BEST PRACTICE MANUAL

HVAC-CHILLERS

Prepared for

Bureau of Energy Efficiency,
(under Ministry of Power, Government of India)
Hall no.4, 2[nd] Floor, NBCC Tower,
Bhikaji Cama Place,
New Delhi – 110066.
.

Indian Renewable Energy Development Agency,
Core 4A, East Court,
1[st] Floor, India Habitat Centre,
Lodhi Road,
New Delhi – 110003

By

Devki Energy Consultancy Pvt. Ltd.,
405, Ivory Terrace, R.C. Dutt Road,
Vadodara – 390007, India.

2006

CONTENTS

LIST OF FIGURES

LIST OF TABLES

1 INTRODUCTION

1.1 Refrigeration & Air-conditioning

India, being a warm tropical country, most of the refrigeration and HVAC applications involve cooling of air, water, other fluids or products. Heating is used only for a very small period in winter in the northern parts of the country and in places at high altitudes.

Refrigeration and Air-conditioning accounts for a significant portion of the energy consumption in many manufacturing industries (like chemicals, pharmaceuticals, dairy, food etc.), agricultural & horticultural sectors (mainly cold stores) and commercial buildings (like hotels, hospitals, offices, airports, theatres, auditoria, multiplexes, data processing centers, telecom switching exchanges etc).

Refrigeration and air conditioning systems cover a wide variety of cooling applications, using both standard and custom-made equipments.

Some commonly used applications are process cooling by chilled water or brine, ice plants, cold stores, freeze drying, air-conditioning systems etc.

Comfort air-conditioning generally implies cooling of room air to about 24°C and relative humidity in the range of 50% to 60%. Industrial process air conditioning and precision air conditioning may require temperatures ranging from 18°C to 24°C with relative humidity values ranging from 10% to 60%.

This manual highlights some issues related to energy efficiency and energy conservation in refrigeration and air conditioning, along with some emerging innovative techniques to eliminate or minimize conventional refrigeration and air conditioning.

1.2 The Need for Radical Thinking

With industrial development, the demand for process related refrigeration and air conditioning is increasing. However, modern lifestyles with the rising demand for comfort air conditioning in commercial buildings and homes, using conventional air conditioning methods and equipments, are a source of concern for an energy scarce country like India.

Significant developments have taken place in the technology related to the hardware and controls related to refrigeration and air conditioning systems to help improve energy efficiency. However, the right attitude and design philosophy can play a larger role than technology in conserving energy. This implies that use of nature-assisted cooling techniques and minimal use of energy guzzling refrigeration equipments is the key energy conservation.

There is an urgent need to promote and commercialize all proven techniques that use natural processes to eliminate or minimize the use of conventional refrigeration and air conditioning. Basic design of industrial processes and also architectural spaces should strive to minimize the demand for conventional refrigeration and air conditioning. A strategy focusing on both refrigeration load reduction and energy efficiency improvement for conventional refrigeration is necessary to limit the unmitigated growth of conventional refrigeration.

2 REFRIGERATION & AIR CONDITIONING SYSTEMS

2.1 Introduction

The most commonly used systems for industrial and commercial refrigeration and air conditioning are *Vapour Compression Refrigeration System* and *Vapour Absorption Refrigeration System*.

Vapour compression machines, usually with electrically driven compressors, are the most commonly used machines for refrigeration and air conditioning for temperatures ranging from 25°C to -70°C.

Vapour Absorption Refrigeration machines, wherein heat energy is consumed, are being increasingly used. Absorption refrigeration machines may be economical in situations where process waste heat or cheap fuels (usually coal or agro-waste) are available.

The uncertainty of energy availability and prices is likely to increase the preference for *Hybrid Systems*, incorporating both Vapour Compression and Vapour Absorption. With the availability natural gas, engine driven vapour compression system or engine waste heat based vapour absorption systems may also find increasing use.

2.2 Refrigeration System Efficiency

The *cooling effect* of refrigeration systems is generally quantified in *tons of refrigeration.* The unit is derived from the cooling rate available per hour from 1 ton (1 short ton = 2000 pounds = 907.18 kg) of ice, when it melts over a period of 24 hours.

British measuring units are still popularly used by refrigeration and air conditioning engineers; hence it is necessary to know the energy equivalents.

1 Ton of Refrigeration (TR) = 3023 kcal/h
= 3.51 $kW_{thermal}$
= 12000 Btu/hr

The commonly used figures of merit for comparison of refrigeration systems are *Coefficient of Performance* (COP), *Energy Efficiency Ratio* (EER) and *Specific Power Consumption* (kW/TR). These are defined as follows:
The
If both refrigeration effect and work done by the compressor (or the input power) are taken in the same units (TR or kcal/hr or kW or Btu/hr), the ratio is

$$COP = \frac{\text{Refrigeration Effect}}{\text{Work done}}$$

If the refrigeration effect is quantified in Btu/hr and work done is in Watts, the ratio is

$$EER = \frac{\text{Refrigeration Effect (Btu/hr)}}{\text{Work done (Watts)}}$$

Higher COP or EER indicates better efficiency.

The other commonly used and easily understood figure of merit is

$$\text{Specific Power Consumption} = \frac{\text{Power Consumption (kW)}}{\text{Refrigeration effect (TR)}}$$

Lower Specific Power Consumption implies better efficiency.

2.3 Vapour Compression Systems

2.3.1 Operating Principle

Vapour compression systems are commercially used for both very small systems (like window air conditioners, refrigerators etc.) to very large systems (in industries and commercial buildings).

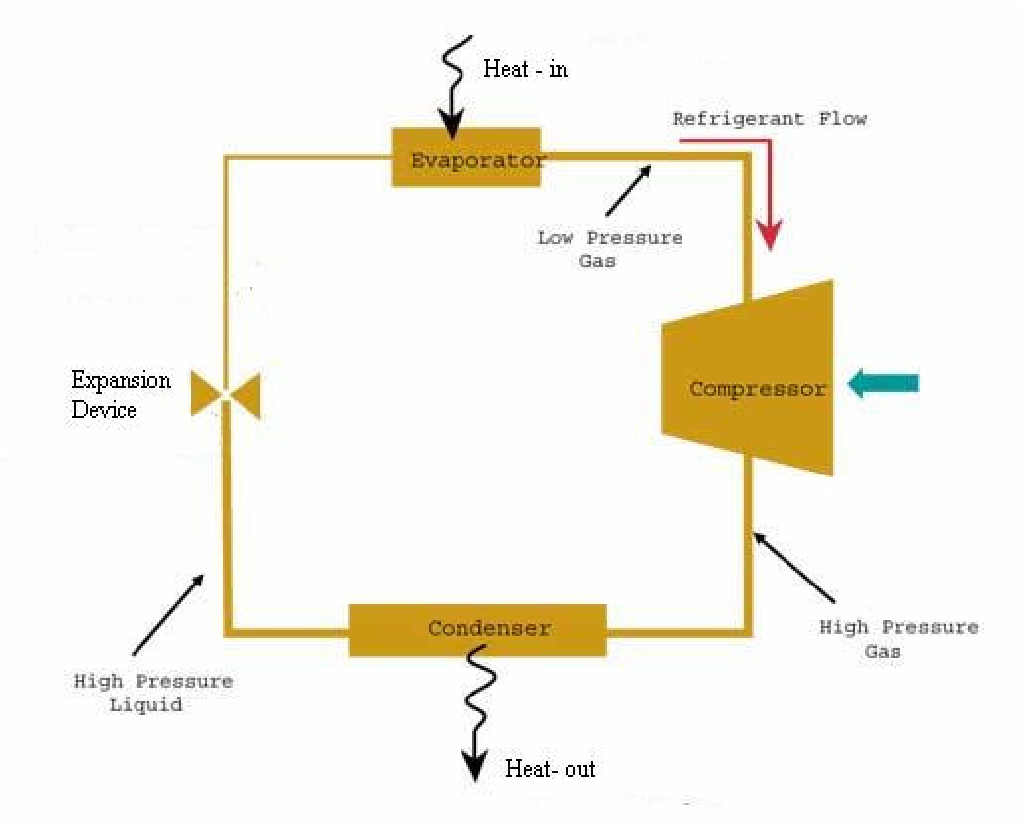

Figure 2-1: ***Schematic Diagram of Vapor Compression System***

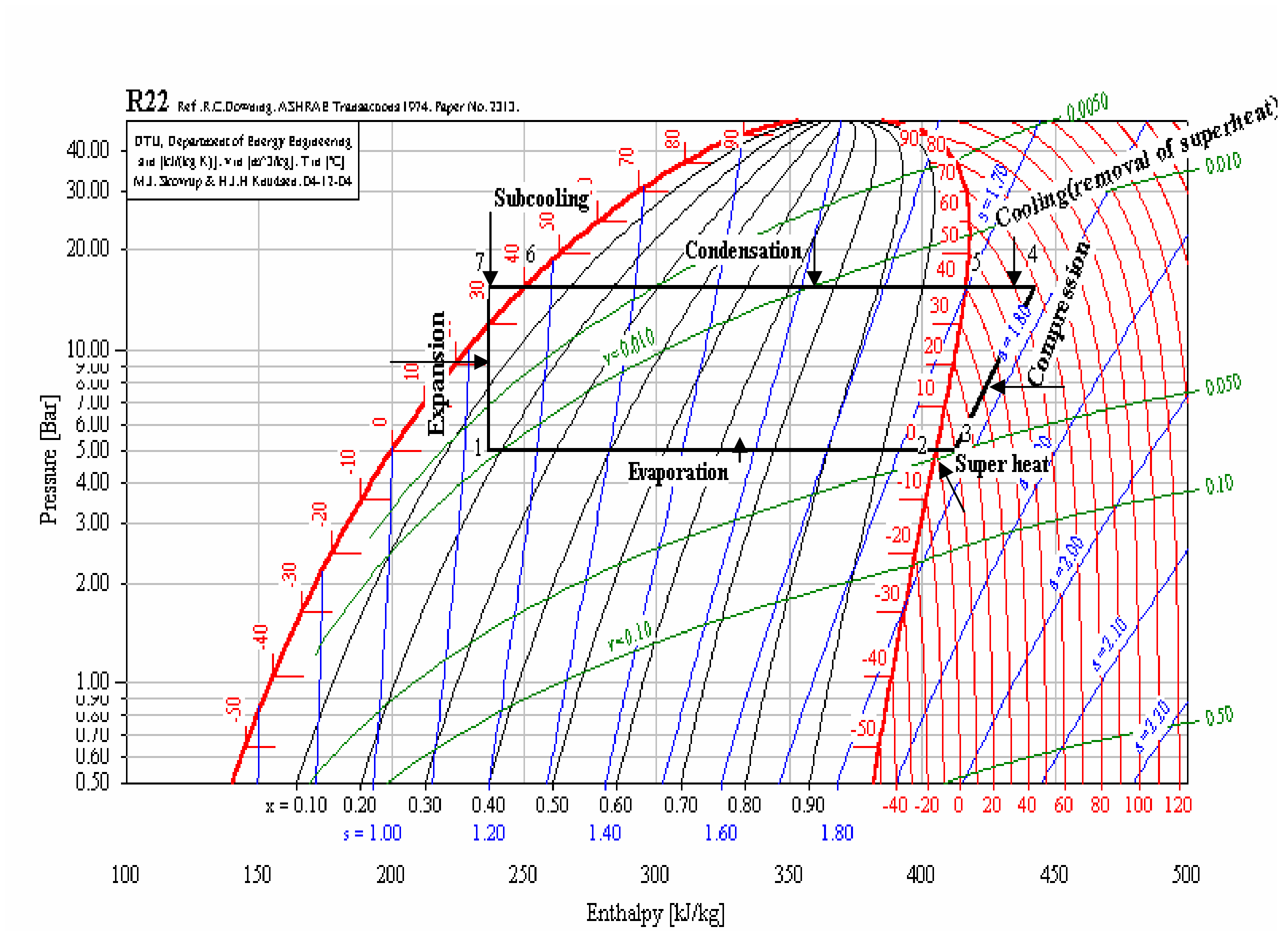

Figure 2-2: Pressure – Enthalpy Diagram for Vapour Compression System

The vapor compression system comprises the following basic steps (fig. 2.1 & 2.2):

1. Step 1-2-3: Absorption of heat by the liquid refrigerant and conversion to gas in the evaporator. Refrigerant is a substance with low boiling point at a desired working pressure. The surface of the evaporator is in contact with air, water or any other fluid or substance that it may be cooling. (From 1 to 2, liquid refrigerant absorbs latent heat energy, at constant pressure and temperature, converting itself to saturated vapour without increase in temperature; from 2 to 3, it absorbs additional heat at constant pressure to become superheated vapor with some increase in temperature. Superheating is necessary to prevent liquid refrigerant from entering the compressor).
2. Step 3-4: Compression of low temperature, low pressure refrigerant gas from the evaporator to high temperature, high pressure gas in the compressor. (From 3 to 4, reduction in vapour volume during compression, with increase in pressure; the joule equivalent of work done during compression, increases the temperature of the vapour to level higher than ambient).
3. Step 4-5-6-7: Rejection of heat to the in the condenser, resulting in condensation of the gaseous refrigerant to liquid at high pressure. The condenser surface is cooled by moving air or water. (From 4 to 5, vapour temperature drops when superheat is removed and the vapour is saturated; from 5 to 6, saturated vapour condenses back to liquid phase at constant temperature and pressure; from 6-7, the liquid is sub-cooled below its saturation temperature).

4. Step 7-1: Expansion of the liquid refrigerant from high condenser pressure to low evaporator pressure through a throttling valve, called expansion device or valve. The opening of the expansion valve is controlled to enable capacity regulation. (There is no heat addition or removal during this process, but some flash gas is formed as a very small portion of the liquid refrigerant evaporates between the expansion valve and the evaporator. To reduce flash gas formation, condensers are sometimes giving more heat transfer area to sub-cool the refrigerant or a very small quantity of refrigerant is separately expanded in a sub-cooler to cool the liquid refrigerant before the expansion valve.)

The heat transfer rate in the evaporator influences the refrigerant suction temperature and pressure. The ambient conditions and heat transfer rate in the condenser influence the discharge temperature and pressure. The suction and discharge pressures, that is, the compression ratio basically decides the extent of work to be done by the compressor and hence the energy consumption in the compressor.

Since the compressor suction and discharge pressures are governed by the heat transfer in the evaporator and condenser, the heat transfer efficacy of these heat exchangers and the regulation of refrigerant flow through the system by the expansion valve and the performance of the compressor at partial loads plays a major role in deciding the overall operating efficiency of the vapour compression system.

2.3.2 Refrigerants: Ozone Depletion and Global Warming

Refrigerants are substances with low boiling points and large latent heats, at pressures above atmospheric pressure. Refrigerants usually fall into one of the following groups:

- CFCs – chlorofluorocarbons;
- HCHCs – hydro chlorofluorocarbons;
- HFCs – hydro fluorocarbons;
- HCs – hydrocarbons;
- NH_3 – ammonia.

CFCs deplete stratospheric ozone and, following the Montreal Protocol, are no longer produced. HCFCs also deplete ozone, but to a lesser extent, and will be phased out in Europe by 2015. Small quantities are manufactured under strict regulation for few older systems. Developing countries have been permitted more time for the transition. Hence new refrigeration systems with HCFCs are still sold in developing countries. HFCs have been developed in the 1990s to replace CFCs and HCFCs. HCs are also being used as replacements.

Refrigerants consume energy during their manufacture and refrigeration plants also contribute to global warming as they consume electricity and other fuels to operate. Combustion of fuels in power stations or refrigeration equipments result in emission of carbon dioxide, which is a greenhouse gas that contributes directly to global warming. The total direct and indirect global warming impact of refrigerants is called its Total Equivalent Warming Impact (TEWI) value.

Table 2-1: Summary of Status of Some Refrigerant Groups

Type	Examples	ODP*	GWP**	Uses	Other Issues
CFC	R12 R502 R11	High	High	Widely used in most applications until 1990.	Now phased out of production
HCFC	R22 R409A R411B	Low	High	Widely used in many applications. Not recommended for us after 1999.	To be phased out of production in 2015. Their use is also regulated increasingly strictly.
NH_3 Ammonia	R717	Zero	Very low	Used in industrial systems since the birth of refrigeration.	Toxic and flammable, reacts with copper.
HFC	R134a R404A R407C R410C R507	Zero	High	Started to be used in place of CFCs from about 1990.	Different compressor oil needed performance of some HFCs not as good as CFCs. Some reliability problems.
HC E.G. propane, iso-butane, iso-pentane	R600a R290 Care 30 Care 50 R1270	Zero	Very low	R290 used in some industrial systems for decades. R600a now used in domestic systems. Care 30 and Care 50 now used in some commercial systems.	Flammable, but are very good refrigerants with few changes needed to a CFC/HCFC system.
CO_2		Zero	Very low	Widely used before the 1950s but superseded by halocarbons. Now being 'rediscovered' as a primary and secondary refrigerant.	Not yet widespread commercial use as a primary refrigerant, but an interesting prospect. (High operating pressures require special materials and construction.)

*ODP – Ozone Depleting Potential; **GWP – Global Warming Potential

Many of the new refrigerants are blends of different substances, which fall into two categories:

- Those with a low ODP which are used as transitional substances, these are usually blends based on HCFC R22 (e.g. R409A, R411B);
- Those with zero ODP, which have a longer term future, these are usually based on HFCs (e.g. R404A, R407C), or HCs (e.g. Care 30, Care 50).

The transitional substances are primarily used to convert existing systems. They should not be specified for new equipment. As they include R22, they can operate with the existing (mineral) oil in the system and the conversion procedure is relatively simple.

The blends based on HCFCs or HCs can be used to replace the HCFC or CFC refrigerant in existing systems without changing the oil. Blends based on HFCs are usually only used in new systems. The blends are mostly of two or three substances and are classified as Zeotropic blends or Azeotropic blends.

<u>Zeotropic Blends</u>
Zeotropic blends behave differently in a system compared to a single substance. While a single substance has a single evaporating/condensing temperature, zeotropic blends evaporate and condense across a small temperature range, and are said to have a 'temperature glide'. The temperature glide is the difference between the 'bubble temperature' (when the refrigerant is saturated and about to evaporate and the 'dew

temperature' (when the saturated vapour is about to condense). This property may lead to increased refrigeration capacity when the refrigerant and the cooled fluid are in counter flow, may lead to uneven ice build up leading to problems in defrosting and may be difficult to use in a flooded system. If required, the refrigerant will also have to be removed as a liquid from the system to prevent change in the composition of the remaining refrigerant in the system.

Azeotropic Blends
Blends that do not have a temperature glide are called azeotropic blends and they behave like a single substance. e.g. R502 and R507.

Hydrocarbons (HCs)
HCs are inflammable and safety is a serious issue. These refrigerants are being used in domestic refrigerators. However, in India, care is required from the view point of safety because a major portion of the refrigerator repair market is in the unorganized sector.

Ammonia (R717 or NH_3)
Anhydrous ammonia is popularly used as a refrigerant in industrial systems, especially chemical and dairy industries. Ammonia has high latent heat and hence less mass flow is required, resulting in slight reduction in the power consumption. Lubricating oil, being denser than ammonia, can be easily drained from the lower points of the system. Air, being heavier than ammonia, can be purged easily from the highest point in the system. All these properties contribute to the design of ammonia systems that are slightly more efficient operationally.

Ammonia is toxic in high concentrations. The occupational exposure limit is 25 ppm when the smell is quite obvious. Most people can tolerate up to about 250 ppm, with some irritation and discomfort. At 3500 ppm, ammonia is quite lethal. A 16% to 27% ammonia-air mixture can be ignited with some difficulty. Ammonia also attacks copper, zinc, tin, calcium and their alloys.

- **The refrigerant type can affect the efficiency of the system by about 10%.**
- **Relative performance of a refrigerant is affected by the type of compressor and the operating conditions.**
- **Zeotropic blends can give advantage in capacity and efficiency when used correctly i.e. advantage is taken of the temperature glide in the evaporator.**
- **Too much or too little charge of refrigerant can reduce efficiency.**
- **Insufficient refrigerant reduces the wetted area in the evaporator, increases the superheat, reduces the suction pressure, increases the temperature lift and reduces the efficiency.**
- **Refrigerant, contaminated with air or other gases, will affect the efficiency of the system.**

2.3.3 Types of Compressor & Capacity Control

The commonly used compressors for most vapour compression systems are *reciprocating, screw, scroll* and *centrifugal*. For very small applications like window air conditioners and split air conditioners, small *rotary roller or sliding vane or reciprocating* compressors are used.

The compressors are constructed as *Open, Hermetically Sealed* and *Semi-Hermetic* (semi-bolted) compressors. In the *open type* compressor, the shaft extends out of the compressor and is connected externally to the prime mover (electric motor or engine). In the *hermetically sealed* motor-compressor unit, the entire assembly is encapsulated, only the refrigerant lines and electrical connections extend out of the housing. In *semi-hermetic* compressors, while the motor and compressor are encapsulated, the heads of the compressors can be removed to gain access to the pistons and valves for servicing. The motor is also accessible for repair by removing the bolted plate.

In both hermetic and semi-hermetic compressors, the refrigerant is in contact with the motor windings. So only halocarbon refrigerants, which do not attack copper, can only be used; ammonia cannot be used. Presently, ammonia compressors are always of the open type.

Open type compressors are usually more efficient than the *hermetic and semi-hermetic types* because the suction vapour in a hermetic compressor passes over the motor to cool it, resulting in super heating of the vapour and thereby requiring more power for compression. However, proper refrigeration system design can minimize the impact on energy consumption.

The *rotary* compressors are usually suitable for small, fraction tonnage capacity machines like window air-conditioners. These compressors are roller*, vane* or *reciprocating* type. In the roller type compressor (fig. 2.3), the roller is eccentrically mounted in a cylindrical space having spring loaded separator. The low pressure vapour enters, gets compressed and finally discharged to the condenser. *Roller* type compressors are manufactured up to 5 ton capacity. In the vane type compressor (fig. 2.4), a number of vanes are mounted in the slots of an eccentrically mounted rotor. The position of vanes changes as the rotor rotates. The compression ratio is limited to about 7:1. The capacity control is by cycling the compressor on and off.

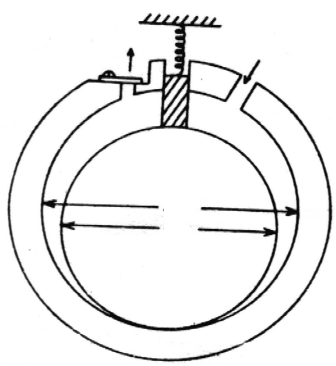

Figure 2-3: Roller Compressor

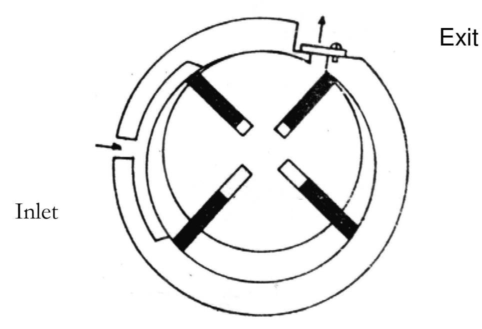

Figure 2-4: Rotary Sliding Vane Compressor

Figure 2-5: Reciprocating Compressor

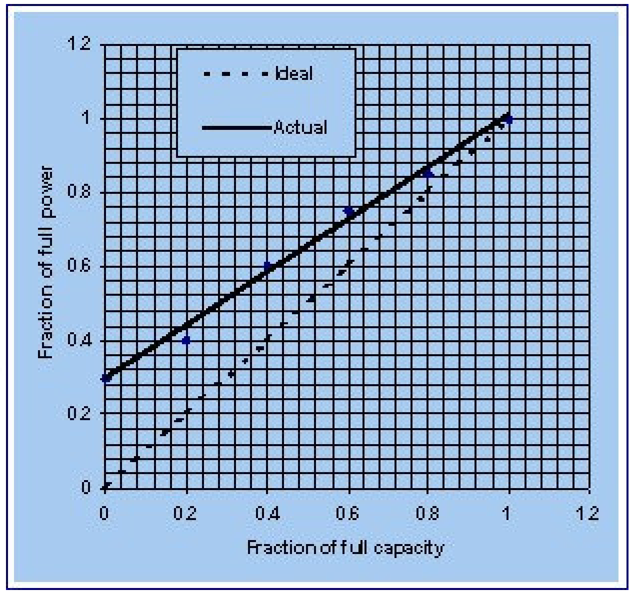

Figure 2-6: Recips - Power at Part Load Operation

In a *reciprocating* compressor, the refrigerant vapour is compressed by movement of the piston in a cylinder (fig. 2.5). Reciprocating compressors are commonly used up to single machine capacities of 250 TR. In the case of small machines, the arrangement may be cyclic on/off control based on temperature sensing. However, in larger machines, as frequent starts and stops of motors are not permitted, other methods of capacity control are adopted. In reciprocating compressors with multiple cylinders, cylinders are selectively loaded or unloaded, based on set pressures (reflecting the temperatures); the variation in power with cylinder unloading is shown in fig. 2.6. Unloading implies that the suction valve is kept open so that the vapour, taken in during the suction stroke, returns back through the suction valves itself during the discharge stroke. Reciprocating compressors cannot tolerate liquid slugging, which can happen when the evaporator load is less and the superheat controlled expansion valve is unable to regulate the flow of refrigerant correctly, resulting in excess liquid refrigerant entering the evaporator and getting sucked into the compressor. Liquid slugging can cause serious damage to the compressor.

In a *screw* compressor (fig.2.7), there is male rotor with lobes and female rotor with gullies. As both the rotors rotate in opposite directions, the gas gets drawn in, gets sealed between the rotors and the housing, gets compressed as the cavity bears against the end of the housing and, finally, as the screw thread reaches the discharge port, the compressed gas flows into the discharge line. The compression ratio in a single stage can go up to 25:1, which is significantly higher than the pressure ratios of reciprocating compressors. *Screw* compressors are available for refrigeration capacities from about 10 TR to 1200 TR, but commonly used in the 100 to 300 TR range. For capacity control, sliding valve control is used to bypass some gas back to the suction (depending on the position of the sliding valve) and hence reduce the volumetric efficiency of the compressor. The variation in power consumption and cooling capacity for a screw compressor is shown in fig. 2.8. Due to internal losses, capacity control by sliding valve below 60% capacity is not very efficient. However, screw compressors can tolerate liquid slugging.

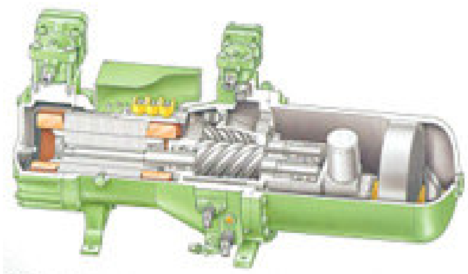

Figure 2-7: Screw Compressor

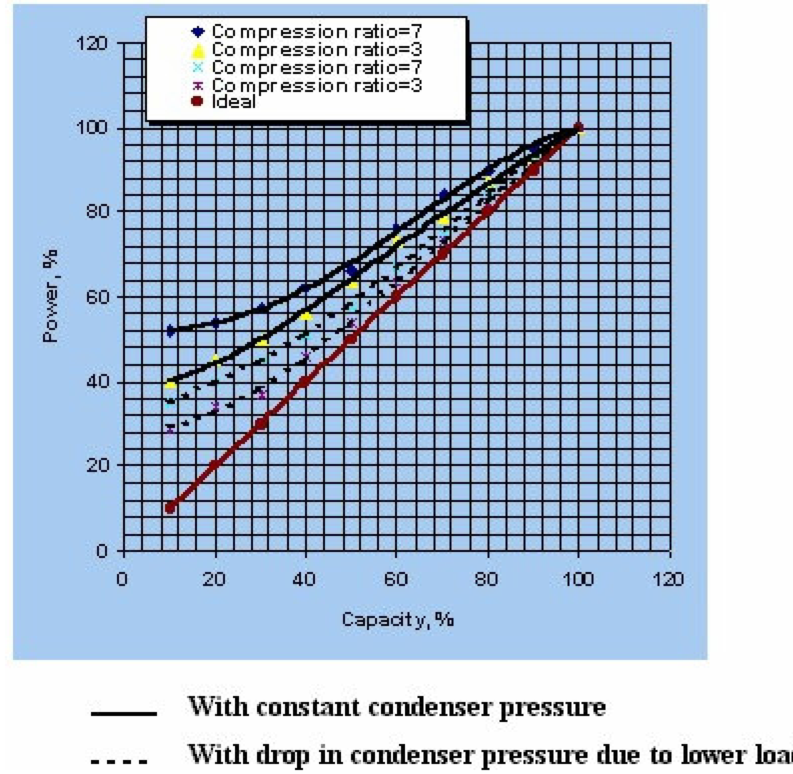

Figure 2-8: Screw - Power at Part Loads

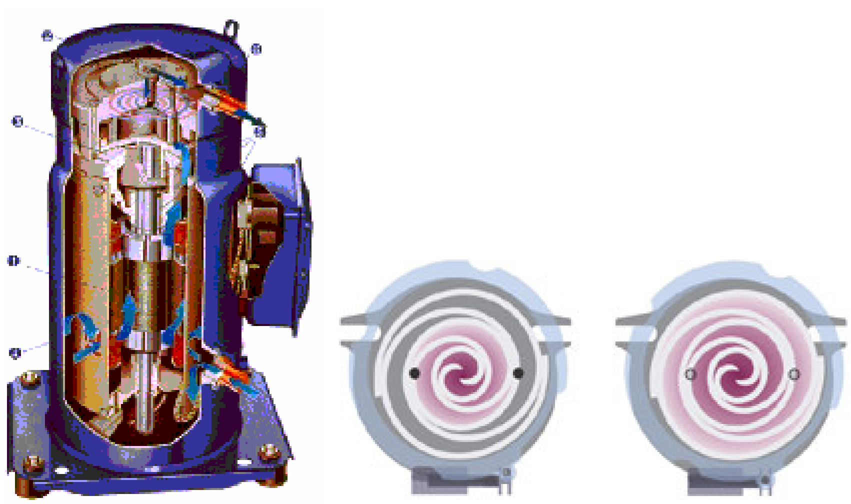

Figure 2-9: Scroll Compressor

In the *Scroll* compressor (fig. 2.9), the main components are two involute scrolls that intermesh. The top scroll, which contains the gas discharge port, is fixed and the bottom scroll orbits. The two scrolls are maintained with a fixed annular phase relation (180°) by an anti-rotation device. As the bottom scroll orbits within the other, crescent shape gas pockets are formed, their volumes are reduced until they vanish at the centre of the scroll. Suction, compression and discharge are simultaneously performed in a continuous sequence by the orbiting motion of the scroll. Scroll compressors are available for capacities up to 30 TR; multiple compressors are used to build larger packages. The capacity control of Scroll compressors is by cyclic on/ off control. Some new Scroll designs have the facility to operate at 100% and 67%. Scroll compressors can tolerate some liquid slugging and particulate contamination. Scrolls can tolerate high discharge temperatures and pressures.

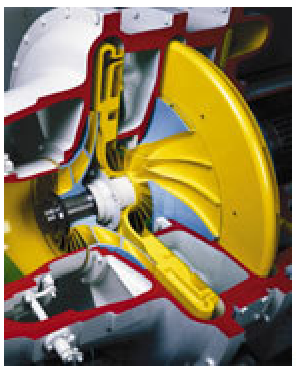

Figure 2-10: Centrifugal Compressor

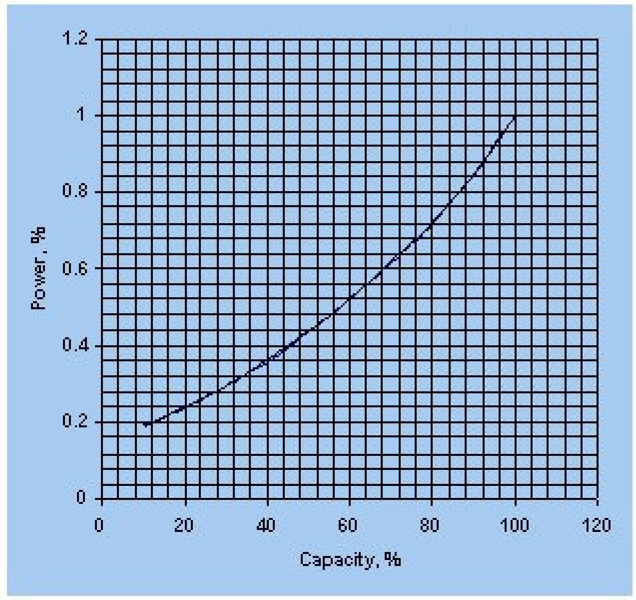

Figure 2-11: Centrifugals – Power at Part Loads

In the *Centrifugal compressor* (fig 2.10), large volumes of vapour are centrifugally accelerated to high velocity and this velocity energy is converted to pressure. The compression ratio is around 5:1 at 3600 rpm. The speed of the centrifugal compressor goes very high at around 20,000 rpm. These machines are manufactured in the range of 35 to 10,000 ton capacity. Centrifugals are generally used beyond 150 ton single machine capacity. In centrifugal compressors, capacity control is ususally by inlet guide vanes at the suction. Capacity control in the range of 10% to 100% is possible; the variation of power with capacity is shown in fig. 2.11.

The partial load operation of all compressors is inefficient compared to full load operation as the power does not drop in proportion to the drop in capacity. Capacity control for all types of compressors can be done very efficiently by varying the speed of the compressor. The permitted range of speed variation will vary depending on the compressor type and the lubrication arrangement.

- **The efficiencies of compressors of different types and makes may be different. So accurate comparisons are required at the time of selection. Compressor efficiency at part loads is key issue to be considered.**
- **It is desirable to avoid the use of single large compressor where the refrigeration load is variable. Multiple smaller machines operating close to full load are more desirable.**
- **Compressors with variable speed drives always result in better efficiency at part loads.**

2.3.4 Evaporators

The evaporator is the heat exchanger where the heat is removed from the system by the boiling of the refrigerant in the evaporator. The heat may be removed from air, water or any other process coolant. The evaporator may be refrigerant cooled coils in an air stream (Air Handling Unit – AHU) (fig. 2.12) for air conditioning, shell and tube heat exchangers (fig. 2.13) with refrigerant in the tube or shell sides, plate heat exchangers (fig. 2.14) or refrigerant coils submerged in water or brine tanks.

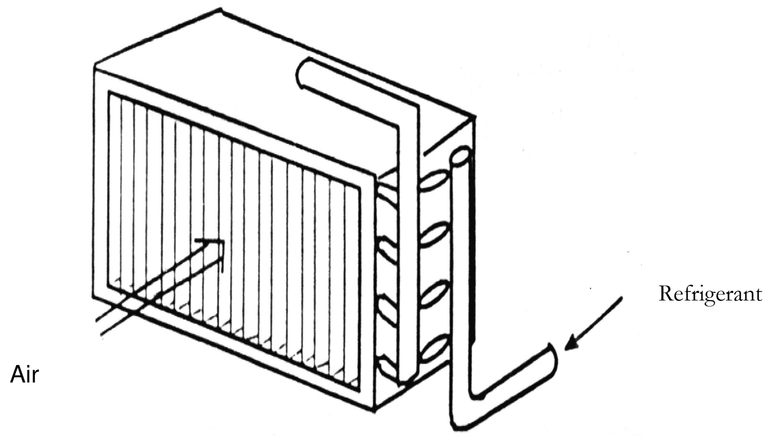

Figure 2-12: Air Handling Unit for Cooling Air

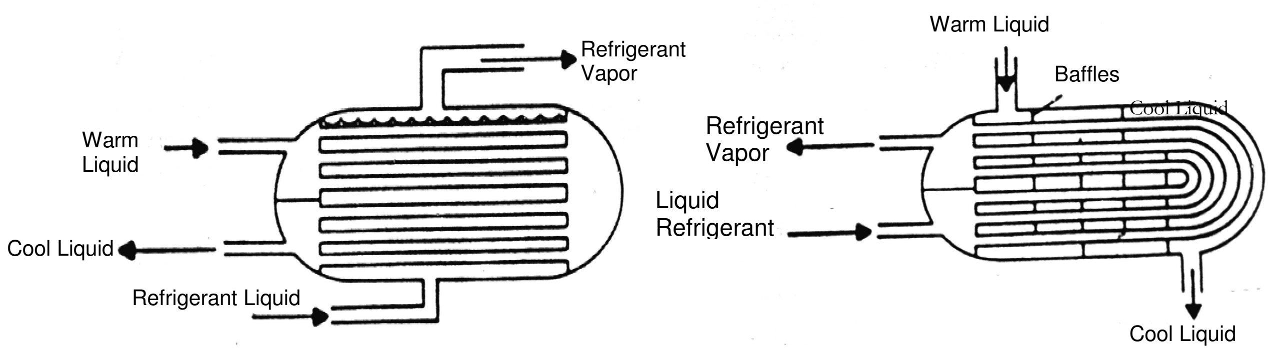

Figure 2-13: Shell & Tube Heat Exchangers

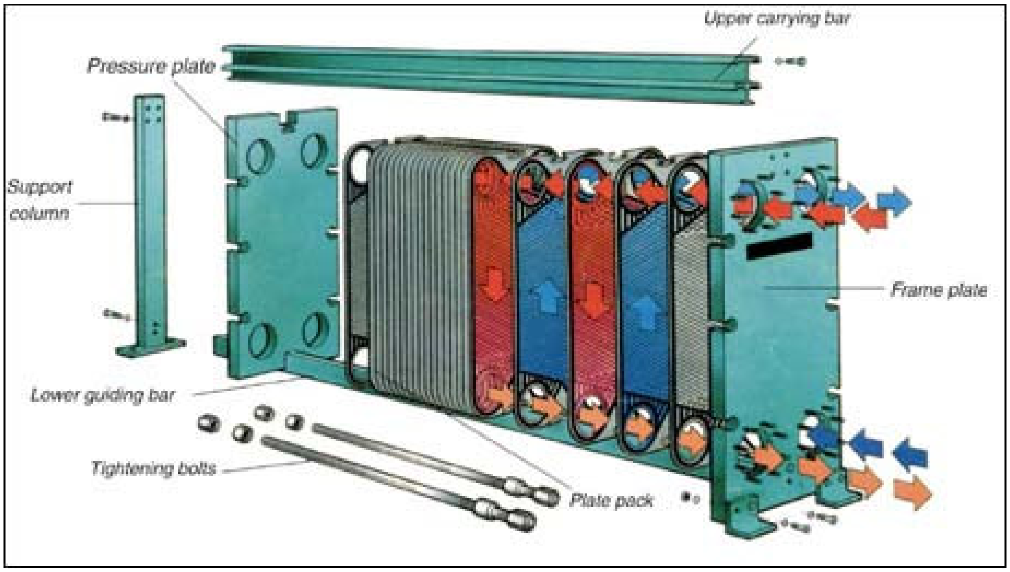

Figure 2-14: Plate Heat Exchanger

The coils are generally of two types: (a) Direct Expansion (DX) type, wherein regulated quantity of refrigerant liquid is allowed to enter through the expansion valve and the vapour get sucked by the compressor or (b) Flooded type, wherein the entire shell or large coil is flooded with refrigerant liquid by gravity from a surge drum or by pumping, the vapour from the boiling liquid move back into the surge drum and then get sucked by the compressor. The surge drum gets liquid refrigerant from a liquid receiver due to pressure difference. In flooded evaporators, the entire heat transfer surface is wetted with liquid refrigerant. Flooded evaporators tend to accumulate lubricating oil and suitable arrangements are provided to drain the oil.

Evaporators are designed to with parallel tubes to maximize heat transfer area, minimize pressure drop and also ensure adequate refrigerant velocity to facilitate return of lubricating oil back to the compressor. Formation of oil film on heat transfer surfaces can significantly reduce the heat transfer rate.

Baudelot cooler is used when water is required very close to freezing temperature, say Baudelot cooler, which comprises vertical tubes or plates, with water tricking down in film form on one side and boiling refrigerant on the other side. The water collects in a tank below.

The heat transfer in the evaporator depends on the surface area, the fluids involved, the turbulence in the fluid streams and the operating temperature and pressure. The heat transfer coefficients may range from 1500 to 5000 W/m^2°K, depending on the evaporator design and refrigerants used.

The benefits of Plate Heat Exchangers (PHEs) over conventional shell and tube heat exchangers are smaller refrigerant volumes (5 to 20%) due to low internal holding volume, easy access for maintenance and inspection, easy augmentation of heat transfer area by addition of plates, high heat transfer coefficients resulting from the intense turbulence in the plate channels and reduced fouling tendencies and higher reliability. PHEs can be designed to have closer approach temperatures i.e. higher evaporating temperature (for same end-use temperature) to achieve higher COP. PHEs have been in use for secondary heat transfer to products (especially in food and pharmaceutical industries) for a long time. More recently, PHEs have been also used as evaporators and condensers in refrigeration systems in place of shell and tube heat exchangers.

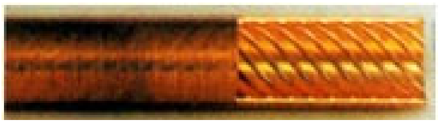

Figure 2-15: Grooved Tubes to Increase Surface Area

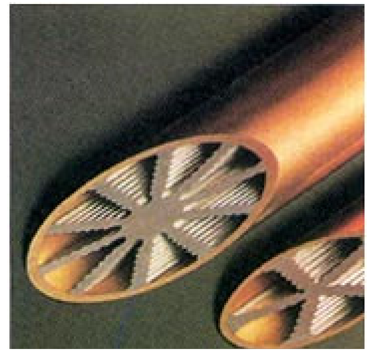

Figure 2-16: Tube Inserts to Increase Turbulence

Larger heat transfer area in the evaporator leads to better heat transfer from the process coolant to the refrigerant, resulting in higher refrigerant temperature and pressure and less energy consumption. Heat transfer area can be increased in heat exchangers by providing more tubes or larger number of smaller tubes or by surface area enhancement techniques like use of grooved tubes (fig.2.15), finned tubes, helical wire inserts in tubes (fig.2.16) etc. Within certain limits, the coefficient of heat transfer can be increased by operation at higher process coolant velocities. Trade offs have to be made depending on pressure drops, energy cost and material cost.

2.3.5 Condensers

The condenser is the heat exchanger where the refrigerant gas condenses, giving up its heat to the atmosphere. The condensers may be natural air draft cooled for small equipment like refrigerators, forced air draft cooled condensers, water cooled shell and tube condensers or water cooled plate heat exchangers or evaporative condensers. Some of the commonly use condensers are shown in fig. 2.17.

In the case of water-cooled condensers, the water is generally cooled in cooling towers or spray ponds. In some cases, evaporative condensers are used where water is sprayed on coils; the heat is directly dissipated to the atmosphere, avoiding the need for a separate cooling tower or spray pond.

The heat transfer coefficients range from 1400 to 11000 $W/m^2°K$, depending on the condenser design and refrigerant used. The issues relating to various types of heat exchangers in section 2.2.4 are also relevant to condensers.

Most refrigerants have low surface tension, which promotes the formation of a thin condensate film on the external profile of the tube. Heat transfer is an inverse function of condensate film thickness and therefore increases with thin condensate films. Hence, short, vertical fins on a horizontal tube will gave a smaller film thickness than on plain tube.

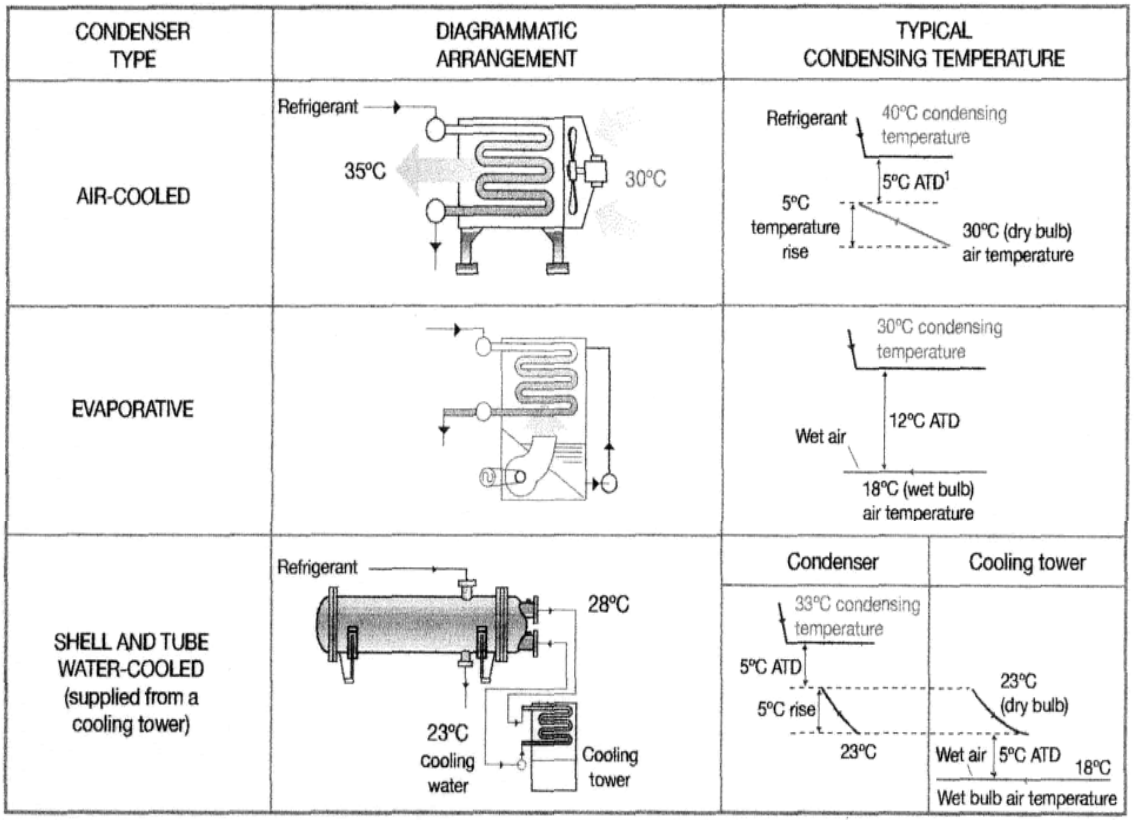

Figure 2-17: Some Commonly Used Condensers

Integral externally finned tubes, additional longitudinal or spiral grooves and ridges in finned tubes etc. create turbulence and augment water side heat transfer. These features promote turbulence and may also help retard the adverse effect of fouling. However, finned tubes may retain condensate, restricting its performance. Several enhanced surfaces with complex surface geometries have been developed to promote surface tension-drained condensation.

Air cooled condensers, being limited by the dry bulb temperature of air, generally result in higher refrigerant condensing temperature and pressure, compared to water cooled condensers, leading to higher compression ratios and higher power consumption (about 20% or higher) in compressors. However, water scarcity and absence of water treatment facilities is forcing many users to prefer air cooled condensers. In such cases, providing evaporative cooling pads before the condenser can help reduce the condensing temperature and pressure when the weather is dry i.e. low relative humidity conditions. Air cooled condensers facilitate the use of warm air for space heating in during cold weather.

- Condenser performance improves when the coolant (air or water) temperature is low.
- Evaporative condensers are more efficient than shell and tube condensers. However, these are difficult to repair in case of tube puncture. Evaporative condensers are also more prone to corrosion due to the constant presence of air around the tubes.

- **The overall heat transfer coefficient of a heat exchanger is determined by the designer and the application engineer has no control over it.**

- **For the same duty conditions, the heat exchanger with more primary tube surface area will have higher efficacy, all other parameters being equal.**

2.3.6 Expansion Valves

The expansion device is a refrigerant flow control device that helps match the refrigerant flow rate with the *boil off rate* in the evaporator, in the process reducing the pressure of the refrigerant from the high condensing pressure to the low evaporator pressure. To utilize the coil more effectively, liquid refrigerant should be present right up to the end of the coil, instead of getting superheated over some area toward the end section of the coil. However, superheat section is a trade-off, to protect the compressor from the possibility of liquid entry to the compressor.

The high pressure, liquid refrigerant is passed through the expansion device into the evaporator, where the refrigerant vaporizes, due to the heat from the process coolant or material being cooled. The expansion device may be a fixed orifice or capillary tube, manually controlled valve, thermostatic, electronic or balanced port valve, float valve or superheat controlled expansion valve.

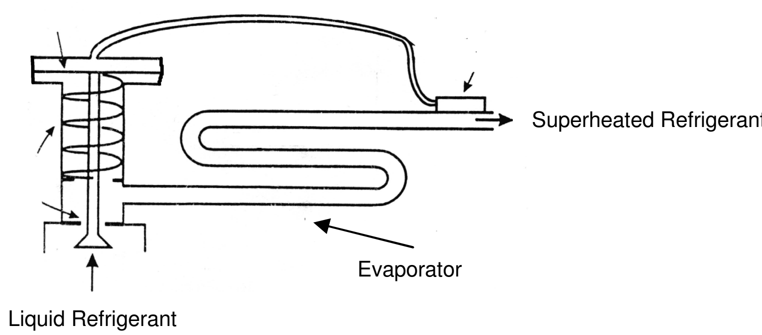

Figure 2-18: Schematic Diagram of Superheat Sensing Expansion Valve

Orifice plates and capillary tubes are used only for small machines like window air conditioners, refrigerators etc. where the capacity control is by cycling the compressor on and off and no refrigerant flow control is required.

Thermostatic valves have an orifice to provide a pressure drop and a needle valve and diaphragm to regulate the flow of refrigerant based on the evaporator outlet refrigerant vapour temperature, which is kept about 5ºK above the evaporating temperature, to prevent liquid entering the compressor.

Balanced port valves are similar to thermostatic valves, apart from a special internal balanced port design, which enable better flow control.

Electronic valves are also similar to thermostatic valves with the difference that the temperature is sensed electronically and the orifice opening is changed either by heated fluid pressure or by electrically actuated valve. Electronic valves can be easily integrated into an electronic microprocessor based control system.

The superheat sensing valve (fig. 2.18) is the most commonly used refrigerant flow control device. This valve controls the refrigerant superheat leaving the evaporator. The valve stem is positioned by the pressure difference on opposite sides of the diaphragm. The pressure under the diaphragm is provided by the refrigerant at the inlet to the evaporator and the pressure on the top of the diaphragm by a power fluid, which is the same refrigerant used in the system. A slight force exerted by the spring on the valve stem keeps the valve closed until the pressure above the diaphragm overcomes the combined forces of the spring and the evaporator pressure. For the pressure above the diaphragm to be higher than the evaporator pressure below the diaphragm, the power fluid temperature must be higher than the saturation temperature in the evaporator. The refrigerant gas at the outlet of the evaporator, therefore, must be superheated to bring the power fluid up to the valve opening pressure.

The operation of the superheat controlled expansion valve can be sluggish at partial loads i.e. when the compression ratio is low. The electronic expansion valve, which can work on very low superheat settings allows more coil length to be used for liquid refrigerant, compared to a conventional expansion valve, and thus improves coil performance. A novel expansion valve controller has been developed in India to sense the process coolant (air, water, brine etc.) temperature and heat or cool the sensing bulb faster to make the operation of the superheat sensing expansion valve more sensitive at partial loads. Use of these devices has led to better temperature stability and energy savings.

Float valves are used in flooded systems where the liquid refrigerant level in the surge drum is sensed and maintained. Alternatively, manually adjusted valves with a level switch are also provided for flooded evaporators.

2.4 Sub-cooling

Sub-cooling of the liquid refrigerant before the expansion valve reduces the flash gas formation and helps increase the evaporator capacity. Sub-cooling can be provided by providing more heat transfer area in the condenser or providing a compressor suction gas to refrigerant liquid heat exchanger or flashing of minuscule quantity of liquid refrigerant to cool the liquid refrigerant. In the case of flooded evaporators, location of the receiver in a cool location can help achieve some free sub-cooling. The COP can improve by 5% to 10% by sub-cooling.

2.5 Superheating

Superheat is the increase in temperature of refrigerant gas above the evaporating temperature. The higher the suction gas superheat, lower the gas density and therefore, lower the compressor mass flow rate. This reduces the compressor capacity without reducing its power consumption, increasing running costs.

Superheat may be picked up while doing useful cooling or in the suction piping from the evaporator to the compressor. To some extent, superheat, is a necessary evil to prevent liquid refrigerant from entering the compressor. The advent of electronic expansion valves has helped minimize superheat without the danger of liquid slugging. Suction lines should always be insulated to minimize superheat.

2.5.1 Secondary Coolants

In air conditioning air may be directly cooled in the heat exchanger (air handling unit) or through a secondary coolant. In most industrial applications, the final material to be cooled is generally cooled through an intermediate process coolant like water or water with an anti-freeze agent (generally termed as brines). The concentration of the anti-freeze agent affects the freezing point and also the heat transfer rate. Fig 2.19 shows the variation of freezing points with the concentration of the anti-freeze agents of some commonly used anti-freeze solutions.

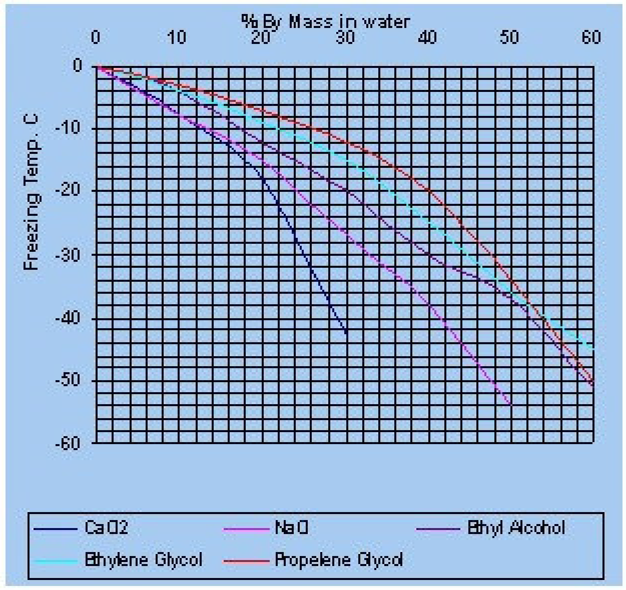

Figure 2-19: Anti-freeze Solutions – Change in Freezing Point with Concentration

2.5.2 Specific Power Consumption of Vapour Compression Systems

The efficiency of refrigeration systems depends on the operating temperature and hence all figures of merit related to efficiency have to specified at a certain operating temperature.

Well designed, well maintained, water cooled vapour compression systems, using reciprocating, screw or scroll compressors, for chilled water at 6°C to 8°C have COP of 4 to 5.8, EER in the range of 14 to 20 Btu/hr/W or Specific Power Consumption in the range of 0.61 to 0.87 kW/TR.

Open-type compressors are more efficient than semi-hermetic compressors; in semi-hermetic compressors, the motor is also cooled by the refrigerant.

Centrifugal compressors, which are generally used for cooling loads about 150 TR, can have COP of about 6, EER greater than 20 and Specific Power Consumption of 0.59 kW/TR.

The COP of systems with air cooled condensers is generally about 20% to 40% higher. In areas with dry weather, the COP improves significantly, if evaporatively cooled saturated air is provided (using cooling pad and booster fan) instead of ambient dry air.

The COP of well designed small machines like window air conditioners and split air conditioning units are generally in the vicinity of 2.5. The COP is poor due the compactness of the machine, which limits heat transfer area, and the fact that the condenser is air cooled.

It may be noted that COP is calculated taking into account only the compressor power consumption. The power consumption of other parasitic loads like pumps, fans etc. are not considered.

COP is affected by the evaporator temperature and the condenser temperature. Higher the evaporator temperature and lower the condenser temperature, better is the COP.

1ºC higher temperature of refrigerant in the evaporator or 1ºC lower in the condenser improves the COP by 2% to 4%.

The evaporator temperature can be increased by:

- **Changing process temperature settings.**
- **Installing an evaporator of higher rating i.e. more heat transfer area.**
- **Keeping the heat transfer surface clean i.e. avoiding fouling, defrosting as per requirement etc.**

The condenser temperature can be decrease by:

- **Installing a condenser of higher rating i.e. more heat transfer area.**
- **The condenser temperature is allowed to float down with ambient temperature**
- **Water cooled or evaporatively cooled condensers are used instead of air cooled condensers**

COP is also affected by:

- **The efficiency of the type of compressor used.**
- **The amount of refrigerant charged in the system. Systems with refrigerant leaks consume more power.**
- **The physical properties of refrigerant used.**

COSP (Coefficient of System Performance) is another parameter, which has been evolved to estimate the system performance by also including the power consumption of parasitic loads like fans, pumps etc.. This is a good parameter to bench mark the overall system efficiency at a particular site but it does facilitate comparison with other similar installations as system configurations vary from site to site.

2.6 Vapour Absorption Refrigeration System

The *Vapour Absorption Systems or Absorption Chillers* use heat source to produce cooling effect. There is no compression of refrigerant vapour in this system. Absorption chillers are used in locations with access to cheap heat energy source or process waste heat. Absorption chillers have few moving parts, which means less noise and vibrations, an important issue in commercial buildings.

Vapour absorption process requires two substances with strong chemical affinity at low temperatures. The commercially available absorption chillers use the following combinations:

a) Lithium Bromide (absorbent) and water (refrigerant) for chilled water at 5ºC and above. Newer versions provide slightly lower temperatures.
b) Water (absorbent) and Ammonia (refrigerant) for temperatures below 5ºC. A secondary coolant (brine) is required to transfer heat to ammonia.

Absorption Chillers may be *single effect* or *multiple effect* machines; the COP of multiple effect machines is higher. In India, single effect and double effect machines are being manufactured. Triple effect absorption machines are available from international manufacturers.

Most of the absorption chillers in use in India are for chilled water using LiBr and water. Single machine capacities in excess of 1200 TR are available. Both steam heated and direct fired models are being widely used. A small number of chillers, using ammonia and water, are also in operation. Ammonia-water absorption chillers are single effect machines, hence their COP is poor.

2.6.1 Operating Principle

Single-Effect Chiller

Fig. 2.20 shows a Single Effect Chiller. The evaporator contains a bundle of tubes that carry the system water to be cooled/chilled. This is the part of the chiller where cooling of the system's water occurs. The cooling cycle begins when high-pressure liquid refrigerant from the condenser passes through an orifice into the lower-pressure evaporator, and collects in the evaporator pan or sump. This step of the cycle causes "flashing" of some of the refrigerant at the entrance to the evaporator and cools the remaining refrigerant liquid that is sprayed over the tubes carrying the system water. Transfer of heat from the comparatively warm system water to the cooled refrigerant causes the latter to evaporate and cool the water in the chiller tubes.

The refrigerant vapors, generated in the evaporator, migrate to the lower-pressure absorber, where the vapors are absorbed by the lithium-bromide absorbent solution spayed over the absorber tube bundle. The absorber and evaporator share the same vapor space, allowing refrigerant vapors generated in the evaporator to migrate continuously to the absorber. The absorption process creates a lower pressure, drawing a continuous flow of refrigerant vapor from the evaporator to the absorber. In addition, the absorption process condenses the refrigerant vapors and releases the heat removed from the evaporator by the refrigerant. The heat released from the condensation of refrigerant vapors and their absorption in the solution is removed from the absorber by transferring it to the cooling coil, installed in the absorber, through which cooling tower water is circulated. The cooling tower water releases the heat removed from the absorber to the cooling tower installed at the facility.

The diluted lithium-bromide solution form the absorber is pumped continuously from the absorber to the generator, which operates at a higher pressure. In the generator, heat (from natural gas, steam, or hot water) is added to boil off the refrigerant vapors absorbed in the absorber. The concentrated lithium-bromide solution produced in the generator is returned to the absorber.

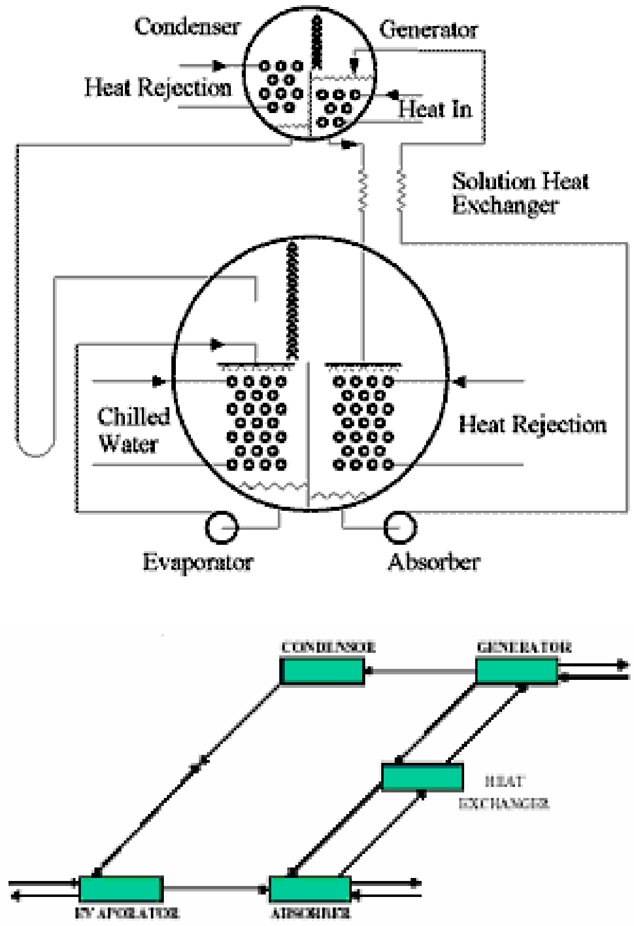

Figure 2-20: Schematic Diagram of Single Effect Absorption Chiller

The refrigerant vapors created in the generator migrate to the cooler condenser. The cooling tower water circulating through the condenser turns the refrigerant vapors to a liquid state and picks up the heat of condensation, which it rejects to the cooling tower. The liquid refrigerant returns to the evaporator and completes the cycle.

A heat exchanger is used to recover some of heat from the concentrated hot lithium-bromide solution going from the generator to the absorber to heat the dilute cold lithium-bromide solution going from the absorber to the generator.

Multiple-Effect Chiller

Double Effect Chiller

A double-effect chiller (fig. 2.21) is very similar to the single-effect chiller, except that it contains an additional generator, condenser, and heat exchanger that operate at higher temperatures than those for a single-effect chiller. The higher temperature generator is called the first stage generator, the higher temperature condenser is called the first stage condenser, and the higher temperature heat exchanger is called the high-temperature heat exchanger.

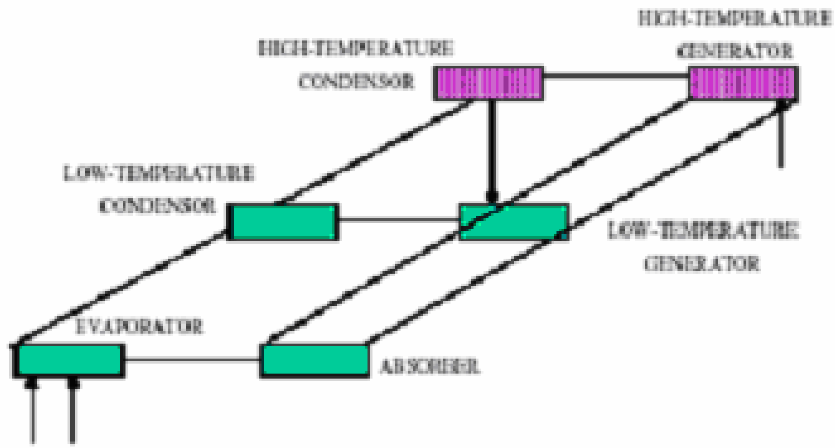

Figure 2-21: Schematic Diagram of Double Effect Absorption Chiller

The refrigerant vapors from the first-stage generator are condensed in the first stage condenser. The heat recovered from condensation in this condenser is used to boil off additional refrigerant vapors from the lower-temperature second stage generator. This additional refrigerant vapor generated increases the capacity of the evaporator for the same heat input, resulting in increased performance efficiency. The high-temperature heat exchanger is used for recovering heat from the concentrated lithium-bromide solution from the first-stage generator to heat the lithium-bromide solution form the second-stage generator.

<u>Triple Effect Chiller</u>

A recent development is the direct fired Triple Effect Absorption Chiller (fig. 2.22) wherein the one more middle level generator and condenser are added. This chiller is now available commercially from some international refrigeration majors.

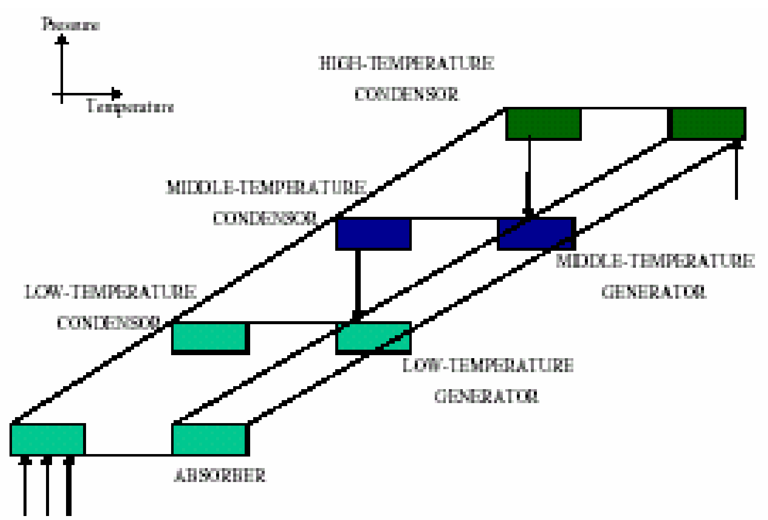

Figure 2-22: Schematic Diagram of Triple Effect Absorption Chiller

Lithium Bromide – Water Absorption Chiller

The LiBr-Water absorption system is commonly used for chilled water applications at 5ºC and above. Single effect absorption chillers can operate with hot water at 85ºC to 90ºC or low pressure steam at pressures up to 3 kg/cm^2g. Double effect chillers require a temperature of at least 140ºC or above, which is available from steam at 7 to 9 kg/cm^2g or pressurized hot water at about 160ºC or direct fuel firing. The evaporator-absorber is at a pressure of 6 mmHg and temperature of about 4ºC. The generator is at about 160ºC at full load; it reduces as part loads. The condensation of water in the condenser-generator takes place at a pressure of 75 mmHg and temperature of about 45ºC.

Water – Ammonia Absorption Chiller

Water – ammonia absorption single effect chillers can be used for applications requiring temperatures in the range of -40ºC to +5ºC. The preferred heat source temperature is 95ºC to 180ºC. Normal cooling water is used for cooling the absorber and condenser. For temperatures below -40ºC up to -60ºC, the cooling will have to be provided by chilled water at 10ºC to 15ºC. These chillers operate at moderate pressures and no vacuum is required till -30ºC.

<u>Other Issues</u>

Absorption chillers require vacuum pumps only for short duration while starting the machine; after that, equilibrium condition is maintained by physical and chemical phenomena.

Crystallisation of LiBr in the chiller was a problem earlier. But now improved controls have overcome this problem. The problem of tube failures due to Li Br corrosion has also been over come by the use of corrosion inhibitors and providing adequate allowance in selection of tube material and thickness.

2.7 Capacity Control

Capacity control is achieved by controlling the heat input rate chiller i.e. controlling the steam flow, fuel firing rate etc. This can be achieved by steam flow control or burner modulation. Absorption chillers can be turned down typically to about 25% capacity.

2.7.1 Specific Fuel Consumption Of Vapour Absorption Systems

Lithium Bromide – Water Absorption Chillers

Single Effect Absorption chillers at about 8°C have COP of about 0.5 to 0.6, EER is about 2.1 and Specific Steam Consumption is about 8.75 kg/hr/TR, at a steam pressure of 3 bar or lower.

Double Effect Absorption chillers at about 8°C have COP in the range of 1 to 1.2, EER in the range of 3.5 to 4 Btu/hr/W and Specific Steam Consumption in the range of 4.5 to 5.25 kg/hr/TR, at a steam pressure of 8 to 8.5 bar. In the case of directly fired double effect absorption chillers, the Specific Fuel Consumption is likely to be about 0.35 m^3/hr/TR of natural gas or about 0.30 kg/h/TR of fuel oil.

Triple Effect Absorption chillers are now available from international refrigeration companies and the reported COP is 1.6 i.e. EER of over 5. The steam consumption is likely to be about 2.3 kg/hr/TR.

Water – Ammonia Absorption Chillers

Single Effect Absorption chillers at about -10°C have COP of about 0.55, EER is about 1.9 and Specific Steam Consumption is about 10.2 kg/hr/TR, at a steam pressure of 3 bar. At -40°C, the COP drops to about 0.45.

The COP of absorption chillers improves when the evaporator is operated at higher temperature.

The COP of the chiller being poor implies that the load dumped on the condenser is very large. It should also be noted that the cooling water passes through both the absorber and condenser. Hence any deficiency in the cooling water system can affect the cooling capacity and the COP of the machine.

The electrical power consumption of the vapor absorption machine is only about 2% of a comparable vapor compression machine, excluding the power consumption of cooling water system. . It may also be noted that for the same heat load, the heat rejected in an absorption machine is about 60% more than in a vapor compression machine; hence the cooling water pumping power consumption is likely to be significantly higher. It may be noted that electricity savings are large; however, the economics would depend on the cost of heat energy used in the absorption system.

To improve the part load efficiency of absorption chillers, variable speed drive can be used for controlling the speed of the dilute LiBr solution pump; this prevents the unnecessary sensible heating of surplus solution in the generator during low refrigeration loads.

Lithium bromide will be supplied with inhibitors to limit corrosion, but regular checks (say once in a year) of chemical composition of the solution is advisable. Slow corrosion by inhibited lithium bromide will produce non-condensable gases, which will cause a very gradual loss of vacuum. If vacuum degrades more quickly than that expected from manufacturer's guidelines, this may be an early warning of corrosion.

The chilled water and condenser water circuits should also be protected against corrosion and scaling by using conventional water treatment.

Vapor compression system with water cooled condensers, using any reciprocating, screw or scroll compressors, can be designed to achieve COP of 5.5 at full load operation. The differences in compressor efficiencies are marginal at full load and can be compensated by system design.

Centrifugal compressors with good system design can reach COP of 6.2 or higher.

Well design vapour compression systems with air cooled condensers are likely to consume about 20% more energy. The performance can be improved by providing evaporatively cooled air to condenser.

The COP of well designed small machines like window air conditioners and split air conditioning units are generally in the vicinity of 2.5.

The conditions and performance of the evaporator and the condenser decide the energy efficiency of the system to a great extent. Hence one refrigeration chiller package can be compared with another, but comparison of COPs merely based on compressor types can be misleading.

The COP of vapour compression systems at part loads is generally lower than that at full load. The extent of drop in COP will depend on the type of capacity control used. If capacity control is achieved by Variable Speed Drives, the COP improves and is generally higher than the full load value.

While selecting or modifying a vapour compression system, the average COP (or kW/TR) of the system should be evaluated, keeping in view, the likely average loading on the plant and the anticipated COP (or kW/TR) at different loading levels.

Vapour absorption chillers have low COP but are economical when operated with waste heat or cheap fuels.

2.8 Cooling Towers

Cooling Towers are an integral part of water cooled refrigeration systems. Poor performance of cooling towers can lead to serious inefficiencies in plant processes and equipment, leading to higher energy consumption, higher vent losses etc. Usually, Induced Draft type, Forced Draft type or Natural Draft type towers are used in the industry. The fans used are usually axial flow fans.

Induced and forced draft cooling towers operate by inducing or forcing air to flow through a matrix that is wetted by water from the condenser. Natural draft cooling towers spray water through nozzles from a height, resulting in cooling of water by the natural upward movement of air.

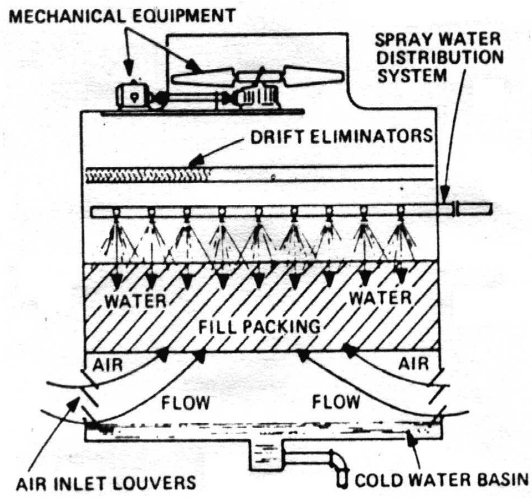

Figure 2-23: Counter-flow Induced Draft

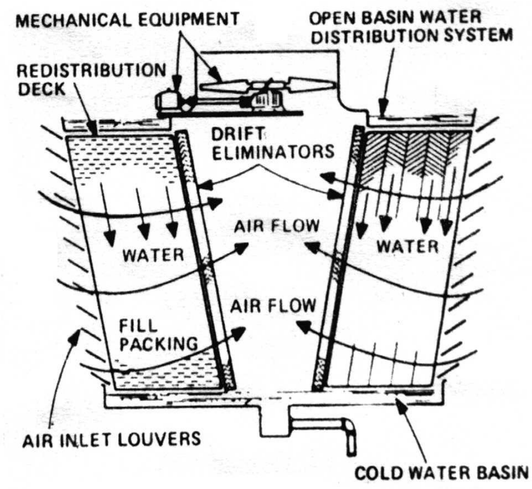

Figure 2-24: Cross-flow Induced Draft

Two parameters, which are useful for determining the performance of cooling towers, are the *Temperature Range* (difference between the cooling tower inlet and outlet water temperatures) and *Temperature Approach* (difference between the cooling tower cold water temperature and the ambient wet bulb temperature).

Though both parameters should be monitored, the *Approach* is a better indicator of cooling tower performance; lower the approach, better is the performance.

For the same heat load, water flow and fill material, lower the *Approach*, larger will be size of the cooling tower and higher would be air flow requirement.

Cooling towers are designed for dissipating a specified heat load at a specified ambient wet bulb temperature. However, actual wet bulb temperatures are continuously varying. Fig 2.25 shows information on variation in Temperature Approach for a particular cooling tower when it is subject to constant flow and heat load under varying ambient wet bulb temperatures. It may be noted that at lower ambient wet bulb temperatures, with constant heat load and water flow rate, the approach rises but the actual cold water temperature goes lower than the specified design value. The actual water flow rates and heat loads through the towers are also usually very different from the rated design flow rate and heat load.

As site tests are difficult, at least it may be ensured that the water flow to airflow is within an acceptable range. For example, for cooling towers designed for wet bulb temperatures of 26°C and an approach of 3°C, the air flow to water flow ratio is approximately 100 cfm/USGPM (1 US GPM = 3.785 lpm), which is equivalent an Liquid to Gas ratio (L/G) of 1.16.

The performance of induced draft cooling towers would depend on the condition of the fill, the heat load, the water flow, the airflow and the ambient wet bulb. The fill materials generally used are lumber, PVC honeycomb fill, PVC slats or PVC 'V' bars.

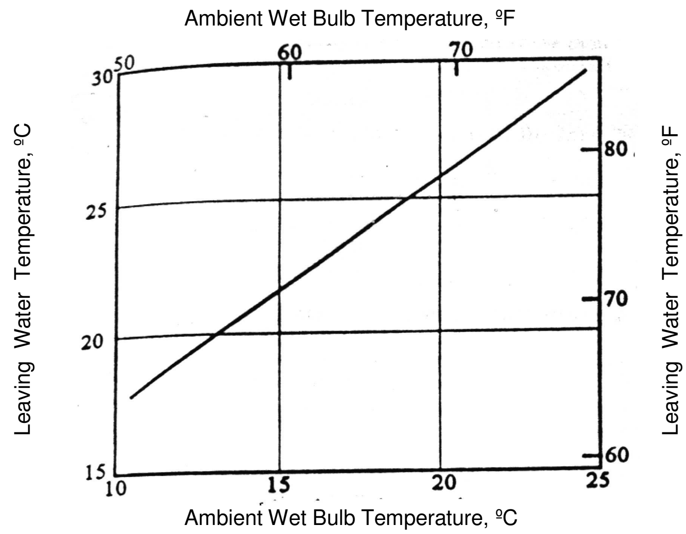

Figure 2-25: Variation of Cooling Water Leaving Temperature with Variation in Ambient WBT

For every 1kW (i.e. 860 kcal/h) of heat rejected, approximately 1.5 litres of water will be evaporated. Water quality if of paramount importance. It should not be turbid or contain suspended impurities. The pH value must be between 6.5 and 7.5 to prevent salt deposition when alkaline or corrosion when acidic. The total permanent ($CaCO_3$) hardness should be below 120 ppm to prevent scale buildup on condenser tubes, resulting in poor heat transfer and higher power consumption in compressor. Since water, available at most places in India, has impurities and high hardness level, the make-up water has to be filtered and softened.

3 STRATEGIES AND OPPORTUNITIES FOR ENERGY SAVING

3.1 Minimising Refrigeration & Air-conditioning

Use Cooling Tower Water at Higher Flows for Process Cooling
For process applications, the possibility of replacing chilled water with cooling water at higher flows and increased heat transfer area is a possibility for some process cooling applications. This approach can also help in replacing sub-zero brine with chilled water in some applications. Technology imported from colder countries generally specifies chilled water for cooling applications as temperatures between 10°C to 20°C are usually available from cooling towers for most part of the year in these countries. At the time of technology transfer or equipment purchase, redesign of the heat exchangers for summer cooling water temperatures, prevalent in India, or highest permissible chilled water temperatures should be seriously explored.

Use Evaporative Cooling for Comfort Cooling in Dry Areas
In areas having dry summer, reasonable comfort temperatures can be achieved by use of evaporative cooling i.e. by humidification of air by small desert coolers or central humidification plants. The energy consumption is likely to be about only about 10% to 20% of a conventional air conditioning system with HVAC chillers.

Building Structure Cooling
A significant portion of the air conditioning load is due the heat transmitted through the walls and heat stored in the structure. This results in the room inner wall surface being at temperatures higher than that of the human body. This results in discomfort as the human body is unable to radiate heat to the walls; hence heat transfer (body cooling) is possible only by convection (air movement) and perspiration. Cooling of the building structure (walls and roof) can lead to dramatic reduction in the wall temperature and human comfort.

Building structure cooling helps neutralize this solar heat load by cooling the structure itself. The Central Building Research Institute, Roorkee, has developed evaporative roof cooling methods (using wetted mats on roof tops) to neutralize the heat load. Wetted mats can be spread on the roof, the water will evaporate, effectively neutralising the solar heat load. The mats can be kept wet by small pump which is controlled by a moisture sensor or timer. It may be noted that the mat should be only moist and not flooded with water. This system can very effectively eliminate or significantly reduce energy consumption air-conditioning plants.

A Mumbai based company, specializing in innovations in refrigeration and air conditioning, has extended this concept further and attempted cooling the roof and the floor by burying a grid of water filled pipes (fig. 3.1), under vacuum, in the roof and floor. Water evaporates at 25ºC and the grid is connected to a small cooling tower, which acts as the heat sink and condenses the water vapour.

This concept was first attempted in a pent house at Ahmedabad. In May, with Ahmedabad ambient temperature at 41.6°C, the terrace floor temperature was 61.9°C, but the structure cooling system ensured that the room temperature was only 27°C. In fact, observations reveal that the room temperature remained almost constant through out the day and night.

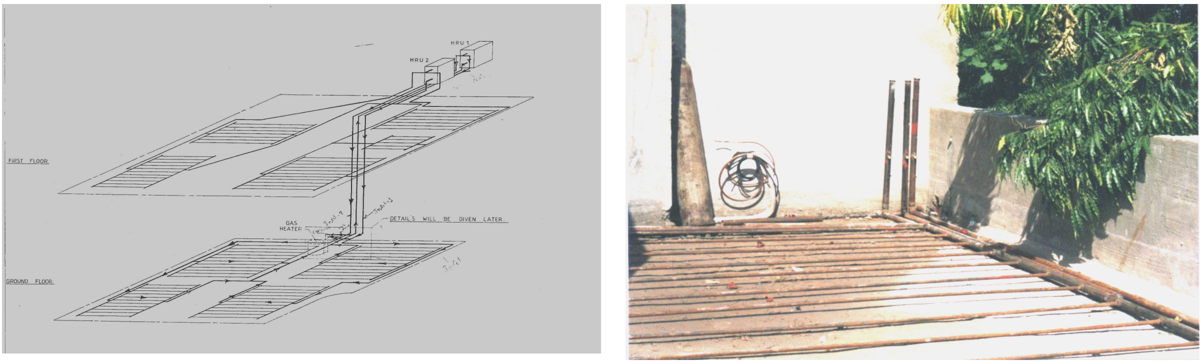

Figure 3-1: Building Structure Cooling: Grid of Pipes on the Roof and Floor (picture before application of cement plaster)

At Jaipur, the comparison of a house with structure cooling system and a neighbouring conventional house without structure cooling is shown in table 3.1. It may be noted that the roof underside temperature with structure cooling is 29.3ºC compared to 45.2ºC without structure cooling. With structure cooling the cooled part of the building becomes a heat sink for the entire structure. A cooler structure helps more efficient heat transfer from the human body to the structure and increase the comfort level. This concept has also been implemented by a leading pharmaceutical company in Madya Pradesh for ware house cooling. This can help eliminate or reduce the need for air conditioning.

Table 3-1: Reduction in Roof Underside Temperature due to Structure Cooling

	Roof Temperature (ºC)		
	Top side	Under side	Difference
House with Roof Cooling	54.6	29.3	25.3
Neighbouring house without roof cooling	52.8	45.2	7.6
Note: Reduction in roof heating load is 75% for air conditioning at 24ºC			

Building structure cooling potential has to be aggressively exploited with innovative methods to eliminate or minimize the need for air conditioning.

3.2 Operating at Higher Evaporator Temperature

Table 3.2 shows the variation of refrigeration capacity, power consumption and specific power consumption for a vapour compression system with variation in evaporator refrigerant gas temperature. It may be observed that higher the evaporator temperature, higher the system capacity, higher the power input and lower the specific power consumption (kW/TR). This clearly indicates that the cooling effect increases in greater proportion than the power consumption, thus, at higher evaporator temperatures, the refrigeration system will cool faster and shut off or unload.

The approximate thumb rule is that for every 1°C higher temperature in the evaporator, the specific power consumption will decrease by about 2 to 3%.

Table 3-2: Effect of Evaporator and Condenser Temperatures on Refrigeration Machine Performance

Evaporator Temperature °C		Condenser Temperatures °C			
		+35	+40	+45	+50
+5	Capacity (TR)	151	143	135	127
	Power cons. (kW)	94	102.7	110.6	117.8
	Sp. Power (kW/TR)	0.62	0.72	0.82	0.93
0	Capacity (TR)	129	118	111	104
	Power cons. (kW)	90	96.8	103	108.9
	Sp. Power (kW/TR)	0.70	0.82	0.93	1.05
-5	Capacity (TR)	103	96	90	84
	Power cons. (kW)	84.2	89.6	94.7	99.4
	Sp. Power (kW/TR)	0.82	0.93	1.05	1.19

<u>Increasing the Chilled Water Temperature Set Point</u>

The rationale behind temperature settings for all process applications needs to be reviewed, keeping in view the high energy costs. The aim is to avoid unnecessary super-cooling, without affecting production, quality and safety. Increasing energy costs have forced many industries to experiment and stabilize process cooling operation at higher temperatures.

<u>Improve air Distribution in Cold Storages</u>

In cold storage units, single floor mounted large coil and fans may lead to non-uniform temperatures in the cold store, which in turn leads to lower temperature settings. Replacement of single large floor mounted coil-fan-diffuser units (for better air distribution) by smaller wall or ceiling mounted small fan-coil units helps in more uniform distribution of cold air, which, in turn, helps lower temperature settings. resulting large energy savings. This measure has been implemented in some potato cold stores.

<u>Improve Air Distribution and Circulation in Air Conditioned Rooms</u>

In some air-conditioning systems, lower temperatures are set to over come problems of poor air distribution; making changes in ducting may be a more economical solution than permanently paying higher energy bills. In air-conditioned spaces, use of circulation fans can provide *apparent comfort* and help raise the room temperature settings to about 26°C or 28°C instead of 24°C. Quiet fans can be concealed behind suspended grids to ensure that the décor is not affected. The reduction in energy consumption in the refrigeration machines will be significantly more than the power consumed by the circulation fans.

3.3 Accurate Measurement and Control of Temperature

Most vapour compression machines use superheat sensing expansion valves, which do not give accurate temperature control especially when the compressors are operating at partial loads, resulting in significantly lower temperatures and higher energy consumption.

This can be avoided by the use of electronic expansion valves, which are modulating valves that operate based on electronic sensing of the end-use temperature.

A Mumbai based company has invented a new temperature controller that senses the temperature of the return air/water/brine and controls the superheat expansion valve by heating or cooling, as necessary, to give very accurate temperature control. This controller can be conveniently retrofitted on the existing superheat sensing system i.e. with out disturbing the existing expansion valve control system. This technology is already commercialized. Some industries and commercial buildings have reported energy savings along with good temperature control between +/-1°C (see case study).

3.4 Reduction in Heat Loads

<u>Keep Unnecessary Heat Loads Out</u>
Unnecessary heat loads may be kept outside air-conditioned spaces. Often, laboratory ovens are kept in air-conditioned spaces. Such practices may be avoided. Provide dedicated external air supply and exhaust to kitchens, cleaning rooms, combustion equipment etc. to prevent easy mixing of air between warm and cool rooms due to pressure differentials. In cold stores, idle operation of fork lift trucks should be avoided in case of any unforeseen stoppage of material movement.

<u>Use False Ceilings</u>
Air-conditioning of unnecessary space wastes energy. In rooms with very high ceiling, provision of false ceiling with return air ducts can reduce the air-conditioning load. Relocation of air diffusers to optimum heights in areas with high ceilings is recommended.

<u>Use Small Control Panel Coolers</u>
CNC machine shops, telecom switching rooms etc. are air-conditioned. As cooling is required only for the panels housing critical circuitry, use of small power panel coolers and hydraulic oil coolers (0.1 to 0.33 TR are available) can make the whole centralised air-conditioning redundant and save energy.

<u>Use Pre-Fabricated, Modular Cold Storage Units</u>
Cold stores should be designed with collapsible insulated partitions so that the space can be expanded or contracted as per the stored product volumes. The idea is to match product volumes and avoid unnecessary cooling of space and reduce losses. Modular cold store designs are commercially available.

<u>Cycle Evaporator Fans in Cold Stores</u>
In cold stores that remain shut for long periods, the heat load of the fans can be the major load. After attainment of temperature, refrigerant flow in evaporator fan-coils units and fan operation can be cycled on and off, using a programmable controller. This will reduce the heat load of fans and save energy (see case study).

3.5 Minimising Heat Ingress

<u>Cold Stores</u>
Many cold stores face serious problems with high energy consumption, ice buildup around doorways, frost and wet floors. They are largely caused by excessive amounts of outside warm air entering the facility at the loading bay. At - 10°C, air can hold only about 2g/m^3. Table 3.3 shows the comparison of heat ingress through identically sized open doors of air conditioned room and a cold store at -16ºC. Providing an anteroom, providing a high speed door (fig. 3.2) between the cold store and the anteroom and providing a dock leveler (fig. 3.3) to seal the back of the truck to the building are some of the methods to reduce air ingress to cold stores during loading and unloading operations. High speed doors are constructed of light weight, flexible materials for durability. They have minimal insulation properties, but since air exchange is the dominant mode of refrigeration loss, the door's speed becomes more important in controlling refrigeration

loss than its insulating properties. A modern high speed door will be open for about 10 seconds per passage. The anteroom in combination with a speed door to the cold store reduces the air infiltration from 2,900 to 700 cubic meters per truck load of cargo moved, a 76 percent reduction.

Table 3-3: *Heat Ingress into Air-conditioned Space through Open Doors*
(Door Size: 2m x 1m; Ambient Condition: 30°C & 60% Relative Humidity)

	Normal Air-conditioned room	Cold Storage
Room temperature, °C	24	-16
Room relative humidity, %	60	90
Additional heat load, TR	2.5	25

Figure 3-2High Speed Door

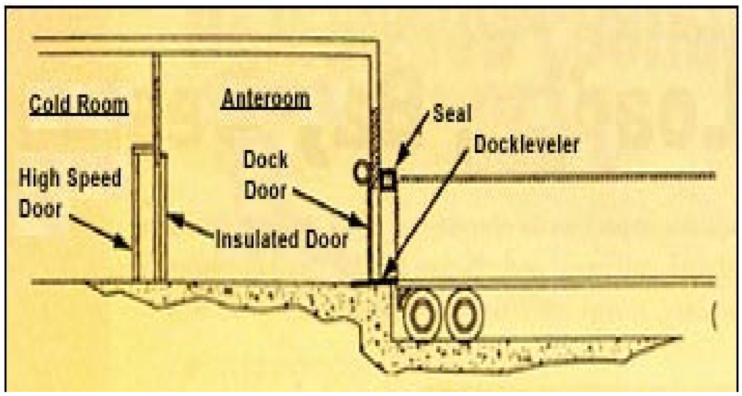

Figure 3-3: Dock Leveler helps seal back of truck with building

Figure 3-4: Example of good thermal insulation in a chilled water system
(Observe that chilled water pumps, valves and pipes are al well insulated)

Check and Maintain Thermal Insulation

Repair damaged insulation after regular checks. Insulate any hot or cold surfaces. Replace wet insulation. Insulate HVAC ducts running outside and through unoccupied spaces. Provide under-deck insulation on the ceiling of the top most floor of air-conditioned buildings.

Insulate Pipe Fittings

Generally, chilled water/brine tanks, pipe lines and end-use equipment in the industry are well insulated. However, valves, flanges etc. are often left uninsulated. With rising energy costs, it pays to insulate pipe flanges, valves, chilled water & brine pumps etc. also (fig. 3.4). Pre-formed insulation or 'home-made' box type insulation can be used. For out door piping, provided metal cladding for weather protection.

Use Landscaping to the Reduce Solar Heat Load

At the time of design of the building, fountains and water flow can be used provide evaporative cooling and act as heat sinks. Trees may be grown around buildings to reduce the heat ingress through windows and also reduce glare. Terrace lawns can help reduce the solar heat gain.

Figure 3-5: Typical Modern Building with Glass Façade

Reduce Excessive Use of Glass on Buildings

Modern commercial buildings (fig. 3.5) use glass facades or large window area (almost 30% of the wall area) resulting in large solar heat gain and heat transmission. Such architecture is suitable for cold countries; in India, it increases the air conditioning load for about eight to ten months in a year. In existing buildings, the possibility of replacing glass panes with laminated insulation boards should be seriously considered. The colour of the laminations can be chosen to suit the internal and external décor. Glass facades, if desired, can be provided in the form of a glass curtain external to a convention wall with necessary window area.

Use Glass with Low Solar Heat Gain Coefficient and Thermal Conductivity

Table 3.4 shows the solar heat gain coefficient (SHGC), thermal Conductivity and daylight transmittance for different types glass panes. Use of glass with low SHGC and thermal conductivity is recommended. Daylight transmittance is important, if electric lighting (another heat load on air conditioning) has to be minimized.

Table 3-4: ***Properties of Different Types of Window Glass***

Product	Solar Heat Gain Coefficient (SHGC)	Thermal Conductivity	Daylight Transmittance
Clear Glass	0.72	3.16	79
Body Tinted Glass	0.45	3.24	65
Hard Coated Solar Control Glass	0.26	3.27	24
Soft Coated Solar Control Glass	0.18	3.08	15
Low Emissivity Glass	0.56	2.33	61
Solar Control + Low Emissivity Glass	0.23	1.77	41

Use Low Conductivity Window Frames
Consider the use of plastic window frames in place of steel and Aluminium frames. This can reduce the heat ingress by conduction. However, security and safety issues related to the use of plastics should also be given due consideration.

Provide Insulation on Sun-Facing Roofs and Walls
Building insulation has not received much attention in India in spite of the very hot summers. Air-conditioned hotels and corporate office buildings should be constructed with insulated walls (hollow bricks with insulation, double walls with insulation fill etc.) to reduce the heat ingress. Providing roof under-deck insulation is a common practice.

Use Doors, Air-Curtains, PVC Strip Curtains for Air Conditioned Spaces
Add vestibules or revolving door or self-closing doors to primary exterior doors. Air-curtains and/or PVC strip curtains are recommended for air-conditioned spaces with heavy traffic of people or pallet trucks. Use intermediate doors in stairways and vertical passages to minimise building stack effect.

The Biological Sciences Building at Indian Institute of Technology, Kanpur, Roof and Wall Insulation has reduced the cooling load by 23%. The windows to wall ratio is only 7% and double glass insulated glass windows has reduced the cooling load by another 9%.

A study of a seven storied modern, air conditioned office building in Mumbai with about 70000 ft^2 has quantified an air conditioning load of 185 TR in summer. Thet window to wall ratio is about 30%. A simulation revealed that blocking 50% of the windows with laminated rigid insulation boards can reduce the air conditioning load by 13%. Providing gypsum board panels along walls with a one inch air gap can reduce the air conditioning load by an additional 4%.

The air conditioning system comprising 56 air cooled package air conditioners has an average COP of 2.7. Replacement of air cooled condensers by water cooler condensers can reduce the specific energy consumption by about 40%.

3.6 Reducing Ventilation Heat Load

Ventilation is required to for ensuring healthy conditions to the occupants of air conditioned rooms. Most designs provide ventilation of about 15 cfm per person in non-smoking area. In India, the issue of Indoor Air Quality (IAQ) is usually ignored; ventilation ports are generally kept closed and fresh air is usually available only through door openings.

Indoor Air Quality is a serious issue in the developed countries and, in future, buildings in India may also have to adhere to norms. However, with ventilation, the heat load increases as the heat content of fresh air is generally higher in summer. Air to air heat exchangers can help reduce this heat load by pre-cooling the incoming air with out-going exhaust air.

Air-to-Air Heat Exchangers for Pre-cooling Ventilation Air

Plate Heat Exchangers
These heat exchangers use a series of thin Aluminium sheets to transfer heat between two air streams. Sufficient turbulence is created between the plates to improve heat transfer.

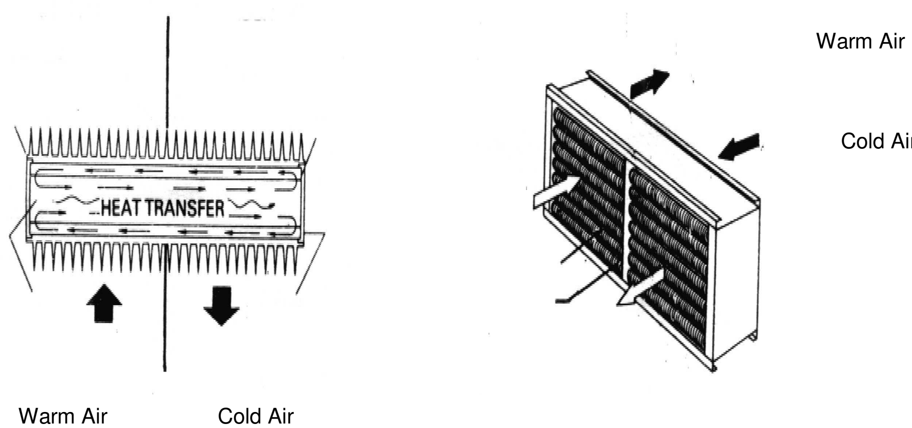

Figure 3-6: Heat Pipes

Heat Pipes
Heat Pipes (fig. 3.6) usually consist of sealed finned tubes with a wick lining on the inner side. The tube contains a working fluid, which evaporates from the hot end of the tube and condenses at the cold end, thus transferring heat. The working fluid is returned to the hot end by capillary action of the wick. An alternative heat pipe design (without the wick), working on the lines of a re-boiler is also being used.

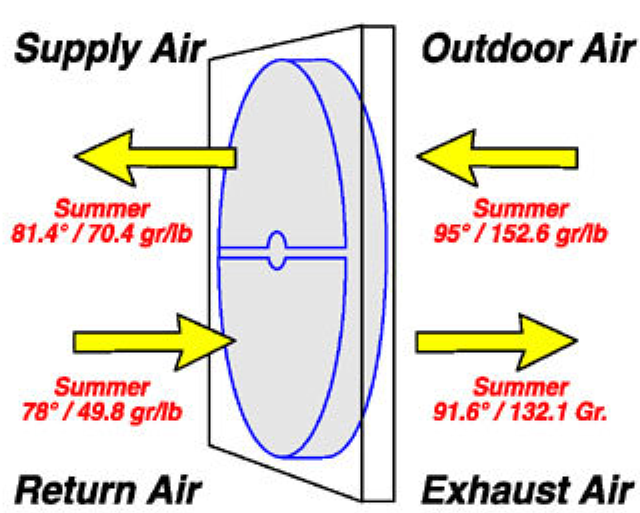

Figure 3-7: Heat Wheel

Heat Wheels
Heat wheels (fig. 3.7) are rotary heat exchangers packed with Aluminium honey comb fill. One half of the wheel is in contact with the warm air and the other half with the cold air. The sensible heat of the incoming warm air is transferred to one half of the fill; as it rotates slowly and moves into the cold half, the fill transfers the heat to the cold stream. Desiccant coated heat wheels are also available; here, in addition to sensible heat transfer, latent heat transfer also takes place as the desiccant absorbs moisture also.

(Also see case study of CII Green Building for an innovative method of pre-cooling ventilation air for an auditorium).

Reducing Ventilation Air Requirement by Ozone Dosing

The oxygen in the fresh air oxidises the Volatile Organic Compounds (VOC) and odours. The air also dilutes the CO_2. The minimum ventilation requirement is 15 cfm per person in a non-smoking room and 30 cfm per person if smoking is permitted. The load on the air conditioning system increases with ventilation. Use of controlled injection of ozone can help reduce the quantity of fresh air. Ozone is a powerful oxidant, which removes odour, VOC and even fungi by oxidation. This reduces the oxygen requirement in the form of ventilation and air is mainly required for diluting CO_2. This is a much smaller requirement, thus fresh air can be modulated rationally down to 5 cfm or less per person. However, if residual concentration of ozone exceeds limits, ozone is a toxic gas. The ASHRAE limit is 0.05 ppm. A plate type corona generator in the air duct is recommended along with an auto VOC controller.

3.7 Using Favourable Ambient Conditions

Use Cooling Tower Water Directly for Cooling in Winter
In locations with dry climate, the winter dry bulb (air) and wet bulb (water) temperatures are very low, especially at nights. During winter, cooling towers may be able to give temperatures of 8°C to 12°C. Plate heat exchangers can be used to transfer the heat load directly to the cooling tower, bypassing the chillers and shutting off the compressors or absorption machines. This system is used in very cold countries to prevent ice build-up in cooling towers. Some industries have successfully implemented this concept

during winter, especially during nights when the wet bulb temperature is close to chilled water temperatures.

Design New Air-conditioning Systems with Facility for 100% Fresh Air during Winter

In air-conditioned systems with centralised AHUs, fresh air and exhaust air ducts can be provided with dampers (and blowers, if necessary) to mix fresh air or use 100% fresh, filtered air, depending on the ambient conditions. These systems should also have the facility to exhaust the stale, warm air. Such air conditioning systems can be fitted with enthalpy sensors and motorised dampers to maximize the use of ambient fresh air during winter and cool summer nights.

The normal energy consumption of the air conditioning system may be comparatively less in winter, but even this reduced load can be eliminated if the system design incorporates the facility to take advantage of favourable ambient conditions. Fresh air systems can also be easily retrofitted in most air conditioning systems with centrally located air handling units.

Use Ground Source Heat Pumps

The near constant temperature in the ground (usually around 26ºC) can be used to cool the air for comfort cooling. Warm air is cooled by drawing it through very long air tunnels created by buring pipes at depths of around 4m. This method is suitable during the dry season. It may not be suitable when the relative humidity is high. This technique has been practically demonstrated at The Energy Research Institute (TERI), Gurgaon.

3.8 Use Evaporators and Condensers with Higher Heat Transfer Efficacy

Use Heat Exchangers with Larger Surface Area

In the Indian industry, the specific power consumption for chilled water at 6° to 8°C, in reasonably well maintained vapour compression systems, is likely to be around 0.8 kW/TR (only for compressor; pumps & fans are not included). The best systems available in India today can give specific power consumption lower than 0.6 kW/TR (compressor power). In the USA, the specific power consumption figure of chillers is expected to be below 0.56 kW/TR to qualify as an high efficiency chiller. A high efficiency chiller developed by Trane, USA, has a specific power consumption of 0.48 kW/TR.

This low specific power consumption has been achieved mainly by use of larger and more effective heat transfer area in the chillers and condensers. Larger area implies more effective heat transfer. This, in turn, implies that the refrigerant temperatures, for the same heat load, will be higher in the evaporator and lower in the condenser. Table 3.2 shows very clearly that higher evaporator temperatures and lower condenser temperatures lead to significant drop in the specific power consumption in the compressor. Hence replacement of chillers/ condensers or increase of heat transfer area by adding additional chillers/condensers in parallel can lead to significant energy savings.

1°C higher temperature in the evaporator or 1°C lower temperature in the condenser can reduce the specific power consumption by 2 to 3%.

Use Plate Heat Exchangers for Process and Refrigeration Machine Condenser Cooling

The use of Plate *Heat Exchangers* for condenser cooling can lead to lower temperature approach, hence reducing the compressor energy consumption. Plate heat exchangers have a temperature approach of 1°C to 5°C instead of around 5°C to 10°C for shell and tube heat exchangers.

Avoid the Use of Air Cooled Condensers
To take advantage of the wet bulb temperature, use of air-cooled condensers should be avoided for large cooling loads. Air cooled condensers may be permitted only for small cooling loads or in conditions of extreme scarcity of water or lack of space for cooling tower. Condenser water may be provided at the lowest acceptable temperature.

Evaporative Pre-coolers for Air-cooled Condensers
The performance of air-cooled condensers is limited by the dry bulb temperature. The performance of these condensers can be improved, in dry weather conditions, by providing humidified air near wet bulb temperature. This pre-cooler consists of a cooling pad (with trickling water) through which the air is drawn. Depending on the design, a booster fan may be required to overcome the additional resistance to air flow. The potential for energy saving in dry summer months may be about 30% to 40%.

3.9 Energy Saving Opportunities in Normal Operation

Use Building Thermal Inertia in Air Conditioning for Early Switch Off
Once the entire building structure is cooled, it takes a few hours to regain normal temperature. This building thermal lag can be used to minimise HVAC equipment operating time by shutting the air-conditioning system half hour or one hour before closing time.

Timers or Occupancy Sensors for Window and Split Air Conditioners
Window air conditioners and split air conditioners installed for office cabins may operate unnecessarily for long time without any occupancy. The use of timers or infra-red occupancy sensors can help switch of these machines automatically.

Interlock Fan Coil Units in Hotels with Door Lock or Master Switch
In hotels, unnecessary operation of Fan Coil Units can be prevented by providing an interlock with the door locking system or by switching control at the reception desk. The fan of the fan-coil unit should get switched off or go to low speed mode and the chilled water flow should be cut off by a solenoid valve. This can reduce the air-conditioning load in business hotels, during day time, when rooms are mostly not occupied.

Improve Utilisation Of Outside Air.
In systems with facility for using fresh air, the use of fresh air should be maximized when ambient conditions are favourable.

Maintain Correct Anti-freeze Concentration
In systems operating below 5°C, brine or glycol concentration should maintained at the correct levels as this has a significant impact on heat transfer and/or pumping energy.

Install Chiller Control System to Co-Ordinate Multiple Chillers.
Study part-load characteristics and cycling costs to determine the most efficient mode for operating multiple chillers. Run the chillers with lowest operating costs to serve the base load. Link Chiller control system to the Building Automation System to maximize savings.

Permit Lower Condenser Pressures during Favourable Ambient Conditions
Favourable ambient conditions reduce cooling air or cooling water temperatures, which will reduce condensing temperatures and pressures, thus reducing the compressor power. The control systems may set to allow the machines to operate at lower condenser pressure.

Optimise Water/Brine/Air Flow Rates
Optimise condenser water flow rate and chilled water/brine flow rate to maintain the design temperature difference across the chillers and condensers. For normal process

cooling and air conditioning applications, the chilled water flow rate is generally maintained around 2.4 gpm/TR and the cooling water flow rate is maintained around 3 gpm/TR. For comfort air-conditioning, the air flow rate is generally around 400 cfm/TR. In situations where the flows are higher, speed reduction should considered. With electronic variable speed drives, the flow can be changed to match the operating load by maintaining a constant temperature drop or pick-up across the evaporator and condenser at all times.

Defrosting

In cold stores, accumulation of frost on the evaporator tube reduces the air flow rate and hence the heat transfer rate significantly. The most widely used methods for defrosting are:

1. Shutting down the compressor, keeping the fan running and allowing the space heat to melt the frost.
2. Using out side warm air to melt the frost after isolating the coil from the cold room.
3. Using electric resistance heaters in thermal contact with the coil,
4. Bypass the condenser and let the hot gas into the evaporator to melt the frost,
5. Spray water on the coils to melt the frost.

The most popular method is the hot gas defrost, this is also relatively less expensive as the heat is a by-product of the refrigeration system. Water spray defrosting is used when quick defrosting is required in production mode, where quick return to production is essential. Electric defrosting is very expensive due to the high cost of electricity.

Irrespective of method, regular defrosting is an essential to maintain the heat transfer efficacy in the evaporator. The frequency of defrosting would depend on the rate of frost build-up, which again depends on the materials being stored. “Defrost on demand” control, which initiates defrost as per requirement rather than by a timer or manual intervention, can save energy where the freezing moisture load is variable.

Match the Refrigeration System Capacity to the Actual Requirement

Most of the refrigeration systems are oversized. Close matching of compressor capacity to the actual requirement will automatically raise the evaporator temperature and pressure, improved heat transfer efficacy and lower energy consumption in the compressor. This can be achieved by switching off some machines or by varying the speed of the compressor. For steady refrigeration loads, the speed can be changed by changing the pulley ratio. For fluctuating refrigeration loads, the use of variable speed drives may be required.

Monitor Performance of Refrigeration Machines

The specific power consumption (kW/TR) of all major chillers should be estimated periodically. It is often observed that the performance of similar machines is significantly different. This information can be used to maximise the operation of efficient machines. Preventive maintenance can also be scheduled based on this information.

The actual cooling capacity of a chiller can be estimated from the water/brine flow rate and the temperature difference across the evaporator or other heat exchanger. The cooling load on an air handling unit can be estimated from the air flow and the enthalpy difference between the inlet and outlet air. After measurement of dry bulb and wet bulb temperatures, enthalpy can be read off from the psychrometric chart. More detailed information on measurements and calculations related to capacity estimation of chillers is available in the “BEE Code on HVAC Chillers”.

3.10 Maintenance to Ensure Energy Efficient Operation

Temperature Settings

Regular check on control settings is required as these settings can drift over a period of time. Instrumentation should be programmable and settings should be locked with password protection.

Clean Fouled Heat Exchangers

Inefficient operation of refrigeration machines is usually due to fouling of condensers. This happens generally due to poor water treatment practices. Scaling of condenser tubes reduces the heat transfer efficacy, increases the refrigerant temperature and pressure in the condenser, reduces the cooling capacity, and increases the power consumption in the compressor. If this problem is ignored, it can also lead to tripping to the compressor on high discharge pressure. Chemical cleaning of heat exchangers is necessary to maintain the heat transfer efficacy. On-line monitoring and dosing systems are available for water treatment; this can ensure scale free operation on a continuous basis.

In the case of evaporative condensers, cleaning air side of condenser tubes helps in maintaining good heat transfer efficacy.

Air handling unit coil tube and fins should also be regularly cleaned externally.

Specify Appropriate Fouling Factors for Condensers

Fouling factors are considered in heat exchanger design to oversize the heat exchangers to offset the effect of fouling. However, equipment suppliers generally consider a fouling factor of 0.0005; a good water treatment programme is required to contain fouling within this limit. Ordinary scale of $CaCO_3$ of only 0.6 mm is equivalent to a fouling factor of 0.002. Studies have shown that 0.6 mm scale can result in an energy loss of about 20%. Poorly managed water treatment programs can very easily lead to scale build-up of this magnitude. Hence proper sizing of heat exchangers, based on realistic fouling factors, and a scientific water treatment programme (based on regular water quality measurements) are essential to maintain efficiency of refrigeration systems.

Purging the Condenser of Air

Air and other non-condensable gases may enter a system through leaks in seals, gaskets or uncapped valves. Air may also be present because of imperfect evacuation before initial charging of the system or due to impurities in the refrigerant or oil.

The non-condensable in condensers add partial pressure to the refrigerant vapour and thus increase the pressure against which the compressor has to work. The heat transfer coefficient also drops as the refrigerant has to diffuse through non-condensable to the tube surface before condensing.

The methods used for air purging are:

- ♣ Direct venting of the air-refrigerant mixture, which is a primitive manual technique
- ♣ A small compressor draws a sample of the refrigerant gas and compresses the mixture, condensing as much as possible of the refrigerant, and vents the vapour mixture that is now rich in non-condensable
- ♣ A low temperature evaporator, in-built in the system, condenses most of the refrigerant from the refrigerant-air mixture drawn from the condenser or receiver and vents the non-condensable. This method does not require a separate compressor and is used widely.

Purging of non-condensable plays an important role in maintaining the efficiency of refrigeration machines.

Do Not Overcharge Oil
The oil level in compressor should be checked through sight glass regularly. Both higher and lower oil levels can damage the condenser. Drop in oil level implies leaks or entrapment of oil elsewhere in the system. Excessive oil can result in film formation in heat exchangers can reduce heat transfer very significantly and increase the operating time and energy consumption.

Pumping Systems
Some of the methods of reducing the energy consumption is pumping systems are:

- ♣ Increasing fluid temperature differentials to reduce pumping rates. After a critical study of the requirement, this can be experimentally done by throttling valves. Variable speed drives can be programmed to maintain constant temperature differentials across heat exchangers.
- ♣ Balancing the system to minimise flows and reduce pumping power requirements. In systems with hot and cold wells, the over flow from one to the other should be reduced to the bare minimum
- ♣ Use of small booster pumps for small loads requiring higher pressures, instead of raising the entire flow to the high pressure.
- ♣ Using siphon effect to advantage: don't waste pumping head with a free fall return. This can be detected by measuring the pressure in the chilled water / brine / cooling water return lines near the discharge point.
- ♣ Operation of pumps near their best efficiency point. Both throttling of valves as well as excessive circulation of flow may move the pump away from the best efficiency point, leading to significant drop in efficiency.
- ♣ Modification of pumps to minimise throttling. This may involve change of impellers or pumps.
- ♣ Flow control by use of variable speed drives or sequenced control of smaller units.
- ♣ Seals and packing should be maintained to minimise water waste, especially chilled water.

Fans/Blowers
The following points are useful to reduce energy consumption in blowers and fans.

- ♣ Turning off fans when they are not needed.
- ♣ Regular cleaning of screens, filters and fan blades.
- ♣ Minimising fan speed.
- ♣ Checking belt tension regularly.
- ♣ Eliminating ductwork air leaks.

3.11 Energy Saving in Low Relative Humidity Air Conditioning

Some industrial applications require that the relative humidity be strictly maintained at levels ranging from 25% to 45% or less with room air temperatures ranging from 18° to 24°C. Conventionally, this is achieved by lowering the air temperature to very low levels (say 6° to 8°C) to condense out more moisture, followed by heating of the cold air (using electric heaters or steam coils in the duct) to about 14° to 18°C; this enabled maintenance of the specified room temperature and relative humidity conditions. However, this results in a double energy penalty i.e. additional energy consumption in duct heaters and additional energy consumption in the compressors to dissipate the additional heat added in the duct.

The modern method is to use of air-to-air heat exchangers that transfer heat from the warm, return air at 24°C (say) to the cold air from the AHU at 8°C (say), thus totally eliminating the duct heaters. This saves the duct heater energy and also reduces the cooling load on the compressor. Use of these heat recovery heat exchangers have lead to savings ranging from 30% to 60% in air-conditioning energy consumption in some industries (see case study).

3.12 Desuperheater for Recovering Condenser Waste Heat

The compressor discharge gas temperatures are likely to be in the vicinity of 100ºC to 120ºC. Use of de-superheaters can help recovery this heat in the form of hot water. Heat can also be recovered from the oil from screw compressors where the oil temperature is likely to be around 80ºC. De-superheaters have helped some hotels to minimize the operation of hot water generators.

A novel heat pump has been developed at the Heat Pump Laboratory, IIT, Bombay, which is capable of providing room air conditioning, heating tap water to 45°C and cooling potable water to 15°C. This heat pump has been designed to cater to the residential and light commercial market with nominal air conditioning capacity of 1, 1.5 and 2 TR in window and split models. In a typical home, this heat pump can provide night time bed room air conditioning, chilled potable drinking water and stored hot water for bathing.

3.13 Inter-fuel Substitution: Electricity Savings by Use of Absorption Chillers

The economics of changeover from vapour compression system to vapour absorption system would depend on the cost of heat energy used and the relative price of electricity. It is likely to be economical in locations where waste heat or low priced heat energy sources are available. The techno-economics is highly dynamic (due to fluctuating fuel prices) and situation specific.

For new projects, use of absorption chillers can facilitate working with a lower contract electrical demand, smaller transformer etc. leading to significant savings in the cost of the plant's electrical installation; this should also be considered in calculation of savings.

For very low temperature refrigeration (i.e. < -40 °C), the possibility of *Hybrid Chillers* can be considered. In Hybrid Chillers, the condensers of the low temperature vapour compression machines can be cooled by chilled water from Absorption Chillers.

In the U.S.A., with the availability of cheap natural gas, absorption chillers and *gas fired engine driven compressors* is also being aggressively promoted. A similar situation is likely in India with increasing availability of natural gas.

3.14 General Tips to Save Energy in Cooling Towers

- Control to the optimum temperature as determined from cooling tower and chiller performance data.
- Use two-speed or variable speed drives for cooling tower fan control if the fans are few. Stage the cooling tower fans with on-off control in multi-cell towers.
- Turn off unnecessary cooling tower fans when loads are reduced.
- Cover hot water basins (to minimise algae growth that contributes to fouling).
- Balance flow to cooling tower hot water basins.
- Periodically clean plugged cooling tower distribution nozzles.

- Install new nozzles to obtain a more uniform water pattern.
- Replace splash bars with self-extinguishing PVC cellular film fill.
- On old counter flow cooling towers, replace old spray type nozzles with new square spray ABS practically non-clogging nozzles.
- Replace slat type drift eliminators with low pressure drop, self extinguishing, and PVC cellular units.
- Follow manufacturer's recommended clearances around cooling towers and relocate or modify structures that interfere with the air intake or exhaust.
- Optimise cooling tower fan blade angle on a seasonal and/or load basis.
- Correct excessive and/or uneven fan blade tip clearance and poor fan balance.
- Use a velocity pressure recovery fan ring.
- Consider on-line water treatment.
- Restrict flows through large loads to design values.
- Shut off loads that are not in service.
- Take blow down water from return water header.
- Optimise blowdown flow rate.
- Send blowdown water to other uses or to the cheapest sewer to reduce effluent treatment load.
- Install interlocks to prevent fan operation when there is no water flow.

4 THERMAL STORAGE FOR MAXIMUM DEMAND CONTROL

4.1 Introduction

In most states of India, industrial consumers pay separate charges for electrical energy (kWh) consumed and maximum kVA or kW demand. *Time of Use* tariff has been introduced in some states. Maharashtra has four different tariff time zones for High Tension consumers; the off-peak energy tariff, between 10 pm to 6 am, is about 30% of the peak time rate. This provides an incentive for users to shift their large loads to off-peak period (night time). Consumers with large refrigeration and air-conditioning load can use the concept of thermal storage to reduce their maximum demand and also peak time energy consumption.

4.2 Technologies

Thermal Storage implies storing the cooling effect as latent heat in ice banks (fig. 4.1) or eutectic salts (which undergo a phase change) and using it when required by melting. This concept can be used to reduce energy cost by operating the refrigeration machines during off-peak hours to store "cold" for subsequent use during peak hours. This reduces the number of refrigeration machines that may have to run to satisfy the peak cooling load. The energy consumption will generally increase with thermal storage; however, this gets compensated by the off peak energy tariff benefits and reduction in maximum demand charges, the quantum of savings would depend on the load profile of the plant. Ice banks are being used in dairies for the past many years to overcome their peak cooling loads.

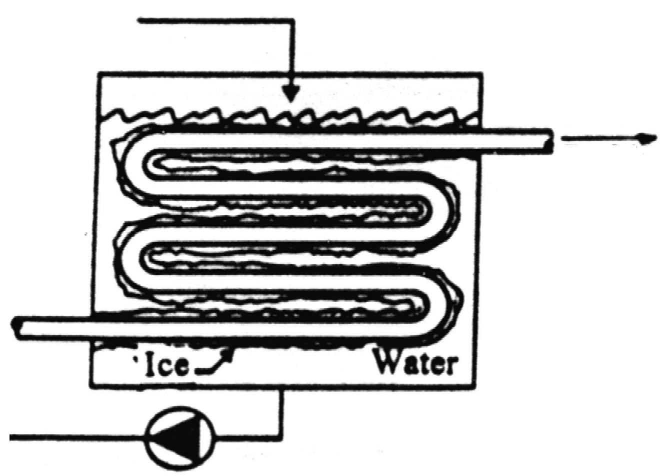

Fig. 4.1: Conventional Ice Storage Tanks with Flooded Ammonia Coils

The conventional ice bank consisted of mild steel coil in tank, flooded with liquid ammonia. During low load periods, ice was built up on the coils up to 2" thickness. Agitation of the chilled water around the ice was maintained for better heat transfer to the process.

Fig. 4.2: Modern Ice Storage tanks with Brine filled Plastic tubes

The modern ice bank system is a modular, insulated, polyethylene tank containing a spiral wound plastic tube heat exchanger surrounded with water. The tank is available in many sizes ranging from 45 to over 500 ton-hours. At night, 25% ethylene glycol solution, is cooled by a chiller and is circulated through the heat exchanger, extracting heat until eventually about 95% of the water in the tank is frozen solid. The ice is built uniformly throughout the tank by the patented temperature averaging effect of closely spaced counter-flow heat exchanger tubes (fig. 4.2). Water does not become surrounded by ice during the freezing process and can move freely as ice forms, preventing damage to the tank. At night, the water-glycol solution circulates through the chiller and the tank's heat exchanger, bypassing the process or air handling unit. The fluid is around –3ºC to –4 ºC and freezes the water surrounding the heat exchanger. The following day, the stored ice cools the solution from 1ºC to 11ºC. A temperature modulating valve, set at 6ºC in a bypass loop around the tank, permits a sufficient quantity of 11ºC fluid to bypass the tank, mix with 1ºC fluid, and achieve the desired 6ºC temperature. The 6ºC fluid enters the coil, where it cools air typically from 24ºC to 13ºC. The fluid leaves the coil at 16ºC, enters the chiller and is cooled to 11ºC.

It should be noted that, while making ice at night, the chiller must cool the water-glycol solution to -3ºCto -4ºC, rather than produce 6ºC water temperatures required for conventional air conditioning systems. This has the effect of "de-rating" the nominal chiller capacity by approximately 30 to 35 percent. Compressor efficiency, however, will vary only slightly (either better or worse) because lower night time temperatures result in cooler condenser temperatures and help keep the unit operating efficiently.

The temperature-modulating valve in the bypass loop has the added advantage of providing unlimited capacity control. During many mild temperature days in the spring and fall, the chiller will be capable of providing all the necessary cooling for the building without assistance from stored cooling. When the building's actual cooling load is equal to or lower than the chiller's capacity, all of the system coolant flows through the bypass loop. Ethylene glycol-based industrial coolant, which is specially formulated for low viscosity and superior heat transfer properties, is used. These contain a multi-component corrosion inhibitor system which permits the use of standard system pumps, seals and air handler coils. Because of the slight difference in heat transfer coefficient between water-glycol and plain water, the supply liquid temperature may have to be lowered by one or two degrees.

More recently, eutectic salts which can change from solid to liquid phase at temperatures ranging from –33° to +27°C. The eutectic salts are enclosed in plastic nodules; a number of such nodules are enclosed in a tank through which chilled water or brine can be allowed to flow. Some hotels and industries are already using this technology.

5 SYSTEM DESIGN AND EQUIPMENT SELECTION: ENERGY ISSUES

5.1 Introduction

The energy cost of refrigeration and air conditioning systems may be about six to ten times their first cost during their life time. Incorporation of energy efficiency into the design of the system can pay rich dividends.

It is important that, when different options are being considered, an exercise to estimate the running cost of the system is done. The designer should be provided adequate information on the load profile and ambient conditions throughout the year. While the system may be designed for the worst case condition, facility for efficient modulation of the system should be available by design or at least the possibility of easy retrofit should exist.

Manufacturers have improved the COP of their top-end chiller packages by about 40 percent, offering dramatic reductions in energy use and peak demand. In view of the phase out of most of the older refrigerants, selection of the new refrigerant and suitable compressor should be done with a long term view.

5.2 Important Issues

5.2.1 Energy Cost

The energy cost and availability for different energy sources like electricity, liquid & gaseous fuels, waste heat, agro waste etc. should be known and carefully analysed as it can have a bearing on the type of machines to be selected.

5.2.2 Refrigeration Load Estimation

This is the most important basis on which the system will be designed. The best way to estimate the load is from theoretical calculations, tempered by past experience of process engineers and energy auditors. Systems should never be designed with vague data. Process engineers, if not aware or committed to energy efficiency issues, may overstate the load or keep huge margins of safety. It should also be ascertained whether the entire proposed load would be installed together or over a period of a few years. The variation of process load over a 24 hour period, the batch cycle times, seasonal variations etc. should all be approximately quantified.

"Free cooling" or "Economical cooling" available from nature in the form of cool air or cool water from cooling towers should not be over looked. To the extent possible pre-cool products using natural means before using refrigeration. Options like building structure cooling, ground source heat pumps etc. should be seriously considered.

Comfort and process related air conditioning should be minimised to the extent possible. Table 5.1 shows typical air conditioning requirements for office spaces.

Conventional reheat based systems for low relative humidity applications should not be used.

Table 5-1: ***Thumb Rules for Calculating Comfort Air-conditioning Load***

Type of Office	Heat Load TR per 100 ft^2
Small office cabins	1.0
Medium size offices (seating 10 to 30 people, with centralised air-conditioning)	0.55 to 0.65
Large multistoried office buildings (with centralised air-conditioning)	0.35

5.2.3 System Design

Pumping System

Pumping systems in complicated networks are difficult to optimize and may eventually be a significant heat load on the system due to inefficient operation. Consider combining variable speed drives with primary-only water loops using two-way valves on end-use cooling coils. Primary-only pump strategies deliver chilled water directly from the chilled-water pump to the terminal equipment, rather than using hierarchic secondary (or even tertiary) loops for pressure control. Modern chillers are much more tolerant of variable water flow rates and inlet temperatures. In case the end-use flows are highly variable, fixed speed primary pump and variable speed secondary pump with a "tie-line" near the chiller can be considered.

Air Handling Systems – Some Emerging Concepts

Two relatively new concepts for saving energy in air conditioning system by addressing the air supply system and its temperature are discussed here. Both systems have a potential for saving energy but have to be applied very carefully, keeping in view location specific issues.

Under floor Supply Air System using Warmer Supply Air

Most air conditioning systems provide cool air at about 13ºC through over head supply air ducts. The air flow is generally close to about 400 cfm per TR for comfort air conditioning.

Under floor air supply through an under floor plenum is an emerging concept that has the potential to save energy. In this arrangement, ducting is eliminated and the air is supplied to a common plenum below the floor. Air outlets with controls are provided near the work spaces which can be controlled as per individual requirement. Since the air is being supplied very close to the user, the supply air temperature settings can be raised to about 17ºC to 20ºC. The air will pick-up heat and rise into the return air plenum above the ceiling and return at a temperature of about 25ºC to 30ºC. This system can work well in dry weather or in situations where higher humidity is required or can be tolerated.

The energy savings result from reduced chiller operation due to higher supply air temperature settings. The absence of ducting leads to reduced static pressure requirements and hence reduced fan power. There is also a possibility of operating lower air flow rates in this system.

Over Ceiling Supply Air System using Cold Air System

In a conventional air conditioning system (CAC), the supply air temperature is about 13°C for comfort air conditioning. This gives a temperature differential of 11°C between room temperature of 24°C and supply air. In a cold air system (CAS), the supply air temperature will be at about 7°C for a room temperature of 25.6°C which, as will be seen later, is acceptable in CAS, yielding a temperature differential of 18.6ºC. Accordingly, in a CAS, the dehumidified air flow rate will be only about 50-60% of the value in CAC for the mentioned temperatures. Cold air distribution systems supply air at 7ºC instead of the conventional 13ºC, cutting down the air flow rate, air handling unit size and duct dimensions.

Brine is supplied to the cooling coil at 1.5ºC (instead of chilled water at 7ºC). Thus, the air that leaves the coil, can be at about 7ºC. The relative humidity in CAS may be higher (about 35%), permitting the operating at higher room temperature for the same comfort level.

In a carefully designed system, CAS may save energy due reduce fluid handling quantity and to lower fan and pumping power.

5.2.4 Minimise Heat Ingress – Select Right Thermal Insulation

Ensure that plant location is selected to ensure minimum heat loss in piping, insulation and issues related to hot air ingress to cooled spaces are adequately address to reduce extraneous heat loads to the minimum. The key properties of the insulating material are suitability for operating temperature, thermal conductivity, water vapour permeability and moisture absorption. Tables 5.2 and 5.3 provides information that help selection of the right thermal insulation of the appropriate thickness.

In addition to running pipes, flexible insulation should be used to “box up” flanges, valves, and pumps. Prefabricated, pre-insulated HVAC ductwork for commercial applications also offers a double wall design. Pumps, chillers etc should also be well insulated.

In the case of cold stores, polyethylene sheet of adequate thickness must be laid on the floor as a vapour barrier, before laying the insulation. All joints must be lapped and bonded together and must continue under the wall panels. All projections and loose stones in the sub-floor must be removed to prevent puncturing the vapour barrier.

Buildings with large air conditioning loads, should be provided wall insulation and glass area should be minimized. Building insulation in the form of double wall with air gap, double wall with insulation fill or single wall with retrofit gypsum panel with air gap etc. should be considered.

Table 5-2: Thermal Conductivities of Some Insulating Materials

Material	Conductivity W/m°K
Cellular foam glass	0.050
Cellular polyurethane	0.023
Expanded polystyrene	0.035
Extruded polystyrene	0.027
Glass fibre	0.036
Poly isocyanate	0.020

Table 5-3: Insulation thickness for Refrigeration Piping

Nominal Dia of pipe	Mean temperature (ºC)														
	10			5			0			-10			-20		
	Thermal conductivity at mean temperature														
	0.02	.03	.04	.02	.03	.04	.02	.03	.04	.02	.03	.04	.02	.03	.04
1”	10	14	17	14	18	23	17	23	29	23	32	41	29	41	53
1.5”	11	16	20	15	21	27	19	27	33	26	37	47	33	47	62
2”	13	18	23	17	25	31	22	31	40	30	44	57	38	57	77
4”	14	20	27	20	30	38	25	38	51	37	57	73	49	92	92
6”	15	23	31	22	35	45	30	45	57	43	62	79	55	99	99
10”	17	26	34	25	37	48	33	48	61	47	67	86	60	10	10

5.2.5 Sizing & Selecting the Right Refrigeration Machine

Assess energy availability at the location i.e. electricity, cheap heat source or waste heat. Energy economics for Vapour Compression System or Vapour Absorption System or a Hybrid System will be decided by the refrigeration load profile and the cost of energy.

For very low temperatures, evaporator temperatures below -20ºC, the two stage vapour compression systems with inter-cooling will have to be considered to limit the discharge pressures. For evaporator temperatures below -50ºC, cascade systems may have to be considered. If cheap heat source or waste heat is available, the possibility of using Water – Ammonia Absorption chillers also exists.

Avoid over sizing to the extent possible – try to match the actual load, provide efficient method of modulation. Use of larger chillers will increase the parasitic loads like pumps, fans etc. Sizes and number of chillers should be selected to match the load profile as closely as possible; the same applies to parasitic loads like pumps, fans etc. This may even imply selecting unequally sized machines.

Water-cooled machines are generally preferable for refrigeration systems. In case of water scarcity, absence of water treatment facility, non-availability of space for cooling towers etc., slightly oversized air cooled condensers may be used.

In the case of air-conditioning systems, the system type can have a significant impact on the energy consumption. Air can be cooled directly by refrigerant in the AHU cooling coil (DX-type chiller) or by chilled water in the AHU cooling coil. Table 5.4 shows a comparison of likely energy consumption for a typical 100 TR air-conditioning system with different types of systems. The DX chiller with the water cooled condenser is the most efficient.

The selection of equipment for vapour compression systems should be done such that most of the compressors will work near their full load and near their best operating efficiency points at assessed plant refrigeration load at various points of time. It should also be ensured that associated equipments like pumps; fans etc. are also sized to ensure minimize energy consumption and reduce the “parasitic load” on the system. Select cooling coils for high temperature drops at design conditions to reduce pumping energy.

Table 5-4: Comparison of Likely Energy Consumption for a Typical 100 TR Air-Conditioning System

Type	DX chiller (air cooled condenser)	DX chiller (water cooled condenser)	Chilled water system
Capacity, TR	100	100	100
Saturated Suction temp, °C	6.1	6.1	4.4
Saturated Discharge temp, °C	52.7	37.8	37.8
Compressor power, kW (a)	104.5	62.0	62.0
Chilled water pump, kW (b)	0.0	0.0	12.8
Condenser cooling fan/pump (c)	7.8	13.5	13.5
Total power, kW (a) + (b) +(c)	112.3	75.5	88.3
Total Specific power, kW/TR	1.12	0.76	0.88

When comparing efficiencies of different chillers, efficiencies should be compared at both full load and part loads. The purchase decision should be based on the most likely load that is likely to prevail for the longest time period.

In applications where over sizing is inevitable, consider the use of variable speed drives on compressors and pumps.

In the case of Absorption Chillers, since capacity control up to 25% can be done without drop in efficiency, single chiller configurations can be permitted. However, variable speed drives may be installed to optimize the flow of pumps depending on the load.

Don't oversize refrigeration plants unnecessarily, it may cost more to buy and run!

Purchase only high efficiency machines, even at a premium. Use larger heat transfer areas for evaporators and condensers. Consider realistic fouling factors based on water quality and general immediate environment. It should be noted that energy efficient refrigeration systems always operate with smaller temperature lifts (or compression ratios), resulting in lower discharge temperatures and the resultant benefits of lower energy consumption, lower maintenance costs and better reliability.

Manufacturers of equipments may be overly optimistic about performance. In the Indian context, marketing personnel exaggerate the positive features and underplay or conceal the negative features of the equipment. Information provided by manufacturers should be vetted and confirmed from independent, reliable sources. The ASHRAE and ISHRAE handbooks are some of the reliable sources

Standard chilling packages may not have the most optimum configuration. It is always sensible to customize the package to improve the COP. In most cases, this implies providing more heat transfer area in the evaporator and condenser to reduce the temperature lift or (compression ratio). The expansion device should ideally be electronic valve or port valve, instead of the normal superheat sensing valve. In case a normal superheat valve is used and part load operation of the refrigeration machine is likely, retrofit of *a precision temperature controller* may be considered.

Cooling tower capacity is much less expensive than chiller capacity. A larger tower will provide cooler water to the chiller at very low cost and thus improve its efficiency at "off-design" conditions. Two-speed motor systems give almost all of the benefits of variable speed drives for cooling tower fans, at much lower cost.

Select the refrigerant keeping in view the Ozone depletion issue, the long term availability, the operating temperature and pressure conditions, the type of compressor and safety. Consider zeotropic blends that may give advantages like increased evaporator capacity and efficiency. Zeotropic blends are more suited for DX systems, check with the supplier before using them in flooded systems.

In the case of absorption chillers, the user has no choice as the machine is sold only as a package.

Use energy efficient motors for continuous or near continuous operation applications should be preferred.

5.2.6 Controls for Energy Efficiency

Variable Speed Drives

Most equipments are likely to marginally oversized or may operate at part loads during certain periods. Variable speed drives (VSDs) should be considered for pumps, fans, and chillers to improve part-load efficiency (see case study) and minimize the inefficiencies associated with oversized equipment.

Process & Building Automation Systems

Advanced controls allow closed-loop feedback to optimize many more variables than earlier systems could accommodate. Using new sensors and variable speed drives, controls can now optimize approach temperatures, water, brine and air flows. Modern energy management systems are sophisticated and can to optimize performance. They allow simultaneous protection of equipment and maximum energy efficiency and are strongly recommended for chillers consuming significant amount of energy.

To minimize energy and demand costs, it is important to have both energy efficient chillers and an appropriate chiller plant control strategy. The chiller plant control strategy will determine the leaving chilled water temperature that meets the current process or building requirements while minimizing energy consumption. Connectivity between the chiller control system and the Building Automation System (BAS) is necessary to maximise the energy saving potential. ASHRAE'S BAC net and Echelon's Lon Works are two standard protocols that are used for connectivity between the chiller and Building Automation Systems.

6 CASE STUDIES

6.1 Case Study 1: Operational Saving – Correct Refrigerant Charging

Equipment

Refrigeration Compressor

Industry/sector

Hikal Ltd. Panoli/Chemical

Details of Measure

The company situated in Panoli manufactures Pesticides. The Utility systems account for a large portion of energy consumption. 3 refrigeration compressors of 75 kW rated power each were operated for chilled water plant.

Measurements indicated that the specific power consumption of chillers were higher than expected. The suction pressure was about 30 to 40 psig only. The specific power consumption was initially 0.98 kW/TR at compressor.

The energy audit report recommended that more refrigerant be charged and suction pressure be increased to 60 psig. This resulted in increase in capacity by 35% and efficiency of these machines improved and one compressor out of three was switched off permanently.

Undercharging of refrigerant is done deliberately sometimes to reduce liquid carry over when chilling load is low. However, this affects the efficiency of the system. For 8ºC chilled water system, the suction pressure is expected to be about 60 to 65 psig. A chiller having a capacity of 75 TR at 57 psig suction pressure will deliver only 49 TR at 37 psig.

The power saving was 21 kW.

Cost benefit analysis

- Type of Measure: No Investment
- Annual Energy Savings: 68,000 kWh
- Actual Cost Savings: Rs 3.5 lakhs
- Actual Investment: Very small
- Payback: Immediate

6.2 Case Study 2: Matching Compressor Capacity To Actual Load By Speed Variation

Equipment

Refrigeration compressor

Name of Industry/sector

Clarisis Organics- Baroda/Chemicals

Details of Measure

The company situated near Baroda manufactures benzene derivatives. The major load in the plant was refrigeration. Two compressors were working; one for chilled water and the other for brine.

Refrigeration compressors were operating at 50 to 60% of the time only. This indicated excess capacity of compressor. The spedific energy consumption was also very high. It was suggested to reduce the speed of compressors by 40% by changing pulley size, keeping in view of the minimum speed of operation recommended by the manufacturer.

Equipment	Rating	Before modification		After modification	
	Power	Actual kW	KW/TR	Actual kW	KW/TR
Chilled water plant compressor	90 kW	74	1.2	35.6	0.7
Brine plant compressor	55 kW	53	1.6	32.3	1.2

The refrigeration capacity is proportional to the speed of the compressor. Hence it was suggested to reduce the speed of brine compressor from 750 rpm to 500 rpm and the speed of chilled water compressor from 780 rpm to 400 rpm. The compressors' operating hours, after reducing the speed, were expected to increase. Since, under this derated condition, the existing evaporator and the condenser are oversized, the specific power consumption was expected to reduce resulting in energy savings.

Cost benefit analysis

- Type of Measure: Marginal Investment
- Annual Energy Savings: 1,15,000 kWh
- Actual Cost Savings: Rs 5.2 lakhs
- Actual Investment: Rs 10,000/-
- Payback: 18 days

6.3 Case Study 3: Replacement Of Inefficient Chiller

Equipment

Refrigeration

Industry/sector

Pennwault Agru Plastics, Baroda

Details of Measure

The company manufactures Plastic pipes for natural gas distribution and municipal sewage pipes.

The process heat load was estimated to be 10 TR. The semi-hermetic compressor specific power consumption was 3.7 kW/TR, which indicated a very inefficient compressor. Usually 0.8 to 1.0 kW/TR at motor input is expected from a good compressor.

A recommendation was given to replace the existing semi-hermetic compressor with a new open type compressor.

The efficiency of the compressor was poor due to a manufacturing defect. The new compressor is installed by compressor manufacturer, free of cost, as it was in warranty period. A new open type compressor was installed and tested. The specific power consumption was 1.0 kW/TR at motor input.

Cost benefit analysis

- Type of Measure: Major retrofit
- Annual Energy Savings: 1,35,000 kWh
- Actual Cost Savings: Rs 6,75,000
- Actual Investment: Nil. The old compressor was on warranty period. Hence replaced free of cost by manufacturer
- Payback: Immediate

6.4 Case Study 4: Innovative Retrofit Precision Temperature Controller

Equipment

Refrigeration compressor for air conditioning

Industry/sector

Mahindra Tractors - Mumbai/Engineering

Details of Measure

The company manufactures tractor components. This air conditioning plant, comprising 3 x 80 TR refrigeration machines, caters to the engine assembly section and offices. The plant operates for 12 hours per day.

This measure is implemented to achieve steady room temperature within +/-1ºC.

At part loads, the operation of the superheat sensing expansion valve is sluggish. This indigenous precision temperature controller senses the return air temperature and heats or cools the sensing bulb of the expansion valve and makes it act faster. This controller, a novel invention of a Mumbai based company, can be installed without disturbing the existing set up of the machine and controller.

Cost benefit analysis

- Annual Energy Savings: 41,868 kWh
- Actual Cost Savings: Rs 1,67,472.
- Actual Investment: Rs 1.2 lakhs approximately.
- Payback: 8.5 months

6.5 Case Study 5: Elimination Of Re-Heat In Low Relative Humidity Air Conditioning

Equipment

Refrigeration compressor for air conditioning

Industry/sector

Asia Brown Boveri - Baroda/Engineering

Background

The company manufactures electrical switchgear components, turbochargers etc.. This air conditioning plant caters to the CVT winding shop. The plant operates continuously to maintain 24ºC and 45% to 50% relative humidity in the CVT winding department.

This was being achieved by the use of electrical duct heaters in the supply air duct. The Air-to-Air Heat Exchanger was installed recover heat from the return air duct to eliminate or reduce the requirement of electrical duct heaters.

Details of Measure

To achieve low relative humidity in air conditioned rooms, the conventional system super cools the air to about 6ºC to condense out more moisture and then reheats the air with electrical duct heaters to about 13ºC to maintain the room conditions. The paradoxical operation results in additional energy consumption in heaters and extra heat load on the air conditioning system.

In the new system, a reboiler filled with refrigerant (behaving like a heat pipe) was installed to transfer heat between the return air stream at 24ºC and the supply air stream at 6ºC, heating the supply air to 13ºC with out electrical heaters.

Cost benefit analysis

- Annual Energy Savings: 97,000 kWh
- Actual Cost Savings: Rs 5,00,000.
- Actual Investment: Rs 5 lakhs approximately.
- Payback: 1 year

6.6 Case Study 6: Larger Heat Exchangers Improve Cop

Use of Larger Condenser to Improve Refrigeration Efficiency

Excel Logistics has a 34000 m^3 (operating at -20ºC) cold store at Melton Mowbray, U.K., which was running on CFC refrigerants. During plant refurbishment to eliminate the use of CFCs, they selected larger evaporative condensers than would normally be specified to improve energy efficiency.

The normal practice is to design condensers for 35ºC condensing temperature at a wet bulb temperature of 20ºC. In this case, Excel Logistics specified a 30ºC condenser i.e. a condenser with 49% more capacity.

Monitoring showed that the installation of larger condensers reduced annual compressor energy consumption by 10.4%. As only two of the four systems were modified, the savings achieved represent a reduction in site energy bill of 2340 pounds per year. The combined consumption of the condenser fan and pump was slightly lower in the new system, saving 1600 pounds per year. The total annual savings were therefore 2500 pounds for the two condensers converted.

The additional cost of the two larger condensers over the standard sized equivalent was 5000 pounds, giving a payback period of two years.

(Source: ETSU, U.K.)

6.7 Case Study 7: Electronic Expansion Valves Save Energy

Use of Electronic Expansion Valves for Improving Refrigeration Efficiency

Doble Quality Foods operate a small cold store in St. Agnes, U.K., and distribute frozen and chilled foods to the catering and butchery trades. The company successfully replaced an old refrigeration system with a new CFC-free system in 1993, achieving 30% energy savings. Even so, the company recognized that additional energy savings could be made by using electronic expansion valves and associated controls.

The original valves were of the thermostatic type and the defrost is electric. The cold store is maintained at -22ºC. The total cooling capacity of the plant is 23 TR and the annual electricity cost was some 11000 pounds prior to the alterations described here.

The improved control features provided by the electronic expansion valves enabled savings in refrigeration energy use because:

- The need for a fixed, high condensing temperature (discharge/head pressure) is avoided;
- Superheat (the temperature difference between the vapor leaving the evaporator and the boiling refrigerant liquid entering it) can be minimized.

The benefits derived from the installation of electronic expansion valves are shown are as follows:

- Improved control of liquid refrigerant flow to evaporator
- Improved heat transfer, since more evaporator surface area used for boiling liquid.

- Raised evaporating temperature and higher suction pressure, reducing energy use by 2% to 3% per 1ºC in evaporating temperature.
- Reduced risk of liquid carry-over to compressor, reducing risk of compressor damage.
- Avoids need for constant pressure drop across expansion valve.
- Allows condensing temperature (discharge/head pressure) to educe at times of low ambient temperatures.

Reducing superheat improves evaporator heat transfer because more evaporator heat transfer because more of the evaporator surface area is devoted to liquid boiling and less to heating of vapor. The improved heat transfer allows the plant to operate at a higher evaporating temperature 9suction pressure). Earlier, Doble plant used to operate with fixed, high condensing temperature, which is, generally, a constant required by thermostatic expansion valves. This constraint is removed if electronic valves and controllers are used. For each 1ºC rise in evaporating temperature, compressor energy costs are reduced by 2%.

The expansion valve controllers have two additional energy saving features.

- 'Defrost by experience' – this allows controllers to decide whether a defrost is required by monitoring the duration of previous defrosts. This means that defrosts can be omitted when not required (e.g. at night when the store is closed and moisture ingress is minimized).
- Evaporator fan cycling – at times of low cooling demand, evaporator fans can be cycled on and off. This reduces energy use while ensuring that air temperatures in the store remain uniform.

Doble Foods also chose to install additional related compressor controllers and to use remote monitoring. The compressor controller determines when compressors should be switched on and off and also operates safety cut-outs and alarms. This enhanced level of control saves energy by:

- More accurate sensing and more stable control of evaporating temperature at the higher levels permitted by the new system;
- Raising the evaporating temperature set point at times of low cooling demand on the store (e.g. at night).

A 'spin off' benefit of the improved controllers is improved monitoring. The controllers store key measurements which can be made available to the management using a hand held display, local or remote PCs. This data can be used to ensure performance is maintained and to calculate the COP of the system.

The energy saving by use of electronic expansion valve controller for "floating head pressure" and "defrost by experience "is 19%, a cost saving of 2090 pounds. The pay back period was 1.4 years.

The energy saving by use of "floating suction pressure" is about 660 pounds per annum. The cost of the compressor controller was 3000 pounds. The payback period was 4.5 years.

Additional 2000 pounds were invested in the monitoring system. The overall payback of the whole project was 2.9 years.

(Source: ETSU, U.K)

6.8 Case Study 8: Pre-cooling Of Auditorium Ventilation Air

The CII-Godrej Centre for Environmental Excellence is India's first Green Building and only the World's 3rd Building to be awarded the Platinum rating by the Green Buildings Council of USA. The building has been designed and constructed keeping in view the following:

In the air conditioned auditorium, the design fresh air requirement is 4000 cfm, which would increase the air conditioning load. A unique method of natural pre-cooling of warm fresh air has been attempted. The concept has evolved from the study of the Moorish wind towers in Spain and the Hawa Mahal at Jaipur.

TIME (HRS)	AMBIENT CONDITIONS		AIR FROM WIND TOWER		AIR FLOW	SENSIBLE COOLING	EQUIVALENT REF. EFFECT TR
	DBT	WBT	DBT	WBT	CFM	BTU/HR	
1130	81	73	73	70	12000	103680	8.64
1215	84	72	72	69		155520	12.96
1300	82	70	74	68		103680	8.64
1330	82	71	74	68		103680	8.64
1400	83	70	74	68		116640	9.72
1415	85	71	74	68		142560	11.88
1430	83	71	74	68		116640	9.72
1445	85	71	74	68		142560	11.88
1500	83	69	74	68		116640	9.72

On an average, the air conditioning load reduction is about 10 TR.

For 10 hours operation of the AHU per day, the savings would amount to 100 TR-hrs
Gross saving in energy @ 0.8 kW/Ton = 80 kWh per day.
Less Pump work spent = 0.75 kW X 2 hrs = 1.5 kWh.
Fan work spent = 0.5 kW X 4 hrs = 2 kWh.

Amount of water consumed = 760 litres.

The net saving is about 76.5 kWh/day in cool climate. The savings are expected to be higher in summer.

(Source: Panasia Engineers, Mumbai)

6.9 Case Study 9: Energy Saving In Fruit Cold Stores

The fruit storage sector presents an outstanding opportunity for energy efficiency. Refrigeration systems account for most energy use at these facilities, and potential for savings can range from 10% to over 50%.

Ammonia or freon-based refrigeration systems are used at most fruit storage warehouses. At facilities with no packing line, refrigeration can use 90% to 95% or more of total utility energy use. With a packing line, refrigeration energy use can range from 70% to 80% of total facility energy, with the balance required by packing lines and lighting. Although evaporator fans account for only one-fourth of total refrigeration system horsepower, they use well over half of refrigeration energy. This is clearly a reflection of a system configuration that is designed for peak pull-down loads. The longer CA rooms are held, the greater the fraction that evaporator fans account for.

Sub-system	Refrigeration connected load %	Refrigeration energy %
Evaporators	29	54
Compressors	69	41
Condensers	6	5

SPECIFIC RECOMMENDATIONS
Although each facility is unique, the following specific energy efficiency recommendations are most common:

1. Computer Control
2. Reduced Minimum Condensing Pressure
3. Evaporator Fan VFD Control
4. Condenser Fan VFD Control
5. Screw Compressor VFD Control

Computer Control
The control system can save tremendous amounts of energy by proper control of VFDs or evaporator fan cycling, compressor sequencing, automated suction pressure optimization, and better condensing pressure control relative to pressure switches. The control system also plays an important role in monitoring and managing room temperature and atmosphere conditions.

Reduced Minimum Condensing Pressure
Most refrigeration systems are designed around peak refrigeration loads during peak summer conditions. However, fruit storage systems operate primarily during the fall, winter, and early spring. During this time, there are thousands of hours per year when ambient temperatures are extremely low, and refrigeration condensing pressure could operate as low as 80 psig (54 °F) to 90 psig (59 °F). Unfortunately, many systems maintain a higher condensing pressure during this period. There are several common reasons for the elevated condensing pressure:

- Screw compressor liquid injection oil cooling may not work properly below 125 to 140 psig.
- Gas pressure systems (i.e., pumper drum) may not operate correctly due to controlled pressure receiver (CPR) pressure or other system limitations.

- A common water tank is used as a condenser sump and defrost water storage. The need (or desire) for warm defrost water necessitates an elevated condensing pressure.
- Often, an elevated condensing pressure is a “tradition”. This can be the result of misconceptions about issues such as screw compressor volume ratios.

Each of these barriers has a solution. Whether retrofitting a screw compressor with thermosiphon oil cooling, or installing separate tanks for condenser water and defrost water, there is rarely an insurmountable barrier to achieving 80 to 90 psig minimum condensing pressure.

Evaporator Fan VFD Control

Evaporator fan VFD control is typically the single greatest opportunity to reduce refrigeration energy use. In a CA facility, fans are typically operated at full speed for several weeks following room seal. At that point, fan speed can be immediately reduced to 50%, or can be staged down over several weeks, again with a minimum of 50% speed. *(There is little incentive to reduce speed below 50% speed, since power has already been reduced by over 80%, and additional speed reduction only diminishes airflow in the room).* In general, one VFD is installed for each CA room. Where VFDs are installed on common storages, one VFD is installed per refrigeration zone. It is important that the VFD be correctly.

Condenser Fan VFD Control

Rather than cycling condenser fans for capacity control, VFD control can be utilized. Similar to evaporator fans, the affinity laws provide excellent savings relative to simple cycling. However, a second convincing benefit also plays a part in the decision to utilized speed control.

VFD control eliminates the rapid cycling of the condenser fan required for proper pressure control. With the VFD, average condensing pressure is smoother and lower. Since lower condensing pressure reduces compressor energy use, condenser fan VFDs often achieve compressor energy savings that is larger than the fan energy savings! Condenser belt and sheave wear is also reduced as a result of VFD control.

Screw Compressor VFD Control

Often, operating a screw compressor unloaded can be avoided by a diverse selection of machines that can be properly sequenced by the control system. In other systems, reciprocating compressors are used to efficiently trim system capacity and power. However, in some systems, there is no avoiding operation of a screw compressor in the unloaded condition. In this situation, a VFD can be installed for the screw compressor. Rather than using the conventional slide valve for unloading, the compressor speed is reduced from 3600 to 1800 rpm, keeping the slide valve fully open. Once at 1800 rpm, the slide valve is then closed to further reduce capacity. Note that the effectiveness of compressor VFD control is dependent on a variety of issues, including the shape of the basic compressor part load power curve. However, there are certainly times when this VFD application is viable.

New construction projects present several additional opportunities for refrigeration energy efficiency. These include:

• **High-Efficiency Condensers:** Condensers are selected with heat rejection per horsepower ratings of 300 MBH/hp or higher.

• **Larger Condensers:** Condensers are selected at lower design condensing temperature (e.g., 85 °F rather than 95 °F), saving both compressor and condenser energy.

Diverse Compressors: Rather than a few large compressors, a diverse selection of machines can be made to allow for optimum sequencing. This helps avoid operating screw compressors unloaded. A combination of screw compressors (for harvest) and reciprocating compressors (holding season) can be beneficial.

• **Incremental Cost for VFDs:** On new construction projects, the cost of VFD control is tempered by savings from eliminated magnetic starters. VFD cost can be as much as 20% to 50% lower during new construction.

• **Premium Efficiency Motors:** Upgrading to premium efficiency motors may be viable for some loads. Condenser pumps and holding-season compressors are two examples.

Two non-refrigeration opportunities are commonly encountered with fruit storage warehouses: lighting upgrades and fast-acting doors.

Lighting Upgrades

In general, lighting can be upgraded in packing warehouses and common storages. In packing areas, standard fluorescent lighting can be upgraded to electronic ballasts and T8 or T10 lamp technology. Obviously, any incandescent lighting should be retrofit with fluorescent or metal halide technology. In common storages, metal halide light fixtures can be installed (or retrofit) with bi-level lighting. A single fixture or group of fixtures is controlled by a motion detector. When no activity is seen for 5 to 15 minutes, the fixture dims. When dimmed, a 400-Watt metal halide fixture that normally draws 465 input Watts may only draw 180 to 200 Watts. When a lift truck or other motion is detected, the fixture immediately increases light output, with none of the delay common to initial startup of metal halide and other high-intensity discharge fixtures.

Fast-Acting Doors

Common storages are notorious for significant infiltration loads. Doors are often left open, or strip curtains are damaged to the point of reduced effectiveness. In some situations, an automated, fast-acting door can be installed to reduced infiltration load. Reducing infiltration can benefit evaporator fan VFDs or fan cycling by reducing room load.

The examples of five fruit warehouses that implemented some of these measures are summarise here. The range in size from 193 to 1,448 hp of refrigeration. Annual energy use ranges from 618,000 to 4,099,000 kWh/yr. Savings ranges from 21% to 47% of total facility energy use. Simple payback ranged from 3 to 6 years without utility incentives, and 1.5 to 3.1 years with utility incentives.

FRUIT WAREHOUSE ENERGY EFFICIENCY CASE STUDIES					
Case Study:	**#1**	**#2**	**#3**	**#4**	**#5**
Refrigeration Horsepower:	277	193	976	777	1448
Rooms (Common & CA):	4	3	25	19	24
Total Facility Energy (kWh/yr):	618,400	713,600	1,641,300	2,114,260	4,098,880
Refrigeration Energy (kWh/yr):	595,976	510,599	1,625,757	1,676,594	3,734,135
Refrigeration Percentage:	96%	72%	99%	79%	91%
Computer Control					X
Reduced Condensing Pressure	X	X	X	X	X
Evaporator Fan VFD Control	X	X	X	X	X
Condenser FanVFD Control	X	X	X	X	X
Lighting Upgrades		X			
Miscellaneous			X		
Energy Savings (kWh/yr):	290,245	333,080	661,225	892,894	879,489
Percent Savings of Total Facility:	47%	47%	40%	42%	21%
Cost Savings ($/yr):	$ 9,683	$ 12,939	$ 24,227	$ 30,555	$ 30,791
Installation Cost:	$ 29,349	$ 80,504	$ 118,251	$ 112,894	$ 152,515
Simple Payback (years):	3.0	6.2	4.9	3.7	5.0
Utility Incentive ($0.12/kWh, up to 50%):	$ 14,675	$ 39,970	$ 59,126	$ 56,447	$ 76,258
Final Customer Cost:	$ 14,675	$ 40,534	$ 59,126	$ 56,447	$ 76,258
Simple Payback After Incentive (years):	1.5	3.1	2.4	1.8	2.5

Source: Marcus H. Wilcox, P.E., President, Cascade Energy Engineering
2001 Proceedings, Energy Efficiency in Fruit Storage Warehouses
WSU-TFREC Post harvest Information Network, USA.

6.10 Case Study 10: Tri-Fuel Chiller Options Saves Cost

The 48 story Time & Life Building had steam-turbine-drive chillers, installed in 1959, for building cooling. An electric-drive chiller was added in the mid '80s to supply additional cooling and introduce an alternative-fuel option. Today, the building boasts a one-of-a-kind tri-fuel plant that uses electric-, steam- and natural gas-powered chillers to meet increased cooling demands and achieve substantial savings in energy costs. It also provides energy redundancy, something that has become critical in California and other parts of the country.

The Rockefeller Group selected four YORK chillers for the Time & Life Building's tri-fuel plant, including one 2100-TR electric-drive chiller, one 1500-TR steam-turbine-drive chiller, and
two 1850-TR gas-engine-drive chillers. In addition, YORK provided one of the gas-engine-drive
chillers with an 1850-TR electric-motor-drive parallel driveline, allowing operators to switch between gas and electric energy sources, depending on which source is most economical at a given time. In addition, the second gas engine-drive chiller has provisions made for a parallel electric-motor driveline to be added in the future, if desired. The plant also features YorkTalk communication interfaces, linking
the chillers to the facility's existing Johnson Controls building-automation system.

The first chiller came on line March 2000, and the plant was fully operational in July. Already, the project is saving $750,000 per year in energy costs. During the worst weeks of the summer, the two gas-engine-drive chillers were operated, avoiding peak-demand electric and steam rates, and more than justifying the investment in the natural-gas component of the plant.

Improved equipment efficiencies also produce savings for the Rockefeller Group. The new steam-turbine-drive chiller consumes just 9.9 pounds of steam per TR-hr, compared to 15 pounds per TR-hr consumed by the two original steam-turbine-drive chillers.

Similarly, the new electric-drive chiller, with a performance of 0.60 kW/TR, significantly improves upon the 0.76 kW/TR efficiency rating of the original electric chiller. "By adding two gas engine-drive chillers, each with an impressive 1.8 coefficient of performance, along with plate heat exchangers for winter free-cooling.

The plant capable of increasing overall performance efficiencies by 40 to 50 percent.

Time & Life Building Tri-Fuel Chiller Plant

	ORIGINAL PLANT	NEW PLANT
Steam-chiller efficiency	15 lbs / TR-hr	9.9 lbs / TR-hr
Electric-chiller efficiency	0.76 kW / TR	0.60 kW / TR
Gas-chiller efficiency	N/A	1.8 COP

Source: York International Corporation, USA

7 REFERENCES

1. ETSU Good Practice Guides, *Energy Efficiency Best Practice Programme, U.K.*

2. Saving Electricity in Utility Systems of Industrial Plants, *Devki Energy Consultancy Pvt. Ltd., Baroda, 1996.*

3. Efficient Use & Management of Electricity, *Devki Energy Consultancy Pvt. Ltd., Baroda, 2004.*

4. Industrial Refrigeration Handbook, *Wilber F. Stoeker, McGraw Hill (1995).*

5. Refrigeration and Air conditioning, *Manohar Prasad, New Age International (P) Ltd., 1996.*

6. ASHRAE Handbooks, *ASHRAE, Atlanta, Georgia, USA.*

7. Cooling Tower Technology – Maintenance, Upgrading and Rebuilding, *Robert Burger, The Fairmont Press Inc., Georgia, USA.*

8. Low-E Glazing Design Guide, *Timothy E. Johnson, Butterworth Architecture.*

9. ISHRAE Journals (1998-2004), *Indian Society for Heating, Refrigeration and Air Conditioning Engineers, Mumbai.*

10. Catalogues of Manufacturers.

8 CONVERSION TABLES

1 Kcal	3.9685 Btu
1 kWh	3413 Btu
1 kWh	860 kcal
1 Btu	1.055 kJ
1 calorie	4.186 Joules
1 hp	746 Watts
1 kg	2.2 lb (pounds)
1 metre	3.28 feet
1 inch	2.54 cm
1 kg/cm^2	14.22 psi
1 atmosphere	1.0332 kg/cm^2
1 kg/cm^2	10 metres of water column @ 4°C
1 kg/cm^2	9.807×10^4 Pascals
1 Ton of Refrigeration	3023 kcal/hour
1 Ton of Refrigeration	12000 Btu/hour
1 US Gallon	3.785 litres
1 Imperial Gallon	4.546 litres
°F	1.8 x °C + 32
°K	°C + 273