REFRIGERATION & AIR CONDITIONING SYSTEM

1. INTRODUCTION

This section briefly describes the main features of the refrigeration and air conditioning system.

1.1 What is Refrigeration and Air Conditioning

Refrigeration and air conditioning is used to cool products or a building environment. The refrigeration or air conditioning system (R) transfers heat from a cooler low-energy reservoir to a warmer high-energy reservoir (see figure 1).

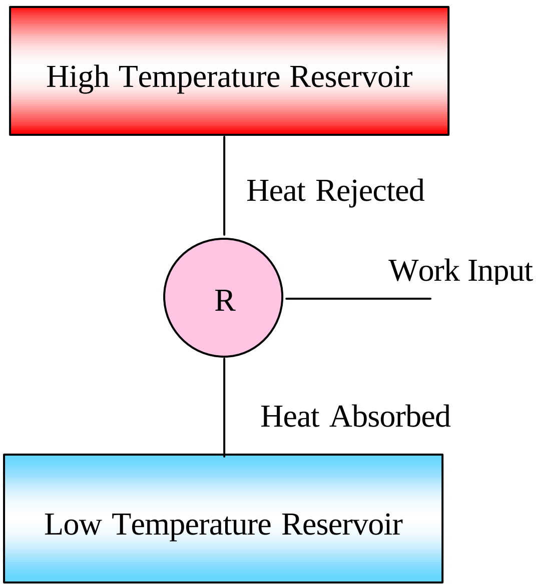


Figure 1. Schematic representation of refrigeration system

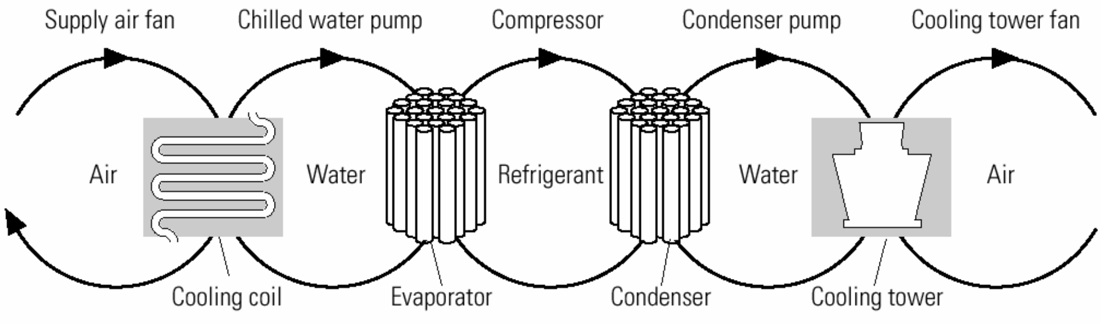


Figure 2. A typical Heat Transfer Loop in Refrigeration System
(Bureau of Energy Efficiency, 2004)

There are several heat transfer loops in a refrigeration system as shown in Figure 2. Thermal energy moves from left to right as it is extracted from the space and expelled into the outdoors through five loops of heat transfer:

- ***Indoor air loop***. In the left loop, indoor air is driven by the supply air fan through a cooling coil, where it transfers its heat to chilled water. The cool air then cools the building space.
- ***Chilled water loop***. Driven by the chilled water pump, water returns from the cooling coil to the chiller's evaporator to be re-cooled.
- ***Refrigerant loop***. Using a phase-change refrigerant, the chiller's compressor pumps heat from the chilled water to the condenser water.
- ***Condenser water loop***. Water absorbs heat from the chiller's condenser, and the condenser water pump sends it to the cooling tower.
- ***Cooling tower loop***. The cooling tower's fan drives air across an open flow of the hot condenser water, transferring the heat to the outdoors.

1.2 Air-Conditioning Systems

Depending on applications, there are several options / combinations of air conditioning, which are available for use:

- Air conditioning (for space or machines)
- Split air conditioners
- Fan coil units in a larger system
- Air handling units in a larger system

1.3 Refrigeration Systems (for processes)

The following refrigeration systems exists for industrial processes (e.g. chilling plants) and domestic purposes (modular units, i.e. refrigerators):

- Small capacity modular units of the direct expansion type similar to domestic refrigerators.
- Centralized chilled water plants with chilled water as a secondary coolant for a temperature range over typically 5 ^{o}C. They can also be used for ice bank formation.

- Brine plants, which use brines as a lower temperature, secondary coolant for typically sub-zero temperature applications, which come as modular unit capacities as well as large centralized plant capacities.
- The plant capacities up to 50 TR (tons of refrigeration) are usually considered as small capacity, 50 – 250 TR as medium capacity and over 250 TR as large capacity units.

A large company may have a bank of units, often with common chilled water pumps, condenser water pumps, cooling towers, as an off site utility. The same company may also have two or three levels of refrigeration and air conditioning such as a combination of:

- Comfort air conditioning (20 – 25 ^{o}C)
- Chilled water system (8^0 – 10^0 C)
- Brine system (sub-zero applications)

2. TYPES OF REFRIGERATION AND AIR CONDITIONING

This section describes the two principle types of refrigeration plants found in industry: Vapour Compression Refrigeration (VCR) and Vapour Absorption Refrigeration (VAR). VCR uses mechanical energy as the driving force for refrigeration, while VAR uses thermal energy as the driving force for refrigeration.

2.1 Vapour Compression Refrigeration System

2.1.1 Description

Compression refrigeration cycles take advantage of the fact that highly compressed fluids at a certain temperature tend to get colder when they are allowed to expand. If the pressure change is high enough, then the compressed gas will be hotter than our source of cooling (outside air, for instance) and the expanded gas will be cooler than our desired cold temperature. In this case, fluid is used to cool a low temperature environment and reject the heat to a high temperature environment.

Vapour compression refrigeration cycles have two advantages. First, a large amount of thermal energy is required to change a liquid to a vapor, and therefore a lot of heat can be removed from the air-conditioned space. Second, the isothermal nature of the vaporization allows extraction of heat without raising the temperature of the working fluid to the temperature of whatever is being cooled. This means that the heat transfer rate remains high, because the closer the working fluid temperature approaches that of the surroundings, the lower the rate of heat transfer.

The refrigeration cycle is shown in Figure 3 and 4 and can be broken down into the following stages:

- **1 – 2.** Low-pressure liquid refrigerant in the evaporator absorbs heat from its surroundings, usually air, water or some other process liquid. During this process it changes its state from a liquid to a gas, and at the evaporator exit is slightly superheated.
- **2 – 3.** The superheated vapour enters the compressor where its pressure is raised. The temperature will also increase, because a proportion of the energy put into the compression process is transferred to the refrigerant.
- **3 – 4.** The high pressure superheated gas passes from the compressor into the condenser. The initial part of the cooling process (3-3a) de-superheats the gas before it is then turned

back into liquid (3a-3b). The cooling for this process is usually achieved by using air or water. A further reduction in temperature happens in the pipe work and liquid receiver (3b - 4), so that the refrigerant liquid is sub-cooled as it enters the expansion device.

- **4 - 1** The high-pressure sub-cooled liquid passes through the expansion device, which both reduces its pressure and controls the flow into the evaporator.

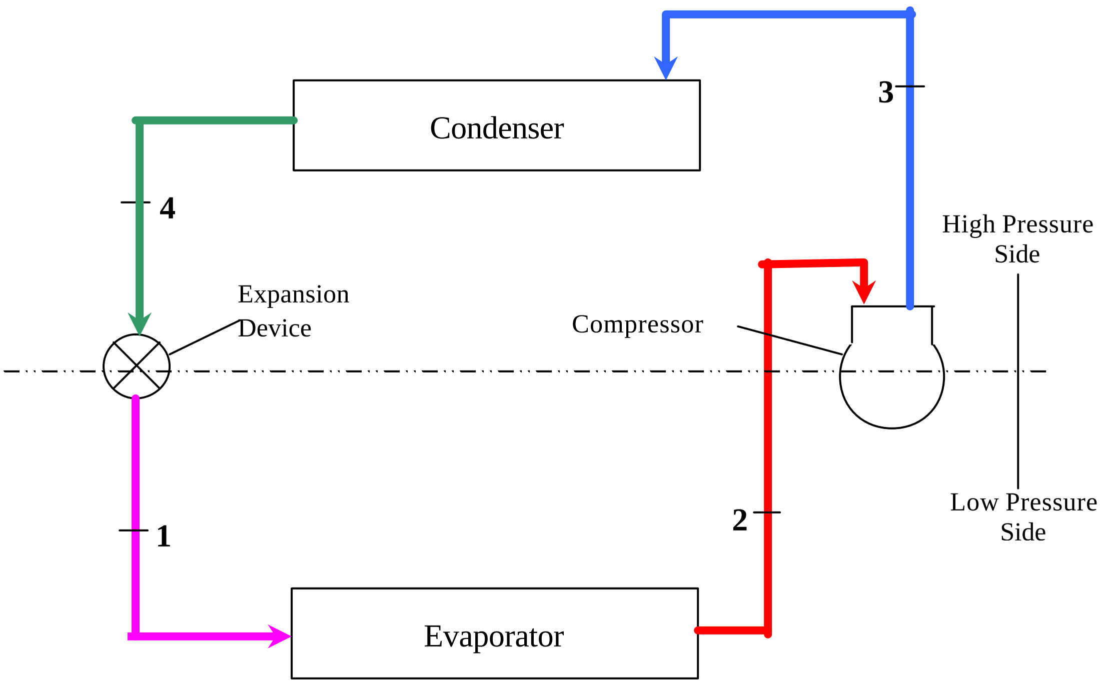


Figure 3. Schematic representation of the vapour compression refrigeration cycle

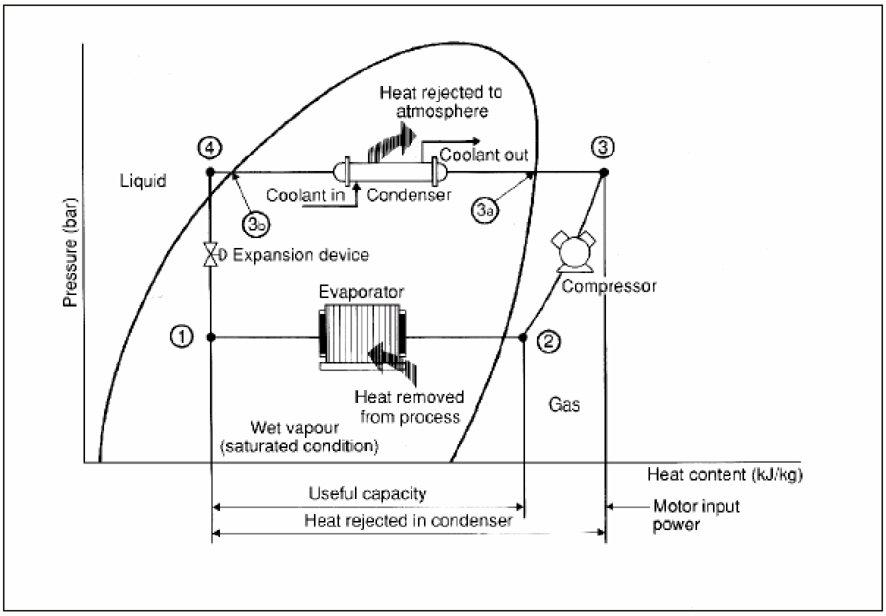


Figure 4. Schematic representation of the refrigeration cycle including pressure changes
(Bureau of Energy Efficiency, 2004)

The condenser has to be capable of rejecting the combined heat inputs of the evaporator and the compressor. In other words: (1 - 2) + (2 - 3) has to be the same as (3 - 4). There is no heat loss or gain through the expansion device.

2.1.2 Types of refrigerant used in vapour compression systems

A variety of refrigerants are used in vapor compression systems. The required cooling temperature largely determines the choice of fluid. Commonly used refrigerants are in the family of chlorinated fluorocarbons (CFCs, also called Freons): R-11, R-12, R-21, R-22 and R-502. The properties of these refrigerants are summarized in Table 1 and the performance of these refrigerants is given in Table 2 below.

Table 1. Properties of commonly used refrigerants (adapted from Arora, C.P., 2000)

Refrigerant	Boiling Point ** (^oC)	Freezing Point (^oC)	Vapor Pressure * (kPa)	Vapor Volume * (m^3 / kg)	Enthalpy *	
					Liquid (kJ / kg)	Vapor (kJ / kg)
R – 11	-23.82	-111.0	25.73	0.61170	191.40	385.43
R – 12	-29.79	-158.0	219.28	0.07702	190.72	347.96
R – 22	-40.76	-160.0	354.74	0.06513	188.55	400.83
R – 502	-45.40	---	414.30	0.04234	188.87	342.31
R – 7 (Ammonia)	-33.30	-77.7	289.93	0.41949	808.71	487.76

* *At -10 ^oC*

** *At Standard Atmospheric Pressure (101.325 kPa)*

Table 2. Performance of commonly used refrigerants (adapted from Arora, C.P., 2000)

Refrigerant	Evaporating Press (kPa)	Condensing Press (kPa)	Pressure Ratio	Vapor Enthalpy (kJ / kg)	COP^{**}_{carnot}
R – 11	20.4	125.5	6.15	155.4	5.03
R – 12	182.7	744.6	4.08	116.3	4.70
R – 22	295.8	1192.1	4.03	162.8	4.66
R - 502	349.6	1308.6	3.74	106.2	4.37
R - 717	236.5	1166.5	4.93	103.4	4.78

* *At -15 ^oC Evaporator Temperature, and 30 ^oC Condenser Temperature*

** *COP_{carnot} = Coefficient of Performance = $Temp._{Evap.}$ / ($Temp._{Cond.}$ -$Temp_{Evap}$)*

The choice of refrigerant and the required cooling temperature and load determine the choice of compressor, as well as the design of the condenser, evaporator, and other auxiliaries. Additional factors such as ease of maintenance, physical space requirements and availability of utilities for auxiliaries (water, power, etc.) also influence component selection.

2.2 Vapour Absorption Refrigeration System

2.2.1 Description

The vapour absorption refrigeration system consists of:

- Absorber: Absorption of refrigerant vapour by a suitable absorbent or adsorbent, forming a strong or rich solution of the refrigerant in the absorbent/ adsorbent
- Pump: Pumping of the rich solution and raising its pressure to the pressure of the condenser
- Generator: Distillation of the vapour from the rich solution leaving the poor solution for recycling

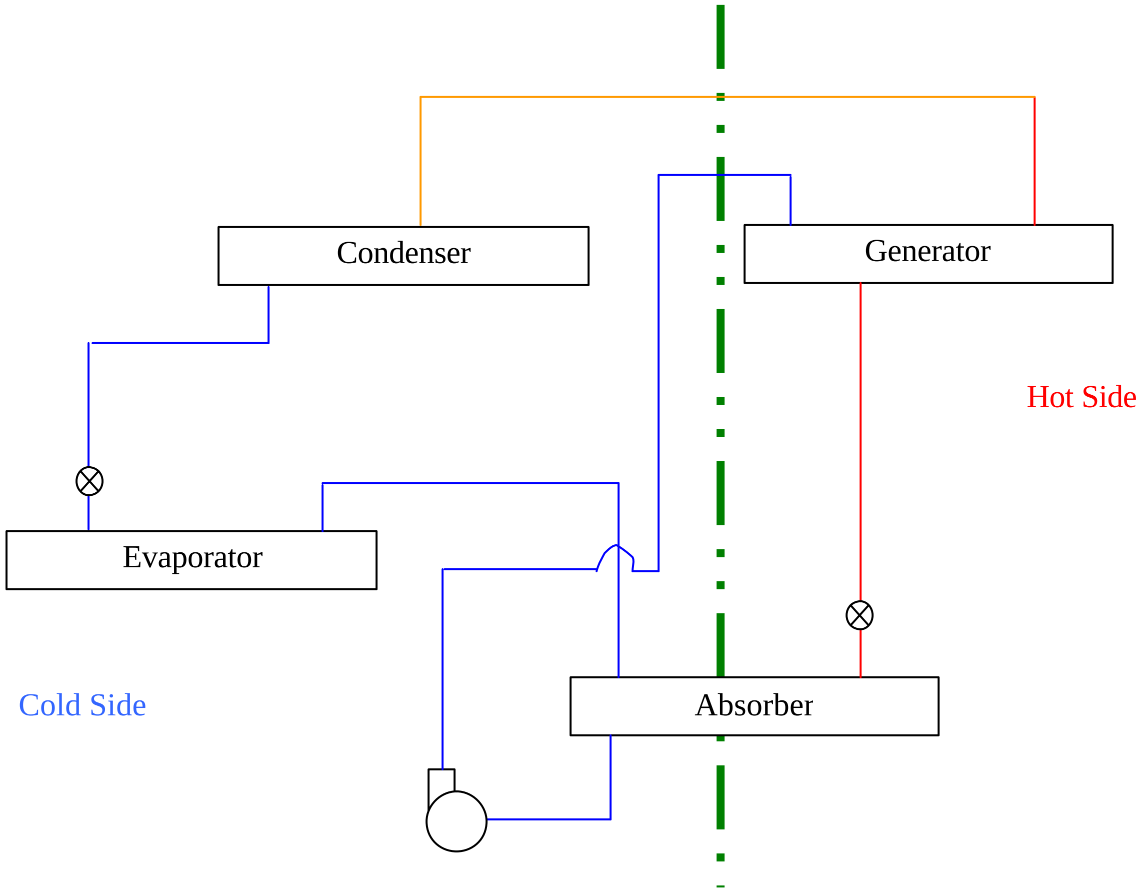


Figure 5: A simple schematic of a vapour absorption refrigeration system

The absorption chiller is a machine, which produces chilled water by using heat such as steam, hot water, gas, oil etc. Chilled water is produced based on the principle that liquid (i.e. refrigerant, which evaporates at a low temperature) absorbs heat from its surroundings when it evaporates. Pure water is used as refrigerant and lithium bromide solution is used as absorbent.

Heat for the vapour absorption refrigeration system can be provided by waste heat extracted from the process, diesel generator sets etc. In that case absorption systems require electricity for running pumps only. Depending on the temperature required and the power cost, it may even be economical to generate heat / steam to operate the absorption system.

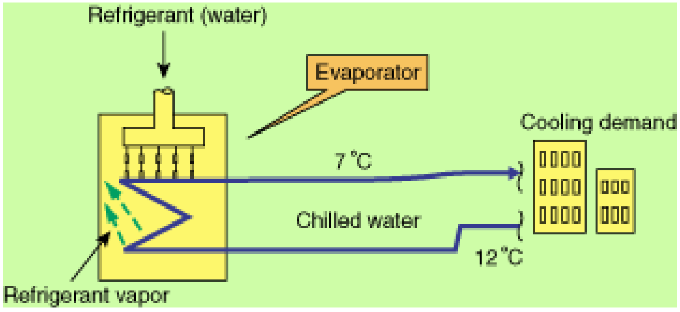
A description of the absorption refrigeration concept is given below (references for the pictures are unknown).

Evaporator

The refrigerant (water) evaporates at around 4°C under a high vacuum condition of 754 mm Hg in the evaporator.

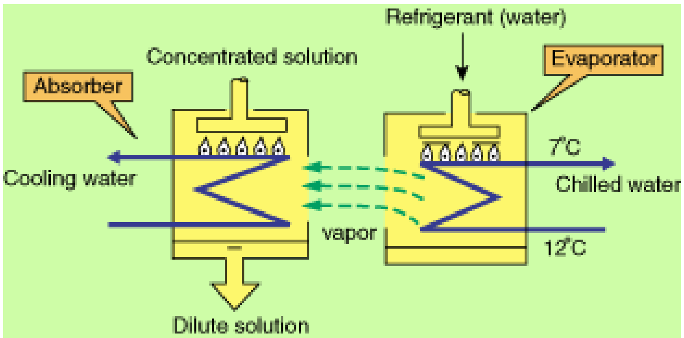
Chilled water goes through heat exchanger tubes in the evaporator and transfers heat to the evaporated refrigerant.

The evaporated refrigerant (vapor) turns into liquid again, while the latent heat from this vaporization process cools the chilled water (in the diagram from 12 °C to 7 °C). The chilled water is then used for cooling purposes.



Absorber

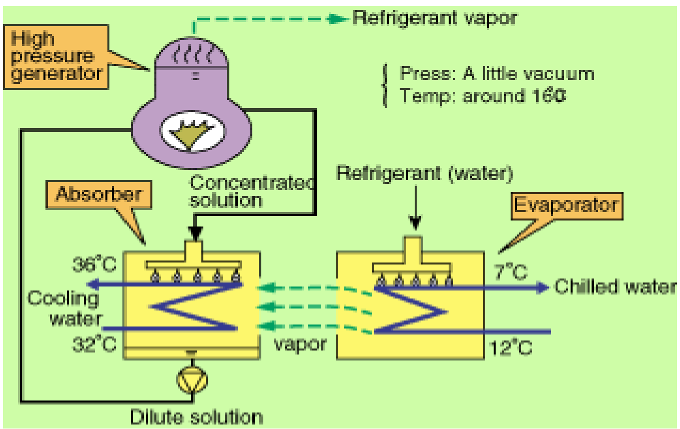
In order to keep evaporating, the refrigerant vapor must be discharged from the evaporator and refrigerant (water) must be supplied. The refrigerant vapor is absorbed into lithium bromide solution, which is convenient to absorb the refrigerant vapor in the absorber. The heat generated in the absorption process is continuously removed from the system by cooling water. The absorption also maintains the vacuum inside the evaporator.

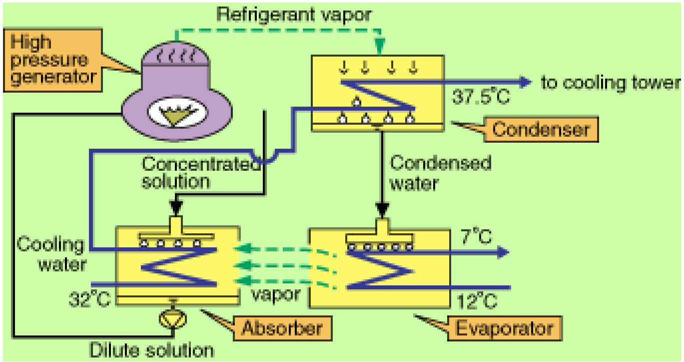


High Pressure Generator

As lithium bromide solution is diluted, the ability to absorb the refrigerant vapor reduces. In order to keep the absorption process going, the diluted lithium bromide solution must be concentrated again.

An absorption chiller is provided with a solution concentrating system, called a generator. Heating media such as steam, hot water, gas or oil perform the function of concentrating



solutions. The concentrated solution is returned to the absorber to absorb refrigerant vapor again.	
Condenser To complete the refrigeration cycle, and thereby ensuring the refrigeration takes place continuously, the following two functions are required 1. To concentrate and liquefy the evaporated refrigerant vapor, which is generated in the high-pressure generator. 2. To supply the condensed water to the evaporator as refrigerant (water) For these two functions a condenser is installed.	

Absorption refrigeration systems that use Li-Br-water as a refrigerant have a Coefficient of Performance (COP) in the range of 0.65 - 0.70 and can provide chilled water at 6.7 °C with a cooling water temperature of 30 °C. Systems capable of providing chilled water at 3 °C are also available. Ammonia based systems operate at above atmospheric pressures and are capable of low temperature operation (below 0°C). Absorption machines are available with capacities in the range of 10-1500 tons. Although the initial cost of an absorption system is higher than that of a compression system, operational costs are much lower if waste heat is used.

2.2.2 Evaporative cooling in vapor absorption refrigeration systems

There are occasions where air conditioning, which stipulates control of humidity of up to 50% for human comfort or for processes, can be replaced by a much cheaper and less energy intensive evaporative cooling.

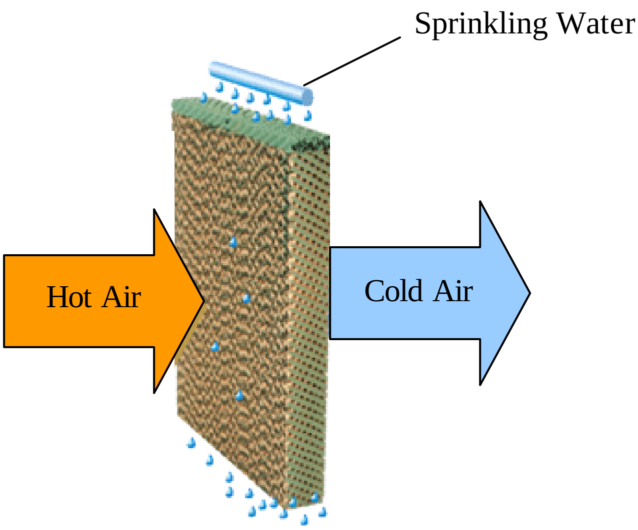


Figure 5. Schematic of evaporative cooling
Adapted from: Munters (2001)

The concept is very simple and is the same as that used in a cooling tower. Air is brought in close contact with water to cool it to a temperature close to the wet bulb temperature. The cool air can be used for comfort or process cooling. The disadvantage is that the air is rich in moisture. Nevertheless, it is an extremely efficient means of cooling at very low cost. Large commercial systems employ cellulose filled pads over which water is sprayed. The temperature can be controlled by controlling the airflow and the water circulation rate. The possibility of evaporative cooling is especially attractive for comfort cooling in dry regions. This principle is practiced in textile industries for certain processes.

3. ASSESSMENT OF REFRIGERATION AND AIR CONDITIONING

This section describes how the performance of refrigeration / air conditioning plants and be assessed.

3.1 Assessment of Refrigeration

3.1.1 TR

We start with the definition of TR.

- TR: the cooling effect produced is quantified as tons of refrigeration, also referred to as "chiller tonnage".
- $TR = Q \times C_p \times (T_i - T_o) / 3024$

  Where Q is mass flow rate of coolant in kg/hr
  C_p is coolant specific heat in kCal /kg deg C
  T_i is inlet, temperature of coolant to evaporator (chiller) in ^0C
  T_o is outlet temperature of coolant from evaporator (chiller) in ^0C.
  1 TR of refrigeration = 3024 kCal/hr heat rejected

3.1.2 Specific Power Consumption

- The specific power consumption kW/TR is a useful indicator of the performance of a refrigeration system. By measuring the refrigeration duty performed in TR and the kW inputs, kW/TR is used as an energy performance indicator.
- In a centralized chilled water system, apart from the compressor unit, power is also consumed by the chilled water (secondary) coolant pump, the condenser water pump (for heat rejection to cooling tower) and the fan in the cooling tower. Effectively, the overall energy consumption would be the sum of:
  - Compressor kW
  - Chilled water pump kW
  - Condenser water pump kW
  - Cooling tower fan kW, for induced / forced draft towers
- The kW/TR, or the specific power consumption for a certain TR output is the sum of:
  - Compressor kW/TR
  - Chilled water pump kW/TR
  - Condenser water pump kW/TR
  - Cooling tower fan kW/TR

3.1.3 Coefficient of Performance

- The theoretical Coefficient of Performance (Carnot), ($COP_{Carnot,}$ a standard measure of refrigeration efficiency of an ideal refrigeration system) depends on two key system temperatures: evaporator temperature T_e and condenser temperature T_c. COP is given as:

$$COP_{Carnot} \quad = \quad T_e / (T_c - T_e)$$

This expression also indicates that higher COP_{Carnot} is achieved with higher evaporator temperatures and lower condenser temperatures. But COP_{Carnot} is only a ratio of temperatures, and does not take into account the type of compressor. Hence the COP normally used in industry is calculated as follows:

$$COP = \frac{\text{Cooling effect (kW)}}{\text{Power input to compressor (kW)}}$$

where the cooling effect is the difference in enthalpy across the evaporator and expressed as kW.

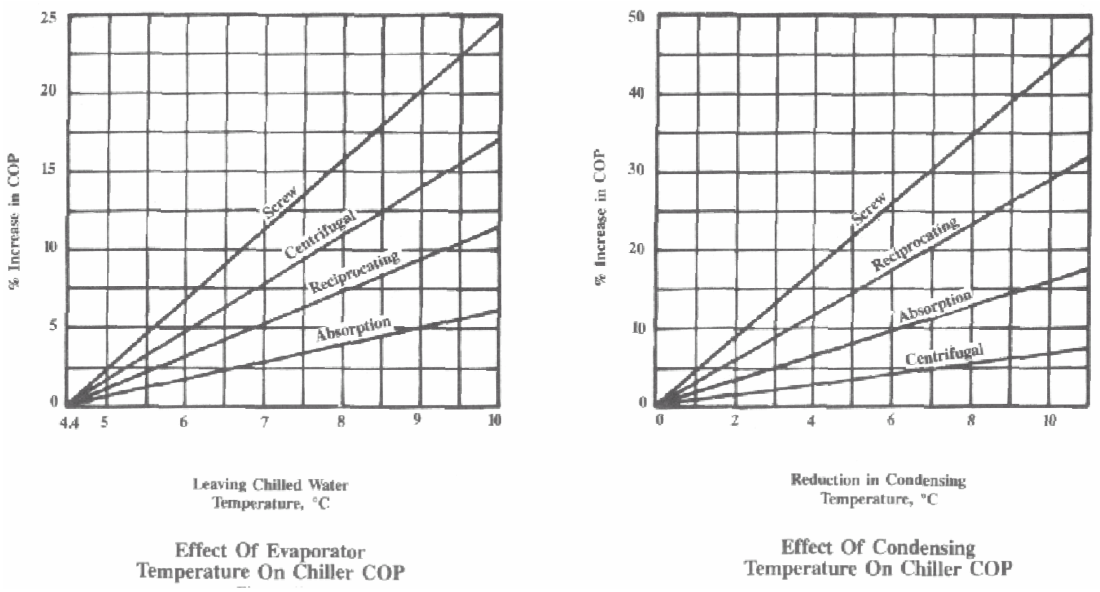


Figure 6: The effect of condensing temperature and evaporator temperature on the chiller (Bureau of Energy Efficiency, 2004)

3.2 Assessment of Air Conditioning

For air conditioning units, the airflow at the Fan Coil Units (FCU) or the Air Handling Units (AHU) can be measured with an anemometer. Dry bulb and wet bulb temperatures are measured at the inlet and outlet of the AHU or the FCU and the refrigeration load in TR is assessed as:

$$TR = \frac{Q \times ? \times (h_{in} - h_{out})}{3024}$$

Where, Q is the air flow in m^3/h
ρ is density of air kg/m^3
h_{in} is enthalpy of inlet air kCal/kg
h_{out} is enthalpy of outlet air kCal/kg

Use of psychometric charts can help to calculate h_{in} and h_{out} from dry bulb and wet bulb temperature values which are measured during trials by a whirling psychrometer. Power measurements at compressor, pumps, AHU fans, cooling tower fans can be taken with a portable load analyzer.

Estimation of the air conditioning load is also possible by calculating various heat loads, sensible and latent, based on inlet and outlet air parameters, air ingress factors, air flow, number of people and type of materials stored.

An indicative TR load profile for air conditioning is presented as follows:

- Small office cabins = $0.1\ TR/m^2$
- Medium size office i.e., 10 – 30 people occupancy with central A/C = $0.06\ TR/m^2$
- Large multistoried office complexes with central A/C = $0.04\ TR/m^2$

3.3 Considerations when Assessing Performance

3.3.1 Accuracy of flow and temperature measurements

In a field performance assessment, accurate instruments are required to measure the inlet and outlet temperatures of chilled water and condenser water, preferably with a count of at least 0.1 ^oC. Flow measurements of chilled water can be made with an ultrasonic flow meter directly or can be determined based on pump duty parameters. Adequacy checks of chilled water are often needed and most units are designed for a typical 0.68 m^3/hr per TR (3 gpm/TR) chilled water flow. Condenser water flow can also be measured with a non-contact flow meter directly or determined by using pump duty parameters. Adequacy checks of condenser water are also needed often, and most units are designed for a typical 0.91 m^3/hr per TR (4 gpm / TR) condenser water flow.

3.3.2 Integrated Part Load Value (IPLV)

Although the kW/ TR can serve as an initial reference, it should not be taken as an absolute since this value is based on a 100% equipment capacity level and on design conditions that are considered most critical. These conditions may only occur during % of the total time the equipment is in operation throughout the year. For this reason, it is essential to have data that reflects how the equipment operates with partial loads or under conditions that demand less than 100% capacity. To overcome this, an average kW/TR with partial loads has to be determined, which is called the Integrated Part Load Value (IPLV).

The IPLV is the most appropriate reference, although not considered the best, because it only captures four points within the operational cycle: 100%, 75%, 50% and 25%. Furthermore, it assigns the same weight to each value, and most equipment operate between 50% and 75% of their capacity. This is why it is so important to prepare a specific analysis for each case that addresses the four points mentioned, as well as developing a profile of the heat exchanger's operations during the year.

4. ENERGY EFFICIENCY OPPORTUNITIES

This section includes areas for energy conservation in refrigeration plants.

4.1 Optimization of Process Heat Exchangers

There is a tendency to apply high safety margins to operations, which influence the compressor suction pressure / evaporator set point. For instance, a process-cooling requirement of 15 ^{o}C would need chilled water at a lower temperature, but the range can vary from 6 ^{o}C to about 10 ^{o}C. At chilled water of 10 ^{o}C, the refrigerant side temperature has to be lower (about –5^{o}C to +5^{o}C). The refrigerant temperature determines the corresponding suction pressure of the refrigerant, which in turn determines the inlet duty conditions for the refrigerant compressor. Applying the optimum / minimum driving force (temperature difference) can thus help to reach the highest possible suction pressure at the compressor, thereby minimizing energy consumption. This requires proper sizing of heat transfer areas of process heat exchangers and evaporators as well as rationalizing the temperature requirement to highest possible value. A 1^{o}C raise in evaporator temperature can save almost 3 % of the power consumed. The TR capacity of the same machine will also increase with the evaporator temperature, as given in the table below.

Table 3. Typical values illustrating the effect of variation in evaporator temperature on the compressor power consumption (National Productivity Council, unpublished)

Evaporator Temperature (^{0}C)	Refrigeration Capacity* (tons)	Specific Power Consumption	Increase in kW/ton (%)
5.0	67.58	0.81	-
0.0	56.07	0.94	16.0
-5.0	45.98	1.08	33.0
-10.0	37.20	1.25	54.0
-20.0	23.12	1.67	106.0

* *Condenser temperature 40^{0}C*

In order to rationalize the heat transfer areas, the heat transfer coefficient on the refrigerant side can range from 1400 – 2800 watts /m^{2}K. The refrigerant side heat transfer areas are of the order of 0.5 m^{2}/TR and above in evaporators.

Condensers in a refrigeration plant are critical equipment that influence the TR capacity and power consumption demands. For any refrigerant, the condensation temperature and corresponding condenser pressure are dependent on the heat transfer area, the effectiveness of heat exchange and the type of cooling chosen. A lower condensation temperature means that the compressor has to work between a lower pressure differential as the discharge pressure is fixed by design and performance of the condenser.

The choice of condensers in practice is between air-cooled, air-cooled with water spray, and heat exchanger cooled. Larger shell and tube heat exchangers that are used as condensers and that are equipped with good cooling tower operations allow operation at low discharge pressure values and improve the TR capacity of the refrigeration plant.

If the refrigerant R22 is used in a water-cooled shell and tube condenser then the discharge pressure is 15 kg/cm^2. If the same refrigerant is used in an air-cooled condenser then the discharge pressure is 20 kg/cm^2. This shows how much additional compression duty is required, which results in almost 30 % additional energy consumption by the plant.

One of the best options at the design stage would be to select large sized (0.65 m^2/TR and above) shell and tube condensers with water-cooling, rather than less expensive alternatives like air cooled condensers or water spray atmospheric condenser units.

The effect of condenser temperature on refrigeration plant energy requirements is given in the table below

Table 7. Typical values illustrating the effect of variation in condenser temperature on compressor power consumption (National Productivity Council, unpublished)

Condensing Temperature (^{0}C)	Refrigeration Capacity (tons)	Specific Power Consumption (kW / TR)	Increase in kW/TR (%)
26.7	31.5	1.17	-
35.0	21.4	1.27	8.5
40.0	20.0	1.41	20.5

* *Reciprocating compressor using R-22 refrigerant.*
Evaporator temperature -10^{0}C

4.2 Maintenance of Heat Exchanger Surfaces

Once compressors have been purchased, effective maintenance is the key to optimizing power consumption. Heat transfer can also be improved by ensuring proper separation of the lubricating oil and the refrigerant, timely defrosting of coils, and increasing the velocity of the secondary coolant (air, water, etc.). However, increased velocity results in larger pressure drops in the distribution system and higher power consumption in pumps / fans. Therefore, careful analysis is required to determine the optimum velocity.

Fouled condenser tubes force the compressor to work harder to attain the desired capacity. For example, a 0.8 mm scale build-up in condenser tubes can increase energy consumption by as much as 35 %. Similarly, fouled evaporators (due to residual lubricating oil or infiltration of air) result in increased power consumption. Equally important is proper selection, sizing, and maintenance of cooling towers. A reduction of 0.55^{o}C in temperature of the water returning from the cooling tower reduces compressor power consumption by 3%.

Table 8. Typical values illustrating the effect of poor maintenance on compressor power consumption (National Productivity Council)

Condition	Evaporation Temp (^{0}C)	Condensation Temp (^{0}C)	Refrigeration Capacity* (tons)	Specific Power Consumption (kW/ton)	Increase in kW/Ton (%)
Normal	7.2	40.5	17.0	0.69	-
Dirty condenser	7.2	46.1	15.6	0.84	20.4
Dirty evaporator	1.7	40.5	13.8	0.82	18.3
Dirty condenser and evaporator	1.7	46.1	12.7	0.96	38.7

* *15 ton reciprocating compressor based system. The power consumption is lower than that for systems typically available in India. However, the percentage change in power consumption is indicative of the effect of poor maintenance.*

4.3 Multi-Staging For Efficiency

Efficient compressor operation requires that the compression ratio be kept low, to reduce discharge pressure and temperature. For low temperature applications involving high compression ratios, and for wide temperature range requirements, it is preferable (due to equipment design limitations) and often economical to employ multi-stage reciprocating machines or centrifugal / screw compressors.

There are two types of multi-staging systems, which are applicable to all types of compressors: compound and cascade. With reciprocating or rotary compressors, two-stage compressors are preferable for load temperatures from –20^{o}C to –58^{o}C, and with centrifugal machines for temperatures around –43^{o}C.

In a multi-stage operation, a first-stage compressor that sized to meet the cooling load, feeds into the suction of a second-stage compressor after inter-cooling of the gas. A part of the high-pressure liquid from the condenser is flashed and used for liquid sub-cooling. The second compressor, therefore, has to meet the load of the evaporator and the flash gas. A single refrigerant is used in the system, and the two compressors share the compression task equally. Therefore, a combination of two compressors with low compression ratios can provide a high compression ratio.

For temperatures in the range of –46^{o}C to –101^{o}C, cascaded systems are preferable. In this system, two separate systems using different refrigerants are connected so that one rejects heat to the other. The main advantage of this system is that a low temperature refrigerant, which has a high suction temperature and low specific volume, can be selected for the low-stage to meet very low temperature requirements.

4.4 Matching Capacity to System Load

During part-load operation, the evaporator temperature rises and the condenser temperature falls, effectively increasing the COP. But at the same time, deviation from the design operation point and the fact that mechanical losses form a greater proportion of the total power negate the effect of improved COP, resulting in lower part-load efficiency.

Therefore, consideration of part-load operation is important, because most refrigeration applications have varying loads. The load may vary due to variations in temperature and process cooling needs. Matching refrigeration capacity to the load is a difficult exercise, requiring knowledge of compressor performance, and variations in ambient conditions, and detailed knowledge of the cooling load.

4.5 Capacity Control and Energy Efficiency

The capacity of compressors is controlled in a number of ways. Capacity control of reciprocating compressors through cylinder unloading results in incremental (step-by-step) modulation. In contrast, continuous modulation occurs in centrifugal compressors through vane control and in screw compressors through sliding valves. Therefore, temperature control requires careful system design. Usually, when using reciprocating compressors in applications with widely varying loads, it is desirable to control the compressor by monitoring the return water (or other secondary coolant) temperature rather than the temperature of the water leaving the chiller. This prevents excessive on-off cycling or unnecessary loading / unloading of the compressor. However, if load fluctuations are not high, the temperature of the water leaving the chiller should be monitored. This has the advantage of preventing operation at very low water temperatures, especially when flow reduces at low loads. The outgoing water temperature should be monitored for centrifugal and screw chillers.

Capacity regulation through speed control is the most efficient option. However, when employing speed control for reciprocating compressors, it should be ensured that the lubrication system is not affected. In the case of centrifugal compressors, it is usually desirable to restrict speed control to about 50 % of the capacity to prevent surging. Below 50%, vane control or hot gas bypass can be used for capacity modulation.

The efficiency of screw compressors operating at part load is generally higher than either centrifugal compressors or reciprocating compressors, which may make them attractive in situations where part-load operation is common. Screw compressor performance can be optimized by changing the volume ratio. In some cases, this may result in higher full-load efficiencies as compared to reciprocating and centrifugal compressors. Also, the ability of screw compressors to tolerate oil and liquid refrigerant slugs makes them preferred in some situations.

4.6 Multi-level Refrigeration for Plant Needs

The selection of refrigeration systems also depends on the range of temperatures required in the plant. For diverse applications requiring a wide range of temperatures, it is generally more economical to provide several packaged units (several units distributed throughout the plant) instead of one large central plant. Another advantage would be the flexibility and reliability. The selection of packaged units could also be made depending on the distance at which cooling loads need to be met. Packaged units at load centers reduce distribution losses in the system. Despite the advantages of packaged units, central plants generally have lower power consumption since at reduced loads power consumption can reduce significantly due to the large condenser and evaporator surfaces.

Many industries use a bank of compressors at a central location to meet the load. Usually the chillers feed into a common header from which branch lines are taken to different locations in the plant. In such situations, operation at part-load requires extreme care. For efficient operation, the cooling load, and the load on each chiller must be monitored closely. It is more efficient to operate a single chiller at full load than to operate two chillers at part-load. The distribution system should be designed such that individual chillers can feed all branch lines. Isolation valves must be provided to ensure that chilled water (or other coolant) does not flow through chillers not in operation. Valves should also be provided on branch lines to isolate sections where cooling is not required. This reduces pressure drops in the system and reduces power consumption in the pumping system. Individual compressors should be loaded to their full capacity before operating the second compressor. In some cases it is economical to provide a separate smaller capacity chiller, which can be operated on an on-off control to meet peak demands, with larger chillers meeting the base load.

Flow control is also commonly used to meet varying demands. In such cases the savings in pumping at reduced flow should be weighed against the reduced heat transfer in coils due to reduced velocity. In some cases, operation at normal flow rates, with subsequent longer periods of no-load (or shut-off) operation of the compressor, may result in larger savings.

4.7 Chilled Water Storage

Depending on the nature of the load, it is economical to provide a chilled water storage facility with very good cold insulation. Also, the storage facility can be fully filled to meet the process requirements so that chillers need not be operated continuously. This system is usually economical if small variations in temperature are acceptable. This system has the added advantage of allowing the chillers to be operated at periods of low electricity demand to reduce peak demand charges. Low tariffs offered by some electric utilities for operation at nighttime can also be taken advantage of by using a storage facility. An added benefit is that lower ambient temperature at night lowers condenser temperature and thereby increases the COP.

If temperature variations cannot be tolerated, it may not be economical to provide a storage facility since the secondary coolant would have to be stored at a temperature much lower than required to provide for heat gain. The additional cost of cooling to a lower temperature may offset the benefits. The solutions are case specific. For example, in some cases it may be possible to employ large heat exchangers, at a lower cost burden than low temperature chiller operation, to take advantage of the storage facility even when temperature variations are not acceptable. Ice bank systems, which store ice rather than water, are often economical.

4.8 System Design Features

In overall plant design, adoption of good practices improves the energy efficiency significantly. Some areas for consideration are:

- Design of cooling towers with FRP impellers and film fills, PVC drift eliminators, etc.
- Use of softened water for condensers in place of raw water.
- Use of economic insulation thickness on cold lines, heat exchangers, considering cost of heat gains and adopting practices like infrared thermography for monitoring - applicable especially in large chemical / fertilizer / process industry.
- Adoption of roof coatings / cooling systems, false ceilings / as applicable, to minimize refrigeration load.
- Adoption of energy efficient heat recovery devices like air to air heat exchangers to pre-cool the fresh air by indirect heat exchange; control of relative humidity through indirect heat exchange rather than use of duct heaters after chilling.

Adopting of variable air volume systems; adopting of sun film application for heat reflection; optimizing lighting loads in the air conditioned areas; optimizing number of air changes in the air conditioned areas are few other examples.

5. OPTION CHECKLIST

This section includes most important energy efficiency options.

- Cold Insulation: Insulate all cold lines / vessels using economic insulation thickness to minimize heat gains; and choose appropriate (correct) insulation.
- Building Envelope: Optimize air conditioning volumes by measures such as use of false ceiling and segregation of critical areas for air conditioning by air curtains.
- Building Heat Loads Minimization: minimize the air conditioning loads by measures such as roof cooling, roof painting, efficient lighting, pre-cooling of fresh air by air- to-air heat exchangers, variable volume air system, optimal thermo-static setting of temperature of air conditioned spaces, sun film applications, etc.
- Process Heat Loads Minimization: Minimize process heat loads in terms of TR capacity as well as refrigeration level, i.e., temperature required, by way of:
  - Flow optimization
  - Heat transfer area increase to accept higher temperature coolant
  - Avoiding wastages like heat gains, loss of chilled water, idle flows.
  - Frequent cleaning / de-scaling of all heat exchangers
- At the Refrigeration A/C Plant Area:
  - Ensure regular maintenance of all A/C plant components as per manufacturer guidelines.
  - Ensure adequate quantity of chilled water and cooling water flows and avoid bypass flows by closing valves of idle equipment.
  - Minimize part load operations by matching loads and plant capacity on line and adopt variable speed drives for varying process load.
  - Make efforts to continuously optimize condenser and evaporator parameters for minimizing specific energy consumption and maximizing capacity.
  - Adopt a VAR system where economics permit as a non-CFC solution.
- Ensure that the AC does not get overloaded and check the fuse or circuit breaker if the AC does not operate.
- Replace or clean the filter and clean the evaporator and condenser coils regularly, for the air conditioner to cool efficiently.
- Clean the thermostat regularly and replace it if necessary.
- If a compressor does not work properly, call a service person immediately
- Any noise that your AC makes needs to be checked by your mechanic.
- A good air filter will extend the life of your air conditioner because the important parts, like the blower assembly, the cooling coil, and other inner parts will stay cleaner, operate more efficiently and last longer.
- Avoid frequent opening of doors/windows. A door kept open can result in doubling the power consumption of your AC.
- Ensure direct sunlight and heat do not enter the air-conditioned space, particularly in the afternoons.
- Most people believe that a thermostat set to a lower temperature than desired will force your air-conditioner to cool faster, not really, all it does, is make your air-conditioner operate for longer. Moreover, you will have an unnecessarily chilly room and wasted power. Every degree lower on the temperature setting results in an extra 3-4% of power consumed. Hence, once you found yourself a comfortable temperature and set the thermostat at that level, avoid changing the thermostat settings.

- Once an air-conditioning system has been designed and installed avoid any major change in the heat-load on the AC. This will add to wasted power
- A clogged drain line is usually caused by algae (the green moss-like stuff!) build-up inside the drain line. The air handler provides a cool, damp environment for development of molds and mildew and if left untreated these growths can spread into your ductwork. Get rid of these molds by using a disinfectant (consult your dealer). Make sure that the face of the cooling or evaporator coil is clean so that air can pass through freely.
- If you have an air return duct in a hot area such as an attic or garage, make sure that this duct is not broken, split, or disconnected and sucking in hot air.
- Window unit should tilt down slightly on the outside. The part that removes humidity (where water accumulates) is the front coil, which is inside your home. Normally, there is a trough and/or a drain tube that lets the water run to the rear of the unit. If the drain gets clogged, water will back up and leak inside. Ask your mechanic to clean the chassis and make sure all screws are tight.
- Heat load can be reduced by keeping a false ceiling in offices. Curtains/ blinds /sun film on windows reduces heat input into the room. Insulating the ceiling, which is exposed to the sun with 50-mm thermocole drastically, reduces heat input into the room.
- Check for duct leaks and crushed ductwork. All air leaks should be sealed with a good quality duct sealant (not duct tape).
- Inspect the chiller as recommended by the chiller manufacturer. Typically, this should be done at least quarterly.
- Routinely inspect for refrigerant leaks.
- Check the compressor operating pressures.
- Check all oil levels and pressures.
- Examine all motor voltages and amps.
- Check all electrical starters, contactors, and relays.
- Check all hot gas and unloader operations.
- Use superheat and subcooling temperature readings to obtain a chiller's maximum efficiency.
- Take discharge line temperature readings.

Some **"Rules of Thumb"** are:

- Refrigeration capacity reduces by 6 percent for every 3.5 °C increase in condensing temperature.
- Reducing condensing temperature by 5.5 °C results in a 20–25 percent decrease in compressor power consumption.
- A reduction of 0.55 °C in cooling water temperature at condenser inlet reduces compressor power consumption by 3 percent.
- 1 mm scale build-up on condenser tubes can increase energy consumption by 40 percent.
- 5.5 °C increase in evaporator temperature reduces compressor power consumption by 20–25 percent.

6. WORKSHEETS

This section includes following worksheets:

- Refrigeration & AC System Rated Specifications
- Refrigeration Plant Performance

Worksheet 1: REFRIGERATION AND AC SYSTEM RATED SPECIFICATIONS

Section no.	**Refrigeration compressor**	**Units**	**Machine reference**			
			1	**2**	**3**	**4**
1.	Make					
2.	Type					
3.	Capacity (of cooling)	TR				
4.	**Chiller:**					
A.	No. of tubes	--				
B.	Dia. of tubes	m				
C.	Total heat transfer area	m^2				
D.	Chilled water flow	m^3/hr				
E.	Chilled water temp. difference	°C				
5.	**Condenser:**					
A.	No. of tubes					
B.	Dia. of tubes					
C.	Total heat transfer area	m				
D.	Condenser water flow	m^3/hr				
E.	Condenser water temp. diff.	°C				
6.	**Chilled water pump:**					
A.	Nos.	--				
B.	Capacity	m^3/hr				
C.	Head developed	mWC				
D.	Rated power	kW				
E.	Rated efficiency	%				
7.	**Condenser water pump:**					
A.	Nos.	--				
B.	Capacity	m^3/hr				
C.	Head developed	mWC				
D.	Rated power	kW				
E.	Rated efficiency	%				

Worksheet 2: REFRIGERATION PLANT PERFORMANCE

No	Parameter reference	Units	Refrigeration compressor reference			
			1	2	3	4
1.	Chilled water flow (using a flow meter or assessed by level difference)	m^3/hr				
2.	Chilled water pump motor input power	kW				
3.	Chilled water pump suction pressure	kg/cm^2g				
4.	Chilled water pump discharge pressure	kg/cm^2g				
5.	Chiller water inlet temperature to chiller	°C				
6.	Chiller water outlet temperature from chiller	°C				
7.	Condenser water inlet temperature	°C				
8.	Condenser pump suction pressure	kg/cm^2				
9.	Condenser pump discharge pressure	kg/cm^2				
10.	Condenser water outlet temperature	°C				
11.	Chiller (evaporator) outlet refrigerant temperature	°C				
12.	Refrigerant pressure	kg/cm^2 (or psig)				
13.	Condenser inlet refrigerant temperature	°C				
14.	Refrigerant pressure	kg/cm^2 (or psig)				
15.	Actual cooling capacity [(1)*(6-5)/3024]	TR				
16.	COP [11/(10-11)]	--				
17.	Compressor motor input power	kW				
18.	Specific energy consumption	kW/TR				
19.	Input power to CT fan	kW				
20.	Input power to chilled water pumps in operation	kW				
21.	Input power to condenser water pumps in operation	kW				
22.	Overall system specific power consumption [(2+17+19+20)/15]	kW/TR				

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