity ratio

The humidity ratio (*w*) is the mass of the water vapor contained in the moist air per unit mass of dry air. Other terms used for humidity ratio are absolute humidity, moisture content and specific humidity.

d) Dry-bulb temperature

The dry-bulb temperature (T_{db}) is the temperature of moist air indicated by an ordinary thermometer.

e) Wet-bulb temperature

A distinction should made between the psychometric and thermodynamic wet-bulb temperature. The psychometric wet-bulb temperature (T_{wb}) is the temperature of moist air indicated by a thermometer whose bulb is covered with a wet wick. The airflow passing over the wick should have a velocity of at least 4.6 m/s (Soponronnarit, 1997).

The thermodynamic wet-bulb temperature (T_{wb}) is the temperature reached by moist air and water if the evaporating water adiabatically saturates the air. The psychometric and thermodynamic wet-bulb temperatures of moist air are nearly equal.

f) Enthalpy

The enthalpy (*h*) of a dry air-water mixture is the heat content of the moist air per unit mass of dry air above a certain reference temperature. Since only differences in enthalpy are of practical engineering interest, the choice of the reference temperature is non-consequential. For liquid water the reference temperature usually chosen is 0°C.

g) Specific volume

The specific volume (*v*) of moist air is defined as the volume per unit mass of dry air. The specific density of the moist air is equal to the reciprocal of its specific volume.

2.1.4 Psychometric chart and drying process

A psychometric chart is a special chart that contains values of the most common thermodynamic properties of moist air: dry-bulb temperature, wet-bulb temperature, dew point temperature, humidity ratio, relative humidity, specific volume and enthalpy. Several processes relative to solid conditioning can be represented conveniently on the psychometric chart. Sensible heating and drying processes is presented in this section. Other processes are presented in Brooker *et al.*, 1992.

a) Sensible heating and cooling

During sensible heating and cooling of the air at constant humidity ratio, heat is added to or withdraws from the drying air in a heat exchanger as in an indirect heater (for solid drying) or in an evaporator (for solid chilling)

The processes of sensible heating and cooling are represented on the psychometric chart by straight lines parallel to the abscissa (Figure. 2.1); both result in changes in dry-bulb and wet-bulb temperatures, enthalpy, specific volume, and relative humidity of the moist air. Assuming that no drying take place, the humidity (moisture content) of the air remains the same because no water is added or removed.
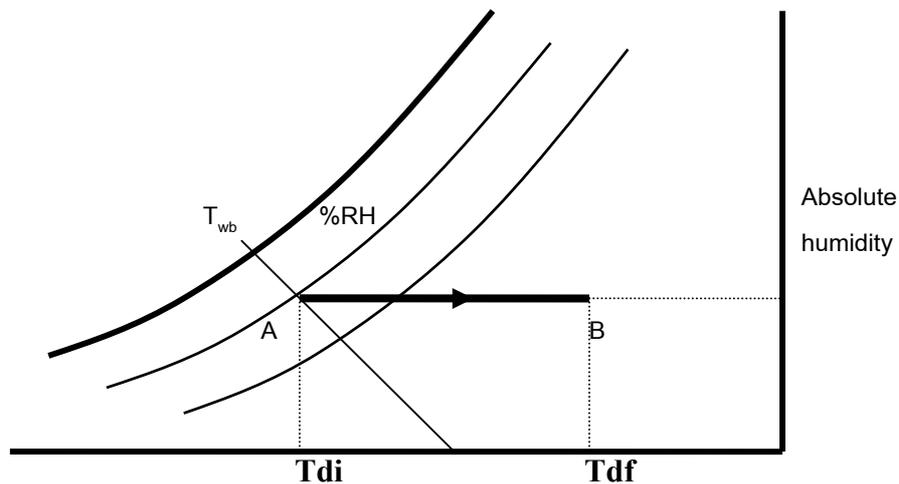

Figure 2.1 Representation on the psychometric chart of the sensible heating and cooling process of moist air. (Brooker *et al*., 1992.)

b) Drying processes

The drying of a mass of solid in a dryer can be considered an adiabatic process. The drying air, without transfer of heat supplies the heat required for evaporation of the solid moisture by conduction or radiation to or from the surroundings. As the air pass through the wet solid mass, a large part of the sensible heat of the air is transformed into latent heat as a result of the increasing amount of water held in the air as vapor. During the adiabatic process there will be an increase in the humidity ratio and relative humidity, the vapor pressure, and the dew point temperature. The enthalpy and wet-bulb temperature remain practically constant during the adiabatic drying process. The process of solid drying is illustrated in Figure 2.2
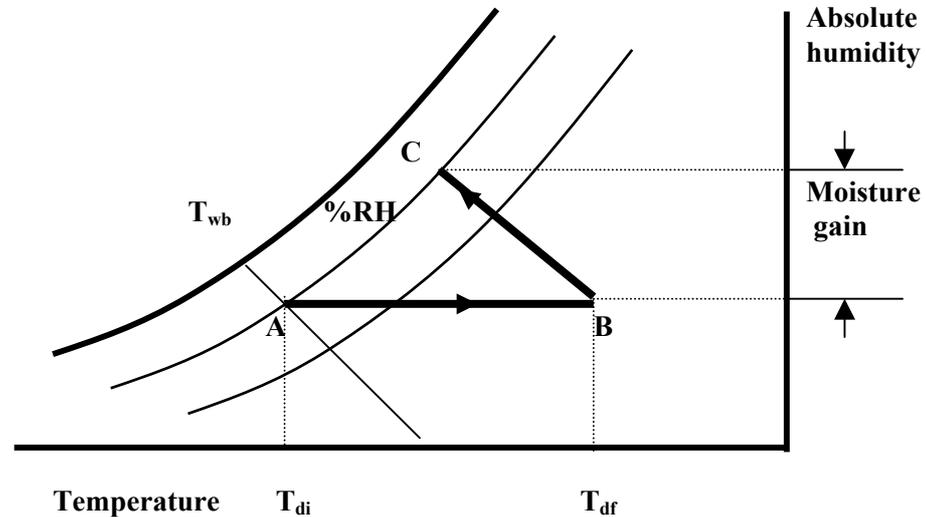
Figure 2.2 Representation on the psychometric chart of the solid drying adiabatically.
(Brooker *et al.*, 1992.)

Assuming the dry-bulb temperature, T_{di}, and wet-bulb temperature, T_{wi}, of the ambient air are known, then the initial state can be determined (point A in Fig. 2.2). The corresponding relative humidity can be read directly from the diagram, as also the absolute humidity. The horizontal line from A shows the process when the air is heated without adding or losing moisture to B. This is the sensible heating processes. It can also be seen that the relative humidity of the air decreases. From state B, the hot air is driven past the product. No heat gain or loss occurs in the system during this process, called an adiabatic process or drying process. This is shown along the part BC. The drying processes take place because the hot air become cooler and receive moisture from the product. The part BC carried out at constant enthalpy that should be identical with a process at constant wet-bulb temperature.

2.1.5 Equilibrium moisture content

The equilibrium moisture content (M_{eq}) is the limiting moisture to which a given material can dry under specific conditions of air temperature and relative

humidity. This value depends on the relative humidity and temperature conditions of the environment as well as on the species, variety and maturity of fruit or food products. The relationship between the moisture content of any material and its equilibrium relative humidity at a constant temperature can be represented by a curve and this curve is called an isotherm. A sorption isotherm is a plot of a material, which has been subjected to a wetting environment. A desorption isotherm is a similar plot for a material which has been subjected to a drying environment.

The M_{eq} can be defined as the moisture content at which the internal product vapor pressure is in equilibrium with the vapor pressure of the environment. The M_{eq} usually decreases with an increase in temperature, solid (food material) properties such as species, variety and maturity and depend on whether the amount of humidity is shifted from high to low (desorption) or from low to high (absorption). The following direction of the mass transfer between the surface of the material and the surrounding air are possible:

i)   From the air to the solid when the partial vapor pressure of air is greater than internal product vapor pressure

ii)   From the solid to the air when the partial vapor pressure of air is smaller than internal product vapor pressure

iii)   Rate of water absorption by the solid equals the rates of water desorption from the solid when the partial vapor pressure of air is equal to internal product vapor pressure.

Several theoretical, semi-theoretical and empirical models have been proposed to describe the isotherm of agricultural products. In the literature there are several relationships describing experimental sorption isotherms for different temperatures by means of approximation. The important M_{eq} models compiled of Oswin model is shown in equation

$$M_{eq}=A[RH/(1-RH)]^{B} \tag{2.5}$$

Where, M_{eq} = the equilibruim moisture content

RH = the relative humidity

A, B = constant whivh are the function of absorption temperature.

This model have been developed for longan drying by Vongvanichakul *et al.* (1997) shown in equation 2.6 to 2.8 as follow:

$$M_f = M_i - (M_{in} - M_{eq})\Delta tK \exp(-Kt) \tag{2.6}$$

$$K = 0.0023(T_{di} + 273.15) - 0.739 \tag{2.7}$$

$$M_{eq} = A[RH_{di} / (1 - RH_{di})]^B \tag{2.8}$$

Where,

A = 2.3015-0.00615 (T_{di}+273.15)

B = -1.3453 + 0.00507 (T_{di}+273.15)

M_{eq} is equilibrium moisture content, RH is relative humidity and T is absolute temperature. A, B are constants and determined experimentally.

2.2 Theory and working principle of heat pump

The heat pump is the system to recover the waste heat. So, application of the heat pump with conventional dryer is possible. The design of heat pump is developed around the application of a refrigeration system, which uses heat rejected from the condenser instead of simply being dissipated to the atmosphere.

A heat pump system (HPS) can be applied instead of the conventional dryer as in Figure 2.3. A condenser unit of the HPS releases heat to the inlet air and increases its temperature. A heater could be used to keep the air temperature at a specified value required for an agricultural product. After passing the drying bed, the air will be drawn through an evaporator section of the HPS. The moisture of the air will be removed keeping the humidity at very low value. The air can either be mixed with the fresh air or recirculated before passing it to the condenser again according to the required condition. By this method, higher quality of hot air (no burnt-fuel smell) can be achieved. This system will be named Heat-Pump Dryer (HPD).

All refrigeration systems are heat pumps, because they absorb heat energy at a low temperature level and discharge at a high temperature level. The design of heat pump is arrived at by using the principles of a refrigeration system where the heat rejected at condenser is used instead of simply throwing it to the atmosphere.

2.2.1 Carnot cycle heat pump

The principle of heat pump described by Holland *et al.* (1987) can be explained as; " A heat pump is a heat engine operating in reverse". The principle of a heat pump system can be illustrated as shown in Figure 2.3
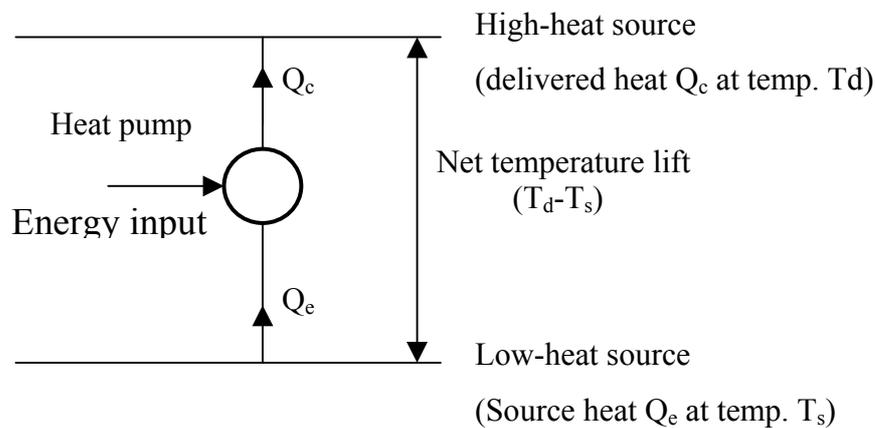

Figure 2.3: Schematic of heat pump.

(Source: Holland *et al.* 1987)

From the first law of thermodynamics, the amount of heat delivered, Q_c at the higher temperature Td is related to the amount of heat extracted Q_e at the low temperature T_s and the amount of high grade energy input Pc by the equation:

$$Q_c = Q_e + P_c \qquad (2.9)$$

A coefficient of performance of heat pump (COP_h) can be defined as

$$COP_h = Q_c / P_c \qquad (2.10)$$

A heat engine operating between a higher temperature Td and a lower temperature T_s have a theoretical maximum thermodynamic efficiency (η). It is written as:

$$\eta = (Td-Ts)/Td \tag{2.11}$$

This equation is known as the Carnot efficiency. The Carnot coefficient of performance defined by Equation 2.12 is:

$$COP_h = Td/(Td-Ts) \tag{2.12}$$

All heat pumps must cool and heat at the same time. A refrigerator is a heat pump that is designed to cool at the lower temperature T_s rather than to heat at the higher temperature T_d. The coefficient of performance of refrigeration is defined as:

$$COP_r = Q_e / P_c \tag{2.13}$$

Equation 2.10-13 can be combined to show that the coefficient of performance of heat pump is related to the coefficient of performance of a refrigerator by the equation.

$$COP_h = COP_r + 1 \tag{2.14}$$

A conventional mechanical vapor compression of heat pump is illustrated in Figure 2.4. Which consists of two heat exchangers, a compressor, an expansion valve and a working fluid. In the evaporator heat exchanger, the working fluid evaporates at a temperature Te by extracting heat from the source. It is then compressed and gives up its latent heat as it condenses at a higher temperature Tc in the condenser. The condensate then expanded through an expansion valve and returns to the evaporator to complete the cycle. The difference between the condensing and evaporating temperature (Tc - Te) is the gross or maximum possible temperature lift. The net temperature lift (Td - Ts) is less than the gross temperature lift by the sum of temperature difference driving forces in the evaporator and condenser. The ratio of the corresponding pressure in the condenser and evaporator P_c / P_e is the compression

ratio (CR). (Tc - Te), CR and the condensing temperature (Tc) are the critical parameters which determine the feasible operation of a heat pump operating on a particular working fluid.
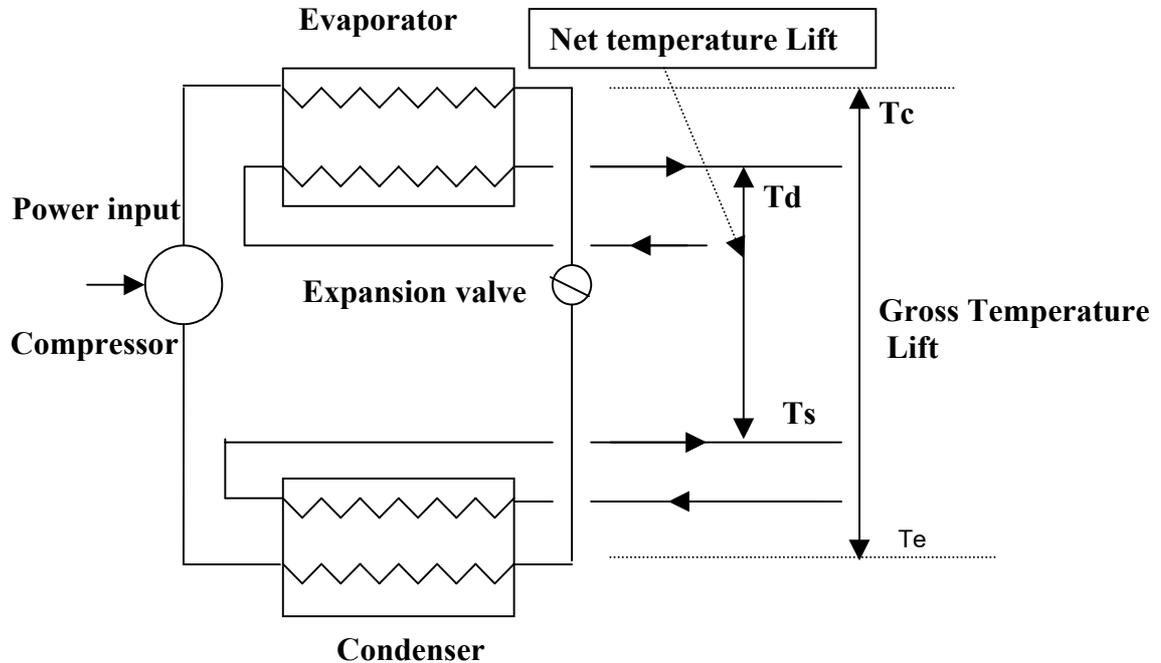

Figure 2.4: Vapor compression heat pump.

(Source: Holland *et al*. 1987)

2.2.2 Ideal Rankine cycle heat pumps

In practice the operation of a vapor compression heat pump approximates more closely to the Rankine heat pump cycle than the theoretical Carnot cycle. The ideal Rankine heat pump cycle is presented in Figure 2.5.

In Figure 2.5 the working fluid at point 1 is in the form of saturated vapor. It is isentropically compressed to point 2 in the superheated vapor region. The superheat ($h_2 - h_2'$) is then removed and it is isothermally condensed from saturated vapor at point $2'$ to saturated liquid at point 3. From point 3, it is isenthalpically expanded to a mixture of liquid and vapor to point 4 from which it is isothermally evaporated to point 1 from which it is isentropically evaporated to point 2. With reference to Figure

2.5, the theoretical Rankine coefficient of performance $(COP)_R$ of a heat pump can be defined as Thermodynamic Design Data for Heat Pump Systems.
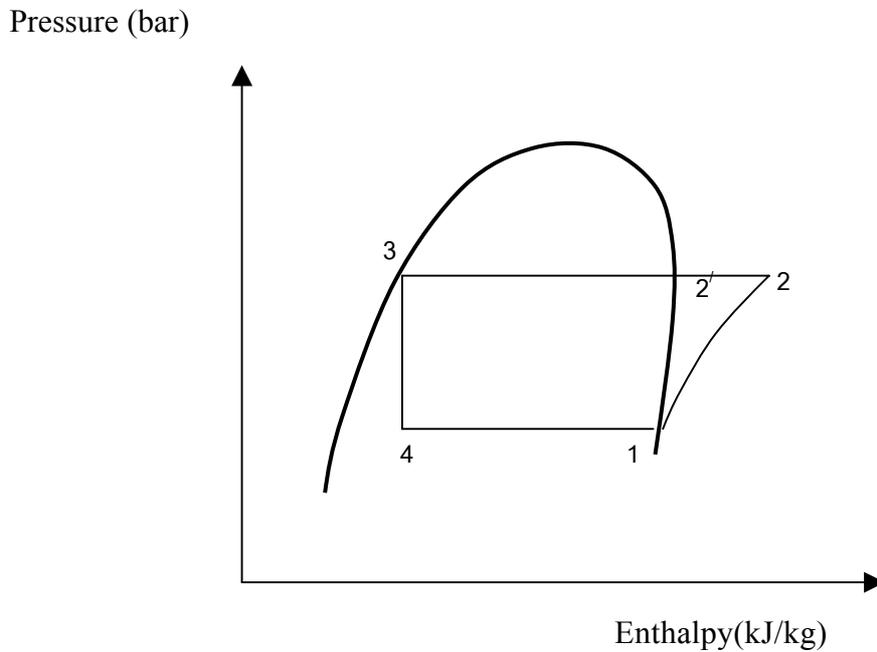
Pressure (bar)

Enthalpy(kJ/kg)

Figure 2.5: Pressure against enthalpy per unit mass

The theoretical Rankine coefficient of performance $(COP)_R$ can be written as:

$$(COP)_R = (H_2 - H_3)/ (H_2 - H_1) \qquad (2.15)$$

Where, H is the enthalpy per unit mass. The compression from point 1 to point 2 is at constant entropy.

The enthalpy per unit mass of superheated vapor at point 2 is related to the enthalpy per unit mass at the saturated vapor point $2^/$ by equation

$$H_2 = H_2^/ + C_p(T_2 - T_2^/) \qquad (2.16)$$

Where, C_p is the heat capacity per unit mass at constant pressure. Equation (2.16) can be approximated with accuracy sufficient for design purposes by

$$H_2 = H_2^/ + (\phi_1 - \phi_{2^/})\, Tc \qquad (2.17)$$

Where, ϕ_1 = entropy at point one

ϕ_2 = entropy at point two

2.2.3 Components of a vapor compression system

A simple vapor compression heat pump system consists of four main devices: compressor, condenser, evaporator and expansion valve. The details are described in the following paragraphs.

2.2.3.1 Compressor

The compressor is the heart of the vapor compression system. There are four common types of refrigeration or heat pump compressor, namely, the reciprocating, screw, centrifugal and vane. Reciprocating compressor are normally used in the refrigeration system. Compressors are single-acting and may be single-cylinder or multi-cylinder. During the suction stroke of the piston, low pressure refrigerant gas is drawn in through the suction valve, which may be located in the piston or in the head. The discharge occurs at the discharge valve, which is usually located in the cylinder head (Stoecker and Jones, 1983).

2.2.3.2 Condenser

The condenser is a heat exchanger. One classification of condenser is according to whether the refrigerant is on the inside of the tubes and whether the fluid cooling the condenser is a gas or liquid. The most widely used type of condenser is finned coil exchanger. The rate of heat transfer in the condenser is a function of the refrigerating capacity and temperature of evaporation and condensation. The condenser rejects the energy, which is absorbed by the evaporator and the heat of compression added by the compressor.

2.2.3.3 Evaporator

Any device in which a refrigerant is boiled for the purpose of removing heat from the surrounding medium is called an evaporator. Evaporators are constructed in a wide variety of types, shapes, sizes and designs. The most usually used is a finned type evaporator. The fins serve as heat absorbing surface and increase

the overall heat transfer area of the evaporator. The capacity of the evaporator depends on the quantity of air being passed through the evaporator, the air condition and the temperature of the refrigerant. The evaporator has greater capacity if the temperature difference between the refrigerant and the air is increased. Since the evaporator increases capacity, it will be able to evaporate a greater quantity of refrigerant. Any increases in the rate of evaporation will raise the suction pressure or temperature. As the suction pressure or temperature is increased, the capacity of the compressor will change due to the density change of the refrigerant.

2.2.3.4 Expansion valve

The purpose of the expansion valve is to reduce the pressure of the liquid refrigerant and to regulate the flow of the refrigerant to the evaporator. The four common types of expansion valves are the capillary tubes, the superheat controlled expansion valve, the float valve and the constant pressure expansion valve. The capillary tube and the superheat-controlled expansion valve are commonly used in the refrigeration system.

2.2.3.5 Effect of changing conditions at the evaporator

The power required by the ideal compressor is the product of the mass rate of flow and the increase in enthalpy during the isentropic compression.

$$P = w. \Delta h_i \qquad\qquad (2.18)$$

Where, P = power, kW

w = mass rate of flow, kg/s

Δh_i = isentropic work of compression, kJ/kg

Figure 2.6 shows the variation in Δh_i as the evaporating temperature changes. The value of Δh_i is large at low evaporating temperature and drops to zero when the suction pressure equals the discharge pressure (when the evaporating temperature equals the condensing pressure). The curve of the power requirement in Fig. 2.6 therefore shows a zero value at two points, where the evaporating temperature equals the condensing temperature and where the mass rate of flow is zero. Between the two extremes the power requirement reaches a peak.
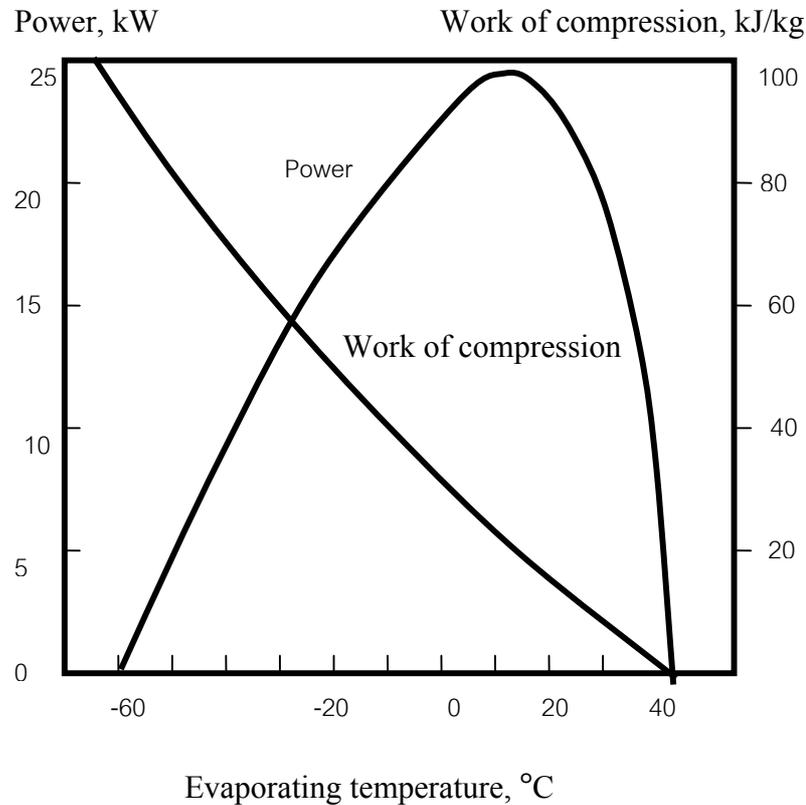
Power, kW   Work of compression, kJ/kg

Evaporating temperature, °C

Figure 2.6 Effect of evaporating temperature on work of compression and power
required by an ideal compressor (Stoecker and Jones, 1983)

The power curve merits close attention because it has important implications. Most refrigeration systems operate on the left side of the peak of the power curve.

During the period of pull down of temperature of following start-up with a warm evaporator, however, the power requirement passes through its peak and may demand more power than the motor; which is selected for design condition. Sometimes motors have to be oversized just to take the system down through the peak in the power curve. To avoid over sizing the motor, the suction pressure is sometimes reduced artificially by throttling the suction gas until the evaporator pressure drops below the peak in the power curve.

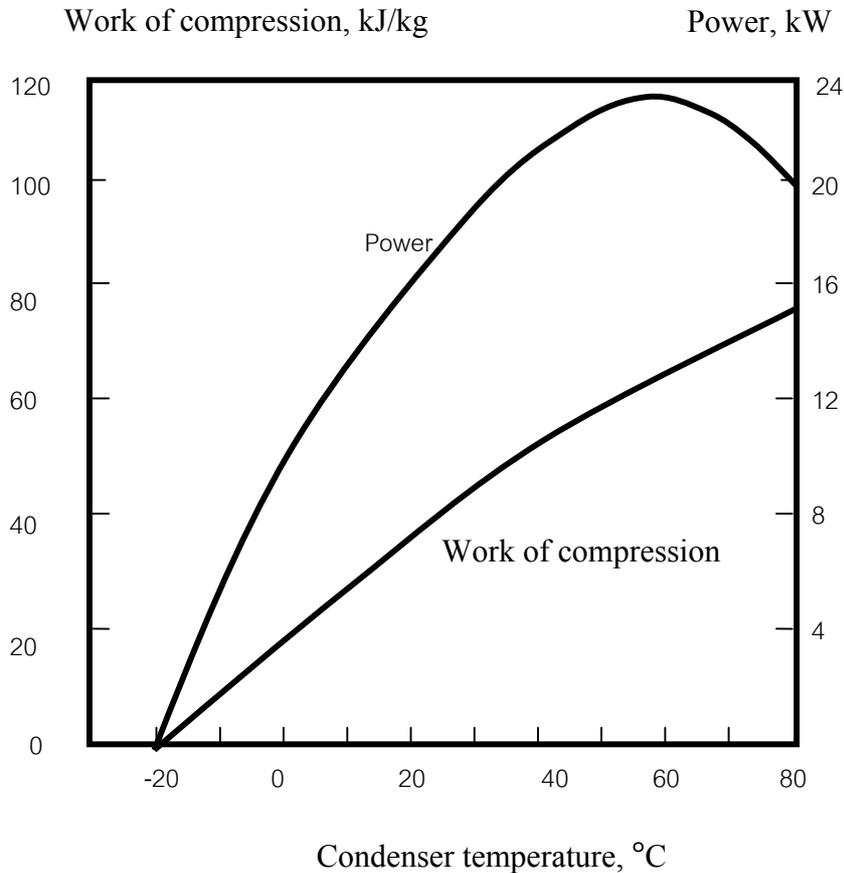2.2.3.6 Effect of changing conditions at the condenser

Work of compression, kJ/kg   Power, kW

Condenser temperature, °C

Figure 2.7 Effect of condenser temperature on work of compression and power
required by an ideal compressor (Stoecker and Jones, 1983)

Most refrigerating systems reject heat to the atmosphere, and ambient conditions
change throughout the year. Process refrigeration plants that operate year round are
particularly subject to a wide range of condensing temperature. The response of
reciprocating compressor to changes in condensing temperature can be analyzed
similar to the evaporating temperature. Figure 2.7 shows the characteristic of the
power. The compressor power is the product of the work of compression in kilojoules
per kilogram and the mass rate of flow. The work of compression in kilojoules per
kilogram increases with increased condenser temperature. The mass rate of flow
decreases as the condensing temperature increases, so that the power increases to a

peak and then drops off, a trend similar to the power as a function of the evaporating temperature shown in Figure 2.6.

2.3 Theory and working principle of thermosyphon

The thermosyphon employs evaporating and condensing process of working fluid. Generally, it is a closed end pipe, which comprises of an air-evacuated hollow tube and some amount of working fluid sealed within the tube. The thermosyphon components, classified by their functions, are; evaporator section, adiabatic section and condenser section. There is no wick employed and the condensate returns to the evaporator section by gravitational force. This requires that the evaporator section must always be situated at a point lower than the condenser. The application of thermosyphon has been limited in several ways, for instance, the thermosyphon cannot operate if the evaporator section is higher than the condenser section since it is against the gravity. In order to be successful in return of the condensate to the evaporator section, many mechanisms have been introduced, e.g. using the capillary structure, osmotic membrane, or employing the centrifugal force etc.

The theory of the thermosyphon, which is a simple heat transfer device simplified to evaluate its performance, can also be applied with the heat pump dryer. The main principle of the method to calculate its performance is to determine total resistance. (Engineering Science Data Unit, 1981). The total resistance of the thermosyphon can be divided into:

i) External resistance (Z_1 and Z_9)
ii) Resistance from material property (Z_2 and Z_8)
iii) Internal resistance (Z_3, Z_4, Z_5, Z_6, Z_7)

The external resistance is resistance from the outside wall of the thermasyphon at both the evaporator and condenser section. The internal resistance is resistance from the phase change, pool boiling or film boiling, and vapor pressure drop along the pipe. The resistance from the material property is the resistance that depends on the type of material used. Each resistance can be explained as:
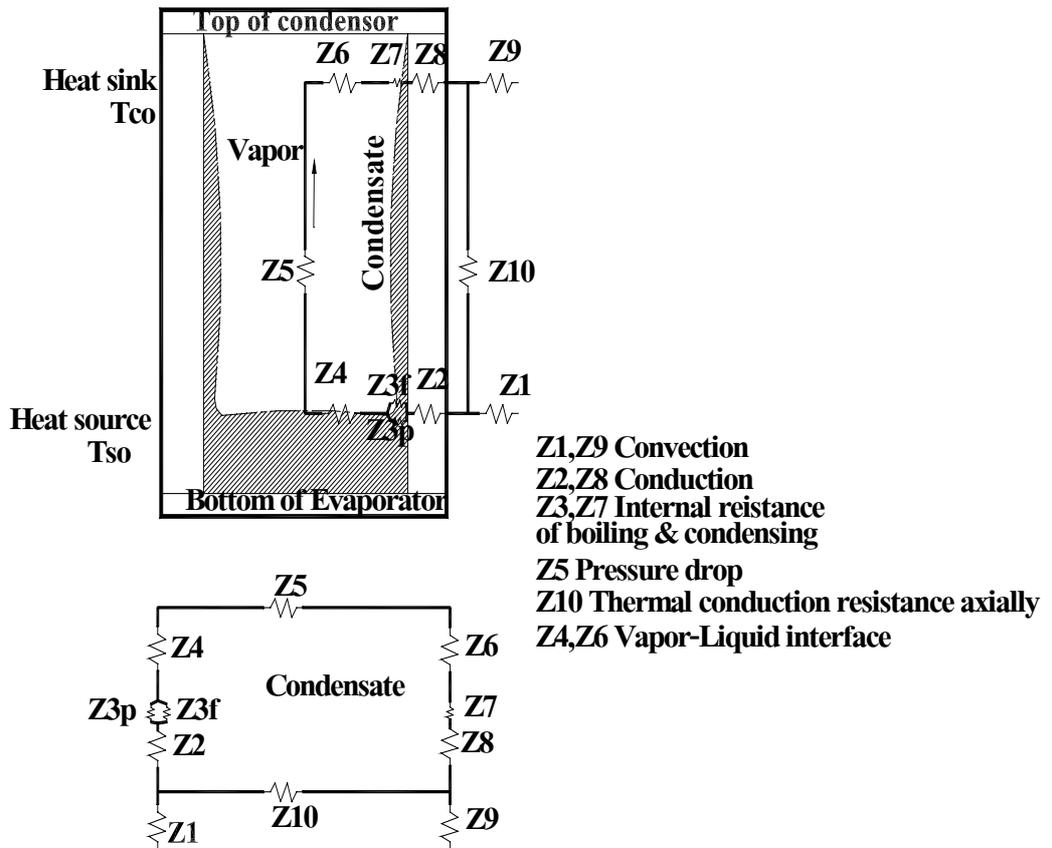
Figure 2.8: Model of total resistance of thermosyphon

2.3.1 External resistance

Z_1 and Z_9 are resistance from external convection of the pipe :

$$Z_1 = \frac{1}{h_{eo} A_{eo}} \tag{2.19}$$

$$Z_9 = \frac{1}{h_{co} A_{co}} \tag{2.20}$$

where, h_{eo} is heat transfer coefficient at the evaporator section (W/m^{2o}C)

     h_{co} is heat transfer coefficient at the condenser section (W/m^{2o}C)

     A_{eo} is outside wall area of the evaporator section (m^2)

     A_{co} is outside wall area of the condenser section (m^2)

2.3.2 Resistance from material property

Z_2 and Z_8 are resistance from the thermal conductivity of the material:

$$Z_2 = \frac{\ln(D_o / D_i)}{2\pi L_e k_x} \tag{2.21}$$

$$Z_8 = \frac{\ln(D_o / D_i)}{2\pi L_c k_x} \tag{2.22}$$

where, D_o is outside diameter of the pipe (m)

     D_I is inside diameter of the pipe (m)

     L_e is length of the evaporator section (m)

     L_c is length of the condenser section (m)

     k_x is thermal conductivity of the material (W/m ^{o}C)

2.3.3 Internal resistance

Z_3 and Z_7 are internal resistance due to pool and film boiling of the working fluid divide into Z_{3p}, which is resistance from pool boiling:

$$Z_{3p} = \frac{1}{\Phi_3 g^{0.2} Q^{0.4} (\pi D_i Le)^{0.6}} \qquad (2.23)$$

Z_{3f} is resistance from film boiling at the evaporator section:

$$Z_{3f} = \frac{CQ^{1/3}}{D_i^{4/3} g^{1/3} Le \Phi_2^{4/3}} \qquad (2.24)$$

where, g   is gravity (m/s^2)

     C   is constant of cylinder tube C = (1/4)(3/π)$^{4/3}$ = 0.235

     Φ_2 is Figure of Merit (2) :

$$\Phi_2 = \left(\frac{L k_l^3 \rho_l^2}{\mu_l} \right)^{1/4} \qquad (2.25)$$

     L   is latent heat of working fluid (kJ/kg)

     k_l is thermal conductivity of working fluid as liquid phase (W/m^{o}C)

     ρ_l is density of working fluid as liquid phase (kg/m^3)

     μ_l is viscosity of working fluid as liquid phase (N.s/m^2)

     Φ_3 Figure of Merit (3) :

$$\Phi_3 = 0.325 \times \frac{\rho_l^{0.5} k_l^{0.3} C_{pl}^{0.7}}{\rho_v^{0.25} L^{0.4} \mu_l^{0.1}} \left[\frac{P_v}{P_a} \right]^{0.23} \qquad (2.26)$$

     C_{pl} is specific heat of working fluid as liquid phase (kJ/kg^{o}C)

     ρ_v is density of working fluid as vapor phase (kg/m^3)

     P_v is vapor pressure of working fluid (Pa)

     P_a is atmospheric pressure 101.3 kPa

And the condition for using Z_{3p} and Z_{3f} as Z_3 is

if $Z_{3p} > Z_{3f}$ so

$$Z_3 = Z_{3p} \qquad (2.27a)$$

if $Z_{3p} < Z_{3f}$ so

$$Z_3 = Z_{3p}F + Z_{3f}(1-F) \tag{2.27b}$$

Where, F   is filling ratio which is defined by

$$F = \frac{V_l}{AL_e} \tag{2.28}$$

V_l is volume of working fluid (m^3)

A   is area cross section of the pipe (m^2)

Z_7 is the resistance from film boiling of working fluid at the condenser section:

$$Z_7 = \frac{CQ^{1/3}}{D_i^{4/3}g^{1/3}Lc\Phi_2^{4/3}} \tag{2.29}$$

Z_4 and Z_6 are resistance due to phase change at the evaporator and condenser section respectively.

Z_5 is resistance due to pressure drop along the pipe.

Z_{10} is resistance due to heat conduction along the axial pipe.

Normally Z_4, Z_5, Z_6 and Z_{10} are small values and can be neglected.

After that the heat transfer by the thermosyphon can be evaluated by:

$$Q = \frac{\Delta T}{Z} \tag{2.30}$$

Where, ΔT is temperature difference between heat source and sink.

2.4 Working fluid property

Table 2.2 Working fluid property for copper tube (Waikual, 1996)

Working fluid	Freezing point (℃)	Boiling point (℃)	Pressure(kPa)		Latent heat (kJ/kg)	
			(10 ℃)	(60 ℃)	(10 ℃)	(60 ℃)
R-11	-111	24	60.68	313.29	186.28	166.93
R-12	-158	-30	422.76	1522.7	147.64	114.49
R-22	-160	-41	681.19	2427.4	196.44	139.58
R-113	-35	48	23.92	150.45	155.25	139.64
R-123*	-107	28	50.57	286.34	176.79	156.19
R-134a	-97	-26	414.49	1682.5	190.87	139.37
R-502	-	-46	773.05	2601.4	139.14	87.65
H₂O	0	100	1.23	19.92	2477.20	2357.9