Fundamentals of Refrigeration

Part 1

This series of articles by M. M. Roy covers the fundamental principles of refrigeration and air conditioning, applications, design, equipment, components and systems. The articles will serve as a source of reference for new comers joining our industry as well as for experienced personnel wanting to brush up on basics.

It is a natural phenomenon that 'heat' always flows from a body or space at a higher temperature to another body or space at a lower temperature.

Heat will not by itself, flow from a low-temperature level to a higher-temperature level unless aided by an external agency.

'Refrigeration' is the process by which it is possible to remove heat from a region of lower-temperature, transfer this heat to a higher-temperature level by input of mechanical work and then reject this heat to a heat-sink (such as ambient air or cooling water). The quantum of heat removed from the lower-temperature region is called "refrigeration effect"

The concept of refrigeration is better understood by an analogy with a simple hydraulic system – see box below.

Refrigerants

Refrigerants or heat-carriers, in refrigeration systems are volatile liquids which change from liquid to vapour phase or vice-versa at different temperatures depending on the pressures these are subjected to.

For example, water boils at 212°F at normal atmospheric pressure. But when the water is subjected to higher pressures like in pressure-cookers, its boiling temperature becomes higher than 212°F; whereas if water is subjected to pressure lower than atmospheric, its boiling temperature falls below 212°F.

Different refrigerants have different pressure-boiling point characteristics. Refrigerants such as Ammonia (NH_3), halogenated hydrocarbons such as R-22 ($CHClF_2$) or even water-vapour are chosen with most desirable properties, keeping requirements of specific applications or systems in mind.

Terms used in refrigeration parlance

There are some terms relating to properties and processes occurring in the condition of a refrigerant during its operating cycle, which need to be explained.

Saturation Temperature: By referring to standard tables giving properties of commonly used refrigerants, it is seen that for each pressure there is a corresponding temperature called 'saturation temperature', at which a particular refrigerant will boil or condense back to liquid form, depending on whether heat is added or removed.

Hydraulic System

Transfers water, say from a ground-level tank, filled with water to an empty roof-level tank in a building, by using a bucket as 'water-carrier', in the following steps :

Step 1: Lower the water-carrier i.e. bucket below ground-level tank and allow water from this tank to drain into and fill the bucket by gravity flow.

Step 2: Lift the bucket full of water from ground-level to a level higher than roof-level tank by doing mechanical work.

Step 3: Empty the bucket by allowing its water to drain by natural gravity flow into the roof-level tank. Empty bucket is now ready to resume water-lifting operation from Step 1 onwards.

Refrigeration System

Transfers heat from a lower-temperature level body to higher-temperature level by using a heat-carrier substance known as 'refrigerant' in the following steps:

Step 1 : Lower the heat-carrier's i.e. refrigerant's temperature below that of low-temperature level body, to allow natural heat-flow from the lower-temperature body to the refrigerant.

Step 2 : Raise the temperature level of the heat-carrier refrigerant now loaded with heat, to a temperature higher than that of heat-sink, by raising its pressure by doing mechanical work of compression.

Step 3 : Reject the heat from the refrigerant, already raised to higher temperaure level in Step 2, by natural heat-flow to the heat-sink (ambient air or cooling water)

After draining its heat-content to the heat-sink, the refrigerant is now ready to resume heat-transfer operations, from Step 1 onwards.

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'Saturation temperature' for any given pressure is also defined as that temperature, at which liquid refrigerant and its vapour co-exist in a closed container, in equilibrium with each other.

Superheating: If vaporised refrigerant is separated from the liquid portion, then any heat added to the already vaporised refrigerant will raise the temperature of this vapour above its saturation temperature corresponding to its pressure.

Thus when temperature of the vapour is above 'saturation temperature' corresponding to its pressure, the vapour is said to be superheated.

Sub-cooling: Is the term used to describe the cooling of the liquid refrigerant at constant pressure to a temperature below the saturation temperature corresponding to this condensing pressure.

Sensible Heat: When heat either absorbed or rejected by a substance causes a change in the temperature of the substance, this heat is called 'sensible heat', as temperature change may be detected by sense of touch or measured by a thermometer. Superheating is an example of sensible heating whereas sub-cooling is an example of sensible cooling.

Latent Heat: When heat, either absorbed or rejected by a substance brings about change in the physical state of the substance without any change in temperature, this heat is called 'latent heat'.

For example, when any liquid refrigerant changes to saturated vapour, 'latent heat of vaporisation' is absorbed for this change of state from liquid to vapour.

Similarly when saturated vapour changes to saturated liquid phase during condensation, the heat rejected for this phase-change is called 'latent heat of condensation'.

These changes in physical form of the refrigerant while absorbing or rejecting 'latent heat', take place at a saturation temperature corresponding to its pressure.

Enthalpy: Denotes heat content of the refrigerant measured from a 'base saturation temperature of –40°F.' At this temperature and corresponding pressure, heat content of the liquid refrigerant is arbitrarily fixed as zero. Thus 'enthalpy' of liquid refrigerant above 'minus 40°F' will be positive and below 'minus 40°F' negative.

Enthalpy of the liquid refrigerant is different for different pressures and corresponding saturation temperatures.

Enthalpy of refrigerant in saturated vapour form at a certain pressure = enthalpy of liquid refrigerant at the same pressure and temperature + latent heat of vaporisation

Adiabatic Compression: Any process performed without addition or removal of heat is an adiabatic process.

Thus, compression of gaseous refrigerant without addition or removal of heat is called 'adiabatic compression'.

In the theoretical vapour compression cycle, it is assumed that there is no transfer of heat between the refrigerant and the cylinder walls during compression i.e. the compression process is considered 'adiabatic'.

Throttling: Is the process of adiabatic expansion of

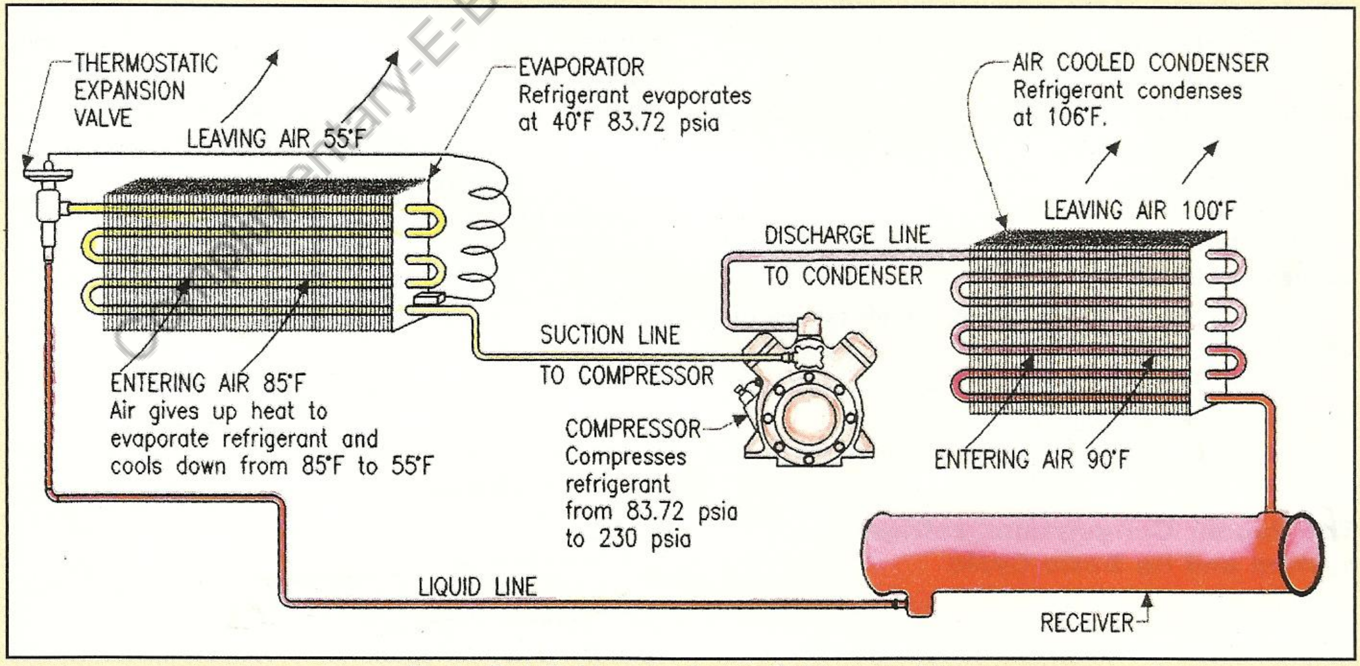

Fig. 1 : Basic Refrigeration System (Refrigerant – 22)

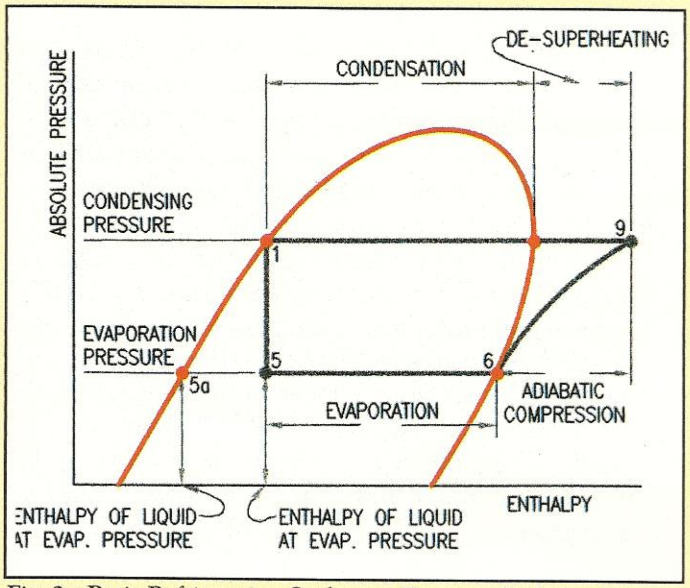

Fig. 2 : Basic Refrigeration Cycle on a Pressure–Enthalpy diagram

liquid refrigerant to a lower pressure at constant enthalpy (i.e. without addition or removal of heat) while passing through an orifice or expansion valve.

During the 'throttling process', a portion of the liquid flashes to vapour by drawing necessary 'latent heat of evaporation' from the remaining liquid, thereby reducing the temperature of the refrigerant fluid at the outlet of the orifice or expansion valve.

Atmospheric pressure: Is expressed in inches of mercury or pounds per square inch (psi) and measured by a barometer. Atmospheric air has weight and therefore experts a pressure on the surface of the earth.

Gauge Pressure: As measured by a pressure-gauge is the difference between the pressure of a fluid in a container and that of the atmosphere. When the pressure exerted by the fluid is more than atmospheric pressure, gauge pressure is positive. On the other hand if the fluid pressure inside the container is less than atmospheric pressure, the gauge pressure is negative indicating a vacuum condition.

Gauge pressure is expressed as pounds per sq. inch gauge (psig)

Absolute Pressure: Is the algebraic sum of the gauge pressure and the atmospheric pressure.

Absolute pressure is expressed as 'pounds per sq.inch absolute' (psia)

The Vapour Compression refrigeration system and its Basic Cycle

The majority of refrigeration systems for air-conditioning as well as process cooling applications use the 'vapour compression refrigeration cycle' in which the heat added to the refrigerant at the lower temperature (from the space or body being cooled) is added as 'heat of vaporisation' in the 'evaporator' and the heat rejected by refrigerant vapour after being compressed and raised to the higher temperature level is rejected as 'heat of condensation' in a 'condenser'

The 'Basic System' and the 'Cycle' are explained below for a R-22 refrigeration system often adopted for air conditioning applications.

Figure 1 shows the various components which integrate together to form the refrigeration system along with typical operating pressures and temperatures marked therein (data taken from Table of properties of R-22 Refrigerant and reproduced below)

Figure 2 shows the 'Basic Refrigeration Cycle' plotted on a Pressure-Enthalpy diagram, and indicates various changes in the condition of the refrigerant as it flows from one part of the cycle to another.

Saturation Temp (°F)	Absolute Pressure (psia)	Enthalpy in BTUs per pound of refrigerant		
		Sat. Liq.	Latent Heat	Sat. Vapor
(Evap.) 40°F	(Evap.) 83.72	21.70	87.39	109.09
(Cond.) 105°F	(Cond.) 230.70	42.98	70.22	113.20

Properties of R-22 Refrigerant

Refrigerant flows through the system components shown in Fig 1, alternating between the liquid and vapour phases as indicated in the Basic Cycle diagram Fig 2. Following is the sequence of operation:

- Saturated liquid leaves the condenser at condensing pressure and temperature and is stored in the receiver to maintain steady supply of refrigerant for the system.
- This high-pressure liquid refrigerant then enters the expansion valve (point 1 in Cycle diagram).

Throttling (Expansion) process (1-5)

Expansion of the liquid refrigerant from condenser pressure to evaporator pressure is an adiabatic (constant enthalpy) process

Point 5 is located in the liquid-vapour mixture area of Pressure-Enthalpy diagram since during this throttling process 'flash' vapour is produced by taking the required latent heat of evaporation from the liquid which is cooled down to a saturation temperature of 40°F at a reduced pressure of 83.72 psia.

c) Evaporation process (5-6)

Liquid refrigerant mixed with a small portion of flash gas, from outlet of the expansion valve at a reduced pressure of 83.70 psia, now enters the evaporator coil and fully evaporates at the corresponding saturated evaporation temperature of 40° F by taking the required latent heat of vaporisation from the air (flowing across the evaporator coil) being chilled in the process from 85°F to 55°F

d) Compression Process (6-9)

Saturated refrigerant vapour leaves the evaporator and enters the compressor suction port at point 6 of the cycle and its pressure and temperature is increased by compression until the superheated discharge gas at a pressure of 230 psia (i.e. same as condensing pressure) leaves the discharge port at point 9 of the cycle. Work done on the refrigerant gas during compression is known as 'heat of compression' and is absorbed by the gas.

e) Desuperheating & Condensation Process (9-1)

This process takes place in the condenser. Initially the hot discharge gas is desuperheated by cooling down to the 'saturated condensing temperature' corresponding to the condensing pressure of 230 psia and thereafter further removal of heat for condensing refrigerant fully to reach point 1 on the 100% saturated liquid line of Pressure-Enthalpy Diagram is effected by rejecting the 'heat of condensation' to the ambient air flowing over the condenser tubes.

Total heat rejected by the refrigeration system to the condenser cooling medium = heat absorbed by the refrigerant in the evaporator (refrigeration effect) + heat-equivalent of work input during compression

Co-Efficient of Performance (COP)

The term COP has been devised as an Index of Performance for refrigeration systems and is defined as the ratio of 'refrigeration effect divided by heat-equivalent of work input in compression'.

Higher the evaporator pressure and corresponding evaporation temperature and lower the condensing pressure and corresponding condensing temperature, less will be the work input required in compression, for the desired refrigeration effect (i.e. capacity output) with consequent increase in COP and better cycle efficiency.

A good system designer will always aim for a higher system COP through optimum equipment selection with adequate heat-transfer areas in the evaporator and the condenser in order to operate the system at a higher evaporation temperature and lower condensing temperature.

Next Issue : Part 2 - The practical vapour compression cycle and deviations from the theoretical cycle. Liquid sub-cooling and suction gas superheat are explained. ❖

Fundamentals of Refrigeration

Part 2

This series of articles by M. M. Roy covers the fundamental principles of refrigeration and air conditioning, applications, design, equipment, components and systems. The articles will serve as a source of reference for new comers joining our industry as well as for experienced personnel wanting to brush up on basics.

An *actual* vapour compression refrigeration system differs in many respects from the theoretical cycle explained in Part 1 (Oct-Dec 2000 issue). Heat transfer takes place between the refrigerant and the surrounding air in all components of the system as shown in Figure 1; whereas the theoretical cycle considers only the "refrigeration effect" Q_E (heat gained by the refrigerant in the evaporator from the conditioned space) and "heat rejection" Q_C in the condenser to the ambient air, the *actual* cycle in practise must also account for the following additional heat gains and losses:

- Q_{SL} or heat gained by the already evaporated refrigerant in the suction line, from the ambient air
- Q_{COMP} or the net heat transfer taking place from the refrigerant during the compression process to the surrounding air.

Hence the actual compression process deviates from "adiabatic" compression assumed in the case of the theoretical cycle because at the start of the compression process, heat is transferred to the cold suction gas from the warmer cylinder walls of the compressor. Towards the later part f the compression process, heat is transferred from the hot compressed refrigerant vapour to the cylinder walls and then to the surrounding air. Overall there is a net heat transfer from the compressed refrigerant to the surroundings.

- Q_{DL} is the heat loss from the discharge gas in the discharge pipeline to the surroundings.
- Q_{LL} is the heat loss from the hot condensed refrigerant in the liquid line to the surroundings

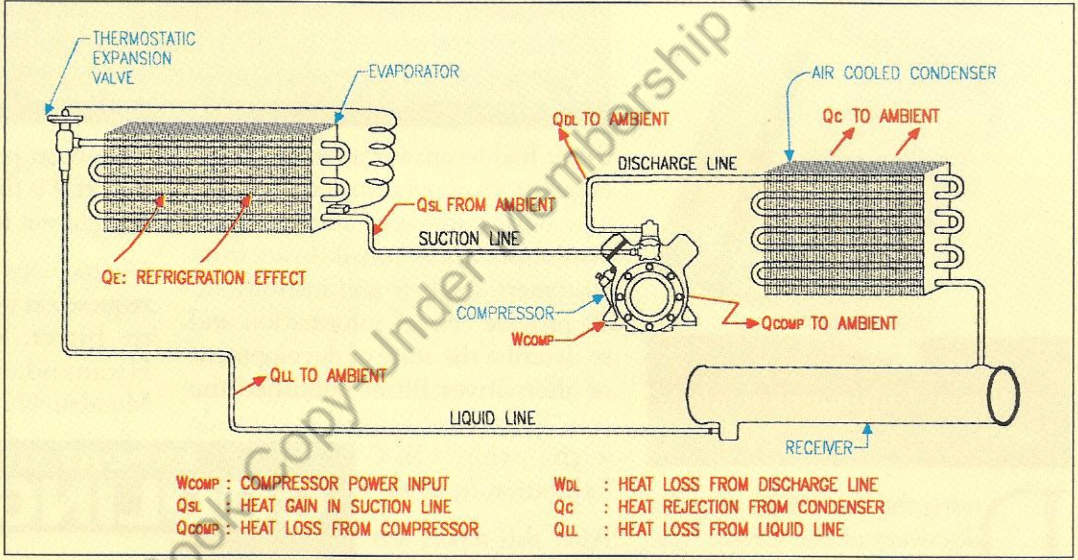

Figure 1 : Schematic of Practical Vapour Compression system

Compressor Volumetric Efficiency

An ideal compressor should have no clearance between the piston and the cylinder walls or the top of the cylinder where the valve plate sits. In reality this is not practical since clearances must exist for lubrication purposes and a gasket has to be used for sealing the gap between the cylinder head and the valve plate. Discharge gas expands into these clearances during the suction stroke thus reducing the swept volume of the cylinder. Volumetric efficiency is the ratio of the actual volume of refrigerant pumped on each stroke to the theoretical piston displacement and is always less than 1.

Power Requirement

The above factors affect the capacity of the compressor as well as the horsepower requirement per ton of refrigeration. Theoretically the heat equivalent of work done in the adiabatic compression process is computed from the Ideal cycle drawn on the P-H diagram, as H9-H6 i.e. "enthalpy of compressed and superheated vapour minus enthalpy of saturated vapour at evaporator

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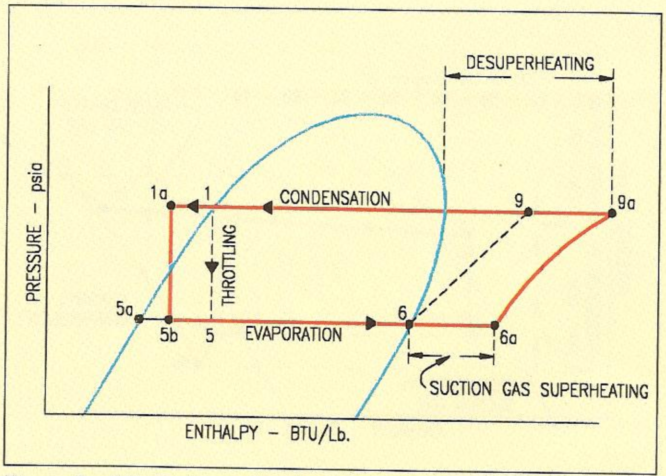

Figure 2 : Effect of superheating suction gas and liquid subcooling

pressure." Please refer to the Basic Refrigeration Cycle diagram, Part 1 for further clarification.

In the *actual* compression process the power required to be applied at the shaft of a compressor is greater than that computed theoretically because of the frictional losses in the compressor and also because of the heat exchange taking place between the vapour and the cylinder walls during compression.

Suction Gas Superheat

Unlike an Ideal cycle wherein the suction vapour entering the compressor is assumed to be exactly saturated, in the *actual* cycle, a thermostatic expansion valve controls the amount of refrigerant flow so as to obtain 5°F to 10°F superheat in the evaporator itself. This also increases the enthalpy of the vapour leaving the evaporator resulting in a slight increase of refrigeration effect per unit weight of refrigerant. A slight increase in the superheat of the evaporated refrigerant takes place also in the suction line (outside the evaporator) due to the heat gain by cold suction vapour from the ambient air at higher temperature.

The condition of the superheated suction gas at the entry to the compressor in the *actual* cycle is shown as point 6a in the P-H diagram in Figure 2.

Inspite of the increase in enthalpy of suction gas due to superheat, the capacity increase is marginal as superheat increases the specific volume of the vapour with consequent reduction in the weight of refrigerant pumped.

Suction gas superheat is adopted in practical systems since:

(i) superheating ensures complete evaporation of all the liquid refrigerant before entering the compressor, thus preventing harmful effects of liquid slugging which can damage the compressor.

(ii) variation in the suction superheat temperature can help in modulating the size of opening of the expansion valve depending on the system load.

Liquid Subcooling

In an *actual* cycle it is normal practise to subcool the condensed liquid refrigerant by 5° to 15°F below the saturation temperature, corresponding to the condensing pressure, before the liquid enters the expansion valve for the throttling operation. It can be seen from Figure 2 that "throttling operation" in the theoretical cycle follows the dotted line path 1-5 whereas throttling of sub-cooled liquid in the *actual* cycle follows the path 1a to 5b. The proportion of flash gas in the liquid vapour mixture at point 5b in the *actual* cycle is less compared to the higher proportion in the liquid-vapour mixture at point 5 after throttling of saturated liquid in a theoretical cycle.

Since flashgas does not produce any refrigeration and it is only the portion of refrigerant in liquid form which evaporates to produce useful "refrigeration effect", a proportionately higher content of liquid refrigerant at entry point 5b to the evaporator, after throttling of "sub-cooled liquid", produces more "refrigeration effect", during evaporation and superheating in the evaporator in the case of the *actual* cycle.

It is also evident from the P-H diagram in Figure 2 that the enthalpy of liquid-vapour mixture at point 5b after throttling of sub-cooled in the *actual* cycle, is less than the enthalpy of liquid vapour mixture at point 5 after throttling of saturated liquid in the theoretical cycle.

Thus the net heat gained by the refrigerant during evaporation and superheating in the evaporator till reaching point 6a, Figure 2, i.e. $H_{6a} - H_{5b}$ being the "useful refrigeration effect" in an *actual* cycle resorting to sub-cooling of liquid in the condenser and slight superheating of vaporised refrigerant in the evaporator is greater than the "refrigeration effect", H_6-H_5 produced in the theoretical cycle. Calculations will show that every 2°F sub-cooling of liquid refrigerant in the condenser results in approximately 1% increase in the system capacity.

Subcooling arrangement in a Condenser

A certain number of tubes in the lower portion of the tube bundle in a shell-and-tube water-cooled condenser are designed to be submerged in the condensed liquid so as to provide a specific amount of sub-cooling. In a multi-pass condenser, effective sub-cooling is achieved by including the sub-cooling tubes in the first pass so that the coolest cooling water comes in contact with the hot condensate liquid to impart good sub-cooling.

Medium and large capacity air-cooled condensers are provided with a sub-cooling circuit in the condensate portion of the condenser coil on the air-entry side, so as to achieve optimum sub-cooling.

Liquid Suction Heat Exchangers

Superheating of suction gas and sub-cooling of liquid refrigerant before throttling, can be carried out simultaneously by using a liquid-suction heat exchanger. Heat is transferred from the liquid (after leaving the condenser and receiver) to the vapour (after leaving the evaporator) in the liquid-suction heat exchanger, thereby sub-cooling the "liquid refrigerant" and superheating the suction vapour.

In this case since superheating of vapour takes place outside the evaporator, it does not increase the "refrigeration effect", but the sub-cooling of liquid before throttling in the expansion valve, reduces the "entering enthalpy" of refrigerant at the entry to the evaporator and since the enthalpy of vaporised refrigerant leaving the evaporator (determined by the evaporator pressure) remains unchanged, the net effect is an increase in "refrigeration effect" per unit weight of refrigerant, equal to the amount transferred by the sub-cooled liquid in the heat-exchanger.

A liquid-suction heat exchanger leads to the following additional benefits:

- Reduction of suction line sweating on account of rise in temperature of suction gas by superheating
- Reduction of liquid slugging into the compressor as superheating of suction gas ensures that refrigerant is fully vaporised before entering the compressor
- Reduction of flash gas during the throttling process on account of sub-cooling of the liquid prior to throttling ensures the availability of a higher quantity of liquid refrigerant for evaporation thereby increasing the useful "refrigeration effect"
- Since the saturation pressure corresponding to the temperature of the sub-cooled liquid is less than the saturation pressure of the liquid at the condensing temperature, the system designer can utilise this pressure difference in over coming the pressure drop in long liquid lines or liquid line risers, where the evaporator is located at a higher level than the condenser, without flashing of refrigerant and reduction of cooling capacity.

Pressure Drops in the Practical Cycle

Other variations take place in the practical vapour compression cycle as shown in Figure 3, due to the following system pressure drops which are inevitable in the "practical cycle".

- Pressure drop of refrigerant in the evaporator coil, from the entering pressure condition of liquid-vapour

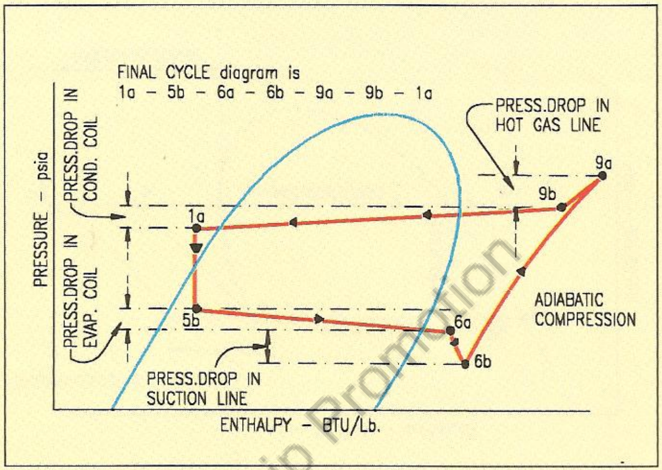

Figure 3 : Effect of pressure drop in coils and piping

flash gas mixture at the entry point to the evaporator till it reaches the state of fully evaporated refrigerant at the "exit" from the evaporator.

Pressure drop in the evaporator coil in the *actual* cycle is evident in Figure 3 as the evaporation line 5b-6a is slanting downwards instead of being horizontal.

- Pressure drop of refrigerant vapour in the suction line is from 6a, at the exit from the evaporator, to 6b i.e. point of entry to the suction port of the compressor.
- Pressure drop in the hot gas line is from point 9a at the discharge from the compressor upto point 9b i.e. the entry point of hot gas to the condenser.
- The desuperheating-condensing-subcooling line from point 9b till it reaches point 1a is also shown sloping downwards on account of the pressure drop of refrigerant in the condenser.

The system designer therefore takes into consideration the pressure drops in the refrigerant suction, discharge and liquid lines as well as the estimated pressure drops in the evaporator and condenser in the practical system, while selecting equipment for the required performance at the expected *actual* suction pressure/temperature and discharge pressure/temperature conditions.

Also the refrigerant liquid, suction and discharge lines must be adequately sized for the required system capacity after considering the location and layout of the refrigeration equipment so as not to exceed the permissible pressure-drop criteria in the refrigerant pipelines.

Next Issue : Part 3 – Two Stage and Cascade Refrigeration Systems.

Fundamentals of Refrigeration

Part 3

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Two-Stage and Cascade Refrigeration Systems

In the single stage vapour compression refrigeration cycle explained in Part 1 (Oct-Dec 2000 issue) and Part 2 (Jan-Mar 2001 issue) it will be observed that the hot gas entering the condenser is in a superheated state. The degree of superheat depends on the "compression ratio" i.e. the ratio between the absolute discharge pressure and the absolute suction pressure.

For any given condensing temperature and pressure, which are determined mainly by the temperature of ambient air (in air-cooled systems) or temperature of entering cooling water (in water-cooled systems) as well as the heat-transfer area of the condenser, the "compression ratio" will depend on the suction pressure and temperature of the evaporating refrigerant, which in turn depends on the application of the Refrigeration System either for air conditioning (high evaporating temperature) or freezing application (low evaporating temperature)

In Refrigeration Systems applied to low temperature cold storage and freezing applications, the refrigerant temperature can be as low as –40°F or lower, resulting in a very low suction pressure at the entry port of the compressor. Consequently the range through which the refrigerant gas has to be compressed to bring it to the condensing pressure condition becomes very large. In other words the "compression ratio" becomes very large for such low temperature cold storage or freezing applications, even though the condensing temperature and pressure may be practically of the same order as in a normal air conditioning application.

In such high "compression ratio" systems the hot gas

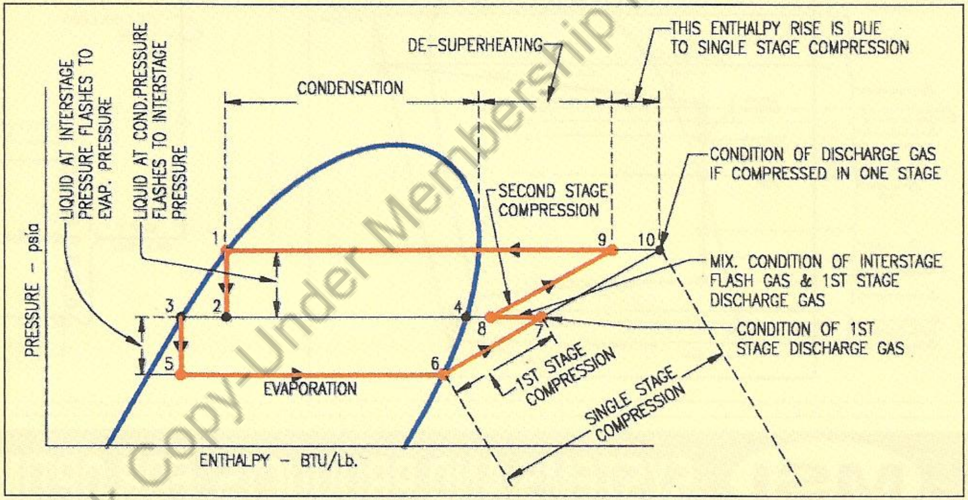

Figure 1 : Two-Stage refrigeration cycle

discharged by the compressor becomes highly superheated with very high discharge gas temperature. Since it is not desirable to operate the compressor with such a high discharge gas temperature, a "two-stage compression system" is adopted for such applications.

The first stage of compression compresses the suction gas to an intermediate pressure. The somewhat superheated discharge gas from the first-stage is usually cooled in an "intercooler" with the help of flash gas generated in the second-stage while expanding the high pressure liquid refrigerant from the condensing pressure to an intermediate pressure.

The mixing of discharge gas at intermediate pressure after the first-stage of compression with the cool flash gas from the second-stage of expansion will result in partial removal of superheat from the first-stage discharge gas. Consequently and as evident from the cycle diagram in Figure 1, the discharge gas enthalpy at point 9 and corresponding discharge gas temperature after the second stage of compression in a two-stage system becomes less than the discharge gas enthalpy at point 10 and corresponding discharge gas temperature in a single-stage compression system, where the refrigerant is compressed

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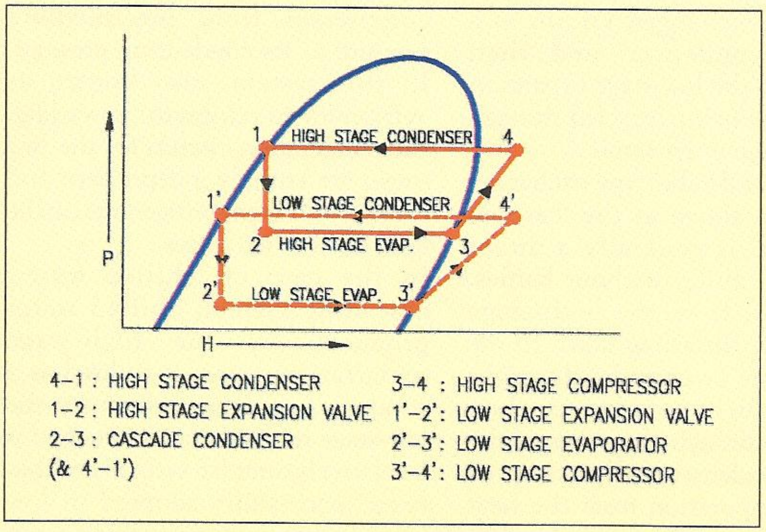

Figure 2 : P-H Diagram of Cascade Refrigeration Cycle

in one step from suction pressure to final discharge pressure.

There is an added benefit of a two-stage refrigeration system, in as much as a greater part (approximately 62%) of the flash gas, which is formed during the second-stage of expansion, needs to be compressed through a smaller range i.e. from interstage pressure to condensing pressure (instead of the entire quantity of flash gas requiring to be compressed through the total range of compression from "evaporator pressure" to the "condensing pressure" in the single-stage refrigeration system). Thus the work input during the compressing process and the power consumption per ton of refrigeration in a two-stage refrigeration system becomes almost 15% less than for a single-stage refrigeration system.

Due to the benefits of better cycle efficiency and power saving in a two-stage refrigeration system with interstage cooling, the two-stage system is also adopted by some manufacturers for making Centrifugal chillers used in air conditioning applications even though the moderate "compression ratios" in chilled water air conditioning applications can be handled by a single-stage system. In fact, manufacturers of single-stage Centrifugal machines tend to design their systems to operate at higher evaporating temperature and lower condensing temperature by somewhat oversizing the evaporator and condenser in order to offset the power disadvantage compared to two-stage Centrifugal systems.

The two-stage refrigeration cycle in Figure 1 (Pressure-Enthalpy diagram) is explained below by charting the sequence of flow of the refrigerant in different phases:

1–2: Refrigerant liquid leaving the condenser at condensing pressure, flashes to 'interstage pressure'

At 2 : Refrigerant is partly gaseous (flash gas) and partly liquid

6-7 : First-stage compression process of refrigerant vapour from 'evaporator' pressure to 'interstage' pressure

Flash gas at point 2 at inter-stage pressure mixes with the first-stage discharge gas (point 7) to reach point 8, with lower superheat, before entering the second-stage compressor

8–9: Refrigerant vapour is compressed from inter-stage pressure to condensing pressure in the second stage of compression

9-1: De-superheating of hot discharge gas and condensation to liquid phase in the condenser

At 3: Represents remaining portion of refrigerant in liquid at 'inter-stage' pressure, after removal of flash gas

3–5: Liquid at inter-stage pressure flashes to evaporator pressure

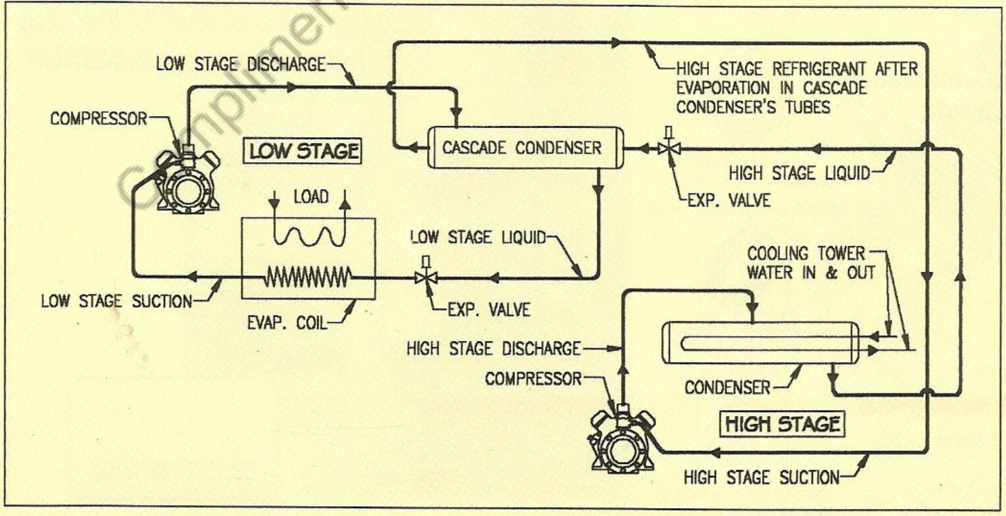

Figure 3 : Schematic Flow Diagram of Cascade Refrigeration System

5–6: Evaporation of liquid refrigerant in the evaporator, before being drawn into the first-stage compressor

6–7–10: If the entire compression process was carried out in one stretch, the compression line would have been 6–7–10, the point 10 representing the highly superheated condition of the discharge gas in a single-stage compression system.

Enthalpy difference between discharge vapour enthalpy at point 10 and the discharge vapour enthalpy at point 9 can be considered as the heat equivalent of power saved in a two-stage compression system compared to a single-stage system.

Cascade System

A cascade refrigeration system is but a variation of the two-stage compression system. The cascade system is often adopted for low temperature refrigeration and freezing applications.

There are two types of Cascade systems – one based on 'refrigerant-to-refrigerant' cascading and the other using 'chilled-water'. As is evident from Figure 2, P-H diagram and the schematic flow diagram Figure 3 of a Cascade system, refrigerant evaporating at a low pressure and temperature in the low-stage evaporator is first compressed to an intermediate pressure in the low-stage compressor, condensed in the shell side of a 'shell-and-tube' type heat exchanger, known as a 'cascade condenser' and then throttled in the low-stage expansion valve before being recycled through the low-stage evaporator.

The 'shell-and-tube' type exchanger referred to above as the 'cascade condenser' is generally a direct expansion chiller without baffles. Refrigerant from the high-stage system after throttling down to the intermediate pressure by the high-stage expansion valve is then circulated through the tubes of the cascade condenser and picks up its heat of evaporation from the heat rejected by the condensing refrigerant of the low-stage system (in the shell side) in the cascade condenser.

The high-state refrigerant evaporating inside the tubes of the cascade condenser, is then compressed by a separate high-stage compressor from intermediate pressure to its condensing pressure. In this system, also known as 'refrigerant-to-refrigerant' cascading, the refrigeration circuits for the two stages are entirely independent and even two different refrigerants can be used for the two stages.

In the case of 'chilled water' cascading system, chilled water produced by the high-stage refrigeration system is utilised as a condenser cooling medium for the low-stage refrigeration system and is a relatively simpler system and has been successfully adopted in low temperature systems requiring not lower than –40°F temperature.

Next Issue : Part 4 – Fundamentals of a Vapour Absorption refrigeration system, theory and operating cycle

Fundamentals of Refrigeration

Part 4

Vapour Absorption Refrigeration Systems – Theory and Operating Cycles

A vapour absorption refrigeration system is heat-energy driven unlike the conventional vapour compression system which uses a compressor.

Vapour absorption systems work with non-CFC refrigerants such as water or ammonia which evaporate at low-temperature in the "evaporator" maintained at low-pressure thus producing cooling at low temperature.

Absorption Process

Operation of the absorption cycle also requires a secondary fluid called "absorbent", (having great affinity for the refrigerant) which is used to absorb the gaseous refrigerant.

Figure 1 shows the basic flow diagram of the vapour absorption cycle. Instead of the low-pressure and temperature refrigerant being sucked into a compressor, (as happens in the vapour compression system) the refrigerant vapour is drawn into an adjoining "absorber" vessel containing the "absorbent" solution and gets readily absorbed into the solution, due to its strong affinity for the refrigerant. The heat of condensation and the heat of mixing released during the absorption process is removed by another fluid such as "cooling water" or ambient air so as to maintain the low vapour pressure condition required for continuous evaporation of refrigerant and its onward absorption by absorbent solution to continue the cycle.

Generation

The absorbent solution which has become dilute after absorbing the refrigerant vapour in the absorber is then pumped by a solution pump to the higher temperature and pressure "generator".

In the generator the dilute solution is heated by means of steam / hot water or direct gas/oil firing so as to concentrate the solution and boil off and discharge the hot refrigerant vapour to the condenser.

The hot concentrated solution which has now regained its strong affinity for absorbing more refrigerant, then returns to the 'absorber' via a heat-exchanger after transferring its sensible heat to the cold dilute solution (being pumped from the absorber to the generator). This 'solution heat-exchanger' improves the 'cycle efficiency' as pre-heating of the dilute solution in the heat exchanger before entering the generator reduces the heat-energy input to the generator and the simultaneous pre-cooling of the concentrated solution returning to the 'absorber' will reduce the extent of cooling to be done to the solution by the coolant, in the absorber section.

About the Author
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This series of articles by M. M. Roy covers the fundamental principles of refrigeration and air conditioning, applications, design, equipment, components and systems. The articles will serve as a source of reference for new comers joining our industry as well as for experienced personnel wanting to brush up on basics.

Characteristics of *Refrigerant – Absorbent* Pairs Chosen for Absorption Cooling

A fluid pair comprising lithium bromide salt solution as 'absorbent' and water as refrigerant is commonly used for air-conditioning applications.

- At the operating temperatures and pressures encountered, water as refrigerant is much more volatile compared to lithium bromide which is practically non-volatile. Hence it is feasible to separate the refrigerant

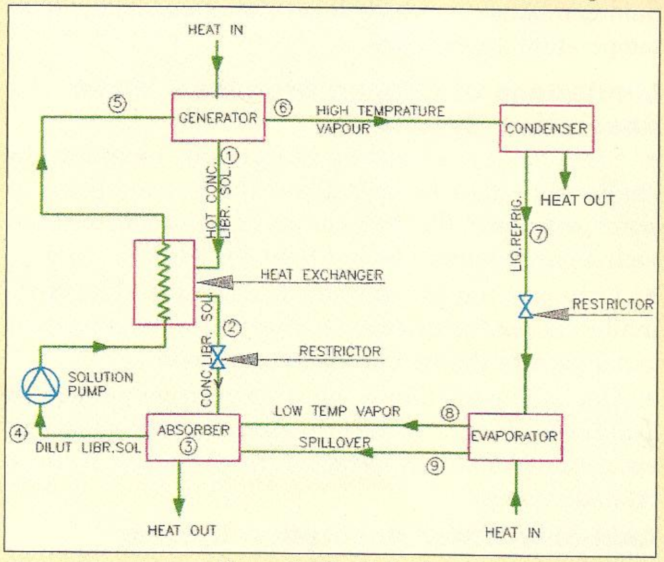


Figure 1 : Basic Cycle
Lithium Bromide-Water single stage absorption refrigeration cycle

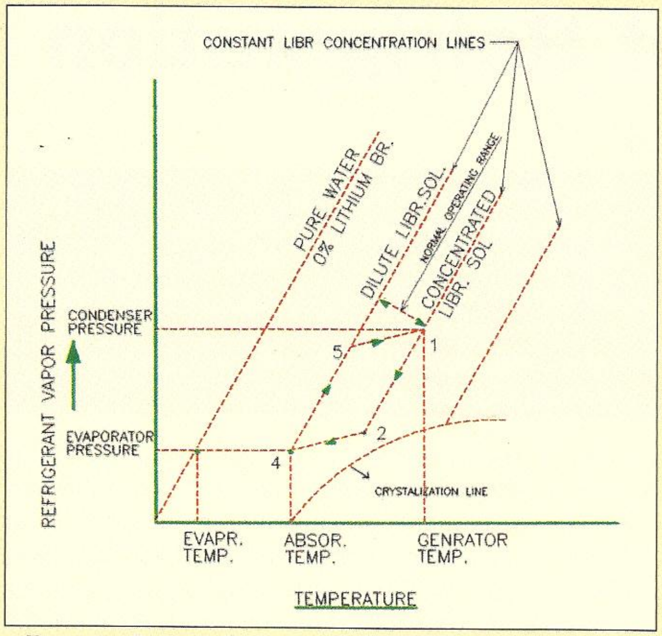


Figure 2 : Pressure Temperature Concentration points for the Lithium Bromide-Water single stage absorption refrigeration cycle
1–2 : Cooling of hot conc. LiBr soln. in Heat Exchanger
2–4 : Dilution and cooling of LiBr soln. in Absorber
4–5 : Preheating of cold dilute LiBr soln. in Heat Exchanger
5–1 : Heating and concentration of LiBr soln. in Generator

from the absorbent for proper evaporation of the refrigerant in the evaporator.

- A concentrated lithium bromide solution has great affinity for water.
- Since the operating pressures are low in the lithium bromide-water absorption system, the pumping cost is low and also the wall thickness of the shell gets reduced compared to the earlier combination of ammonia-water fluid combination, which is now used mainly for low-temperature applications.

Limitations of Lithium Bromide – Water Absorption System

- Since water is used as refrigerant, evaporation temperature must be kept above the freezing point of water and hence the temperature of chilled water from such a system cannot be less than 5°C (41°F).
- Lithium bromide solution is corrosive. Therefore inhibitors need to be added to the system to protect the metal parts of the system against corrosion.
- Coolant temperatures must be relatively lower to avoid crystallization of the lithium bromide which makes air cooling of the absorber, difficult in the lithium bromide system.

Ammonia-Water Absorption System

A fluid pair using ammonia as refrigerant and water as absorbent is now mainly used for low temperature refrigeration applications and to a limited extent for small capacity air-conditioning units with air-cooled absorption systems.

The advantages are: Water has great affinity for ammonia and can dissolve enormous amounts of ammonia vapour.

and

There is no solidification problem encountered with ammonia refrigerant over a very wide range of evaporating temperatures even upto very low temperature applications.

The limitations are: Since water also evaporates when the ammonia water solution is heated in the 'generator', the ammonia vapour from the 'generator' is mixed with water vapour. If the ammonia vapour mixed with water-vapour reaches the condenser, condensation of water vapour will interfere with the evaporation of the ammonia liquid in the evaporator and reduce the refrigeration capacity.

Therefore, the vapours from the generator are passed through the 'rectifier' where the vapour mixture is cooled so as to condense the water vapour and return the water to the generator and the rectified ammonia vapour then passes to the condenser, as shown in the flow-diagram in Figure 3, to continue the remaining operations as in the case of the lithium bromide-water absorption system.

Single Effect Absorption Chiller

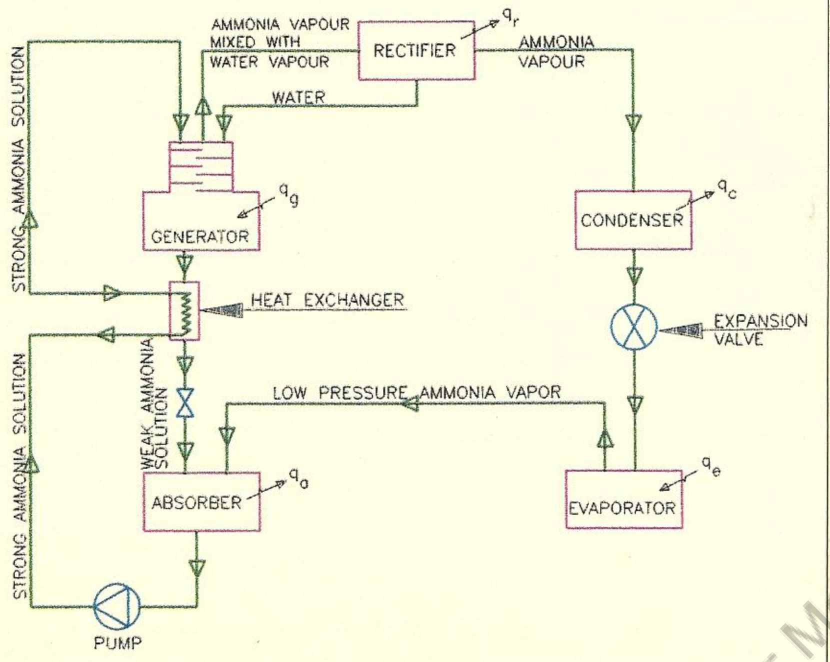
Single effect or single stage absorption cycle, as shown, in Figure 1 and Figure 2 uses "single-stage generation" for increasing concentration of the dilute solution by heating the solution in a single generator and boiling off the refrigerant vapour from the solution with heat input from low-pressure steam (around 1 kg/cm^2 g. pressure) or from hot water at (85°C to 95°C)

Single effect system also has one "solution heat exchanger" for heat exchange between incoming cold dilute solution from the absorber and the returning hot concentrated solution from the generator.

General Arrangement of Typical Single-Effect Absorption Chiller

Figure 4 shows a schematic diagram of a single effect steam-fired absorption chiller, working on lithium bromide-water system, consisting of two sections in one shell. The lower section has two tube bundles - absorber tube-bundle at the bottom and the evaporator tube-bundle located above the absorber - both operating at high vacuum conditions i.e. very low pressures of the order of 7 mm mercury absolute (1/100 atmosphere).

The upper section also contains two tube bundles - the generator and the condenser - operating at pressures



q_e = Heat input to low pressure ammonia vapour in evaporator i.e. Refrigeration Effect
q_g = Heat input to ammonia solution in generator to drive off ammonia vapour to condenser via rectifier
q_a = Heat rejected during process of absorption of ammonia (refrigerant) vapour by weak solution in absorber
q_c = Heat rejected during condensation of Ammonia refrigerant in condenser
qr = Heat rejected during condensation of water vapour for seperation from ammonia vapour in rectifier

Figure 3 : Ammonia absorption refrigeration cycle

of the order of 70 mm mercury absolute (i.e. 1/10 atmosphere).

In addition there is a solution pump, a refrigerant pump, a heat exchanger, control valve and temperature controller and a purge unit (to remove non-condensable gases from the system).

Function of Main Components

Generator & Condenser Section

- During operation, heat supplied by steam circulating inside the generator-tubes causes a portion of the refrigerant (water) to boil off, thus concentrating the dilute solution on the outside of the tubes.
- The hot refrigerant vapour flows through eliminators (which prevent carry-over of lithium bromide) to the condenser, where it is condensed on the outside of the tubes by giving up its heat of condensation to cooling water passing through the tubes. This cooling water is the same water that has been previously used to cool the absorber.
- Condensed refrigerant flows by gravity and pressure differential through a restrictor to the evaporator.

Evaporator & Absorber Section

The liquid refrigerant (water) evaporates in the evaporator, at a low evaporation temperature of about 4°C (corresponding to the low evaporator pressure of 7 mm mercury absolute) by taking the heat of evaporation from the return chilled water, say at 12°C (from the air conditioning system) which is cooled down to the desired outlet chilled water temperature, say 7°C for supply to the air conditioning system.

Liquid refrigerant from the evaporator sump is pumped by a refrigerant pump and sprayed over the evaporator tubes through a spray header and nozzles, for improved heat transfer.

Refrigerant vapours from the evaporator flow through eliminators, (to prevent carry over of liquid refrigerant) to the absorber, attracted and absorbed by the lithium bromide solution flowing over the outside of the absorber tubes, thus diluting the solution. Heat released during the process of absorption is removed by cooling water (from a cooling tower) flowing through the absorber tubes.

The solution is circulated through the distribution system and sprayed over the absorber tubes by the solution pump for better heat transfer.

Heat Exchanger

The heat exchanger provided for heat exchange between the cold dilute solution before entering the generator and the hot concentrated solution leaving the generator, improves the 'cycle efficiency' by reducing the heat input required in the generator and reducing the cooling water flow in the absorber.

Capacity Control

Capacity of the unit is automatically controlled to meet the load variation, by varying the rate of reconcentration of the dilute solution in the generator by regulating the steam supply to the unit with a control valve at the steam condensate outlet getting a command for opening or closing, from the 'sensor' sensing the outlet chilled water temperature.

Duhring Diagram

The absorption cycle's performance can be read in Figure 5 which depicts the temperature-pressure concentration relation for the lithium bromide aqueous solution in the entire cycle, as explained below:

1. **At Generator Outlet:** shows concentrated and heated state of lithium bromide solution (65% lithium bromide by weight) and temperature 103°C

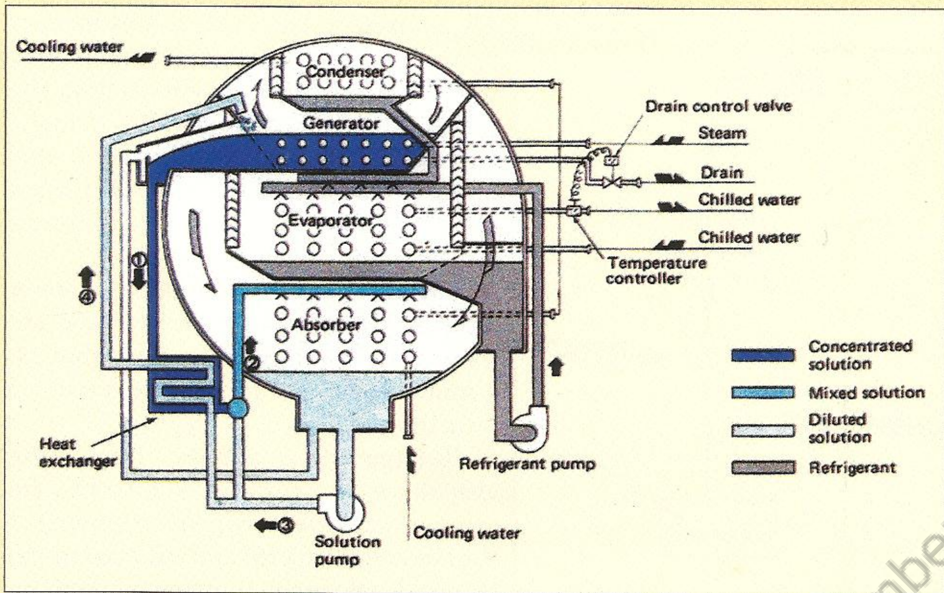


Figure 4 : Schematic diagram of a single effect steam fired absorption chiller

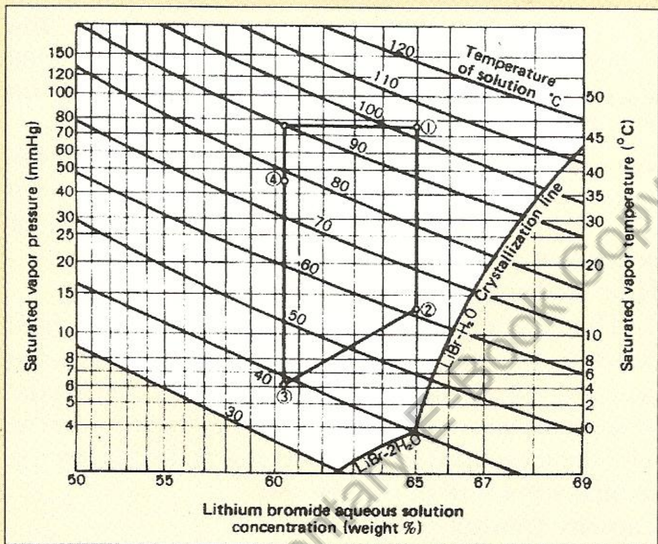


Figure 5 : Duhring diagram for determining the performance of an absorption cycle

and corresponding vapour pressure of 75 mm mercury Absolute.

2. **At Absorber Inlet :** lithium bromide concentration remaining unchanged i.e. 65% but solution temperature coming down to 61°C (after passing through heat-exchanger) and vapour pressure 13 mm mercury Absolute.
3. **At Absorber Outlet :** Lower solution concentration of 60.3% reached after dilution and cold solution temperature of approx. 38.5°C.
4. **Inlet of Generator :** Solution concentration remaining unchanged i.e. 60.3% (as in 3 above) but higher solution temperature of approx. 78°C attained due to pre-heating in the heat-exchanger.

Co-efficient of performance (i.e. useful refrigeration effect in the evaporator divided by the heat input in generator) of single effect absorption chillers is rather low (ranging between 0.68 to 0.75). However, a single-effect vapour absorption system offers the unique benefits of obtaining useful refrigeration for air-conditioning or process chilling through heat recovery from low-pressure and temperature waste steam in certain industrial processes or power plants using condensing back-pressure turbines, as well as heat recovery from a relatively lower-temperature flue-gas (such as a diesel engine / gas engine exhaust) to produce hot water for operating the single effect vapour absorption cycle.

Next Issue : Part 5 – Double Effect operating cycles, Steam Fired and Direct Fired Absorption machines. ❖

Fundamentals of Refrigeration

Part 5

Double-Effect Operating Cycles, Steam-Fired and Direct-Fired Absorption Machines

This series of articles by M. M. Roy covers the fundamental principles of refrigeration and air conditioning, applications, design, equipment, components and systems. The articles will serve as a source of reference for new comers joining our industry as well as for experienced personnel wanting to brush up on basics.

The current range of absorption chillers typically operate on the double-effect absorption technology, for effective utilization of high temperature heat available from heat sources such as high pressure steam (pressure 8 kg/cm^{2}g & temperature 180° C) and direct combustion of natural gas and fuel oils.

Double effect absorption machines utilizing indirect heating by steam are known as steam fired types whereas the direct fired models are those utilizing the heat derived from direct combustion of natural gas or fuel oils.

The double effect absorption machines operating on the double effect absorption cycle, came into vogue around three decades ago and enjoy a dominant position in the market today, due to significant performance enhancement of 50% over the basic "single effect cycle". The COP (co-efficient of performance) of modern double effect absorption machines utilizing high temperature heat, ranges between 1.1 to 1.2 compared to the low COP of 0.68 to 0.75 for single effect machines, operating with lower temperature heat input at 90° C to 120° C from hot water / low pressure steam. The enhanced performance in double effect cycle is attained by providing two generators and two condensers and additional heat exchangers instead of only one generator and one condenser in the single effect machine. Figure 1 shows major components and working of a two-stage (i.e. double-effect) steam-fired absorption machine.

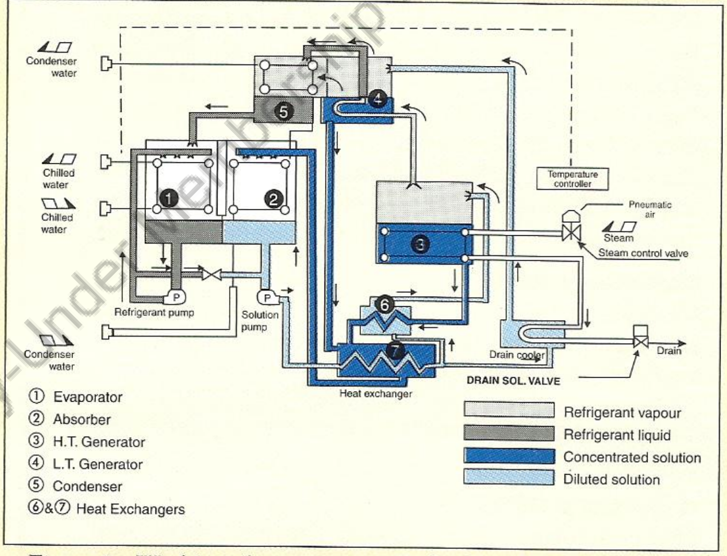


Figure 1 : Working of two stage steam–fired machines (Paraflow)

Major Components of Double Effect Machines

High-Temperature Generator (H T G)

In the double effect cycle, the first stage or high temperature generator (HTG) receives the externally supplied heat from external heat sources at high temperature to boil refrigerant vapor from the dilute lithium bromide solution in the HTG, thereby concentrating the Lithium Bromide solution in the HTG-shell.

Low-Temperature Generator (LTG)

The tube-bundle in the low temperature generator (LTG) acts as the "additional" condenser for condensing hot refrigerant vapor coming from the HTG, by rejecting the heat of condensation to dilute solution on the shell side of LTG, so as to concentrate the lithium bromide solution in LTG by boiling off refrigerant vapor from this solution.

Thus additional refrigerant vapor is produced in the second generator i.e. LTG without any additional "primary heat input", thereby improving the COP of the system.

High Temperature Heat Exchanger (H T H X)

This is a solution to solution shell and tube heat exchanger placed in the solution streams flowing to and from the HTG, for pre-heating the dilute solution going to the HTG. This reduces the primary heat input required from external sources, resulting in energy-saving.

Drain-Cooler (Condensate Heat Exchanger)

This is an additional heat exchanger incorporated in the double-effect steamfired absorption machines for recovering heat from steam-condensate and utilize this heat

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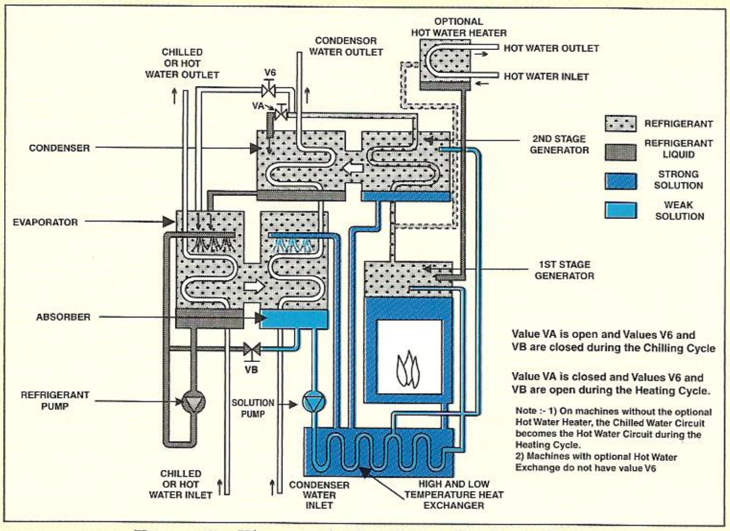

Figure 2 : Flow cycle for direct-fired machine

to preheat the dilute or intermediate concentration solution flowing to LTG or HTG, for pre-heating the solution thus reducing the steam flow required for producing the required "refrigeration effect".

Thus the steam input required at 8 kg/cm^{2}g pressure in a typical steam fired double effect absorption chiller producing chilled water, say at 6.7° C for an air-conditioning application is approx. 4.5kg per hour per ton of refrigeration i.e. 42% less than the steam input of approx. 7.8 kg/hour / ton of refrigeration at inlet pressure of 1 kg/cm^{2}g in a single effect machine.

Function of Other Components of Double-Effect Vapor Absorption System

The remaining components namely condenser, evaporator, absorber, solution pump, refrigerant pump and low temperature solution heat exchanger in a double-effect machine function in a similar way as enumerated earlier for single-effect machines (Reference: Fundamentals of Refrigeration - Part 4, July - September 2001 Issue), but explained briefly in the context of special features of double-effect machines.

Condenser

The condenser in a double effect machine is designed to handle refrigerant from two sources:

1. Collect liquid resulting from the condensing of hot refrigerant vapor produced by the first stage generator (i.e. high temperature generator), inside tubes of the second-stage generator, (i.e. low-temperature generator) which discharges into the condenser shell.

2. Refrigerant vapor produced in the second stage generator i.e. low temperature generator shell (due to concentration of solution in this generator on account of rejection of heat of condensation of 1st stage refrigerant vapor inside the tubes of the 2nd stage generator which migrates into the condenser to get liquefied in the condenser shell by rejecting its heat of condensation to cooling water circulating inside the condenser tube-bundle).

Both the streams of refrigerant liquid collecting in the condenser sump then flow down by pressure differential through a restrictor to the Evaporator.

Evaporator

Liquid refrigerant which is sprayed as a fine mist over the evaporator tubes by means of the refrigerant pump, readily evaporates at a low evaporation temperature of about 4° C at the low evaporator pressure of 7 mm mercury absolute.

This heat of evaporation i.e. refrigeration effect is derived from cooling of the returning chilled water from the air conditioning system (say from 12° C to 7° C), inside the evaporator tube-bundle. The chilled water is then supplied back to the air conditioning system.

Absorber

The refrigerant water vapor produced in the evaporator readily migrates into the absorber maintained at low pressure of 6.5 mm mercury to get readily absorbed by the concentrated lithium bromide solution due to its strong affinity for water vapor, which results in dilution of the lithium bromide solution. This process of absorption of water vapor, in lithium bromide solution gives off heat which is removed by cooling water (coming from the cooling tower) circulating inside the absorber tubes. The absorbent solution is sprayed by a solution spray pump over the external surfaces of the absorber tubes for efficient and uniform absorption of incoming water vapor by the lithium bromide solution and thus helps to maintain the vacuum condition in the Evaporator. Performance additive usually Octyl alcohol is added in minute quantity to the Lithium Bromide solution to reduce surface tension and improve convection and the rate of absorption of water vapour by the solution.

Double Effect Direct-Fired Absorption Machines

Figure 2 is a schematic of a double-effect i.e. two-stage direct-fired vapor absorption machine. Except for substitution of direct-fired 1st Stage generator instead of steam-fired and elimination of the drain-cooler (steam-condensate heat-exchanger) not required in the direct-fired machine operation, all other major components of a direct-fired machine are similar to what is described above in case of the steam-fired type. Sometimes the direct-fired machines are designed to simultaneously provide chilled water as well as hot water by providing an optional shell and tube hot water heat-exchanger receiving a portion of hot refrigerant vapor from the 1st stage generator to provide necessary heating.

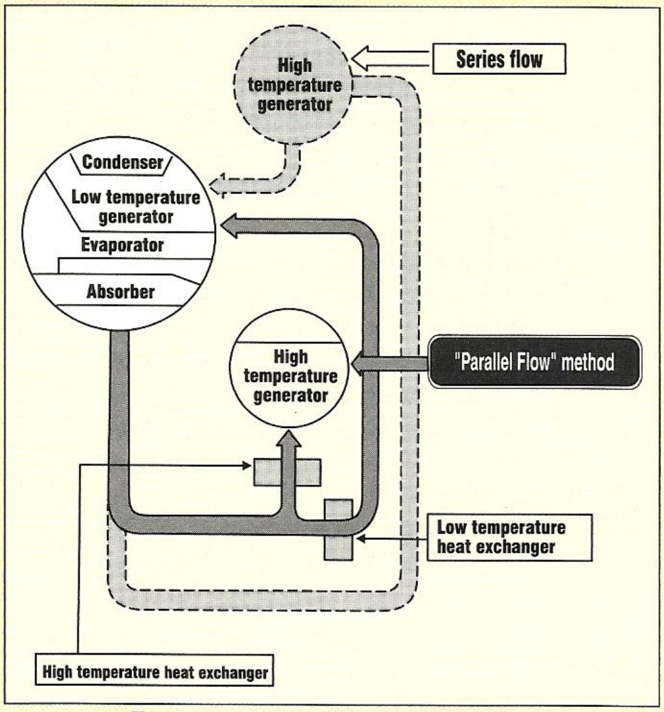

Figure 3 : Basic solution flow diagram

There are **direct-fired chiller-cum-heater** models available for application in extreme climate areas, where the same machine can work on heating cycle in winter by providing additional valves as shown in Figure 2.

Chilling Cycle

Valve V_A is open and valves V_6 and V_8 are closed during the chilling cycle.

Heating Cycle

Valve V_A is closed and valves V_6 and V_8 are open during the heating cycle.

Operational and Safety Controls in Double Effect (Steam-Fired/Direct Fired) Machines

Modern machines are equipped with microprocessor based control systems for automatic modulation of capacity and external heat intake in response to load variation. Temperature of chilled water at outlet of chiller is preset at desired value. Any deviations from the set point is sensed by the sensor placed at chiller outlet, which gives the command signal to automatically modulate the "heat input" till the chilled water temperature is restored to preset value. Modulation of heat input results in changes to concentration of absorbent solution supplied to the absorber, if pumped solution flow remains constant.

In some machines, provision is made for reducing solution flow through speed control of solution pump in part load operation so as to improve part load efficiency. In addition to capacity controls, all double effect absorption machines are provided with safety cut-out features as follows:

- Low chilled water temperature cut out to shut down chiller when preset low temperature is reached.
- Low refrigerant temperature cut out: a sensor in the evaporator monitors refrigerant temperature and as refrigerant low limit temperature is reached, it prevents further loading, then unloads and finally brings about a shut down.
- Flow switches: in chilled water and cooling water circuits of the chiller, will trip the chiller in the event of stoppage / less flow in either of these circuits.
- Pump motor overload / over temperature will trip chiller cut-outs when current limit is exceeded or safe temperature is exceeded.

Absorbent Concentration Limit to Prevent Crystallization

Solution and refrigerant temperatures are sensed during operation and used to keep a safe margin between solution temperature and solution crystallization temperature at key points in the absorption cycle so that whenever the safety margin is reduced, the control system prevents further chiller loading and unloads / shuts down the chiller if required to prevent solution crystallization.

Many absorption chillers also incorporate a "built-in overflow system" between liquid refrigerant sump and the absorber sump, so that in the event of increase of absorbent solution concentration resulting in reduction of solution level in absorber sump and increase in refrigerant liquid level in the evaporation sump, refrigerant will overflow from the evaporator sump to the absorber sump to dilute the absorbent solution to a safe level.

Additional controls for Direct-Fired Machines

- Burner fault

  Operation of the burner is monitored by its own control system and in case of a burner fault, a signal is passed to chiller control system for necessary chiller shut down.

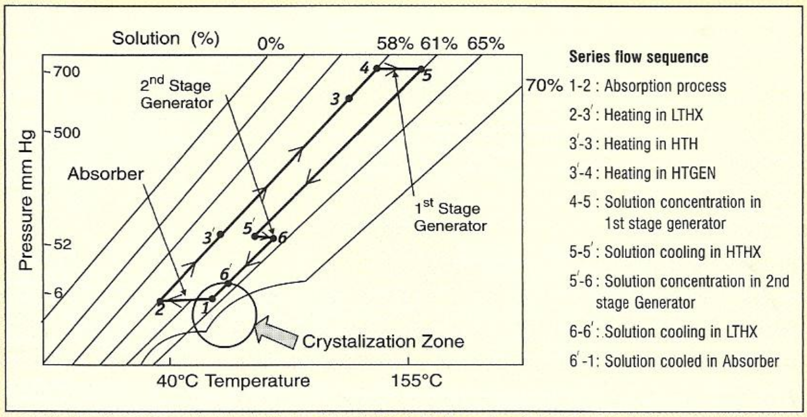

Figure 4 : Series flow cycle

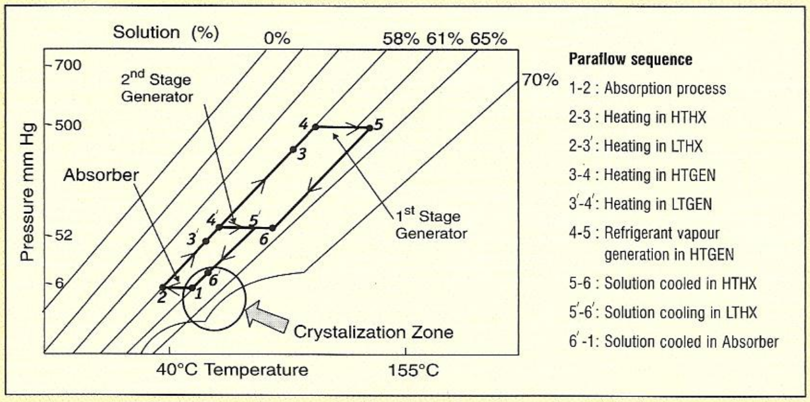

Figure 5 : Parallel flow cycle

- High temperature limit
  Direct fired chillers also have a temperature sensor in liquid absorbent solution near the burner fire tube and whenever temperature reaches high limit, the control prevents further loading, then unloads and further causes the chiller to shut down if necessary.
- High pressure limit
  A pressure sensor is placed in the vapor space in the 1st Stage i.e. (high temperature) generator shell so that in the event of the first stage generator pressure exceeding the high limit the control prevents further loading, then unloads and finally can cause chiller to shut down.

Purge system

Non-condensibles (air) which inevitably enter the system (being in a vacuum condition) in spite of leak proof construction and some non-condensable gases generated internally during operation are collected in the purge separator and tank and purged out periodically by running the purge pump.

Efficient purging is necessary to avoid corrosion. Check the depletion of corrosion inhibitor and get desired capacity / efficiency.

Different Solution Flow Cycles Used in Double Effect VAMs (Steam-Fired / Direct- Fired)

Three different solution flow cycles namely "Series Flow", "Parallel Flow" and "Reverse Parallel Flow" are being used by different manufacturers of double effect absorption chillers, for taking the dilute solution from the solution circulation pump discharge through the heat-exchangers and both the generators.

1. **Series Flow**

   This cycle was the first to be developed and is still being commercially used by some manufacturers.

   The "Series Flow" method essentially sends all of the diluted solution from the absorber outlet to the 1st stage i.e. high-temperature generator for first stage of concentration of solution and refrigerant vapor generation and thereafter the entire intermediate concentration solution is sent to the 2nd stage i.e. low temperature generator for further concentration of solution and additional refrigerant vapor generation, as shown in the basic diagram - Figure 3.

   The exact sequence of solution flow in "Series Flow" through heat exchangers and generators is given below and also illustrated in Figure 6.

   The entire dilute solution from absorber outlet passes through a pump and then flows through the following in sequence:

   - Low temperature heat exchanger (for pre-heating)
   - High temperature heat exchanger (for further heating)
   - 1st stage i.e. high temperature generator (to concentrate solution and generate refrigerant vapor by external heat input)
   - High temperature heat exchanger (to cool the intermediate concentration solution)
   - Second stage i.e. low temperature generator (for further concentration of solution and additional refrigerant vapor generation)
   - Low temperature heat exchanger (for cooling of concentrated solution)
   - Absorber (for dilution process)

2. **Parallel Flow**

   In the "Parallel Flow" system, diluted solution from absorber outlet after passing through the solution pump is divided into two streams at the pump discharge, with one stream of solution going to the high temperature generator, attaining a higher concentration compared to the second stream of solution sent to the low temperature generator, reaching moderate concentration level.

   Both the streams of concentrated solution are then joined together and cooled in the low temperature heat exchanger before going back to he absorber for dilution process.

   Schematic diagram in Figure 3 shows the basic schemes of solution flow as well as the detailed sequence of flow from the Absorber through the heat exchangers and respective generators.

Duhring Diagrams Series and Parallel Flow (Figures 4 and 5)

The sequence of solution flow, vapor pressures, and typical concentration levels of the Libr solution at different operating points on the cycle are drawn on the same Duhring diagram to illustrate the difference between the traditional "Series-Flow" and subsequently

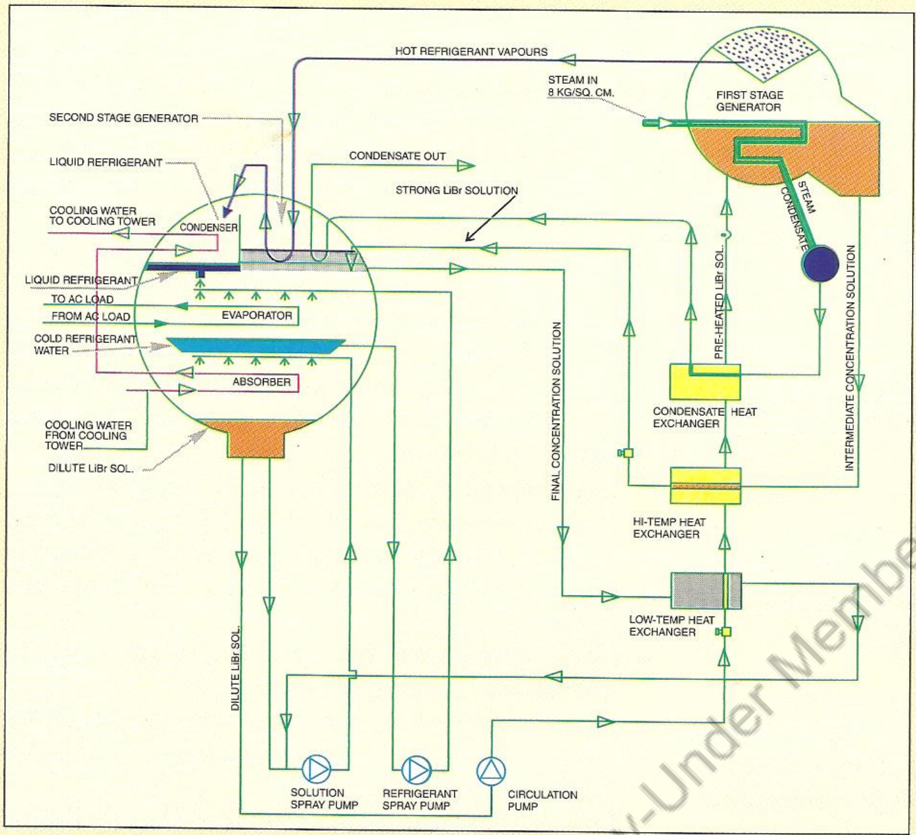

Figure 6 : Double effect steam-fired absorption chiller series flow solution cycle

commercialized "Paraflow" system of solution flow. Paraflow cycle operates at lower concentration cycle and away from crystallization line compared to "Series Flow".

3. **Reverse Parallel Flow**

   This is a later development with some modifications to the parallel flow system.

   As shown in the schematic diagram Figure 4, the entire solution leaving the absorber is pumped through the low-temperature heat exchanger and then to the low - temperature generator. After reaching intermediate level of concentration in the LTG, the solution flow is then split, with a porion going to the low-temperature heat-exchanger. The remaining solution is then pumped by another pump through high temperature solution heat exchanger to the high temperature generator and after attaining high concentration and temperature this solution stream returns via the shell side of the high temperature heat exchanger and then mixed with the other stream of solution (returning from the LTG) before returning back to the absorber at a mixed concentration condition. Reverse Paraflow cycle also operates at lower concentration level and away from crystallization line compared to "Series-Flow".

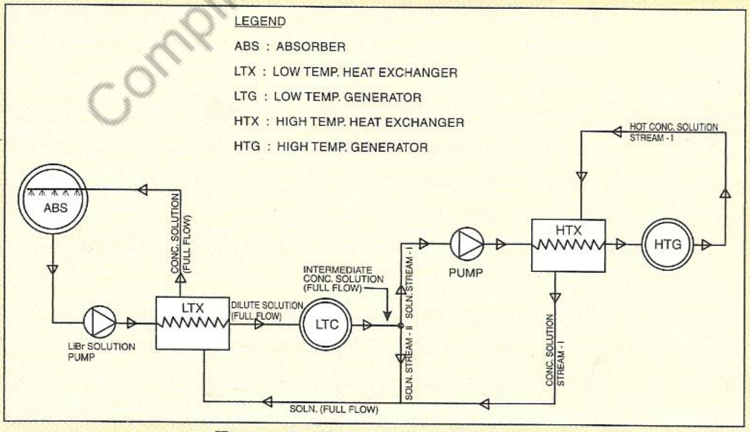

Figure 7 : Reverse parallel flow

New Developments in Vapor - Absorption Cycles

Triple effect cycles have been recently developed by providing three generators and three condensers instead of two generators and two condensers in the double effect absorption system.

In the triple effect system, primary heat (from natural gas or fuel oil burner) concentrates absorbent solution in the first stage generator at about 230° C. A fluid pair such as aqeous mixture of nitrate salts which is more stable at high temperature compared to aqeous lithium bromide can be used for the high temperature cycle. The refrigerant vapor produced in the 1st stage generator is then used to concentrate additional absorbent solution in the 2nd stage generator at about 150° C. Finally the refrigerant vapor produced in the 2nd stage generator concentrates additional absorbent solution in a 3rd stage generator at about 90° C. Solution heat exchangers are used at these stages for preheating solution before entering respective generators for concentrating the solution, in order to improve the cycle efficiency.

COP is improved to about 1.7 in triple effect cycle as compared to 1.1 to 1.2 in double effect cycles.

However, corrosion inhibition and monitoring of stability of solution and performance additives (octyl alcohol) becomes critical and pressure vessel design for 1st stage generator becomes necessary in triple effect systems.

Next Issue : Design and selection of a refrigeration system for cold storage applications

Fundamentals of Refrigeration

Part 6

Refrigeration Systems for Cold Storages and Heat Load Estimation

Refrigeration systems for cold storage applications are designed to maintain simultaneously the desired low temperature along with relative humidity and air movement within a vapour–sealed and insulated chamber for preservation of food products like fruits, vegetables, fish, meat, poultry, dairy products, yeast etc. as well as life saving drugs, blood plasma and vaccines/serums.

Holding food products at low temperature within the cold room helps to retard product deterioration by slowing down the growth of micro-organisms and chemical changes. An important design criterion is to maintain high relative humidity (in addition to the desired low air temperature) within the refrigerated chamber which reduces water loss and prevents shrinkage and dis-coloration of products during storage.

The recommended storage temperature, relative humidity, air movement, depend on the type of product to be preserved, desired shelf life (long or short), condition of the product – whether fresh or processed and pre-cooked or frozen before loading into the cold store. Recommended design storage conditions and storage life for different food products are available in ASHRAE Refrigeration Handbook and in application catalogs of various refrigeration equipment manufacturers.

The Cold Chain in Preservation of Food Products

1. Initial Product Chilling

It is desirable to reduce the initial product temperature rapidly to an acceptable level for storage and maintain the required low-temperature air quantity and flow to minimise bacteria growth and reduce water loss/shrinkage.

Fruits and vegetables direct from harvest need to be cooled down to a temperature of 2 to 4°C in approximately 18 to 24 hours and to maintain relative humidity around 85% to 90% in the chilling room.

Even freshly harvested marine products on board fishing trawlers or freshly killed meat in abattoirs require to be

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chilled to a temperature of 1 to 2°C and to maintain high relative humidity of 85 to 90% during chilling, to retain freshness, flavour, texture and reduce shrinkage/discoloration.

2. Combined Chilling & Holding of Products in the Same Room

Chilling and holding storage are combined in the case of several products like pears, apples, potatoes adn some dairy products, which can tolerate some temperature fluctuation in the cold room during the loading period.

For example in case of a potato cold storage, during post harvest loading months of February and March the cold room temperature is allowed to range between 10 to 13°C during the loading period when refrigeration load on the system is high on account of the pull-down load as potatoes don't sprout and shrinkage is less at a storage temperature of 10 to 13°C during this post harvest period of 2 to 2½ months. However, storage at a higher temperature of 16 to 18°C is not desirable because shrinkage, sprouting and decay can start. Subsequently, during the long holding season, cold room temperature is maintained around 3 to 4°C to avoid sprouting and relative humidity of around 85 to 90%.

3. Processing & Freezing of Food Products Prior to Long-term Storage in Freezers Rooms.

Frozen products can generally be preserved even for 6 to 12 months in frozen-storage chambers maintained at –18 to –22°C and relative humidity around 85 to 90%.

Products which are only precooled or partly frozen or in a thawed condition should not be transferred into the frozen-storage chambers as this will lead to slow-freezing of the products causing water loss and loss of flavour and also bring about temperature fluctuations in the frozen-storage chambers, due to the additional refrigeration load on account of the latent load of product freezing.

Hence the design of suitable freezing plants for product freezing needs to be integrated into any well designed refrigeration system for long term preservation of food

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products at low temperature of –18 to –22°C.

Two popular methods of freezing food products are adopted depending on the size and shape and whether the products are processed and packed into cartons of suitable thickness prior to freezing.

- Plate Type Contact Freezers are built within insulated cabinets and incorporate horizontal refrigerated plates with adjustable distance between plates ranging from 25 mm to 85 mm depending on the thickness of package of the food products.

Refrigerant temperatures are maintained around –35°C in the plates and the products are pre-cooled to approximately 2°C before loading inside the plate freezer unit for freezing to take place by upper and lower surfaces of the food packages being in direct contact with the refrigerated plates.

- Air blast Freezer and Tunnel Freezers are adopted to freeze larger sizes of products or products of irregular shapes and sizes which cannot be handled in plate freezers.

Applications are for freezing of pre-chilled meat carcasses, big fish, large fish fillets or for freezing of loose materials like shelled peas and sweet corn. Air blast freezers utilise rapid blast of ultra low temperature air at –40°C in insulated freezer rooms or freezing tunnels employing continuous steel mesh belt for conveying the products through the blast tunnel.

Refrigeration Load Calculations for Cold Storages

Accurate estimation of the heat loads for a refrigerated storages includes the following steps :

1. Establish Basic Design Data such as :

- Design ambient dry bulb and wet bulb temperatures at place of installation
- Required storage temperature and relative humidity (depending on type of products to be stored, condition of products before loading and desired shelf life)
- Required product's storage capacity, daily loading rate and incoming product temperature
- Dimensions, insulation (type, thickness and density), type of construction and orientation of the building
- Internal load data

The above data will enable one to work out different elements of the heat load as follows :

2. Solar & Transmission Heat Gains

The following equation is used for estimation :

Heat gains through walls/roof/floor = Area × (outside temp. – inside temp. + temp. differential correction) × (heat transfer co-efficient "U")

Figures of temperature difference corrections are for solar radiation effect in respect of various wall facings (North/South/East/West) and "U" factors depend on type of wall construction and thickness/type of insulation and are listed in ASHRAE Handbook. See Table 1 for recommended insulation thickness for different temperatures.

Table 1
General standards for insulation thickness in storage rooms

Storage Temperature		Desirable Insulation Thickness in Inches	
°F	°C	Styrofoam	Urethane
–50° to –25°	–45° to – 32°	8	6
–25° to –0°	–32° to –18°	6	4
0° to 25°	–18° to –4°	4	4
25° to 40°	–4° to 5°	4	3–4
40° and up	+5° and up	2	2

3. Infiltration of Outside Air

Outside air that is much higher in temperature and moisture content enters cold storage because of door openings for loading/unloading of products and also due to infiltration through even minor cracks around doors. Considering one 3 ft. wide door for a cold storage upto 5000 cft. volume, infiltration air flow rates of 14 cfm in case of average usage and 22 cfm for heavy usage are considered normal.

Accordingly:

Outside air sensible heat gain in BTU/hr. = (infiltration cfm) × 1.08 × (temp. difference between outside and cold room)

Outside air latent heat gain in BTU/hr. = (infiltration cfm) × 0.68 × (grains per lb difference between outside and room air read from a psychrometric chart)

However, for a refrigerated warehouse for long term storage, which is protected by a vestibule, lower infiltration rate of 12 cfm is to be considered for calculating infiltration air sensible and latent heat gains. Infiltration air cfm is to be proportionately increased for wider doors and additional doors.

4. Internal Loads

a) Lights – typical requirement will be 1 to 1.5 watt per sq. ft. (i.e. 10 to 15 watts per sq. metre). Based on estimation of total lighting wattage in the entire cold room space, using this norm, sensible heat gain due to lights in BTU/hr. can be calculated by using a multiplier of 3.42 BTU/watt.

b) Heat dissipation by cold diffuser's fan-motors can be estimated depending on the fan motors hp ratings and considering that usually the cold diffuser units with drive motors are located inside the cold storage chambers,

$$\textbf{sensible heat gain} = \frac{\textbf{(motors total consumed hp)} \times 2545}{\textbf{\% efficiency}}$$

c) Material handling equipment such as fork lift trucks which are used for loading/unloading of products in a refrigerated warehouse, cause heat gain, into the refrigerated space, depending on the loading capacity of the fork lift truck. See Table 2.

Table 2 : Heat gain from fork lift trucks

Fork Lift loading capacity lb	Heat gain per hour of truck operation (BTU/hr.)
2000	14000
4000	21000
6000	23000
8000	26000

d) Occupancy – people working in cold storage chambers for loading and unloading of products, dissipate both sensible and latent heat depending on the temperature. See Table 3.

Table 3 : Heat dissipation from people working in cold storages

	Temperature of cold storage			
	45°F	35°F	0°F	-10°F
Sensible Heat BTU/hr	580	650	890	910
Latent Heat BTU/hr	110	110	110	110

5. Product Cooling Load

Since the products to be stored enter the cold storage chambers at a higher temperature than the storage temperature, the refrigeration system has to pull down the temperature of the quantity of products loaded daily to the desired storage temperature within a period of 18 to 24 hours before the next day's loading takes place. Product cooling load is estimated as follows:

Product cooling load =

$$\frac{\textbf{(wt. of product loaded daily)} \times \textbf{(incoming product temp.} - \textbf{storage temp.)} \times \textbf{(specific heat)}}{\textbf{(pull down time in hrs.)}}$$

ASHRAE Handbook Refrigeration Systems and Applications contains data of specific heats of different food products, before freezing and after freezing, and depending on the application, i.e. whether the cold room will store fresh or frozen prouducts, appropriate "specific heat" is taken for calculation of product cooling load.

Product Freezing Load

This must be estimated only for freezing applications.

Product freezing load =

$$\frac{\textbf{(wt. of the product)} \times \textbf{(latent heat of freezing)}}{\textbf{(freezing time)}}$$

Data on the latent heat of freezing is available in ASHRAE Handbook or several engineering manuals from manufacturers of refrigeration equipment for cold storages.

6. Respiration Heat

Fresh fruits and vegetables stored inside cold storage rooms emit heat of respiration which varies with the type and temperature of products being stored and is calculated as follows :

Total respiration heat in BTU/hr. =

(total weight of products stored in pounds) × (unit reaction heat in BTU per lb/hr.)

Data on "Respiration Heat" for different fruits and vegetables is listed in the ASHRAE Handbook.

7. Safety Factor for Total Refrigeration Load

Based on the estimation of main elements of refrigeration load as above and after adding a safety margin of 10%, the total refrigeration load is calculated. A summary of all loads is given in Table 4.

Table 4 : Summary of refrigeration load estimate

I. Solar and transmission heat gains through Walls/Roof/Floor

II. Infiltration air :
   Sensible heat gain
   Latent heat gain

III. Internal heat :
   a) Lights
   b) Cold diffuser fan motors
   c) Material handling (fork lift trucks)
   d) Occupancy – Sensible heat
   – Latent heat

IV. a) Product sensible cooling load
   b) Product freezing load (where applicable)
   c) Respiration heat
   (applicable for fresh fruits and vegetables)

V a) Total Heat Gain in BTU/hr.
   (I + II + III + IV)
   b) Add safety factor 10%
   Grand Total Refrigeration Load
   V(a) + V(b)

Next Issue : Selection of a refrigeration system for cold storage applications.

Fundamentals of Refrigeration

Part 7 (Concluding part)

This series of articles by M. M. Roy covers the fundamental principles of refrigeration and air conditioning, applications, design, equipment, components and systems. The articles will serve as a source of reference for new comers joining our industry as well as for experienced personnel wanting to brush up on basics.

Selecting a Refrigerating System for Cold Storage Application

A well designed refrigerating system for preservation of perishable products should have all the components properly sized, with adequate safety margin over the calculated refrigeration load.

It is equally necessary to ensure a proper balance between the capacity of the condensing unit and the cold diffuser unit's capacity at a desirable temperature difference (TD) between refrigerant temperature in the coil and the air temperature in the cold room, depending on the nature of the stored product.

Another important criterion from the point of view of product safety and flexibility for load management, particularly for a large storage, is that the total refrigeration load be divided equally among two or more condensing units and cold diffusers to avoid product spoilage in the event of electro-mechanical failures and permit switching off one or more condensing unit / cold diffuser during low load situations for energy saving.

Yet another important consideration is ensuring adequate air quantity and air flow as necessary, for maintaining the desired relative humidity condition. Therefore, the fans of the cold diffusers need to be sized to circulate the required air quantity uniformly, through the stacks, in the entire volume of the storage space.

Based on the above criteria, a detailed procedure for selecting various components of the refrigerating system are explained below.

Condensing Unit

The sum total of the refrigeration load from various sources of heat, including safety margin, as enumerated in the last issue gives us an estimate of average cooling capacity required in BTUs or Kcal per hour with 24 hours run time of the refrigeration plant.

In practice, the condensing units are sized with higher capacity in consideration of 18 to 20 hours actual run time in 24 hours. Thus the main refrigeration equipment i.e. the condensing unit needs to be sized for increased capacity to an extent of 20% to 30% more than the average hourly cooling capacity derived by heat load calculations.

The capacity of the condensing unit is selected at a suction temperature 2°F lower than the design refrigerant evaporating temperature in the cold diffuser coil and suitable condensing temperature, depending on the ambient air temperature (for an air-cooled unit) or cooling water temperature (for a water-cooled unit) available at the location of installation.

A compact air-cooled condensing unit with reciprocating compressor is often selected in case of small capacity direct expansion type cold storage plants. A single stage compressor is suitable for storage temperatures not lower than –25°C (–13°F). However, for applications requiring –30°C (–22°F) and lower temperatures, such as blast-freezers, a two stage compressor is necessary since the high compression ratio required for –30°C and lower temperature storages cannot be attained with single stage reciprocating compressors.

In case of large capacity, low temperature refrigeration systems, rotary screw refrigeration compressors (available in single stage and compounded designs) are gaining wider acceptance due to smooth part load operation and higher energy efficiency, made possible by slide valve mechanism for stepless capacity control.

Selecting Drive-motors for Compressors

Drive motors for open-type reciprocating compressors, in case of low temperature cold storage and freezer applications, need to be oversized even upto 100% above full load power requirement, due to high pull down load and minimum HP required for starting. A disadvantage is reduced power factor and lower motor efficiency during normal holding operations.

However, since screw refrigeration compressors start virtually unloaded, it would suffice to add about 10 to 15% to the full load power for coping with the pull-down load.

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Selecting a Condenser

The heat rejected to condensers during initial pull-down operations after periodic loading of incoming products, becomes much more than the heat rejected during normal operation.

Thus it becomes necessary to select the condenser for a higher heat rejection tonnage encountered pull-down so as to prevent tripping of the compressor on high condensing pressure build-up during pull-down operations.

Ammonia refrigeration systems for low-temperature applications

While HCFC (R-22) and HFC refrigeration systems are adopted for small capacity direct expansion cold storage plants, flooded type ammonia refrigeration systems are preferred for large size, multi-product cold-storage's having several rooms requiring different temperatures for different food products.

Cold Diffuser Unit

Generally a cold diffuser unit is specified in terms of its cooling capacity in BTU/Hr/°F temperature difference (TD) between the room air temperature and refrigerant evaporating temperature in the cooling coil.

Thus any cold diffuser unit can handle a higher cooling load if selected to operate on a higher TD, (i.e. with the lower refrigerant temperature). The same cold diffuser unit can cope with less cooling load when selected to operate on a lower TD (i.e. with higher refrigerant evaporating temperature), room air temperature remaining constant.

Design cold room temperature remaining the same, a higher room relative humidity requires higher ADP (apparatus dew point) i.e. higher coil surface temperature and higher refrigerant evaporating temperature in the cooling coil, resulting in lower TD. Thus a cold storage system requiring a high relative humidity will need a cold diffuser unit selected on a lower TD as compared to another cold storage system designed with a lower room relative humidity which will need a cold diffuser unit selected for a higher TD.

Table1: Recommended design TD for different relative humidities

Type of product to be stored/ chilled	Approximate Design RH	Recommended design TD for cold diffuser selection
Vegetables, flowers, unpacked fruits, chilling room etc.	90% high RH maintained to minimize moisture-loss from products	7 to 9°F
Packaged meats & vegetables & fruits packed in cartons etc.	80% to 85% (packaged food products immune from water loss & thus lower RH acceptable)	10 to 12°F
Fruits with tough skins such as coconuts, dry fruits, onions, garlic, cheese etc.	65% to 75%	12 to 16°

Recommended design TD for selection of cold diffuser units for cold storages for different type of products, requiring different room relative humidity conditions are indicated in Table 1.

Calculation of required supply air quantity for sizing cold diffuser fans

The dehumidified supply air quantity, in cfm, is calculated by using the following formula :

$$\text{Dehumidified air cfm} = \frac{\text{Room sensible heat in B.T.U/hr.}}{\text{Air constant} \times \text{dehumidified air temp. rise °F}}$$

However, the standard air constant of 1.08 (used for air-conditioning calculations) which is based on air density of 0.075 lbs/ cu.ft. at 70°F air temperature cannot be used for cold storage supply air cfm calculations as the figure of air constant increases proportionately to higher density of air at lower supply air temperatures, as shown in Table 2.

Table2 : Air Constant for Different Supply Air Temperatures

Cold Room Temp. °F	Approx. supply air temp. °F	Air constant
40°F(potato storage)	30°F	1.16
32°F (fresh foods storage)	20°F	1.19
–5°F (frozen foods storage)	–15°F	1.30
–30°F (freezing tunnels)	–40°F	1.37

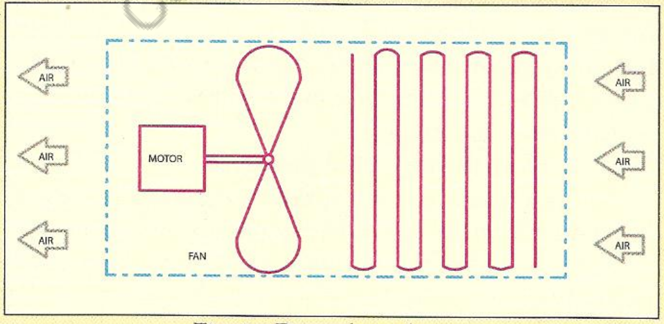

Fig. 1 : Draw-through type

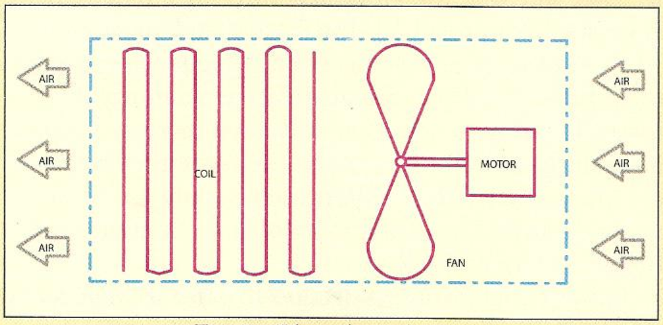

Fig. 2 : Blow-through type

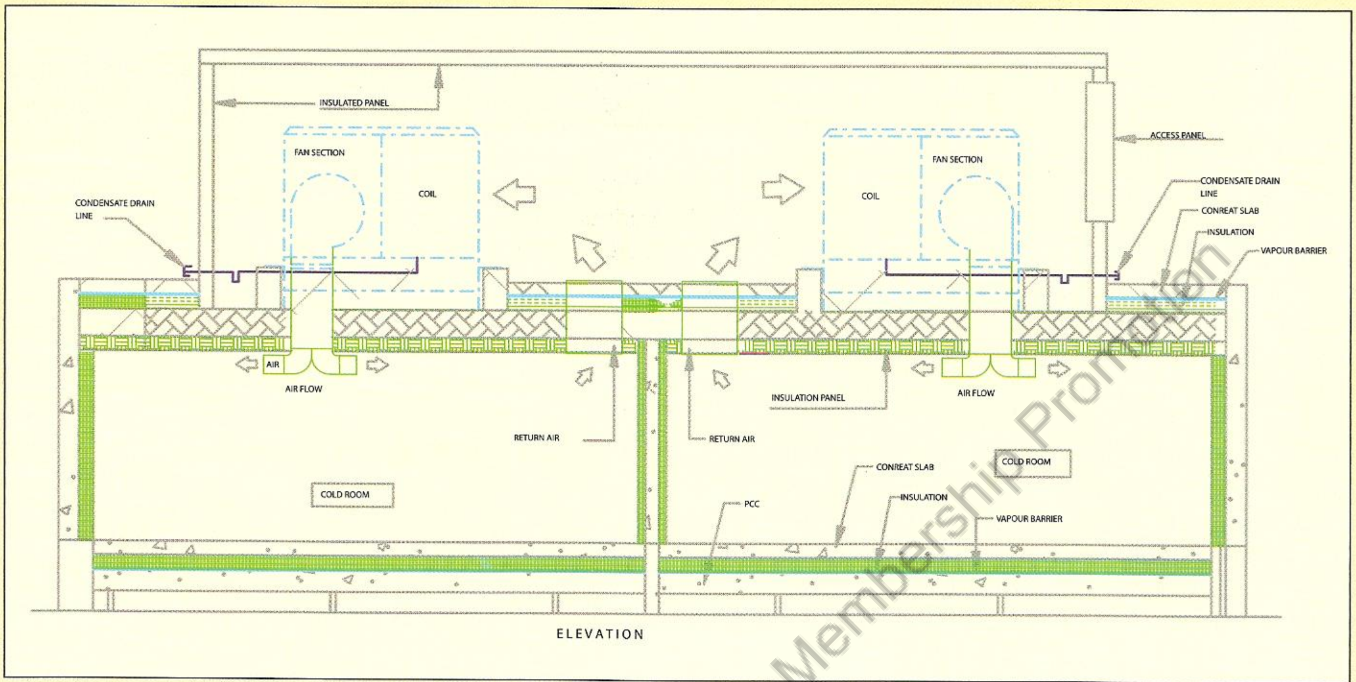

Fig. 3 : Cold diffuser installed at roof top with enclosure

Dehumidified air temperature rise in °F is calculated by using the formula :

Dehumidified air temp. rise °F = (1–Bypass factor)(Room temp. –ADP)

The Bypass factor depends on the number of rows of coil, coil face velocity and fin spacing (i.e. no. of fins/ inch).

In practice, since most manufacturers of cold diffusers list their range in terms of fan capacity in cfm and cooling capacity in BTU /hr. /°F TD, it is a question of selecting one or more cold diffusers to cope with the total room refrigeration load, based on acceptable TD for a specific product stored.

There are also broad guidelines such as required air changes per hour (between 30 to 60 for vegetables and fruit storage and between 40 to 80 for holding freezers). Therefore, the total air capacity of standard cold diffusers if selected from manufacturer's catalogues can be checked to make sure that the minimum air change requirement is satisfied, for proper air distribution and air motion.

A cold storage system when designed for storage conditions requiring a high relative humidity with a lower TD for coil selection will have increased supply air quantity as compared to another system of the same cooling capacity that requires less supply air quantity when designed for a lower room relative humidity and a higher TD for coil selection.

Thus a cold diffuser selection to cater to high room relative humidity requirement will lead to selecting a bigger cold diffuser unit with bigger size coil and higher air capacity blower unit with increased fan motor kW rating, but the condensing unit selection will be more economical and energy-efficient due to the higher operating suction temperature on account of higher evaporating temperature in the coil selected at lower TD.

Materials used for coil fabrication

The evaporator coil of the cold diffuser units can be copper tube and aluminium fin, direct expansion type, for cold storage plants working on R-22 refrigerant.

However, in case of ammonia refrigeration systems, flooded steel tube coils are used in evaporators. Steel/ stainless steel tube finned coils are in vogue for ammonia systems since copper cannot be used with ammonia because of adverse chemical reaction between the ammonia and copper.

Generally, direct expansion coils selected for cold storage applications are with a 6 row or 8 row configuration, coil face velocity around 600fpm and 4 fins per inch. However for deep-freezers and air blast freezing applications, coils with 3 fins per inch are also used (inspite of a 7.5% drop in capacity rating), so as to reduce defrosting frequency.

Draw or blow-through cold diffusers

The fans used in cold diffusers are usually of axial flow, propeller type, in order to make the units compact.

The location of the fan will determine whether the unit is of :

- Draw-through type – in which the return air is drawn through the coil and discharged through the fan, whereby the supply air temperature increases on account of motor heat, before discharging to cold room.

OR

- Blow-through type – in which heat from the motor is absorbed by the cooling coil and thus the supply air temperature is almost at the ADP before discharging to the cold room.

Blow-through cold diffusers are selected for applications of low TD where the refrigerant evaporating temperature and air temperature are quite close, such as in cold stores for preservation of flowers with minimum moisture loss.

In the case of large size cold rooms with high refrigeration loads and large design air flow requirement, with a view to distribute the supply air uniformly without duct work, multiple cold diffuser units of ceiling mounted type must be uniformly spaced.

However, in large refrigerated warehouses if ducted supply air is necessary, centrifugal fans can be selected to handle the higher static pressure. Wherever feasible cooling units with downward discharge fans (connected to ducts) can be installed in insulated roof top enclosures and air-cooled condensing units can be mounted on the roof close to the cooling units as shown in Figure 3. Return air (from the refrigerated space below) can rise up through a steel grating for rechilling and recirculation.

A roof deck installation of the cold diffuser units, if feasible, can eliminate ceiling suspended fan-coil units within the refrigerated chamber and thus periodic inspection and access for maintenance to these units can be done without entering the refrigerated chamber. The cooling units will also not be liable to any damage from material handling operations.

Potato Cold Storages

Potato cold stores are extremely common throughout India and many new stores are being built to replace the older designs. Hence a few details about such stores will be useful to the reader.

Present day economic considerations of scale favour large capacity cold storage units to hold 3000 tonnes and still larger quantities.

A warehouse design with floor to ceiling height of 9 m with two mezzanine levels created inside for storing potatos in bags upto a stacking height of 3 m or so at all three levels can be adopted for a greenfield project. 1000 tonnes storage capacity can be created on each level based on 3 levels of stacking. This design can bring down the land cost and construction cost.

Space planning of refrigerated space can be based on 3.4 m^3 (120 ft^3) volume per tonne of product.

The required number of typical storage rooms of 1000 tonnes capacity each can be planned on either side of a central corridor approximately 2.4 m wide (acting as an antichamber). Additional space is to be earmarked for refrigeration machine room, electrical and other utilities, office space, truck dock area for receiving, sorting, grading and other handling operations.

Design conditions for potatoe storage

	Dry Bulb	Relative Humidity
Long carry (holding months)	3.5°C	85%
Pull down period (post harvest loading period Feb & March)	10 to 13°C	Not critical

$$\text{Daily loading rate in each cold room} = \frac{\text{Storage capacity per chamber}}{\text{45 Days}}$$

Refrigeration load estimate

Refrigeration load estimates are to be made both for the holding season and for pull down period during the loading season.

Since maintaining room dry bulb temperature and relative humidity during the long carry holding months is critical for product preservation in prime condition (without sprouting and shrinkage) proper refrigeration equipment and cold diffuser selection is required to match the cooling load and dehumidified air cfm required for the holding conditions.

However, it is prudent to oversize the condenser, considering the higher heat rejection duty during pull down period.

The same compressors and cooling coils (selected for the lower dry bulb temperature and high relative humidity requirements during holding operations) will give approximately 25% higher capacity during pull down operations because of higher evaporating/suction pressure and this will partially offset the need for greater refrigeration capacity during pull down operations.

Since some stand-by condensing unit capacity is always recommended for cold storage installations, if required the stand-by condensing unit can be put into operation during loading operations to cope with the peak cooling load during the pull down period. Subsequently, during the long holding months, the standby condensing unit can be held in reserve.

Cold diffusers with 8 rows, copper tube, aluminum finned cooling coils with 4 fins per inch can be used for R-22 refrigeration system. Multiple cold diffuser units can be suitably placed for uniform air distribution in each cold room.

Next Issue : Starting a new series on Fundamentals of Air Conditioning. ❖