

Module 3

**ENERGY EFFICIENCY  
IN  
ELECTRICAL SYSTEMS**



Sustainable and Renewable  
Energy Development Authority

**Sustainable and Renewable Energy Development Authority  
Power Division**

Second Edition'2021



### ***Preamble***

In order to ensure energy efficiency and conservation and to determine the future course of action, Sustainable and Renewable Energy Development Authority (SREDA) has developed the Energy Efficiency & Conservation Master Plan up to 2030 in 2016. According to this plan, the target of energy saving has been set 20% per GDP by 2030 which will be achieved by the use of energy efficient machinery and equipment as well as by improving energy management system in the demand side.

In order to achieve the above mentioned target & to ensure the energy efficiency and conservation in industrial & commercial sector, SREDA has formulated the Energy Audit Regulation'2018. Based on this regulation, SREDA will conduct the Energy Auditor Certification Examination to create energy auditors and energy managers in Bangladesh.

SREDA has prepared the following modules as reading material for four paper examinations in cooperation with various National and Foreign partner organizations.

Module No	Examination Paper	Subject
Module 01	Paper 01	Fundamentals of Energy Management and Energy Audit
Module 02	Paper 02	Energy Efficiency in Thermal Systems
Module 03	Paper 03	Energy Efficiency in Electrical Systems
Module 04	Paper 04	Energy Performance Assessment for Equipment and Utility Systems

This module 03 on Energy Efficiency in Electrical Systems is the reading material for the preparation of Paper 03 Examination for prospective candidates.

We hope that these modules will also act as valuable resource for practicing engineers in comprehending and implementing energy efficiency measures in the facilities.

Since it is the first iteration of these modules, in some areas there might be a lack of information. Up-gradation of these modules is an ongoing process. Any suggestion and comments (please email to [ad.eaa@sreda.gov.bd](mailto:ad.eaa@sreda.gov.bd) ) on the contents of those modules will be highly appreciated.

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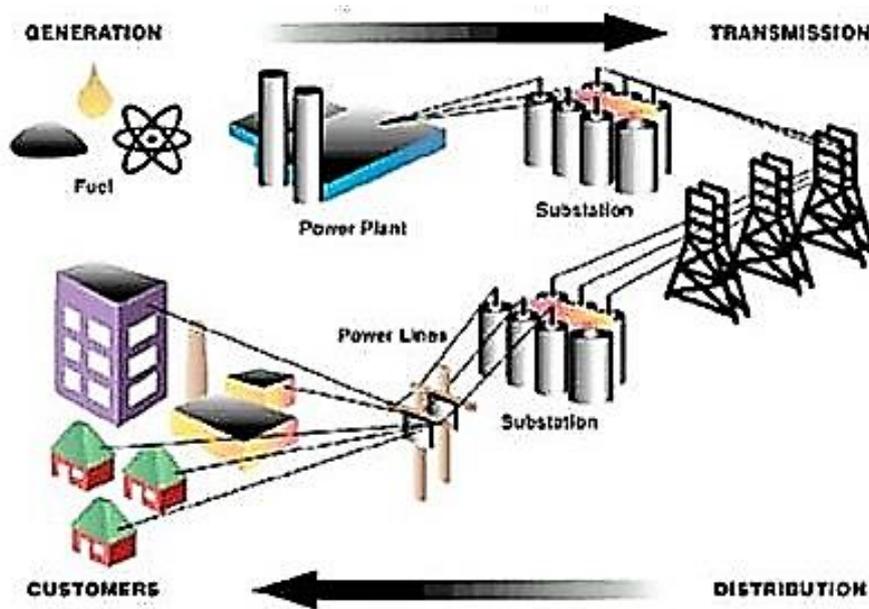
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# Chapter 1: Electrical Systems

## 1.1 Introduction to Electric Power Supply Systems

An electric power supply system in a country is comprised of generating units that produce electricity; high voltage transmission lines that transport electricity over long distances; distribution lines that deliver the electricity to customers; substations that connect the pieces to each other; and energy control centers to coordinate the operation of the components. Figure 1.1 shows a simple electric supply system with transmission and distribution network and linkages from electricity sources to end-user.



*Figure 1.1: Electric supply system*

### Power Generation Plant

Fossil fuels such as coal, oil and Natural Gas, nuclear energy, and falling water (hydel) are commonly used energy sources in the power generating plant. A wide and growing variety of unconventional generation technologies and fuels also have been developed, including cogeneration, solar energy, wind generators, and waste material. About 58% of power generation capacity in Bangladesh is from Natural gas based power plant.

The Installed capacity of Power Generation (by plant type) as on Sept, 2018 is given below:

<i>Plant Type</i>	<i>Installed Capacity (MW)</i>
<b><i>Power Import</i></b>	<b><i>1160</i></b>
<b><i>Solar PV</i></b>	<b><i>3</i></b>
<b><i>Hydro</i></b>	<b><i>230</i></b>
<b><i>Steam Turbine</i></b>	<b><i>2404</i></b>
<b><i>Combined Cycle (CC)</i></b>	<b><i>5871</i></b>
<b><i>Gas Turbine (GT)</i></b>	<b><i>1322</i></b>
<b><i>Reciprocating Engine (RE)</i></b>	<b><i>6053</i></b>
<b><i>Total*</i></b>	<b><i>17043</i></b>

\*Including Captive Power & Renewable Energy Total Installed Capacity (17,043 + 2,800+290) = 20,133 MW

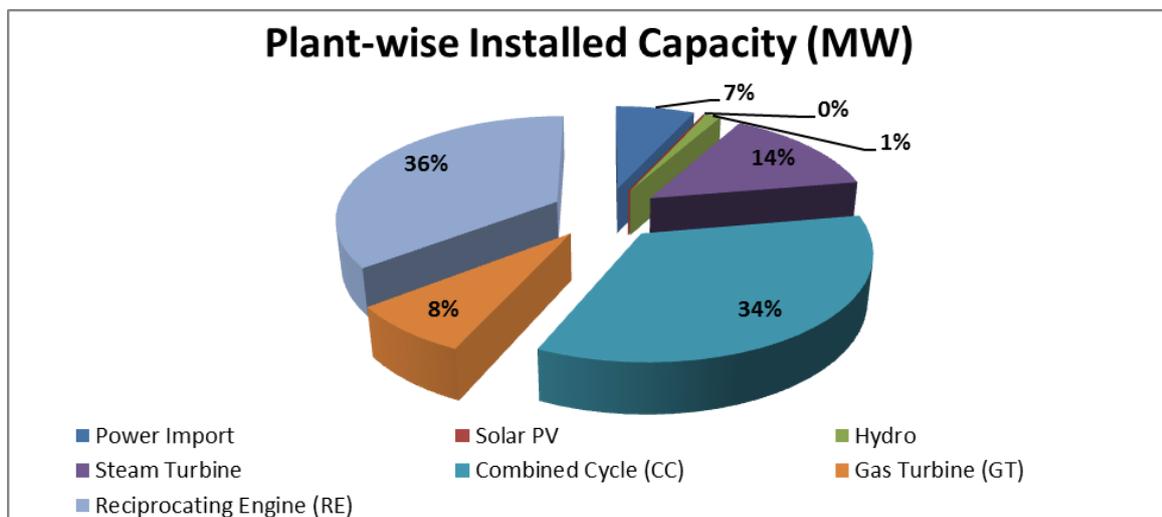


Figure 1.2 a: Primary Grid System in Bangladesh

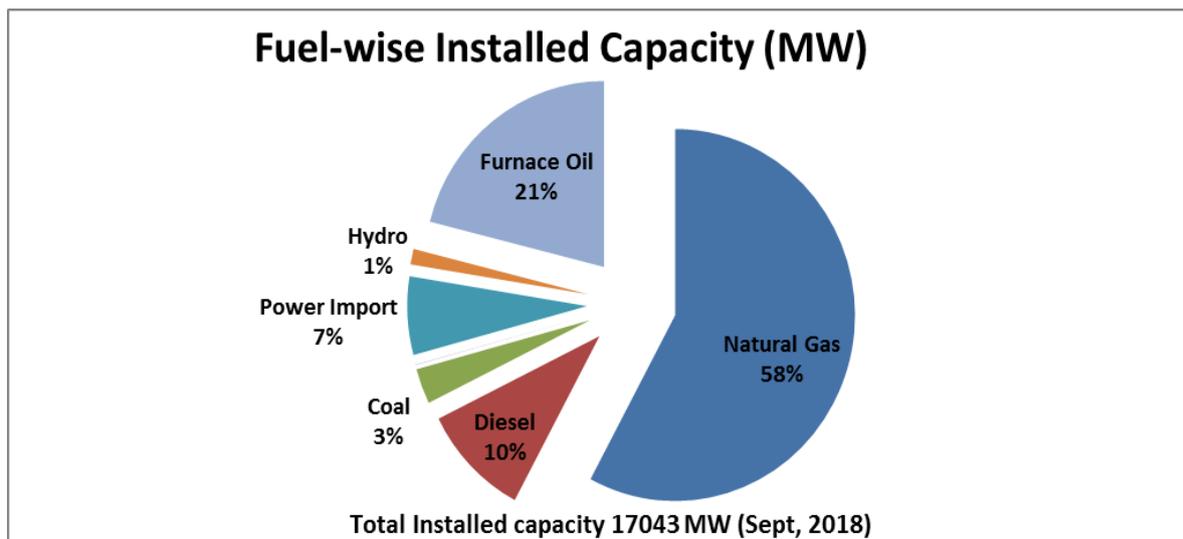


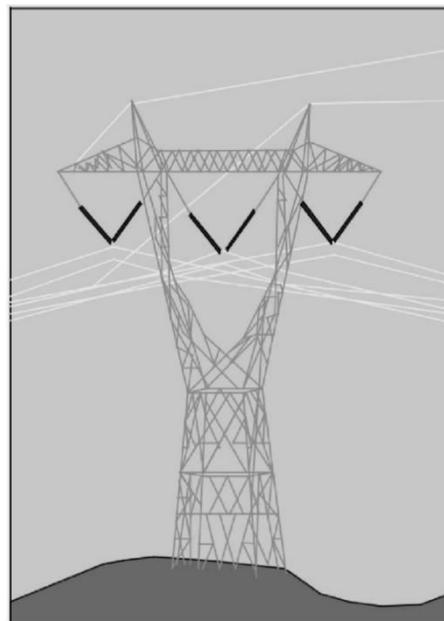
Figure 1.2 b: Primary Grid System in Bangladesh

The peak demand is expected to be about 17,304 MW in FY2020 and 25,199 MW in 2025 and 33,708 MW in 2030 based on 7% GDP growth rate.

The maximum power station in Bangladesh run via natural gas, some of them are run by HFO (Heavy fuel oil). Mainly Diesel oil and Furnace oil are used for thermal power plant.

### **Transmission and distribution lines**

As per Grid Code of Bangladesh, allowable range for frequency variation is 49.0 to 51.0 Hz. Power plants typically produce 50 cycle/second (Hertz) alternating-current (AC) electricity with Terminal voltage of different generators are 11 KV, 11.5 KV and 15.75 KV. The transmission and distribution network include sub-stations, lines and distribution transformers. At the power plant, the 3-phase voltage is stepped up to a higher voltage for transmission on cables strung on cross-country towers. High voltage transmission is used so that smaller, more economical wire sizes can be employed to carry the lower current and to reduce losses. Sub-stations, containing step-down transformers, reduce the voltage for distribution to industrial users. The voltage is further reduced for commercial facilities. Electricity must be generated, as and when it is needed since electricity cannot be stored virtually in the system here is no difference between a transmission line and a distribution line except for the voltage level and power handling capability. Transmission lines are usually capable of transmitting large quantities of electric energy over great distances. They operate at high voltages.



*Figure 1.3 Cross Country*

*Tower*

Distribution lines carry limited quantities of power over shorter distances. High voltage (HV) and extra high voltage (EHV) transmission is the next stage from power plant to transport A.C. power over long distances at voltages like; 400kV, 230 kV and 132 kV. These are called as the primary grid system. Where transmission is over 1000KM, high voltage direct current transmission (HVDC) is also favored to minimize the losses.

Sub-transmission network at 230 kV, 132kV constitutes the next link towards the end user. High voltage transmission network that transmits the power to grid substation transformers to be stepped down at 33 kV, 11 kV and 0.4 kV for delivery to the consumers of various categories.

Distribution at 11kV/6.6kV/3.3kV constitutes the last link to the consumer, who is connected directly or through transformers depending upon the drawl level of service.

Voltage drop in the line is in relation to the resistance and reactance of line, length and the current drawn. For the same quantity of power handled, lower the voltage, higher the current drawn and higher the voltage drop. The current drawn is inversely proportional to the voltage level for the same quantity of power handled.

The power loss in line is proportional to resistance and square of current. (i.e.  $P_{Loss}=I^2R$ ). Higher voltage transmission and distribution thus would help to minimize line voltage drop in the ratio of voltages, and the line power loss in the ratio of square of voltages. For instance, if distribution of power is raised from 11 kV to 33 kV, the voltage drop would be lower by a factor 1/3 and the line loss would be lower by a factor  $(1/3)^2$  i.e., 1/9. Lower voltage transmission and distribution also calls for larger quantity of conductor on account of current handling capacity needed.



**Figure 1.4: Primary Grid System in Bangladesh**

(Source: PGCB)

## Cascade Efficiency

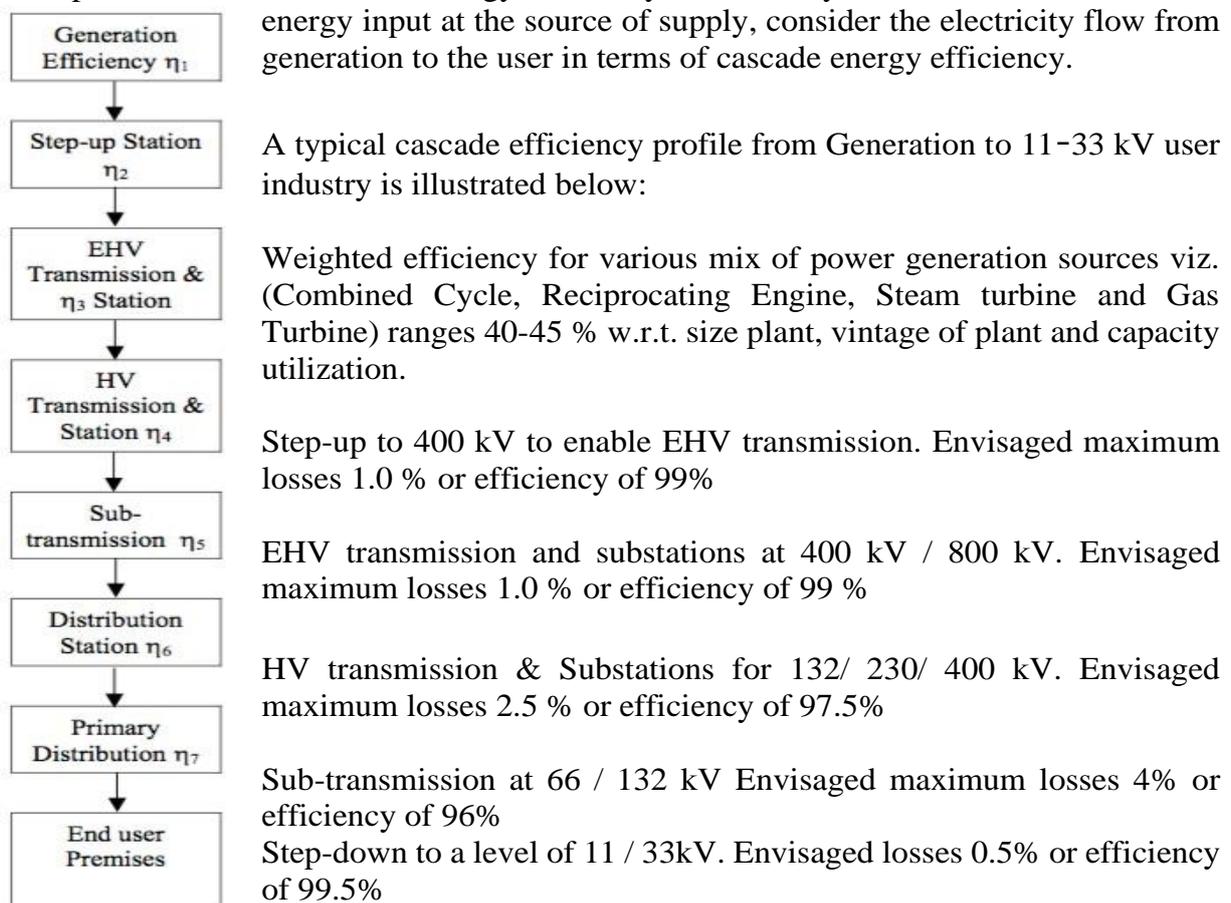
The primary function of transmission and distribution equipment is to transfer power economically and reliably from one location to another. Conductors in the form of wires and cables strung on towers and poles carry the high-voltage, AC electric current. A large number of copper or aluminum conductors are used to form the transmission path. The resistance of the long-distance transmission conductors is to be minimized. Energy loss in transmission lines is wasted in the form of  $I^2R$  losses.

Capacitors are used to correct power factor by causing the current to lead the voltage. When the AC currents are kept in phase with the voltage, operating efficiency of the system is maintained at a high level.

Circuit-interrupting devices are switches, relays, circuit breakers, and fuses. Each of these devices is designed to carry and interrupt certain levels of current. Making and breaking the current carrying conductors in the transmission path with a minimum of arcing is one of the most important characteristics of this device. Relays sense abnormal voltages, currents, and frequency and operate to protect the system.

Transformers are placed at strategic locations throughout the system to minimize power losses in the T&D system. They are used to change the voltage level from low-to-high in step-up transformers and from high-to-low in step-down units. Since the power loss of a transmission line is based on  $I^2R$ , losses can be reduced to an acceptable value by stepping up the source voltage to a high value to proportionally reduce the source current.

The power source to end user energy efficiency link is a key factor which influences the energy input at the source of supply, consider the electricity flow from generation to the user in terms of cascade energy efficiency.



Distribution is finally link to end user at 11 / 33 kV. Envisaged losses maximum 5 % of efficiency of 95 %

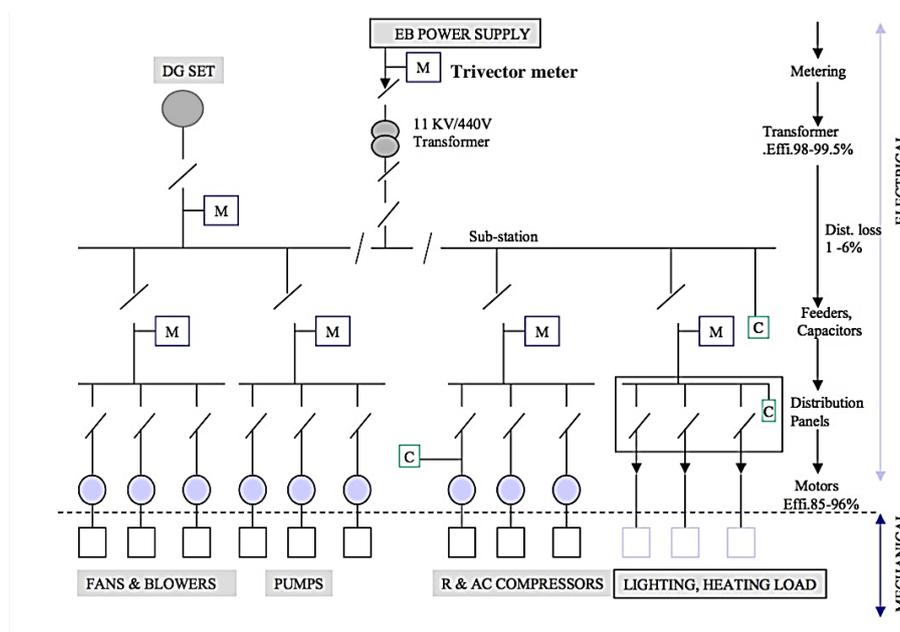
Cascade efficiency from Generation to end user =  $\eta_1 \times \eta_2 \times \eta_3 \times \eta_4 \times \eta_5 \times \eta_6 \times \eta_7$

The cascade efficiency in the T&D system from output of the power plant to the end use is 87% (i.e.,  $0.995 \times 0.99 \times 0.975 \times 0.96 \times 0.995 \times 0.95 = 87\%$ )

## Industrial End User

At the industrial end user premises again the plant network elements like transformers at receiving sub-station, switch gear, lines and cables, load-break switches, capacitors cause losses which affect the input received energy.

A typical plant single line diagram of electrical distribution system is shown in Figure 1.5



**Figure 1.5: Electrical Distribution System – Single Line Diagram**

The likely network elements that are encountered at industry up to the motor, i.e., pre-motor system can include:

- Outdoor circuit breakers with typical full load losses of 0.002 – 0.015%
- Receiving transformers with typical operating efficiency of 99 % or above.
- Medium voltage switch gear 5.15kV where maximum full load losses can be between 0.005 - 0.02%
- Load break switches where maximum of full load losses can be between 0.003– 0.025 %.
- Current limiting reactors can have a maximum full load losses ranging from 0.09% to 0.3 %.
- Medium voltage starters can have a maximum full load losses of 0.02% to 0.15%.

- Lines and cables can have a maximum loss ranging for 1% to 6% depending upon lengths, voltage levels, power factor, condition of network in plant
- Motor control centers can have a full load losses range from 0.01% to 0.4 %.
- Low voltage switchgear can have a full load loss ranging from 0.13 % to 0.34 %.

Thus, as per the links available in the in-plant distribution network, the cascade efficiency of pre-motor system can be computed, as a product of efficiencies of the actual links in cascade.

When problems like low voltage at motor terminals are encountered, this pre-motor system needs to be looked into, for improvement opportunities like;

- Relocating transformers close to load centers.
- Increasing cable / line size addition of parallel cable, and minimizing jumpers / loose connections and optimizing line lengths etc.
- Tap changing as needed at the transformers.
- Capacitor relocation close to load centers or motor terminals, as discussed later.
- Adopting best practices like infrared thermograph of distribution network, for identifying hotspots, which indicate potential of break-down/overloading etc., for attention / maintenance.

### **ONE Unit saved = TWO Units Generated**

After power generation at the plant, it is transmitted and distributed over a wide network. The standard technical losses are around 9.5% in Bangladesh (Efficiency=90.5%). But overall T & D (Transmission and Distribution Loss) losses range from 9–17%. All these may not constitute technical losses, since un-metered and pilferage are also accounted in this loss.

When the power reaches the industry, it is received by the transformer. The energy efficiency of the transformer is generally very high. Next it goes to the motor through internal plant distribution network. A typical distribution network efficiency including transformer is 95% and motor efficiency is about 90%. Another 30% (Efficiency=70%) is lost in the mechanical system which includes coupling / drive train, a driven equipment such as pump and flow control valves/throttling etc. Thus the overall energy efficiency becomes 50%. ( $0.90 \times 0.95 \times 0.9 \times 0.70 = 0.54$ , i.e 54% efficiency)

Hence one unit saved in the end user is equivalent to two units generated in the power plant. (1 Unit /0.5 Eff =2 Units)

### **1.2 Electricity billing**

The electricity billing by utilities for medium & large industries and enterprises (Category - F to Category – H) is often done on two-part tariff structure, i.e. one part for capacity (or demand) drawn and the second part for actual energy drawn during the billing cycle. Capacity or demand is in kW. The reactive energy (i.e.) KVARh drawn by the customer / service is also recorded and billed for in some utilities, because this would affect the load on the utility. Accordingly, utility charges for maximum demand, active energy and

reactive power drawn as reflected by the power factor in their billing structure. In addition, other fixed and variable expenses are also levied.

The tariff structure generally includes the following components:

- a) Maximum demand Charges
- b) These relate to maximum demand registered during month / billing period and corresponding rate of utility.
- c) Energy Charges
- d) These relate to energy (kilo watt hours) consumed during month /billing period and corresponding rates, often levied in slabs of use rates. Some utilities now charge on the basis of apparent energy (kVAh), which is a vector sum of kWh and KVARh.
- e) Power factor penalty or bonus rates, as levied by most utilities, are to contain reactive power drawn from grid.
- f) Fuel cost adjustment charges as levied by some utilities are to adjust the increasing fuel expenses over a base reference value.
- g) Electricity duty charges levied w.r.t units consumed.
- h) Meter rentals
- i) Lighting and fan power consumption is often at higher rates, levied sometimes on slab basis or on actual metering basis.
- j) Interruptible and adjustable rates like night tariff concessions and time of use rates are also prevalent in tariff concessions and time of use rates are also prevalent in tariff structure provisions of some utilities.

“Analysis of utility bill data and trending of the same helps energy manager to identify ways for electricity bill reduction through available provisions in tariff framework, apart from budgeting”.

The utility employs a tri vector meter of electromagnetic or the state of the art static electronic tri vector meter, for billing purposes. As apparent, active and reactive energy are vectorial in nature, the monitoring meter is called Tri vector meter. The minimum outputs from the electromagnetic meters are:

- Maximum demand registered during the month, which is measured in preset time intervals (say of 30 minute duration) and this is reset at the end of every billing cycle.
- Maximum demand (MD) in kW shall be registered using the technique of cumulating on integration period controlled by built-in process and the MD shall be continuously recorded and the highest shall be indicated. (Integration period: thirty minutes)
- Active energy in KWH during billing cycle
- Average PF for billing period.
- Reactive energy in KVARH during billing cycle and
- Apparent energy in KVAH during billing cycle

It is important to note that while maximum demand is recorded, it is not the instantaneous demand drawn, as is often misunderstood, but the time integrated demand over the predefined recording cycle.

As example, in an industry, if the drawl over a recording cycle of 30 minutes is:

- 2500 KW for 4 minutes
- 3600 KW for 12 minutes
- 4100 KW for 6 minutes
- 3800 KW for 8 minutes

The MD recorder will be computing MD as:

$$\frac{(2500 \times 4) + (3600 \times 12) + (4100 \times 6) + (3800 \times 8)}{30} = 3606.7 \text{ kW}$$

The month's maximum demand will be the highest among such demand values recorded over the month. The meter registers only if the value exceeds the previous maximum demand value and thus, even if, average maximum demand is low, the industry/facility has to pay for the maximum demand charges for the highest value registered during the month, even if it occurs for just one recording cycle duration i.e., 30 minutes during whole of the month (1440 such intervals in a month).

The LCD electronic tri-vector meters have some excellent provisions that can help the utility as well as the industry. These provisions include:

- Substantial memory for logging and recording all relevant events
- High accuracy of 0.2 class
- Amenability to time of use tariffs
- Tamper detection /recording
- Measurement of harmonics and Total Harmonic Distortion (THD)
- Long service life due to absence of moving parts
- Amenability for remote data access/downloads

Analysis and trending of purchased electricity for the past 12 months in a year and cost components can help the industry to identify key areas such as avoiding power factor penalty, contract demand reduction and availing time of the tariff advantages etc. for bill reduction within the utility tariff available framework.

Compiling monthly electricity use data, including all sources like; cogeneration, captive diesel power generation; doing cost comparison by source, linking power consumption to production by specific power consumption assessment, would serve as a powerful information tool for energy manager / auditor to optimize electricity costs for the industry or facility.

### **1.3 Electrical load management and maximum demand control**

#### **Need for Electrical load management**

1. In a macro perspective, the growth in the electricity use and diversity of end use segments in time of use has led to shortfalls in capacity to meet demand. As capacity addition is costly and only a long time prospect, better load management at user end helps to minimize peak demands on the utility infrastructure as well as better utilization of plant capacities.

2. The utilities use power tariff structure to influence end user in better load management through measures like time of use tariffs, penalties on exceeding allowed maximum demand, night tariff concessions etc. Load management is a powerful means of efficiency improvement both for end user as well as utility.
3. As the demand charges constitute a considerable portion of the electricity bill, from user angle too there is a need for integrated load management to effectively control the maximum demand.

## **Step By Step Approach for Maximum Demand Control**

### **1. Load Curve Generation**

Presenting the load demand of a consumer against time of the day is known as a 'load curve'. If it is plotted for the 24 hours of a single day, it is known as an 'hourly load curve' and if daily demands plotted over a month, it is called daily load curves. These types of curves are useful in predicting patterns of drawl, peaks and valleys and energy use trend in a section or in an industry or in a distribution network as the case may be.

The load factor can also be defined as the ratio of the energy consumed during a given period to the energy, which would have been used if the maximum demand had been maintained throughout that period.

Load Factor = (Energy Consumed in 24 Hours) / (Maximum Load Recorded × 24 Hours).

### **2. Rescheduling of Loads**

Rescheduling of large electric loads and equipment operations, in different shifts can be planned and implemented to minimize the simultaneous maximum demand. For this purpose, it is advisable to prepare an operation flow chart and a process chart. Analyzing these charts and with an integrated approach, it would be possible to reschedule the operations and running equipment in such away as to reduce the maximum demand and improve the load factor.

### **3. Staggering of Motor loads**

When running of motors of large capacities are involved, it is advisable to stagger the running of these motors with a suitable planning (as the process may permit) so as to minimize the simultaneous maximum demand (depending on the conditions of load) offered by these motors.

### **4. Storage of Products/in process material/ process utilities like refrigeration**

It is possible to reduce the maximum demand by building up storage capacity of products/ materials, water, chilled water / hot water using electricity during off peak periods. Off peak hour operations also help to save energy due to favorable conditions such as lower ambient temperature etc.

#### **Example 1.1**

Ice bank system is used in milk & dairy industry. Ice is made in lean period and used in peak load period and thus maximum demand is reduced.

#### **Shedding of Non-Essential Loads**

When the maximum demand tends to reach preset limit, shedding some of non-essential loads temporarily can help to reduce it. It is possible to install direct demand monitoring systems, which will switch off non-essential loads when a preset demand is reached. Simple systems give an alarm, and the loads are shed manually. Sophisticated microprocessor controlled systems are also available, which provide a wide variety of control options like:

- Accurate prediction of demand
- Graphical display of present load, available load, demand limit
- Visual and audible alarm
- Automatic load shedding in a predetermined sequence
- Automatic restoration of load
- Recording and metering

### **5. Operation of Captive Generation, Diesel Generation Sets and Gas Engines**

When Diesel/Gas generation sets are used to supplement the power supplied by the electric utilities, it is advisable to connect the Diesel or Gas sets for durations when demand reaches the peak value. This would reduce the load demand to a considerable extent and minimize the demand charges.

### **6. Reactive Power Compensation**

The maximum demand can also be reduced at the plant level by using capacitor banks and maintaining the optimum power factor. Capacitor banks are available with microprocessor based control systems. These systems switch on and off the capacitor banks to maintain the desired Power factor of system and optimize maximum demand thereby.

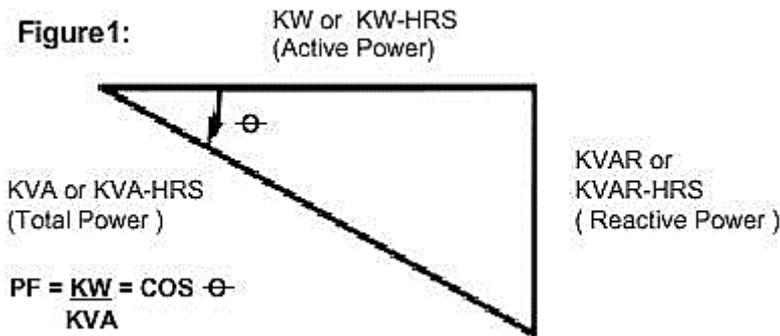
## **1.4 Power factor improvement and benefits**

### **Power factor Basics**

In all industrial electrical distribution systems, the predominant loads are resistive and inductive. Resistive loads are incandescent lighting and resistance heating. In case of pure resistive loads, the voltage (V), current (I), resistance (R) relations are linearly related, i.e.  $V = I \times R$  and  $kW = V \times I$

Inductive loads are A.C. Motors, induction furnaces, transformers and ballast-type lighting. Inductive loads require two kinds of power: (1) active (or working) power to perform the work and (2) reactive power to create and maintain electro-magnetic fields. The vector sum of the active power and reactive power make up the total (or apparent) power used. This is the power generated by the Utilities (Distribution companies) for the user to perform a given amount of work.

- Active power is measured in KW (Kilo Watts)
- Reactive power is measured in KVAR (Kilo Volt-Amperes Reactive)
- Total Power is measured in KVA (Kilo Volts-Amperes)



*Figure 1.6: Power factor*

The active power (shaft power required or true power required) in kW and the reactive power required (KVAR) are  $90^\circ$  apart vectorically in a pure inductive circuit i.e., reactive power KVAR lagging the active kW. The vector sum of the two is called the apparent power or kVA, as illustrated above and the kVA reflects the actual electrical load on distribution system.

The ratio of kW to kVA is called the power factor which is always less than or equal to unity. Theoretically, when electric utilities supply power, if all loads have unity power factor, maximum power can be transferred for the same distribution system capacity. However, as the loads are inductive in nature with the power factor ranging from 0.2 to 0.9, the electrical distribution network is stressed for capacity at low power factors.

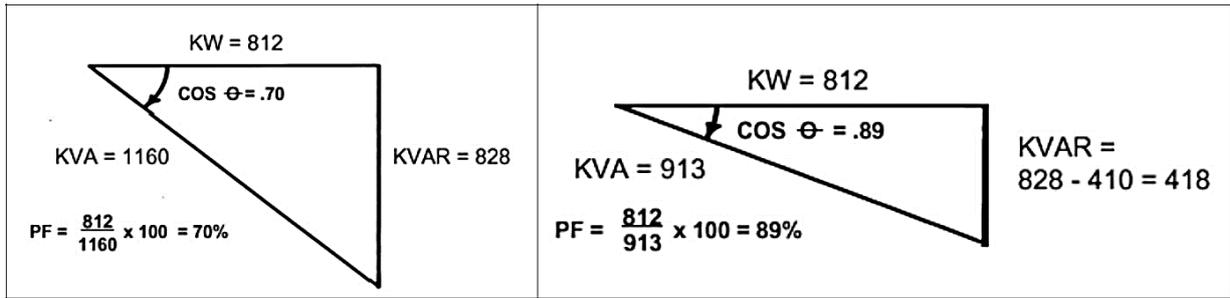
### **Improving Power Factor**

The solution to improve the power factor is to add power factor correction capacitors to the plant power distribution system. They act as reactive power generators, and provide the needed reactive power to accomplish KW of work. This reduces the amount of reactive power and thus total power generated by the Utilities (Distribution companies)

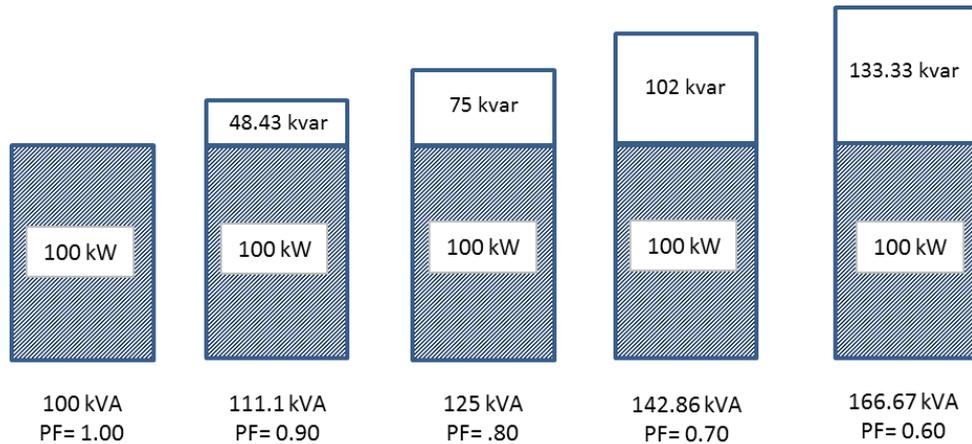
### **Example 1.2**

A chemical industry had installed a 1500 KVA transformer. The initial demand of the plant was 1160 KVA with power factor of 0.70. The % loading of transformer was about 78% ( $1160/1500=77.3\%$ ). To improve the power factor and thereby avoiding the penalty, the unit had added about 410 KVAR in motor load end. This improved the power factor to 89%, and reduced the required KVA to 913, which is the vector sum of KW and KVAR.

After improvement the plant had avoided penalty and the 1500 KVA transformer now loaded only to 60% of capacity. This will allow the addition of more loads in the future to be supplied by the transformer.



**Figure 1.7a: Power factor before and after Improvement**



**Figure 1.7b: Increase in the apparent and reactive powers as a function of the load power factor, holding the real power of the load constant**

### The advantages of improvement by capacitor addition

- Reactive component of the network is reduced and also the total current in the system from the source end.
- $I^2R$  power losses are reduced in the system because of reduction in current.
- Voltage level at the load end is increased.
- KVA loading on the source generators as also on the transformers and lines up to the capacitors reduces giving capacity relief. A high power factor can help in utilizing the full capacity of the electrical system.

### Cost benefits of PF improvement

While costs of PF improvement are in terms of investment needs for capacitor addition the benefits to be quantified for feasibility analysis are:

- Reduced KVA (Maximum demand) charges in utility bill
- Reduced distribution losses (KWH) within the plant network
- Better voltage at motor terminals and improved performance of motors
- A high power factor eliminates penalty charges imposed when operating with a low power factor
- Investment on system facilities such as transformers, cables, switchgears etc. for delivering load is reduced

## Selection, Location and Sizing of Capacitor

The figures given in table 1 are the multiplication factors which are to be multiplied with the input power (kW) to give the KVAR of capacitance required to improve present power factor to a new desired power factor.

**Table 1.1: Multiplication factors for selection of capacitors**

Original P.F.	Desired P.F.				
	1.0	0.95	0.90	0.85	0.80
0.55	1.518	1.189	1.034	0.899	0.763
0.60	1.333	1.004	0.849	0.714	0.583
0.65	1.169	0.840	0.685	0.549	0.419
0.70	1.020	0.691	0.536	0.400	0.270
0.75	0.882	0.553	0.398	0.262	0.132
0.80	0.750	0.421	0.266	0.130	
0.85	0.484	0.291	0.136		
0.90	0.328	0.155			
<b>0.95</b>	<b>0.620</b>				

Having known the existing power factor, the multiplication factor may be calculated for raising the power factor from the present value to the desired value.

### Example 1.3

If power factor of 30 kW load is to be improved from 0.80 to 0.95, then  
 Size of the capacitor = kW × multiplication factor = 30 × 0.421 = 12.63 (or) 13 KVAR

In case of induction motors of different ratings and speeds, in order to improve their power factor to 0.95 and above, the rating of the capacitor (in KVAR) for direct connection to induction motor can be referred to in the chapter on electric motors.

### Direct relation can also be used for capacitor sizing

$$\text{KVAR Rating} = \text{KW} [\tan \phi_1 - \tan \phi_2]$$

where, KVAR rating is the size of the capacitor needed, KW is the average power drawn,  $\tan \phi_1$  is the trigonometric ratio for the present power factor, and  $\tan \phi_2$  is the trigonometric ratio for the desired PF.

$$\phi_1 = \text{Existing } (\text{Cos}^{-1} \text{PF}_1) \text{ and } \phi_2 = \text{Improved } (\text{Cos}^{-1} \text{PF}_2)$$

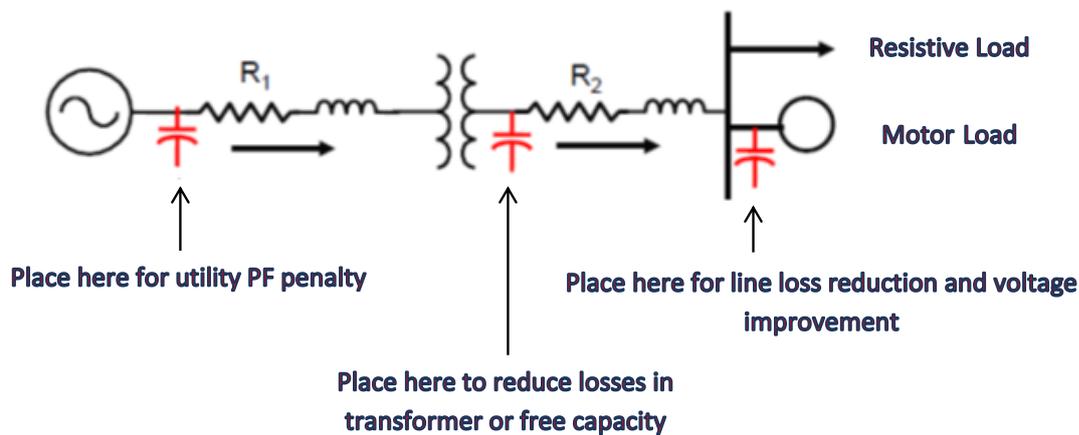
### Location of Capacitors

Location of capacitors is an important factor to be considered. For the benefit of electricity boards, connection of capacitors on H.T. side is good enough. Although the cost of H.T. capacitor per KVAR is low, the cost of the associated switchgear is quite high.

Alternatively, the capacitors can be connected on L.T. side of the main substation. The capacitors may be placed at load centers viz., directly with motors or group of motors at motor control centers. Correction of PF at the motors has number of advantages, as the induction motors are the main source of reactive currents in every industrial plant. The

advantages include the absence of additional switchgear; no separate control of capacitor is required in switching on and off operations and reduced effect of motor inrush currents.

From energy efficiency point of view, capacitor location at receiving substation only helps the utility in loss reduction. Locating capacitors at user end motors will help to reduce loss reduction within the plants distribution network as well and directly benefit the user by reduced demand cost. Reduction in the distribution loss in KWH when tail end power factor is raised from  $PF_1$  to a new power factor  $PF_2$ , will be proportional to  $[1 - (PF_1 / PF_2)^2]$



*Figure 1.8: Effect of Location*

### Capacitors for other loads

The other types of load requiring capacitor application include induction furnaces, induction heaters, arc welding transformers etc. The capacitors are normally supplied with control gear for the application of induction furnaces and induction heating furnaces. The P.F of arc furnaces experiences a wide variation over melting cycle as it changes from 0.7 at starting to 0.9 at the end of the cycle.

Power factor for arc welders and resistance's welders is corrected by connecting capacitors across the primary winding of the transformers, as the normal PF would be in the range of 0.35. The commended capacitor ratings for various sizes of welding transformers are given in table below.

<i>Welding Transformer Rating kVA</i>	<i>Capacitor Rating KVAR</i>
Single Phase	
9	4
12	6
18	8
24	12
30	18
Three Phase	
57	16.5

95	30
128	45
160	60

### Performance assessment of power factor capacitors

**Voltage effects:** Ideally capacitor voltage rating is to match the supply voltage. If the supply voltage is lower, the reactive KVAR produced will be the ratio  $V_1^2/V_2^2$  where  $V_1$  is the actual supply voltage,  $V_2$  is the rated voltage.

On the other hand, if the supply voltage exceeds rated voltage, the life of the capacitor is adversely affected.

*Table 1.2: Effect of addition of KVAR capacitor*

<b>KVAR Added</b>	<b>Phase Voltage</b>	<b>Phase Current</b>	<b>Total (kW)</b>	<b>Total (kVA)</b>	<b>Power Factor</b>
0	269	121	69	96	0.72
15	268	109	69	84	0.8
30	270	100	70	80	0.87
45	271	92	70	74	0.94
60	272	88	70	71	0.98
75	273	87	70	70	0.99
90	274	89	70	73	0.95 (1.05)
105	276	95	70	79	0.89 (1.11)

**Material of capacitors:** Power factor capacitors are available in various types by dielectric material used as; paper/poly propylene etc. The watt loss per KVAR as well as life vary with respect to the choice of the dielectric material and hence is a factor to be considered while selection.

**Connections:** Shunt capacity or connections are adopted for almost all industry/end user applications, while series capacitors are adopted for voltage boosting in distribution networks.

**Operational performance of capacitors:** This can be made by monitoring capacitor charging current vis- a- vis the rated charging current. Capacity of fused elements can be replenished as per requirements. Portable analyzers can be used for measuring KVAR delivered as well as charging current.

Some checks that need to be adopted in use of capacitors are:

- i. Name plates can be misleading with respect to ratings. It is good to check by charging currents.
- ii. Capacitors boxes may contain only insulated compound and insulated terminals with no capacitor elements inside.

- iii. Capacitors for single phase motor starting and those used for lighting circuits for voltage boost, are not power factor capacitor units and these cannot withstand power system conditions.

## **Transformers**

A transformer can accept energy at one voltage and deliver it at another voltage. This permits electrical energy to be generated at relatively low voltages and transmitted at high voltages and low currents, thus reducing line losses.

Transformers consist of two or more coils that are electrically insulated, but magnetically linked. The primary coil is connected to the incoming power and the secondary coil connects to the load. The turn's ratio is the ratio between the number of turns on the primary to the turns on the secondary.

The secondary voltage is equal to the primary voltage times the turn's ratio. Ampere- turns are calculated by multiplying the current in the coil times the number of turns. Primary ampere-turns are equal to secondary ampere-turns. Voltage regulation of a transformer is the percent increase in voltage from full load to no load.

## **Types of Transformers**

Transformers are classified as two categories as given below:

Power transformers: It is used in transmission network of higher voltages, deployed for step-up and step down transformer application. (400 kV, 200 kV, 110kV, 66kV, 33kV)

Distribution transformers: It is used for lower voltage distribution networks as a means to end user connectivity. (11.kV, 6.6kV, 3.3kV, 440V, 230V)

## **Rating of transformer**

Rating of the transformer is calculated based on the connected load and applying the diversity factor on the connected load, applicable to the particular industry and arrive at the KVA rating of the Transformer. Diversity factor is defined as the ratio of overall maximum demand of the plant to the sum of individual maximum demand of various equipment. Diversity factor varies from industry to industry and depends on various factors such as individual loads, load factor and future expansion needs of the plant. Diversity factor will always be less than one.

## **Location of transformer**

Location of the transformer is very important as far as distribution loss is concerned. Transformer receives HT voltage from the grid and steps it down to the required voltage. Transformers should be placed close to the load center, considering other features like optimization needs for centralized control, operational flexibility etc. This will bring down the distribution loss in cables.

## **Transformer Losses and Efficiency**

The efficiency varies anywhere between 96 to 99 percent. The efficiency of the transformers not only depends on the design but also on the effective operating load.

Transformer losses consist of two parts:

No-load loss (also called core loss) is the power consumed to sustain the magnetic field in the transformer's steel core. Core losses are caused by two factors: hysteresis and eddy current losses. Hysteresis loss is that energy lost by reversing the magnetic field in the core as the magnetizing AC rises and falls and reverses direction. Eddy current loss is a result of induced currents circulating in the core. Core loss occurs whenever the transformer is energized; core loss does not vary with load.

Load loss (also called copper loss) is associated with full-load current flow in the transformer windings. Copper loss is power lost in the primary and secondary windings of a transformer due to the ohmic resistance of the windings. Copper loss varies with the square of the load current. ( $P=I^2R$ ).

For a given transformer, the manufacturer can supply values for no-load loss,  $P_{NO-LOAD}$ , and load loss,  $P_{LOAD}$ . The total transformer loss,  $P_{TOTAL}$ , at any load level can then be calculated from:

$$P_{TOTAL} = P_{NO-LOAD} + (\% \text{ Load}/100)^2 \times P_{LOAD}$$

Where transformer loading is known, the actual transformer's loss at given load can be computed as:

$$\text{No load loss} + \left( \frac{\text{KVA Load}}{\text{Rated KVA}} \right)^2 \text{ full load loss}$$

**Table 1.3: Typical Transformer Loss for Distribution Transformers (DT's) above 100kVA**

KVA Rating	Voltage Rating (V)	No load loss(W)	Load Loss(W)	Impedance%
160		425	3000	5
200		570	3300	5
250		620	3700	5
315		800	4600	5
500	11000/433	1100	6500	5
630		1200	7500	5
1000		1800	11000	5
1600		2400	15500	5
2000		3000	20000	6
630		1450	7500	5
1000	33000/433	2200	11500	5
1600		3000	16000	6.25
2000		3500	21000	6.25

### Voltage fluctuation control

A control of voltage in a transformer is important due to frequent changes in supply voltage level. Whenever the supply voltage is less than the optimal value, there is a chance of nuisance tripping of voltage sensitive devices. The voltage regulation in transformers is

done by altering the voltage transformation ratio with the help of tapping. There are two methods of tap changing facility available.

### **Off-circuit tap changer**

It is a device fitted in the transformer, which is used to vary the voltage transformation ratio. Here the voltage levels can be varied only after isolating the primary voltage of the transformer.

### **On load tap changer (OLTC)**

The voltage levels can be varied without isolating the connected load to the transformer. To minimize the magnetization losses and to reduce the nuisance tripping of the plant, the main transformer (the transformer that receives supply from the grid) should be provided with On Load Tap Changing facility at design stage. The downstream distribution transformers can be provided with off-circuit tap changer.

The On-load gear can be put in auto mode or manually depending on the requirement. OLTC can be arranged for transformers of size 250kVA onwards. However, the necessity of OLTC below 1000 kVA can be considered after calculating the cost economics.

### **Parallel operation of transformers**

The design of Power Control Centre (PCC) and Motor Control Centre (MCC) of any new plant should have the provision of operating two or more transformers in parallel. Additional switch gears and bus couplers should be provided at design stage.

Whenever two transformers are operating in parallel, both should be technically identical in all aspects and more importantly with same impedance level. This will minimize the circulating current between transformers.

Where the load is fluctuating in nature, it is preferable to have more than one transformer running in parallel, so that the load can be optimized by sharing the load between transformers. The transformers can be operated close to the maximum efficiency range by this operation.

### **Energy Efficient Transformers**

Most energy loss in dry-type transformers occurs through heat or vibration from the core. The new high-efficiency transformers minimize these losses. The conventional transformer is made up of a silicon alloyed iron (grain oriented) core. The iron loss of any transformer depends on the type of core used in the transformer. The latest technology is to use for the amorphous core. Amorphous material has great advantage in reducing No load loss.

Amorphous core material (AM) offers both reduced hysteresis loss and eddy current loss because this material has a random grain and magnetic domain structure which results in high permeability giving a narrow hysteresis curve compared to conventional core material. Eddy current losses are reduced by the high resistivity of the amorphous material, and the reduced thickness of the film (thickness is approximately 0.03 mm, which is about

1/10 comparing with silicon steel). Amorphous core transformers offer a 70 to 80% reduction in no-load losses compared to transformers using conventional core material.

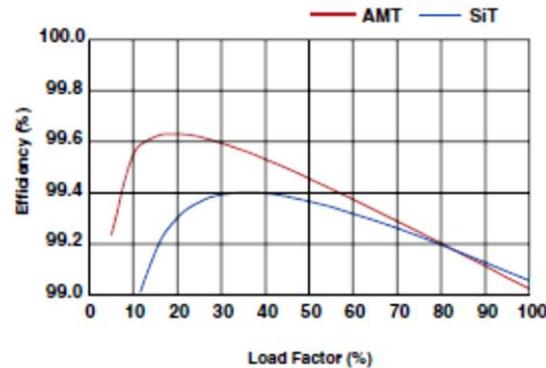


Figure 1.9: Comparison of Conventional and Amorphous Core Transformers

### Example 1.4

#### Transformer loss calculation

An engineering industry has installed three numbers of 1000KVA transformers for an electrical load of 1500KVA. The No-load loss and the full load loss of the transformers were collected from the transformer certificates as 2.8KW and 11.88KW respectively. Estimate the total loss when 3 transformers in parallel operation and also 2 transformers parallel operation. The transformer losses can also be obtained from manufacturers test certificate which are available in the plant.

#### a) Total loss when Two transformers in parallel operation:

$$\begin{aligned} \text{No load loss} &= 2 \times 2.8 = 5.6 \\ \text{Load Loss} &= 2 \times (750/1000)^2 \times 11.88 \\ \text{Total Loss} &= 5.6 + 13.36 = 18.96 \end{aligned}$$

#### b) Total loss when Three transformers in parallel operation: No load loss = 3 x 2.8 = 8.4KW

$$\begin{aligned} \text{Load loss} &= 3 \times (500/1000)^2 \times 11.88 = 8.91 \text{ kW} \\ \text{Total loss} &= 17.31 \text{ kW} \\ \text{Savings by operating 3 transformers in parallel} &= 18.96 - 17.31 = 1.65 \text{ kWh} \\ &= 1.65 \text{ kWh} \times 24 \text{ Hrs} \times 365 \text{ days} = 14454 \text{ kWh /year} \end{aligned}$$

### 1.5 System distribution losses

In an electrical system often the constant no load losses and the variable load losses are to be assessed alongside, over long reference duration, towards energy loss estimation. Identifying and calculating the sum of the individual contributing loss components is a challenging one, requiring extensive experience and knowledge of all the factors impacting the operating efficiencies of each of these components.

For example the cable losses in any industrial plant will be of the order of 2 to 4 percent. Note that all of these are current dependent, and can be readily mitigated by any technique that reduces facility current load.

In system distribution loss optimization, the various options available include:

- Relocating transformers and sub-stations near to load centers
- Re-routing and re-conducting such feeders and lines where the losses/voltage drops are higher.
- Power factor improvement by incorporating capacitors at load end.
- Optimum loading of transformers in the system.
- Opting for lower resistance All Aluminum Alloy Conductors (AAAC) in place of conventional Aluminum Cored Steel Reinforced (ACSR) lines
- Minimizing losses due to weak links in distribution network such as jumpers, loose contacts, old brittle conductors.
- Distribution loss assessment and optimization studies today are feasible on account of accurate metering developments on the one hand and availability of powerful computer based load flow analysis packages on the other.
- Using full infrared thermography system, each electrical panel can be scanned to identify points of high system heat. Called “hotspots”, these high heat points result from connections become looser corroded overtime. The resulting increase in resistance at that spot in the system can add wattage losses to the electrical energy consumption. These hot spots also create safety risks and risks to abrupt system failure. Fixing them is often as simple as de-energizing that point in the system, and then using a wrench to tighten a bolt.

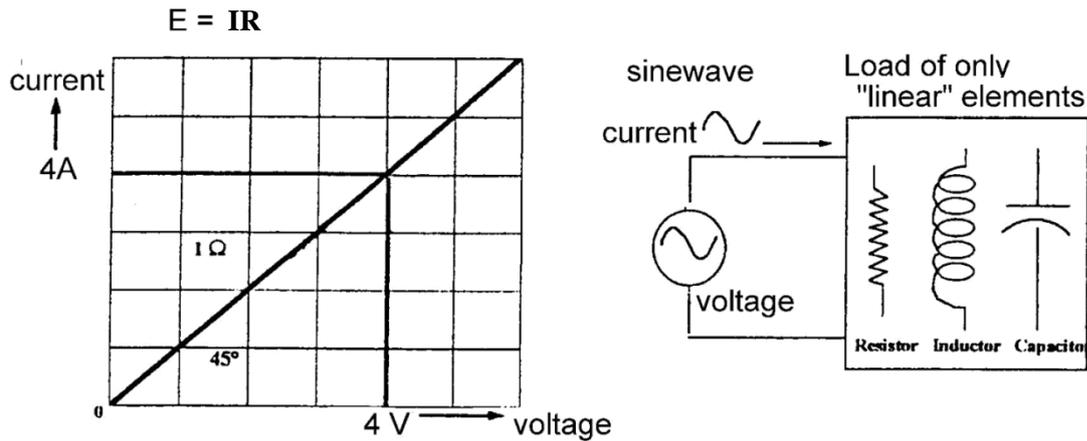
As far as electricity distribution utilities are concerned, involving large network and complex connectivity features, there exist well proven computer based application packages which can be used for network load flow analysis. The analysis outputs can help a utility engineer to assess the extent of transmission and distribution losses, to identify sections for improvement where voltage drops are high, to identify avenues for loss reduction such as ideal location of sub-stations, feeder augmentation, etc.

## **1.6 Harmonics and its Effects**

In any alternating current network, flow of current depends upon the voltage applied and the impedance (resistance to AC) provided by elements like resistances, reactance of inductive and capacitive nature. As the value of impedance in above devices is constant, they are called linear whereby the voltage and current relation is of linear nature.

### **Example for Linear loads**

Linear loads occur when the impedance is constant; then the current is proportional to the voltage (A straight line graph, as shown in Figure–1.10). Simple loads, composed of one of the elements shown in Figure–1.10, do not produce harmonics.

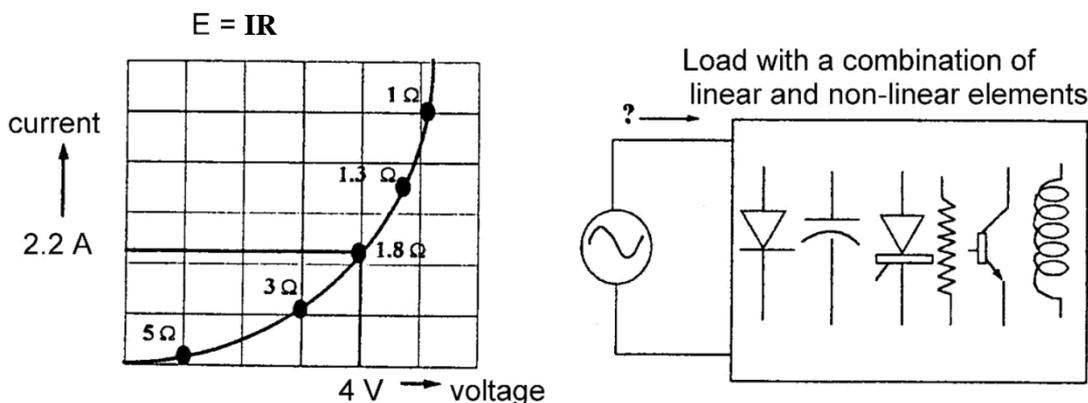


*Figure 1.10: Linear loads*

However in real life situation, various devices like diodes, silicon controlled rectifiers, thyristors, voltage & current controllers, induction & arc furnaces are also deployed for various requirements and due to their varying impedance characteristic, these Non-Linear devices cause distortion in voltage and current waveforms which is of increasing concern in recent times.

### Example for Non-Linear loads

Non-linear loads occur when the impedance is not constant; then the current is not proportional to the voltage (as shown in Figure 1.11). Combinations of the components shown in Figure 1.11 normally create non-linear loads and harmonics.



*Figure 1.11: Non-Linear loads*

Harmonics occurs as spikes at intervals which are multiples of the mains (supply) frequency and these distort the pure sine wave form of the supply voltage & current. Thus harmonics are multiples of the fundamental frequency of an electrical power system. If for example, the fundamental frequency is 50Hz, then the 5<sup>th</sup> harmonic is five times that frequency, or 250 Hz. Likewise, the 7<sup>th</sup> harmonic is seven times the fundamental or 350Hz, and so on for higher order harmonics.

The magnitude and order of harmonics is governed by the nature of the device being used and the impact is expressed as Total Harmonic Distortion (THD). Harmonics can be expressed in terms of current or voltage.

In terms of voltage, it is expressed as a percentage of fundamental voltage by the expression:

$$\%THD = \sqrt{\sum_{n=2}^{n=n} \frac{V_n^2}{V_1^2}} * 100$$

where  $V_1$  is the fundamental frequency voltage and  $V_n$  is  $n^{\text{th}}$  harmonic voltage component.

**In terms of current, it is expressed as below:**

A 5th harmonic current is simply a current flow at 250Hz on a 50Hz system. The 5th harmonic current flowing through the system impedance creates a 5<sup>th</sup> harmonic voltage. The following is the formula for calculating the THD for current:

$$= \frac{\sqrt{(I_5^2 + I_7^2)}}{I_1}$$

$I_1$  = current at 50 Hz = 250 Amps,  $I_5$  = current at 250 Hz = 50 Amps  $I_7$  = current at 350 Hz = 35 Amps

If  $I_1$  = 250 Amps,  $I_5$  = 50 Amps and  $I_7$  = 35Amps

Then...

$$I_{THD} = \frac{\sqrt{(50^2 + 35^2)}}{250} \times 100 = 24\%$$

When harmonic currents flow in a power system, they are known as poor “power quality”. Other causes of poor power quality include transients such as voltage pikes, surges, sags, and ringing. Because they repeat every cycle, harmonics are regarded as a steady-state cause of poor power quality.

The harmonic assessment can be carried out at site by using a load analyzer. The wave form is sampled and analyzed through various harmonic frequencies, i.e. multiples of the mains frequency for assessing THD. Load analysers are available in market, which can measure THD up to 63<sup>rd</sup> harmonic.

### **Causes and Effects of Harmonics in electrical systems**

Devices that draw non-sinusoidal currents when a sinusoidal voltage is applied create harmonics. Frequently these are devices that convert AC to DC. Listed below are some of these devices.

#### **Electronic Switching Power Converters**

- Computers, Uninterruptible power supplies (UPS), Solid-state rectifiers
- Electronic process control equipment, PLC’s, etc.
- Electronic lighting ballasts, including light dimmer
- Reduced voltage motor controllers

## Arcing Devices

- Discharge lighting, e.g. Fluorescent, Sodium and Mercury vapor
- Arc furnaces, Welding equipment, Electrical traction system

## Ferromagnetic Devices

- Transformers operating near saturation level
- Magnetic ballasts (Saturated Iron core)
- Induction heating equipment, Chokes, Motors

## Appliances

- TV sets, air conditioners, washing machines, microwave ovens
- Fax machines, photocopiers, printers

These devices use power electronics like diodes and thyristors which are a growing percentage of the load in industrial power systems. Normally each load would manifest a specific harmonic spectrum. Many problems can arise from harmonic currents in a power system. Some problems are easy to detect. Higher RMS current and voltage in the system are caused by harmonic currents, which can result in any of the problems listed below.

## Effects of Harmonics on Network

The effects of harmonics on distribution network include:

- Metering errors in electromagnetic type meters.
- Overloading and overheating of motors due to increased iron losses & overheating of conductors.
- Overloading of neutral conductor especially in low voltage distribution network and High neutral currents
- Malfunctioning of control equipment and protection relays due to false signals.
- Blown Fuses (no apparent fault)
- Misfiring of AC and DC Drives
- Tripped Circuit Breakers Voltage distortion
- High neutral to ground voltages Increased system losses (heat)
- Rotating and electronic equipment failures
- Capacitor bank over-load and failures
- Reduced power factor

Source	Typical Harmonics
6 Pulse Drive/Rectifier	5,7,11,13,17,19...
12 Pulse Drive/Rectifier	11,13,23,25...
18 Pulse Drive	17,19,35,37...
Switch-Mode Power Supply	3,5,7,9,11,13...
Fluorescent Lights	3,5,7,9,11,13...
Arcing Devices	2,3,4,5,7...
Transformer Energization	2,3,4

Generally, the magnitude decreases as harmonic order increases.

$$h = n_p \pm 1$$

$h$  = order of harmonics,  $n$  = an integer 1, 2, 3, ...,  $p$  = number of pulses per cycle

For a three phase bridge rectifier, since the number of pulses  $p = 6$  per line frequency cycle, the characteristic or dominant harmonics are:  $h = n \cdot 6 \pm 1 = 5, 7, 11, 13, 17, 19, 23, 25, 35, 37...$

## Harmonic Filters

Harmonic filters consist of a capacitor bank and reactor in series are designed and adopted for suppressing harmonics, by providing low impedance path for harmonic component. The Harmonic filters connected suitably near the equipment generating harmonics help to reduce THD to acceptable limits. In present context where no Electro Magnetic Compatibility regulations exist, an application of Harmonic filters is very relevant for industries having diesel power generation sets and co-generation units. Energy managers / auditors can address the issue of harmonics from the point of view of energy efficiency and power quality assurance.

The Harmonic Mitigation solutions currently in use in the industry broadly fall into the following categories:

1. Passive Harmonic Filter (PHF)
2. Advance Active Filters (AAF)
3. Active Front End based VFDs (AFE)

### Passive Harmonic Filter (PHF)

It is the most common method for the cancellation of harmonic current in the distributed system. These filters are basically designed on principle either single tuned/double tuned or band pass filter technology. Passive filters offer very low impedance in the network at the tuned frequency to divert all the harmonic current at the tuned frequency.

### Advance Active Filters (AAF)

It is connected parallel with the distribution system. Distribution system consists of a wide percentage of harmonics produced by non- linear loads. Active filters compensate current harmonics by injecting equal magnitude but opposite phase harmonic compensating current.

### Active Front End based VFDs (AFE)

It is used in VFDs because it:

- has the major advantage of mitigation of harmonics without using external filter,
- maintains unity power factor at the point of common coupling,
- supports Bidirectional power flow which makes recovery of energy to the mains by saving it,
- provides clean power to the grid which in turn does not affect the other loads connected to it,
- maintains the DC voltage irrespective of the supply variations.

### Harmonics Limits:

The permissible harmonic limit for different current ( $I_{sc} / I_L$ ) as per IEEE standard is given in Table 1.4 and for different bus voltage are given in Table 1.5 Current Distortion Limits for General Distribution System's end-User limits (120 Volts To 69,000 Volts)

<b>Table 1.4 Maximum Harmonic Current Distortion in % of <math>I_L</math></b>						
<b>Individual Harmonic Order (Odd Harmonics)</b>						
$I_{sc}/I_L$	$h < 11$	$11 \leq h < 17$	$17 \leq h < 23$	$23 \leq h < 35$	$35 \leq h$	TDD
$< 20^*$	4.0	2.0	1.5	0.6	0.3	5.0
$20 < 50$	7.0	3.5	2.5	1.0	0.5	8.0
$50 < 100$	10.0	4.5	4.0	1.5	0.7	12.0
$100 < 1000$	12.0	5.5	5.0	2.0	1.0	15.0
$> 1000$	15.0	7.0	6.0	2.5	1.4	20.0

Even harmonics are limited to 25% of the odd current harmonic limits above.

Current distortions that result in a direct current offset, e.g. half wave converters are not allowed.

\*All power generation equipment is limited to these values of current distortion, regardless of actual  $I_{sc}/I_L$ .

Where,  
 $I_{sc}$  = Maximum short circuit current at PCC (Point of Common Coupling).  
 And  $I_L$  = Maximum Demand Load Current (fundamental frequency component) at PCC.  
 TDD = Total Demand Distortion, harmonic current distortion in % of maximum demand load current (15 or 30 min demand).

<b>Table 1.5 Total Harmonic Distribution for Different Voltage Levels in %</b>		
<b>Bus Voltage at PCC</b>	<b>Individual Voltage Distortion (%)</b>	<b>Total Voltage Distortion THD (%)</b>
69 kV and below	3.0	5.0
69.001 kV Thru 161 kV	1.5	2.5
161 kV and above	1.0	1.5

*Note:*

High voltage systems can have up to 2.0% THD where the cause is an HVDC terminal that will attenuate by the time it is tapped for a user.

**Two very important points must be made in reference to the above.**

1. The customer is responsible for maintaining current distortion to within acceptable levels, while the utility is responsible for limiting voltage distortion.
2. The limits are only applicable at the point of common coupling (PCC) between the utility and the customer. The PCC, while not explicitly defined, is usually regarded as the point at which the utility equipment ownership meets the customer's or the metering point.

Therefore, the above limits cannot be meaningfully applied to distribution panels or individual equipment within a plant. The entire plant must be considered complying with these limits.

### 1.7 A Glossary of Basic Electrical Terms

V	Symbolizes volts or the electromotive force or electric pressure.
I	Symbolizes the electric current flowing in the circuit in amperes.
P	Power in watts or kilowatts, indicates the real working component of the energy put in.
KVA	Indicates kilovolt amperes or the apparent power that determines the heating effect on the AC equipment and systems. All elements of the power systems must be sized to accommodate this burden.
KVAR	Kilovolt amperes reactive, or the ‘Phantom Component’ that vectorally combines with real power, to determine KVA on electric systems.
PF	Power factor or the ratio of real power (KW) to apparent power (KVA) percentage or as a decimal value.
$\Phi$	Phase angle on the alternating current system, or the measure of the vector displacement between true power and apparent power, power factor expressed as the decimal value is the cosine of $\phi$
kWh	Kilowatt-hours are the unit of electrical energy. KWh is obtained by integrating power, expressed in KW with time. For example, a power of 2KW appliance for 15 minutes (1/4 hour) indicates an energy consumption of $2 \times 1/4 = 0.5$ kWh.
Power	$\sqrt{3} \times V \times I \times PF$ for 3 phase systems, where V = Line voltage, I = line current • In delta connected electrical system, V line = V phase, I lines = $\sqrt{3}$ I phase • In star connected electrical system, V line = $\sqrt{3}$ V phase, I lines = I phase and $V \times I \times PF$ for single phase systems
KVA	$\sqrt{3} \times V \times I$ for 3 phase systems and $V \times I$ for single-phase systems.
Load factor	Load factor is the ratio of the average demand (KVA or KW) to the peak demand for a power system. High load factor leads to better utilization of capacity.
Demand Factor	Demand factor is the ratio of maximum demand to the connected load.
Peak Demand or Peak Load	The highest demand on an electric utility system is called Peak Load or Peak Demand. Demand varies with time every day and also from season to season. As electricity cannot be stored easily, utilities have to provide the generating capacity to meet peak demand even it lasts for a short duration.
Connected Load	Connected load is the summation of nameplate ratings (kW or kVA) of the electrical equipment installed in a consumer's premises

Load Management	Load management is a set of techniques for control of power supply and demand to increase the system load factor. Electric utilities as well as consumers can reduce the peak (maximum demand) by load shifting or load shedding (power cuts)
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### Analysis of Electrical Power Systems

System Problem	Common Causes	Possible Effects	
Voltage imbalances among the three phases	Improper transformer tap settings, single-phase loads not balanced among phases, poor connections, bad conductors, transformer grounds or faults.	Motor vibration, premature motor failure A 5% imbalance causes a 40% increase in motor losses.	Balance loads among phases.
Voltage deviations from rated voltages (too low or high)	Improper transformer settings, Incorrect selection of motors.	Over-voltages in motors reduce efficiency, power factor and equipment life	Correct transformer settings, motor ratings and motor input voltages
Poor connections in distribution or at connected loads.	Loose bus bar connections, loose cable connections, corroded connections, poor crimps, loose or worn contactors	Produces heat, causes failure at connection site, leads to voltage drops and voltage imbalances	Use Infra-Red camera to locate hot-spots and correct.
Undersized conductors.	Facilities expanding beyond original designs, poor power factors	Voltage drop and energy waste.	Reduce the load by conservation load
Insulation leakage	Degradation over time due to extreme temperatures, abrasion, moisture, chemicals	May leak to ground or to another phase. Variable energy waste.	Replace conductors, insulators
Low Power Factor	Inductive loads such as motors, transformers, and lighting ballasts Non-linear loads, such as most electronic loads.	Reduces current-carrying capacity of wiring, voltage regulation effectiveness, and equipment life	Add capacitors to counteract reactive loads.
Harmonics (non-sinusoidal voltage and/or current wave forms)	Office-electronics, UPSs, variable frequency drives, high intensity discharge lighting and electronic and core-coil ballasts.	Over-heating of neutral conductors, motors, transformers, switch gear. Voltage drop, low power factors, reduced capacity.	Take care with equipment selection and isolate sensitive electronics

An analysis of an electrical power system may uncover energy waste, fire hazards, and equipment failure. Facility / Energy managers increasingly find that reliability-centered maintenance can save money, energy, and downtime.

## Chapter 2: Electrical Motors

### 2.1 Introduction

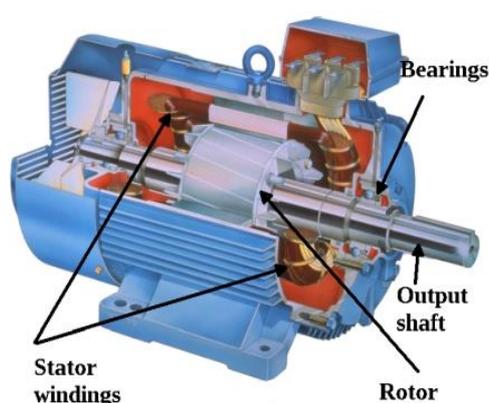
Electric motors convert electrical power into mechanical power by the interaction between the magnetic fields set up in the stator and rotor windings within a motor. In industrial applications, electric motor driven systems are used for various applications such as pumping, compressed air, fans, conveyors etc.

All industrial electric motors can be broadly classified as Induction Motors, Direct Current Motors or Synchronous Motors. All motor types have the same four operating components: Stator (stationary windings), Rotor (rotating windings), Bearings, and Frame (enclosure). All motors convert electrical energy in to mechanical energy by the interaction between the magnetic fields set up in the stator and rotor windings.

### 2.2 Motor Types

#### 2.2.1 Induction Motors

Induction motors are the most commonly used in industrial applications. The induction motor is the most popular type of ac motor because of its simplicity and ease of operation. An induction motor does not have a separate field circuit; instead, it depends on transformer action to induce voltages and currents in its field circuit. In fact, an induction motor is basically a rotating transformer. Its equivalent circuit is similar to that of a transformer, except for the effects of varying speed. There are two types of induction motor rotors, cage rotors and wound rotors. Cage rotors consist of a series of parallel bars all around the rotor, shorted together at each end. Wound rotors are complete three-phase rotor windings, with the phases brought out of the rotor through slip rings and brushes. Wound rotors are more expensive and require more maintenance than cage rotors, so they are very rarely used (except sometimes for induction generators).



*Figure 2.1: Induction Motor*

In induction motors, the induced magnetic field of the stator winding induces a current in the rotor. In induction machines, rotor currents are induced in the rotor windings by a combination of the time-variation of the stator currents and the motion of the rotor relative to the stator. If a 3-phase supply is fed to the stator windings of a 3-phase motor, a magnetic flux of constant magnitude, rotating at synchronous speed is set up. At this point, the rotor

is stationary. The rotating magnetic flux passes through the air gap between the stator & rotor and sweeps past the stationary rotor conductors. This rotating flux, as it sweeps, cuts the rotor conductors, thus causing an e.m.f to be induced in the rotor conductors. As per the Faraday's law of electromagnetic induction, it is this relative motion between the rotating magnetic flux and the stationary rotor conductors, which induces an e.m.f on the rotor conductors. Since the rotor conductors are shorted and form a closed circuit, the induced e.m.f produces a rotor current whose direction is given by Lenz's Law, is such as to oppose the cause producing it. In this case, the cause which produces the rotor current is the relative motion between the rotating magnetic flux and the stationary rotor conductors. Thus to reduce the relative speed, the rotor starts to rotate in the same direction as that of the rotating flux on the stator windings, trying to catch it up. The frequency of the induced e.m.f is same as the supply frequency. An induction motor normally operates at a speed near synchronous speed ( $n_{sync}$ ), but it can never operate at exactly  $n_{sync}$ .

### **Slip-ring motor**

The slip-ring motor or wound-rotor motor is a variation of the squirrel cage induction motor. While the stator is the same as that of the squirrel cage motor, the rotor of a slip-ring motor is wound with wire coils. A slip ring induction motor is an asynchronous motor, as the rotor never runs in synchronous speed with the stator poles. The ends of the windings are connected to slip rings so that resistors or other circuitry can be inserted in series with the rotor coils through carbon brushes that slide on the slip-rings allowing an electrical connection with the rotating coils. This basically is the difference in construction between a squirrel cage and slip-ring motors. These are helpful in adding external resistors and contactors. The slip necessary to generate the maximum torque (pull-out torque) is directly proportional to the rotor resistance. In the slip-ring motor, the effective rotor resistance is increased by adding external resistance through the slip rings. Thus, it is possible to get higher slip and hence, the pull-out torque at a lower speed. A particularly high resistance can result in the pull-out torque occurring at almost zero speed, providing a very high pull-out torque at a low starting current. As the motor accelerates, the value of the resistance can be reduced, altering the motor characteristic to suit the load requirement. Once the motor reaches the base speed, external resistors are removed from the rotor. This means that now the motor is working as the standard induction motor.

This motor type is ideal for very high inertia loads, where it is required to generate the pull-out torque at almost zero speed and accelerate to full speed in the minimum time with minimum current draw.

Modifying the speed torque curve by altering the rotor resistors, the speed at which the motor will drive a particular load can be altered. At full load the speed can be reduced effectively to about 50% of the motor synchronous speed, particularly when driving variable torque/variable speed loads, such as printing presses, compressors, conveyer belts, hoists and elevators. Reducing the speed below 50%, results in very low efficiency due to higher power dissipation in the rotor resistances. This type of motor is used in applications for driving variable torque/ variable speed loads.

## **2.2.2 Direct-Current Motors**

Direct-Current motors, as the name implies, use direct, i.e. unidirectional, current. Used in special applications, they only represent small percentage of motors used in industry, *e.g.* where high torque starting or where smooth acceleration over a broad speed range is required. Before the widespread use of power electronic rectifier-inverters, dc motors were unexcelled in speed control applications.

### **The working of DC motor is based on the principle**

That when a current-carrying conductor is placed in a magnetic field, it experiences a mechanical force. The direction of mechanical force is given by Fleming's Left-hand Rule. There is no basic difference in the construction of a DC generator and a DC motor. In fact, the same D.C machine can be used interchangeably as a generator or as a motor.

There are five major types of dc motors in general use:

1. The separately excited dc motor
2. The shunt dc motor
3. The permanent-magnet dc motor
4. The series dc motor
5. The compounded dc motor

### **2.2.3 Synchronous Motors**

In synchronous machines, rotor-winding currents are supplied directly from the stationary frame through a rotating contact. AC power is fed to the stator of the synchronous motor. The rotor is fed by dc from a separate source. The rotor magnetic field locks onto the stator rotating magnetic field and rotates at the same speed. The speed of the rotor is a function of the supply frequency and the number of magnetic poles in the stator. While induction motors with a slip, i.e., rpm is less than the synchronous speed, the synchronous motor rotate with no slip, i.e., the rpm is same as the synchronous speed governed by supply frequency and number of poles. The basic principle of synchronous motor operation is that the rotor "chases" the rotating stator magnetic field around in a circle, never quite catching up with it. The slip energy is provided for by the D.C. excitation power.

## **2.3 Motor Characteristics**

### **2.3.1 Motor Speed**

The speed of a motor is the number of revolutions in a given time frame, typically revolutions per minute (RPM). The speed of an AC motor depends on the frequency of the input power and the number of poles for which the motor is wound. The synchronous speed in RPM is given by the following equation, where the frequency is in hertz or cycles per second:

$$\text{Synchronous Speed (RPM)} = \frac{120 * \text{Frequency}}{\text{No.of Poles}}$$

Motors have synchronous speeds like 3000 / 1500 / 1000 / 750 / 600 / 500 / 375 rpm corresponding to no. of poles (always even) being 2, 4, 6, 8, 10, 12, 16 and given the mains frequency of 50 cycles / sec.

The actual speed with which the motor operates, will be less than the synchronous speed. The difference between synchronous and full load speed is called slip and is measured in percent. It is calculated using this equation:

$$\text{Slip (\%)} = \frac{\text{Synchronous Speed} - \text{Full Load Speed}}{\text{Synchronous Speed}} * 100$$

As per relation stated above, the speed of an AC motor is determined by the number of motor poles and by the input frequency. It can also be seen that theoretically speed of an AC motor can be varied infinitely by changing the frequency. For practical limits to speed variation, manufacturer's guidelines should be referred to. With the addition of a Variable Frequency Drive (VFD), the speed of the motor can be decreased as well as increased.

### 2.3.2 Volts/Hz Relationship

It has been seen that by changing the frequency, one can change the speed of the motor. However, frequency is not the only parameter that must be changed. Notice in the motor model below that the impedance of a motor will change with frequency since the impedance of an inductor equals to  $2\pi fL$ . At low frequencies, this impedance approaches zero making the circuit appear to be a short circuit.

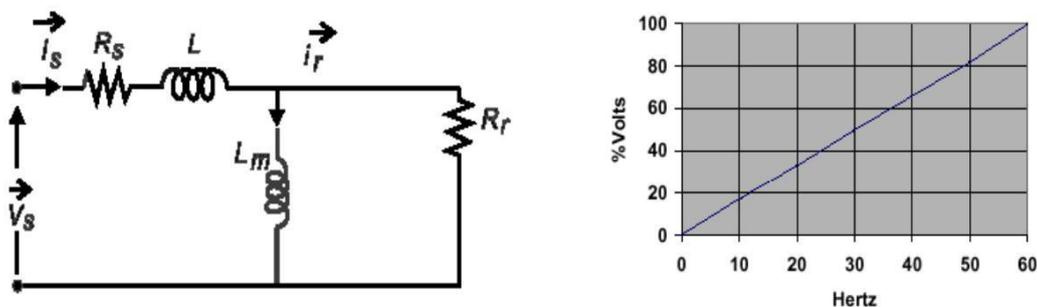
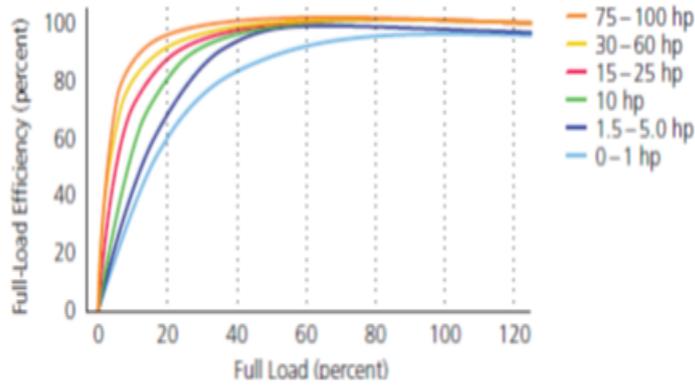


Figure 2.2: Volts/Hz Relationship

To maintain a constant flux in the motor, the voltage to the motor must also be changed. This ratio is constant over most of the entire speed range. By keeping the ratio constant, a fixed speed induction motor can be made to run at variable speed and provide constant torque as required by driven machine. At low speeds, due to the motor having inherent resistance in the windings, the ratio must be altered to provide enough magnetizing flux to spin the motor. The VFD allows this relationship to be altered by changing the voltage boost parameter.



**Figure 2.3:**  
*different load*

*Efficiency at*

### 2.3.3 Power Factor

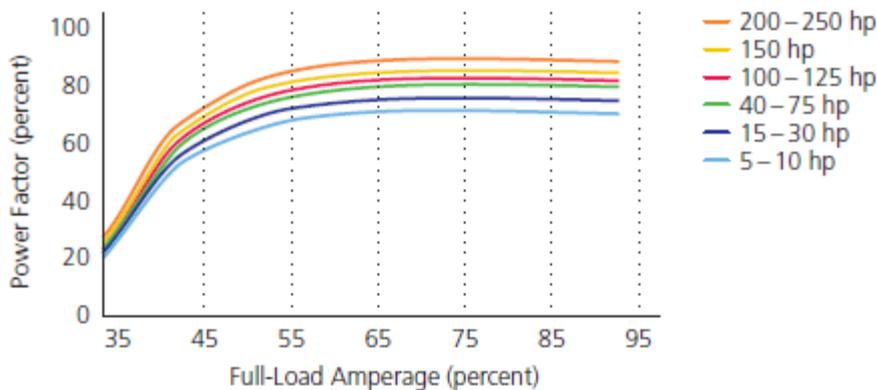
The power factor of the motor is given as:

$$\text{Power Factor} = \cos \phi = \text{kW/kVA}$$

As the load on the motor comes down, the magnitude of the **active current** reduces. However, there is no corresponding reduction in the **magnetizing current**, with the result that the motor power factor reduces, with a reduction in applied load. Induction motors, especially those operating below their rated capacity, are the main reason for low power factor in electric systems.

### 2.3.4 Motor Efficiency Parameters

Two important attributes relating to efficiency of electricity use by A.C. Induction motors are efficiency ( $\eta$ ), defined as the ratio of the mechanical energy delivered at the rotating shaft to the electrical energy input at its terminals, and power factor (PF), defined as the ratio of the real power (kW) to apparent power (kVA) drawn by the motor. Motors, like other inductive loads, are characterized by power factors less than one. As a result, the total current draw needed to deliver the same real power is higher than for a load characterized by a higher PF. An important effect of operating with a PF less than one is that resistance losses in wiring upstream of the motor will be higher, since these are proportional to the square of the current. Thus, both a high value for  $\eta$  and a PF close to unity are desired for efficient overall operation in a plant.



**Figure 2.4:** *Power factor at different loads*

Squirrel cage motors are normally more efficient than slip-ring motors, and higher-speed motors are normally more efficient than lower-speed motors. Efficiency is also a function

of motor temperature. Totally-enclosed, fan-cooled (TEFC) motors are more efficient than screen-protected drip-proof (SPDP) motors. Also, as with most equipment, motor efficiency increases with the rated capacity.

The efficiency of a motor is determined by intrinsic losses that can be reduced only by changes in motor design. Intrinsic losses are of two types—fixed, i.e., independent of motor load, and variable, i.e., dependent on load. Fixed losses consist of magnetic core losses and friction and windage losses. Magnetic core losses (sometimes called iron losses) consist of eddy current and hysteresis losses in the stator. They vary with the core material and geometry and with input voltage. Friction and windage losses are caused by friction in the bearings of the motor and aerodynamic losses associated with the ventilation fan and other rotating parts. Variable losses consist of resistance losses in the stator and in the rotor and miscellaneous stray losses. Resistance to current flow in the stator and rotor result in heat generation that is proportional to the resistance of the material and the square of the current ( $I^2R$ ). Stray losses arise from a variety of sources and are difficult to either measure directly or to calculate, but are generally proportional to the square of the rotor current.

Part-load performance characteristics of a motor also depend on its design. For operating loads in the range of 50 – 100 percent of rated load, the reductions in  $\eta$  decreases significantly, and PF continues to fall. Both  $\eta$  and PF fall to very low levels at low loads.

## **2.4 Motors Selection**

The primary technical consideration defining the motor choice for any particular application is the torque required by the load. Especially important is the relationship between the maximum torque generated by the motor (break-down torque) and the torque requirements for start-up (locked rotor torque) and during acceleration periods. Other load characteristics, e.g., constant versus variable torque requirements or constant versus variable speed also are considered in the selection process.

The duty/ load cycle determines the thermal loading on the motor. One consideration with totally enclosed fan cooled (TEFC) motors is that the cooling may be insufficient when the motor is operated at speeds below its rated value.

Several additional selection criteria are also typically considered. Ambient operating conditions affect motor choice: special motor designs are available for corrosive or dusty atmospheres, high temperatures, restricted physical space etc.

Anticipated switching frequency is an important consideration: an estimate of the frequency of switching (usually dictated by the process), whether automatic or manually controlled, can help in selecting the appropriate motor for the duty cycle.

The demand a motor will place on the balance of the plant electrical system is another consideration: if the load variations are large, for example as a result of frequent starts and stops of large components like compressors, the resulting large voltage drops could be detrimental to other equipment.

There are still other considerations that can influence the motor selection. Reliability is of prime importance. In many cases, however, designers and process engineers seeking

reliability will grossly over size equipment, leading to sub-optimal energy performance. Good knowledge of process parameters and a better understanding of the plant power system can aid in reducing over sizing with no loss of reliability. Inventory is another consideration.

Many large industries use standard equipment, which can be easily serviced or replaced, thereby reducing the stock of spare parts that must be maintained and minimizing shut-downtime. This practice affects the choice of motors that might provide better energy performance in specific applications. Shorter lead times for securing individual motors from suppliers would help reduce the need for this practice. Price is another issue. Many users are first-cost sensitive, leading to the purchase of less expensive motors that may be more costly on a lifecycle basis because of lower efficiency. For example, energy efficient motors or other specially designed motors typically save within a few years an amount of money equal to several times the incremental cost for an energy efficient motor, over a standard- efficiency motor.

#### 2.4.1 Field Tests for Determining Efficiency

##### **No Load Test:**

The motor is run at rated voltage and frequency without any shaft load. Input power, current frequency and voltage are noted. The no load P.F. is quite low and hence low PF wattmeter is required. From the input power, stator  $I^2R$  losses under no load are subtracted to give the sum of friction, wind age and core losses. To separate core and F&W losses, test is repeated at variable voltages. It is worthwhile plotting no-load input kW versus Voltage; the intercept is F& W kW loss component.

##### **Stator and Rotor $I^2R$ Losses:**

The stator winding resistance is directly measured by a bridge or volt amp method. The resistance must be corrected to the operating temperature. For modern motors, the operating temperature is likely to be in the range of 100°C to 120°C and necessary correction should be made. Correction to 75°C may be inaccurate. The correction factor is given as follows:

$$\frac{R_2}{R_1} = \frac{235 + t_2}{235 + t_1}$$

The rotor resistance can be determined from locked rotor test at reduced frequency, but rotor  $I^2R$  losses are measured from measurement of rotor slip.

Rotor  $I^2R$  losses = Slip × (Stator Input – Stator  $I^2R$  Losses – Core Loss)

Accurate measurement of slip is possible by stroboscope or non- contact type taco meter. Slip also must be corrected to operating temperature.

##### **Stray Load Losses:**

These losses are difficult to measure with any accuracy. IEEE Standard 112 gives a complicated method, which is rarely used on shop floor. IS and IEC standards take a fixed value as 0.5 % of output. It must be remarked that actual value of stray losses is likely to be more. IEEE – 112 specifies values from 0.9 % to 1.8%.

Motor Rating	Stray Losses
1 – 125HP	1.8 %
125 – 500HP	1.5 %
501 – 2499HP	1.2 %
2500 and above	0.9 %

### Points for Users:

It must be clear that accurate determination of efficiency is very difficult. The same motor tested by different methods and by same methods by different manufacturers can give a difference of 2%. In view of this, for selecting high efficiency motors, the following can be done:

- When purchasing large number of small motors or a large motor, ask for a detailed test certificate. If possible, try to remain present during the tests.
- See that efficiency values are specified without any tolerance
- Check the actual input current and kW, if replacement is done
- For new motors, keep a record of no load input current and power
- Use values of efficiency for comparison and for confirming; rely on measured inputs for all calculations.

### Estimation of efficiency in the field can be done as follows:

- Measure stator resistance and correct to operating temperature. From rated current value,  $I^2R$  losses are calculated.
- From rated speed and output, rotor  $I^2R$  losses are calculated
- From no load test, core and F & W losses are determined.

### The method is illustrated by the following example:

#### Example 2.1

#### Motor Specifications

Rated power	=	34 kW/45 HP
Voltage	=	415 Volt
Current	=	57 Amps
Speed	=	1475 rpm
Insulation class	=	F
Frame	=	LD 200L

Connection	=	Delta
No load test Data		
Voltage, V	=	415 Volts
Current, $I_{NL}$	=	16.1 Amps
Frequency, F	=	50 Hz
Stator phase resistance at 30°C	=	0.264 Ohms
No load power, $P_{nl}$	=	1063.74 Watts

- Calculate iron plus friction and windage losses
- Calculate stator resistance at 120°C

$$R_2 = R_1 * \frac{235 + t_2}{235 + t_1}$$

- Calculate stator copper losses at operating temperature of resistance at 120°C
- Calculate full load slip (s) and rotor input assuming rotor losses are slip times rotor input.
- Determine the motor input assuming that stray losses are 0.5% of the motor rated power
- Calculate motor full load efficiency and full load power factor

### Solution

- Iron plus friction and windage loss,  $P_i + P_{fw} = P_{nl} = 1063.74$  Watts  
Stator Copper loss,  $P_{st}(30^\circ\text{C}) = 3 \times (16.1/\sqrt{3})^2 \times 0.264 = 68.43$  Watts  
 $P_i + P_{fw} = P_{nl} - P_{st} = 1063.74 - 68.43 = 995.3$
- Stator Resistance at 120°C,  $R(120^\circ\text{C}) = 0.264 \times (120 + 235) / (30 + 235) = 0.354$  ohms
- Stator copper losses at full load,  
 $P_{st}(120^\circ\text{C}) = 3 \times (57/\sqrt{3})^2 \times 0.354 = 1150.1$  Watts
- Full load slip  
 $S = (1500 - 1475) / 1500 = 0.0167$   
Rotor input,  $P_r = P_{output} / (1 - S) = 34000 / (1 - 0.0167) = 34577.4$  Watts
- Motor full load input power  
 $P_{input} = P_r + P_{st(120^\circ\text{C})} + P_i + P_{fw} + P_{stray} = 34577.4 + 1150.1 + 995.3 + (0.005 \times 34000) = 36892.8$  Watts
- Motor efficiency at full load Efficiency =  $P_{out}/P_{input} \times 100 = 34000/36892.8 \times 100 = 92.2\%$   
Full Load PF =  $P_{input} / (\sqrt{3} \times V \times I) = 36892.8 / (\sqrt{3} \times 415 \times 57) = 0.9$

### Comments:

- The measurement of stray load losses is very difficult and not practical even on test beds.
- The actual value of stray loss of motors up to 200HP is likely to be 1% to 3% compared to 0.5 % assumed by standards.
- The value of full load slip taken from the name plate data is not accurate. Actual measurement under full load conditions will give better results.

- d) The friction and windage losses really are part of the shaft output; however, in the above calculation, it is not added to the rated shaft output, before calculating the rotor input power. The error however is minor.
- e) When a motor is rewound, there is a fair chance that the resistance per phase would increase due to winding material quality and the losses would be higher. It would be interesting to assess the effect of a nominal 10 % increase in resistance per phase.

## 2.5 Energy-Efficient Motors

Energy-efficient motors are the ones in which, design improvements are incorporated specifically to increase operating efficiency over motors of standard design. Design improvements focus on reducing intrinsic motor losses. Improvements include the use of lower-loss steel a longer core (to increase active material), thicker wires (to reduce resistance), thinner laminations, smaller air gap between stator and rotor, copper instead of aluminum bars in the rotor, superior bearings and a smaller fan, etc. energy-efficient motors now available operate with efficiencies that are typically 3 to 4 percentage points higher than standard motors. As per the standard IEC 60034-30-1, energy-efficient motors are designed to operate without loss in efficiency at loads between 75 % and 100 % of rated capacity. This may result in major benefits in varying load applications. The power factor is about the same or may be higher than for standard motors. Furthermore, energy-efficient motors have lower operating temperatures and noise levels, greater ability to accelerate higher-inertia loads, and are less affected by supply voltage fluctuations.

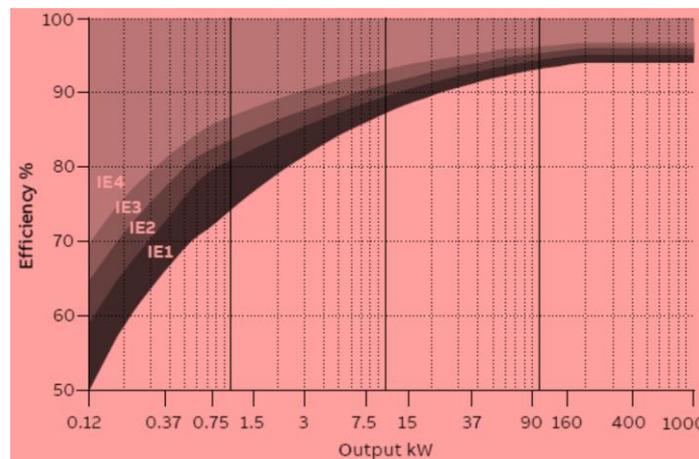


Figure 2.5: IE efficiency classes for 4 pole motors at 50 Hz

**Measures adopted for energy efficiency address each loss specifically as under:**

### 2.5.1 Stator and Rotor $I^2R$ Losses

These losses are major losses and typically account for 55% to 60% of the total losses.  $I^2R$  losses are heating losses resulting from current passing through stator and rotor conductors.  $I^2R$  losses are the function of a conductor resistance, the square of current. Resistance of conductor is a function of conductor material, length and cross-sectional area. The suitable selection of copper conductor size will reduce the resistance. Reducing the motor current is most readily accomplished by decreasing the magnetizing component of current. This involves lowering the operating flux density and possible shortening of air gap. Rotor  $I^2R$

losses are a function of the rotor conductors (usually Aluminum) and the rotor slip. Utilization of copper conductors will reduce the winding resistance. Motor operation closer to synchronous speed will also reduce rotor  $I^2R$  losses.

### **2.5.2 Core Losses**

Core losses are those found in the stator-rotor magnetic steel and are due to hysteresis effect and eddy current effect during 50 Hz magnetization of the core material. These losses are independent of load and account for 20 – 25 % of the total losses.

The hysteresis losses which are a function of flux density, are reduced by utilizing low-loss grade of silicon steel laminations. The reduction of flux density is achieved by suitable increase in the core length of stator and rotor. Eddy current losses are generated by circulating current within the core steel laminations. These are reduced by using thinner laminations.

### **2.5.3 Friction and Windage Losses**

Friction and Windage losses result from bearing friction, windage and circulating air through the motor and account for 8–12% of total losses. These losses are independent of load. The reduction in heat generated by stator and rotor losses permits the use of smaller fan. The windage losses also reduce with the diameter of fan leading to reduction in windage losses.

### **2.5.4 Stray Load-Losses**

These losses vary according to square of the load current and are caused by leakage flux induced by load currents in the laminations and account for 4 to 5% of total losses. These losses are reduced by careful selection of slot numbers, tooth/slot geometry and air gap.

As a result of the modifications to improve performance, the costs of energy-efficient motors are higher than those of standard motors. The higher cost will often be paid back rapidly in saved operating costs, particularly in new applications or end-of-life motor replacements. In cases where existing motors have not reached the end of their useful life, the economics will be less clearly positive.

Energy efficient motors cover a wide range of ratings and the full load efficiencies are higher by 3 to 7 %. The mounting dimensions are also maintained as per BDS 1196:1988 to enable easy replacement.

Because the favorable economics of energy-efficient motors are based on savings in operating costs, there may be certain cases which are generally economically ill-suited to energy-efficient motors. These include highly intermittent duty or special torque applications such as hoists and cranes, traction drives, punch presses, machine tools, and centrifuges. In addition, energy efficient designs of multi-speed motors are generally not available. Furthermore, most energy-efficient motors produced today are designed only for continuous duty cycle operation.

Given the tendency of over sizing on the one hand and ground realities like; voltage, frequency variations, efficacy of rewinding in case of a burnout, on the other hand, benefits

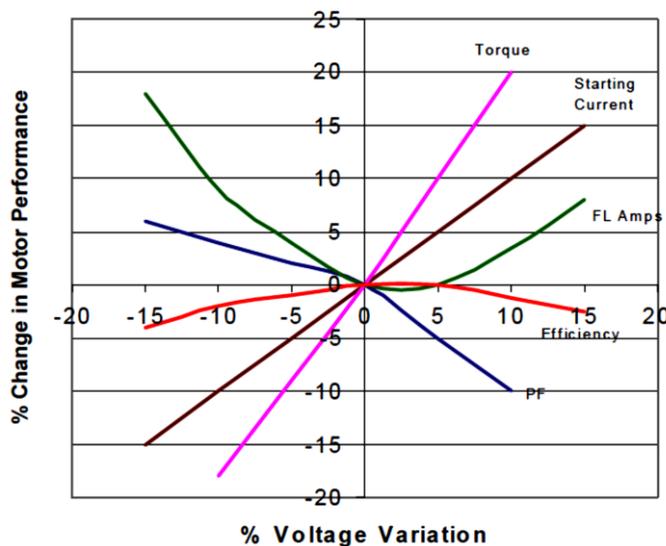
of EEM's can be achieved only by careful selection, implementation, operation and maintenance efforts of energy managers.

<i>Energy Efficient Motors</i>	
<i>Power Loss Area</i>	<i>Efficiency Improvement</i>
1. Iron	Use of thinner gauge, lower loss core steel reduces eddy current losses. Longer core adds more steel to the design, which reduces losses due to lower operating flux densities.
2. Stator $I^2R$	Use of more copper and larger conductors increases cross sectional area of stator windings. This lowers resistance (R) of the windings and reduces losses due to current flow (I).
3. Rotor $I^2R$	Use of larger rotor conductor bars increases size of cross section, lowering conductor resistance (R) and losses due to current flow (I).
4. Friction & Windage	Use of low loss fan design reduces losses due to air movement.
5. Stray Load Loss	Use of optimized design and strict quality control procedures minimizes stray load losses.

## 2.6 Factors Affecting Energy Efficiency & Minimizing Motor Losses in operation

### 2.6.1 Power Supply Quality

Motor performance is affected considerably by the quality of input power that is the actual volts and frequency available at motor terminals vis-à-vis rated values as well as voltage and frequency variations and voltage unbalance across the three phases. Motors in Bangladesh must comply with standards set by the Bangladesh Standards and Testing Institution (BSTI) for tolerance to variations in input power quality. The BSTI standards



specify that a motor should be capable of delivering its rated output with a voltage variation and frequency variation. Voltage fluctuations can have detrimental impacts on motor performance. The general effects of voltage and frequency variation on motor performance are presented in following figure 2.6:

Figure 2.6: Effect of Voltage Variation on Induction Motors

### Effect of Frequency Variation

Motors built in accordance to NEMA standards are designed to operate successfully at rated load and at rated voltage with a variation in the frequency of up to 5% above or below the rated frequency. The information in the following table is based on voltage being held constant.

Freq.	Starting and Max. Torque	Synchronous Speed	% Slip	Full Load Speed	Full Load Eff	Full Load PF	Full Load Current	Locked Rotor Current	Temp. Rise @ Full Load	Max. Overload Capacity	Magnetic Noise (No load)
105%	Decrease 10%	Increase 11%	Practically No Change	Increase 5%	Slight Increase	Slight Increase	Decrease Slightly	Decrease 5-6%	Decrease Slightly	Decrease Slightly	Decrease Slightly
95%	Increase 11%	Decrease 10%	Practically No Change	Decrease 5%	Slight Decrease	Slight Decrease	Increase Slightly	Increase 5-6%	Increase Slightly	Increase Slightly	Increase Slightly

### Effect of Voltage Variation

Induction motors are normally designed to give satisfactory performance on a line voltage of up to 10% above or 10% below the rated value per NEMA standards

Voltage	Starting and Max. Torque	Synchronous Speed	% Slip	Full Load Speed	Full Load Eff	Full Load PF	Full Load Current	Locked Rotor Current	Temp. Rise @ Full Load	Max. Overload Capacity	Magnetic Noise (No load)
110%	Increase 21%	No change	Decrease 17%	Increase 1%	Increase 0-1 point	Decrease 2-8 points	Decrease 0-7%	Increase 10-14%	Decrease 4-6OC	Increase 21%	Increase slightly
90%	Decrease 21%	No change	Increase 23%	Decrease 1%	Decrease 1-3 points	Increase 1-3 points	Increase 10-12%	Decrease 10-12%	Increase 4-8OC	Decrease 19%	Decrease slightly

**The options available for an energy manager to ensure near to rated voltage at motor terminals include:**

- a) Load end power factor improvement by providing matching PF capacitors.
- b) Minimizing line / cable voltage drops from sub-station to motor terminals.
- c) Transformers tap changing as required in case of consistent and continuous low voltage situations.

Voltage unbalance, the condition where the voltages in the three phases are not equal, can be still more detrimental to motor performance and motor life. Unbalance typically occurs as a result of supplying single-phase loads disproportionately from one of the phases. It can also result from the use of different sizes of cables in the distribution system.

**Table 2.1: Example of the Effect of Voltage Unbalance on Motor Performance**

Unbalance in current (%)	Percent unbalance in voltage*		
	0.30	2.30	5.40
	0.4	17.7	40.0
Increased temperature rise( <sup>0</sup> C)	0	30	40

Percent unbalance in voltage is defined as  $100 (V_{\max} - V_{\text{avg}}) / V_{\text{avg}}$ , Where  $V_{\max}$  and  $V_{\text{avg}}$  are the largest and the average of the three phase voltages, respectively.

**The NEMA (National Electrical Manufacturers Association of USA) standard definition of voltage unbalance is given by the following equation:**

Voltage unbalance = Maximum deviation from mean of  $V_{ab}, V_{bc}, V_{ca}$   
 Mean of ( $V_{ab}, V_{bc}, V_{ca}$ )

For example, if the line voltages are  $V_{ab} = 410, V_{bc} = 417,$  and  $V_{ca} = 408$

$$\% \text{ Voltage unbalance} = (417 - 411.7 / 411.667) \times 100 = 1.29 \%$$

Where,

$$\text{Mean} = (410 + 417 + 408) / 3 = 411.7$$

Hence the voltage unbalance is 1.29%.

**Common Causes of Voltage Unbalance**

It is recommended that the voltage unbalance at the motor terminals not exceed 1%, anything above this will lead to de rating of the motor. The common causes of voltage unbalance are

Some of the more common causes of unbalanced voltages are:

- Unbalanced incoming utility supply
- Unequal transformer tap settings
- Large single phase distribution transformer on the system
- Open phase on the primary of a 3 phase transformer on the distribution system
- Faults or grounds in the power transformer
- Open delta connected transformer banks

- A blown fuse on a 3 phase bank of power factor improvement capacitors
- Unequal impedance in conductors of power supply wiring
- Unbalanced distribution of single phase loads such as lighting
- Heavy reactive single phase loads such as welders

Voltage unbalance is probably the leading power factor problem that results in motor overheating and premature motor failure.

Voltage unbalance causes extremely high current imbalance. The magnitude of current imbalance may be 6 to 10 times as large as the voltage imbalance. A motor will run hotter when operating on a power supply with voltage unbalance. The additional temperature rise is estimated with the following equation

$$\text{Additional temperature rise} = 2 \times (\% \text{ Voltage unbalance})^2$$

For example, if the voltage unbalance is 2% for a motor operating at 100°C, the additional temperature rise will be 8°C. The winding insulation life is reduced by one half for each 10 °C increase in operating temperature.

The options that an Energy Manager can exercise to minimize voltage unbalance include:

- Balancing any single phase loads equally among all the three phases
- Segregating any single phase loads which disturb the load balance and feed them from a separate line / transformer.

## 2.6.2 Motor Loading

### Measuring Load

Knowing the load on the motor over its typical operating cycle is critical to understanding the potential for improving motor use efficiency. Under-loading and variable loading can produce inefficient motor operation. However, it is normally quite difficult to ascertain the load on the motor, as it requires measuring input power, current, voltage, frequency and motor speed under both load and no-load conditions. Measurement of the stator resistance is also required. It is generally inadequate to measure only the current drawn under load, as this can give misleading results. The no-load measurements provide the basis for estimating fixed losses, which, together with the measurements under load, permit motor efficiency to be estimated (IEEE, 1984). Proper instrumentation is critical to making accurate measurements.

### Reducing Under-loading

Probably the most pervasive practice contributing to sub- optimal motor efficiency is that of under-loading. Under-loading results in lower efficiency and power factor, and higher than necessary first cost for the motor and related control equipment. Under-loading is common for several reasons. Original equipment manufacturers tend to use a large safety factor in motors they select. Under-loading of the motor may also occur from under-utilization of the equipment. For example, machine tool equipment manufacturers provide for a motor rated for the full capacity load of the equipment. The user may need this full

capacity rarely, resulting in under-loaded operation most of the time. Another common reason for under-loading is selection of a larger motor to enable the output to be maintained at the desired level even when input voltages are abnormally low. Finally, under-loading also results from selecting a large motor for an application requiring high starting torque where a special motor, designed for high torque, would have been suitable.

A careful evaluation of the load would determine the capacity of the motor that should be selected. Another aspect to consider is the incremental gain in efficiency achievable by changing the motor. Larger motors have inherently higher rated efficiencies than smaller motors. Therefore, their placement of motors operating at 60–70% of capacity or higher is generally not recommended. However, there are no rigid rules governing motor selection; the savings potential needs to be evaluated on a case-to-case basis. When downsizing, it may be preferable to select an energy-efficient motor, the efficiency of which may be higher than that of a standard motor of higher capacity.

For motors which consistently operate at loads below 50% of rated capacity, an inexpensive and effective measure might be to operate in star mode. A change from the standard delta operation to star operation involves re-configuring the wiring of the three phases of power input at the terminal box.

Operating in the star mode leads to a voltage reduction by a factor of ' $\sqrt{3}$ '. Motor output falls to one-third of the value in the delta mode, but performance characteristics as a function of load remain unchanged. Thus, full-load operation in star mode gives higher efficiency and power factor than partial load operation in the delta mode. However, motor operation in the star mode is possible only for applications where the torque-to-speed requirement is lower at reduced load.

For applications with high initial torque and low running torque needs, Delta-Star starters are also available in market, which help in load following de-rating of electric motors after initial start-up.

### **Sizing to Variable Load**

Industrial motors frequently operate under varying load conditions due to process requirements. A common practice in cases where such variable-loads are found is to select a motor based on the highest anticipated load. In many instances, an alternative approach is typically less costly, more efficient, and provides equally satisfactory operation. With this approach, the optimum rating for the motor is selected on the basis of the load duration curve for the particular application. Thus, rather than selecting a motor of high rating that would operate at full capacity for only a short period, a motor would be selected with a rating slightly lower than the peak anticipated load and would operate at overload for a short period of time. Since operating within the thermal capacity of the motor insulation is of greatest concern in a motor operating at higher than its rated load, the motor rating is selected as that which would result in the same temperature rise under continuous full-load operation as the weighted average temperature rise over the actual operating cycle. Under extreme load changes, e.g. frequent starts / stops, or high inertia loads, this method of calculating the motor rating is unsuitable since it would underestimate the heating that would occur.

Where loads vary substantially with time, in addition to proper motorizing, the control strategy employed can have a significant impact on motor electricity use. Traditionally, mechanical means (e.g. throttle valves in piping systems) have been used when lower output is required. More efficient speed control mechanisms include multi-speed motors, eddy-current couplings, fluid couplings, and solid-state electronic variable speed drives.

### 2.6.3 Power Factor Correction

As noted earlier, induction motors are characterized by power factors less than unity, leading to lower overall efficiency (and higher overall operating cost) associated with a plant's electrical system. Capacitors connected in parallel (shunted) with the motor are typically used to improve the power factor. The impacts of PF correction include reduced kVA demand (and hence reduced utility demand charges), reduced  $I^2R$  losses in cables up stream of the capacitor (and hence reduced energy charges), reduced voltage drop in the cables (leading to improved voltage regulation), and an increase in the overall efficiency of the plant electrical system.

The size of capacitor required for a particular motor depends upon the no-load reactive kVA (KVAR) drawn by the motor, which can be determined only from no-load testing of the motor. In general, the capacitor is then selected to not exceed 90% of the no-load KVAR of the motor. (Higher capacities could result in over-voltages and motor burn-outs). Alternatively, typical power factors of standard motors can provide the basis for conservative estimates of capacitor ratings to use for different size motors.

**Table 2.2: Capacitor ratings for power factor correction by direct connection to induction motors**

Motor Rating(HP)	Capacitor rating (KVAR) for Motor Speed					
	3000	1500	1000	750	600	500
5	2	2	2	3	3	3
7.5	2	2	3	3	4	4
10	3	3	4	5	5	6
15	3	4	5	7	7	7
20	5	6	7	8	9	10
25	6	7	8	9	9	12
30	7	8	9	10	10	15
40	9	10	12	15	16	20
50	10	12	15	18	20	22
60	12	14	15	20	22	25
75	15	16	20	22	25	30
100	20	22	25	26	32	35
125	25	26	30	32	35	40
150	30	32	35	40	45	50
200	40	45	45	50	55	60
250	45	50	50	60	65	70

Since a reduction in line current, and associated energy efficiency gains, are reflected backwards from the point of application of the capacitor, the maximum improvement in overall system efficiency is achieved when the capacitor is connected across the motor

terminals, as compared to somewhere further upstream in the plant's electrical system. However, economies of scale associated with the cost of capacitors and the labor required to install them will place an economic limit on the lowest desirable capacitor size.

Energy managers can, by a motor load survey, arrive at capacitor ratings, locations and cost benefits. One factor to be considered "operating hours" of motor.

#### **2.6.4 Maintenance**

Inadequate maintenance of motors can significantly increase losses and lead to unreliable operation. For example, improper lubrication can cause increased friction in both the motor and associated drive transmission equipment. Resistance losses in the motor, which rise with temperature, would increase. Providing adequate ventilation and keeping motor cooling ducts clean can help dissipate heat to reduce excessive losses. The life of the insulation in the motor would also be longer: for every 10<sup>0</sup>C increase in motor operating temperature over the recommended peak, the time before rewinding would be needed is estimated to be halved.

A check list of good maintenance practices to help ensure proper motor operation would include:

- Inspecting motors regularly for wear in bearings and housings (to reduce frictional losses) and for dirt/dust in motor ventilating ducts (to ensure proper heat dissipation).
- Checking load conditions to ensure that the motor is not over or under loaded. A change in motor load from the last test indicates a change in the driven load, the cause of which should be understood.
- Lubricating appropriately. Manufacturers generally give recommendations for how and when to lubricate their motors. Inadequate lubrication can cause problems, as noted above. Over-lubrication can also create problems, e.g. excess oil or grease from the motor bearings can enter the motor and saturate the motor insulation, causing premature failure or creating a fire risk.
- Checking periodically for proper alignment of the motor and the driven equipment. Improper alignment can cause shafts and bearings to wear quickly, resulting in damage to both the motor and the driven equipment.
- Ensuring that supply wiring and terminal box are properly sized and installed. Inspect regularly the connections at the motor and starter to be sure that they are clean and tight.

#### **2.6.5 Age**

Most motor cores are manufactured from silicon steel or de-carbonized cold-rolled steel, the electrical properties of which do not change measurably with age. However, poor maintenance (inadequate lubrication of bearings, insufficient cleaning of air cooling passages, etc.) can cause a deterioration in motor efficiency overtime. Ambient conditions can also have a detrimental effect on motor performance. For example, excessively high temperatures, high dust loading, corrosive atmosphere, and humidity can impair insulation properties; mechanical stresses due to load cycling can lead to mis-alignment. However, with adequate care, motor performance can be maintained.

### 2.6.6 Rewinding Effects on Energy Efficiency

It is common practice in industry to rewind burnt-out motors. The population of rewind motors in some industries exceeds 50% of the total population. Careful rewinding can sometimes maintain motor efficiency at previous levels, but in most cases, losses in efficiency result. Rewinding can affect a number of factors that contribute to determining motor efficiency: winding and slot design, winding material, insulation performance, and operating temperature. For example, a common problem occurs when heat is applied to strip old windings: the insulation between laminations can be damaged, thereby increasing eddy current losses. A change in the air gap may affect power factor and output torque.

However, if proper measures are taken, motor efficiency can be maintained, and in some cases increased, after rewinding. Efficiency can be improved by changing the winding design, though the power factor could be affected in the process. Using wires of greater cross section, slot size permitting, would reduce stat or losses thereby increasing efficiency. However, it is generally recommended that the original design of the motor be preserved during the rewind, unless there are specific, load-related reasons for redesign.

The Impact of rewinding on motor efficiency and power factor can be easily assessed if the no-load losses of a motor are known before and after rewinding. Maintaining documentation of no-load losses and no-load speed from the time of purchase of each motor can facilitate assessing this impact.

#### Monitoring Format for Rewound Motors

Section	Equipment Code	Motor Code	Motor Type		No Load Current				Starter Resistance/phase		No load loss	
			Sq.Cage	Slip Ring	New Motor		After Rewinding		New	Rewound	New	Rewound
					A	V	A	V			Watts	Watts

### 2.6.7 Speed Control of AC Induction Motors

Traditionally, DC motors have been employed when variable speed capability was desired. By controlling the armature (rotor) voltage and field current of a separately-excited DC motor, a wide range of output speeds can be obtained. DC motors are available in a wide range of sizes, but their use is generally restricted to a few low speed, low-to-medium power applications like machine tools and rolling mills because of problems with mechanical commutation at large sizes. Also, they are restricted for use only in clean, non-hazardous areas because of the risk of sparking at the brushes. DC motors are also expensive relative to AC motors.

Because of the limitations of DC systems, AC motors are increasingly the focus for variable speed applications. Both AC synchronous and induction motors are suitable for variable speed control. Induction motors are generally more popular, because of their

ruggedness and lower maintenance requirements. AC induction motors are inexpensive (half or less of the cost of a DC motor) and also provide a high power to weight ratio (about twice that of a DC motor).

An induction motor is an asynchronous motor, the speed of which can be varied by changing the supply frequency. The speed can be also be varied through a number of other means, including varying the input voltage, varying the resistance of the rotor circuit, using multi- speed windings, using mechanical means such as gears and pulleys or static voltage and frequency converters. The control strategy to be adopted in any particular case will depend on a number of factors including investment cost, load reliability and any special control requirements. Thus, for any particular application, a detailed review of the load characteristics, historical data on process flows, the features required of the speed control system, the electricity tariffs and the investment costs would be a prerequisite to the selection of a speed control system.

The characteristics of the load are particularly important. Load refers essentially to the torque output and corresponding speed required. Loads can be broadly classified as constant torque, variable torque and constant power. Constant torque loads are those for which the output power requirement may vary with the speed of operation but the torque does not vary. Conveyors, rotary kilns, and constant-displacement pumps are typical examples of constant torque loads. Variable torque loads are those for which the torque enquired varies with the speed of operation. Centrifugal pumps and fans are typical examples of variable torque loads (torque varies as the square of the speed). Constant power loads are those for which the torque requirements typically change inversely with speed. Machine tools are atypical example of a constant power load.

The largest potential for electricity savings with variable speed drives is generally in variable torque applications, for example centrifugal pumps and fans, where the power requirement changes as the cube of speed. Constant torque loads are also suitable for VSD application.

Quantifying the magnitude and temporal variation of the load in any specific case is difficult without adequate instrumentation and an effective monitoring system. For example, for pumping application, measured data on variations in flow, pressure, temperature, head, voltage, current and power would be required along with matched data on production, product mix or grade (if applicable), power supply interruptions, changes due to seasonal variations (if applicable), planned or unplanned shut downs, and effect of start- upon process energy requirements. Extensive data of this form typically are not available. However, in many applications, equipment is over sized due to the safety margins applied at each stage of the system design, in which cases a simpler analysis based on fewer or not significant savings are possible.

### **Electro-Mechanical Speed Control Systems**

Electro-mechanical speed control mechanisms include purely mechanical systems, multi-speed motors, eddy-current drives, and fluid couplings. The characteristics of these are summarized in the table below.

**i. Gears, pulleys, etc.**

A variety of purely mechanical systems are available for motor speed control, including variable pulley sheaves, gears, chains, and friction drives. These systems provide limited speed variation and do not lend themselves to automatic control, but are suitable for applications which require operation at a few fixed speeds and adjustment at frequent intervals.

**ii. Multi-speed motors**

Motors can be wound such that two speeds, in the ratio of 2:1, can be obtained. Motors can also be wound with two separate windings, each giving 2 operating speeds, for a total of four speeds. Multi-speed motors can be designed for applications involving constant torque, variable torque, or for constant output power. Multi-speed motors are suitable for applications which require limited speed control (two or three fixed speeds instead of continuously variable speed), in which cases they tend to be very economical.

*Table 2.3: Electro Mechanical Speed Control alternatives for AC Induction Motors*

VSD Type (Power, Speed Range)	Advantages	Disadvantages
<b>ELECTRO-MECHANICAL CONTROL METHODS</b>		
Gears, Pulleys, etc.		
Variable Pulley Sheaves	Low Cost	Low power savings; high maintenance costs
Gears	Low Cost	Low power savings; high maintenance costs
Chains	Low Cost	Low power savings; high maintenance costs
Friction Drives	Low Cost	Low power savings; high maintenance costs
Multispeed Motors	Operational at 2 or 4 Fixed Speeds	Stepped speed control; lower efficiency than single-speed motors
Eddy-current Drives >0kW, 10:1	Simple; relatively low cost; step less speed control	Needs low efficiency at below 50% rated speed
Fluid Coupling Drives >0 kW, 5: 1	Simple; relatively low cost; stepless speed control	Low efficiency at below 50% rated speed

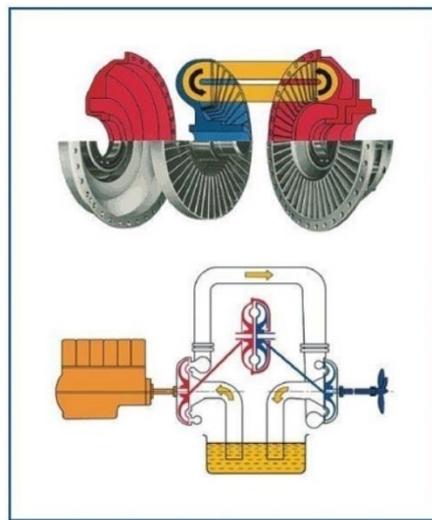
**iii. Eddy-current drives**

This method employs an eddy-current clutch to vary the output speed. The clutch consists of a primary member coupled to the shaft of the motor and a freely revolving secondary member coupled to the load shaft. The secondary member is separately excited using a DC field winding. The motor starts with the load at rest and a DC excitation is provided

to the secondary member which induces eddy-currents in the primary member. The interaction of the fluxes produced by the two currents gives rise to a torque at the load shaft. By varying the DC excitation the output speed can be varied to match the load requirements. The major disadvantage of this system is relatively poor efficiency, particularly at low speeds.

#### iv. Fluid-coupling drives

Power can be transmitted through a fluid either by hydro-static, hydro-kinetic, hydro-viscous, or hydro-dynamic forces. In most commercial fluid coupling drives, normally a light mineral oil, which in turn transmits the energy to the load.



*Figure 2.7: Fluid Coupling*

#### Construction

Fluid couplings work on the hydrodynamic principle. It consists of a pump-generally known as impeller and a turbine generally known as rotor, both enclosed suitably in a casing. The impeller and the rotor are bowl-shaped and have large number of radial vanes. They face each other with an air gap. The impeller is suitably connected to the prime mover while the rotor has a shaft bolted to it. This shaft is further connected to the driven machine through a suitable arrangement. Oil is filled in the fluid coupling from the filling plug provided on its body. A fusible plug is provided on the fluid coupling which blows off and drains out oil from the coupling in case of sustained overloading.

#### Operating Principle

There is no mechanical inter-connection between the impeller and the rotor and the power is transmitted by virtue of the fluid filled in the coupling. When the impeller is rotated by the prime mover, the fluid flows out radially and then axially under the action of centrifugal force. It then crosses the air gap to the runner and is directed towards the bowl

axis and back to the impeller. To enable the fluid to flow from impeller to rotor it is essential that there is difference in head between the two and thus it is essential that there is difference in RPM known as slip between the two. Slip is an important and inherent characteristic of a fluid coupling resulting in several desired advantages. As the slip increases, more and more fluid can be transferred. However when the rotor is at a standstill, maximum fluid is transmitted from impeller to rotor and maximum torque is transmitted from the coupling. This maximum torque is the limiting torque. The fluid coupling also acts as a torque limiter.

### **Characteristics**

Fluid coupling has a centrifugal characteristic during starting thus enabling no-load start-up of prime mover, which is of great importance. This feature also provides inherent overload protection. The slipping characteristic of fluid coupling provides a wide range of choice of power transmission characteristics. By varying the quantity of oil filled in the fluid coupling, the normal torque transmitting capacity can be varied. The maximum torque or limiting torque of the fluid coupling can also be set to a pre-determined safe value by adjusting the oil filling. The fluid coupling has the same characteristics in both directions of rotation.

The losses in this system are circulation and slip losses. Circulation losses are a constant percentage of the rated capacity of the unit—typically 1.5%. Slip losses are the product of torque output and slip speed (motor speed minus load speed). If the slip speed increases significantly, a higher motor rating, or a motor designed for high slip may have to be selected.

### **v. Variable Pitch Drives Description**

This method of speed control uses the mechanical means of belts and variable pitch sheaves or pulleys to change speed. The power source is a standard induction motor. Often these units are enclosed and have a gear reducer built in for reduced speed ranges. The horse power range is generally limited from 5 to 50HP with not much available outside of that range.

### **Variable Pitch Features**

First cost – These systems are among the lowest cost methods of achieving variable speed.  
Simplicity – The principle of operation is well known and easy to understand. Also the construction is simple.

### **Variable Pitch Disadvantages**

Control – Remote control is not an inherent feature. Since the drive uses mechanical means to vary the speed, electrical control signals must be adapted to existing mechanical controls.

Belt wear–The stress of variable speed operation requires periodic checks and replacement of the belts.

High inertia loads – This condition may cause problems. It may require over sizing the drive or custom motors. Special shutdown and start up procedures may be required to prevent overloading the motor.

Sheave Wear – running at a constant speed for extended periods of time may cause grooving in the sheaves. This degrades speed control and decreases belt life.

## **Motor Speed Control Systems:**

### **Multi-speed motors**

#### **Direct Current Drives (DC)**

The DC drive technology is the oldest form of electrical speed control. The drive system consists of a DC motor and a controller. The motor is constructed with armature and field windings. Both of these windings require a DC excitation for motor operation. Usually the field winding is excited with a constant level voltage from the controller.

Then, applying a DC voltage from the controller to the armature of the motor will operate the motor. The armature connections are made through a brush and commutator assembly. The speed of the motor is directly proportional to the applied voltage.

The controller is a phase controlled bridge rectifier with logic circuits to control the DC voltage delivered to the motor armature. Speed control is achieved by regulating the armature voltage to the motor. Often a tachogenerator is included to achieve good speed regulation. The tachogenerator would be mounted on the motor and produces a speed feedback signal that is used within the controller.

#### **Wound Rotor AC Motor Drives (Slip Ring Induction Motors)**

Wound rotor motor drives use a specially constructed motor to accomplish speed control. The motor rotor is constructed with windings which are brought out of the motor through slip rings on the motor shaft. These windings are connected to a controller which places variable resistors in series with the windings. The torque performance of the motor can be controlled using these variable resistors. Wound rotor motors are most common in the range of 300 HP and above.

### **Slip Power Recovery Systems**

Slip power recovery is a more efficient alternative speed control mechanism for use with slip-ring motors. In essence, a slip power recovery system varies the rotor voltage to control speed, but instead of dissipating power through resistors, the excess power is collected from the slip rings and returned as mechanical power to the shaft or as electrical power back to the supply line. Because of the relatively sophisticated equipment needed, slip power recovery tends to be economical only in relatively high power applications and where the motor speed range is 1:5 or less.

## Application of Variable Speed Drives (VSD)

Although there are many methods of varying the speeds of the driven equipment such as hydraulic coupling, gear box, variable pulley etc., the most possible method is one of varying the motor speed itself by varying the frequency and voltage by a variable frequency drive.

### Concept of Variable Frequency Drive

The speed of an induction motor is proportional to the frequency of the AC voltage applied to it, as well as the number of poles in the motor stator. This is expressed by the equation:

$$\text{RPM} = (f \times 120) / p$$

Where f is the frequency in Hz, and p is the number of poles in any multiple of 2.

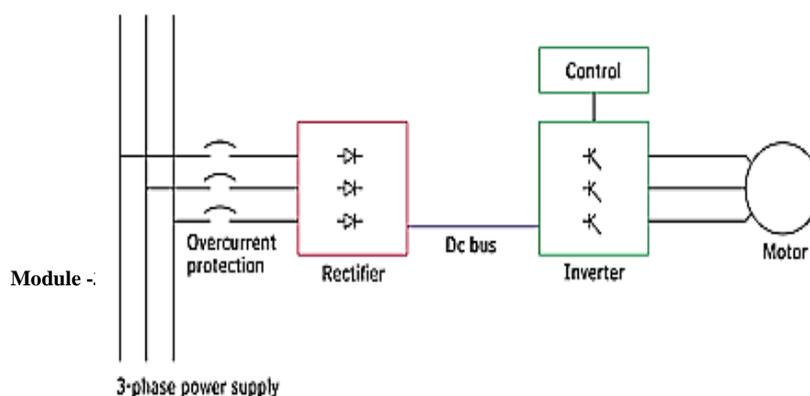
Therefore, if the frequency applied to the motor is changed, the motor speed changes in direct proportion to the frequency change. The control of frequency applied to the motor is the job given to the VSD.

The VSD's basic principle of operation is to convert the electrical system frequency and voltage to the frequency and voltage required to drive a motor at a speed other than its rated speed. The two most basic functions of a VSD are to provide power conversion from one frequency to another, and to enable control of the output frequency.

### Need for VFD

Earlier motors tended to be over-designed to drive a specific load over its entire range. This resulted in a highly inefficient driving system, as a significant part of the input power was not doing any useful work. Most of the time, the generated motor torque was more than the required load torque.

In many applications, the input power is a function of the speed like fan, blower, pump and so on. In these types of loads, the torque is proportional to the square of the speed and the power is proportional to the cube of speed. Variable speed, depending upon the load requirement, provides significant energy saving. A reduction of 20% in the operating speed of the motor from its rated speed will result in an almost 50% reduction in the input power to the motor. This is not possible in a system where the motor is directly connected to the supply line. In many flow control applications, a mechanical throttling device is used to limit the flow. Although this is an effective means of control, it wastes energy because of the high losses and reduces the life of the motor valve due to generated heat



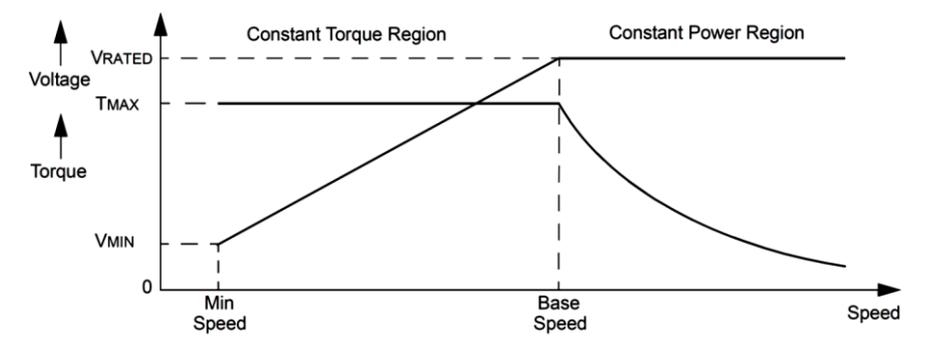
*Figure: 2.8 Components of a Variable Speed Drive*

### Principles of VFD's

The VFD is a system made up of active/passive power electronics devices (IGBT, MOSFET, etc.), a high speed central controlling unit and optional sensing devices, depending upon the application requirement. A typical modern-age intelligent VFD for the three phase induction motor is shown in Figure 2.8.

The basic function of the VFD is to act as a variable frequency generator in order to vary speed of the motor as per the user setting. The rectifier and the filter convert the AC input to DC with negligible ripple. The inverter, under the control of the microcontroller, synthesizes the DC into three-phase variable voltage, variable frequency AC.

The base speed of the motor is proportional to supply frequency and is inversely proportional to the number of stator poles. The number of poles cannot be changed once the motor is constructed. So, by changing the supply frequency, the motor speed can be changed. But when the supply frequency is reduced, the equivalent impedance of electric circuit reduces. This results in higher current drawn by the motor and a higher flux. If the supply voltage is not reduced, the magnetic field may reach the saturation level. Therefore, in order to keep the magnetic flux within working range, both the supply voltage and the frequency are changed in a constant ratio. Since the torque produced by the motor is proportional to the magnetic field in the air gap, the torque remains more or less constant throughout the operating range.



*Figure 2.9: V/f Control*

As seen in Figure 2.9, the voltage and the frequency are varied at a constant ratio up to the base speed. The flux and the torque remain almost constant up to the base speed. Beyond the base speed, the supply voltage cannot be increased. Increasing the frequency beyond the base speed results in the field weakening and the torque reduces. Above the base speed, the torque governing factors become more nonlinear as the friction and windage losses increase significantly. Due to this, the torque curve becomes nonlinear. Based on the motor type, the field weakening can go up to twice the base speed. This control is the most popular in industries and is popularly known as the constant V/f control.

By selecting the proper V/f ratio for a motor, the starting current can be kept well under control. This avoids any sag in the supply line, as well as heating of the motor. The VFD also provides over current protection. This feature is very useful while controlling the motor with higher inertia. Since almost constant rated torque is available over the entire operating range, the speed range of the motor becomes wider. User can set the speed as per the load requirement, thereby achieving higher energy efficiency (especially with the load where power is proportional to the cube speed). Continuous operation over almost the entire range is smooth, except at very low speed. This restriction comes mainly due to the inherent losses in the motor, like frictional, windage, iron, etc. These losses are almost constant over the entire speed. Therefore, to start the motor, sufficient power must be supplied to overcome these losses and the minimum torque has to be developed to overcome the load inertia.

A single VFD has the capability to control multiple motors. The VFD is adaptable to almost any operating condition.

### **VFD Selection**

The size of the VFD depends mainly on driven load type and characteristics. This will determine the drive capacity in terms of full load current (FLC) and power delivered (kW).

### **Driven Load Types and Characteristics**

Mechanical load, which is the load on the motor shaft, can be of two types- Constant Torque (CT) or Variable Torque (VT). There is a basic difference between the two loads with respect to load torque variation at different speeds.

A CT load implies that the load torque seen at motor shaft is independent of motor speed. This means that the load torque remains approximately the same at all speeds. Examples of CT loads include material handling conveyers, reciprocating & screw compressors and certain types of blowers such as roots blower.

A VT load implies that the load torque seen at the motor shaft is dependent upon the motor speed.

Examples of VT loads include centrifugal fans & pumps and centrifugal compressors.

The graphs (Figures 2.10 & 2.11) below describe the torque requirements at various speeds.

*Load Torque*                      *Motor Shaft Power*

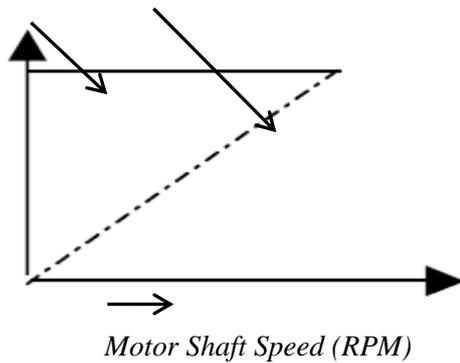


Figure 2.10 A CT load characteristic

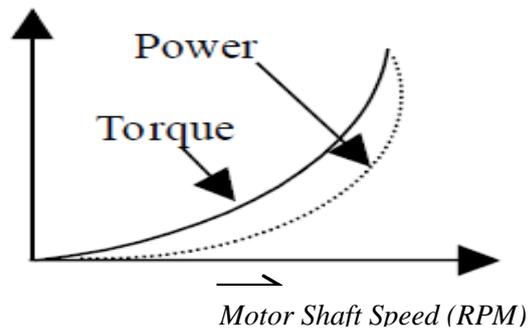


Figure 2.11 A VT load characteristic

### Variable Frequency Drives: Precautions

- Ensure that the power voltage supplied to VFDs is stable within plus or minus 10% to prevent tripping faults.
- Motors operating at low speeds can suffer from reduced cooling. For maximum protection on motors to be run at low speeds, install thermal sensors that interlock with the VFD control circuit. Standard motor protection responds only to over-current.
- Speed control wiring, which is often 4 mA to 20 mA or 0 VDC to 5 VDC, should be separated from other wiring to avoid erratic behavior. Parallel runs of 115V and 24V control wiring may cause problems.
- Precautions for specifying, installing and operating VFDs are numerous. Improper installation and start-up accounts for 50 % of VFD failures
- Use the VFD start-up sheet to guide the initialization check prior to energizing the VFD for the first time.
- Corrosive environments, humidity above 95%, ambient air temperatures exceeding 40<sup>0</sup>C (104<sup>0</sup>F), and conditions where condensation occurs may damage VFDs.
- If a VFD is started when the load is already spinning, the VFD will try to pull the motor down to a low, soft-start frequency. This can result in high current and a trip unless special VFDs are used.
- Switching from grid power to emergency power while the VFD is running is not possible with most types of VFDs. If power switching is anticipated, include this capacity in the specification.
- If electrical disconnects are located between the VFD and motor, interlock the run- permissive circuit to the disconnect.
- If a motor always operates at rated load, a VFD will increase power use, due to electrical losses in the VFD.

### Harmonics

A key concern with VSD operation is the generation of harmonics-multiples of the fundamental frequency (50Hz) which result in a non-sinusoidal high for inverters with only a few pulses per second as compared to more complex ones that produce a 12-step (or higher) voltage waveform or that employ pulse width modulation.

Harmonics increase motor losses, and can adversely affect the operation of sensitive auxiliary equipment. Then on-sinusoidal supply results in harmonic currents in the stator

which increases the total current draw. In addition, the rotor resistance (or more precisely impedance) increases significantly at harmonic frequencies, leading to less efficient operation. Also, stray load losses can increase significantly at harmonic frequencies. Overall motor losses increase by about 20% with a six-step voltage wave form compared to operation with a sinusoidal supply. In some cases the motor may have to be derated as a result of the losses. Alternatively, additional circuitry and switching devices can be employed to minimize losses.

Motor instability, characterized by hunting of the rotor (a phenomenon where the rotor accelerates or decelerates about a stable speed), may occur for certain critical frequency ranges and loading conditions. Motors are inherently unstable at low frequencies. Instability can also occur due to the interaction between the motor and the converter. This is especially true of motors of low rating, which have low inertia. Instabilities can be reduced by changing the machine parameters (e.g., motor impedance) or by employing closed loop feedback systems. The VSD supplier should be consulted to ascertain the impact of inverters on motor performance.

Harmonics can also contribute to low power factor. Shunt capacitors that might be used to compensate for low power factor also generate harmonics. If parallel resonance occurs between the shunt capacitor and VSD (at frequency determined by the capacitance and system inductance), the voltage waveform may undergo significant distortion causing motor overheating, capacitor failure or mis-operation of the control. To avoid resonance, reactive filters can be designed for the VSD. Filters are designed to eliminate resonance at the fifth and seventh frequency harmonics which are the most harmful. Low level distortions may still be present, but these do not pose a serious problem.

### **2.6.8 Other operating concerns**

Steep voltage transients, e.g. caused by switching of shunt capacitors or transient system faults, can affect VSD operation. Isolation transformers or special VSD inverter designs can be used to prevent such transients from tripping or otherwise affecting VSD operation. A power supply with unstable voltage and frequency also hampers regenerative braking, thus eroding efficiency.

When a totally-enclosed fan-cooled motor is controlled to operate at very low speed, some derating may be required if the cooling is inadequate.

In cases where some of the operating time is at close to rated load, system efficiency might be increased by using a bypass arrangement to avoid the losses in the VSD. The VSD would be engaged at lower loads, when the VSD-related reductions in power would more than off-set in efficiencies of the VSD. Such a bypass arrangement would also provide for greater reliability in case of a VSD failure.

### **2.6.9 Transmission Efficiency**

Power transmission equipment linking motors to driven machines including shafts, belts, chains, and gears should be properly selected, installed and maintained. When possible, use flat belts in place of V-belts. Helical gears are more efficient than worm gears; use worm gears only with motors under 10hp. As far as possible it is better to have a direct drive thus avoiding losses in motor power transmission to the driven machine.

*Power transmission Efficiency margin from V belt drive to flat belt drive 5 to 6 % and from Worm gearbox to Helical Gearbox 8 to 10 %*

### **2.6.10 Soft Starters**

A soft starter is another form of reduced voltage starter for A.C. induction motors which facilitates gradual acceleration of motor and consequent elimination of shocks during starting (soft start). The starting current with a soft starter is only 1.5 to 2 times the full load current as against 5–7 times in the case of other conventional starters. Because of this cable sizes, contactors and motors can be sized lower during the initial selection. The soft starter is similar to a primary resistance or primary reactance starter in that it is in series with the supply to the motor. The current into the starter equals the current out. The soft starter employs solid state devices to control the current flow and therefore the voltage applied to the motor.

#### **Working of Soft Starter / Energy Saver**

- i. The soft-start motor starter combines the advantages of reduced voltage starting and reduced energy consumption, in one compact economic package. Because they are solid state, burnout of starter parts are eliminated. The reduced voltage starter gracefully increases motor voltage from the set minimum voltage to full voltage allowing the motor to smoothly and gradually accelerate to full speed, when it is set on soft-start mode.
- ii. The soft starter offers a ramp input voltages source to a motor which can be programmable, and this factor is the essence of soft starting as against conventional step voltage starter or D.O.L. starter.

#### **Advantages:**

- Nanosecond response on account of micro-electronics involved.
- Reduced power consumption and hence reduced energy bills, at part loads.
- Power factor improvement on a continuous basis
- Reduction in maximum demand
- Reduced peak motor current during starting
- Reduced motor temperature, increased motor life and decreased maintenance needs
- When using a generator, the power generated can be used to operate more equipment, since the starting and running currents of motors are lower, when motors are fitted with energy-saver-soft-starters.

#### **Typical Applications**

Plastic injection moulding machines, Machine tools / motor generator welding sets, chemical process equipment, unloading type air compressors, punch presses, center less

grinding and polishing machines, blowers, wherever the motor load is varying continuously.

## 2.7 Case Studies

### CASE STUDY- 1

Based on study of humidification plants in a Rayon industry, it was established that the air flow demand in supply air and return air varies as per season and the supply and return fans rated 16 kW, 12 kW respectively were running at constant load. Towards energy efficiency improvement, the drives were converted variable frequency drives. The part load effects one summarized as follows:

Supply Frequency C/S	Supply Air Fan		Return Air Fan	
	KW Drawn	% Drop in	KW Drawn	% Drop in
50	16.4	Capacity Ref.	12	Capacity Ref.
45	12.0	10.0	8.7	10.0
40	8.4	20.0	6.1	20.0
35	5.6	30.0	4.1	30.0
30	3.5	40.0	2.6	40.0

As the flow requirements vary, the following schedule was adopted with power savings indicated alongside.

Months	Recommended	Average %kW Savings
October, Nov, Dec, Jan	35	51
Feb, March, Sept.	40	35
April, August, July	45	18
May, June	50	-

Covering all 10fans, with 142kW consumption among five sets of supply air & return air fans, the savings achieved were 377223kWh worth BDT.13.16lakhs/yea against an investment of BDT 12.5lakhs @ BDT 1.25lakh/ drive yielding a simple payback period of less than one year.

### CASE STUDY-2

Refrigeration plant study in a chemical complex indicated part load operation of reciprocating compressor drives with unloader mechanism, as illustrated below:

Item Reference	Brine Unit	Chilled Water Unit
Motor input kW	53	74.1

Rpm	750	780
Op. hours / day	14.5	9.0
% ON time	58	38
Power consumption/day (kWh)	768.5	666.9

The Compressor speed can be increased or decreased with a change in the drive pulley or the driven pulley or in some cases, both pulleys. By pulley diameter modification, the machines were de-rated to match required capacity. A higher sized reciprocating compressor operating with higher unloading percentage was downsized by reducing the motor (drive) pulley size from 12” to 7.5” in case of Brine unit and 12” to 6” in case of Chilled water unit .

Despite increased use hours due to derating effects, the specific power consumption reduced as the evaporator and condenser became oversized for the de-rated condition, and the energy consumption reduced as well as follows:

Item Reference	Brine Unit	Chilled Water Unit
Motor input kW	32.3	35.6
Speed Rpm	480	409.5
Op. hours / day	18.0	14.0
Average energy consumption/day (kWh)	581.4	498.4

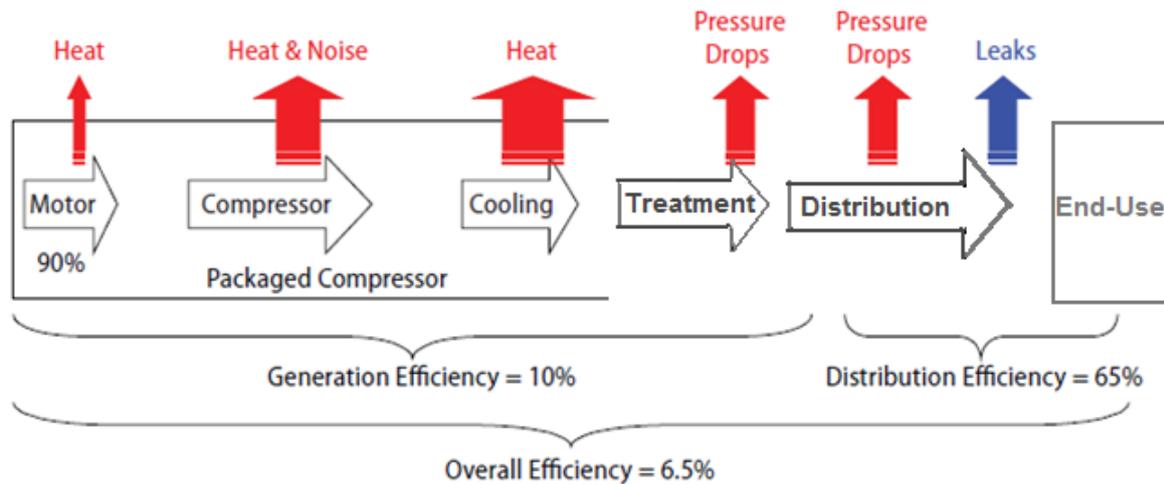
The energy savings of 355.6kWh/day by the simple modifications, are a significant 25%. The interesting part of the exercise was that the investment is nominated, and the pay back is in order of days. Most cases, where load-unload cycling of reciprocating machines takes place for capacity control, pulley diameter modification offers a simple, cheap solution to derate the machine capacity with energy savings potential of significant order.

## **Chapter 3: COMPRESSED AIRSYSTEMS**

### **3.1 Introduction**

Air compressors account for a significant amount electricity used in industrial sector. Air compressors are used in a variety of industries to supply process requirements, operate

pneumatic tools and equipment, and for instrumentation. The generation efficiency is only about 10% as shown in the figure 3.1 and balance 90% of energy of the power of the prime mover being converted to unusable heat energy and to a lesser extent lost in form of friction, misuse and noise. Further considering the distribution efficiency, the overall efficiency can be as low as 6.5% when pressure drops, leaks, and part-load control losses are considered.



*Figure 3.1: Efficiency of Compressed Air System*

## 3.2 Compressor Types

Compressors are broadly classified as dynamic and positive displacement types.

Dynamic compressors are centrifugal compressors and are further classified as radial and axial flow types. Positive displacement compressors are further classified as reciprocating and rotary compressors, under which there are further sub-classifications.

Dynamic compressors increase the fluid's velocity, which is then converted to increased pressure at the outlet. Positive displacement compressors increase the pressure of the gas by reducing the volume. The flow and pressure requirements of a given application determine the suitability of a particular type of compressor.

## 3.3 Positive Displacement Compressors

### 3.3.1 Reciprocating Compressors

In industry, reciprocating compressors are the most widely used type for both air and refrigerant compression. They are characterized by a flow output that remains nearly constant over a range of discharge pressures. Also, the capacity is directly proportional to the speed. The output, however, is a pulsating one. Reciprocating compressors are available in many configurations, the four most widely used of which are horizontal, vertical, horizontal balance-opposed and tandem. Vertical type reciprocating compressors are used in the capacity range of 50–150cfm. Horizontal balance opposed compressors are used in the capacity range of 200–5000 cfm in multi-stage design and upto 10,000 cfm in single stage designs. In refrigeration cycles, reciprocating compressors are best suited for applications requiring up to 150 tone of refrigeration.

Reciprocating compressors are available in variety of types, such as single or multiple cylinder, single or multi stage, lubricated and non-lubricated, and water or air cooled. Non-lubricated compressors are especially useful for providing instrument air and for processes, which require oil free discharge. However non-lubricated machines have marginally higher specific power consumption (kW/cfm) as compared to lubricated models. In the case of lubricated machines, oil has to be separated from the discharge. Single cylinder machines are generally air-cooled while multi-cylinder machines are water cooled, though multi-stage air cooled models are available for machines up to 100 kW. Two stage machines are used for high pressures and are characterized by lower discharge temperature (140 – 160<sup>0</sup>C) compared to single-stage machines (205 – 240<sup>0</sup>C). In some cases, multi-stage machines may have a lower specific power consumption compared to single stage machines operating over the same total pressure differential. Multi-stage machines generally have higher investment costs, particularly for applications with high discharge pressure (up to 7 bar) and low capacities (less than 25 cfm). Multi staging has other benefits, however, including a reduced pressure differential across cylinders, which reduces the load and stress on valves and piston rings.

### **3.3.2 Rotary Compressors**

Rotary compressors have rotors in place of pistons and give a continuous, pulsation free discharge. They operate at high speed and generally provide higher throughput than reciprocating compressors. They are directly coupled to the prime mover and require lower starting torque as compared to reciprocating machines. Also they do not require specialized foundations, operate with limited vibration, and have a lower number of parts - which means less failure rate. Among rotary machines, the lobe compressor (also known as a Roots blower) and screw compressors are among the most widely used. The lobe compressor is essentially a low-pressure blower and is limited to a discharge pressure of 1bar in single-stage design and up to 2.2 bar in two stage design. Screw compressors are high speed, high capacity machines and are available in both dry and oil injected types with single or twin screw compressors deliver oil-free air and are available in sizes up to 20,000cfm and pressure up to 15 bar, and in the size range of 100 – 1000cfm in lubricated designs upto 10 bar discharge pressure. They cannot operate at discharge pressures below 3.5 bar. In the case of oil injected (lubricated) rotary screw compressors, oil has to be separated from the discharged air (gas). In refrigeration systems, screw compressors are preferable for intermediate temperatures (10 to 50<sup>0</sup>C).

Other types of rotary compressors such as rotary vane and liquid-ring compressors are also available. Rotary vane compressors are small in size and have a pulsation-free output. The two common types of rotary vane compressors are oil-flooded with non-metallic blades and water-cooled with steel blades. Both designs have many of the advantages of the oil injected screw compressor and are generally designed as multi-stage machines. Single-stage designs are available for discharge pressures up to 7 bar. Liquid ring compressors operate on the same principle as rotary vane compressors. They have comparatively high power consumption and are generally designed for discharge pressures up to 5.5 bar.

### **3.4 Centrifugal Compressors**

Centrifugal compressors have appreciably different characteristics as compared to reciprocating machines. A small change in compression ratio produces a marked change in compressor output and efficiency. They operate on the same principle as centrifugal

pumps. Centrifugal machines are less efficient than reciprocating machines, but are better suited for applications requiring very high capacities, typically above 12,000cfm in the case of air (gas) compression, and for high capacity (150–5800 tons), low temperature (10–100°C) refrigeration applications. Centrifugal machines are low in initial cost and compact in size for large capacity requirements (a single stage centrifugal machine may provide the same capacity as a multi-stage reciprocating compressor). Machines with either radial or axial flow impellers are available. Axial compressors have characteristics similar to positive displacement compressors. Also, they are suitable for higher compression ratios and are 5–10 % more efficient than radial compressors. Axial compressors typically are multi-stage machines while radial machines are usually single-stage designs.

**Table 3.1: General Selection Criteria for Compressors**

Type of Compressor	Capacity (m <sup>3</sup> /h)		Pressure(bar)	
	From	To	From	To
Roots power compressor single stage	100	30000	0.1	1
Reciprocating				
- Single / Two stage	100	12000	0.8	12
- Multi stage	100	12000	12.0	700
Screw				
- Single stage	100	2400	0.8	13
- Two stage	100	2200	0.8	24
Centrifugal	600	300000	0.1	450

### Comparison of Different Compressors

The power consumption of various compressors depends on the operating pressure, free air delivery and efficiency etc. The variations in power consumption during unloading/part load operation are more significant and depend on the type of compressor and method of capacity control. The relative efficiencies and part load power consumption of different compressors are given in Table 3.2

**Table 3.2 Comparison of Different Compressors**

Item	Reciprocating	Rotary vane	Rotary Screw	Centrifugal
Efficiency at full load	High	Medium-high	High	High
Efficiency at part load	High due to staging	Poor: below 60% of full load	Poor: below 60% of full load	Poor: below 60% of full load
Efficiency at no load (power as % of full load)	High (10-25%)	Medium (30-40%)	High-poor (25-60%)	High-medium (20-30%)

In case of reciprocating machines, the unload power consumption is in the order of 25% of full load power. While in screw compressors, the unload power consumption is marginally higher compared to reciprocating machines.

It is preferable to use screw compressors for constant air requirement. If screw compressors have to be installed for fluctuating loads, it is desirable to have screw compressor with variable speed drive to further optimize unload power consumption.

Some of the plants have adopted the strategy of operating screw compressor at full load for meeting the base-load requirement and reciprocating compressor for fluctuating load to optimize on unload power consumption.

### 3.5 System Components

Compressed air systems, depending on the requirement, consist of a number of components compressors filters, air dryers, inter-coolers, after coolers, oil separators, valves, nozzles, and piping.

- **Intake Air Filters:** Prevent dust and atmospheric impurities from entering compressor. Dust causes sticking valves, scored cylinders, excessive wear etc.
- **Inter-stage Coolers:** Reduce the temperature of the air(gas) before it enters the next stage to reduce the work of compression and increase efficiency. They can be water-or air-cooled.
- **After Coolers:** Reduce the temperature of the discharge air, and thereby reduce the moisture carrying capacity of air.
- **Air-dryers:** Air dryers are used to remove moisture, as air for instrument and pneumatic equipment needs to be relatively free of any moisture. The moisture is removed by using adsorbents or refrigerant dryers, or state of the art heat less dryers.
- **Moisture Traps:** Air traps are used for removal of moisture in the compressed air distribution lines. They resemble steam traps in which the air is trapped and moisture is removed.
- **Receivers:** Depending on the system requirements, one or more air receivers are generally provided to reduce output pulsations and pressure variations.

### 3.6 Compressor Performance

#### Capacity of a Compressor: Free Air Delivery (FAD)

Free air, as defined by CAGI (Compressed Air & Gas Institute) is air at ATMOSPHERIC conditions at any specific location. Because the barometer and temperature may vary at different localities and at different times, it follows that this term does not mean air under standard conditions.

Measured in CFM (Cubic feet per minute) this is the amount of compressed air converted back to the actual inlet (free air) conditions before it was compressed. In other words, the volume of air, which is drawn in from the atmosphere by the compressor, then compressed and delivered at a specific pressure.

#### Compressor Efficiency Definitions

Compressor efficiency is often expressed as either an adiabatic or isothermal or mechanical efficiency. These are computed as the isothermal and adiabatic power respectively, divided by the actual power consumption. The calculation of isothermal power excludes that needed to overcome friction and generally gives an efficiency that is lower than adiabatic efficiency. This is an important consideration when selecting compressors based on reported values of efficiency. Manufacturers generally provide the

adiabatic (theoretical) horsepower required for compression. The actual power intake would be slightly higher because of mechanical losses.

For practical purposes, the most effective guide in comparing compressor efficiencies is the specific power consumption for different compressors that would provide identical duty.

### **Isothermal Efficiency**

The iso-thermal efficiency of a multi-stage air compressor can be calculated as a ratio of the theoretical kW required for a duty conditions and the actual kW input measured. This efficiency would reflect the combined efficiency of the compressor and the drive motor and the method can be adopted to assess the performance for identifying margins with respect to rated values, merit rating of compressors, maintenance planning, etc.

$$\text{Theoretical kW} = \left( \frac{NK}{K-1} \right) \left( Q \frac{P_s}{0.612} \right) \left[ \left( \frac{P_d}{P_s} \right)^{\frac{K-1}{NK}} - 1 \right]$$

N = No. of stages

K = Ratio of specific heats (1.35 for air)

$P_s$  = suction pressure in  $\text{kg/cm}^2$

$P_d$  = Discharge pressure in  $\text{kg/cm}^2$

Q = Actual air flow ( $\text{m}^3/\text{min.}$ ) Actual kW =  $\sqrt{3} V I \times \text{PF}$  as measured

$$\text{Efficiency of compressor and motor combination} = \frac{100 \times \text{Theoretical kW}}{\text{Actual kW}}$$

It is also a done thing, to compute and add-up stage wise work of compression (theoretical kW) in case the performance of intercoolers is not optimal.

### **Volumetric Efficiency**

$$\text{Volumetric efficiency} = \frac{\text{Free air delivered}(\text{m}^3 / \text{min})}{\text{Compressor displacement}(\text{m}^3 / \text{min})} \times 100$$

$$\text{Compressor Displacement} = \frac{\pi}{4} \times D^2 \times L \times S \times \chi \times n$$

<i>D</i>	=	<i>Cylinder bore, metre</i>
<i>L</i>	=	<i>Cylinder stroke, metre</i>
<i>S</i>	=	<i>Compressor speed rpm</i>
<i>χ</i>	=	<i>1 for single acting and 2 for double acting cylinders</i>

$$n = \text{No. of cylinders}$$

### 3.7 Efficient Compressor Operation

#### 3.7.1 Reciprocating Compressors

The capacity of reciprocating compressors can be controlled by throttling suction or discharge pressure, by using the cylinder expansion (clearance) volume, bypassing gas externally, using cylinder un-loaders, or by speed control. External bypass involves feeding some of the output back to the suction (the gas may have to be cooled before being fed back to the compressor). Though this method can provide 0 – 100 % capacity control, it does not result in any significant power reduction at reduced flow. However, at very low capacities, gas bypass is generally the only solution. Other methods may result in unstable operation. Throttling is another alternative, but saves relatively little energy. The most common and more efficient method is to employ automatic or manual cylinder un-loaders or to use clearance pockets in the cylinder. Reduction in power consumption is proportional to capacity control. The most efficient method of controlling capacity, however, is to vary the speed of the prime movers since capacity is directly proportional to the speed.

The performance of reciprocating compressors varies based on the altitude, inlet air (gas) temperature, discharge pressure, the effectiveness of inter stage cooling and operational speed. Increase in altitude, with the resultant reduction in pressure, increases the compression ratio leading to higher discharge temperature and reduced efficiency. However, for a given compressor ratio, the specific power requirement, which varies directly with the suction pressure, decreases with an increase in altitude. The effect of altitude on multi- staged compressors is slight.

The discharge pressure should be kept at the minimum required for the process or for operation of pneumatic equipment for a number of reasons, including minimizing power consumption. Also, compressor capacity varies inversely with discharge pressure. Another disadvantage of higher discharge pressures is the increased loading on compressor piston rods and their subsequent failure. Lower pressure also results in lower leakage losses. In general, when compressing gas starting at ambient temperature and pressure through a pressure ratio exceeding 4, multistage compression with inter cooling should be considered to maintain the temperature of the compressed gas.

Multi-stage compressors are usually provided with inter coolers to reduce the temperature of the air (gas) discharged between stages. Ideally, the intake temperature at each stage should be the same as that at the first stage (referred to as perfect cooling) so that the volume of air to be compressed does not increase. The provision of well- designed inter cooling systems reduces power consumption. However, use of very cold water can result in condensation which may result in water entering the cylinder, thereby reducing valve life, accelerating wear and scoring of piston, piston rings and cylinder. Condensation can also occur when the relative humidity of inlet air is high and the compressor cylinder temperature is lower than inlet air temperature (for example, as a result of higher than required flow of cooling water). The condensed water may also wash away the oil film on the cylinder and cause rust which will result in an abrasion during compressor operation and significantly reduce efficiency. At the other extreme, if cooling is

insufficient, the discharge temperature increases. High temperature operation reduces oil viscosity and the oil film thickness can be reduced.

The location of the compressor should be considered during the selection process. Locations with high moisture or high temperature can cause operational trouble and increase power consumption. Cooling should be adequate in such cases. When locating the suction inside buildings, care should be taken to ensure adequate clearance between the suction and the walls to reduce pulsation and vibrations.

### **3.7.2 Centrifugal Compressors**

Manufacturers specify discharge pressure and power requirement as a function of the inlet volumetric flow rate. These performance curves are valid at only the given inlet conditions. Therefore, to optimize process efficiency and to predict performance at different process and inlet conditions, performance curves which provide the polytropic head and polytropic efficiency is a function of the inlet volume flow is useful for such analysis. The manufacturer should be consulted in such cases.

The major limitation of a centrifugal compressor is that it operates at peak efficiency at design point only and any deviation from the operating point is detrimental to its performance. When selecting centrifugal compressors, close attention should be paid during system design to ensure that at high pressure, with the consequent reduction in flow, the surge point is not reached. Surge point is the point on the performance curve where a further decrease in flow (typically in the region of 50 - 70 % of rated capacity) causes instability, resulting in a pulsating flow, which may lead to overheating, failure of bearings due to thrust reversals, or excessive vibration. Bypass valves are commonly used to prevent surging.

Using variable inlet-guide vanes, throttling of suction pressure or throttling of discharge pressure can control the output of centrifugal compressors. However, since centrifugal compressors follow the same affinity laws as centrifugal pumps another efficient way to match compressor output to meet varying load requirements is through speed control.

### **3.7.3 Screw Compressors**

The capacity of screw compressors is normally controlled by a hydraulically operated slide valve, which bypasses gas without compression. However, speed control is the most efficient means of capacity control. Unlike reciprocating and centrifugal compressors, screw compressors develop full pressure regardless of speed. Also, they are more stable at low capacity (up to 10% load)—unlike centrifugal compressors which surge at capacities lower than about 50% of the rated value. The volume handled and the power consumption are directly proportional to speed. Therefore, variable speed drives can be used to efficiently control capacity. Part load power consumption of screw compressors is generally higher than that for reciprocating and centrifugal compressors. Additionally, the discharge pressure of the screw compressor should be closely matched to discharge line pressure to avoid over or under compression that result in higher power consumptions. Therefore, when considering screw compressors, the volume ratio should be properly specified.

### 3.8 Energy Efficiency Practices In Compressed Air Systems

#### 3.8.1 Location of Compressors

Location of air compressors and the quality of air drawn by the compressors will have a significant bearing on the amount of energy consumed. Compressor performance as a breathing machine improves with cool, clean, dry air at intake.

##### Cool air intake

Every 4°C rise in inlet air temperature results in a higher energy consumption by 1% to achieve equivalent output. Hence, cool air intake leads to a more efficient compression.

*Table 3.3: Effect of Intake Air temperature on Power Consumption*

Inlet Temperature(°C)	Relative Air Delivery (%)	Power Saved (%)
10.0	102.0	+1.4
15.5	100.0	Nil
21.1	98.1	- 1.3
26.6	96.3	- 2.5
32.2	94.1	- 4.0
37.7	92.8	- 5.0
43.3	91.2	- 5.8

It is preferable to draw cold ambient air, as the temperature of air inside the compressor room will be a few degrees higher than the ambient temperature. A sheltered inlet, protected from rain on a north wall is desirable. While extending air intake to the outside of building, care should be taken to minimize excess pressure drop in the suction line, by selecting a bigger diameter duct with minimum number of bends.

#### 3.8.2 Dust Free Air Intake

Dust in the suction air causes excessive wear of moving parts and results in malfunctioning of the valves due to abrasion. Suitable air filters should be provided at the suction side. Air filters should have high dust separation capacity, low pressure drops and robust design to avoid frequent cleaning and replacement.

*Table 3.4: Effect of Pressure Drop across Air Inlet Filter on Power consumption*

Pressure Drop Across air filter (mmWC)	Increase in Power Consumption (%)
0	0
200	1.6
400	3.2

600	4.7
800	7

Air filters should be selected based on the compressor type and installed as close to the compressor as possible. For every 25mbar pressure lost at the inlet due to choked filters, the compressor performance is reduced by about 2percent. Hence, it is advisable to clean inlet air filters at regular intervals to avoid high-pressure drops. Manometers or differential pressure gauges across filters may be provided for monitoring pressure drops so as to plan filter-cleaning schedules.

**Table 3.5: Comparison of Inlet Air Filters**

<b>Type</b>	<b>Filter Action Efficiency, %</b>	<b>Particle Size, Microns</b>	<b>Pressure Drop when Clean, inches of Water Column</b>	<b>Comments</b>
Dry	100, 99, 98	10, 5, 3	3 to 8	Recommended for non-lubricated compressors in a high dust environment
Dry type with Silencer	100, 99	10, 5	5, 7	Same as above
Oil wetted (viscous impingement)	95, 85	20, 10	0.25 to 2.0	Not recommended for dusty areas or for non-lubricated
Oil bath	98, 90	10, 3	2, 6 to 10	Same as above. Recommended for rotary vane compressors in normal service

### 3.8.3 Dry Air Intake

**Table 3.6: Moisture Levels at Various Humidity Levels**

<b>% Relative Humidity</b>	<b>Kg of water vapour compressed per hour for every 1000 m<sup>3</sup>/min. of air at 30<sup>0</sup>C</b>
50	27.6
80	45
100	68.22

Atmospheric air always contains some amount of water vapour, depending on the relative humidity, being high in foggy or rainy weather. The moisture level will also be high if air is drawn from a damp area, cooling tower exhaust and air conditioner warm outlet air.

It is desirable to draw air with a low relative humidity, as otherwise, energy is consumed to compress the water vapour in the air and again to condense and drain the moisture from inter and after coolers. The moisture-carrying capacity of air increases with a rise in temperature and decreases with increase in pressure.

### 3.8.4 Pre-Cooled Air Intake

By cooling the air entering the compressor, the efficiency of the compressor can be improved. This cooling, usually to 25<sup>0</sup>C is achieved by refrigeration, if the chilled water is available at cheap cost. As the temperature of air is reduced, its volume decreases and a greater mass of air is available for the given compressor. Therefore, due to pre-cooling either more air is delivered for a given power input or the power input is reduced for a required volumetric flow rate. Using pre-cooled dry air can save about 20– 30 % of compressor power requirement. Also, the moisture present inlet air is condensed out giving dry air for compression and saving energy which would otherwise be used for compressing water vapour.

After-coolers and dryers are also eliminated as the pre-cooler performs their functions. This represents another capital cost saving as well as an additional energy saving device. However the economics of cooling the air using chilled water must be viable considering the cost of chilled water available and energy savings in the compressor due the reduced intake air temperature.

### 3.8.5 Elevation

The altitude of a place has a direct impact on the volumetric efficiency of the compressor. The effect of altitude on volumetric efficiency is given below:

*Table 3.7: Effect of Altitude on Volumetric Efficiency*

Altitude Meters	Barometric Pressure	Percentage Relative Volumetric Efficiency Compared with Sea Level	
	Mbar	At 4 bar	At 7 bar
Sea level	1013	100	100
500	945	98.7	97.7
1000	894	97	95.2
1500	840	95.5	92.7
2000	789	93.9	90
2500	737	92.1	87

It is evident that compressors located at higher altitudes consume more power to achieve a particular delivery pressure than those at sea level, as the compression ratio is higher.

### 3.8.6 Cooling Water Circuit

Most of the industrial compressors are water-cooled, where in the heat of compression is removed by circulating cold water to cylinder heads, inter-coolers and after-coolers. The resultant warm water is cooled in a cooling tower and circulated back to compressors. The effect of cooling tower performance, total dissolved solids (TDS) in cooling water, pumps and fans on compressor performance is discussed below:

- **Cooling Tower Performance**

The main purpose of a cooling tower is to reduce the inlet warm water temperature to near the wet bulb temperature of ambient air. Cooling towers are generally designed to have an approach temperature of 2 to 5°C depending upon the type of cooling tower. In practice, because of microbial growth, scale formation, corrosion and improper maintenance, the intimate contact between air and water is disturbed, resulting in high temperature outlet water which will affect compressor inter cooler effectiveness and compressor performance. Proper maintenance of cooling tower is very important to achieve the desired approach temperature.

- **Effect of air wet bulb temperature on cooling tower performance**

The cooling tower performance is affected by atmospheric conditions—particularly by the wet bulb temperature of inlet air. In a given location, the wet bulb temperature changes throughout the year, reaching its peak value only occasionally. It would, therefore, be uneconomical to operate the tower designed on the basis of the maximum wet bulb temperature. A compromise between peak and average conditions has to be adopted. While designing or selecting a cooling tower, “5 %” wet bulb temperature is used which is the wet bulb temperature not exceeded more than 5% of the total number of hours during summer months. This is estimated from the study of local meteorological data.

- **Cooling Tower Pump**

Cooling water is supplied to compressors through centrifugal pumps at a particular pressure and flow rate. Any change in water flow rate or pressure will affect the compressor performance. High efficiency pumps and motors have to be selected, as they run continuously. Interconnecting pipelines, inter-coolers, and after-coolers have to be selected or designed for minimum pressure drop. Generally pumps in a central compressor house are over-designed for safety reasons, and are capable of catering to more than one compressor. During lean seasons and night shift, only one or two compressors are in operation but cooling water is circulated through even in idle compressors. To avoid this waste of water supply, idle compressors should be closed or a water pressure switch fixed in the water line so that compressor can be tripped off whenever the cooling water pressure falls below a pre-set value.

- **Cooling Tower fans**

Cooling tower fans are provided to facilitate more air throughput thus increasing cooling tower efficacy. A malfunction of fan will result in less air to water ratio, change in air distribution pattern etc., which will affect the cooling tower performance. Hence, proper fan maintenance and fan energy management has to be adopted for lower energy consumption.

- Once the cooling water temperature approaches set temperature, the cooling tower fan can be switched off or operated intermittently by providing an interlock between water outlet temperature and fan operation.
- If two speed motors are used the cooling tower fan power requirement can be reduced substantially, whenever the ambient wet bulb temperature decreases.
- Automatic variable pitch propeller type fans and inverter type devices can be incorporated to permit variable fan speeds. These can track the cooling load for a constant outlet water temperature.

- Heavy fan blades made up of metals can be replaced with light weight, and aerodynamically designed blades such as FRP to reduce the initial torque required and the power consumption.
- Speed control of cooling tower fans by fluid-coupling drives can decrease power consumption in motors.

### Cooling Water TDS

In most of the installations, raw water with high TDS is used for compressor cooling. Because of inadequate attention to water quality, the TDS levels may shoot up to unacceptable levels, due to water loss through evaporation, drift and other losses. This leads to increase of scaling in cylinder heads, inter-coolers and after-coolers, which reduces heat exchanger efficacy and compressor capacity. The scaling in compressor and interconnecting pipelines not only reduce its effectiveness but also increases pressure drop and thus, water pumping power.

*Table 3.8: Effect of Scaling on Pressure Drop and Inner Pipe Diameter*

Scaling Thickness (mm)	Inside Diameter Reduced		Pressure Drop
	From (mm)	To (mm)	
0.4	64	63.2	6
0.8	64	62.4	14
1.2	64	61.6	21
4.7	64	54.6	134

Use of treated water or purging a portion of cooling water periodically can maintain TDS levels within acceptable limits. It is better to maintain the water pH by addition of chemicals, and avoid microbial growth by addition of fungicides and algacides.

### 3.8.7 Efficacy of Inter and After Coolers

Inter-coolers are provided between successive stages of a multi-stage compressor to cool the air, reduce its specific volume and condense out excess water. This reduces the power requirement in consecutive stages. Ideally, the temperature of the inlet air at each stage of a multi-stage machine should be the same as it was at the first stage. This is referred to as “perfect cooling”. But in actual practice, because of fouled heat exchangers, due to scaling of dissolved solids in cooling water, the inlet air temperatures at subsequent stages are higher than the normal levels resulting in higher power consumption, as a larger volume is handled for the same duty.

*Table 3.9: Effect of Inter-stage Cooling on Specific Power Consumption of a Reciprocating Compressor*

Details	Imperfect Cooling	Perfect Cooling	Chilled Water Cooling
1 Stage inlet temperature <sup>0</sup> C	21.1	21.1	21.1
2 Stage inlet temperature <sup>0</sup> C	26.6	21.1	15.5

Capacity(m <sup>3</sup> /min)	15.5	15.6	15.7
Shaft Power (kW)	76.3	75.3	74.2
Specific energy consumption kW (m <sup>3</sup> /min)	4.9	4.8	4.7
Percent Change	+ 2.1	-	- 2.1

It can be seen from the table 3.9 that an increase of 5.5<sup>0</sup>C in the inlet to the second stage results in a 2% increase in the specific energy consumption. Use of cold water reduces power consumption. However, use of very cold water could result in condensation of moisture in the air leading to cylinder damage. An after-cooler is located after the final stage of the compressor to reduce air temperature and water content, as far as possible, before air enters the receiver. As time passes, dissolved solids in the cooling water coat the after-coolers, thereby reducing the heat transfer effectiveness. So, fouled after-coolers, allow warm, humid air into the receiver, which causes more condensation in air receivers and distribution lines, which in consequence, leads to increased corrosion. Periodic cleaning of both heat exchangers and cylinder heads are therefore necessary.

**Table 3.10: Table: Cooling Water Requirement**

Compressor Type	Minimum quantity of Cooling Water required for 2.85 m <sup>3</sup> /min. FAD at 7 bar (lpm)
Single-stage	3.8
Two-stage	7.6
Single-stage with after- cooler	15.1
Two-stage with after- cooler	18.9

Inter-cooler and after-cooler efficacy also depends upon the quantity of cooling water circulated through the heat exchanger.

### 3.8.8 Pressure Settings

- **Reducing Delivery Pressure:**

The power consumed by a compressor depends on its operating pressure and rated capacity. They should not be operated above their optimum operating pressures as this not only wastes energy, but also leads to excessive wear, leading to further energy wastage. The volumetric efficiency of a compressor is also less at higher delivery pressures. The possibility of down setting the delivery pressure should be explored by careful study of pressure requirements of various equipment, and the pressure drop in the line between the compressed air generation and utilization points. The pressure switches must be adjusted such that the compressor cut-in and cuts-off at optimum levels.

**Table 3.11: Table: Power Reduction through Pressure Reduction**

Pressure Reduction		Power Reduction (%)		
From(bar)	To (bar)	Single-stage Water-cooled	Two-stage Water-cooled	Two-stage Air-cooled
6.8	6.1	4	4	2.6
6.8	5.5	9	11	6.5

A reduction in the delivery pressure of a compressor would reduce the power consumption. This has been practically achieved, as discussed in the relevant case study.

- **Compressor modulation by Optimum Pressure Settings**

Very often in an industry, different types, capacities and makes of compressors are connected to a common distribution network. In such situations, proper selection of a right combination of compressors and optimal modulation of different compressors can conserve energy. Where more than one compressor feeds a common header, compressors have to be operated in such a way that the cost of compressed air generation is minimal. If all compressors are similar, the pressure setting can be adjusted such that only one compressor handles the load variation, whereas the others operate more or less at full load.

If compressors are of different sizes, the pressure switch should be set such that only the smallest compressor is allowed to modulate. If different types are operated together, for example, both reciprocating and screw compressors, the reciprocating compressor must be allowed to modulate, while keeping the screw compressor at full load always as its part load operation consumes more power. In general, the compressor with lower no-load power consumption must be modulated. Compressors can be graded according to their specific energy consumption, at different pressures and energy efficient ones must be made to meet most of the demand.

*Table 3.12: Expected Specific Power Consumption of Reciprocating Compressors (based on motor input)*

Pressure bar	No. of Stages	Specific Power kW/170 CMHg
1	1	6.29
2	1	9.64
3	1	13.04
4	2	14.57
7	2	18.34
8	2	19.16
10	2	21.74
15	2	26.22

- **Mains air pressure reduction**

It is often necessary to reduce the mains pressure when supply groups of plants or complete workshops. This requires a pressure reducing valve of large capacity and good flow characteristics. In these circumstances, a pressure reducing station may be used.

If the low pressure air requirement is considerable, it is advisable to generate low pressure and high pressure air separately, and feed to the respective sections instead of reducing the pressure through pressure reducing valves, which invariably waste energy.

- **Minimum pressure drop in air lines**

The air mains and their associated branches, hoses, couplings and other accessories offer considerable opportunities for energy conservation.

**Table 3.13: Energy Wastage due to Smaller Pipe Diameter**

Pipe Nominal Bore (mm)	Pressure drop (bar) per 100meters	Equivalent power losses (kW)
40	1.80	9.5
50	0.65	3.4
65	0.22	1.2
80	0.04	0.2
100	0.02	0.1

Excess pressure drop due to inadequate pipe sizing, choked filter elements, improperly sized couplings and hoses represent energy wastage. The above table illustrates the energy wastage, if the pipes are of smaller diameter.

- **Equivalent lengths of fittings**

When long runs of distribution mains are involved, the pressure drops maybe higher than acceptable levels; in such cases it is desirable to check for actual pressure drops.

**Table 3.14: Resistance of Pipe Fittings in Equivalent Lengths (in metres)**

Type of Fitting	Nominal Pipe Size in mm									
	15	20	25	32	40	50	65	80	100	125
Gate Valve	0.11	0.14	0.18	0.27	0.32	0.40	0.49	0.64	0.91	1.20
Run of standard	0.12	0.18	0.24	0.38	0.40	0.52	0.67	0.85	1.20	1.52
Tee 90° long bend	0.15	0.18	0.24	0.38	0.46	0.61	0.76	0.91	1.20	1.52
Elbow	0.26	0.37	0.49	0.67	0.76	1.07	1.37	1.83	2.44	3.20
Return bend	0.46	0.61	0.76	1.07	1.20	1.68	1.98	2.60	3.66	4.88
Through side	0.52	0.70	0.91	1.37	1.58	2.14	2.74	3.66	4.88	6.40
Outlet of tee globe valve	0.76	1.07	1.37	1.98	2.44	3.36	3.96	5.18	7.32	9.45

### 3.8.9 Blowers in place of Compressed Air System

Since the compressed air system is already available, facilities engineers may be tempted to use compressed air to provide air for low pressure requirements such as agitating plating tanks or pneumatic conveying. Using a blower that is designed for lower pressure operation will cost only a fraction of compressed air generation cost.

### 3.8.10 Capacity Control of Compressors

In many installations, the use of air is intermittent. Therefore, some means of controlling the output of the compressor is necessary. This is achieved by regulation of pressure, volume, temperature or some other factors. The type of capacity control employed has a direct impact on the compressor power consumption. Some control schemes commonly used are discussed below:

- **On / Off Control**

Automatic start and stop control, as its name implies, starts or stops the compressor by means of a pressure activated switch as the air demand varies. This is a very efficient method in controlling the capacity of compressor, where the motor idle-running losses are eliminated, as it completely switches off the motor when the set pressure is reached. This is suitable for small compressors (less than 10 kW).

- **Load and Unload**

This is a two-step control where compressor is loaded when there is air demand and unloaded when there is no air demand. During unload, the reciprocating compressor motor runs without air compression, thereby consuming only 20- 30% of the full load power. In screw compressors, the unloading is achieved by closing the inlet valve. The idling power is about 40 to 50 % of the full load power depending configuration, operation and maintenance practices. While in screw compressors, the unload power consumption is marginally higher compared to reciprocating machines.

- **Multi-step Control**

Motor-driven reciprocating compressors above 75 kW are usually equipped with a multi-step control. In this type of control, unloading is accomplished in a series of steps, varying from full load down to no-load. A relevant case study has been appended for this opportunity.

**Table 3.15: Power Consumption of Reciprocating Compressor at Various Loads**

Load %	Power Consumption as % of full load Power
100	100
75	76-77
50	52-53
25	27-29
0	10-12

Five-step control (0%, 25%, 50%, 75% & 100%) is accomplished by means of clearance pockets. In some cases, a movable cylinder head is provided for variable clearance in the cylinder.

- **Throttling Control**

This kind of control is achieved using an inlet valve or a variable–displacement slide valve and is suitable for screw compressors where the capacity can be varied from 40 to 100%. The variable displacement method reduces the volume of air delivered by venting air from a variable portion of the helical screw length to the inlet side of the compressor. The variable displacement method is more efficient than the inlet valve.

The output of centrifugal compressors can be controlled using variable inlet guide vanes to throttle discharge pressure. However, another efficient way to match compressor output to meet varying load requirements is by speed control.

*Table 3.16: Typical part load gas compression: Power input for speed and vane control of centrifugal compressors*

System Volume	Flow % Speed Control	Power Input (%) Vane Control
111	120	-
100	100	100
80	76	81
60	59	64
40	55	50
20	51	46
0	47	43

At low volumetric flow (below 40 %), vane control may result in lower power input compared to speed control due to low efficiency of the speed control system. For loads more than 40%, speed control is recommended.

### 3.8.11 Avoiding Misuse of Compressed Air

Misuse of compressed air for purposes like body cleaning, liquid agitation, floor cleaning, drying, equipment cooling and other similar uses must be discouraged. Wherever possible, low-pressure air from a blower should be substituted for compressed air, for example secondary air for combustion in a boiler / furnace.

The following illustrations gives an idea of savings by stopping use of compressed air by choosing alternative methods to perform the same task.

Electric motors can serve more efficiently than air-driven rotary devices. The following table gives the comparison of pneumatic grinders and electrical grinders. Refer table 3.17 below:

**Table 3.17: Power Requirements for Pneumatic and Electrical Tools**

Tool	Wheel dia (mm)	Speed (rpm)	Air Cons. (m <sup>3</sup> /h)	Power (kW)
Pneumatic angle grinder	150	6000	102 m <sup>3</sup> /h at 6 bar	10.2
Electric angle grinder	150	5700-8600	N.A.	1.95 –2.90
Pneumatic jet grinder	35	30000	32.3 m <sup>3</sup> /h at 6 bar	3.59
Electric straight grinder	25	22900-30500	N.A.	0.18

It may be noted that in some areas use of electric tools are not permitted due to safety constraints, especially places where inflammable vapours are present in the environment. In those cases, possibility of shifting the machine tool operation to outside the flammable area may be considered when evaluating use of electric tools. It should always be remembered that safety consideration always overrides energy conservation.

- *In place of pneumatic hoists, electric hoists can be used. A comparison is given below:*

**Table 3.18: Comparison of Power Consumption of Pneumatic and Electric Hoists**

Capacity	Type	Compressed Air Required CMH	Equivalent Power Consumption at the Compressor, kW	Motor Rating of an Electric Hoist, kW
0.5	Chain	125	12	0.37
1.0	Chain	118	12	0.37
1.5	Chain	118	12	1.125
2	Chain	118	12	1.5
5	Wire rope	200	20.25	2.7

- Material conveying applications can be replaced by blower systems of preferably by a combination of belt/ screw conveyers and bucket elevators. In a paper manufacturing facility, compressed air was used for conveying wood chips.
- For applications like blowing of components, use of compressed air amplifiers, blowers or gravity-based systems may be possible. Brushes can sweep away debris from work in progress as effectively as high-pressure air. Blowers also can be used for this purpose. When moving air really is required for an application, often sources other than compressed air can do the job. Many applications do not require clean, dry, high-pressure and expensive 6 bar or 7 bar compressed air rather, only moving air is needed to blow away debris,

provide cooling, or other functions. In these cases, local air fans or blowers may satisfy the need for moving air much economically.

- Use of compressed air for cleaning should be discouraged; use of vacuum cleaners is an alternative for some applications. If absolutely necessary, compressed air should be used only with blow guns to keep the air pressure below 2 bar; higher pressure are not permitted as per national safety regulations. Use of compressed air amplifiers can also be considered for some cleaning applications.
- For applications where compressed air is indispensable for cleaning internal crevices of machines etc., installation of a separate cleaning air header with a main isolation valve may be considered. The main valve should be opened only for a few, well-defined time periods during the whole day, no connections for cleaning should be provided from process or equipment air lines.
- Replacement of pneumatically operated air cylinders by hydraulic power packs can be considered.
- Use of compressed air for personal comfort cooling can cause grievous injuries and is extremely wasteful; it should be banned from the safety viewpoint alone. Use of man coolers or air washers (in dry areas) may be encouraged. If a ¼” hosepipe is kept open at a 7bar compressed air line for personal cooling for at least 1000 hours/ annum, it can cost about 1,00,000 BDT/ annum.
- Vacuum systems are much more efficient than expensive venturi methods, which use expensive compressed air rushing past an orifice to create a vacuum.
- Mechanical stirrers, conveyers, and low-pressure air may mix materials far more economically than high-pressure compressed air.

### 3.8.12 Avoiding Air Leaks and Energy Wastage

The major opportunity to save energy is in the prevention of leaks in the compressed air system. Leaks frequently occur at air receivers, relief valves, pipe and hose joints, shutoff valves, quick release couplings, tools and equipment. In most cases, they are due to poor maintenance and sometimes, improper installation in underground lines.

#### Air leakages through Different Size Orifices

The following table gives the amount of free air wasted for different nozzle sizes and pressure:

*Table 3.19: Discharge of Air through Orifice (Cd =1.0)*

Gauge Pressure Bar	0.5mm	1mm	2mm	3mm	5mm	10mm	12.5mm
0.5	0.06	0.22	0.92	2.1	5.7	22.8	35.5
1.0	0.08	0.33	1.33	3.0	8.4	33.6	52.5

2.5	0.14	0.58	2.33	5.5	14.6	58.6	91.4
5.0	0.25	0.97	3.92	8.8	24.4	97.5	152.0
7.0	0.33	1.31	5.19	11.6	32.5	129.0	202.0

### Cost of Compressed Air Leakage

It may be seen from the following table that any expenditure on sealing leaks would be paid back through energy saving.

*Table 3.20: Cost of Air Leakage*

Orifice Size mm	KW Wasted	* Energy Waste (BDT/Year)
0.8	0.2	8000
1.6	0.8	32000
3.1	3.0	120000
6.4	12.0	480000

\* based on BDT 5 / kWh; 8000 operating hours; air at 7.0bar

### Steps in simple shop-floor method for leak quantification

- Shut off compressed air operated equipment (or conduct test when no equipment is using compressed air).
- Run the compressor to charge the system to set pressure of operation
- Note the subsequent time taken for 'on load' and 'off load' cycles of the compressors. For accuracy, take ON & OFF times for 8 – 10 cycles continuously. Then calculate total 'ON' Time (T) and Total 'OFF' time (t).
- The system leakage is calculated as

System leakage ( $\text{m}^3/\text{min}$ ) =  $Q \times T / (T + t)$

Q = Actual free air being supplied during trial, in cubic meters per minute

T = Time on load in minutes

t = Time unload in minutes

#### Example 3.1

*In the leakage test in a process industry, following results were observed*

Compressor capacity (m <sup>3</sup> /minute)	=	35
Cut in pressure, kg/cm <sup>2</sup> (g)	=	6.8
Cut out pressure, kg/cm <sup>2</sup> (g)	=	7.5
Load kW drawn	=	188 kW
Unload kW drawn	=	54 kW
Average 'Load' time, T	=	1.5 minutes
Average 'Unload' time, t	=	10.5 minutes

Comment on leakage quantity and avoidable loss of power due to air leakages.

a) Leakage quantity (m <sup>3</sup> /minute), q	=	$\frac{(1.5)}{(1.5)+(10.5)} \times 35$
	=	4.375 m <sup>3</sup> /min
b) Leakage quantity per day, (m <sup>3</sup> /day)	=	4.375 x 24 x 60 = 6300 m <sup>3</sup> /day
c) Specific power for compressed air generation	=	188 kW / (35 x 60) m <sup>3</sup> /hr
	=	0.0895 kWh/m <sup>3</sup>
d) Energy lost due to leakage/day	=	0.0895 x 6300 = 564 kWh

- **Leakage Test by Ultrasonic Leak Detector**

Leakage tests are conducted by a Leak Detector having a sensing probe, which senses when there are leakages. The leak is detected by ultrasonic vibration. Leak testing is done by observing and locating sources of ultrasonic vibrations created by turbulent flow of gases passing through leaks in pressurized or evacuated systems. Use is made of ultrasonic detectors to reveal air borne and structure-borne vibrations, and translators that convert these in audible high frequency sounds to lower frequencies within the range of human hearing. Detection of leaks in compressed air and gas systems at high temperatures, beneath insulated coverings, and in pipelines and manifolds, can be done.

Leak detection and location from a distance through air or other fluids involves remote scanning of suspected leak areas with a directional probe and coordinating the direction of the source of the characteristic hissing sound of the leak with the relative sound intensity. Probably the greatest advantage of ultrasonic leak detection is that this method can be used with any fluid (liquid, gas or vapour) if the physical conditions for sound generation are met in the leak. When leak conditions generate sound in ambient air, leaks can be detected up to and beyond 30 metre (100feet). This offers advantages when extended structures are to be inspected. Ultrasonic mechanical vibration signal energy is converted to electrical signal energy by an appropriate transducer. The single most significant actor to be noted is the frequency distribution of ultrasonic energy from leaks. All leaks possess energy in the 30/50 kHz. At lower pressure of 480 and 70 kPa (70and10psi), it is seen that there is a distinct maximum around 40 kHz.

### 3.8.13 Compressor Capacity Assessment

- **Need for Capacity Assessment**

The compressor capacity is expressed in terms of quantity of free air delivered at a particular pressure. Due to ageing of the compressors and inherent inefficiencies in the internal components, the free air delivered may be less than the design value, despite adherence to good maintenance practices. Sometimes, other factors such as poor maintenance, fouled heat exchanger and effects of altitude also tend to reduce free air delivery. In order to meet the air demand, the inefficient compressor may have to run for more time, thus consuming more power than actually required.

The power wastage depends on the percentage deviation of FAD capacity. For example, a worn out compressor valve can reduce the compressor capacity by as much as 20 percent. A periodic assessment of the FAD capacity of each compressor has to be carried out to check its actual capacity. If the deviations are more than 10%, corrective measures should be taken to rectify the same.

The ideal method of compressor capacity assessment is through a nozzle test where in a calibrated nozzle is used as a load, to vent out the generated compressed air. Flow is assessed, based on the air temperature, stabilization pressure, orifice constant etc.

### **Simple method of Capacity Assessment in Shop-floor**

- Isolate the compressor along with its individual receiver being taken for test from main compressed air system by tightly closing the isolation valve or blank, thus closing the receiver outlet.
- Open water drain valve and drain out water fully and empty the receiver and the pipe line. Make sure that water trap is tightly closed once again to start the test.
- Start the compressor and activate the stopwatch.
- Note the time taken to attain the normal operational pressure P<sub>2</sub> (in the receiver) from initial pressure P<sub>1</sub>.
- Calculate the capacity as per the formulae given below:

**Actual Free air discharge** 
$$Q = \frac{P_2 - P_1}{P_0} \times \frac{V}{T} N \frac{M^3}{Min}$$

P<sub>2</sub> = Final pressure after filling (kg/cm<sup>2</sup> a)

P<sub>1</sub> = Initial pressure (kg/cm<sup>2</sup>a) after bleeding

P<sub>0</sub> = Atmospheric Pressure (kg/cm<sup>2</sup>a)

V = Storage volume in m<sup>3</sup> which includes receiver, after cooler, and delivery piping

T = Time take to build up pressure to P<sub>2</sub> in minutes

### **Example 3.2**

An instrument air compressor capacity test gave the following results – Comment?

Make: ABC

Date: .....

Time: .....

Test No.:1

Piston displacement: 16.88 m<sup>3</sup>/min

Theoretical compressor capacity: 14.75 CMM @ 7 kg/cm<sup>2</sup>(g)

Compressor rated rpm 750: Motor rated rpm 1445

Receiver Volume: 7.79 m<sup>3</sup>

Additional hold up volume,

i.e., pipe / water cooler, etc., is: 0.4974 m<sup>3</sup> Total volume: 8.322 m<sup>3</sup>

Initial pressure P1: 0.5 kg/cm<sup>2</sup>(g)

Final pressure P2: 7.03 kg/cm<sup>2</sup>(g)

Pump up time: 4.021 min

Atmospheric pressure P<sub>0</sub>: 1.026 kg/cm<sup>2</sup>(g)

$$\text{Compressor output CMM} = \frac{(P2 - P1) * \text{Total Volume}}{\text{Atm. Pressure} * \text{Pumpup time}}$$

$$\frac{(7.03 - 0.5) * 8.322}{1.026 * 4.021} = 13.17 \text{ m}^3/\text{min}$$

**Comment:**

Capacity shortfall with respect to 14.75 m<sup>3</sup>/min rating is 1.577 m<sup>3</sup>/min i.e., 10.69% Compressor performance needs to be investigated further.

**Example 3.3**

Assessing compressed air system study for a plant section gave following results. Compressors online A, B, C, D, E

- All reciprocating type
- Trial observation Summary

Compressor Reference	Measured Capacity m <sup>3</sup> /min (@ 7 kg/cm <sup>2</sup> (g))	'On' Load kW	'Unload' kW	Load Time Min.	Unload Time Min.
A	13.17	115.30	42.3	Full time*	0
B	12.32	117.20	51.8	Full time*	0
C	13.14	108.30	43.3	Full time*	0
D	12.75	104.30	29.8	Full time*	0
E	13.65	109.30	39.3	5.88	39.12

\*Compressors running in load conditions and not getting unloaded during normal operations

**Comments:**

- For a cycle time of 45 minutes (39.12 + 5.88)
  - Compressed air generated in m<sup>3</sup>

$$45 (13.17) + 45 (12.32) + 45 (13.14) + 45 (12.75) + 5.88 (13.65) = 2392.36 \text{ m}^3$$

ii. Power consumption kWh

$$45/60 (115.3) + 45/60 (117.20) + 45 / 60 (108.3) + 45/60 (104.3) + 5.88/60 (109.8) + ((39.12) / 60) 39.3 = 370.21 \text{ kWh}$$

iii. Compressed air generation capacity online in  $\text{m}^3$

$$45 [13.17 + 12.32 + 13.14 + 12.75 + 13.65] = 2926.35 \text{ m}^3$$

a) The consumption rate of the section connected

$$2392.36 / 45 \text{ m}^3/\text{min} = 53.16 \text{ m}^3/\text{min}$$

b) Compressor air draw as a % of capacity on line is  $[2392.36 / 2926.35] \times 100 = 81.75\%$

c) Specific power consumption =  $370.21 \text{ kWh} / 2392.36 = 0.155 \text{ kWh} / \text{m}^3/\text{min}$

d) Idle power consumption due to unload operation =  $25.62 \text{ kWh}$  in every 45 minutes cycle i.e.,  $34.16 \text{ kWh}$  every hour.

e) It would be favorable and energy efficient to keep the compressor 'D' in cycling mode on account of lower un-load losses.

f) A suitable smaller capacity compressor can be planned to replace the compressor with highest unload losses.

g) An investigation is called for, as to why such a large variation of unload power drawn, exists although all compressors have almost the same rated capacity.

### 3.8.14 Line Moisture Separator and Traps

Although, in an ideal system, all cooling and condensing of air should be carried out before the air leaves the receiver, this is not very often achieved in practice. The amount of condensation, which takes place in the lines, depends on the efficiency of moisture extraction before the air leaves the receiver and the temperature in the mains itself. In general, the air main should be given a fall of not less than 1 m in 100 m in the direction of air flow, and the distance between drainage points should not exceed 30m.

Drainage points should be provided using equal tees, as it assists in the separation for water. Whenever a branch line is taken off from the mains it should leave at the top so that any water in the main does not fall straight in to the plant equipment. Further, the bottom of the falling pipe should also be drained.

### 3.8.15 Compressed Air Filter

Although some water, oil and dirt are removed by the separators and traps in the mains, still some of it is always left, which is being carried over. Moreover, pipe systems accumulate scale and other foreign matters, such as small pieces of gasket material, jointing compounds and so on. Burnt compressor oil may also be carried over in pipe work, and this, with other contaminants, forms a gummy substance. To remove these, all of which are liable to have deleterious effects on pneumatic equipment, the air should be filtered as near as possible to the point of use. Water and oil collected in the filter sump must be drained off because, if its level is allowed to build up, then it is forced through the filter element into the very system it is designed to protect.

### 3.8.16 Regulators

In many instances, pneumatic operations are to be carried out at a lower pressure than that of the main supply. For these applications, pressure regulators are required to reduce the pressure to the required value and also to ensure that it remains reasonably constant at the usage point. Pilot operated type regulators are energy efficient than direct-acting and self-relieving regulators. In the self-relieving type, a small relief valve is provided which allows excess air to bleed away, should the downstream pressure exceed the set value. It is suitable for applications where the control pressure has to be varied periodically.

### 3.8.17 Lubricators

Where air is used to drive prime movers, cylinders and valves, they should be fitted with a lubricator. Essentially, a lubricator is a reservoir of oil and has been designed so that when air is flowing, metered amount of oil is fed in mist form into the air stream. This oil is carried with the motive air, to the point of use to lubricate all moving parts. All lubricators require a certain minimum rate of air flow to induce oil into their stream. Their design should be such that once the air flow is more than this minimum rate, they give satisfactory lubrication without causing an excessive pressure drop. Light free-fogging, lubricating oil with a high velocity index and without lead additives is suitable for lubrication. The ratio of oil to air can be decided experimentally. A rough guide is, one drop of oil per minute for every 5 dm<sup>3</sup>/s of free air at 5.5 bar pressure.

It is advisable to fix filters, regulators and lubricators as close as possible to the equipment being served. Where lubricators are used to provide oil for linear actuators or when the direction of air flow is reversed, the volume of pipe work between the lubricator and cylinder should not exceed to 50 % of the volume of free air used by the cylinder per stroke.

### 3.8.18 Air Dryers

There are certain applications where air must be free from moisture and have a lower dew point. This calls for more sophisticated and expensive methods to lower the dew point of compressed air. Three common types of air dryers used are heat-less (absorption), absorption and chiller dryers. They produce dry air with 10°C–40°C dew point, depending on the type of dryers.

*Table 3.21: Moisture Content in Air*

Dew point at Atmospheric Pressure (°C)	Moisture Content (ppm)
0	3800

-5	2500
-10	1600
-20	685
-30	234
-40	80
-60	6.5

**Table 3.22: Pressure Dew Point and Power Consumption Data for Dryers**

Type of Dryer	Atmospheric Dew Point (°C)	First Cost	Operating Cost	Power Cons. For 1000 m <sup>3</sup> /hr
Refrigeration	-20	Low	Low	2.9 kW
Desiccant regenerative (by compressed air purging)	-20	Low	High	20.7 kW
Desiccant regenerative (external or internal heating with electrical or steam heater, reduced or no compressed air purging)	-40	Medium	Medium	18.0 kW
Desiccant regenerative (using heated low pressure air, no compressed air loss)	-40	High	Low	12.0 kW
Desiccant regenerative (by recovery of heat of Compression from compressed air)	-40	High	Very low	0.8 kW

### 3.8.19 Air Receivers

The main purpose of a receiver is to act as a pulsation damper, allowing intermittent high demands for compressed air to be met from a small compressor set, resulting in lesser energy consumption.

If receiver is sized too small for air demand, compressor will run for longer periods. By improving the ability of storage to meet air demand, running time of the compressor is minimised, thereby reducing energy usage as well as wear and tear. Compressed air systems usually have one primary receiver and possibly few secondary receivers near high intermittent air using equipment.

The air receiver should be generously sized to give a large cooling surface and even out the pulsation in delivered air pressure from a reciprocating compressor. As per IS 7938-1976, volume of air in receiver (in m<sup>3</sup>) should be 1/10<sup>th</sup> to 1/6<sup>th</sup> of the output in m<sup>3</sup>/min.

A simple formulae often quoted for air receiver size is to take a value equal to one minute's continuous output of the compressor. However, this should be considered indicative of the minimum size of receiver. A better suggestion is to estimate the peak air

consumption likely and allow for the maximum pressure drop that is acceptable at this peak load.

$$\text{Receiver capacity in } m^3 = \frac{\text{m}^3 \text{ of free air volume required}}{\text{permissible pressure drop in bar}}$$

If peaks cannot be quantified, another approximation can be to size the receiver volume to be 5% of the rated hourly free air output. Providing an air receiver n earth load end, where there is sudden high demand lasting for a short period, would avoid the need to provide extra capacity.

### **3.8.20 Capacity Utilisation**

In many installations, the use of air is intermittent. This means the compressor will be operated on low load or no load condition, which increases the specific power consumption per unit of air generated. Hence, for optimum energy consumption, a proper compressor capacity control should be selected. The nature of the control device depends on the function to be regulated. Regulation of pressure, volume, temperature or some of factor determines the type of regulation required and the type of the compressor drive.

## **Chapter 4: FANS AND BLOWERS**

## 4.1 Introduction

Fans and blowers provide air for ventilation and industrial process requirements. Fans generate a pressure to move air (or gases) against a resistance caused by ducts, dampers, or other components in a fan system. The fan rotor receives energy from a rotating shaft and transmits it to the air.

### Difference between Fans, Blowers and Compressors

Fans, blowers and compressors are differentiated by the method used to move the air, and by the system pressure they must operate against. As per American Society of Mechanical Engineers (ASME) the specific ratio - the ratio of the discharge pressure over the suction pressure - is used for defining the fans, blowers and compressors (see Table 5.1).

*Table 4.1 Differences between Fans, Blower and Compressor*

Equipment	Specific Ratio	Pressure rise (mmWg)
Fans	Up to 1.11	1136
Blowers	1.11 to 1.20	1136 – 2066
Compressors	more than 1.20	-

## 4.2 Fan Types

Fan and blower selection depends on the volume flow rate, pressure, type of material handled, space limitations, and efficiency. Fan efficiencies differ from design to design and also by types. Typical ranges of fan efficiencies are given in Table 4.2.

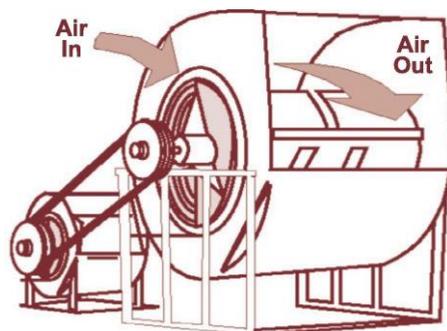
Fans fall into two general categories: centrifugal flow and axial flow.

In centrifugal flow, airflow changes direction twice - once when entering and second when leaving (forward curved, backward curved or inclined, radial) (see Figure 4.1).

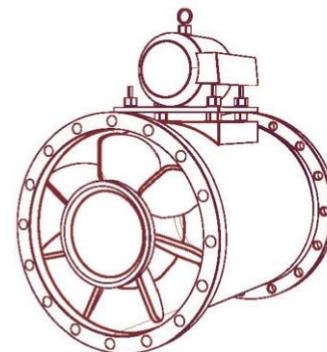
In axial flow, air enters and leaves the fan with no change in direction (propeller, tubeaxial, vaneaxial) (see Figure 5.2).

*Table 4.2 Fan Efficiencies*

Type of fan	Peak Efficiency Range
<b>Centrifugal Fan</b>	
Airfoil, backward curved/inclined	79-83
Modified radial	72-79
Radial	69-75
Pressure blower	58-68
Forward curved	60-65
<b>Axial fan</b>	
Vanaxial	78-85
Tubeaxial	67-72
Propeller	45-50



*Figure 4.1 Centrifugal Fan*



*Figure 4.2 Axial Fan*

### Centrifugal Fan: Types

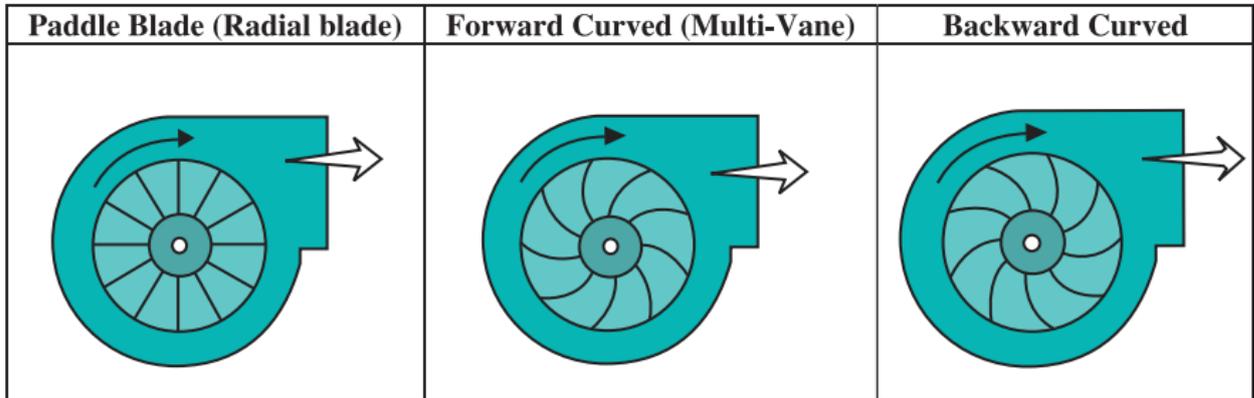
The major types of centrifugal fan are: *radial, forward curved and backward curved* (see Figure 5.3).

**Radial fans** are industrial workhorses because of their high static pressures (upto 1400 mm WC) and ability to handle heavily contaminated airstreams. Because of their simple design, radial fans are well suited for high temperatures and medium blade tip speeds.

**Forward-curved fans** are used in clean environments and operate at lower temperatures. They are well suited for low tip speed and high-airflow work - they are best suited for moving large volumes of air against relatively low pressures.

**Backward-inclined fans** are more efficient than forward-curved fans. Backward-inclined fans reach their peak power consumption and then power demand drops off well within their

useable airflow range. Backward-inclined fans are known as "non-overloading" because changes in static pressure do not overload the motor.



*Figure 4.3 Types of Centrifugal Fans*

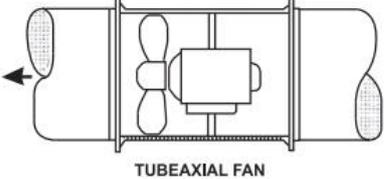
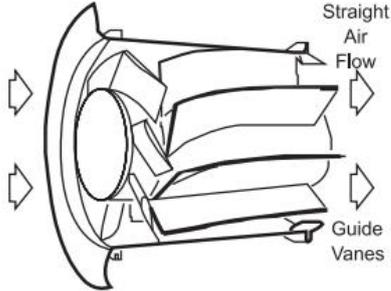
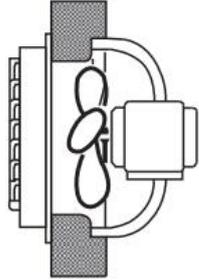
### **Axial Flow Fan: Types**

The major types of axial flow fans are: tube axial, vane axial and propeller (see Figure 5.4.)

Tubeaxial fans have a wheel inside a cylindrical housing, with close clearance between blade and housing to improve airflow efficiency. The wheel turn faster than propeller fans, enabling operation under high-pressures 250 — 400 mm WC. The efficiency is up to 65%.

Vaneaxial fans are similar to tubeaxials, but with addition of guide vanes that improve efficiency by directing and straightening the flow. As a result, they have a higher static pressure with less dependence on the duct static pressure. Such fans are used generally for pressures upto 500 mmWC. Vaneaxials are typically the most energy-efficient fans available and should be used whenever possible.

Propeller fans usually run at low speeds and moderate temperatures. They experience a large change in airflow with small changes in static pressure. They handle large volumes of air at low pressure or free delivery. Propeller fans are often used indoors as exhaust fans. Outdoor applications include air-cooled condensers and cooling towers. Efficiency is low — approximately 50% or less.

Tube Axial	Vane Axial	Propeller
		
 <p style="text-align: center;">TUBEAXIAL FAN</p>	 <p style="text-align: center;">VANEAXIAL FAN</p>	 <p style="text-align: center;">PROPELLER FAN Belt Drive or Direct Connection</p>

**Figure 4.4 Types of Radial Fans**

The different types of fans, their characteristics and typical applications are given in Table

### 4.3 Common Blower Types

Blowers can achieve much higher pressures than fans, as high as 1.20 kg/cm<sup>2</sup>. They are also used to produce negative pressures for industrial vacuum systems. Major types are: centrifugal blower and positive-displacement blower.

Centrifugal blowers look more like centrifugal pumps than fans. The impeller is typically driven and rotates as fast as 15,000 rpm. In multi-stage blowers, air is accelerated as it passes through each impeller. In single-stage blower, air does not take many turns, and hence it is more efficient.

Centrifugal blowers typically operate against pressures of 0.35 to 0.70 kg/cm<sup>2</sup>, but can achieve higher pressures. One characteristic is that airflow tends to drop drastically as system pressure increases, which can be a disadvantage in material conveying systems that depend on a steady air volume. Because of this, they are most often used in applications that are not prone to clogging.

**Table 4.3 Types of Fans, Characteristics, and Typical Applications**

Centrifugal Fans			Axial-flow Fans		
Type	Characteristics	Typical Applications	Type	Characteristics	Typical Applications
Radial	High pressure, medium flow, efficiency close to tube-axial fans, power increases continuously	Various industrial applications, suitable for dust laden, moist air/ gases	Propeller	Low pressure, high flow, low efficiency, peak efficiency close to point of free air delivery (zero static pressure)	Air-circulation, ventilation, exhaust
Forward-curved blades	Medium pressure, high flow, dip in pressure curve, efficiency higher than radial fans, power rises continuously	Low pressure HVAC, packaged units, suitable for clean and dust laden air / gases	Tube-axial	Medium pressure, high flow, higher efficiency than propeller type, dip in pressure-flow curve before peak pressure point.	HVAC, drying ovens, exhaust systems
Backward curved blades	High pressure, high flow, high efficiency, power reduces as flow increases beyond point of highest efficiency	HVAC, various industrial applications, forced draft fans, etc.	Vane-axial	High pressure, medium flow, dip in pressure-flow curve, use of guide vanes improves efficiency	High pressure applications including HVAC systems, exhausts
Airfoil type	Same as backward curved type, highest efficiency	Same as backward curved, but for clean air applications			

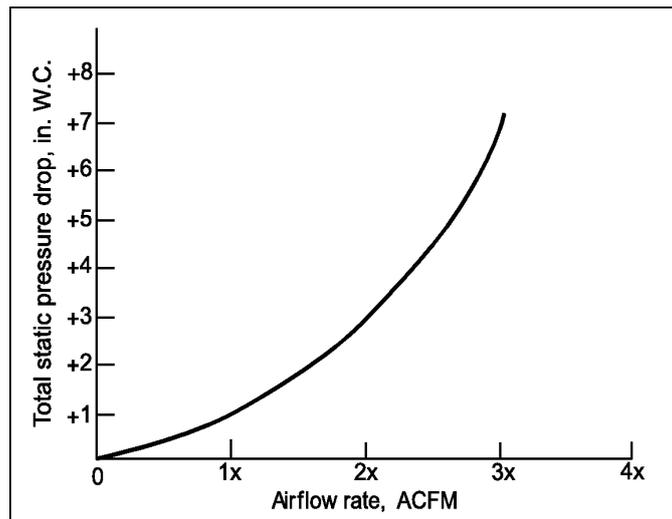
Positive-displacement blowers have rotors, which "trap" air and push it through housing. Positive-displacement blowers provide a constant volume of air even if the system pressure varies. They are especially suitable for applications prone to clogging, since they can produce enough pressure - typically up to 1.25 kg/cm<sup>2</sup> - to blow clogged materials free. They turn much slower than centrifugal blowers (e.g. 3,600 rpm), and are often belt driven to facilitate speed changes.

#### 4.4 Fan Performance Evaluation and Efficient System Operation

##### System Characteristics

The term "system resistance" is used when referring to the static pressure. The system resistance is the sum of static pressure losses in the system. The system resistance is a function of the configuration of ducts, pickups, elbows and the pressure drops across equipment-for example bag filter or cyclone.

The system resistance varies with the square of the volume of air flowing through the system. For a given volume of air, the fan in a system with narrow ducts and multiple short radius elbows is going to have to work harder to overcome a greater system resistance than it would in a system with larger ducts and a minimum number of long radius turns. Long narrow ducts with many bends and twists will require more energy to pull the air through them. Consequently, for a given fan speed, the fan will be able to pull less air through this system than through a short system with no elbows. Thus, the system resistance increases substantially as the volume of air flowing through the system increases; square of air flow.



*Figure 4.5 System Characteristics*

Conversely, resistance decreases as flow decreases. To determine what volume the fan will produce, it is therefore necessary to know the system resistance characteristics. In existing systems, the system resistance can be measured. In systems that have been designed, but not built, the system resistance must be calculated. Typically a system resistance curve (see Figure 5.5) is generated with for various flow rates on the x-axis and the associated resistance on the y-axis.

### **Fan Characteristics**

Fan characteristics can be represented in form of fan curve(s). The fan curve is a performance curve for the particular fan under a specific set of conditions. The fan curve is a graphical representation of a number of inter-related parameters. Typically a curve will be developed for a given set of conditions usually including: fan volume, system static pressure, fan speed, and brake horsepower required to drive the fan under the stated conditions. Some fan curves will also include an efficiency curve so that a system designer will know where on that curve the fan will be operating under the chosen conditions (see Figure 5.6). In the many curves shown in the Figure, the curve static pressure (SP) vs. flow is especially important.

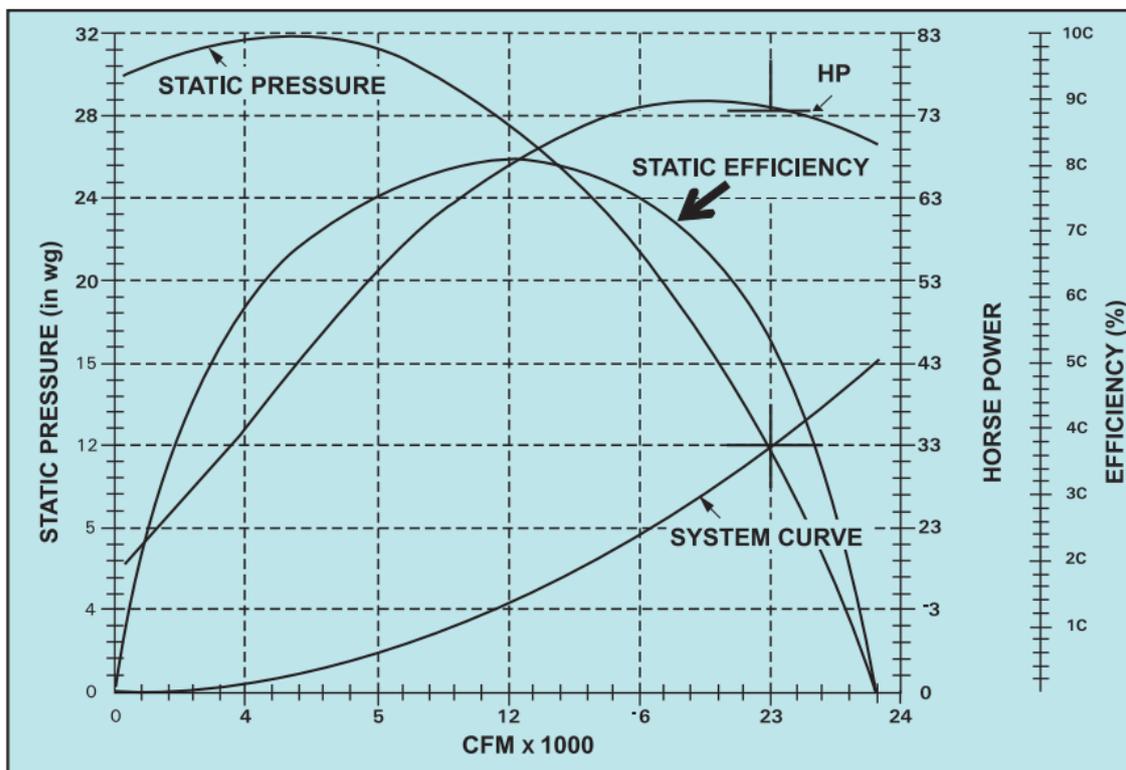
The intersection of the system curve and the static pressure curve defines the operating point. When the system resistance changes, the operating point is also changes. Once the operating point is fixed, the power required could be found by following a vertical line that passes through the operating point to an intersection with the power (**BHP**) curve. A horizontal line drawn through the intersection with the power curve will lead to the

required power on the right vertical axis. In the depicted curves, the fan efficiency curve is also presented.

### System Characteristics and Fan Curves

In any fan system, the resistance to air flow (pressure) increases when the flow of air is increased. As mentioned before, it varies as the square of the flow. The pressure required by a system over a range of flows can be determined and a "system performance curve" can be developed (shown as SC) (see Figure 4.7).

This system curve can then be plotted on the fan curve to show the fan's actual operating point at "A" where the two curves ( $N_1$  and  $SC_1$ ) intersect. This operating point is at air flow  $Q_1$  delivered against pressure  $P_1$ .



*Figure 4.6 Fan Characteristics Curve by Manufacturer*

A fan operates along a performance given by the manufacturer for a particular fan speed. (The fan performance chart shows performance curves for a series of fan speeds.) At fan speed  $N_1$ , the fan will operate along the  $N_1$  performance curve as shown in Figure 5.7. The fan's actual operating point on this curve will depend on the system resistance; fan's operating point at "A" is flow ( $Q_1$ ) against pressure ( $P_1$ ).

### Two methods can be used to reduce air flow from $Q_1$ to $Q_2$ :

First method is to restrict the air flow by partially closing a damper in the system. This action causes a new system performance curve ( $SC_2$ ) where the required pressure is greater for any given air flow. The fan will now operate at "B" to provide the reduced air flow  $Q_2$  against higher pressure  $P_2$ .

Second method to reduce air flow is by reducing the speed from  $N_1$  to  $N_2$ , keeping the damper fully open. The fan would operate at "C" to provide the same  $Q_2$  air flow, but at a lower pressure  $P_3$ .

Thus, reducing the fan speed is a much more efficient method to decrease airflow since less power is required and less energy is consumed.

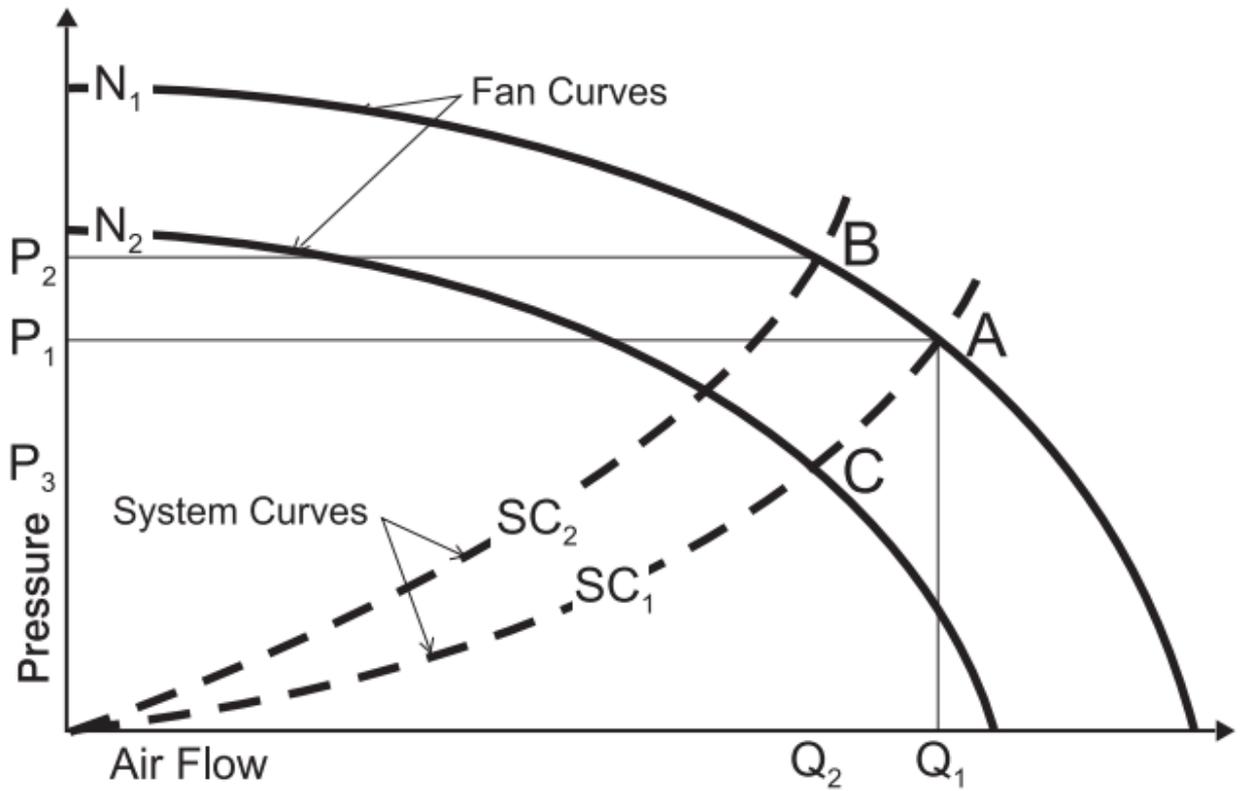


Figure 4.7 System Curve

### Fan Laws

The fans operate under a predictable set of laws concerning speed, power and pressure. A change in speed (RPM) of any fan will predictably change the pressure rise and power necessary to operate it at the new RPM.

<b>Flow ∞ Speed</b>	<b>Pressure ∞ (Speed)<sup>2</sup></b>	<b>Power ∞ (Speed)<sup>3</sup></b>
$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$	$\frac{SP_1}{SP_2} = \left( \frac{N_1}{N_2} \right)^2$	$\frac{kW_1}{kW_2} = \left( \frac{N_1}{N_2} \right)^3$
<p><i>Varying the RPM by 10% decreases or increases air delivery by 10%.</i></p>	<p><i>Reducing the RPM by 10% decreases the static pressure by 19% and an increase in RPM by 10% increases the static pressure by 21%.</i></p>	<p><i>Reducing the RPM by 10% decreases the power requirement by 27% and an increase in RPM by 10% increases the power requirement by 33%.</i></p>

Where Q — Flow, SP — Static Pressure, kW — Power and N — Speed (RPM)

#### 4.5 Fan Design and Selection Criteria

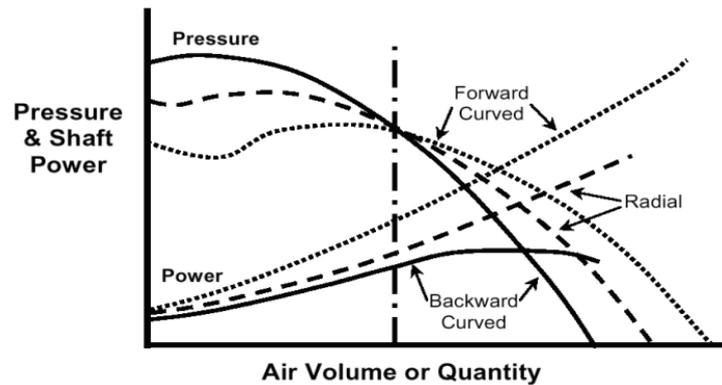
Precise determination of air-flow and required outlet pressure are most important in proper selection of fan type and size. The air-flow required depends on the process requirements; normally determined from heat transfer rates, or combustion air or flue gas quantity to be handled. System pressure requirement is usually more difficult to compute or predict. Detailed analysis should be carried out to determine pressure drop across the length, bends, contractions and expansions in the ducting system, pressure drop across filters, drop in branch lines, etc. These pressure drops should be added to any fixed pressure required by the process (in the case of ventilation fans there is no fixed pressure requirement). Frequently, a very conservative approach is adopted allocating large safety margins, resulting in over-sized fans which operate at flow rates much below their design values and, consequently, at very poor efficiency.

Once the system flow and pressure requirements are determined, the fan and impeller type are then selected. For best results, values should be obtained from the manufacturer for specific fans and impellers.

The choice of fan type for a given application depends on the magnitudes of required flow and static pressure. For a given fan type, the selection of the appropriate impeller depends additionally on rotational speed. Speed of operation varies with the application. High speed small units are generally more economical because of their higher hydraulic efficiency and relatively low cost. However, at low pressure ratios, large, low-speed units are preferable.

## Fan Performance and Efficiency

Typical static pressures and power requirements for different types of fans are given in the Figure 5.8.



**Figure 4.8 Fan Static Pressure and Power Requirements for Different Fans**

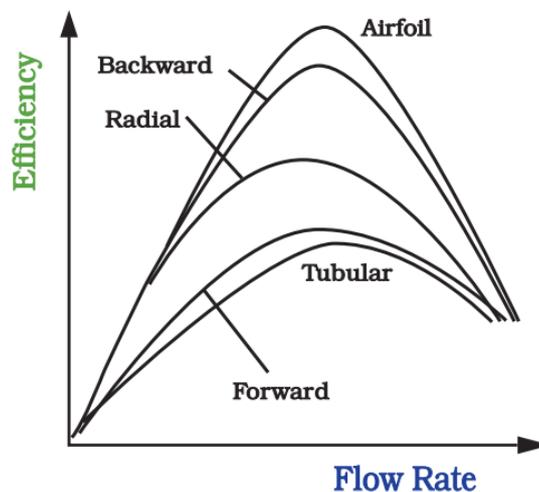
Fan performance characteristics and efficiency differ based on fan and impeller type (See Figure4.9).

In the case of centrifugal fans, the hub- to-tip ratios (ratio of inner-to-outer impeller diameter) the tip angles (angle at which forward or backward curved blades are curved at the blade tip - at the base the blades are always oriented in the direction of flow), and the blade width determine the pressure developed by the fan.

Forward curved fans have large hub-to-tip ratios compared to backward curved fans and produce lower pressure.

Radial fans can be made with different heel-to-tip ratios to produce different pressures.

At both design and off-design points, backward-curved fans provide the most stable operation. Also, the power required by most backward —curved fans will decrease at flow higher than design values. A similar effect can be obtained by using inlet guide vanes instead of replacing the impeller with different tip angles. Radial fans are simple in construction and are preferable for high-pressure applications.



**Figure 4.9 Fan Performance Characteristics based on Fans/ Impellers**

Forward curved fans, however, are less efficient than backward curved fans and power rises continuously with flow. Thus, they are generally more expensive to operate despite their lower first cost.

Among centrifugal fan designs, aerofoil designs provide the highest efficiency (upto 10% higher than backward curved blades), but their use is limited to clean, dust-free air. Axial-flow fans produce lower pressure than centrifugal fans, and exhibit a dip in pressure before reaching the peak pressure point. Axial-flow fans equipped with adjustable / variable pitch blades are also available to meet varying flow requirements.

Propeller-type fans are capable of high-flow rates at low pressures. Tube-axial fans have medium pressure, high flow capability and are not equipped with guide vanes.

Vane-axial fans are equipped with inlet or outlet guide vanes, and are characterized by high pressure, medium flow-rate capabilities.

Performance is also dependent on the fan enclosure and duct design. Spiral housing designs with inducers, diffusers are more efficient as compared to square housings. Density of inlet air is another important consideration, since it affects both volume flow-rate and capacity of the fan to develop pressure. Inlet and outlet conditions (whirl and turbulence created by grills, dampers, etc.) can significantly alter fan performance curves from that provided by the manufacturer (which are developed under controlled conditions). Bends and elbows in the inlet or outlet ducting can change the velocity of air, thereby changing fan characteristics (the pressure drop in these elements is attributed to the system resistance). All these factors, termed as System Effect Factors, should, therefore, be carefully evaluated during fan selection since they would modify the fan performance curve.

Centrifugal fans are suitable for low to moderate flow at high pressures, while axial-flow fans are suitable for low to high flows at low pressures. Centrifugal fans are generally more expensive than axial fans. Fan prices vary widely based on the impeller type and the mounting (direct-or-belt-coupled, wall-or-duct-mounted). Among centrifugal fans, aerofoil and backward-curved blade designs tend to be somewhat more expensive than forward-curved blade designs and will typically provide more favourable economics on a lifecycle basis. Reliable cost comparisons are difficult since costs vary with a number of application-specific factors. A careful technical and economic evaluation of available options is important in identifying the fan that will minimize lifecycle costs in any specific application.

### **Safety margin**

The choice of safety margin also affects the efficient operation of the fan. In all cases where the fan requirement is linked to the process/other equipment, the safety margin is to be decided, based on the discussions with the process equipment supplier. In general, the safety margin can be 5 % over the maximum requirement on flow rate.

In the case of boilers, the induced draft (ID) fan can be designed with a safety margin of 20 % on volume and 30 % on head. The forced draft (FD) fans and primary air (PA) fans do not require any safety margins. However, safety margins of 10 % on volume and 20 % on pressure are maintained for FD and PA fans.

### Some pointers on fan specification

The right specification of the parameters of the fan at the initial stage, is pre-requisite for choosing the appropriate and energy efficient fan.

The user should specify following information to fan manufacturer to enable right selection: Design operating point of the fan — volume and pressure

Normal operating point - volume and pressure

Maximum continuous rating

Low load operation - This is particularly essential for units, which in the initial few years may operate at lower capacities, with plans for upgradation at a later stage. The initial low load and the later higher load operational requirements need to be specified clearly, so that the manufacturer can supply a fan which can meet both the requirements, with different sizes of impeller.

Ambient temperature — The ambient temperatures, both the minimum and maximum, are to be specified to the supplier. This affects the choice of the material of construction of the impeller

The maximum temperature of the gas at the fan during upset conditions should be specified to the supplier This will enable choice of the right material of the required creep strength.

Density of gas at different temperatures at fan outlet

Composition of the gas — This is very important for choosing the material of construction of the fan.

Dust concentration and nature of dust — The dust concentration and the nature of dust (e.g. bagasse — soft dust, coal — hard dust) should be clearly specified.

The proposed control mechanisms that are going to be used for controlling the fan.

The operating frequency varies from plant-to-plant, depending on the source of power supply. Since this has a direct effect on the speed of the fan, the frequency prevailing or being maintained in the plant also needs to be specified to the supplier.

Altitude of the plant

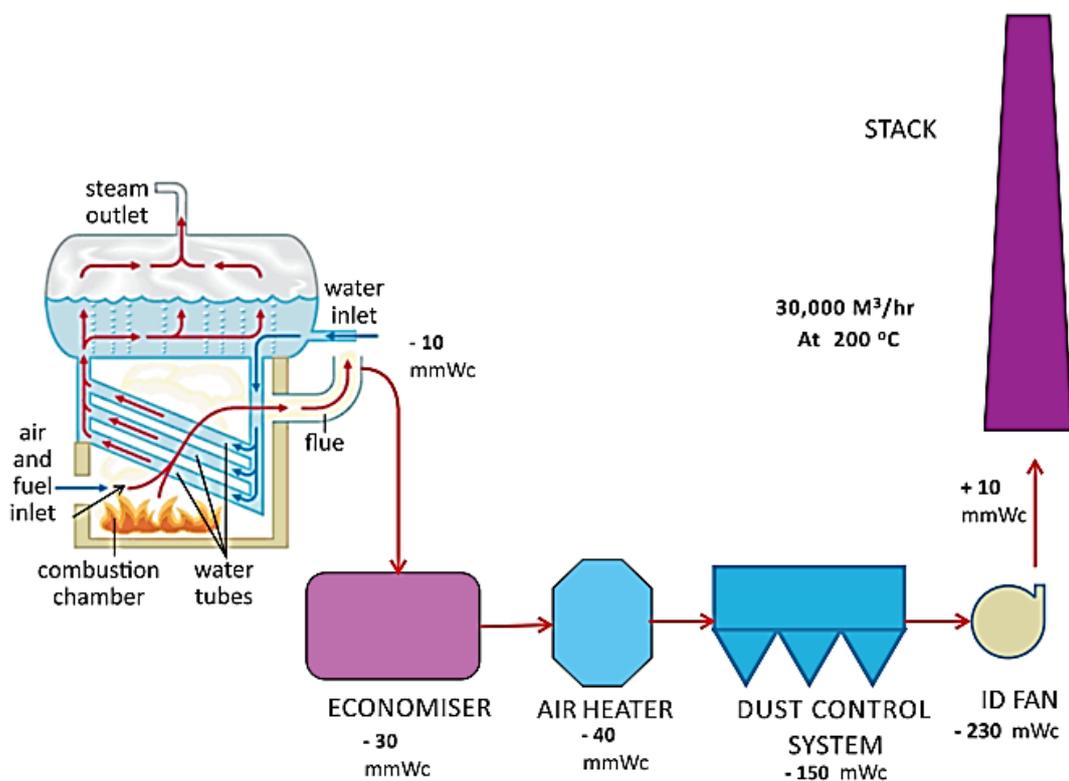
The choice of speed of the fan can be best left to fan manufacturer. This will enable him to design the fan of the highest possible efficiency. However, if the plant has some preferred speeds on account of any operational need, the same can be communicated to the fan supplier.

### System Resistance and Pressure Drop

The system resistance has a major role in determining the performance and efficiency of a fan. The system resistance also changes depending on the process. For example, the formation of the coatings / erosion of the lining in the ducts, changes the system resistance marginally. In some cases, the change of equipment (e.g. Replacement of Multi-cyclones with ESP / Installation of low pressure drop cyclones in cement industry) duct modifications, drastically shift the operating point, resulting in lower efficiency. In such cases, to maintain the efficiency as before, the fan has to be changed.

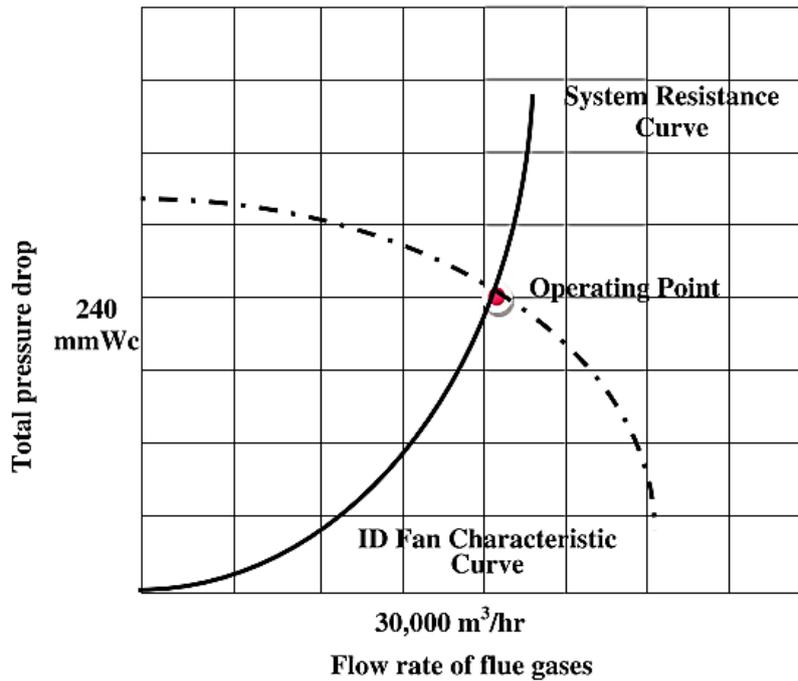
Hence, the system resistance has to be periodically checked, more so when modifications are introduced and action taken accordingly, for efficient operation of the fan.

System resistance in the application of centrifugal fan is the resistance offered by all the process equipment and duct line connected to the inlet of the fan as well as at the outlet of the fan, usually stacks. In the Figure 4.10 draft required to overcome the resistance offered by the boiler to draw the flue gases -10 mmWc (- ve sign indicates the suction), the draft required to draw the flue gases through the economiser, Air heater and dust control system are added to estimate the total resistance at the inlet of the ID fan (i.e.  $-10 - 30 - 40 - 150 = -230$  mmWc). However the ID fan requires a minimum of 10 mmWc pressure to push the gases to the bottom of the stack from where the gases will be taken care by natural draft into atmosphere. Hence the overall resistance (pressure drop) to be build up by the ID fan is the difference between the outlet pressure and the inlet pressure [i.e.  $10 - (-230) = 240$  mmWc].



**Figure 4.10 Pressure Drop across Various Equipment**

System resistance is a function of gas density and the velocity of the gas. In the Figure 5.10 initially when there is no flue gas generation, there is no gas velocity in the process equipment and the ducts and hence the resistance (pressure drop) is zero (see Figure 5.11). After the boiler firing is started, there is continuous increase in the generation of flue gases and accordingly there is continuous increase in the velocity of the gases and continuous increase in draft. When the boiler is fired at rated capacity, the generation of flue gases are maximum resulting in corresponding pressure drop, which is the "operating point" ( 30,000 m³/hr, 240 mmWc) of the boiler ( Figure 5.11). Accordingly the ID fan with a characteristic (Performance) curve passing through the "operating point" of the system at the highest possible fan efficiency has to be selected for better utilisation of energy.



*Figure 4.11 Fan System Curve*

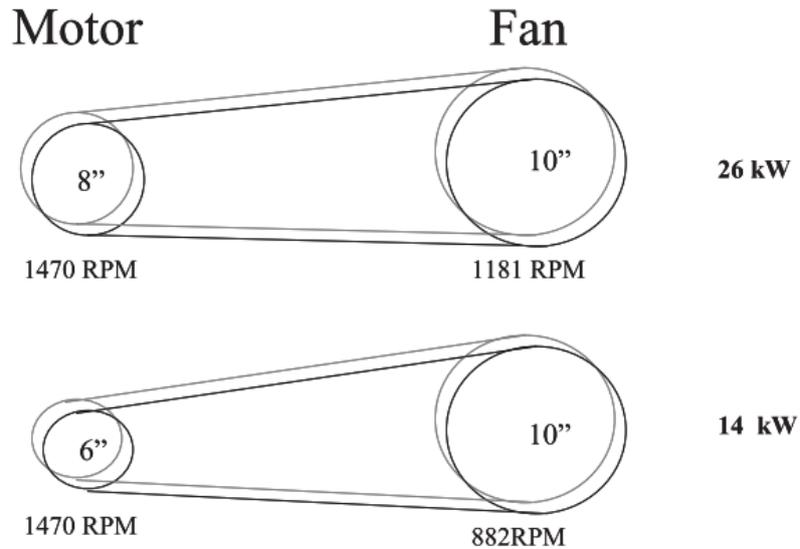
#### 4.6 Flow Control Strategies

Typically, once a fan system is designed and installed, the fan operates at a constant speed. There may be occasions when a speed change is desirable, i.e., when adding a new run of duct that requires an increase in air flow (volume) through the fan. There are also instances when the fan is oversized and flow reductions are required.

Various ways to achieve change in flow are: pulley change, damper control, inlet guide vane control, variable speed drive and series and parallel operation of fans.

#### Pulley Change

When a fan volume change is required on a permanent basis, and the existing fan can handle the change in capacity, the volume change can be achieved with a speed change. The simplest way to change the speed is with a pulley change. For this, the fan must be driven by a motor through a v-belt system. The fan speed can be increased or decreased with a change in the drive pulley or the driven pulley or in some cases, both pulleys. As shown in the Figure 4.12, a higher sized fan operating with damper control was downsized by reducing the motor (drive) pulley size from 8" to 6". The power reduction was 12 kW.



**Figure 4.12 Pulley Change**

### Damper Controls

Some fans are designed with damper controls (see Figure 4.13). Dampers can be located at inlet or outlet. Dampers provide a means of changing air volume by adding or removing system resistance. This resistance forces the fan to move up or down along its characteristic curve, generating more or less air without changing fan speed. However, dampers provide a limited amount of adjustment, and they are not particularly energy efficient.



**Figure 4.13 Damper change**

## Inlet Guide Vanes

Inlet guide vanes are another mechanism that can be used to meet variable air demand (see Figure 4.14). Guide vanes are curved sections that lay against the inlet of the fan when they are open. When they are closed, they extend out into the air stream. As they are closed, guide vanes pre-swirl the air entering the fan housing. This changes the angle at which the air is presented to the fan blades, which, in turn, changes the characteristics of the fan curve. Guide vanes are energy efficient for modest flow reductions — from 100 percent flow to about 80 percent. Below 80 percent flow, energy efficiency drops sharply.



*Figure 4.14 Inlet Guide Vanes*

Axial-flow fans can be equipped with variable pitch blades, which can be hydraulically or pneumatically controlled to change blade pitch, while the fan is at stationary. Variable-pitch blades modify the fan characteristics substantially and thereby provide dramatically higher energy efficiency than the other options discussed thus far.

## Variable Speed Drives

Although, variable speed drives are expensive, they provide almost infinite variability in speed control. Variable speed operation involves reducing the speed of the fan to meet reduced flow requirements. Fan performance can be predicted at different speeds using the fan laws. Since power input to the fan changes as the cube of the flow, this will usually be the most efficient form of capacity control. However, variable speed control may not be economical for systems, which have infrequent flow variations. When considering variable speed drive, the efficiency of the control system (fluid coupling, eddy-current, VFD, etc.) should be accounted for, in the analysis of power consumption.

## Series and Parallel Operation

Parallel operation of fans is another useful form of capacity control. Fans in parallel can be additionally equipped with dampers, variable inlet vanes, variable-pitch blades, or speed controls to provide a high degree of flexibility and reliability.

Combining fans in series or parallel can achieve the desired airflow without greatly increasing the system package size or fan diameter. Parallel operation is defined as having two or more fans blowing together side by side.

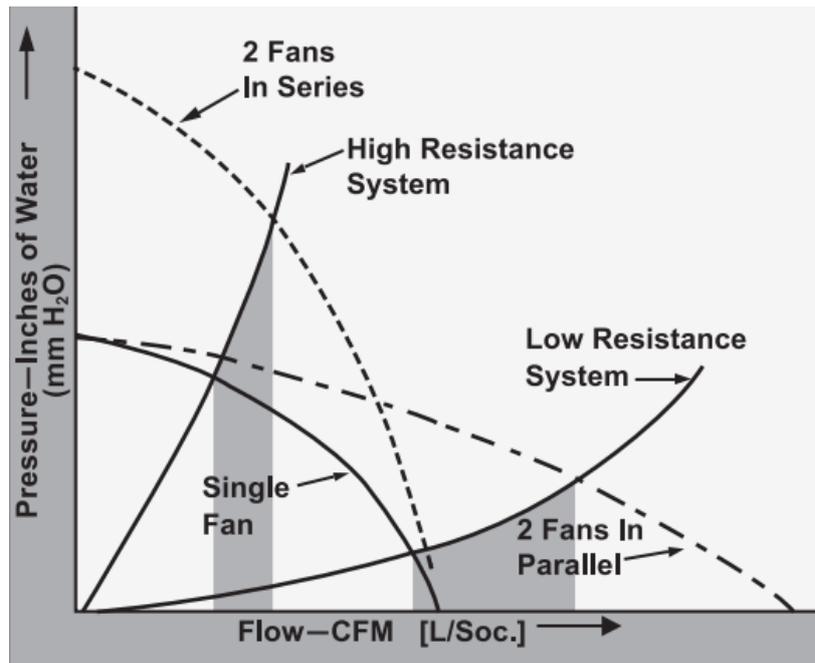


Figure 4.15: Series and Parallel Operation

The performance of two fans in parallel will result in doubling the volume flow, but only at free delivery. As Figure 4.15 shows, when a system curve is overlaid on the parallel performance curves, the higher the system resistance, the less increase in flow results with parallel fan operation. Thus, this type of application should only be used when the fans can operate in a low resistance almost in a free delivery condition.

Series operation can be defined as using multiple fans in a push-pull arrangement. By staging two fans in series, the static pressure capability at a given airflow can be increased, but again, not to double at every flow point, as the above Figure displays. In series operation, the best results are achieved in systems with high resistances.

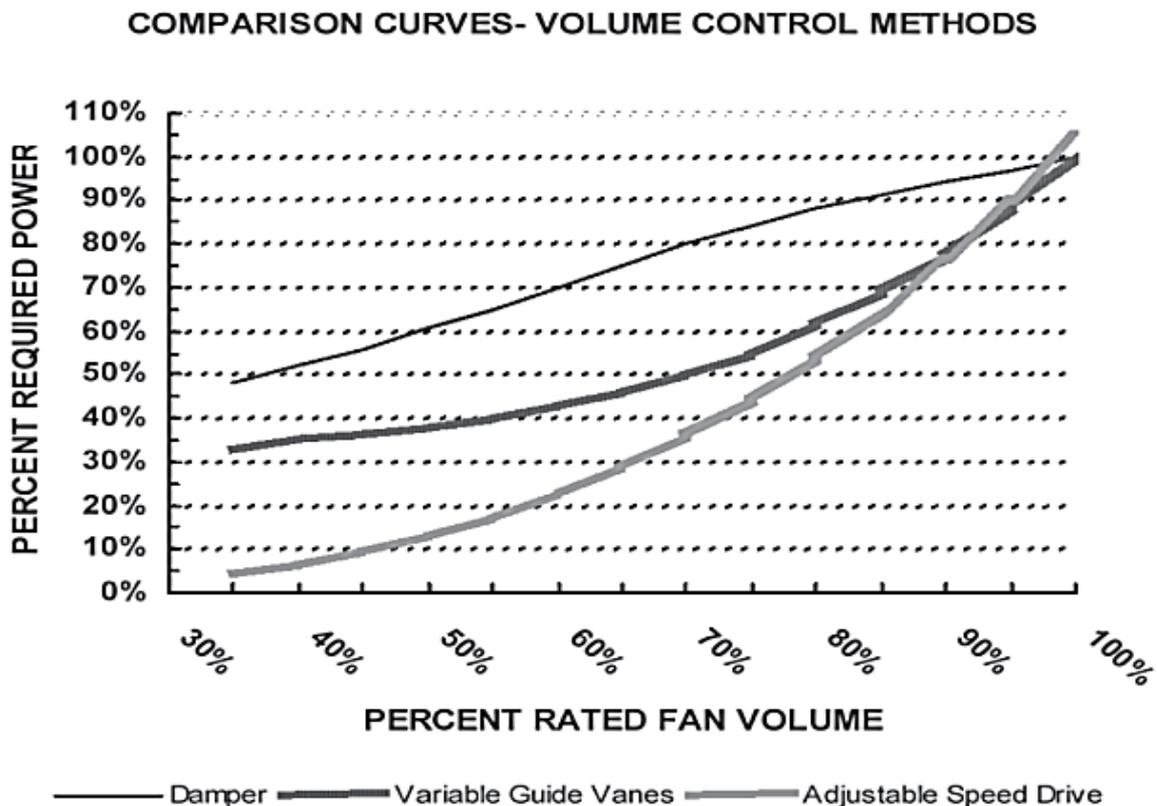
In both series and parallel operation, particularly with multiple fans certain areas of the combined performance curve will be unstable and should be avoided. This instability is unpredictable and is a function of the fan and motor construction and the operating point.

### Factors to be considered in the selection of flow control methods

Comparison on of various volume control methods with respect to power consumption (%) required power is shown in Figure 4.16.

All methods of capacity control mentioned above have turn-down ratios (ratio of maximum—to minimum flow rate) determined by the amount of leakage (slip) through the control elements. For example, even with dampers fully closed, the flow may not be zero due to leakage through the damper. In the case of variable-speed drives the turn-down ratio is limited by the control system. In many cases, the minimum possible flow will be determined by the characteristics of the fan itself. Stable operation of a fan requires that it operate in a region where the system curve has a positive slope and the fan curve has a negative slope.

The range of operation and the time duration at each operating point also serves as a guide to selection of the most suitable capacity control system. Outlet damper control due to its simplicity, ease of operation, and low investment cost, is the most prevalent form of capacity control. However, it is the most inefficient of all methods and is best suited for situations where only small, infrequent changes are required (for example, minor process variations due to seasonal changes. The economic advantage of one method over the other is determined by the time duration over which the fan operates at different operating points. The frequency of flow change is another important determinant. For systems requiring frequent flow control, damper adjustment may not be convenient. Indeed, in many plants, dampers are not easily accessible and are left at some intermediate position to avoid frequent control.



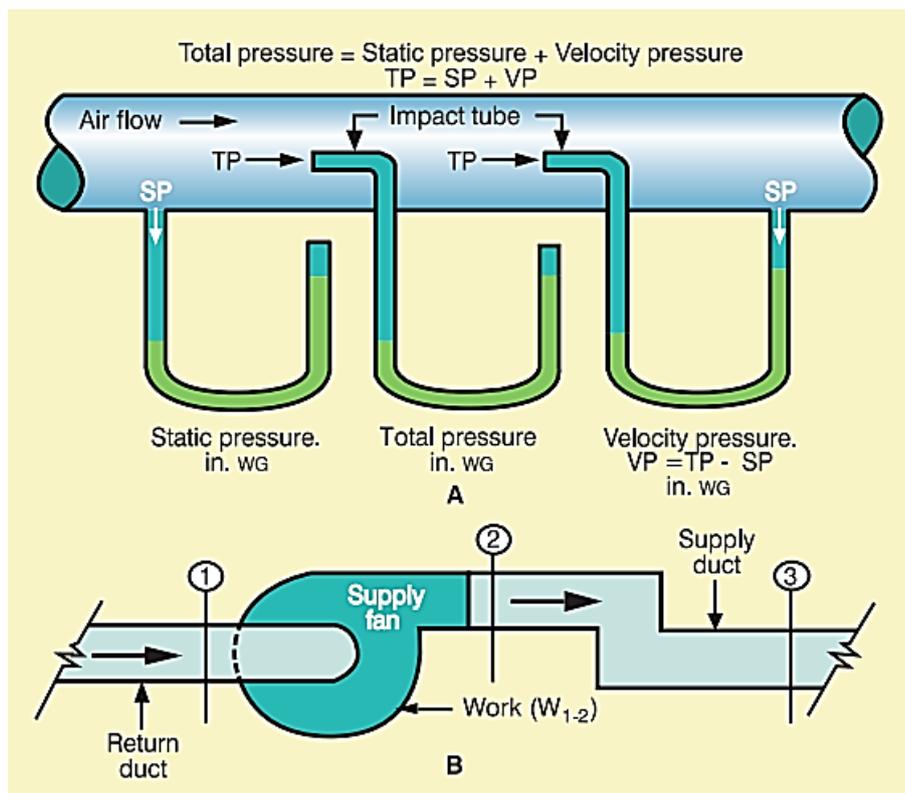
*Figure- 4.16: Comparison: Various Volume Control Methods*

#### 4.7 Fan Performance Assessment

The fans are tested for field performance by measurement of flow, head and temperature on the fan side and electrical motor kW input on the motor side.

### Air flow measurement *Static pressure*

Static pressure is the potential energy put into the system by the fan. It is given up to friction in the ducts and at the duct inlet as it is converted to velocity pressure. At the inlet to the duct, the static pressure produces an area of low pressure (see Figure 4.17).



**Figure 4.17: Static, Total and Velocity Pressure**

### Velocity pressure

Velocity pressure is the pressure along the line of the flow that results from the air flowing through the duct. The velocity pressure is used to calculate air velocity.

### Total pressure

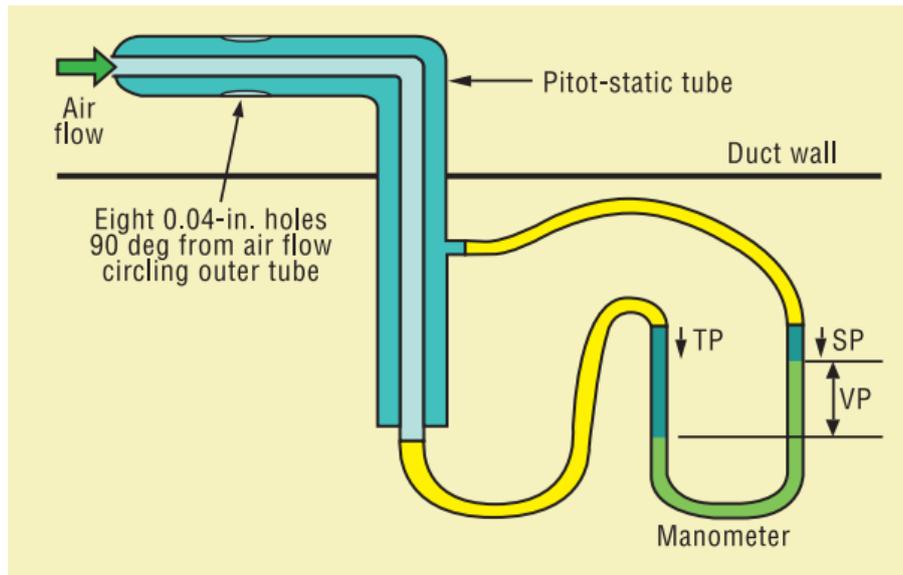
Total pressure is the sum of the static and velocity pressure. Velocity pressure and static pressure can change as the air flows through different size ducts accelerating and decelerating the velocity. The total pressure stays constant, changing only with friction losses. The illustration that follows shows how the total pressure changes in a system.

The fan flow is measured using pitot tube manometer combination, or a flow sensor (differential pressure instrument) or an accurate anemometer. Care needs to be taken regarding number of traverse points, straight length section (to avoid turbulent flow regimes of

measurement) upstream and downstream of measurement location. The measurements can be on the suction or discharge side of the fan and preferably both where feasible.

### Measurement by Pitot tube

The Figure 5.18 shows how velocity pressure is measured using a pitot tube and a manometer. Total pressure is measured using the inner tube of pitot tube and static pressure is measured using the outer tube of pitot tube. When the inner and outer tube ends are connected to a manometer, we get the velocity pressure. For measuring low velocities, it is preferable to use an inclined tube manometer instead of U tube manometer.

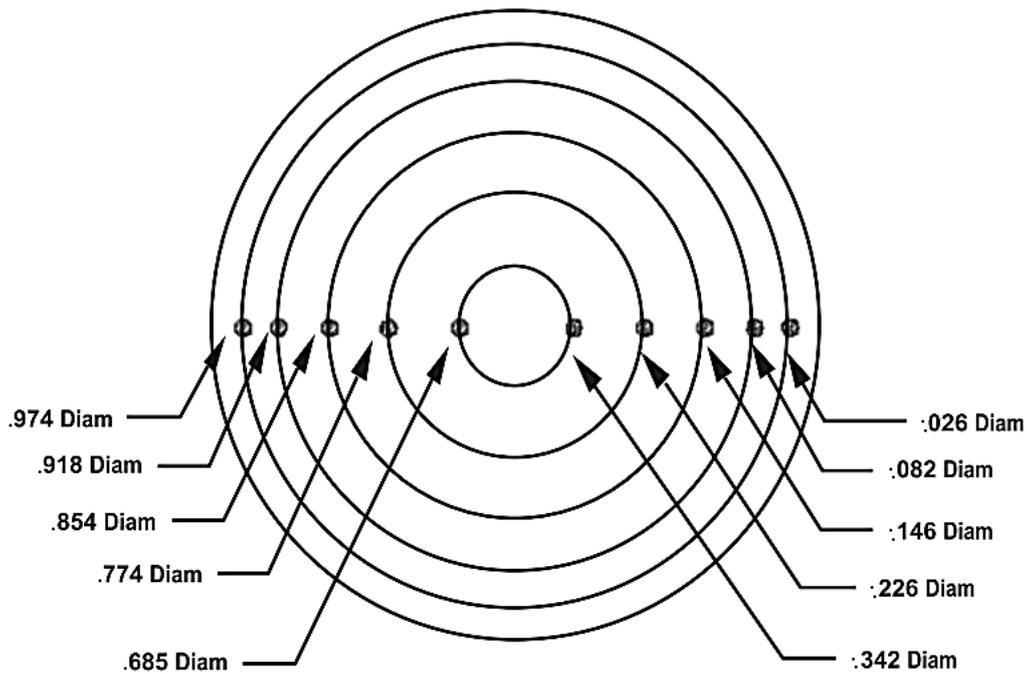


*Figure 4.18 Velocity Measurement Using Pitot Tube*

### Measurements and Calculations

#### Velocity pressure/velocity calculation

When measuring velocity pressure the duct diameter (or the circumference from which to calculate the diameter) should be measured as well. This will allow us to calculate the velocity and the volume of air in the duct. In most cases, velocity must be measured at several places in the same system.



**Figure 4.19** *Traverse Points for Circular Duct*

The velocity pressure varies across the duct. Friction slows the air near the duct walls, so the velocity is greater in the center of the duct. The velocity is affected by changes in the ducting configuration such as bends and curves. The best place to take measurements is in a section of duct that is straight for at least 3-5 diameters after any elbows, branch entries or duct size changes.

To determine the average velocity, it is necessary to take a number of velocity pressure readings across the cross-section of the duct. The velocity should be calculated for each velocity pressure reading, and the average of the velocities should be used. Do not average the velocity pressure; average the velocities. For round ducts over 6 inches diameter, the following locations will give areas of equal concentric area (see Figure 4.19).

For best results, one set of readings should be taken in one direction and another set at a 90° angle to the first. For square ducts, the readings can be taken in 16 equally spaced areas. If it is impossible to traverse the duct, an approximate average velocity can be calculated by measuring the velocity pressure in the center of the duct and calculating the velocity. This value is reduced to an approximate average by multiplying by 0.9.

**Calculation of Velocity:** After taking velocity pressures readings, at various traverse points, the velocity corresponding to each point is calculated using the following expression.

$$\text{Velocity (m/s)} = C_p \times \sqrt{\frac{2 \times 9.81 \times \Delta p}{\gamma}}$$

Where  $C_p$  = The pitot tube coefficient (Take manufacturer's value or assume 0.85)  
 $\Delta P$  = The average velocity pressure measured using pitot tube and inclined manometer by taking number of points over the entire cross-section of the duct, mm Water Column

$\gamma$  = Gas density at flow conditions, kg/m<sup>3</sup>

#### EXAMPLE 4.1

Air flow measurements using the pitot tube, in the primary air fan of a coal fired boiler gave the following data, calculate the velocity of air.

Air temperature = 38°C

Velocity pressure = 47 mmWC

Pitot tube constant,  $C_p = 0.9$

Air density at 38°C = 1.135 kg /m<sup>3</sup>

Find out the velocity of air in m/sec.

$$\begin{aligned}\text{Velocity (m/s)} &= C_p \times \sqrt{\frac{2 \times 9.81 \times \Delta p}{\gamma}} \\ &= 0.9 \times \sqrt{\frac{2 \times 9.81 \times 47}{1.135}} \\ &= 25.6 \text{ m/s}\end{aligned}$$

#### Calculation of gas density

To calculate the velocity and volume from the velocity pressure measurements, it is necessary to know the density of gas. The density is dependent on altitude, temperature, molecular weight and pressure.

$$\text{Density } (\gamma), \text{ kg / m}^3 = \frac{P \times M}{R \times T}$$

P – Absolute gas pressure, mmWC

M – Molecular weight of the gas, kg/kg mole (in the case of air M = 28.92 kg/kg mole)

T – Gas temperature, K

R – Gas constant, 847.84 mmWC m<sup>3</sup>/kg mole K

#### Calculation of molecular weight for flue gas consisting of CO<sub>2</sub>, CO, O<sub>2</sub>, N<sub>2</sub> (M) (dry basis), kg/kg mole

$$= \{ \% \text{ CO}_2 \times M_{\text{CO}_2} + \% \text{ O}_2 \times M_{\text{O}_2} + \% \text{ CO} \times M_{\text{CO}} + \% \text{ N}_2 \times M_{\text{N}_2} \} / 100$$

#### Volume calculation

The volume in a duct can be calculated for the velocity using the equation:

$$\text{Volumetric flow } (Q), \text{ m}^3 / \text{sec} = \text{Velocity}, V \text{ (m/sec)} \times \text{Area} (\text{m}^2)$$

#### Fan efficiency

Fan manufacturers generally use two ways to mention fan efficiency: mechanical efficiency (sometimes called the total efficiency) and static efficiency. Both measure how well the fan converts horsepower into flow and pressure.

The equation for determining mechanical efficiency is:

$$\text{Fan Mechanical Efficiency } (\eta_{\text{mechanical}}), \% = \frac{\text{Volume in } m^3 / \text{sec} \times \Delta p (\text{total pressure}) \text{ in mmWC}}{102 \times \text{Power input to fan shaft in kW}} \times 100$$

The static efficiency equation is the same except that the outlet velocity pressure is not added to the fan static pressure

$$\text{Fan Static Efficiency } (\eta_{\text{static}}), \% = \frac{\text{Volume in } m^3 / \text{sec} \times \Delta p (\text{static pressure}) \text{ in mmWC}}{102 \times \text{Power input to fan shaft in kW}} \times 100$$

Drive motor kW can be measured by a load analyzer. This kW multiplied by motor efficiency gives the shaft power to the fan.

## 4.8 Energy Savings Opportunities

### Minimizing demands on the fan.

1. Minimising excess air level in combustion systems to reduce FD fan and ID fan load.
2. Minimising air in-leaks in hot flue gas path to reduce ID fan load, especially in case of kilns, boiler plants, furnaces, etc. Cold air in-leaks increase ID fan load tremendously, due to density increase of flue gases and in-fact choke up the capacity of fan, resulting as a bottleneck for boiler / furnace itself.
3. In-leaks / out-leaks in air conditioning systems also have a major impact on energy efficiency and fan power consumption and need to be minimized.

The findings of performance assessment trials will automatically indicate potential areas for improvement, which could be one or a more of the following:

1. Change of impeller by a high efficiency impeller along with cone.
2. Change of fan assembly as a whole, by a higher efficiency fan
3. Impeller derating (by a smaller dia impeller)
4. Change of metallic / Glass reinforced Plastic (GRP) impeller by the more energy efficient hollow FRP impeller with aerofoil design, in case of axial flow fans, where significant savings have been reported
5. Fan speed reduction by pulley dia modifications for derating
6. Option of two speed motors or variable speed drives for variable duty conditions
7. Option of energy efficient flat belts, or, cogged raw edged V belts, in place of conventional V belt systems, for reducing transmission losses.
8. Adopting inlet guide vanes in place of discharge damper control
9. Minimizing system resistance and pressure drops by improvements in duct system

## 4.9 Case Study on Pressure Drop Reduction Across the Bag Filter

One of the Cement filter bag house is experiencing high Differential Pressure (DP) across the bag house while producing one particular type of cement. This high DP is resulting in high power consumption and puffing from the bag house. Upon examination it has been found that particle size distribution for this particular type of Cement associated with the fineness is creating high DP.

The results of this replacement are self-explanatory in the details given below:

Application	Bag Filter for Cement Mill	
Problem	High DP across the bag house while producing one particular type of Cement Product	
Reason for the Problem	Characteristics of particle size distribution associated with its fineness is creating high DP across the bag house	
Solution	Replace the existing filter bags with PTFE Membrane filter bags	
Results	Reduction of 50mm WC in DP across the bag house Reduction of 5kWh in power consumption	
	Existing Bag	Bag with membrane
	40000	41600
	96.8/86.9	98.2/88.7
	Mixed Felt	Mixed Felt with PTFE membrane
	165	115
	51	46
Conclusion	The above results are categorically demonstrating that by changing the filter fabric we can achieve significant performance improvement of the bag house	

### Case Studies VSD Applications

**Case 1:** Cement plants use a large number of high capacity fans. By using liners on the impellers, which can be replaced when they are eroded by the abrasive particles in the dust-laden air, the plants have been able to switch from radial blades to forward-curved and backward-curved centrifugal fans. This has vastly improved system efficiency without requiring frequent impeller changes.

For example, a careful study of the clinker cooler fans at a cement plant showed that the flow was much higher than required and also the old straight blade impeller resulted in low system efficiency. It was decided to replace the impeller with a backward-curved blade and use liners to prevent erosion of the blade. This simple measure resulted in a 53 % reduction in power consumption, which amounted to annual savings of BDT. 2.1 million.

**Case 2:** Another cement plant found that a large primary air fan which was belt driven through an arrangement of bearings was operating at system efficiency of 23 %. The fan was replaced with a direct coupled fan with a more efficient impeller. Power consumption reduced from 57 kW to 22 kW. Since cement plants use a large number of fans, it is generally possible to integrate the system such that air can be supplied from a common duct in many cases.

For example, a study indicated that one of the fans was operated with the damper open to only 5 %. By re-ducting to allow air to be supplied from another duct where flow was being throttled, it was possible to totally eliminate the use of a 55 kW fan.

**Case 3:** The use of variable-speed drives for capacity control can result in significant power savings. A 25 ton-per-hour capacity boiler was equipped with both an induced-draft and forced-draft fan. Outlet dampers were used to control the airflow. After a study of the air-flow pattern, it was decided to install a variable speed drive to control air flow. The average power consumption was reduced by nearly 41 kW resulting in annual savings of BDT. 0.33 million. The investment of BDT. 0.65 million for the variable-speed drive was paid back in under 2 years.

**Case 4:** The type of variable-speed drive employed also significantly impacts power consumption. Thermal power stations install a hydraulic coupling to control the capacity of the induced-draft fan. It was decided to install a VFD on ID fans in a 200 MW thermal power plant. A comparison of the power consumption of the two fan systems indicated that for similar operating conditions of flow and plant power generation, the unit equipped with the VFD control unit consumed, on average, 4 million units / annum less than the unit equipped with the hydraulic coupling.

## **4.10 Computational Fluid Dynamics**

### **Use of Computational Fluid Dynamics (CFD) for Energy Efficiency**

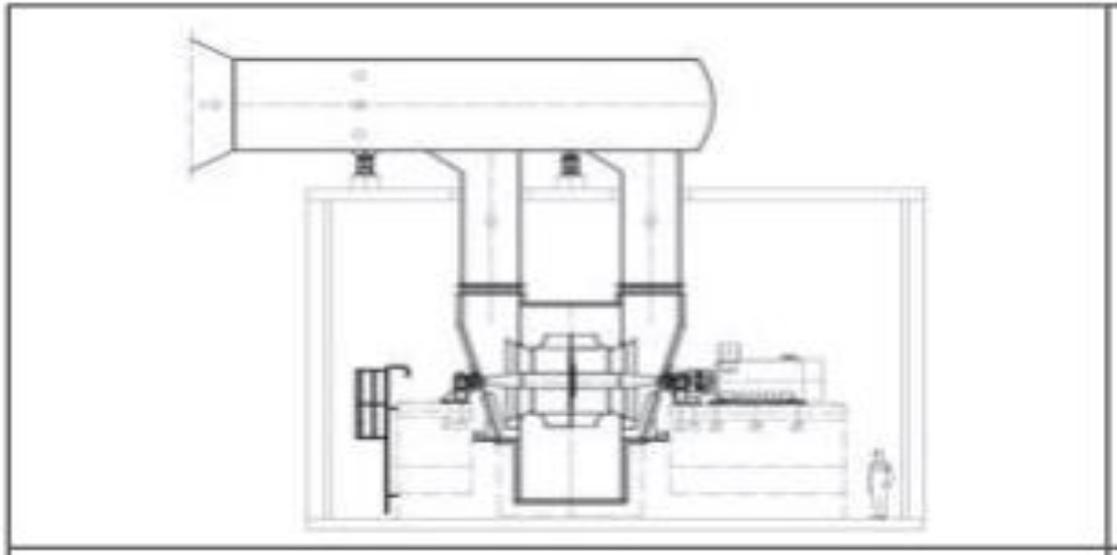
Computational Fluid Dynamics (CFD) is a powerful computer based simulation tool, which uses mathematics to model the physical system and to solve equations to predict the mass, momentum and energy. CFD tools helps to visualize the flow pattern, temperature and pressure profiles as well as particle movement inside the equipment / system under consideration. Being a simulation tool, various designs / operating parameters can be changed to arrive at best operating conditions for the plant. This also helps to determine the effect of changes in key process parameters before making any changes in the manufacturing process at plant. Since it is a proactive analysis and design tool, it can highlight the root cause not just the effect when evaluating plant problems. It is able to reduce scale-up problems because the models are based on fundamental physics and are scale independent. It is also useful in simulating conditions where it is not possible to take detailed measurements because of high temperature or dangerous environment.

### **Application of CFD in a Fan System for Energy Efficiency**

In addition to optimization of the fan itself, the upstream and downstream flow situations are important for trouble free operation of the fan and maximum possible efficiency. If, for instance, the incoming flow is turbulent or swirling because of poorly designed bends or changes in cross-section, this will affect the fan's operating characteristics. Disturbances in the inflow and outflow zones have particularly serious effects in the case of high efficiency fans (> 80 %); as such fans depend on a non-swirling inflow in order to achieve their efficiency figures.

This case study describes the optimization of the induced draught flow of a double-inlet fan using CFD. Figure 4.20 shows the system arrangement drawing. The 3D model derived

from this drawing is shown in Figure 4.21. The very sharp-edged transitions at inlet side affect the smooth flow.

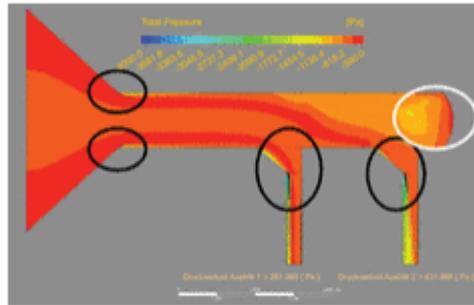


*Figure 4.20 System arrangement drawing with induced draught fan*

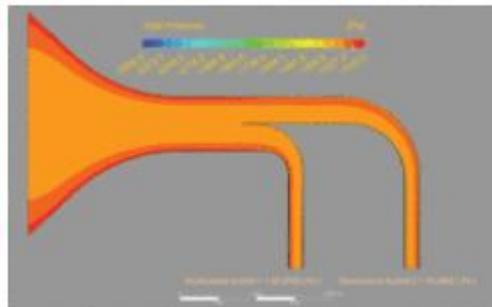


*Figure 4.21 3D model of induced draught fan*

The simulation (Figure 4.22) shows extensive flow disruption and instability. The pressure drop of this system is correspondingly high. From the inlet until the end of the first branch, the pressure drop is about 260 Pa. From the inlet until the next branch the drop is as high as 430 Pa because of the flow disruption near the end of the horizontal duct.



**Figure 4.22 Total pressure profile with original design**



**Figure 4.23 Total pressure profile after optimizing design**

The black circles indicate unfavourable situations because this design involves transitions that are too sharp-edged and thus prevent the flow from following the contours, resulting in turbulence. This causes pressure drops and thus leads to additional electrical energy consumption. By contrast, the white circle indicates a duct end producing an inefficient fluid flow.

Since a portion of the vortex flow is sucked into the fan it has to cope with strongly swirling inflow air and therefore loses efficiency and suffers from mechanical vibrations. Optimization of the geometry involved redesigning the critical points highlighted in Figure 5.22 so that the cross-section transitions were smooth. This redesigning succeeded in reducing the pressure drop at the front inflow duct to the fan by a factor of four, from 261 Pa to 66 Pa, while the pressure drop at the rear inflow duct that had been caused by the poorly-designed horizontal duct end (Figure 5.22, white circle) was even reduced to little more than one sixth the original figure. The severe swirling of the fan intake air (Figure 5.23, red circles) was also significantly reduced, enabling the fan to achieve its rated performance figures.

The mean pressure drop saving of 275 Pa at a volume flow of 5,00,000 m<sup>3</sup>/h significantly reduces the power consumption. The resultant power saving was of the order of 49 kW.

### **Example 4.2**

A V-belt centrifugal fan is supplying air to a process plant. The performance test on the fan gave the following parameters.

Density of air at 0 °C	1.293 kg/m'
Ambient air temperature	40 °C
Diameter of the discharge air duct	0.8 m
Velocity pressure measured by Pitot tube in discharge duct	45 mmWC
Pitot tube coefficient	0.9
Static pressure at fan inlet	- 20 mmWC
Static pressure at fan outlet	185 mmWC
Power drawn by the motor coupled with the fan	75 kW
Belt transmission efficiency	97 %
Motor efficiency at the operating load	93 %

Find out the static fan efficiency.

### Solution

Air temperature	40 °C
Diameter of the discharge air duct	0.8 m
Velocity pressure measured by Pitot tube	45 mmWC
Static pressure at fan inlet	- 20 mmWC
Static pressure at fan outlet	185 mmWC
Power drawn by the motor	75 kW
Transmission efficiency	97%
Motor efficiency	93 %
Area of the discharge duct	$3.14 \times 0.8 \times 0.8 \times 1/4$
	0.5024 m <sup>2</sup>
Pitot tube coefficient	0.9
Corrected gas density	$(273 \times 1.293) / (273 + 40) = 1.1277$
Volume	$\frac{C_p \times A \sqrt{2 \times 9.81 \times \Delta p \times \gamma}}{\gamma}$
	$\frac{0.9 \times 0.5024 \times \text{Sq rt.}(2 \times 9.81 \times 45 \times 1.1277)}{1.1277}$
	12.65 m <sup>3</sup> /s
Power input to the shaft	$75 \times 0.97 \times 0.93$
	67.65 kW
Static Fan Efficiency %	$= \frac{\text{Volume in m}^3 / \text{Sec} \times \text{total static pressure in mmwc}}{102 \times \text{Power input to the shaft in (kW)}}$
Fan static Efficiency	$\frac{12.65 \times (185 - (-20))}{102 \times 67.65}$
	37.58 %

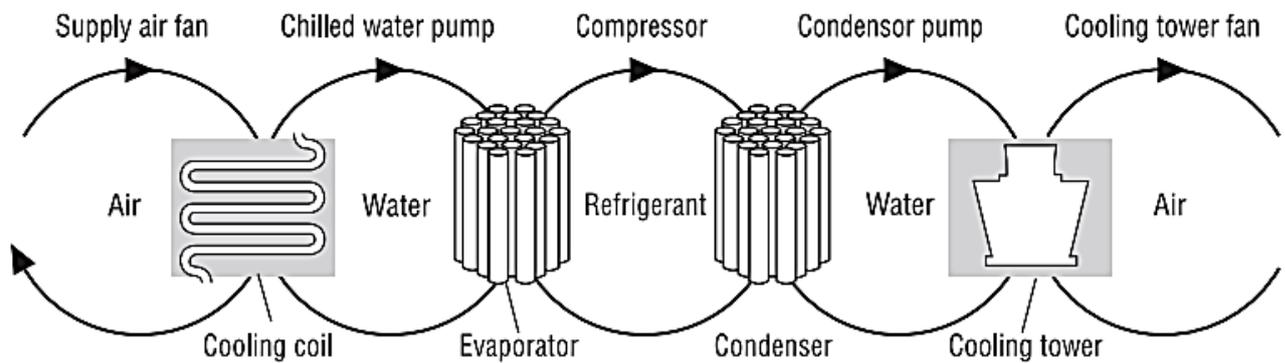
## Chapter 5: HVAC and Refrigeration System

## 5.1 Introduction

The Heating, Ventilation and Air Conditioning (HVAC) and refrigeration system transfers the heat energy from or to the products, or building environment. Energy in form of electricity or heat is used to power mechanical equipment designed to transfer heat from a colder, low-energy level to a warmer, high-energy level.

Refrigeration deals with the transfer of heat from a low temperature level at the heat source to a high temperature level at the heat sink by using a low boiling refrigerant.

There are several heat transfer loops in refrigeration system as described below:



*Figure 5.1 Heat Transfer Loops in Refrigeration System*

In the Figure 5.1, 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 leftmost 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.

## 5.2 Psychrometrics and Air-Conditioning Processes

Psychrometrics is the science of moist air properties and processes, which is used to illustrate and analyze air-conditioning cycles. It translates the knowledge of heating or cooling

loads (which are in kW or tons) into volume flow rates (in m<sup>3</sup>/s or cfm) for the air to be circulated into the duct system.

Water vapor is lighter than dry air. The amount of water vapor that the air can carry increases with its temperature. Any amount of moisture that is present beyond what the air can carry at the prevailing temperature can only exist in the liquid phase as suspended liquid droplets (if the air temperature is above the freezing point of water), or in the solid state as suspended ice crystals (if the temperature is below the freezing point).

The most commonly used psychrometric quantities include the dry and wet bulb temperatures, dew point, specific humidity, relative humidity.

**Psychrometric Chart:**

Psychrometric chart (Figure 5.2) is a chart indicating the psychrometric properties of air such as dry-bulb temperature, wet-bulb temperature, specific humidity, enthalpy of air in kJ/kg dry air, specific volume of air in 111 m<sup>3</sup>/kg and relative humidity  $\phi$  in %. It helps in quantifying and understanding air conditioning process.

**Example 5.1**

Assume that the outside air temperature is 32°C with a relative humidity of  $\phi = 60\%$ . Use the psychrometric chart to determine the air properties. See Figure 5.2.

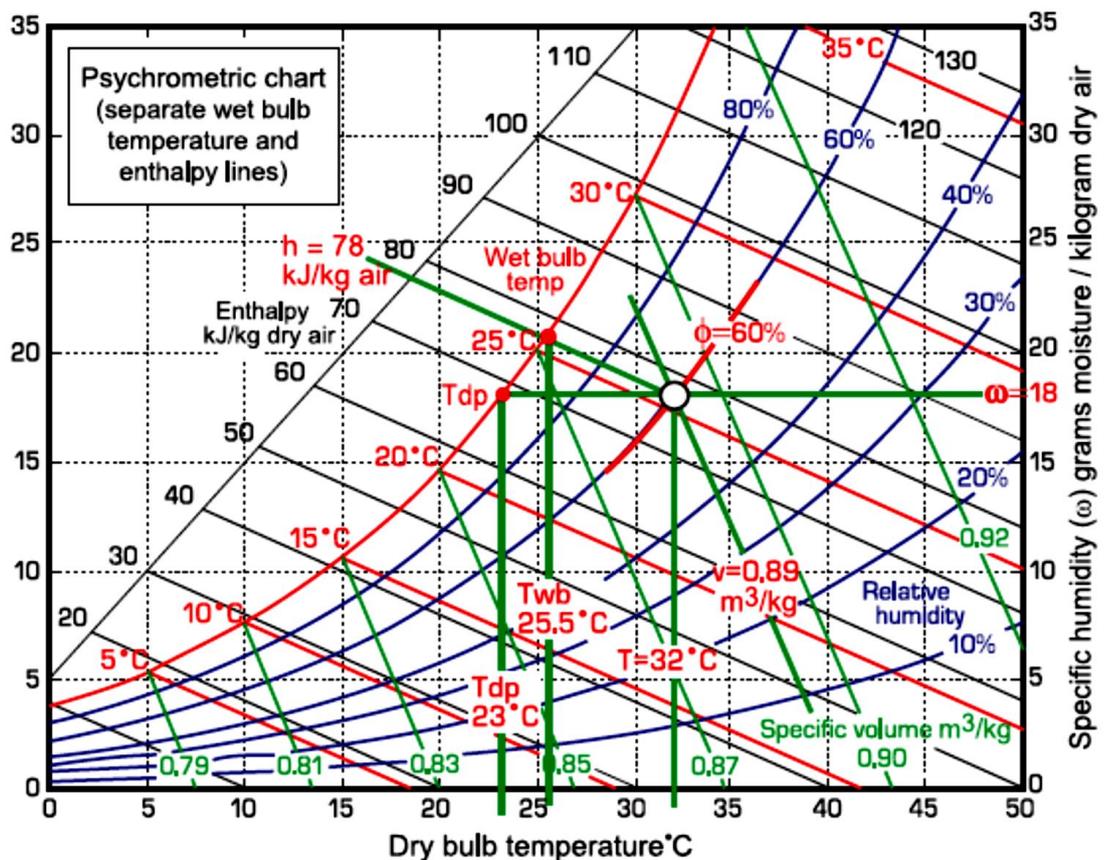


Figure 5.2 Properties of Air at 32°C Dry Bulb Temperature and RH of 60%

**Solution**

Air properties of air at 32°C dry bulb temperature and RH of 60 %

Specific humidity, $\omega$	=	18 gm-moisture/kg-air
Enthalpy, h	=	78 kJ/kg-air
Wet-bulb temperature, $T_{wb}$	=	25.5°C
Dew-point temperature, $T_{dp}$	=	23°C
The specific volume of the dry air, v	=	0.89m <sup>3</sup> /kg

### Comfort Zone

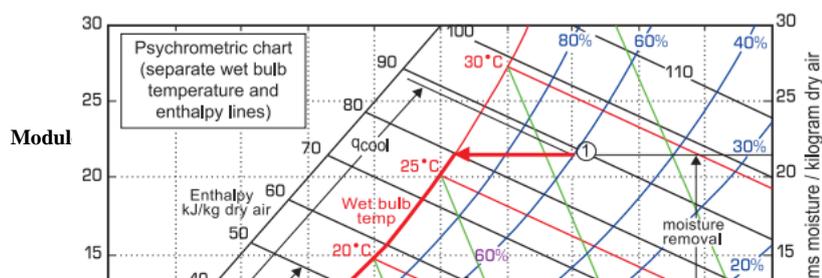
One of the major applications of the Psychrometric Chart is in air conditioning, and we find that most humans feel comfortable when the temperature is between 22°C and 27°C, and the relative humidity  $\phi$  between 40% and 60%. This defines the "comfort zone" which is portrayed on the Psychrometric Chart as shown in Figure 4.3. Thus with the aid of the chart we either heat or cool, add moisture or dehumidify as required in order to bring the air into the comfort zone.

### Example 5.2

Outside air at 35°C and 60% relative humidity is to be conditioned by cooling and re-heating so as to bring the air to the "comfort zone" with the exit temperature of 24°C and 53% RH. Using the Psychrometric Chart neatly plot the required air conditioning process and estimate

(a) the amount of moisture removed, (b) the heat removed, and (c) the amount of heat added.

See Figure 5.3.



*Figure-5.3: Properties of Air*

**Solution**

Using the Figure 5.3

- a) The amount of moisture removed = 11.5g-H<sub>2</sub>O /kg-dry-air
- b) The heat removed = (1)-(2),  $q_{cool} = 48$  kJ/kg-dry-air

The amount of heat added = (2)-(3),  $q_{heat} = 10$  kJ/kg-dry-air Air-

**Conditioning Systems**

Depending on applications, there are several options / combinations, which are available for use as given below:

- Air Conditioning (for comfort / machine)
- Split air conditioners
- Fan coil units in a larger system
- Air handling units in a larger system Refrigeration Systems (for processes)
- Small capacity modular units of direct expansion type similar to domestic refrigerators, small capacity refrigeration units.
- Centralized chilled water plants with chilled water as a secondary coolant for temperature range over 5 °C typically. They can also be used for ice bank formation.
- Brine plants, which use brines as lower temperature, secondary coolant, for typically subzero temperature applications, which come as modular unit capacities as well as large centralized plant capacities.
- The plant capacities upto 50 TR are usually considered as small capacity, 50 — 250 TR as medium capacity and over 250 TR as large capacity units.

A large industry may have a bank of such units, often with common chilled water pumps, condenser water pumps, cooling towers, as an off site utility.

The same industry may also have two or three levels of refrigeration & air conditioning such as:

- Comfort air conditioning (20 — 25 °C)
- Chilled water system (8 —10 °C)
- Brine system (sub-zero applications)

Two principle types of refrigeration plants found in industrial use are: 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.

### 5.3 Types of Refrigeration System

#### Vapour Compression Refrigeration

Heat flows naturally from a hot to a colder body. In refrigeration system the opposite must occur i.e. heat flows from a cold to a hotter body. This is achieved by using a substance called a refrigerant, which absorbs heat and hence boils or evaporates at a low pressure to form a gas. This gas is then compressed to a higher pressure, such that it transfers the heat it has gained to ambient air or water and turns back (condenses) into a liquid. In this way heat is absorbed, or removed, from a low temperature source and transferred to a higher temperature source.

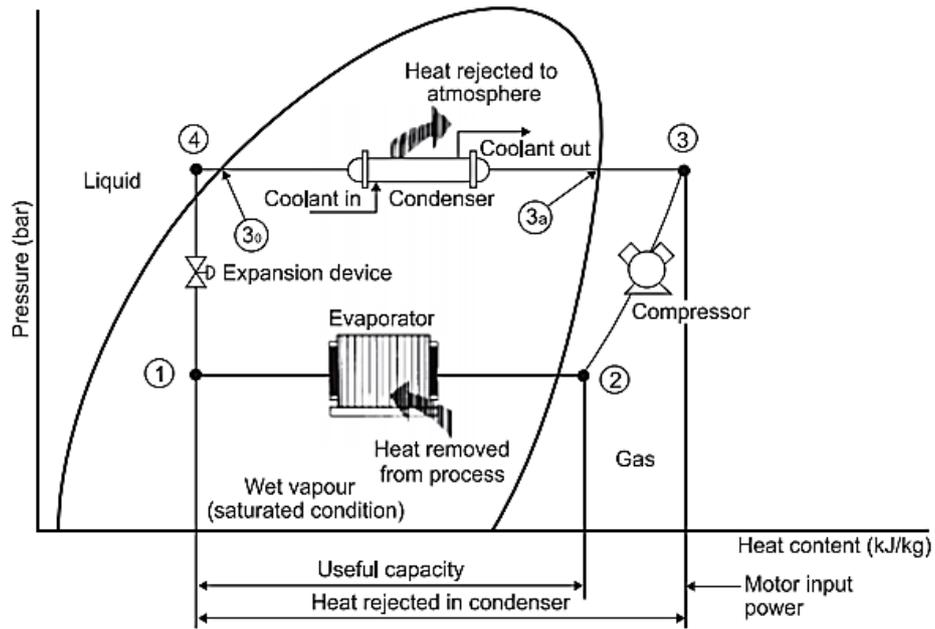
The refrigeration cycle can be broken down into the following stages (see Figure 4.4):

**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. There will also be a big increase in temperature, because a proportion of the energy input 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) desuperheats 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 5.4: Schematic of a Basic Vapour Compression Refrigeration System**

It can be seen that the condenser has to be capable of rejecting the combined heat inputs of the evaporator and the compressor; i.e.  $(1 - 2) + (2 - 3)$  has to be the same as  $(3 - 4)$ . There is no heat loss or gain through the expansion device.

### **Alternative Refrigerants for Vapour Compression Systems**

The use of CFCs is phased out due to their damaging impact on the protective tropospheric ozone layer around the earth. The Montreal Protocol of 1987 and the subsequent Copenhagen agreement of 1992 mandate a reduction in the production of ozone depleting Chlorinated Fluorocarbon (CFC) refrigerants in a phased manner, with an eventual stop to all production by the year 1996. As part of the accelerated phase-out of CFCs, India has completely phased out CFCs by 1st August, 2008.

In response, the refrigeration industry has developed two alternative refrigerants; one based on Hydrochloro Fluorocarbon (HCFC), and another based on Hydro Fluorocarbon (HFC). The HCFCs have a 2 to 10% ozone depleting potential as compared to CFCs and also, they have an atmospheric lifetime between 2 to 25 years as compared to 100 or more years for CFCs (Brandt, 1992). However, even HCFCs are mandated to be phased out, and only the chlorine free (zero ozone depletion) HFCs would be acceptable.

The 19th MOP (Meeting of Parties) took a decision to accelerate the phase-out of HCFC production and consumption for developed and developing countries. The new phase-out schedule for Article 5 parties as per the decision taken at the 19th MOP is as follows:

Base-level for production & consumption: the average of 2009 and 2010

Freeze= 2013 at the base-level

10% reduction in 2015

35% reduction in 2020

67.5% reduction in 2025

100% reduction in 2030 with a service tail of 2.5% annual average during the period 2030-2040 (Source: Ministry of Environment and Forest, Ozone Cell)

Until now, only one HFC based refrigerant, HFC 134a, has been developed. HCFCs are comparatively simpler to produce and the three refrigerants 22, 123, and 124 have been developed. The use of HFCs and HCFCs results in slightly lower efficiencies as compared to CFCs, but this may change with increasing efforts being made to replace CFCs.

### **Absorption Refrigeration**

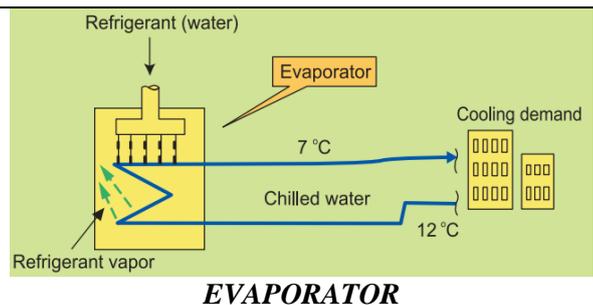
The absorption chiller is a machine, which produces chilled water by using heat such as steam, hot water, gas, oil etc. Chilled water is produced by the principle that liquid (refrigerant), which evaporates at low temperature, absorbs heat from surrounding when it evaporates. Pure water is used as refrigerant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

Heat for the vapour absorption refrigeration system can be provided by waste heat extracted from process, diesel generator sets etc. Absorption systems require electricity to run 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.

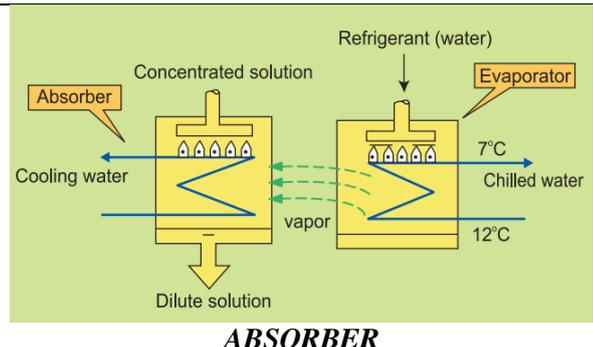
Description of the absorption refrigeration concept is given below:

The refrigerant (water) evaporates at around 4 °C under the high vacuum condition of 754 mmHg in the evaporator. When the refrigerant (water) evaporates, the latent heat of vaporization takes the heat from incoming chilled water.

This latent heat of vaporization can cool the chilled water which runs into the heat exchanger tubes in the evaporator by transfer of heat to the refrigerant (water).

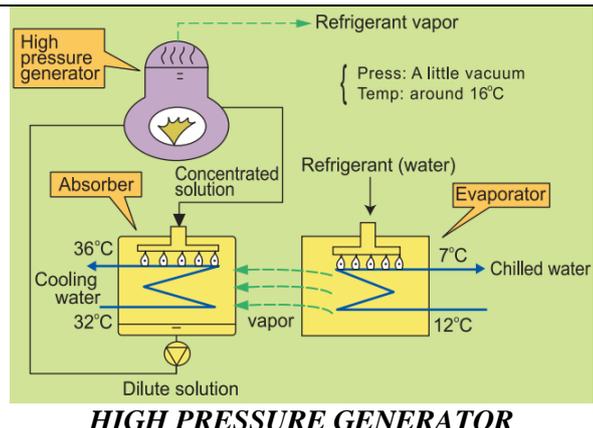


In order to keep evaporating, the refrigerant vapor must be discharged from the evaporator and refrigerant (water) must be supplied. The refrigerant vapor is absorbed into lithium bromide solution which is convenient to absorb the refrigerant vapor in the absorber. The heat generated in the absorption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



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

Absorption chiller is provided with the solution concentrating system by the heating media such as steam, hot water, gas, oil, which performs such function is called generator.

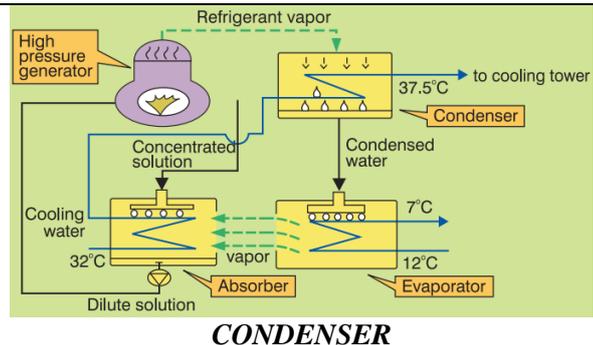


The concentrated solution flows into the absorber and absorbs the refrigerant vapor again.

In order to carryout above works continually and to make complete cycle, 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 function, condenser is installed.



A typical schematic of the absorption refrigeration system is given in the Figure 5.5.

Li-Br-water absorption refrigeration systems 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 of capacities in the range of 10-1500 tons are available. Although the initial cost of absorption system is higher than compression system, operational cost is much lower-if waste heat is used.

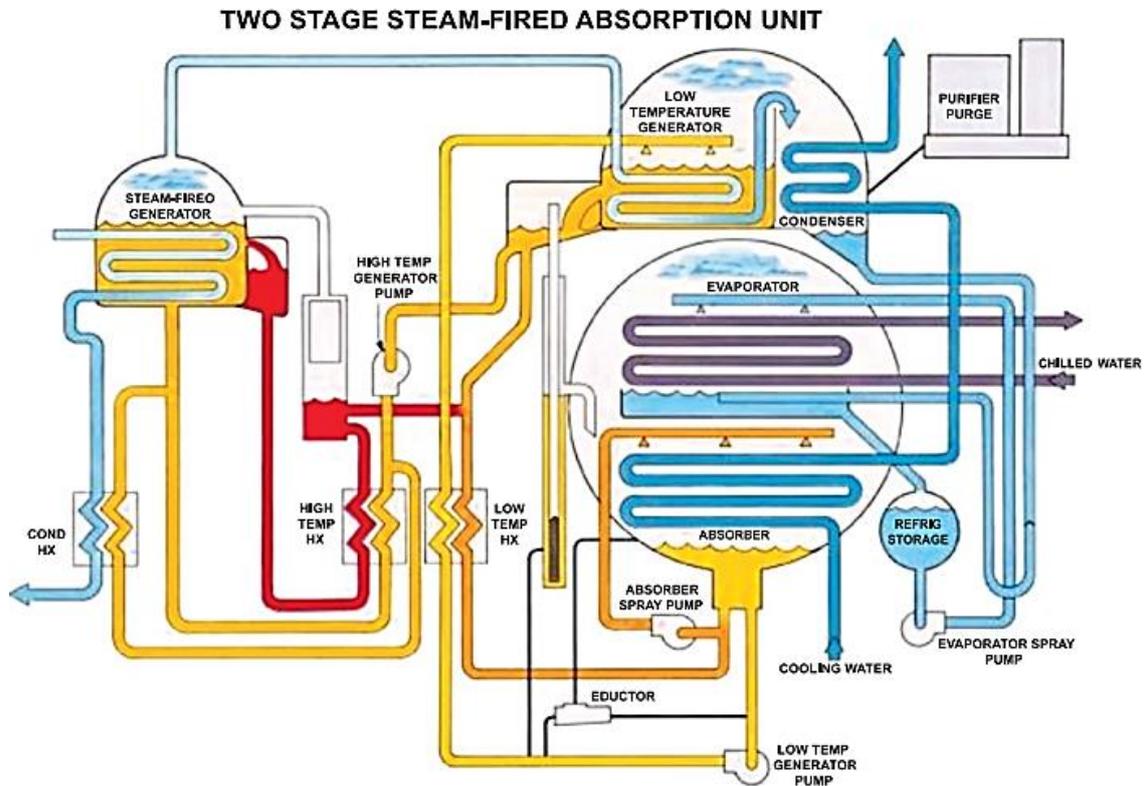


Figure 5.5: Schematic of Absorption Refrigeration System

### Evaporative Cooling

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

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.

### 5.4 Common Refrigerants and Properties

A variety of refrigerants are used in vapor compression systems. The choice of fluid is determined largely by the cooling temperature required. 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 4.1 and the performance of these refrigerants is given in Table 4.2.

<b>Table 5.1 Properties of Commonly used Refrigerants</b>						
Refrigerant	Boiling Point ** (°C)	Freezing Point (°C)	Vapor Pressure * (kPa)	Vapor Volume (m <sup>3</sup> / 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 °C \*\* At Standard Atmospheric Pressure (101.325 kPa)

<b>Table 5.2 Performance of Commonly used Refrigerants*</b>					
Refrigerant	Evaporating Press (kPa)	Condensing Press (kPa)	Pressure Ratio	Vapor Enthalpy (kJ / kg)	COP** <sub>cam</sub>
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 °C Evaporator Temperature, and 30 °C Condenser Temperature

\*\* COP<sub>carnot</sub> = Coefficient of Performance = Temp.<sub>Evap.</sub> / (Temp.<sub>Cond.</sub> - Temp.<sub>Evap.</sub>)

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.

## 5.5 Compressor Types and Application

For industrial use, open type systems (compressor and motor as separate units) are normally used, though hermetic systems (motor and compressor in a sealed unit) also find service in some low capacity applications. Hermetic systems are used in refrigerators, air conditioners, and other low capacity applications. Industrial applications largely employ reciprocating, centrifugal and, more recently, screw compressors, and scroll compressors. Water-cooled systems are more efficient than air-cooled alternatives because the temperatures produced by refrigerant condensation are lower with water than with air.

### Centrifugal Compressors

Centrifugal compressors are the most efficient type (see Figure 4.6) when they are operating near full load. Their efficiency advantage is greatest in large sizes, and they offer considerable economy of scale, so they dominate the market for large chillers. They are able to use a wide range of refrigerants efficiently, so they will probably continue to be the dominant type in large sizes.

Centrifugal compressors have a single major moving part – an impeller that compresses the refrigerant gas by centrifugal force. The gas is given kinetic energy as it flows through the impeller. This kinetic energy is not useful in itself, so it must be converted to pressure energy. This is done by allowing the gas to slow down smoothly in a stationary diffuser surrounding the impeller.



*Figure 5.6 Centrifugal Compressor*

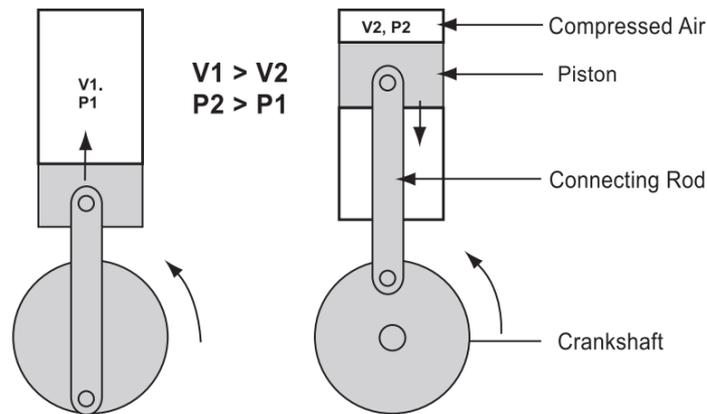
To minimize efficiency loss at reduced loads, centrifugal compressors typically throttle output with inlet guide vanes located at the inlet to the impeller(s). This method is efficient down to about 50% load, but the efficiency of this method decreases rapidly below 50% load.

Older centrifugal machines are not able to reduce load much below 50%. This is because of "surge" in the impeller. As the flow through the impeller is choked off, the gas does not acquire enough energy to overcome the discharge pressure. Flow drops abruptly at this point, and an oscillation begins as the gas *flutters* back and forth in the impeller. Efficiency drops abruptly, and the resulting vibration can damage the machine. Many older centrifugal machines deal with low loads by creating a false load on the system, such as by using hot gas bypass. This wastes the portion of the cooling output that is not required.

Another approach is to use variable-speed drives in combination with inlet guide vanes. This may allow the compressor to throttle down to about 20% of full load, or less, without false loading. Changing the impeller speed causes a departure from optimum performance, so efficiency still declines badly at low loads. A compressor that uses a variable-speed drive reduces its output in the range between full load and approximately half load by slowing the impeller speed. At lower loads, the impeller cannot be slowed further, because the discharge pressure would become too low to condense the refrigerant. Below the minimum load provided by the variable-speed drive, inlet guide vanes are used to provide further capacity reduction.

### **Reciprocating Compressors**

The maximum efficiency of reciprocating compressors (see Figure 5.7) is lower than that of centrifugal and screw compressors. Efficiency is reduced by clearance volume (the compressed gas volume that is left at the top of the piston stroke), throttling losses at the intake and discharge valves, abrupt changes in gas flow, and friction. Lower efficiency also results from the smaller sizes of reciprocating units, because motor losses and friction account for a larger fraction of energy input in smaller systems.



**Figure 5.7: Reciprocating Compressor**

Reciprocating compressors suffer less efficiency loss at partial loads than other types, and they may actually have a higher absolute efficiency at low loads than the other types. Smaller reciprocating compressors control output by turning on and off. This eliminates all part-load losses, except for a short period of inefficient operation when the machine starts.

Larger multi-cylinder reciprocating compressors commonly reduce output by disabling ("unloading") individual cylinders. When the load falls to the point that even one cylinder provides too much capacity, the machine turns off several methods of cylinder unloading are used, and they differ in efficiency. The most common is holding open the intake valves of the unloaded cylinders. This eliminates most of the work of compression, but a small amount of power is still wasted in pumping refrigerant gas to-and-fro through the unloaded cylinders. Another method is blocking gas flow to the unloaded cylinders, which is called "suction cutoff"

Variable-speed drives can be used with reciprocating compressors, eliminating the complications of cylinder unloading. This method is gaining popularity with the drastic reduction in costs of variable speed drives.

### **Screw compressors**

Screw compressors, sometimes called "helical rotary" compressors, compress refrigerant by trapping it in the "threads" of a rotating screw-shaped rotor (see Figure 4.8). Screw compressors have increasingly taken over from reciprocating compressors of medium sizes and large sizes, and they have even entered the size domain of centrifugal machines. Screw compressors are applicable to refrigerants that have higher condensing pressures, such as HCFC-22 and ammonia. They are especially compact. A variety of methods are used to control the output of screw compressors. There are major efficiency differences among the different methods. The most common is a slide valve that forms a portion of the housing that surrounds the screws.

Using a variable-speed drive is another method of capacity control. It is limited to oil-injected compressors, because slowing the speed of a dry compressor would allow excessive internal leakage. There are other methods of reducing capacity, such as suction throttling that are inherently less efficient than the previous two.

## Scroll Compressors

The scroll compressor is an old invention that has finally come to the market. The gas is compressed between two scroll-shaped vanes. One of the vanes is fixed, and the other moves within it. The moving vane does not rotate, but its center revolves with respect to the center of the fixed vane, as shown in Figure 5.9. This motion squeezes the refrigerant gas along a spiral path, from the outside of the vanes toward the center, where the discharge port is located. The compressor has only two moving parts, the moving vane and a shaft with an off-center crank to drive the moving vane. Scroll compressors have only recently become practical, because close machining tolerances are needed to prevent leakage between the vanes, and between the vanes and the casing.

The features of various refrigeration compressors and application criteria are given in the Table 5.3.

**Table 5.3: Comparison of Different Types of Refrigeration Plants**

(Source : ASHRAE & Vendor Information)

S. No.	Parameters	Vapour Compression Chillers			Vapour Absorption Chiller				
		Reciprocating	Centrifugal	Screw	LiBr				Ammonia
					Single Effect	Double Effect	Half Effect	Triple Effect	Single Stage
1	Refrigeration Temp. Range (Brine / Water)	+7 to -30°C	+7 to -0°C	+7 to -25°C	Above 6°C				Upto -33°C
2	Energy Input	Electricity	Electricity	Electricity	Heat (Steam / Hot Water / Hot Oil/ Direct Fired)	Heat (Steam / Hot Water / Hot Oil/ Direct Fired)	Heat (Hot Water)	Heat (Steam /Hot Oil / Direct Fired )	Heat (Steam/ Hot Water / Hot Oil)
3	Heat Input Temp. Range - Maximum -Minimum	-	-	-	Minimum 85°C	Minimum 130°C	Minimum 55°C	Minimum 190°C	Minimum 85°C
4	Typical Energy to TR Ratio								
4	Air Conditioning Temp. Range	0.7-0.9 kW/TR	0.63kW/TR	0.65 kW/TR	5000 kcal/TR	2575 kcal/TR	7500 kcal/TR	2000 kcal/TR	4615 kcal/TR
	Subzero Temp. Range	1.25 to 2.5 kW/TR	---	1.25 to 2.5 kW/TR	---	---	---	---	6666 kcal/hr
5	Refrigerant	R11,R123,R134a Ammonia	R22, R12	R22, R134a Ammonia	Pure Water	Pure Water	Pure Water	Pure Water	Pure Ammonia
6	Absorbent	---	---	---	Water-LiBr solution	Water-LiBr solution	Water-LiBr solution	Water-LiBr solution	Ammonia-LiBr solution
7	Typical single unit capacity range								
7	Air Condition temp. range	1-150 TR	300 TR & above	50-200 TR	30 TR & above	30 TR & above	30 TR & above	50 TR & above	30 TR & above
	Subzero temp. range	10-50 TR	---	50-200 TR	---	---	---	---	30 TR & above
8	Typical COP at Part Load up to 50%	Reduces at part load	Reduces at part load	Improves by 15-20%	Marginal Improvement at Part Load				No variation
9	Typical Internal Pressure Levels - Low -High Typical Internal Temp. Levels	0.15-0.40 bar a 1.20-1.50 bar a -25 to 50°C	2.5-3.5 bar a 11-12 bar -5 to 50°C	2-5.5 bar 18-20 bar -25 to 50°C	5-6 mm Hg (abs) 60-70 mm Hg (abs) +4 to 75°C	5-6 mm Hg (abs) 370-390 mm Hg (abs) +4 to +130°C	5-6 mm Hg (abs) 60-70 mm Hg (abs) +4 to 130°C	5-6 mm Hg (abs) 2 kg/cm <sup>2</sup> (a) +4 to 160°C	1.2kg/cm <sup>2</sup> (a) 18 kg/cm <sup>2</sup> (a) -25 to +150°C

S. No.	Parameters	Vapour Compression Chillers			Vapour Absorption Chiller				
		Reciprocating	Centrifugal	Screw	LiBr				Ammonia
					Single Effect	Double Effect	Half Effect	Triple Effect	Single Stage
10	-Typical Cooling tower capacity range per 100 TR of chillers - Air-conditioning Temperature Range - Subzero temp. range	130 190	120 ---	120 160	260 ---	200 ---	370 ---	170 ---	290 290
11	Typical Make-up water quantity range in Ltrs/Hr. -Air Conditioning temperature range -Subzero temp. range	672 983	620 ---	620 830	1345 ---	1035 ---	1914 ---	880 ---	1500 1500
12	Material of construction -Generator	---	---	---	Cu-Ni or Stainless Steel				Carbon Steel
	-Absorber	---	---	---	Cu-Ni				Carbon Steel
	-Evaporator	Copper / Carbon steel	copper / Carbon steel	Copper/ Carbon steel	Cu-Ni				Carbon Steel
	-Condenser	Copper / Carbon steel	Copper / Carbon steel	Copper / Carbon steel	Cu-Ni				Carbon Steel
	-Solution Heat Exchange	---	---	---	Carbon Steel				Carbon Steel
	-Solution Pump	---	---	---	Cast Iron Hermatically Sealed (Canned motor type)				Cast Iron with Meh.Seal
	-Refrigerant pump	---	---	---	Cast Iron Hermatically Sealed (Canned motor type)				Not needed
13	Expected Life	25-30 years			15-20 years				50 years
14	Normally Expected Repairs / Maintenance	Periodic Compressor Overhaul Tube Replacement after 1-12 years			Tube Replacement due to Corrosion				Practically no repairs
15	Factory Assembled packaged Or Site Assembled	Factory Assembled							Factory Assembled upto 230 TR in A/C & subzero range
16	Beneficial Energy Sources	Low cost Electricity	Low cost Electricity	Low cost Electricity	a) Waste Heat b) Low cost steam / Low cost fuels				

S. No.	Parameters	Vapour Compression Chillers			Vapour Absorption Chiller				
		Reciprocating	Centrifugal	Screw	LiBr				Ammonia
					Single Effect	Double Effect	Half Effect	Triple Effect	Single Stage
17	Critical Parameters	-Electricity supply	-Lubrication System -Compressor Operation & Maintenance -Electrical Power Panel Maintenance	---	a) Vacuum in Chiller b) Purge System for Vacuum c) Corrosion Inhibitors in Absorbent d) Surfactants in Absorbent e) Cooling Water Treatment f) Cooling Water Temperature g) Heat Source Temperature				Sudden Power failure for 45-60 min. or more can disturb the distillation column for continuous operation. Needs D.G. set if there is frequent power failure for periods longer than 30 min.

## 5.6 Selection of a Suitable Refrigeration System

A clear understanding of the cooling load to be met is the first and most important part of designing / selecting the components of a refrigeration system. Important factors to be considered in quantifying the load are the actual cooling need, heat (cool) leaks, and internal heat sources (from all heat generating equipment). Consideration should also be given to process changes and / or changes in ambient conditions that might affect the load in the future. Reducing the load, e.g. through better insulation, maintaining as high a cooling temperature as practical, etc. is the first step toward minimizing electrical power required to meet refrigeration needs. With a quantitative understanding of the required temperatures and the maximum, minimum, and average expected cooling demands, selection of appropriate refrigeration system (single-stage / multi-stage, economized compression, compound / cascade operation, direct cooling / secondary coolants) and equipment (type of refrigerant, compressor, evaporator, condenser, etc.) can be undertaken.

## 5.7 Performance Assessment of Refrigeration Plants

The cooling effect produced is quantified as tons of refrigeration (TR).

1 TR of refrigeration = 3024 kcal/hr heat rejected. The refrigeration TR is assessed as

$$TR = Q C_p (T_i - T_o) / 3024$$

Where,

Q is mass flow rate of coolant in kg/hr

$C_p$  is coolant specific heat in kcal /kg °C

$T_i$  is inlet, temperature of coolant to evaporator (chiller) in °C

$T_o$  is outlet temperature of coolant from evaporator (chiller) in °C The above TR is also called as chiller tonnage.

The specific power consumption kW/TR is a useful indicator of the performance of refrigeration system. By measuring refrigeration duty performed in TR and the kilowatt inputs, kW/TR is used as a reference 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 as well condenser water (for heat rejection to cooling tower) pump and cooling tower fan in the cooling tower. Effectively, the overall energy consumption would be towards:

- Compressor kW
- Chilled water pump kW
- Condenser water pump kW
- Cooling tower fan kW, for induced / forced draft towers

The specific power consumption for certain TR output would therefore have to include:

- Compressor kW/TR
- Chilled water pump kW/TR
- Condenser water pump kW/TR
- Cooling tower fan kW/TR

The overall kW/TR is the sum of the above.

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

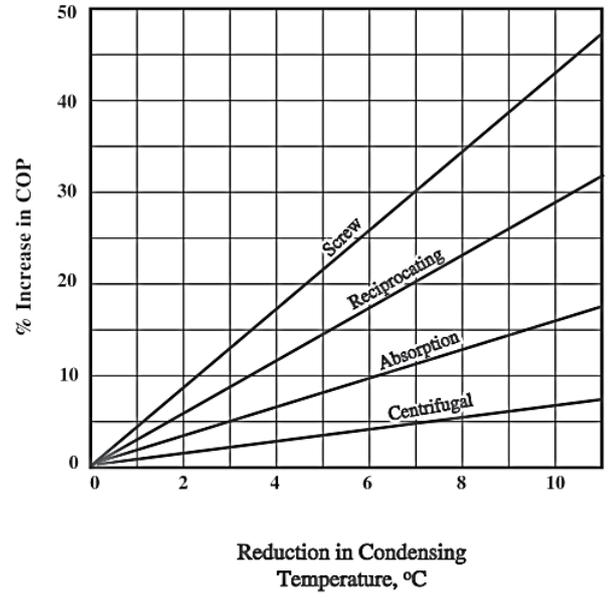
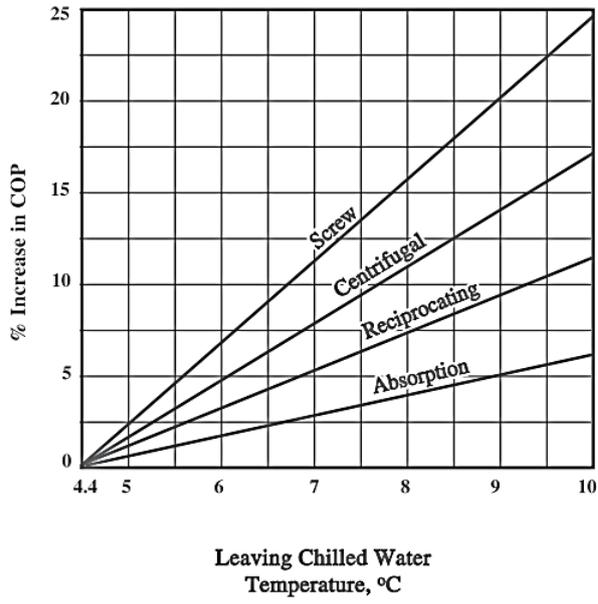
$$COP_{\text{carnot}} = T_e / (T_c - T_e)$$

This expression also indicates that higher  $COP_{\text{carnot}}$  is achieved with higher evaporator temperature and lower condenser temperature.

But  $COP_{\text{carnot}}$  is only a ratio of temperatures, and hence does not take into account the type of compressor. Hence the COP normally used in the industry is given by

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

Where the cooling effect is the difference in enthalpy across the evaporator and expressed as kW. The effect of evaporating and condensing temperatures are given in the Figure 5.8 and Figure 5.9 below:



Effect Of Evaporator Temperature On Chiller COP  
 Effect Of Condensing Temperature On Chiller COP  
**Figure 5.8 & 5.9 Effect of Evaporator and Temperature on Chiller COP**

In the field performance assessment, accurate instruments for inlet and outlet chilled water temperature and condenser water temperature measurement are required, preferably with a least count of 0.1°C. Flow measurements of chilled water can be made by an ultrasonic flow meter directly or inferred from pump duty parameters. Adequacy check of chilled water is needed often and most units are designed for a typical 0.68 m<sup>3</sup>/hr per TR (3 gpm/TR) chilled water flow. Condenser water flow measurement can also be made by a non-contact flow meter directly or inferred from pump duty parameters. Adequacy check of condenser water is also needed often, and most units are designed for a typical 0.91 m<sup>3</sup>/hr per TR (4 gpm / TR) condenser water flow.

In case of 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 AHU or the FCU and the refrigeration load in TR is assessed as;

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

Where,

Q is the air flow in m<sup>3</sup>/h

ρ is density of air kg/m<sup>3</sup>

h<sub>in</sub> is enthalpy of inlet air kcal/kg

h<sub>out</sub> is enthalpy of outlet air kcal/kg

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

Estimation of 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, no. 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<sup>2</sup>
- Medium size office i.e., 10-30 people occupancy with central AC= 0.06 TR/ m<sup>2</sup>
- Large multistoried office complexes with central AC= 0.04 TR/ m<sup>2</sup>

### **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 derived from 100% of the equipment's capacity level and is based on design conditions that are considered the most critical. These conditions occur may be, for example, during only 1% of the total time the equipment is in operation throughout the year. Consequently, it is essential to have data that reflects how the equipment operates with partial loads or in conditions that demand less than 100% of its capacity. To overcome this, an average of kW/TR with partial loads i.e., Integrated Part Load Value (IPLV) have to be formulated.

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 usually operates at between 50 % and 75% of its capacity. This is why it is so important to prepare specific analysis for each case that addresses the four points already mentioned, as well as developing a profile of the heat exchanger's operations during the year.

## **5.8 Factors Affecting Performance & Energy Efficiency of Refrigeration Plants**

### **Design of Process Heat Exchangers**

There is a tendency of the process group to operate with high safety margins which influences the compressor suction pressure / evaporator set point. For instance, a process cooling requirement of 15°C would need chilled water at a lower temperature, but the range can vary from 6°C to say 10°C. At 10°C chilled water temperature, the refrigerant side temperature has to be lower, say —5°C to +5°C. The refrigerant temperature, again sets the corresponding suction pressure of refrigerant which decides the inlet duty conditions for work of compression of the refrigerant compressor. Having the optimum / minimum driving force (temperature difference) can, thus, help to achieve highest possible suction pressure at the compressor, thereby leading to less energy requirement. 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°C raise in evaporator temperature can help to save almost 3 % on power consumption. The TR capacity of the same machine will also increase with the evaporator temperature, as given in Table 5.4.

**Table 5.4 Effect of Variation in Evaporator Temperature on Compressor Power Consumption**

Evaporator Temperature (°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°C

Towards rationalizing the heat transfer areas, the heat transfer coefficient on refrigerant side can be considered to range from 1400 - 2800 watts /m<sup>2</sup>K.

The refrigerant side heat transfer areas provided are of the order of 0.5 Sq.m /TR and above in evaporators.

Condensers in a refrigeration plant are critical equipment that influences the TR capacity and power consumption demands. Given a refrigerant, the condensing temperature and corresponding condenser pressure, depend upon the heat transfer area provided, effectiveness of heat exchange and the type of cooling chosen. A lower condensing temperature, pressure, in best of combinations would mean 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 choices of condensers in practice range from air cooled, air cooled with water spray, and heat exchanger cooled. Generously sized shell and tube heat exchangers as condensers, with good cooling tower operations help to operate with low discharge pressure values and the TR capacity of the refrigeration plant also improves. With same refrigerant, R22, a discharge pressure of 15 kg/cm<sup>2</sup> with water cooled shell and tube condenser and 20 kg/cm<sup>2</sup> with air cooled condenser indicate the kind of additional work of compression duty and almost 30 % additional energy consumption required by the plant. One of the best option at design stage would be to select generously sized (0.65 m<sup>2</sup>/TR and above) shell and tube condensers with water-cooling as against cheaper alternatives like air cooled condensers or water spray atmospheric condenser units.

The effect of condenser temperature on refrigeration plant energy requirements is given in Table 5.5.

**Table 5.5 Effect of Variation in Condenser Temperature on Compressor Power**

Condensing Temperature (°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°C

## Maintenance of Heat Exchanger Surfaces

After ensuring procurement, effective maintenance holds 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 most effective and efficient option.

Fouled condenser tubes force the compressor to work harder to attain the desired capacity. For example, a 0.8 mm scale build-up on 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°C temperature in water returning from the cooling tower reduces compressor power consumption by 3.0 % (see Table 5.6).

**Table 5.6 Effect of Poor Maintenance on Compressor Power Consumption**

Condition	Evaporator Temp. (°C)	Condenser Temp. (°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 Bangladesh. However, the percentage change in power consumption is indicative of the effect of poor maintenance.

## 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 requirements, it is preferable (due to equipment design limitations) and often economical to employ multi-stage reciprocating machines or centrifugal / screw compressors.

Multi-staging systems are of two-types: compound and cascade — and are applicable to all types of compressors. With reciprocating or rotary compressors, two-stage compressors are preferable for load temperatures from  $-20$  to  $-58^{\circ}\text{C}$ , and with centrifugal machines for temperatures around  $-3^{\circ}\text{C}$ .

In multi-stage operation, a first-stage compressor, sized to meet the cooling load, feeds into the suction of a second-stage compressor after inter-cooling of the gas. Apart 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 work of compression is shared equally by the two compressors. Therefore, two compressors with low compression ratios can in combination provide a high compression ratio.

For temperatures in the range of  $-46^{\circ}\text{C}$  to  $-101^{\circ}\text{C}$ , cascaded systems are preferable. In this system, two separate systems using different refrigerants are connected such that one provides the means of heat rejection to the other. The chief 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.

### **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.

### **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 as against continuous capacity modulation of centrifugal through vane control and 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 leaving 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.

### **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 accorded. 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.

### **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 night time 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.

### **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.

### **5.9 Performance Assessment of window, split and package air conditioning units Air Conditioners**

**Energy Efficiency Ratio (EER):** EER is calculated by dividing a chiller's cooling capacity (in watts) by its power input (in watts) at full-load conditions. This definition of EER has been adopted in star labeling programme.

The energy efficiency ratio (EER) = Refrigeration effect in Watts/ Power input in Watts  
Based on the condition of the air, the air properties such as specific volume and enthalpy at both inlet and out let conditions can be obtained from psychrometric charts. From these parameters the capacity delivered by the air conditioner can be evaluated, which when compared with power drawn would reveal its performance in terms of kW/TR and EER.

Performance Assessment of Package Air Conditioner -

#### **Example 5.5**

Package air conditioner capacity	-	10 TR
Average Air Velocity	-	2.27 m/s (across suction side filter)
Cross Sectional Area	-	0.58 m <sup>2</sup>
Air Flow Rate	-	1.32 m <sup>3</sup> /sec - 4751 m <sup>3</sup> /hr
Inlet Air Condition		
DBT	-	20 °C
WBT	-	14 °C
Sp.Vol	-	0.8405 m <sup>3</sup> /kg
Enthalpy	-	9.37 kcal/kg
Outlet Air Condition		
DBT	-	12.7 °C
WBT	-	11.3 °C
Enthalpy	-	7.45 kcal/kg
Cooling Effect Delivered	-	3.6 TR = 12.7 kW
Power Drawn		
Compressor	-	4.71 kW
	-	4.3 kW (shaft power@ 90% motor efficiency)
Pump	-	2.14 kW
C.T Fan	-	0.384 kW
Specific Energy Consumption		
Compressor	-	1.31 kW/TR
Overall	-	2 kW/TR
EER	-	12700/4300
	-	2.95 W/W

Note: The Package A/C unit has two compressors of 5 TR capacity each, of which only one was in operation due to low cooling load.

### 5.10 Cold Storage Systems

A Refrigerated storage which includes cold storage and frozen food storage is the best known method of preservation of food to retain its value and flavor.

The refrigeration system in a cold storage is usually a vapour compression system comprising the compressor, condenser, receiver, air cooling units and associate piping and controls.

In smaller cold rooms and walk-ins the practice is to use air cooled condensing units with sealed, semi-sealed or open type compressors. In the light of the CFC phase out the trend now is to use HCFC22, HFC-134a or other substitute refrigerants. In the medium and large sized units the practice is to use a central plant with ammonia as the refrigerant.

In some present day medium and large sized units with pre-fabricated (insulated) panel construction, the trend is to use modular HCFC-22/HFC units which are compact, lightweight and easy to maintain.

### **Energy Saving Opportunities in Cold Storage Systems:**

Energy cost constitutes a major part of the running cost of a cold store. Apart from the problems of the availability of electrical energy, the ever increasing rate of electrical energy seriously affects the economic viability of cold store units.

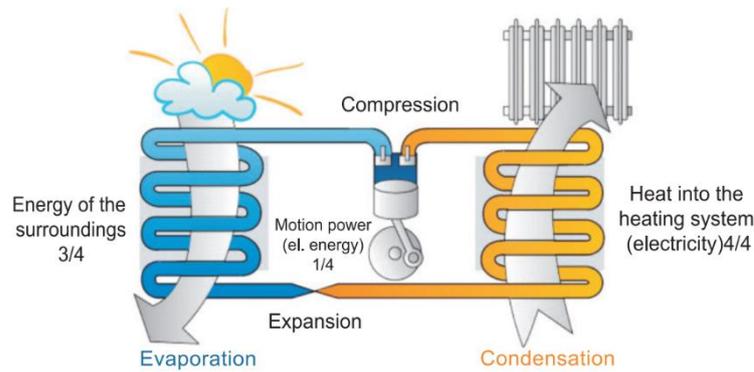
Following are some of the measures adopted to achieve energy efficient operation.

- **Cold Store Building Design:** Proper orientation, compact arrangement of chambers, shading of exposed walls, adequate insulation etc. are some of the important factors.
- **Refrigeration System:** The system must be designed for optimum operating conditions like evaporating and condensing temperatures, as these conditions have a direct bearing on energy consumption.
- **Compressor capacity control system** helps in energy savings during partial load operation.
- **Control System:** The proper control systems for refrigerant level, room temperature, compressor capacity etc., are required to further optimize energy consumption.
- **Air Curtain or Strip Curtain:** The use of air curtains and strip curtains is a common feature in present day cold stores as they help reduce air infiltration due to frequent and sometimes long door openings. Fan operated air curtains are expensive and work on electrical power whereas strip curtains are cheaper and need no energy for operation.
- **Heat Recovery System:** In processing plant cold stores, a heat reclaim system can be installed to recover a part of the heat rejected by the refrigeration system. This can be gainfully utilised in generating hot water free of cost.

## **5.11 Heat Pumps and their Applications**

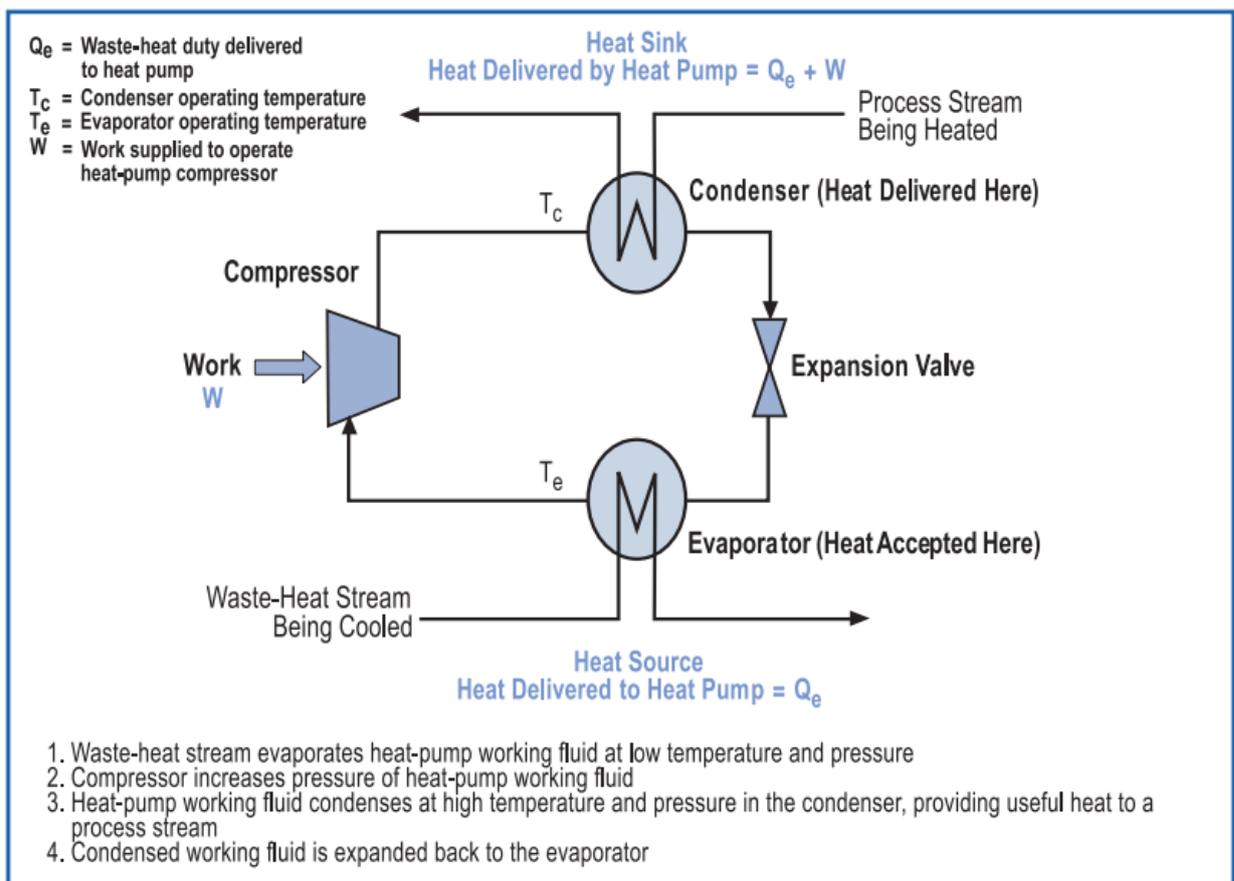
### **Heat Pump Technology**

A heat pump is same as an air conditioner except that the heat rejected in an air conditioner becomes the useful heat output. Heat flows naturally from a higher to a lower temperature. Heat pumps, however, are able to force the heat flow in the other direction, using a relatively small amount of high quality drive energy (electricity, fuel, or high-temperature waste heat). For the example shown in 5.12 the heat pump takes three units of energy from atmosphere and with an additional one unit by way of compressor work is able to provide four units of energy at a higher temperature. Thus heat pumps can transfer heat from natural heat sources in the surroundings, such as the air, ground or water, or from man-made heat sources such as industrial or domestic waste, to a building or an industrial application.



**Figure 5.10 Heat Pump**

In order to transport heat from a heat source to a heat sink, external energy is needed to drive the heat pump. Theoretically, the total heat delivered by the heat pump is equal to the heat extracted from the heat source, plus the amount of drive energy supplied. Electrically-driven heat pumps for heating buildings typically supply 100 kWh of heat with just 20-40 kWh of electricity. Many industrial heat pumps can achieve even higher performance, and supply the same amount of heat with only 3-10 kWh of electricity. The principle of operation of heat pump is shown in the Figure 5.11.



**Figure 5.11 Principle of Operation**

## Heat Pump Applications

Industrial heat pumps are mainly used for:

- Space heating
- Heating of process streams
- Water heating for washing, sanitation and cleaning
- Steam production
- Drying/dehumidification
- Evaporation
- Distillation
- Concentration

When heat pumps are used in drying, evaporation and distillation processes, heat is recycled within the process. For heating of the space, process streams and steam production, the heat pumps utilise (waste) heat sources between 20°C and 100°C.

### 5.12 Ventilation Systems

Ventilation can be simply described as air circulation, the extraction of stale, overheated and contaminated air and supply and distribution of fresh air in amounts necessary to provide healthy and comfortable conditions for the occupants of the room. The ventilation effectiveness is dictated by number of Air Changes per Hour (ACH). The number air changes depend on the purpose and function. Typical air changes are given in Table 5.7 for various operations.

<i>Table 5.7 Typical Air Changes per Hour</i>	
Location	Air changes per hour
Boiler room	15 — 30
Compressor room	10 — 12
Conference rooms	10 — 20
Engine rooms	15 — 30
Lavatories	6 — 15
Offices	6 — 10
Welding shops	15 — 30

**Calculation of ventilation rate:** If the compressor room size is 15 m (L) X 10 m (B) X 4 m (H) then the ventilation rate is  $= L \times B \times H \times ACH$   
 $= 15 \times 10 \times 4 \times 10$   
 $= 6000 \text{ m}^3/\text{hr}$

### 5.13 Ice Bank Systems

Ice Bank System is a proven technology that has been utilized for decades. Thermal energy storage takes advantage of low cost, off-peak electricity, produced more efficiently throughout the night, to create and store cooling energy for use when electricity tariffs are higher, typically during the day. There are full- and partial- load Off-Peak Cooling systems. The essential element for either full- or partial- storage configurations are thermal- energy storage tanks. Each tank contains a spiral-wound, polyethylene-tube heat exchanger surrounded with water. ICEBANK tanks are available in a variety of sizes ranging from 45 to over 500 ton-hours. These systems are economical based on the electricity tariff of particular utility. These systems can be employed to meet the air conditioning requirements in the commercial

buildings as well as to meet the chilling requirements in Dairy and process industry. The main advantage of these systems is to reduce the peak demand of the utility and also reduce the cost of operation for the end user.

### **How Ice Bank Works?**

With a partial-storage system, the chiller can be 40 to 50 percent smaller than other HVAC systems, because the chiller works in conjunction with the ICEBANK tanks during on-peak daytime hours to manage the building's cooling load. During off-peak night time hours, the chiller charges the ICEBANK tanks for use during the next day's cooling. The lowest possible average load is obtained by extending the chiller hours of operation.

## **5.14 Humidification Systems**

This is a process involving reduction in dry bulb temperature and increase in specific humidity. The atmospheric conditions with respect to humidity play a very important part in many manufacturing processes. For example in textile processing the properties like dimensions, weight, tensile strength, elastic recovery, electrical resistance, rigidity etc. of all textile fibre are influenced by humidity maintained. Temperature does not have a great effect on the fibres but the temperature dictates the amount of moisture the air will hold in suspension and, therefore, temperature and humidity must be considered together. Humidification system without chilling helps to maintain only the RH% without much difficulty.

### **Adiabatic saturation or evaporative cooling**

In this process (Figure 5.12) air comes in direct contact with water in the air washer. There is heat and mass transfer between air and water. The humidity ratio of air increases. If the time of contact is sufficient, the air gets saturated. Latent heat of evaporation required for conversion of water into water vapor is taken from the remaining water. When equilibrium conditions are reached, water cools down to the wet bulb temperature of the air. If the air washer is ideal, the dry bulb temperature and wet bulb temperature of the air would be equal. Dry bulb temperature of the air goes down in the process and the effect of cooling is due to the evaporation of some part of the water. That is why it is called **Evaporative Cooling**.

The sensible heat is decreased as the temperature goes down but the latent heat goes up as water vapour is added to the air. The latent heat required by the water which is evaporated in the air is drawn from the sensible heat of the same air. Thus it is transformation of sensible heat to latent heat. During this process the enthalpy of air remains the same. If humidity ratios of saturated air and of the air before saturation are known, then the difference between the two would be the amount of water vapour absorbed by unit weight of dry air.

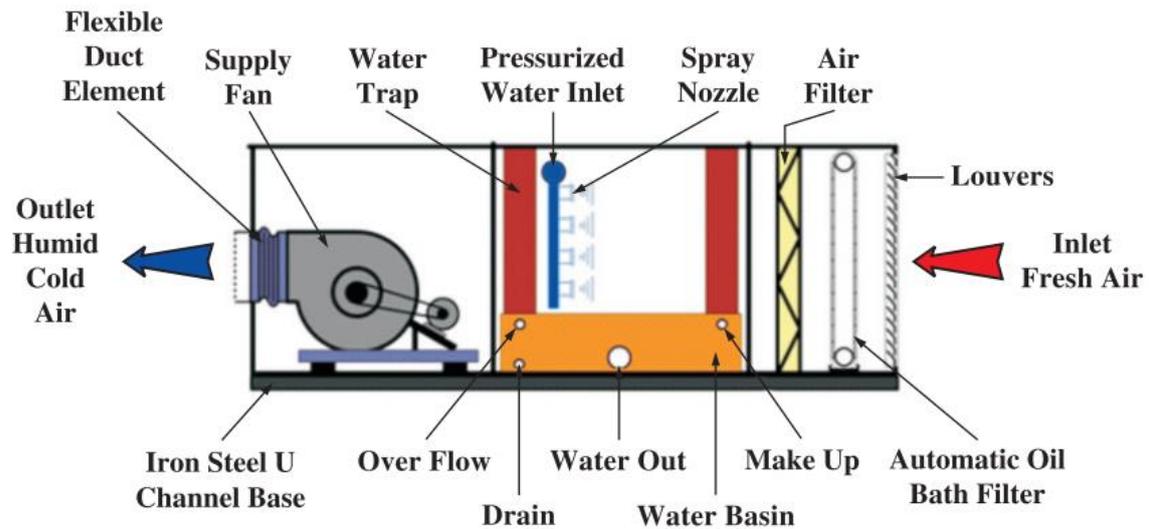


Figure 5.12 Air Washer Unit

### Humidifying Air by adding Water

If water is added to air without any heat supply the state of air change **adiabatic** along a constant enthalpy line in the *psychrometric chart*. The dry bulb temperature of the air decreases.

The amount of added water can be expressed as

$$m_w = v \rho (\dot{\omega}_{out} - \omega_{in})$$

Where,

$m_w$  = mass of added water (kg/hr)

$v$  = volume flow of air ( $m^3/hr$ )

$\rho$  = density of air - vary with temperature,  $1.293 \text{ kg}/m^3$  at  $20^\circ\text{C}$  ( $\text{kg}/m^3$ )

$\dot{\omega}$  = specific humidity of air ( $\text{kg}/\text{kg}$ )

### Example 5.6

#### Humidifying Air by adding Water

In an air washer of textile humidification system airflow of  $3000 \text{ m}^3/\text{h}$  at  $25^\circ\text{C}$  and 10% relative humidity is humidified to 60% relative humidity by adding water through spray nozzles. Calculate the amount of water required. The specific humidity of air at inlet and outlet are  $0.002 \text{ kg}/\text{kg}$  and  $0.0062 \text{ kg}/\text{kg}$  respectively.

#### Solution

The amount of water added can be calculated as:

$$m_w = 3000 \times 1.184 \times (0.0062 - 0.002) = 14.9 \text{ kg/h}$$

### 5.15 Energy Saving Opportunities

#### a) Cold Insulation

Insulate all cold lines / vessels using economic insulation thickness to minimize heat gains; and choose appropriate (correct) insulation.

b) Building Envelope

Optimise air conditioning volumes by measures such as use of false ceiling and segregation of critical areas for air conditioning by air curtains.

c) Building Heat Loads Minimisation

Minimise 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.

d) Process Heat Loads Minimisation

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

e) 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, avoid bypass flows by closing valves of idle equipment.
- Minimize part load operations by matching loads and plant capacity on line; 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 VAR system where economics permit as a non-CFC solution.

## 5.16 Case Study: Screw Compressor Application

### Background

Rotary Screw Compressors are widely used for refrigeration applications to compress ammonia & other refrigerating gases. A typical sectional view of the compressor is shown below in Figure 5.13.

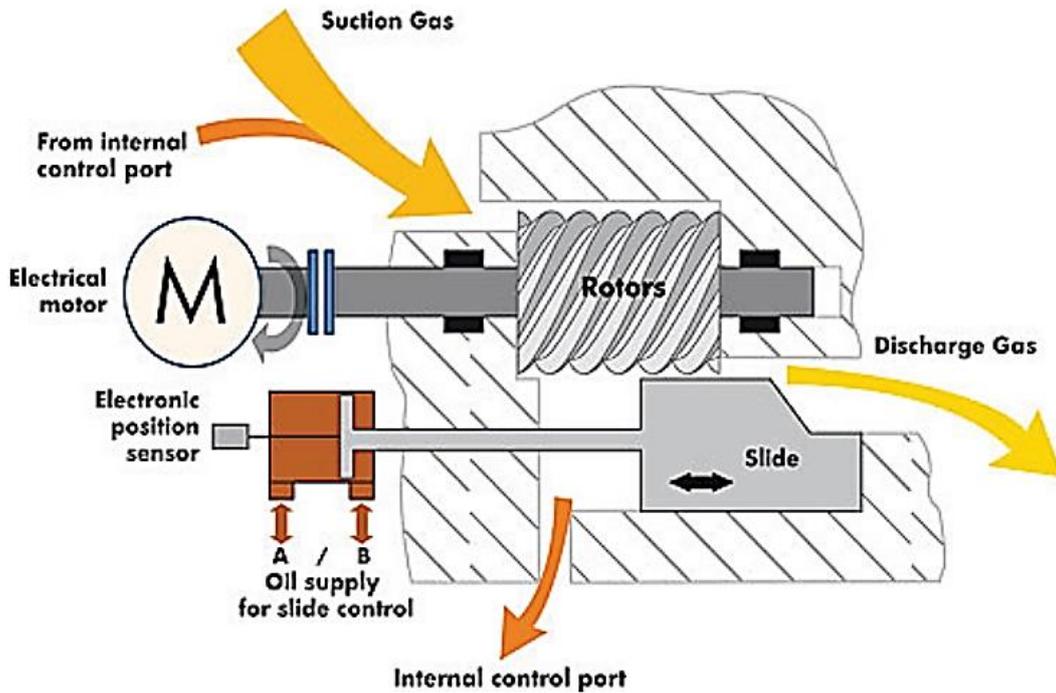


Figure 5.13 Section of Screw Compressor showing the meshing screws & slide valve

The table 5.10 below gives the measured system data during compressor running at part load condition.

Table 5.10 Performance parameters for partial load conditions

Volumetric refrigerant flow (m <sup>3</sup> )	Suction Pressure (kg/cm <sup>2</sup> )	Discharge Pressure (kg/cm <sup>2</sup> )	Inlet Temp. (°C)	Outlet Temp. (°C)	Total refrigeration load (Calories)	Total refrigeration load (TR)
21.03	3.19	11.89	28.08	1.49	558964	184.73
26.62	3.16	11.65	26.85	2.12	658336	217.57
26	3.17	11.63	27.75	1.68	678000	224.07
28	3.22	11.7	27.91	1.92	727650	240.47
32	3.22	11.78	27.98	2.28	822400	271.79
20.13	3.25	12.22	27.28	1.5	518947	171.5
21.71	3.25	12.38	27.77	1.6	568197	187.78
23.36	3.25	12.52	28.4	1.64	625306	206.65
28.15	3.25	11.99	28.58	1.27	714310	236.07
28.51	3.25	12.31	28.7	1.54	774362	255.91

The table 5.11 below gives the power consumption data for partial load operation with and without VFD.

**Table 5.11 Power consumption for partial load operation (with & without VFD)**

Power consumed without VFD (kW)	Power consumed with VFD (kW)	Measured Power consumed per ton of refrigeration load without VFD (kW/TR)	Measured Power consumed per ton of refrigeration load with VFD (kW/TR)	Hourly savings due to VFD operation (kW)
173.14	127.25	0.94	0.69	45.89
203.93	149.87	0.94	0.69	54.06
210.02	154.35	0.94	0.69	55.67
225.4	165.65	0.94	0.69	59.75
254.75	187.22	0.94	0.69	67.53
160.75	118.14	0.94	0.69	42.61
176	129.35	0.94	0.69	46.65
193.69	142.35	0.94	0.69	51.34
221.26	162.61	0.94	0.69	58.65
239.87	176.28	0.94	0.69	63.59

Average hourly power savings = 57.6 kW

Average yearly energy savings (250 days x 10 hours) = 136435 kWh  
 Average yearly monetary savings (BDT. 6.00/kWh) = BDT. 8, 18,610 /-

**Example 5.7**

The measured values of a 20 TR package air conditioning plant are given below: Average air velocity across suction side filter: 2.5 m/s

Cross Sectional area of suction: 1.2 m<sup>2</sup>

Inlet air = Dry Bulb: 20°C, Wet Bulb: 14 °C, Enthalpy: 9.37 kcal/kg

Outlet air = Dry Bulb: 12.7 °C, Wet Bulb: 11.3 °C; Enthalpy: 7.45 kcal/kg

Specific volume of air: 0.85 m<sup>3</sup>/kg

Power drawn: by compressor: 10.69 kW

by Pump: 4.86 kW

by Cooling tower fan: 0.87 kW

**Calculate:**

- a) Air Flow rate in m<sup>3</sup>/hr
- b) Cooling effect delivered in kW
- c) Specific power consumption of compressor in kW/TR
- d) Overall kW/TR
- e) Energy Efficiency Ratio in kW/kW

**Solution**

$$\text{Air flow rate} = 2.5 \times 1.2 = 3 \text{ m}^3/\text{sec} = 10800 \text{ m}^3/\text{hr}$$

$$\begin{aligned} \text{Cooling Effect delivered} &= [(9.37 - 7.45) \times 10800] / (0.85 \times 3024) \\ &= 8.07 \text{ TR} = 28.32 \text{ kW} \end{aligned}$$

$$\text{Compressor kW/TR} = 10.69/8.07 = 1.32$$

$$\text{Overall kW/TR} = (10.69+4.86+0.87)/8.07 = 2.04$$

$$\text{Energy Efficiency Ratio (EER) in kW/kW} = 28.32/10.69 = 2.65$$

## Chapter 6: Pumps And Pumping System

### 6.1 Pump Types

Pumps come in a variety of sizes for a wide range of applications. They can be classified according to their basic operating principle as dynamic or displacement pumps. Dynamic pumps can be sub-classified as centrifugal and special effect pumps. Displacement pumps can be sub-classified as rotary or reciprocating pumps.

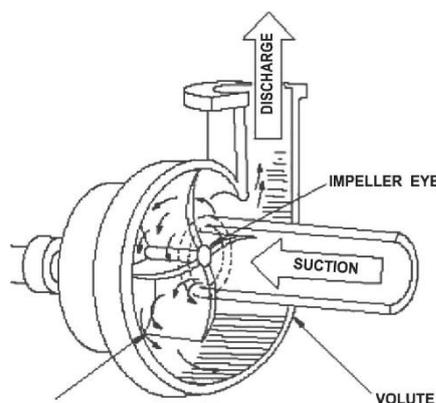
In principle, any liquid can be handled by any of the pump designs. Where different pump designs could be used, the centrifugal pump is generally the most economical followed by rotary and reciprocating pumps. Although, positive displacement pumps are generally more efficient than centrifugal pumps, the benefit of higher efficiency tends to be offset by increased maintenance costs.

Since, worldwide, centrifugal pumps account for the majority of electricity used by pumps, the focus of this chapter is on centrifugal pump

### Centrifugal Pumps

A centrifugal pump (Figure 6.1) is of a very simple design. The two main parts of the pump are the impeller and the diffuser. Impeller, which is the only moving part, is attached to a shaft and driven by a motor. Impellers are generally made of bronze, polycarbonate, cast iron, stainless steel as well as other materials. The diffuser (also called as volute) houses the impeller and captures and directs the water off the impeller.

Water enters the center (eye) of the impeller and exits the impeller with the help of centrifugal force. As water leaves the eye of the impeller a low-pressure area is created, causing more water to flow into the eye. Atmospheric pressure and centrifugal force cause this to happen. Velocity is developed as the water flows through the impeller spinning at high speed. The water velocity is collected by the diffuser and converted to pressure by specially designed passageways that direct the flow to the discharge of the pump, or to the next impeller should the pump have a multi-stage configuration.



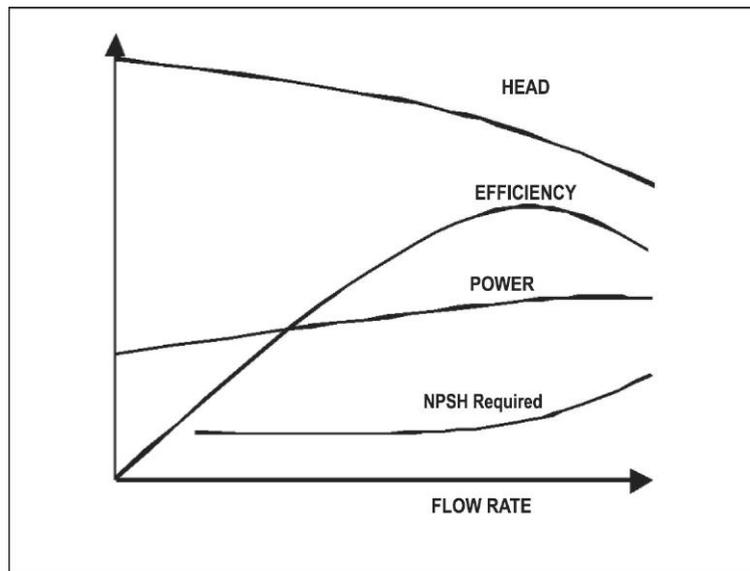
**Figure 6.1 Centrifugal Pump**

The pressure (head) that a pump will develop is in direct relationship to the impeller diameter, the number of impellers, the size of impeller eye, and shaft speed. Capacity is

determined by the exit width of the impeller. The head and capacity are the main factors, which affect the horsepower size of the motor to be used. The more the quantity of water to be pumped, the more energy is required.

A centrifugal pump is not positive acting, it will not pump the same volume always. The greater the depth of the water, the lesser is the flow from the pump. Also, when it pumps against increasing pressure, the less it will pump. For these reasons it is important to select a centrifugal pump that is designed to do a particular job.

Since the pump is a dynamic device, it is convenient to consider the pressure in terms of head i.e. meters of liquid column. The pump generates the same head of liquid whatever the density of the liquid being pumped. The actual contours of the hydraulic passages of the impeller and the casing are extremely important, in order to attain the highest efficiency possible. The standard convention for centrifugal pump is to draw the pump performance curves showing Flow on the horizontal axis and Head generated on the vertical axis. Efficiency, Power & NPSH Required (described later), are also all conventionally shown on the vertical axis, plotted against Flow, as illustrated in Figure 6.2.



**Figure 6.2 Pump Performance**

Given the significant amount of electricity attributed to pumping systems, even small improvements in pumping efficiency could yield very significant savings of electricity. The pump is among the most inefficient of the components that comprise a pumping system, including the motor, transmission drive, piping and valves.

**Hydraulic Power, Pump Shaft Power and Motor Input Power**

$$\text{Hydraulic Power } P_h = Q \text{ (m}^3/\text{s)} \times \text{Total Differential head, } h_d - h_s \text{ (m)} \times \rho \text{ (kg/m}^3) \times g \text{ (m/s}^2) / 1000$$

Where  $h_d$  -discharge head,  $h_s$ -suction head,  $\rho$ -density of the liquid,  $g$ - acceleration due to gravity

$$\text{Pump Shaft Power } P_s = \text{Hydraulic power, } P_h / \text{Pump Efficiency, } \eta_{\text{Pump}}$$

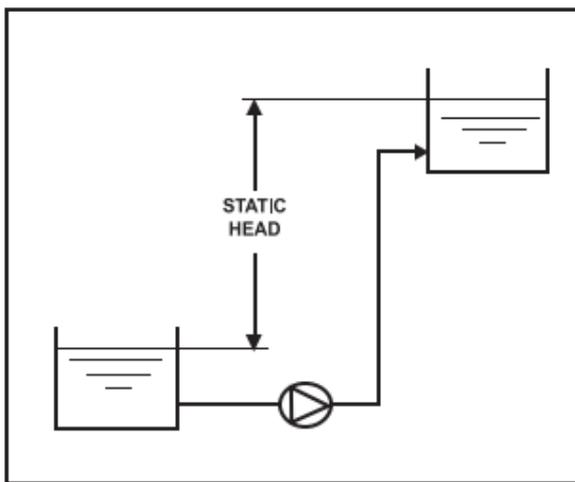
$$\text{Motor Input Power} = \text{Pump shaft power } P_s / \text{Motor Efficiency, } \eta_{\text{Motor}}$$

## 6.2 System Characteristics

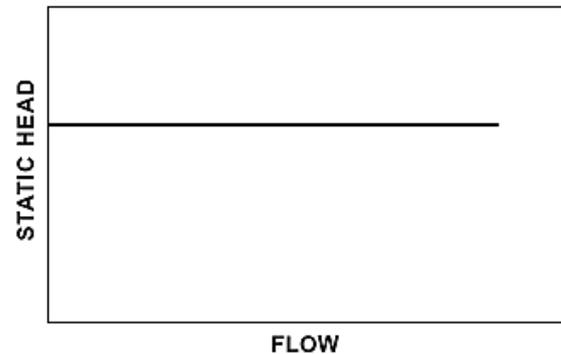
In a pumping system, the objective, in most cases, is either to transfer a liquid from a source to a required destination, e. g. filling a high level reservoir or to circulate liquid around a system e. g. as a means of heat transfer in heat exchanger.

A pressure is needed to make the liquid flow at the required rate and this must overcome head 'losses' in the system. Losses are of two types: static and friction head.

Static head is simply the difference in height of the supply and destination reservoirs, as in Figure 6.3. In this illustration, flow velocity in the pipe is assumed to be very small. Another example of a system with only static head is pumping into a pressurized vessel with short pipe runs. Static head is independent of flow and graphically would be shown as in Figure 6.4.

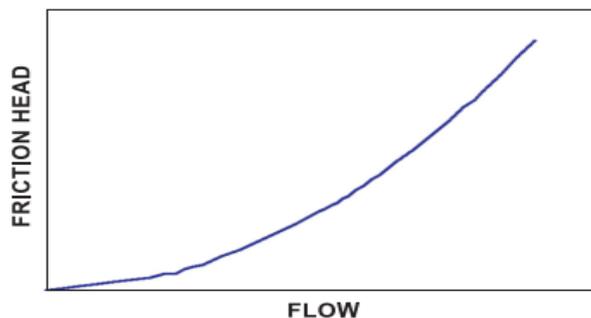


*Figure 6.3 Static Head*



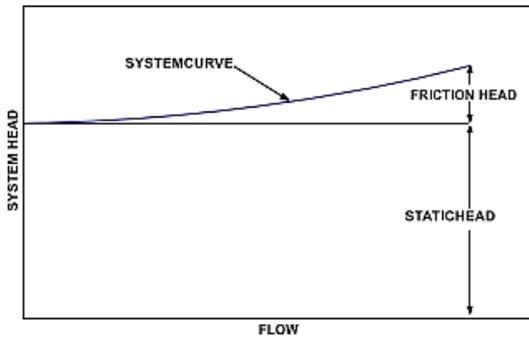
*Figure 6.4 Static Head vs. Flow*

Friction head (sometimes called dynamic head loss) is the friction loss, on the liquid being moved, in pipes, valves and equipment in the system. Friction tables are universally available for various pipe fittings and valves. These tables show friction loss per 100 feet (or meters) of a specific pipe size at various flow rates. In case of fittings, friction is stated as an equivalent length of pipe of the same size. The friction losses are proportional to the square of the flow rate. A closed loop circulating system without a surface open to atmospheric pressure, would exhibit only friction losses and would have a system friction head loss vs. flow curve as Figure 6.5.

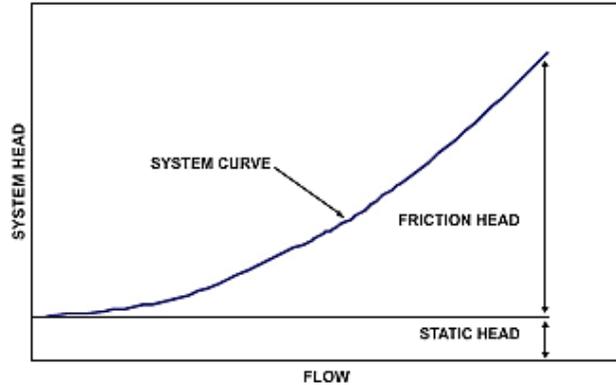


*Figure 6.5 Friction Head vs. Flow*

Most systems have a combination of static and friction head and the system curves for two cases are shown in Figures 6.6 and 6.7. The ratio of static to friction head over the operating range influences the benefits achievable from variable speed drives which shall be discussed later



*Figure 6.6 System with High Static Head*

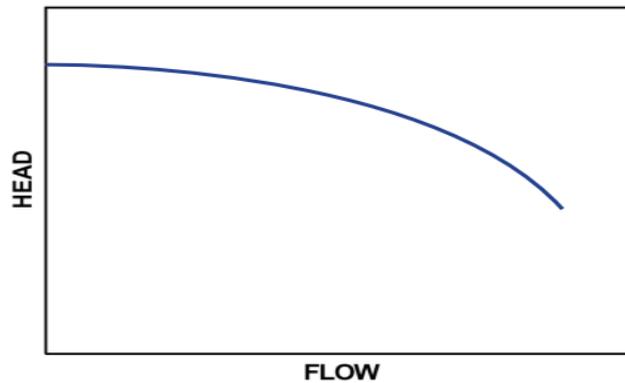


*Figure 6.7 System with Low Static Head*

Static head is a characteristic of the specific installation and reducing this head where this is possible generally helps both the cost of the installation and the cost of pumping the liquid. Friction head losses must be minimized to reduce pumping cost, but after eliminating unnecessary pipe fittings and length, further reduction in friction head will require larger diameter pipe, which adds to capital cost.

### 6.3 Pump Curves

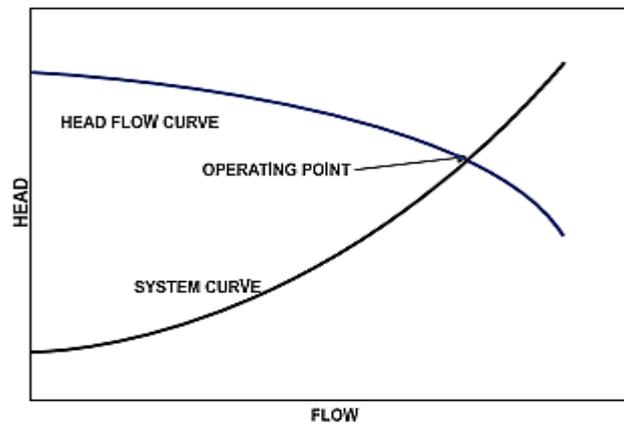
The performance of a pump can be expressed graphically as head against flow rate. The centrifugal pump has a curve where the head falls gradually with increasing flow. This is called the pump characteristic curve (Head — Flow curve). See Figure 6.8.



*Figure 6.8 Head-Flow Curve*

## Pump operating point

When a pump is installed in a system the effect can be illustrated graphically by superimposing pump and system curves. The operating point will always be where the two curves intersect. Figure: 6.9.



*Figure 6.9 Pump Operating Point*

If the actual system curve is different in reality to that calculated, the pump will operate at a flow and head different to that expected. For a centrifugal pump, an increasing system resistance will reduce the flow, eventually to zero, but the maximum head is limited as shown. Even so, this condition is only acceptable for a short period without causing problems. An error in the system curve calculation is also likely to lead to a centrifugal pump selection, which is less than optimal for the actual system head losses. Adding safety margins to the calculated system curve to ensure that a sufficiently large pump is selected will generally result in installing an oversized pump, which will operate at an excessive flow rate or in a throttled condition, which increases energy usage and reduces pump life.

## 6.4 Factors Affecting Pump Performance

### Matching Pump and System Head-flow Characteristics

Centrifugal pumps are characterized by the relationship between the flow rate ( $Q$ ) they produce and the pressure ( $H$ ) at which the flow is delivered. Pump efficiency varies with flow and pressure, and it is highest at one particular flow rate.

The Figure 6.10 below shows a typical vendor-supplied head-flow curve for a centrifugal pump. Pump head-flow curves are typically given for clear water. The choice of pump for a given application depends largely on how the pump head-flow characteristics match the requirement of the system downstream of the pump.

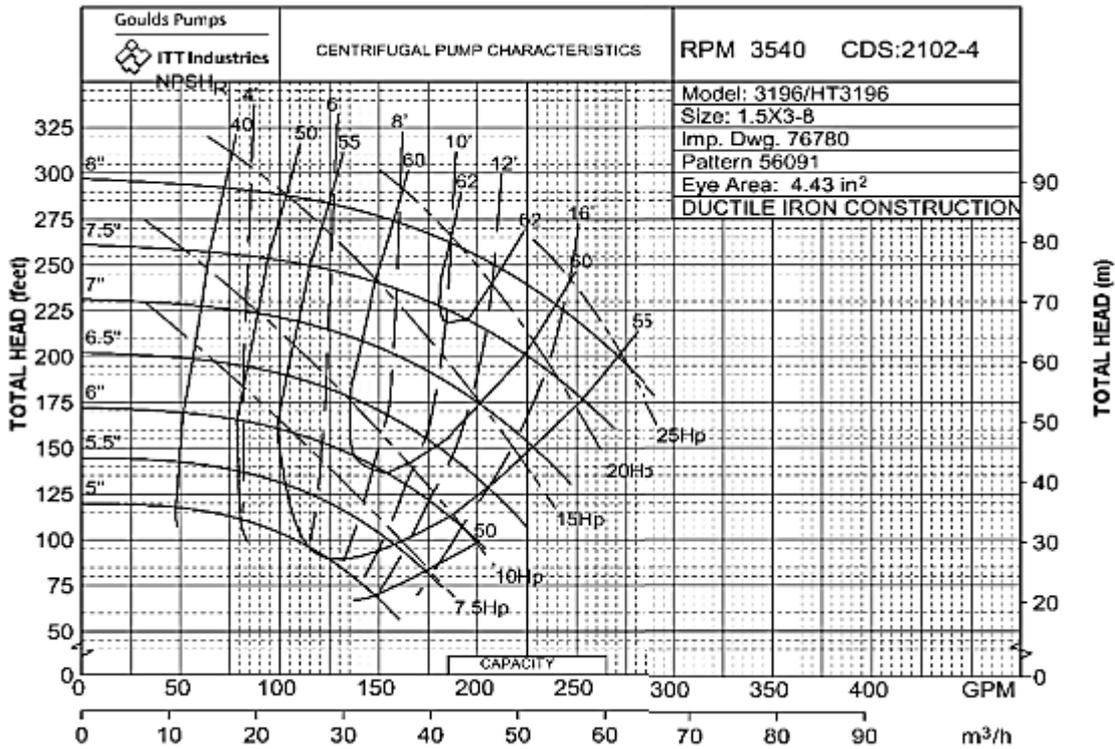


Figure 6.10 Typical Centrifugal Pump Performance Curve

### Effect of over sizing the pump

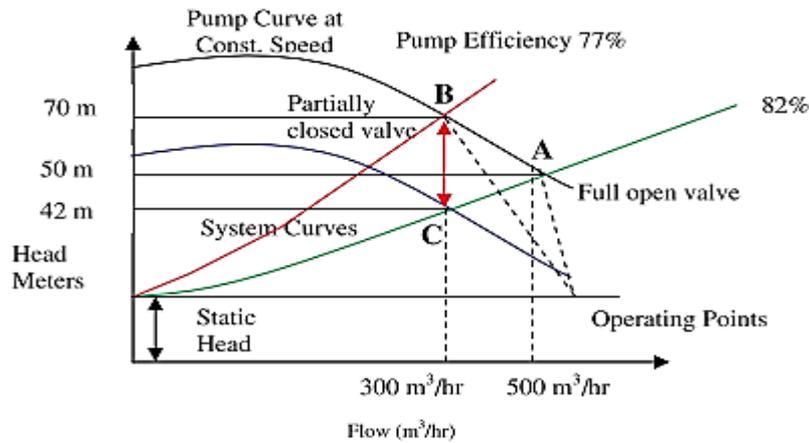
As mentioned earlier, pressure losses to be overcome by the pumps are functions of flow -the system characteristics are also quantified in the form of head-flow curves. The system curve is basically a plot of system resistance i.e. head to be overcome by the pump versus various flow rates. The system curves change with the physical configuration of the system; for example, the system curves depends upon height or elevation, diameter and length of piping, number and type of fittings and pressure drops across various equipment - say a heat exchanger.

A pump is selected based on how well the pump curve and system head-flow curves match. The pump operating point is identified as the point, where the system curve crosses the pump curve when they are superimposed on each other. The Figure 6.11 shows the effect on system curve with throttling.

In the system under consideration, water has to be first lifted to a height-this represents the static head.

Then, we make a system curve, considering the friction and pressure drops in the system-this is shown as the green curve.

Suppose, we have estimated our operating conditions as 500 m<sup>3</sup>/hr flow and 50 m head, we will chose a pump curve which intersects the system curve (Point A) at the pump's best efficiency point (BEP).



**Figure 6.11** Effect on System Curve with Throttling

But, in actual operation, we find that 300 m<sup>3</sup>/hr is sufficient. The reduction in flow rate has to be effected by a throttle valve. In other words, we are introducing an artificial resistance in the system.

Due to this additional resistance, the frictional part of the system curve increases and thus the new system curve will shift to the left - this is shown as the red curve.

So the pump has to overcome additional pressure in order to deliver the reduced flow. Now, the new system curve will intersect the pump curve at point B. The revised parameters are 300 m<sup>3</sup>/hr at 70 m head. The red double arrow line shows the additional pressure drop due to throttling.

It may be noted that the best efficiency point has shifted from 82% to 77% efficiency. So it is actually needed to operate at point C, which is 300 m<sup>3</sup>/hr on the original system curve. The head required at this point is only 42 meters.

Hence a new pump is needed, which will operate with its best efficiency point at C. But there are other simpler options rather than replacing the pump. The speed of the pump can be reduced or the existing impeller can be trimmed (or new lower size impeller). The blue pump curve represents either of these options.

### Energy loss in throttling

Consider a case (see Figure 6.12) where we need to pump 68 m<sup>3</sup>/hr of water at 47 m head. The pump characteristic curves (A. . .E) for a range of pumps are given in the Figure 6.12.

If we select pump E, then the efficiency is 60%

$$\text{Hydraulic Power} = Q \text{ (m}^3\text{/s)} \times \text{Total Differential head, } h_d - h_s \text{ (m)} \times \rho \text{ (kg/m}^3\text{)} \times g \text{ (m/s}^2\text{)} / 1000$$

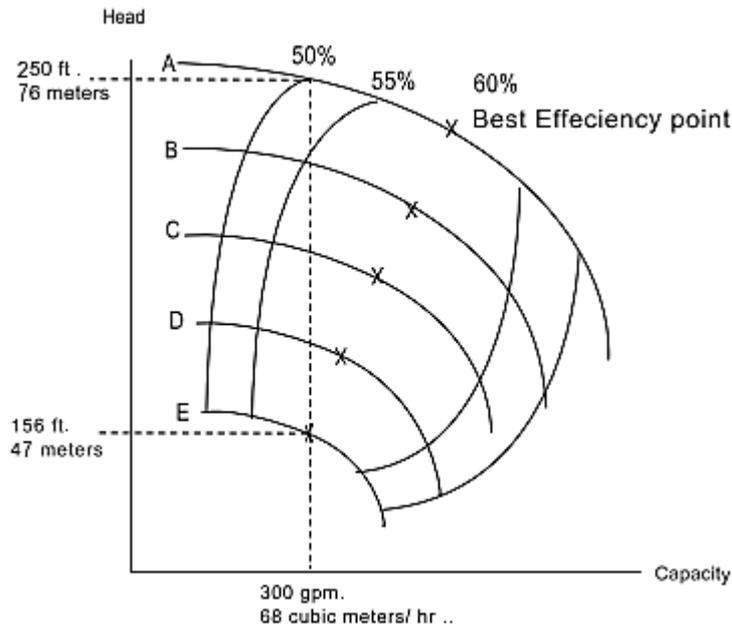
$$= \frac{(68/3600) \times 47 \times 1000 \times 9.81}{1000}$$

$$= 8.7 \text{ kW}$$

$$\text{Shaft Power} = 8.7 / 0.60 = 14.5 \text{ kW}$$

$$\text{Motor Power} = 14.5 / 0.9 = 16.1 \text{ kW (considering a motor efficiency of 90\%)}$$

If we select pump A, then the efficiency is 50% (drop from earlier 60%)



**Figure 6.12 Pump Characteristic Curves**

Obviously, this is an oversize pump. Hence, the pump has to be throttled to achieve the desired flow. Throttling increases the head to be overcome by the pump. In this case, head is 76 meters.

$$\begin{aligned} \text{Hydraulic Power} &= Q \text{ (m}^3\text{/s)} \times \text{Total Differential head, } h_d - h_s \text{ (m)} \times \rho \text{ (kg/m}^3\text{)} \times g \text{ (m/s}^2\text{)} / 1000 \\ &= \frac{(68/3600) \times 76 \times 1000 \times 9.81}{1000} \\ &= 14 \text{ kW} \end{aligned}$$

$$\begin{aligned} \text{Shaft Power} &= 14 / 0.50 = 28 \text{ kW} \\ \text{Motor Power} &= 28 / 0.9 = 31 \text{ kW (considering a motor efficiency of 90\%)} \end{aligned}$$

Extra energy used = 8000 hrs/yr X 14.9 = 1, 19,200 kWh/annum = BDT 6,55,600/annum (BDT 5.50 per kWh)

In this example, the extra cost of the electricity is more than the cost of a new pump.

## 6.5 Efficient Pumping System Operation

To understand a pumping system, one must realize that all of its components are interdependent. When examining or designing a pump system, the process demands must first be established and most energy efficiency solution introduced. For example, does the flow rate have to be regulated continuously or in steps? Can on—off batch pumping be used? What is the flow rates needed and how are they distributed in time?

The first step to achieve energy efficiency in pumping system is to target the end-use. A plant water balance would establish usage pattern and highlight areas where water consumption can be reduced or optimized. Good water conservation measures, alone, may eliminate the need for some pumps.

Once flow requirements are optimized, then the pumping system can be analyzed for energy conservation opportunities. Basically this means matching the pump to requirements by adopting proper flow control strategies. Common symptoms that indicate opportunities for energy efficiency in pumps are given in the Table 6.1.

**Table 6.1 Symptoms that indicate potential opportunity**

Symptom	Likely Reason	Best Solutions
Throttle valve-controlled systems	Oversized pump	Trim impeller, smaller impeller, variable speed drive, two speed drive, lower RPM
Bypass line (partially or completely) open	Oversized pump	Trim impeller, smaller impeller, open variable speed drive, two speed drive, lower RPM
Multiple parallel pump system with the same number of pumps always operating	Pump use not monitored or controlled	Install controls
Constant pump operation in batch environment	Wrong system design	On-off controls
High maintenance cost (seals, bearings)	Pump operated far away from BEP	Match pump capacity with system requirement

**Effect of speed variation**

As stated above, a centrifugal pump is a dynamic device with the head generated from a rotating impeller. There is therefore a relationship between impeller peripheral velocity and generated head. Peripheral velocity is directly related to shaft rotational speed, for a fixed impeller diameter and so varying the rotational speed has a direct effect on the performance of the pump. All the parameters shown in figure 6.2 will change if the speed is varied and it is important to have an appreciation of how these parameters vary in order to safely control a pump at different speeds. The equations relating rotodynamic pump performance parameters of flow, head and power absorbed, to speed are known as the Affinity Laws:

$$Q \propto N$$

$$H \propto N^2$$

$$P \propto N^3$$

Where,

Q = Flow rate

H = Head

P = Power absorbed

N = Rotating speed

Efficiency is essentially independent of speed

Flow: Flow is proportional to the speed

$$Q_1 / Q_2 = N_1 / N_2$$

Example:  $100 / Q_2 = 3000 / 1500$

$$Q_2 = 50 \text{ m}^3/\text{hr}$$

Head: Head is proportional to the square of speed

$$H_1 / H_2 = N_1^2 / N_2^2$$

Example:  $100 / H_2 = 3000^2 / 1500^2$

$$H_2 = 25 \text{ m}$$

Power (kW): Power is proportional to the cube of speed

$$\text{kW}_1 / \text{kW}_2 = N_1^3 / N_2^3$$

### Example 6.1

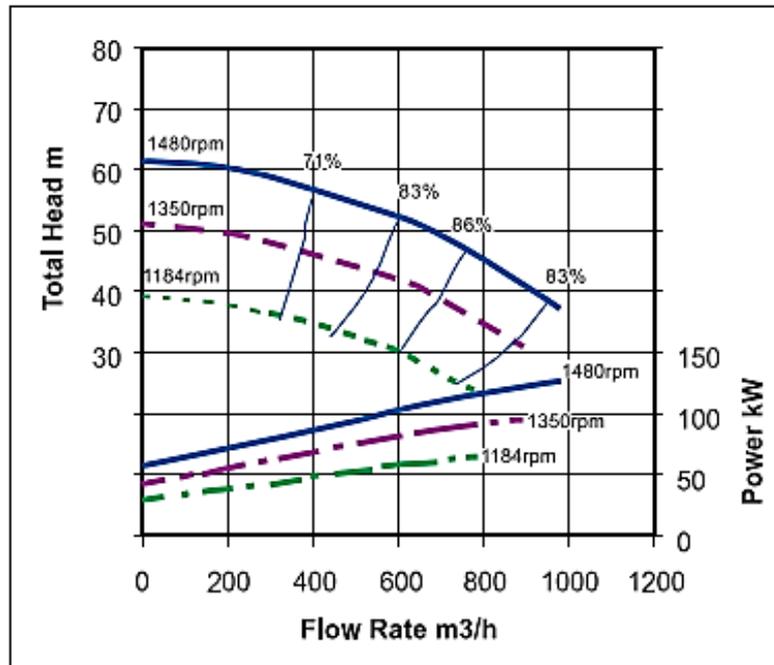
$$40 / \text{KW}_2 = 3000^3 / 1500^3$$

$$\text{KW}_2 = 5 \text{ kW}$$

As can be seen from the above laws, reduction in speed will result in considerable reduction in power consumption. This forms the basis for energy conservation in centrifugal pumps with varying flow requirements. The implication of this can be better understood as shown in an example of a centrifugal pump in Figure 6.13 below.

Points of equal efficiency on the curves for three different speeds are joined to make the iso-efficiency lines, showing that efficiency remains constant over small changes of speed providing the pump continues to operate at the same position related to its best efficiency point (BEP).

The affinity laws give a good approximation of how pump performance curves change with speed but in order to obtain the actual performance of the pump in a system, the system curve also has to be taken into account.



*Figure 6.13 Example of Speed Variation Effecting Centrifugal Pump Performance*

### Effects of impeller diameter change

Changing the impeller diameter gives a proportional change in peripheral velocity, so it follows that there are equations, similar to the affinity laws, for the variation of performance with impeller diameter D:

$$Q \propto D$$

$$H \propto D^2$$

$$P \propto D^3$$

Efficiency varies when the diameter is changed within a particular casing. Note the difference in iso-efficiency lines in Figure 6.14 compared with Figure 6.13. The relationships shown here apply to the case for changing only the diameter of an impeller within a fixed casing geometry, which is a common practice for making small permanent adjustments to the performance of a centrifugal pump. Diameter changes are generally limited to reducing the diameter to about 75% of the maximum, i.e. a head reduction to about 50%. Beyond this, efficiency and NPSH are badly affected. However speed change can be used over a wider range without seriously reducing efficiency. For example reducing the speed by 50% typically results in a reduction of efficiency by 1 or 2 percentage points. The reason for the small loss of efficiency with the lower speed is that mechanical losses in seals and bearings, which generally represent <5% of total power, are proportional to speed, rather than speed cubed. It should be noted that if the change in diameter is more than about 5%, the accuracy of the squared and cubic relationships can

fall off and for precise calculations, the pump manufacturer's performance curves should be referred to.

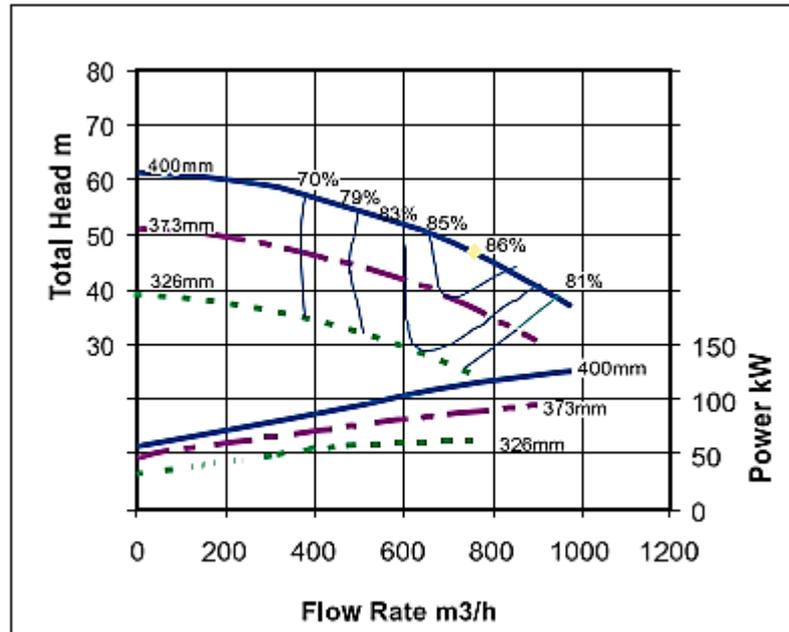


Figure 6.14 Example: Impeller Diameter Reduction on Centrifugal Pump Performance

The illustrated curves are typical of most centrifugal pump types. Certain high flow, low head pumps have performance curve shapes somewhat different and have a reduced operating region of flows. This requires additional care in matching the pump to the system when changing speed and diameter.

### Pump suction performance

Liquid entering the impeller eye turns and is split into separate streams by the leading edges of the impeller vanes, an action which locally drops the pressure below that in the inlet pipe to the pump. If the incoming liquid is at a pressure with insufficient margin above its vapour pressure then vapour cavities or bubbles appear along the impeller vanes just behind the inlet edges. This phenomenon is known as cavitation and has three undesirable effects:

1. The collapsing cavitation bubbles can erode the vane surface, especially when pumping water-based liquids.
2. Noise and vibration are increased, with possible shortened seal and bearing life.
3. The cavity areas will initially partially choke the impeller passages and reduce the pump performance. In extreme cases, total loss of pump developed head occurs.

The value, by which the liquid pressure at the eye of pump exceeds the liquid vapour pressure, is expressed as a head of liquid and referred to as Net Positive Suction Head Available - (NPSHA). This is a characteristic of the system design. The value of NPSH needed at the pump suction to prevent the pump from cavitation is known as NPSH Required - (NPSHR). This is a characteristic of the pump design.

The three undesirable effects of cavitation described above begin at different values of NPSHA and generally there will be cavitation erosion before there is a noticeable loss of

pump head. However for a consistent approach, manufacturers and industry standards, usually define the onset of cavitation as the value of NPSHR when there is a head drop of 3% compared with the head with cavitation free performance. At this point cavitation is present and prolonged operation at this point will usually lead to damage. It is usual therefore to apply a margin by which NPSHA should exceed NPSHR.

As would be expected, the NPSHR increases as the flow through the pump increases, see fig 6.2. In addition, as flow increases in the suction pipework, friction losses also increase, giving a lower NPSHA at the pump suction, both of which give a greater chance that cavitation will occur. NPSHR also varies approximately with the square of speed in the same way as pump head and conversion of NPSHR from one speed to another can be made using the following equations

$$Q \propto N$$

$$NPSHR \propto N^2$$

It should be noted however that at very low speeds there is a minimum NPSHR plateau, NPSHR does not tend to zero at zero speed. It is therefore essential to carefully consider NPSH in variable speed pumping.

## 6.6 Flow Control Strategies

### Pump control by varying speed

To understand how the speed variation changes the duty point, the pump and system curves are over-laid. Two systems are considered, one with only friction loss and another where static head is high in relation to friction head. It will be seen that the benefits are different.

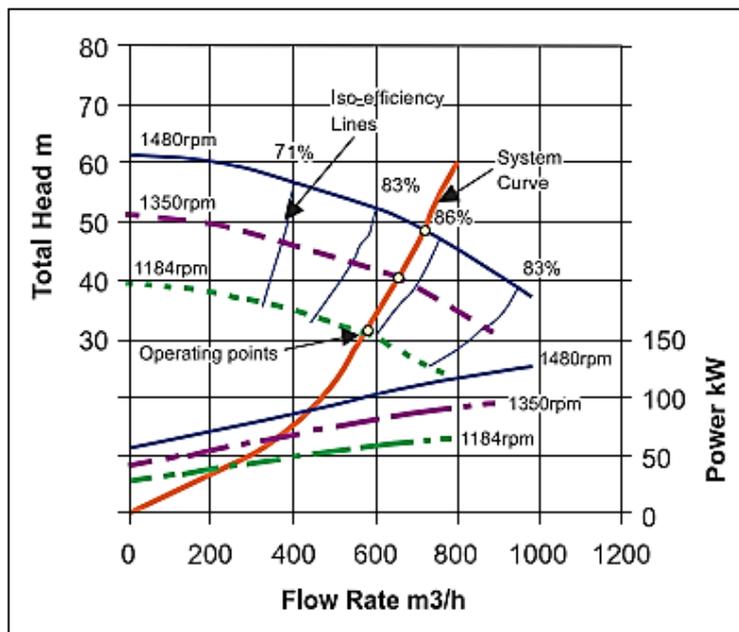
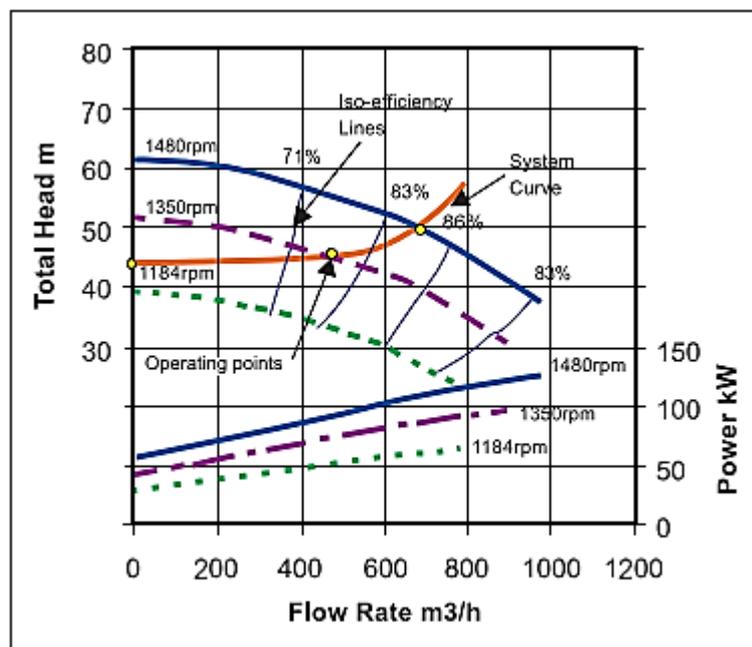


Figure 6.15: Example of the Effect of Pump Speed Change in a System with Only Friction Loss

In Figure 6.15, reducing speed in the friction loss system moves the intersection point on the system curve along a line of constant efficiency. The operating point of the pump,

relative to its best efficiency point, remains constant and the pump continues to operate in its ideal region. The affinity laws are obeyed which means that there is a substantial reduction in power absorbed accompanying the reduction in flow and head, making variable speed the ideal control method for systems with friction loss.

In a system where static head is high, as illustrated in Figure 6.16, the operating point for the pump moves relative to the lines of constant pump efficiency when the speed is changed. The reduction in flow is no longer proportional to speed. A small turn down in speed could give a big reduction in flow rate and pump efficiency, which could result in the pump operating in a region where it could be damaged if it ran for an extended period of time even at the lower speed. At the lowest speed illustrated, (1184 rpm), the pump does not generate sufficient head to pump any liquid into the system, i.e. pump efficiency and flow rate are zero and with energy still being input to the liquid, the pump becomes a water heater and damaging temperatures can quickly be reached.



**Figure 6.16: Example for the Effect of Pump Speed Change with a System with High Static Head**

The drop in pump efficiency during speed reduction in a system with static head, reduces the economic benefits of variable speed control. There may still be overall benefits but economics should be examined on a case-by-case basis. Usually it is advantageous to select the pump such that the system curve intersects the full speed pump curve to the right of best efficiency, in order that the efficiency will first increase as the speed is reduced and then decrease. This can extend the useful range of variable speed operation in a system with static head. The pump manufacturer should be consulted on the safe operating range of the pump.

It is relevant to note that flow control by speed regulation is always more efficient than by control valve. In addition to energy savings there could be other benefits of lower speed. The hydraulic forces on the impeller, created by the pressure profile inside the pump casing, reduce approximately with the square of speed. These forces are carried by the pump bearings and so reducing speed increases bearing life. It can be shown that for a centrifugal pump, bearing life is inversely proportional to the 7th power of speed. In

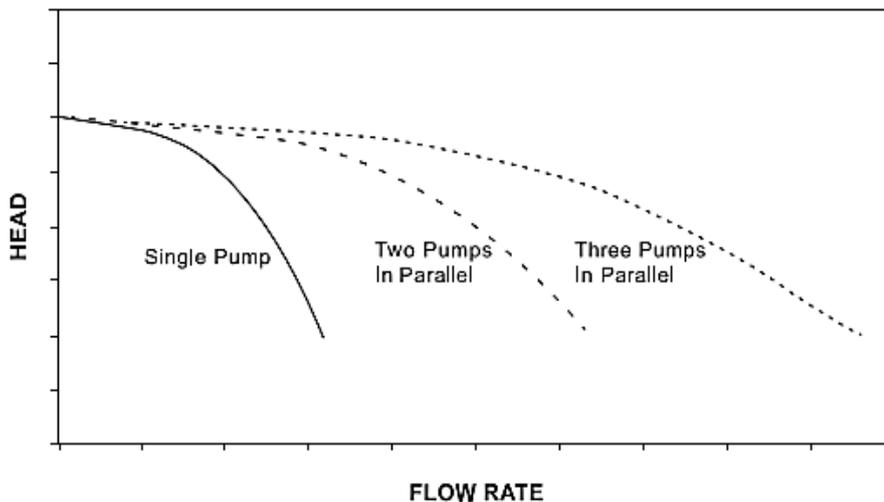
addition, vibration and noise are reduced and seal life is increased providing the duty point remains within the allowable operating range.

The corollary to this is that small increases in the speed of a pump significantly increase power absorbed, shaft stress and bearing loads. It should be remembered that the pump and motor must be sized for the maximum speed at which the pump set will operate. At higher speed the noise and vibration from both pump and motor will increase, although for small increases the change will be small. If the liquid contains abrasive particles, increasing speed will give a corresponding increase in surface wear in the pump and pipework.

The effect on the mechanical seal of the change in seal chamber pressure should be reviewed with the pump or seal manufacturer, if the speed increase is large. Conventional mechanical seals operate satisfactorily at very low speeds and generally there is no requirement for a minimum speed to be specified, however due to their method of operation, gas seals require a minimum peripheral speed of 5 m/s

### **Pumps in parallel switched to meet demand**

Another energy efficient method of flow control, particularly for systems where static head is a high proportion of the total, is to install two or more pumps to operate in parallel. Variation of flow rate is achieved by switching on and off additional pumps to meet demand. The combined pump curve is obtained by adding the flow rates at a specific head. The head/flow rate curves for two and three pumps are shown in Figure 6.17.

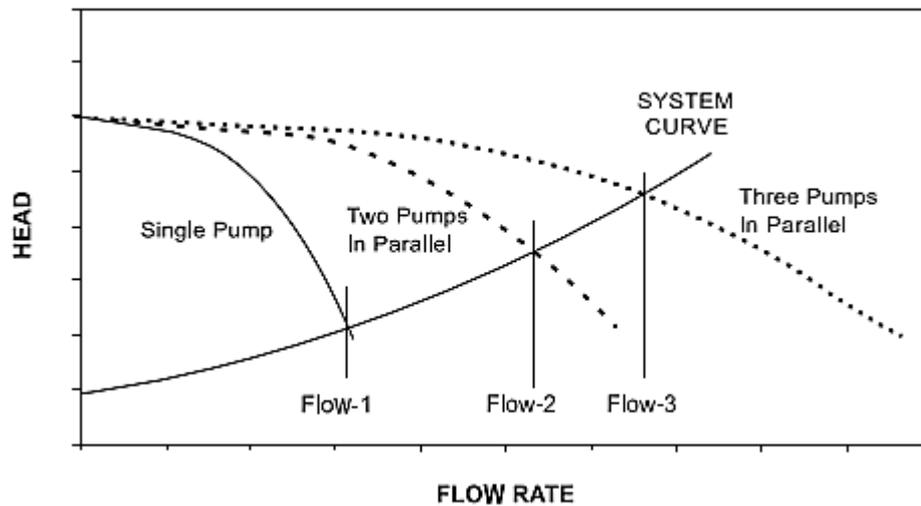


*Figure 6.17: Typical Head-Flow Curves for Pumps in Parallel*

The system curve is usually not affected by the number of pumps that are running. For a system with a combination of static and friction head loss, it can be seen, in Figure 6.18, that the operating point of the pumps on their performance curves moves to a higher head and hence lower flow rate per pump, as more pumps are started. It is also apparent that the flow rate with two pumps running is not double that of a single pump. If the system head were only static, then flow rate would be proportional to the number of pumps operating.

It is possible to run pumps of different sizes in parallel provided their closed valve heads are similar. By arranging different combinations of pumps running together, a larger number of different flow rates can be provided into the system.

Care must be taken when running pumps in parallel to ensure that the operating point of the pump is controlled within the region deemed as acceptable by the manufacturer. It can be seen from Figure 6.18 that if 1 or 2 pumps were stopped then the remaining pump(s) would operate well out along the curve where NPSH is higher and vibration level increased, giving an increased risk of operating problems.



*Figure 6.18: Typical Head-Flow Curves for Pumps in Parallel, With System Curve Illustrated*

### **Stop/start control**

In this control method, the flow is controlled by switching pumps on or off. It is necessary to have a storage capacity in the system e.g. a reservoir, a wet well, an elevated tank or an accumulator type pressure vessel. The storage can provide a steady flow to the system with an intermittent operating pump. When the pump runs, it does so at the chosen (presumably optimum) duty point and when it is off, there is no energy consumption. If intermittent flow, stop/start operation and the storage facility are acceptable, this is an effective approach to minimize energy consumption.

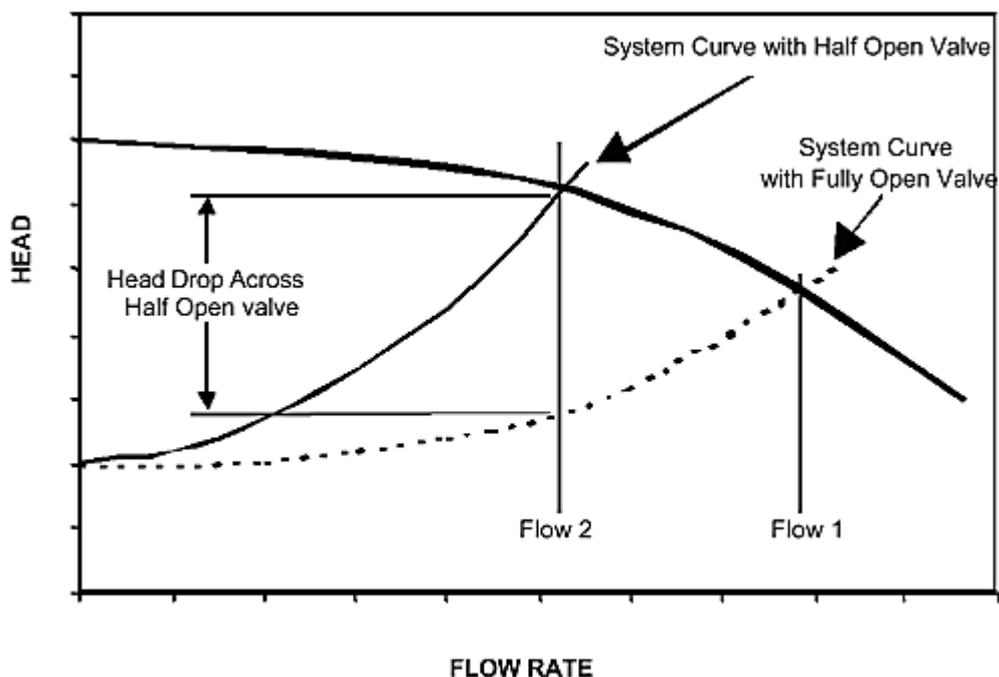
The stop/start operation causes additional loads on the power transmission components and increased heating in the motor. The frequency of the stop/start cycle should be within the motor design criteria and checked with the pump manufacturer.

It may also be used to benefit from “off peak” energy tariffs by arranging the run times during the low tariff periods.

To minimize energy consumption with stop/start control, it is better to pump at as low flow rate as the process permits. This minimizes friction losses in the pipe and an appropriately small pump can be installed. For example, pumping at half the flow rate for twice as long can reduce energy consumption to a quarter. It means it is beneficial to run one pump at full capacity continuously rather than running two pumps at a time with a stop/start control.

## Flow control valve

With this control method, the pump runs continuously and a valve in the pump discharge line is opened or closed to adjust the flow to the required value.



*Figure 6.19: Control of Pump Flow by Changing System Resistance Using 3 Valve.*

To understand how the flow rate is controlled, see Figure 6.19. With the valve fully open, the pump operates at “Flow 1”. When the valve is partially closed, it introduces an additional friction loss in the system which is proportional to flow squared. The new system curve cuts the pump curve at “Flow 2” which is the new operating point. The head difference between the two curves is the pressure drop across the valve.

It is usual practice with valve control to have the valve 10% shut even at maximum flow. Energy is therefore wasted overcoming the resistance through the valve at all flow conditions. There is some reduction in pump power absorbed at the lower flow rate (see Figure 6.19), but the flow multiplied by the head drop across the valve, is wasted energy. It should also be noted that while the pump will accommodate changes in its operating point as far as it is able within its performance range, it can be forced to operate high on the curve where its efficiency is low and its reliability is affected.

Maintenance cost of control valves can be high, particularly on corrosive and solids-containing liquids. Therefore, the lifetime cost could be unnecessarily high.

### **By-pass control**

With this control approach, the pump runs continuously at the maximum process demand duty with a permanent by-pass line attached to the outlet. When a lower flow is required the surplus liquid is bypassed and returned to the supply source.

An alternative configuration may have a tank supplying a varying process demand, which is kept full by a fixed duty pump running at the peak flow rate. Most of the time, the tank overflows and recycles back to the pump suction. This is even less energy efficient than a control valve because there is no reduction in power consumption with reduced process demand.

The small by-pass line sometimes installed to prevent a pump running at zero flow is not a means of flow control, but required for the safe operation of the pump.

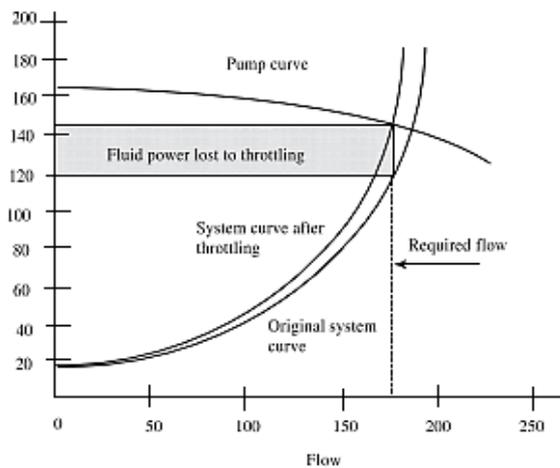
### Fixed Flow reduction

### Impeller trimming

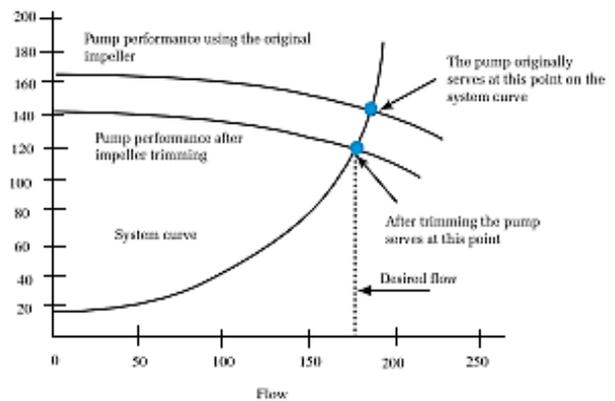
Impeller trimming refers to the process of machining the diameter of an impeller to reduce the energy added to the system liquid.

Impeller trimming offers a useful correction to pumps that, through overly conservative design practices or changes in system loads are oversized for their application.

Trimming an impeller provides a level of correction below buying a smaller impeller from the pump manufacturer. But in many cases, the next smaller size impeller is too small for the pump load. Also, smaller impellers may not be available for the pump size in question and impeller trimming is the only practical alternative short of replacing the entire pump/motor assembly. (See Figures 6.20 & 6.21 for before and after impeller trimming).



*Figure 6.20 Before Impeller Trimming*



*Figure 6.21 After Impeller Trimming*

Impeller trimming reduces tip speed which in turn directly lowers the amount of energy imparted to the system liquid and lowers both the flow and pressure generated by the pump.

The Affinity Laws, which describe centrifugal pump performance, provide a theoretical relationship between impeller size and pump output (assuming constant pump speed):

Where:

Q = flow

H = head  
P = power  
D = diameter of impeller

Subscript 1 = original pump,  
Subscript 2 = pump after impeller trimming

$$Q_2 = \frac{D_2}{D_1} \times Q_1$$

$$H_2 = \left[ \frac{D_2}{D_1} \right]^2 \times H_1$$

$$P_2 = \left[ \frac{D_2}{D_1} \right]^3 \times P_1$$

Trimming an impeller changes its operating efficiency and the non-linearities of the Affinity Laws with respect to impeller machining complicate the prediction of pump performance. Consequently, impeller diameters are rarely reduced below 75 percent of their original size.

### **Meeting variable flow reduction**

#### **Variable Speed Drives (VSDS)**

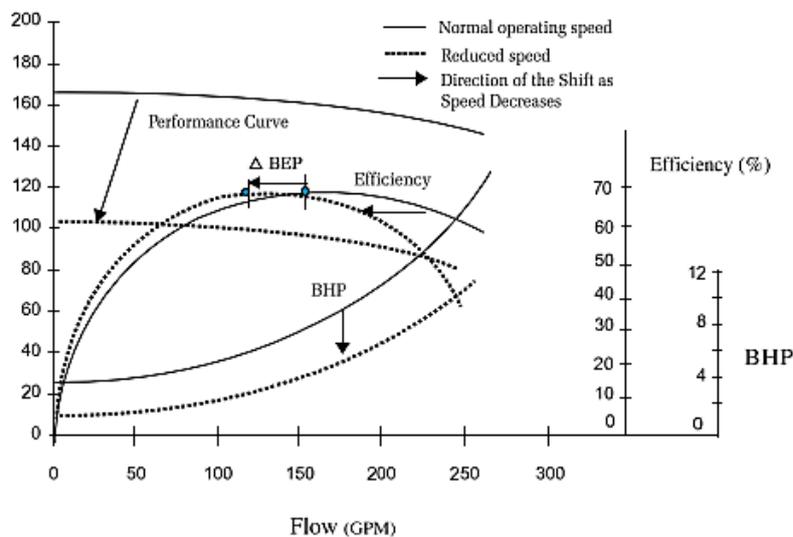
In contrast, pump speed adjustments provide the most efficient means of controlling pump flow. By reducing pump speed, less energy is imparted to the fluid and less energy needs to be throttled or bypassed. There are two primary methods of reducing pump speed: multiple-speed pump motors and variable speed drives (VSDs).

Although both directly control pump output, multiple-speed motors and VSDs serve entirely separate applications. Multiple-speed motors contain a different set of windings for each motor speed consequently they are more expensive and less efficient than single speed' motors. Multiple speed motors also lack subtle speed changing capabilities within discrete speeds.

VSDs allow pump speed adjustments over a continuous range, avoiding the need to jump from speed to speed as with multiple-speed pumps. VSDs control pump speeds using several different types of mechanical and electrical systems. Mechanical VSDs include hydraulic clutches, fluid couplings and adjustable belts and pulleys. Electrical VSDs include eddy current clutches, wound-rotor motor controllers and variable frequency drives (VFDs). VFDs adjust the electrical frequency of the power supplied to a motor to change the motor's rotational speed. VFDs are by far the most popular type of VSD.

However, pump speed adjustment is not appropriate for all systems. In applications with high static head, slowing a pump risks inducing vibrations and creating performance problems that are similar to those found when a pump operates against its shutoff head. For systems in which the static head represents a large portion of the total head, caution should be used in deciding whether to use VFDs. Operators should review the performance of VFDs in similar applications and consult VFD manufacturers to avoid the damage that can result when a pump operates too slowly against high static head.

For many systems, VFDs offer a means to improve pump operating efficiency despite changes in operating conditions. The effect of slowing pump speed on pump operation is illustrated by the three curves in Figure 6.22. When a VFD slows a pump, its head/flow and power curves drop down and to the left and its efficiency curve shifts to the left. This efficiency response provides an essential cost advantage by keeping the operating efficiency as high as possible across variations in the system's flow demand, the energy and maintenance costs of the pump can be significantly reduced.



**Figure 6.22: Effect of VFD**

VFDs may offer operating cost reductions by allowing higher pump operating efficiency but the principal savings derive from the reduction in frictional or bypass flow losses. Using a system perspective to identify areas in which fluid energy is dissipated in non-useful work often reveals opportunities for operating cost reductions.

For example, in many systems, increasing flow through bypass lines does not noticeably impact the backpressure on a pump. Consequently, in these applications pump efficiency does not necessarily decline during periods of low flow demand. By analyzing the entire system, however, the energy lost in pushing liquid through bypass lines and across throttle valves can be identified.

Another system benefit of VFDs is a soft start capability. During startup, most motors experience in-rush currents that are 5 to 6 times higher than normal operating currents. This high current fades when the motor spins up to normal speed. VFDs allow the motor to be started with a lower startup current (usually only about 1.5 times the normal operating current). This reduces wear on the motor and its controller. VFDs will consume 4 to 6% power as a running cost apart from its initial cost.

## 6.7 Boiler Feed Water Pumps (BFP)

The feed water pumps are normally multi stage centrifugal pumps, sized based on boiler design pressure. The operation of a multistage pump is similar to the operation of several single stage pumps, of identical capacity, in series. Since most boilers operate below design pressure, the feed water pump head is often higher than required. This excessive pump head is dropped across pressure reducing valves and manual valves. Installing a VFD on the feed water pump in such cases can decrease pump power consumption and improve control performance. Trimming the impeller, reducing number of stages or changing the feed water pumps may also be feasible depending on variation in operating load of the boiler.

### Boiler Feed Pump Control with VFD

There are several ways of controlling the pump

- **One pump, one boiler, no feed water regulating valve:** In this the pump speed is varied according to the level of water in the boiler. The level control system used for the feed water admission valve transmits its signal directly to the pump VFD controller. With this system it is possible to eliminate not only the feed pump constant discharge but also the boiler feed water regulating control valve and, thereby, cut initial capital investment. The inherent efficiency loss due to throttling is eliminated.
- **Constant discharge pressure control:** The feed pump is controlled to a predetermined pressure setting irrespective of plant load. The advantage of this system is that the pump will not be required to operate near shut off pressures, due to the shifting of the operating point on the curve.
- **Constant differential pressure control:** Feed pump pressure is controlled to produce a predetermined pressure drop across the feed water regulating valve, usually approximately 3.5 to 5.5 kg/cm<sup>2</sup>, thus allowing the boiler feed pump to follow plant demand.

### Optimizing Boiler Feed Water Pump Capacity - Case

A waste heat boiler has two feed water pumps, each of 6 stages and having a capacity of 35 m<sup>3</sup>/hr. The pumps are designed to generate a head of 276 m, normally one pump is operated.

The actual steam demand is 28 TPH at 15 kg/cm<sup>2</sup>. The capacity of the feed water pumps is far in excess of the requirement. This results in throttling of pump discharge leading to energy loss. To save energy in the BFW pumps, it was suggested to remove two impeller stages of the pump to effectively regulate the pressure developed in the pumps.

The impact of this measure on power consumption was then evaluated and the results are given in Table 6.2:

<i>Table 6.2: Power Drawn before and after</i>
Operation of pumps

With 6 stages (impellers)	53
With 4 stages (impellers)	35
Actual power savings achieved	18

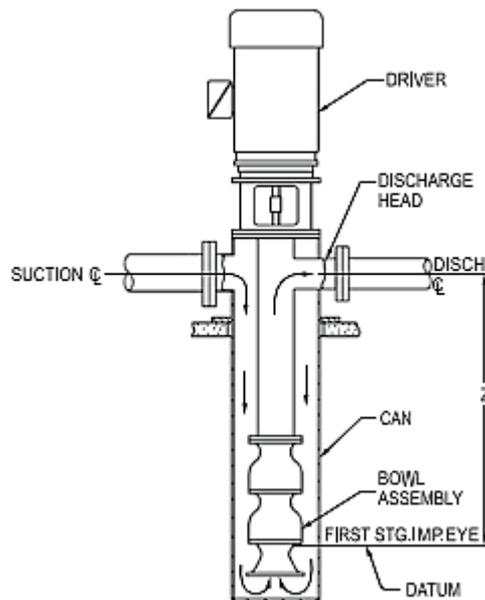
## 6.8 Municipal Water Pumping System

Municipal water pumps are predominantly centrifugal pumps and vertical turbine pumps. The capacity of a water pumping station is normally specified in Million Liters per Day (MLD) of water handled. Municipal water system consists of the following sub systems:

- Raw water pump house, intake pumps at water source/river
- Pure water pump house and filtration plant
- Booster station as per the requirement
- Elevated Storage Reservoirs in the distribution system

### Vertical Turbine Pumps

Vertical turbine pump (Figure 6.23) [deep well turbine pump] is vertical axis centrifugal or mixed flow type pump comprising of stages which accommodate rotating impellers and stationary bowls possessing guide vanes.



*Figure 6.23 Vertical Turbine Pump*

These pumps are used where the pumping water level is below the limits of Volute centrifugal pump. They have higher initial cost and are more difficult to install and repair. The pressure head developed depends on the diameter of impeller and the speed at which it is rotated. The pressure head developed by single impeller is not great. Additional head is obtained by adding more bowl assemblies or stage.

Construction:

It has three parts:

### 1. Pump Element:

The pump element is made up of one or more bowls or stages. Each bowl consists of an impeller and diffuser.

### 2. Discharge Column:

It connects the bowl assembly and pump head and conducts water from former to later.

### Discharge head:

It consists of base from which the discharge column, bowl assembly and shaft assembly are suspended.

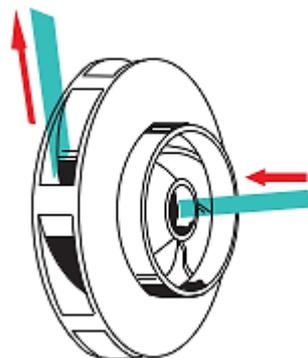
### Submersible Pump:

A vertical turbine pump close coupled to a small diameter submersible electric motor is termed as “submersible pump”. The motor is fixed directly below the intake of the pump. The pump element and the motor operate under submerged condition. It can be used in very deep tube well where a long shaft would not be practical.

## 6.9 Sewage Water Pumps

The fundamental difference between a centrifugal sewage pump impeller and those of clear water pumps is its ability to pass solid material that would normally clog the latter. What differentiates various sewage pump impellers is the method by which they accomplish this. All sewage handling pumps comprise single suction impellers. This is to avoid the necessity of locating the pump shaft in the intake.

The Figure 6.24 is that of a typical radial flow impeller of a pump used to handle coarse solids or fibrous matters. These pumps use volutes because diffuser is prone to clogging. The design of sewage pumps is largely determined by the size of foreign matter that must pass through the pump without clogging.



*Figure 6.24 Radial Flow Impeller*

## 6.10 Agricultural Pumping System

The pumps used in agriculture sector are normally installed by individual farmers based on the guidelines provided by agriculture department/state utilities and feedback from other users. Most of the pumps used are locally manufactured keeping initial investment as the selection criteria rather than efficiency and energy conservation.

The pump sets used are generally inefficient with operating efficiency ranging from 30 - 55%. The wide variation is due to changing water levels in the intake thus forcing the pump to operate away from the best efficiency point. The pump sets are more often oversized so as to draw water from increasingly declining depths and also to withstand large voltage fluctuations.

Mostly centrifugal pumps are used and the capacity of the pumps vary from 1 Hp to 25 Hp. The rating of the pumps is decided based on water table levels. High rating pumps above 25 HP are also used in several areas. Large capacity centrifugal pumps of 75 HP to 500 HP ratings are also used by Irrigation departments to provide water to agricultural consumers. Diesel engine driven pumps are also common in areas where there is erratic or no power supply.

The following energy conservation opportunities have been demonstrated for energy savings in agricultural pumping.

- Installation of low friction foot valves
- Installation of low friction HDPE suction and delivery pipes
- Installation of long bends
- Installation of high efficiency pumps and motors

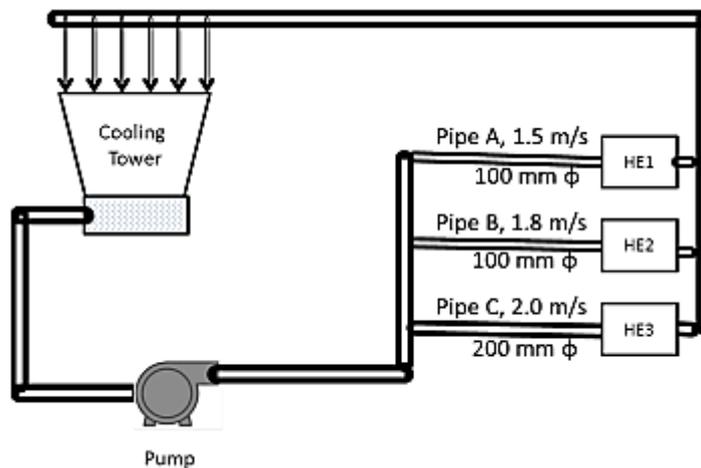
### **6.11 Energy Conservation Opportunities in Pumping Systems**

- Ensure adequate NPSH at site of installation.
- Ensure availability of basic instruments at pumps like pressure gauges, flow meters.
- Operate pumps near Best Efficiency Point.
- Modify pumping system and pumps losses to minimize throttling.
- Adapt to wide load variation with variable speed drives or sequenced control of multiple units.
- Stop running multiple pumps - add an auto-start for an on-line spare or add a booster pump in the problem area.
- Use booster pumps for small loads requiring higher pressures.
- Increase liquid temperature differentials to reduce pumping rates in case of heat exchangers.
- Decrease outlet cold water temperature of cooling tower in order to reduce the pumping flow rates in case of mixing.
- Separate High Pressure and Low Pressure systems
- Repair seals and packing to minimize water loss by dripping.
- Balance the system to minimize flows and reduce pump power requirements.
- Avoid pumping head with a free-fall return (gravity); Use siphon effect to advantage:
- Conduct water balance to minimize water consumption.
- Avoid cooling water re-circulation in DG sets, air compressors, refrigeration systems, cooling towers feed water pumps, condenser pumps and process pumps.

- In multiple pump operations, carefully combine the operation of pumps to avoid throttling.
- Provide booster pump for few areas of higher head.
- Replace old pumps by energy efficient pumps.
- In the case of over designed pump, provide variable speed drive, or downsize / replace impeller or replace with correct sized pump for efficient operation.
- Optimize number of stages in multi-stage pump in case of head margins.
- Reduce system resistance by pressure drop assessment and pipe size optimization.

### Example 6.1

The cooling water circuit of a process industry is depicted in the figure below. Cooling water is pumped to three heat exchangers via pipes A, B and C where flow is throttled depending upon the requirement. The diameter of pipes and measured velocities with non-contact ultrasonic flow meter in each pipe are indicated in the figure.



The following are the other data:

Measured motor power	: 50.7 kW
Motor efficiency at operating load	: 90%
Pump discharge pressure	: 3.4 kg/cm <sup>2</sup>
Suction head	: 2 meters

Determine the efficiency of the pump.

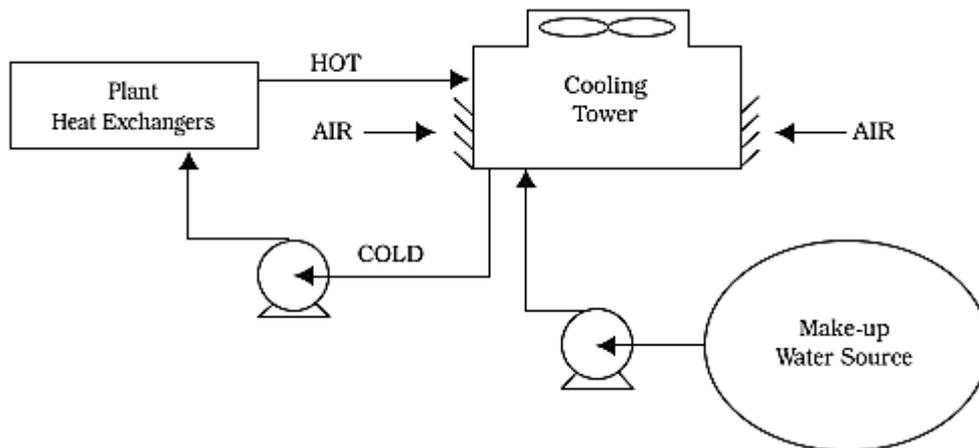
**Solution**

Flow in pipe A	$22/7 \times (0.1)^2/4 \times 1.5$
	0.011786 m <sup>3</sup> /s
Flow in pipe B	$22/7 \times (0.1)^2/4 \times 1.8$
	0.014143 m <sup>3</sup> /s
Flow in pipe C	$22/7 \times (0.2)^2/4 \times 2.0$
	0.062857 m <sup>3</sup> /s
Total flow	0.088786 m <sup>3</sup> /s
Total head	34 m – 2 m = 32 m
Pump hydraulic power	$0.088786 \times 32 \times 9.81$
	27.9 kW
	$27.9 \times 100/50.7 \times 0.9$
Pump efficiency	61 %

## Chapter 7: COOLING TOWER

### 7.1 Introduction

Cooling towers are a very important part of many chemical plants. The primary task of a cooling tower is to reject heat into the atmosphere. They represent a relatively inexpensive and dependable means of removing low-grade heat from cooling water. The make-up water source is used to replenish water lost to evaporation. Hot water from heat exchangers is sent to the cooling tower. The water exits the cooling tower and is sent back to the exchangers or to other units for further cooling. Typical closed loop cooling tower system is shown in Figure 7.1.



*Figure 7.1 Cooling Water System*

### Cooling Tower Types

Cooling towers fall into two main categories: Natural draft and Mechanical draft.

Natural draft towers use very large concrete chimneys to introduce air through the media. Due to the large size of these towers, they are generally used for water flow rates above 45,000 m<sup>3</sup>/hr. These types of towers are used only by utility power stations.

Mechanical draft towers utilize large fans to force or suck air through circulated water. The water falls downward over fill surfaces, which help increase the contact time between the water and the air – this helps maximise heat transfer between the two. Cooling rates of Mechanical draft towers depend upon their fan diameter and speed of operation. Since, the mechanical draft cooling towers are much more widely used; the focus is on them in this chapter.

### Mechanical draft towers

Mechanical draft towers are available in the following airflow arrangements:

1. Counter flows induced draft.
2. Counter flow forced draft.
3. Cross flow induced draft.

In the counter flow induced draft design, hot water enters at the top, while the air is introduced at the bottom and exits at the top. Both forced and induced draft fans are used.

In cross flow induced draft towers, the water enters at the top and passes over the fill. The air, however, is introduced at the side either on one side (single-flow tower) or opposite sides (double-flow tower). An induced draft fan draws the air across the wetted fill and expels it through the top of the structure.

The Figure 7.2 illustrates various cooling tower types. Mechanical draft towers are available in a large range of capacities. Normal capacities range from approximately 10 tons, 2.5 m<sup>3</sup>/hr flow to several thousand tons and m<sup>3</sup>/hr. Towers can be factory built or field erected -for example concrete towers are only field erected.

Many towers are constructed so that they can be grouped together to achieve the desired capacity. Thus, many cooling towers are assemblies of two or more individual cooling towers or “cells.” The number of cells they have, e.g., a eight-cell tower, often refers to such towers. Multiple-cell towers can be lineal, square, or round depending upon the shape of the individual cells and whether the air inlets are located on the sides or bottoms of the cells.

#### Components of Cooling Tower

The basic components of an evaporative tower are: Frame and casing, fill, cold water basin, drift eliminators, air inlet, louvers, nozzles and fans.

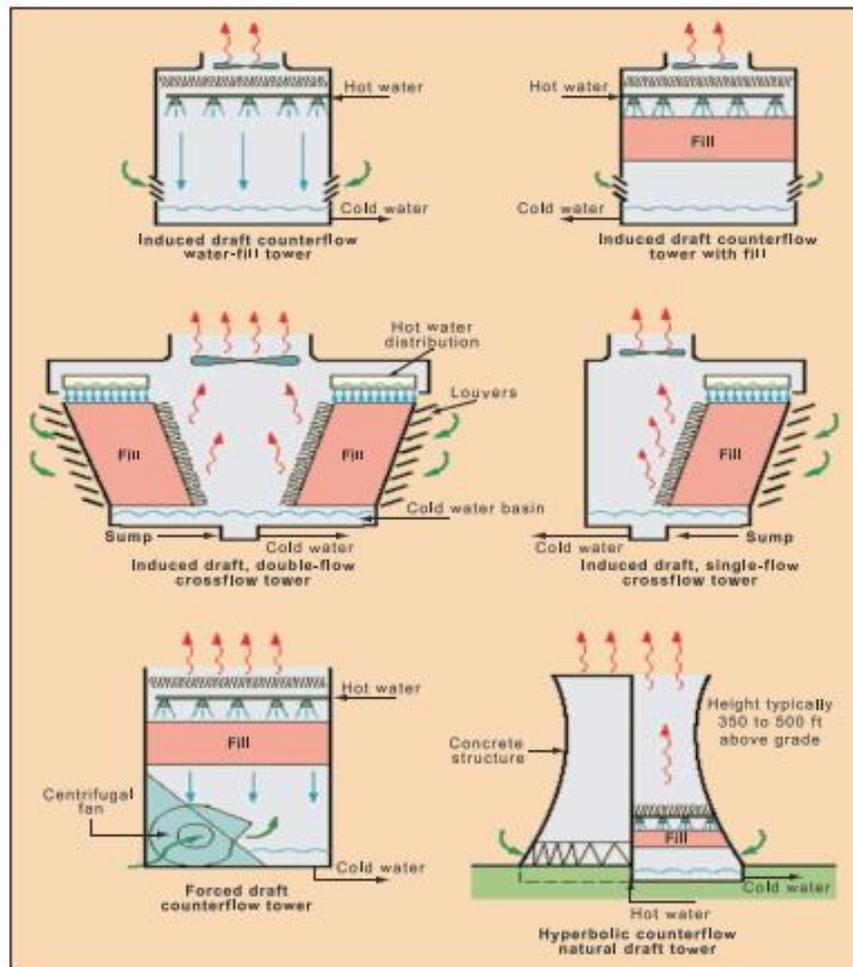
**Frame and casing:** Most towers have structural frames that support the exterior enclosures (casings), motors, fans, and other components. With some smaller designs, such as some glass fiber units, the casing may essentially be the frame.

**Fill:** Most towers employ fills (made of plastic or wood) to facilitate heat transfer by maximising water and air contact. Fill can either be splash or film type.

With splash fill, water falls over successive layers of horizontal splash bars, continuously breaking into smaller droplets, while also wetting the fill surface. Plastic splash fill promotes better heat transfer than the wood splash fill.

Film fill consists of thin, closely spaced plastic surfaces over which the water spreads, forming a thin film in contact with the air. These surfaces may be flat, corrugated, honeycombed, or other patterns. The film type of fill is the more efficient and provides same heat transfer in a smaller volume than the splash fill.

**Cold water basin:** The cold water basin, located at or near the bottom of the tower, receives the cooled water that flows down through the tower and fill. The basin usually has a sump or low point for the cold water discharge connection. In many tower designs, the cold water basin is beneath the entire fill.



*Figure 7.2 Cooling Tower Types*

In some forced draft counter flow design, however, the water at the bottom of the fill is channeled to a perimeter trough that functions as the cold water basin. Propeller fans are mounted beneath the fill to blow the air up through the tower. With this design, the tower is mounted on legs, providing easy access to the fans and their motors.

**Drift eliminators:**

These capture water droplets entrapped in the air stream that otherwise would be lost to the atmosphere.

**Air inlet:** This is the point of entry for the air entering a tower. The inlet may take up an entire side of a tower-cross flow design-or be located low on the side or the bottom of counter flow designs.

**Louvers:** Generally, cross-flow towers have inlet louvers. The purpose of louvers is to equalize air flow into the fill and retain the water within the tower. Many counter flow tower designs do not require louvers.

**Nozzles:** These provide the water sprays to wet the fill. Uniform water distribution at the top of the fill is essential to achieve proper wetting of the entire fill surface. Nozzles can

either be fixed in place and have either round or square spray patterns or can be part of a rotating assembly as found in some circular cross-section towers.

**Fans:** Both axial (propeller type) and centrifugal fans are used in towers. Generally, propeller fans are used in induced draft towers and both propeller and centrifugal fans are found in forced draft towers. Depending upon their size, propeller fans can either be fixed or variable pitch. A fan having non-automatic adjustable pitch blades permits the same fan to be used over a wide range of kW with the fan adjusted to deliver the desired air flow at the lowest power consumption.

Automatic variable pitch blades can vary air flow in response to changing load conditions.

### **Tower Materials**

In the early days of cooling tower manufacture, towers were constructed primarily of wood. Wooden components included the frame, casing, louvers, fill, and often the cold water basin. If the basin was not of wood, it likely was of concrete.

Today, tower manufacturers fabricate towers and tower components from a variety of materials. Often several materials are used to enhance corrosion resistance, reduce maintenance, and promote reliability and long service life. Galvanized steel, various grades of stainless steel, glass fiber, and concrete are widely used in tower construction as well as aluminum and various types of plastics for some components.

Wood towers are still available, but they have glass fiber rather than wood panels (casing) over the wood framework. The inlet air louvers may be glass fiber, the fill may be plastic, and the cold water basin may be steel.

Larger towers sometimes are made of concrete. Many towers—casings and basins—are constructed of galvanized steel or, where a corrosive atmosphere is a problem, stainless steel. Sometimes a galvanized tower has a stainless steel basin. Glass fiber is also widely used for cooling tower casings and basins, giving long life and protection from the harmful effects of many chemicals.

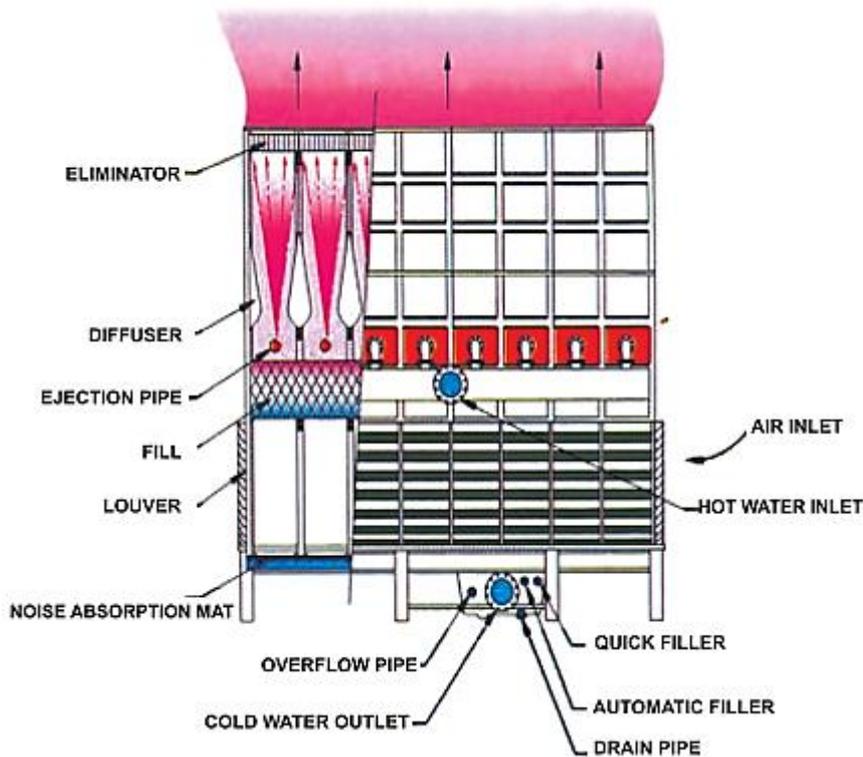
Plastics are widely used for fill, including PVC, polypropylene, and other polymers. Treated wood splash fill is still specified for wood towers, but plastic splash fill is also widely used when water conditions mandate the use of splash fill. Film fill, because it offers greater heat transfer efficiency, is the fill of choice for applications where the circulating water is generally free of debris that could plug the fill passageways.

Plastics also find wide use as nozzle materials. Many nozzles are being made of PVC, ABS, polypropylene, and glass-filled nylon. Aluminum, glass fiber, and hot-dipped galvanized steel are commonly used fan materials. Centrifugal fans are often fabricated from galvanized steel. Propeller fans are fabricated from galvanized, aluminum, or molded glass fiber reinforced plastic.

### **Fanless Cooling Towers**

## Basis of Theory

Fanless cooling tower (Figure 7.3) takes advantage of the water pressure of the existing water circulation pump forming a water screen with specially designed ejection headers. As the water flows through the nozzles at high velocity, based on an ejector principle, low pressure is created which sucks the ambient cold air into the tower. The kinetic energy of Water entering the cooling tower is converted into kinetic energy of the air by the use of specially designed ejector nozzles. Water Pressure required in the Jet Ejector Nozzles is min. 0.5 Bar.



*Figure 7.3 Fanless Cooling Tower*

The incoming air passes through the fills at the bottom while the ejected water falls on the fills thus enabling a counter current heat exchange between water and air. Drift eliminators are provided to contain the drift losses.

## Features of Fanless Cooling Tower

### Energy saving

Since fans are not used in this type of cooling there is a considerable saving of power even though marginally higher power consumption is required for the pump.

### Low noise

The noises of traditional cooling tower originate from the operating fans and motors. Further the vibration caused by these transmission units reinforces the noise resonance. This problem is eliminated in fanless cooling tower since no fan/motor is used.

## Water saving

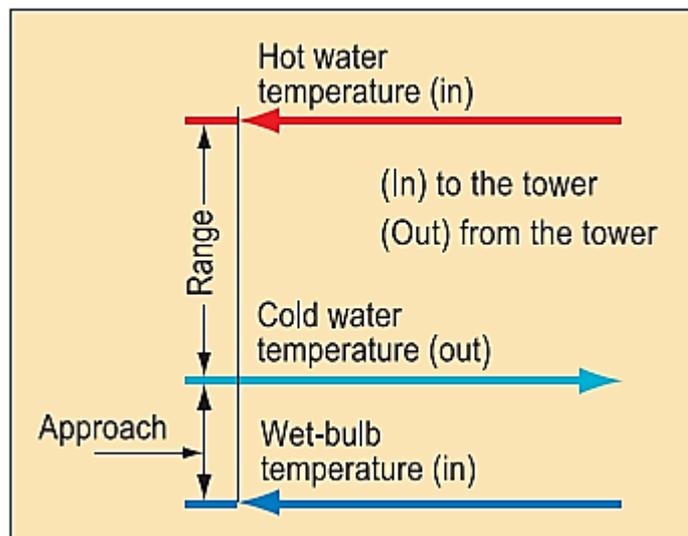
The velocity of water is less than that in conventional cooling tower. In combination with high efficiency drift eliminators this can reduce the drift loss to 0.001% which is much less than that for a conventional tower. Since the water droplets will be less than 50 micron it evaporates immediately without causing any pollution nearby.

## Low maintenance cost

Since the fanless cooling tower has no mechanical equipment such as fan, motor, gearbox etc. there is hardly any maintenance required, provided the quality of circulation water is kept clean and well maintained.

## 7.2 Cooling Tower Performance

The important parameters, from the point of determining the performance of cooling towers, are:



*Figure 7.4 Range and Approach*

- i. "Range" is the difference between the cooling tower water inlet and outlet temperature. (see Figure 7.4).
- ii. "Approach" is the difference between the cooling tower outlet cold water temperature and ambient wet bulb temperature. Although, both range and approach should be monitored, the 'Approach' is a better indicator of cooling tower performance. (see Figure 7.4).
- iii. Cooling tower effectiveness (in percentage) is the ratio of range, to the ideal range, i.e., difference between cooling water inlet temperature and ambient wet bulb temperature, or in other words it is = **Range / (Range + Approach)**.
- iv. Cooling capacity is the heat rejected in kcal/hr or TR, given as product of mass flow rate of water, specific heat and temperature difference

- v. Evaporation loss is the water quantity evaporated for cooling duty and, theoretically, for every 10, 00,000 kcal heat rejected, evaporation quantity works out to 1.8 m3. An empirical relation used often is:

$$\text{*Evaporation Loss (m3/hr) = 0.00085 x 1.8 x circulation rate (m3/hr) x (T}_1\text{-T}_2\text{)}$$

T<sub>1</sub> -T<sub>2</sub> = Temperature difference between inlet and outlet water.

\*Source: Perry's Chemical Engineers Handbook (Page: 12-17)

- vi. Cycles of concentration (CDC) is the ratio of dissolved solids in circulating water to the dissolved solids in makeup water.

$$\text{Blow Down} = \text{Evaporation Loss} / (\text{C.O.C.} - 1)$$

- vii. Liquid/Gas (L/G) ratio, of a cooling tower is the ratio between the water and the air mass flow rates. Against design values, seasonal variations require adjustment and tuning of water and air flow rates to get the best cooling tower effectiveness through measures like water box loading changes, blade angle adjustments.

Thermodynamics also dictate that the heat removed from the water must be equal to the heat absorbed by the surrounding air:

$$L (T_1 - T_2) = G (h_2 - h_1)$$

$$\frac{L}{G} = \frac{h_2 - h_1}{T_1 - T_2}$$

Where,

L/G	=	liquid to gas mass flow ratio (kg/kg)
T <sub>1</sub>	=	hot water temperature (0C)
T <sub>2</sub>	=	cold water temperature (0C)
h <sub>2</sub>	=	enthalpy of air-water vapor mixture at exhaust wet-bulb temperature (same units as above)
h <sub>1</sub>	=	enthalpy of air-water vapor mixture at inlet wet-bulb temperature (same units as above)

### Factors Affecting Cooling Tower Performance

#### Capacity

Heat dissipation (in kcal/hour) and circulated flow rate (m3/hr) are not sufficient to understand cooling tower performance. Other factors, which we will see, must be stated along with flow rate m3/hr. For example, a cooling tower sized to cool 4540 m3/hr through a 13.9°C range might be larger than a cooling tower to cool 4540 m3/hr through 19.5°C range.

## Range

Range is determined not by the cooling tower, but by the process it is serving. The range at the exchanger is determined entirely by the heat load and the water circulation rate through the exchanger and on to the cooling water.

$$\text{Range } ^\circ\text{C} = \text{Heat Load in kcaUhour} / \text{Water Circulation Rate in LPH}$$

As a generalization, the closer the approach to the wet bulb, the more expensive the cooling tower due to increased size. Usually a 2.8°C approach to the design wet bulb is the coldest water temperature that cooling tower manufacturers will guarantee. If flow rate, range, approach and wet bulb had to be ranked in the order of their importance in sizing a tower, approach would be first with flow rate closely following the range and wet bulb would be of lesser importance.

### Factors that affect cooling tower size

Cooling tower size is affected by the heat load, range, approach, and WBT. When three of these four quantities are held constant, tower size varies in the following manner:

- Directly with the heat load
- Inversely with the range
- Inversely with the approach
- Inversely with the entering WBT

## Heat Load

The heat load imposed on a cooling tower is determined by the process being served. The degree of cooling required is controlled by the desired operating temperature level of the process. In most cases, a low operating temperature is desirable to increase process efficiency or to improve the quality or quantity of the product. In some applications (e. g. internal combustion engines), however, high operating temperatures are desirable. The size and cost of the cooling tower is proportional to the heat load. If heat load calculations are low undersized equipment will be purchased. If the calculated load is high, oversize and more costly, equipment will result.

Process heat loads may vary considerably depending upon the process involved. Determination of accurate process heat loads can become very complex but proper consideration can produce satisfactory results. On the other hand, air conditioning and refrigeration heat loads can be determined with greater accuracy.

Information is available for the heat rejection requirements of various types of power equipment. A sample list is as follows:

### Air Compressor

- |   |                  |
|---|------------------|
| - Single-stage                                | - 129 kcal/kW/hr |
| - Single-stage with after cooler              | - 862 kcal/kW/hr |
| - Two-stage with intercooler                  | - 518 kcal/kW/hr |
| - Two-stage with intercooler and after cooler | - 862 kcal/kW/hr |

Refrigeration, Compression	- 63 kcal/min/TR
Refrigeration, Absorption	- 127 kcal/min/TR
Steam Turbine Condenser	- 555 kcal/kg of steam
Diesel Engine, Four-Cycle, Supercharged	- 880 kcal/kW/hr
Natural Gas Engine, Four-cycle (18 kg/cm <sup>2</sup> compression)	- 1523 kcal/kW/hr

## Wet Bulb Temperature

Wet bulb temperature is an important factor in performance of evaporative water cooling equipment.

It is a controlling factor from the aspect of minimum cold water temperature to which water can be cooled by the evaporative method. Thus, the wet bulb temperature of the air entering the cooling tower determines operating temperature levels throughout the plant, process, or system. Theoretically, a cooling tower will cool water to the entering wet bulb temperature, when operating without a heat load. However, a thermal potential is required to reject heat, so it is not possible to cool water to the entering air wet bulb temperature, when a heat load is applied. The approach obtained is a function of thermal conditions and tower capability.

Initial selection of towers with respect to design wet bulb temperature must be made on the basis of conditions existing at the tower site. The temperature selected is generally close to the average maximum wet bulb for the summer months. An important aspect of wet bulb selection is, whether it is specified as ambient or inlet. The ambient wet bulb is the temperature, which exists generally in the cooling tower area, whereas inlet wet bulb is the wet bulb temperature of the air entering the tower. The latter can be, and often is, affected by discharge vapors being recalculated into the tower. Recirculation raises the effective wet bulb temperature of the air entering the tower with corresponding increase in the cold water temperature. Since there is no initial knowledge or control over the recirculation factor, the ambient wet bulb should be specified. The cooling tower supplier is required to furnish a tower of sufficient capability to absorb the effects of the increased wet bulb temperature peculiar to his own equipment.

It is very important to have the cold water temperature low enough to exchange heat or to condense vapours at the optimum temperature level. By evaluating the cost and size of heat exchangers versus the cost and size of the cooling tower, the quantity and temperature of the cooling tower water can be selected to get the maximum economy for the particular process.

The Table 7. 1 illustrates the effect of approach on the size and cost of a cooling tower. The towers included were sized to cool 4540 m<sup>3</sup>/hr through a 16.67°C range at a 267°C design wet bulb. The overall width of all towers is 21.65 meters; the overall height, 15 .25 meters, and the pump head, 10.6 m approximately.

**Table- 7.1: Approach VS Cooling Tower Size (4540 m<sup>3</sup>/hr; 16.67°C Range; 26.7 °C Wet bulb; 10.7 Pump Head)**

Approach °C	2.77	3.33	3.88	4.44	5.0	5.55
Hot Water °C	46.11	46.66	47.22	47.77	48.3	48.88
Cold Water °C	29.44	30	30.55	31.11	31.66	32.22
No. of Cells	4	4	3	3	3	3
Length of Cells Mts.	10.98	8.54	10.98	9.76	8.54	8.54
Overall Length Mts.	43.9	34.15	32.93	29.27	25.61	25.61
No. of Fans	4	4	3	3	3	3
Fan Diameter Mts.	7.32	7.32	7.32	7.32	7.32	6.71
Total Fan kW	270	255	240	202.5	183.8	183.8

### Approach and Flow

Suppose a cooling tower is installed that is 21.65 m wide x 36.9 m long x 15.24 m high, has three 7.32m diameter fans and each powered by 25 kW motors. The cooling tower cools from 3632 m<sup>3</sup>/hr water from 46.1°C to 29.4°C at 26.7°C WBT dissipating 60.69 million kcal/hr. The Table 7.2 shows what would happen with additional flow but with the range remaining constant at 16.67°C. The heat dissipated varies from 60.69 million kcal/hr to 271.3 million kcal/hr.

**Table 7.2 Flow vs. Approach for a Given Tower (Tower is 21.65 m x 36.9 M; Three 7.32 M Fans; Three 25 kw Motor; 16.7°C Range with 26.7°C Wet Bulb)**

Flow m <sup>3</sup> /hr	Approach °C	Cold Water °C	Hot Water °C	Million kcal/hr
3632	2.78	29.40	46.11	60.691
4086	3.33	29.95	46.67	68.318
4563	3.89	30.51	47.22	76.25
5039	4.45	31.07	47.78	84.05
5516	5.00	31.62	48.33	92.17
6060.9	5.56	32.18	48.89	101.28
7150.5	6.67	33.29	50.00	119.48
8736	8.33	35.00	51.67	145.63
11590	11.1	37.80	54.45	191.64
13620	13.9	40.56	57.22	226.91
16276	16.7	43.33	60.00	271.32

For meeting the increased heat load, few modifications would be needed to increase the water flow through the tower. However, at higher capacities, the approach would increase.

### Range, Flow and Heat Load

Range is a direct function of the quantity of water circulated and the heat load. Increasing the range as a result of added heat load does require an increase in the tower size. If the cold water temperature is not changed and the range is increased with higher hot water temperature, the driving force between the wet bulb temperature of the air entering the

tower and the hot water temperature is increased, the higher level heat is economical to dissipate.

If the hot water temperature is left constant and the range is increased by specifying a lower cold water temperature, the tower size would have to be increased considerably. Not only would the range be increased, but the lower cold water temperature would lower the approach. The resulting change in both range and approach would require a much larger cooling tower.

### **Approach & Wet Bulb Temperature**

The design wet bulb temperature is determined by the geographical location. Usually the design wet bulb temperature selected is not exceeded over 5 percent of the time in that area. Wet bulb temperature is a factor in cooling tower selection; the higher the wet bulb temperature, the smaller the tower required to give a specified approach to the wet bulb at a constant range and flow rate.

A 4540 m<sup>3</sup>/hr cooling tower selected for a 16.67°C range and a 4.45°C approach to 21.11°C wet bulb would be larger than a 4540 m<sup>3</sup>/hr tower selected for a 16.67°C range and a 4.45°C approach to a 26.67°C wet bulb. Air at the higher wet bulb temperature is capable of picking up more heat. Assume that the wet bulb temperature of the air is increased by approximately 11.1°C. As air removes heat from the water in the tower, each kg of air entering the tower at 21.1°C wet bulb would contain 18.86 kcal and if it were to leave the tower at 32.2°C wet bulb it would contain 24.17 kcal per kg of air.

In the second case, each kg of air entering the tower at 26.67°C wet bulb would contain 24.17 kcal and were to leave at 37.8°C wet bulb it would contain 39.67 kcal per kg of air.

In going from 21.1°C to 32.2°C, 12.1 kcal per kg of air is picked up, while 15.5 kcal/kg of air is picked up in going from 26.67°C to 37.8°C.

### **Fill Media Effects**

In a cooling tower, hot water is distributed above fill media which flows down and is cooled due to evaporation with the intermixing air. Air draft is achieved with use of fans. Thus some power is consumed in pumping the water to a height above the fill and also by fans creating the draft.

An energy efficient or low power consuming cooling tower is to have efficient designs of fill media with appropriate water distribution, drift eliminator, fan, gearbox and motor. Power savings in a cooling tower, with use of efficient fill design, is directly reflected as savings in fan power consumption and pumping head requirement.

### **Function of Fill media in a Cooling Tower**

Heat exchange between air and water is influenced by surface area of heat exchange, time of heat exchange (interaction) and turbulence in water effecting thoroughness of intermixing. Fill media in a cooling tower is responsible to achieve all of above.

**Splash and Film Fill Media:** As the name indicates, splash fill media generates the required heat exchange area by splashing action of water over fill media and hence breaking into smaller water droplets. Thus, surface of heat exchange is the surface area of the water droplets, which is in contact with air.

**Film Fill and its Advantages**

In a film fill, water forms a thin film on either side of the fill sheets. Thus area of heat exchange is the surface area of the fill sheets, which is in contact with air.

Typical comparison between various fill media is shown in Table 7.3.

*Table 7.3. Comparisons between various fill media*

	<b>Splash Fill</b>	<b>Film Fill</b>	<b>Low Clog Film Fill</b>
Possible L/G Ratio	1.1 – 1.5	1.5 – 2.0	1.4 – 1.8
Effective Heat Exchange Area	30 – 45 m <sup>2</sup> /m <sup>3</sup>	150 m <sup>2</sup> /m <sup>3</sup>	85 – 100 m <sup>2</sup> /m <sup>3</sup>
Fill Height Required	5 – 10 m	1.2 – 1.5 m	1.5 – 1.8 m
Pumping Head Requirement	9 – 12 m	5 – 8 m	6 – 9 m
Quantity of Air Required	High	Much low	Low

Due to fewer requirements of air and pumping head, there is a tremendous saving in power with the invention of film fill. Recently, low-clog film fills with higher flute sizes have been developed to handle high turbid waters. For sea water, low clog film fills are considered as the best choice in terms of power saving and performance compared to conventional splash type fills.

**Choosing a Cooling Tower**

The counter-flow and cross flows are two basic designs of cooling towers based on the fundamentals of heat exchange. It is well known that counter flow heat exchange is more effective as compared to cross flow or parallel flow heat exchange.

Cross-flow cooling towers are provided with splash fill of concrete, wood or perforated PVC. Counter-flow cooling towers are provided with both film fill and splash fill.

Typical comparison of Cross flow Splash Fill, Counter Flow Tower with Film Fill and Splash fill is shown in Table 7.4. The power consumption is least in Counter Flow Film Fill followed by Counter Flow Splash Fill and Cross-Flow Splash Fill.

*Table 7.4: Typical Comparison of Cross flow splash fill, Counter Flow Tower with Film Fill and*

### *Splash Fill*

Number of Towers	:	2		
Water Flow	:	16000 m <sup>3</sup> /hr.		
Hot Water Temperature	:	41.5°C		
Cold Water Temperature	:	32.5°C		
Design Wet Bulb Temperature	:	27.6°C		
		<b>Counter Flow Film Fill</b>	<b>Counter Flow Splash Fill</b>	<b>Cross-Flow Splash Fill</b>
Fill Height, Meter		1.5	5.2	11.0
Plant Area per Cell		14.4 × 14.4	14.4 × 14.4	12.64 × 5.49
Number of Cells per Tower		6	6	5
Power at Motor Terminal/Tower, kW		253	310	330
Static Pumping Head, Meter		7.2	10.9	12.05

### 7.3 Efficient System Operation

#### i. Cooling Water Treatment

Cooling water systems is one of the Critical utility in Power plants, process industries and in Air- conditioning systems. The power plant performance, Chiller performance have direct effect on energy consumption, based on Cooling water temperatures which in turn is maintained by good cooling water treatment.

The various problems in Cooling water system and the corrective measures required are discussed below.

#### a) Water Side Problems

Usually the typical problems that any (Open) cooling system meets with are:

- Corrosion and/or Scale formation
- Biological/Micro-biological fouling

#### **Corrosion:**

Corrosion, being not a precisely understood phenomenon, is a function of various factors of which the following are the main factors responsible for promoting corrosion in the system; high salinity of the water, low PH, low Alkalinity, presence of corrosive gases (mainly oxygen and CO<sub>2</sub>), dissimilarity of the metals etc.

Corrosion can either lead to failure of the metallurgy (leakages in the heat exchangers) and/or deposit formation of corrosion products.

#### **Scale Formation:**

The main sources for the scale formation in the Open Evaporative Condenser circuit are:

Hard water containing, high levels of Calcium and Magnesium, high level of PH and Alkalinity. An open evaporative cooling systems (condenser water systems) operated on softened water can meet with severe scaling problems when

- PH of the circulating water is above 9.0
- The total Alkalinity as CaCO<sub>3</sub> is above 550 ppm
- Temporary hardness in the sources of make-up is above 200 ppm

### **Biological/Micro—Biological Fouling**

Systems exposed to sunlight (mainly cooling tower) often meet with severe problem of algae formation. Other problems associated with algae are slime mass, fungi and various species of bacteria.

Bacteria being miniature bodies, of which growth is not controlled, can lead to the formation of fine masses of suspended particles that lead to fouling and deposit formation. Algae obviously block the nozzles of the cooling tower and thus reduce temperature drop across the tower. Slime masses again are responsible for fouling and deposit formation.

### **Deposit Formation:**

Foreign matter such as; turbidity, sand, silt, mud, air borne debris and other suspended impurities are the sources of deposits formation. Corrosion products that are formed also add to the deposit formation.

#### b) Energy Losses:

Regardless of the type of system, be it open or closed, if it meets with any of the above problems, either the cooling tower nozzles are blocked resulting in reduced Delta 'T' and/or the deposits/scales are formed on the heat transfer surfaces.

For example, the energy losses due to scale and deposit formation in a cooling water circuit of a refrigeration system are significant as shown in Table 7.5. The scale and deposit on the heat transfer area in process equipment can also cause production loss.

*Table 7.5 Effect of Fouling on Efficiency and Power Consumption*

<b>Fouling Factor</b>	<b>Thickness of scale/deposit (mm)</b>	<b>% Reduction in efficiency in terms of heat transfer (Condenser)</b>	<b>% Increase in power consumption due to drop in condenser efficiency</b>
Clean	0.00	0	-
0.0005	0.15	30	5
0.001	0.30	44	10
0.002	0.60	63	20
0.003	0.90	72	30

c) Solution to the Problems

### **ON line / OFF Line Chemical Cleaning**

Depending on the criticality the plant management may adopt ON line/ OFF line cleaning systems.

### **Preventive Treatment**

For preventive treatment, a wide range of chemicals are available in the market and formulations manufactured by reputed companies are generally very safe to use in the system.

### **Corrosion/Scale Inhibitors**

To control corrosion and scale formation depending upon the severity of each of the problem, either or both chemicals should be used and the selection of the chemicals should be made in accordance with the quality of the make-up water available for plant operation.

### **Dispersants (For Deposit Formation)**

Suitable dispersants help in controlling the deposit formation and selection of the dispersants is made in accordance with the nature of suspended solids/deposits forming particulate present in the water.

### **Side Stream Filter**

Circulating water having very high levels of turbidity and/or suspended impurities should be facilitated with side stream filters. Side stream filters are generally selected to handle 2% to 5% of the total rate of circulation, but to ensure that the total water content in the system (hold-up volume) is filtered approximately once in 12 hours.

### **Bio Dispersants and Biocides**

To combat problems arising due to the growth of biological and micro biological species, such as algae, fungi, slime, bacteria etc. It is very essential to select a combination of oxidizing and non- oxidizing biocides. Bio-dispersants are used to remove the upper layer

of the biological masses and allow better penetration of biocides in the lower layers of bio-masses.

## **Chlorination**

Chlorination is the most effective and most economical oxidizing biocide. Chlorination for the smaller systems may be done with hypo chlorite based products and for the larger systems having hold-up volume in excess of 100 m<sup>3</sup> be done with suitable gas chlorinators. The safest gas chlorination equipment are vacuum gravity feed type which can be easily installed on either 50 kg or 100 kg chlorine cylinders.

### ii. Drift Loss in the Cooling Towers

It is very difficult to ignore drift problem in cooling towers. Now-a—days most of the end user specification calls for 0.02% drift loss. With technological development and processing of PVC, manufacturers have brought large change in

the drift eliminator shapes and the possibility of making efficient designs of drift eliminators that enable end user to specify the drift loss requirement to as low as 0.003 — 0.001%.

### iii. Cooling Tower Fans

The purpose of a cooling tower fan is to move a specified quantity of air through the system, overcoming the system resistance which is defined as the pressure loss. The product of air flow and the pressure loss is air power developed/work done by the fan; this may be also termed as fan output and input kW depends on fan efficiency.

The fan efficiency in turn is greatly dependent on the profile of the blade. An aerodynamic profile with optimum twist, taper and higher coefficient of lift to coefficient of drop ratio can provide the fan total efficiency as high as 85-92 %. However, this efficiency is drastically affected by the factors such as tip clearance, obstacles to airflow and inlet shape, etc.

As the metallic fans are manufactured by adopting either extrusion or casting process it is always difficult to generate the ideal aerodynamic profiles. The FRP blades are normally hand moulded which facilitates the generation of optimum aerodynamic profile to meet specific duty condition more efficiently. Cases reported where replacement of metallic or Glass fibre reinforced plastic fan blades have been replaced by efficient hollow FRP blades, with resultant fan energy savings of the order of 20-30% and with simple payback period of 6 to 7 months.

Also, due to lightweight, FRP fans need low starting torque resulting in use of lower HP motors. The

Light weight of the fans also increases the life of the gear box, motor and bearing is and allows for easy handling and maintenance.

### iv. Performance Assessment of Cooling Towers

In operational performance assessment, the typical measurements and observations involved are:

- Cooling tower design data and curves to be referred to as the basis.
- Intake air WBT and DBT at each cell at ground level using a whirling psychrometer.
- Exhaust air WBT and DBT at each cell using a whirling psychrometer.
- CW inlet temperature at risers or top of tower, using accurate mercury in glass or a digital thermometer.
- CW outlet temperature at full bottom, using accurate mercury in glass or a digital thermometer.
- Process data on heat exchangers, loads on line or power plant control room readings, as relevant.
- CW flow measurements, either direct or inferred from pump motor kW and pump head and flow characteristics.
- CT fan motor amps, volts, kW and blade angle settings
- TDS of cooling water.
- Rated cycles of concentration at the site conditions.
- Observations on nozzle flows, drift eliminators, condition of fills, splash bars, etc.

The findings of one typical trial pertaining to the Cooling Towers of a Thermal Power Plant 3 x 200 MW is given below:

### Observations

Unit Load 1 & 3 of the Station	=	398 MW
Mains Frequency	=	49.3
Inlet Cooling Water Temperature °C	=	44 (Rated 43°C)
Outlet Cooling Water Temperature °C	=	37.6 (Rated 33°C)
Air Wet Bulb Temperature near Cell °C	=	29.3 (Rated 27.5°C)
Air Dry Bulb Temperature near Cell °C	=	40.8°C
Number of CT Cells on line with water flow	=	45 (Total 48)
Total Measured Cooling Water Flow m <sup>3</sup> /hr	=	70426.76
Measured CT Fan Flow m <sup>3</sup> /hr	=	989544

### Analysis

CT water Flow/Cell, m <sup>3</sup> /hr	=	1565 m <sup>3</sup> /hr (1565000 kg/hr) (Rated 1875 m <sup>3</sup> /hr)
CT Fan air Flow, m <sup>3</sup> /hr (Avg.)	=	989544 m <sup>3</sup> /hr (Rated 997200 m <sup>3</sup> /hr)
CT Fan air Flow kg/hr (Avg.) @ Density of 1.08 kg/m <sup>3</sup>	=	1068708 kg/hr
L/G Ratio of C.T. kg/kg	=	1.46 (Rated 1.74 kg/kg)

CT Range	=	$(44 - 37.6) = 6.4^{\circ}\text{C}$
CT Approach	=	$(37.6 - 29.3) = 8.3^{\circ}\text{C}$
% CT Effectiveness	=	$\frac{\text{Range}}{(\text{Range} + \text{Approach})} \times 100$
	=	$\frac{6.4}{(6.4 + 8.3)} \times 100$
	=	43.53
Rated % CT Effectiveness	=	$100 * (43 - 33) / (43 - 27.5)$
	=	64.5%

Cooling Duty Handled/Cell in kcal (i.e., Flow * Temperature Difference in kcal/hr)	=	$1565 * 6.4 * 10^3$
Evaporation Losses in m <sup>3</sup> /hr	=	$10016 * 10^3 \text{ kcal/hr (Rated } 18750 * 10^3 \text{ kcal/hr)}$
	=	$0.00085 \times 1.8 \times \text{circulation rate (m}^3\text{/hr)}$
	=	$\times (T_1 - T_2)$
	=	$0.00085 \times 1.8 \times 1565 \times (44 - 37.6)$
	=	15.32 m <sup>3</sup> /hr per cell
Percentage Evaporation Loss	=	$[15.32/1565] * 100$
	=	0.97%
Blow down requirement for site COC of 2.7	=	Evaporation losses / (COC-1)
	=	$15.32/(2.7-1)$ per cell i.e., 9.01 m <sup>3</sup> /hr
Make up water requirement/cell in m <sup>3</sup> /hr	=	Evaporation Loss + Blow down Loss
	=	$15.32 + 9.01 = 2433$

### Comments

- Cooling water flow per cell is much lower, almost by 16.5%, need to investigate CW pump and system performance for improvements. Increasing CW flow through cell was identified as a key result area for improving performance of cooling towers.
- Flow stratification in 3 cooling tower cells identified.

- Algae growth identified in 6 cooling tower cells.
- Cooling tower fans are of GRP type drawing 36.2 kW average. Replacement by efficient hollow FRP fan blades is recommended.

#### 7.4 Flow Control Strategies

Control of tower air flow can be done by varying methods: starting and stopping (ON-OFF) of fans, use of two- or three-speed fan motors, use of automatically adjustable pitch fans, and use of variable speed fans.

ON-OFF fan operation of single speed fans provides the least effective control. Two-speed fans provide better control with further improvement shown with three speed fans. Automatic adjustable pitch fans and variable-speed fans can provide even closer control of tower cold-water temperature. In multi-cell towers, fans in adjacent cells may be running at different speeds or some may be on and others off depending upon the tower load and required water temperature. Depending upon the method of air volume control selected, control strategies can be determined to minimize fan energy while achieving the desired control of the Cold water temperature.

#### 7.5 Energy Saving Opportunities in Cooling Towers

- Follow manufacturer's recommended clearances around cooling towers and relocate or modify structures that interfere with the air intake or exhaust.
- Optimize 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.
- On old counter-flow cooling towers, replace old spray type nozzles with new square spray
- ABS practically non-clogging nozzles.
- Replace splash bars with self-extinguishing PVC cellular film fill.
- Install new nozzles to obtain a more uniform water pattern
- Periodically clean plugged cooling tower distribution nozzles.
- Balance flow to cooling tower hot water basins.
- Cover hot water basins to minimize algae growth that contributes to fouling.
- Optimize blow down flow rate, as per COC limit.
- Replace slat type drift eliminators with low pressure drop, self-extinguishing, PVC cellular units.
- Restrict flows through large loads to design values.
- Segregate high heat loads like furnaces, air compressors, DG sets, and isolate cooling towers for sensitive applications like A/C plants, condensers of captive power plant etc. A 1°C cooling water temperature increase may increase A/C compressor kW by 2.7%. A 1°C drop in cooling water temperature can give a heat rate savings of 5 kcal/kWh in a thermal power plant.
- Monitor L/G ratio, CW flow rates w.r.t. design as well as seasonal variations. It would help to increase water load during summer and times when approach is high and increase air flow during monsoon times and when approach is narrow.
- Monitor approach, effectiveness and cooling capacity for continuous optimization efforts, as per seasonal variations as well as load side variations. — Consider COC improvement measures for water savings.

- Consider energy efficient FRP blade adoption for fan energy savings.
- Consider possible improvements on CW pumps w.r.t. efficiency improvement.
- Control cooling tower fans based on leaving water temperatures especially in case of small units.
- Optimize process CW flow requirements, to save on pumping energy, cooling load, evaporation losses (directly proportional to circulation rate) and blow down losses.

### **Case Study 7.1: Application of VFD for Cooling Tower (CT) Fan**

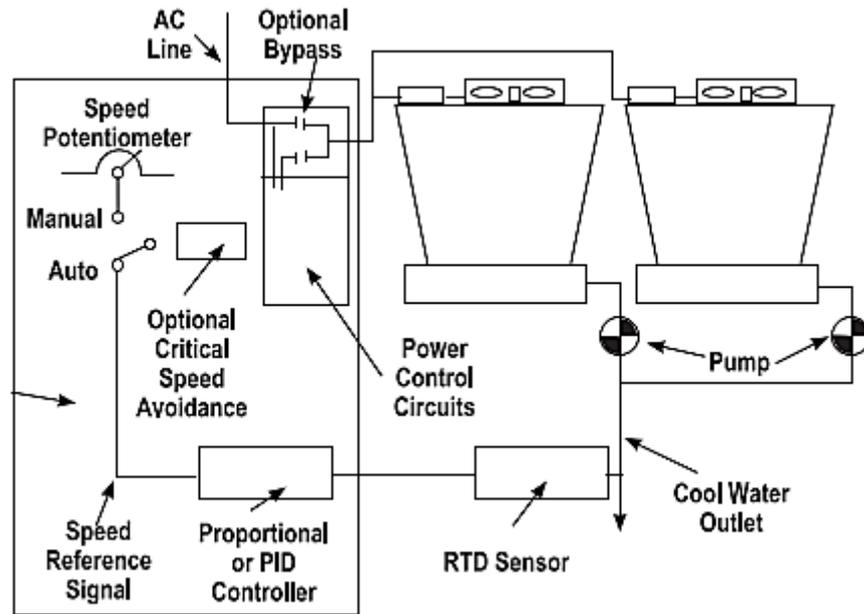
The rating (KW) of the CT fan is selected for the worst case wet and dry bulb temperatures. In areas where such temperature conditions occur for a small portion of the year & which require maximum air flow for this condition, it is possible to improve energy efficiency by reducing the speed of the fan (to obtain reduced air flow), using a VFD.

It is therefore necessary to obtain data for the variations in wet and dry bulb temperatures on an annual basis to arrive at estimates for the energy saved through use of VFD. Alternatively, it is also possible to install a VFD on a trial basis on the CT fans and measure the electrical power consumed with and Without VFD.

The relationship between the power consumed by the CT fan and the airflow delivered by it follows a cube law. The potential for energy savings exists if a proper analysis of the cooling system is made.

### **Implementation with VFD**

An energy efficient system with VFD can be realized through the use of closed loop control. In this control method, the return or cold water temperature is used as the feedback signal to the PID controller which is a standard control block in the drive.



*Figure 7.5 Schematic of CT fan control with VFD & closed loop feedback*

The highlights of control with temperature feedback and drive can be summarized below:

An RTD sensor, installed at the CT outlet generates a 4-20 mA current signal as the feedback to the integrated Process PID Controller in the drive.

The set point for the cooled water temperature is entered in engineering units ( $^{\circ}\text{C}$  or  $0\text{F}$ ) in the drive controller.

Any error between the set point and the feedback signal (temperature in this case) will be integrated by the PID controller so that the same, after correction, is zero. For example, if there is an increase in the outlet temperature (due to wet bulb temperature increase or due to an increase in plant load), the feedback exceeds the set point & the error  $A$  becomes negative. The PID controller output will now try to increase the drive frequency so that the fans deliver more cooling air for evaporation. This has the effect of bringing down the outlet temperature. The correction continues till the feedback signal matches the set point. A similar correction takes place when the outlet temperature reduces. In that case, the CT fan motor speed is reduced to bring the  $A$  value to zero.

This design therefore permits precise control of outlet temperature and conserves energy.

In the event of drive failure, the CT fans can still be operated through an optional built-in bypass circuit which will transfer the power source to the mains supply, thereby ensuring uninterrupted operation.

### **Use of VFD for CT fan motors in Ingot manufacturing plant**

An aluminium ingot manufacturing plant requires large amounts of water for cooling of the ingots.

Hence cooling tower fans are required to cool the water from the ingot plant. The salient features of the application are as given below:

- Drives have been installed on two Cooling Towers.
- Details of drives supplied as follows:

Cooling Tower (CT)	No of CT Fans	CT Fan Motor, KW	Drive, KW
1	3	11	37
2	3	15	45

### Details of control and power consumption

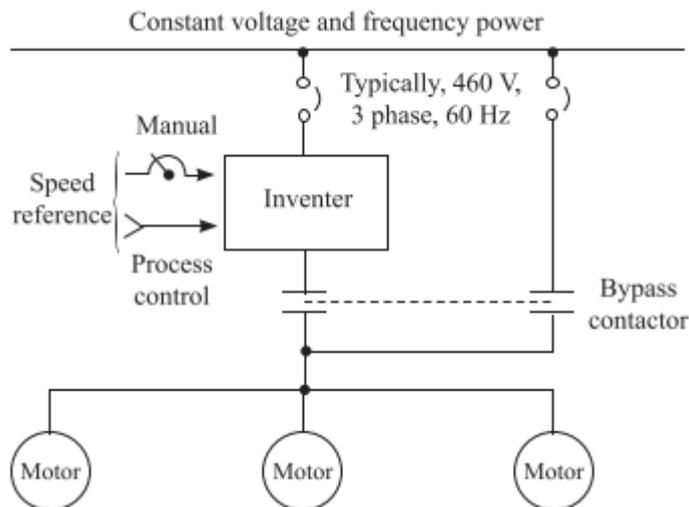
The previous method of control employed a digital temperature controller to switch ON & OFF the CT Fans depending upon the basin temperature. A typical daily regimen employed for CT#1 was as follows:

- 3 Nos. CT Fans running for 8 Hours.
- 2 Nos. CT Fans running for 8 Hours.
- 1 No CT Fan running for 5 hours.

Trials were taken with the VFD (common to 3 nos. drive motors as shown in Figure 7.6 below) installed in CT 1 and run for a period of one month to ascertain the power consumption with and without the drive.

Note:

- The applicable power supply voltage in this case would be 415V, 50Hz.
- There is a bypass line which will enable the motors to operate from mains supply in case of VFD failure.



*Figure 7.6 Typical Power Schematic of VFD and CT fan motors*

It was compared with the power consumed with manual operation (as described above) on a daily basis. It yielded a significant result in terms of power saved daily.

### Savings calculation with VFD operation:

Energy consumed daily with manual control (kWh)	= 392
Energy consumed daily with VFD control (kWh)	= 254
Energy saved on daily basis with VFD control (kWh)	= 138
Energy saved on daily basis with VFD control (%)	= 35.2
Unit energy cost (BDT)	= 5
Number of days of running of fans in a month	= 26
Number of months in a year	= 12
Annual savings due to VFD operation (BDT)	= 215280
Average price of drive panel (BDT)	= 250000
Payback period (Year)	= 1.16

\*Data for savings extrapolated for annual estimates

In practice, the power saved with VFD operation would also depend upon the wet bulb temperature which would vary on a seasonal basis. In case of higher temperatures, the VFD would be required to run at maximum speed during which period, the savings would be negligible. Hence the average annual savings would reduce depending upon the site environmental conditions. The quantum of savings can be optimized by having a closed loop system as shown above which will track the outlet water temperature and determine the drive motor speed accurately.

### Example 7.2

The energy audit observations at a cooling tower (CT) in a process industry are given below:

Cooling Water (CW) Flow: 3000 m<sup>3</sup>/hr

CW in Temperature: 41 deg. C

CW Out Temperature: 31 deg C

Wet Bulb Temperature: 24 deg. C

Find out Range, Approach, Effectiveness and cooling tower capacity in kcal per hour of the CT?

#### Solution

Range = (Inlet -Outlet) Cooling Water Temperature deg. C

Range = (41 — 31) = 10 deg. C

Approach = (Outlet Cooling Water - Air Wet Bulb) Temperature deg. C

Approach = (31 — 24) = 7 deg C

% CT Effectiveness = Range / (Range + Approach) x 100

% Effectiveness = 100 x [Range / (Approach + Range)]

10/ [10+7] x 100 = 58.8 %

Cooling capacity, kcal/hr = heat rejected = CW flow rate in kg per hour x (CW inlet hot water temp. to CT, deg. C - CW outlet cold well temp, deg. C)

Cooling capacity = 3000X1000X (41 - 31) = 30,000,000 kCal per hour = 30 Million kcal/hour.

## **Chapter 8: Lighting System**

### **8.1 Introduction**

Most natural light comes from the sun, including moon light. Its origin makes it completely clean and it consumes no natural resources. But man-made sources generally require consumption of resources, such as fossil fuels, to convert stored energy into light energy.

Light is usually described as the type of electromagnetic radiation that has a wavelength visible to the human eye, roughly 400 to 700 nanometers. Light exists as tiny “packets” called photons and exhibits the properties of both particles and waves. Visible light, as can be seen on the electromagnetic spectrum, as given in Figure 8.1, represents a narrow band between ultraviolet light (UV) and infrared energy (heat). These light waves are capable of exciting the eye’s retina, which results in a visual sensation called sight. Therefore, seeing requires a functioning eye and visible light.

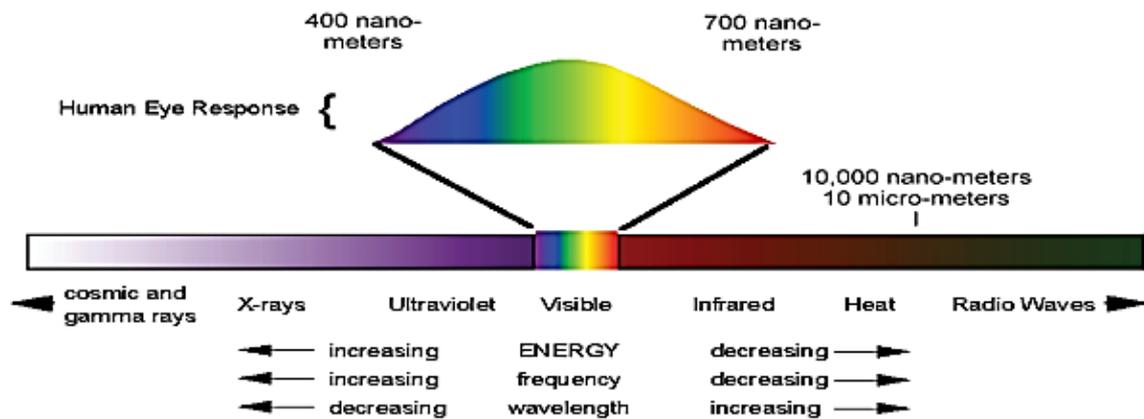


Figure 8.1: Visible Radiation

The lumen (lm) is the photometric equivalent of the Watt, weighted to match the eye response of the “standard observer”. Yellowish—green light receives the greatest weight because it stimulates the eye more than blue or red light of equal radiometric power:

1 Watt = 683 lumens at 555 nm wavelength

The best eye sensitivity, as seen from Figure 8.2 is at 555 nm wavelength having greenish yellow color with a luminous efficacy of 683 lm/Watt.

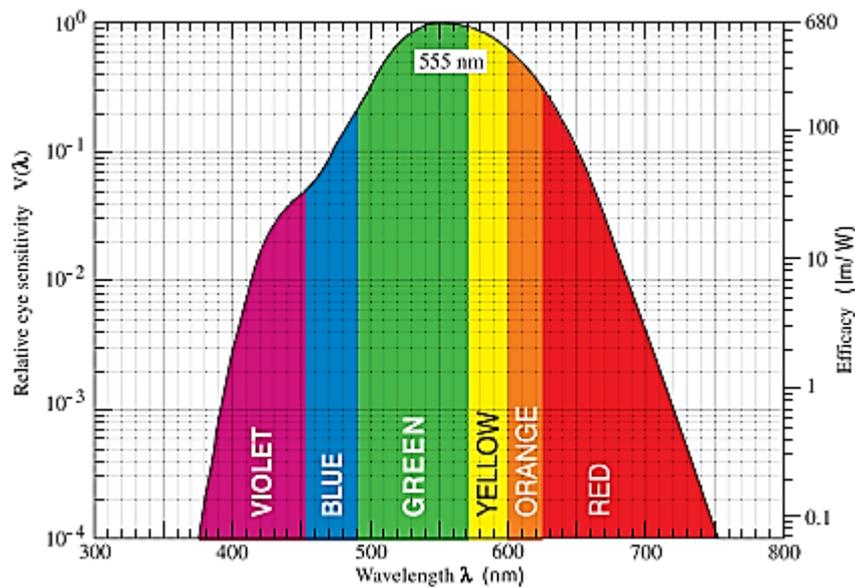


Figure 8.2: Relative Eye Sensitivity and Luminous Efficacy

Three primary considerations to ensure energy efficiency in lighting systems are:

- i. Selection of the most efficient light source possible in order to minimize electricity consumption and cost.
- ii. Matching the proper lamp type to the intended work task or aesthetic application, consistent with color, brightness control and other requirements.
- iii. Establishing adequate light levels without compromising productivity improve security and increase safety.

## 8.2 Basic Parameters and Terms in Lighting System

**Luminous flux:** The luminous flux describes the quantity of light emitted by a light source. It is a measure of a lamp's economic efficiency.

The most common measurement or unit of luminous flux is the lumen (lm).

The lumen rating of a lamp is a measure of the total light output of the lamp. Light sources are labeled with an output rating in lumens.



**Illuminance (E):** is the quotient of the luminous flux incident on an element of the surface at a point of surface containing the point, by the area of that element.

The lighting level produced by a lighting installation is usually qualified by the illuminance produced on a specified plane. In most cases, this plane is the major plane of the tasks carried out in the interior and is commonly called the working plane. The illuminance provided by an installation affects both the performance of the tasks and the appearance of the space. Lux (lx) is the metric unit of measure for illuminance of a surface. One lux is equal to one lumen per square meter. Illuminance decreases by the square of the distance (inverse square law).

The inverse square law defines the relationship between the illuminance from a point source and distance. It states that the intensity of light per unit area is inversely proportional to the square of the distance from the source (essentially the radius).

$$E = \frac{I}{d^2}$$

Where, E = Illuminance in lux (lm/m<sup>2</sup>), I = Luminous flux in lumen (lm) and d = distance in m

An alternate form of this equation which is sometimes more convenient is:

$$E_1 d_1^2 = E_2 d_2^2$$

Distance is measured from the test point to the first luminating surface - the filament of a clear bulb or the glass envelope of a frosted bulb.

### Example 8.1

The illuminance is 10 lm/m<sup>2</sup> from a lamp at 1 meter distance. What will be the illuminance at half the distance?

**Solution:**

$$\begin{aligned} E_{(1m)} &= (d_2 / d_1)^2 * E_2 \\ &= (1.0 / 0.5)^2 * 10.0 \\ &= 40 \text{ lm/m}^2 \end{aligned}$$

**Average maintained illuminance:** is the average of illuminance (lux) levels measured at various points in a defined area.

**Circuit Watts:** is the total power drawn by lamps and ballasts in a lighting circuit under assessment.

**Luminous Efficacy (lm/W):** is the ratio of luminous flux emitted by a lamp to the power consumed by the lamp. It is a reflection of efficiency of energy conversion from electricity to light form. Unit: lumens per lamp Watt (lm/W).

**Lamp Circuit Efficacy:** is the amount of light (lumens) emitted by a lamp for each Watt of power consumed by the lamp circuit, i.e. including control gear losses. This is a more meaningful measure for those lamps that require control gear. Unit: lumens per circuit Watt (lm/W).

**Installed Load Efficacy:** is the average maintained illuminance provided on a horizontal working plane per circuit watt with general lighting of an interior. Unit: lux per Watt per square metre (lux/W/m<sup>2</sup>).

**Installed Power Density:** The installed power density per 100 lux is the power needed per square meter of floor area to achieve 100 lux of average maintained illuminance on a horizontal working plane with general lighting of an interior. Unit: Watts per square metre per 100 lux (W/m<sup>2</sup>/100lux)

**Color rendering index (CRI):** is a measure of the effect of light on the perceived color of objects. To determine the CR1 of a lamp, the color appearances of a set of standard color chips are measured with special equipment under a reference light source with the same correlated color temperature as the lamp being evaluated. If the lamp renders the color of the chips identical to the reference light source, its CR1 is 100. If the color rendering differs from the reference light source, the CR1 is less than 100. A low CR1 indicates that some colors may appear unnatural when illuminated by the lamp.

**Luminaire:** is a device that distributes filters or transforms the light emitted from one or more lamps. The luminaire includes all the parts necessary for fixing and protecting the lamps, except the lamps themselves. In some cases, luminaires also include the necessary circuit auxiliaries, together with the means for connecting them to the electric supply. The basic physical principles used in optical luminaire are reflection, absorption, transmission and refraction.

**Control gear:** The gears used in the lighting equipment are as follows:

- Ballast is a current limiting device, to counter negative resistance characteristics of any discharge lamps. In case of fluorescent lamps, it aids the initial voltage build-up, required for starting. In an electric circuit the ballast acts as a stabilizer.

Fluorescent lamp is basically an electric discharge lamp with two electrodes separated inside a tube with no apparent connection between them. When sufficient voltage is impressed on these electrodes, electrons are driven from one electrode and attracted to the other. The current flow takes place through an atmosphere of low-pressure mercury vapor.

- Since the fluorescent lamps cannot produce light by direct connection to the power source, they need an ancillary circuit and device to get started and remain illuminated. The auxiliary circuit housed in a casing is known as ballast.
- Ignitors are used for starting high intensity discharge lamps such as metal halide and sodium vapor lamps. Ignitors generate a high voltage pulse or a series of pulses to initiate the discharge.

### 8.3 Light Source and Lamp Types

Lamp is equipment, which produces light. Light is that part of the electromagnetic spectrum that is perceived by our eyes. A number of light sources are available, each with its own unique combination of operating characteristics viz., efficacy, color, lamp life, and the percent of output that a lamp loses over its life.

Based on the construction and operating characteristics, the lamps can be categorized into three groups: incandescent, fluorescent and high intensity discharge (HID) lamps. HID lamps can be further classified as sodium vapor, mercury vapor and metal halide lamps. The most commonly used lamps are described briefly as follows

#### 1) Incandescent lamp

The principal parts of an incandescent lamp also known as GLS lamp (General Lighting Service lamp) include the filament, the bulb, the fill gas or vacuum and the cap. Incandescent lamps (Figure 8.3 A&B) produce light by means of a wire or filament heated to incandescence by the flow of electric current through it. The filament is enclosed in an evacuated glass bulb filled with inert gas such as argon, krypton, or nitrogen that helps to increase the brilliance of lamp and to prevent the filament from burning out.

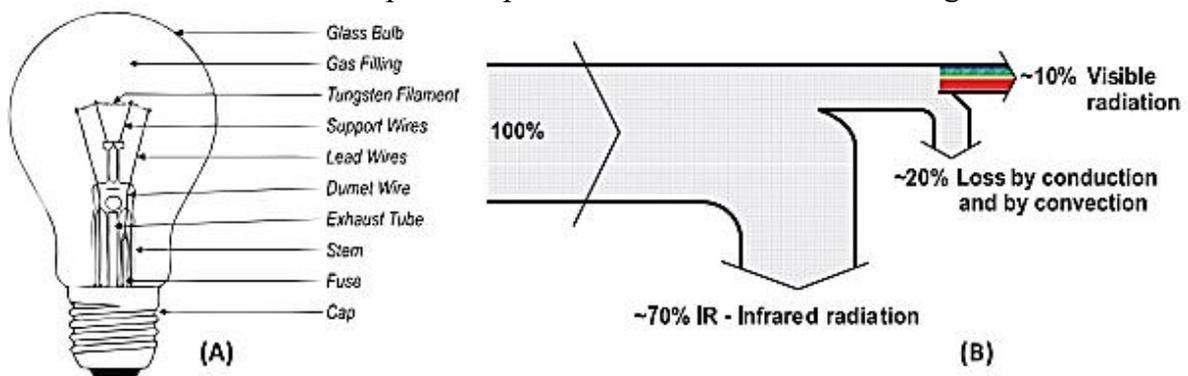
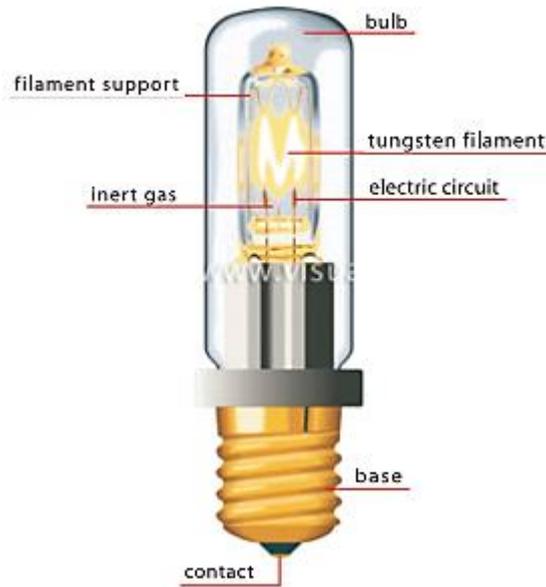


Figure 8.3: Incandescent Lamp and Energy Flow Diagram

**Reflector lamps:** Reflector lamps are basically incandescent, provided with a high quality internal mirror, which follows exactly the parabolic shape of the lamp. The reflector is resistant to corrosion, thus making the lamp maintenance free and output efficient.

## 2) Halogen lamp

It has a tungsten filament and the bulb filled with halogen gas (Figure 8.4). Current flows through the filament and heats it up, as in incandescent lamps. These lamps therefore generate a relatively large amount of heat. The use of halogen increases the efficiency and extends the service life compared with traditional incandescent lamps. Low-voltage types are very small and are ideal for precise direction of light, but they require a transformer.



*Figure 8.4: Halogen Lamp*

Tungsten atoms evaporate from the hot filament and move toward the cooler wall of the bulb. Tungsten, oxygen and halogen atoms combine at the bulb-wall to form tungsten oxyhalide molecules. The bulb-wall temperature keeps the tungsten oxyhalide molecules in a vapor. The molecules move toward the hot filament where the higher temperature breaks them apart. Tungsten atoms are re-deposited on the cooler regions of the filament - not in the exact places from which they evaporated. Breaks usually occur near the connections between the tungsten filament and its molybdenum lead-in wires where the temperature drops sharply.

## 3) Fluorescent tube lamp (FTL)

It works by the fluorescence principle. A fluorescent lamp (Figure 8.5 A&B) is a glass tube containing a small trace of a gas such as mercury vapor (for a white color), carbon dioxide (for green), neon (for red color), etc., with a special fluorescent / phosphorescent coating on the interior surface of the tube.

It contains two filaments, one at each end of the tube and when the electrical supply is switched ON, the contacts of the starter open and the filaments glow to heat up the gas contained inside the tube.

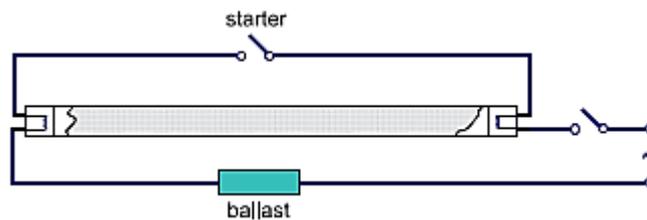
This action provides a voltage across its electrodes that set off an electric (gaseous mercury) arc discharge in the tube. This generates invisible UV radiation that is high enough to ionize the warmed-up gas inside the tube. This ionized gas also called as “plasma”, excites the fluorescent coating so that it gives out visible light. Ballast is needed

to start and operate fluorescent lamps, because of the characteristics of a gaseous arc. The luminous flux is highly dependent on the ambient temperature. Fluorescent Lamps are about 3 to 5 times as efficient as standard incandescent lamps and can last about 10 to 20 times longer.

**The different types of fluorescent lamps and their reference are given below:**

**Linear tubes**

- T12 - 38 mm (1.5”diameter)
- T8 - 25 mm (1”diameter)
- T5 - 16mm (5/8”diameter)
- T2 - 6 mm (1/4”diameter)



*Figure 8.5 (A): Fluorescent Tube Lamp*

**U-bent tubes**

- T12- 38 mm (1.5”diameter)
- T8 - 25 mm (1” diameter)

**Circular tubes**

- T9 - 38 mm (1.5”diameter)
- T5 - 16 mm (5/8”diameter)

These four lamps vary in diameter (ranging from 1.5 inches that is 12/8 of an inch for T12 to 0.625 or 5/8 of an inch in diameter for T5 lamps). Efficacy is another area that distinguishes one from another. T5 & T8 lamps offer a 5-percent increase in efficacy over 40-watt T12 lamps, and have become the most popular choice for new installations.

**4) Compact fluorescent lamp (CFL)**

Compact Fluorescent lamps (Figure 8.6) are compact / miniature versions of the linear or circular fluorescent lamps and operate in a very similar way. The luminous flux depends on temperature. CFL’s use less power and have a longer rated life compared to an incandescent lamp.



*Figure 8.6: Compact Fluorescent Lamp*

They are designed to replace an incandescent lamp and can fit into most existing light fixtures formerly used for incandescent. CFL's are available in screw type/ pin type which fit into standard sockets, and gives off light that is similar to common fluorescent lamps.

## 5) Sodium vapour lamp

### Low pressure sodium vapour lamp

Although low pressure sodium vapour (LPSV) lamps (Figure 8.7) are similar to fluorescent systems (because they are low pressure systems), they are commonly included in the HID family. LPSV lamps are the most efficacious light sources, but they produce the poorest quality light of all the lamp types. Being a monochromatic light source, all colours appear black, white, or shades of gray under an LPSV source. LPSV lamps are available in wattages ranging from 18-180.

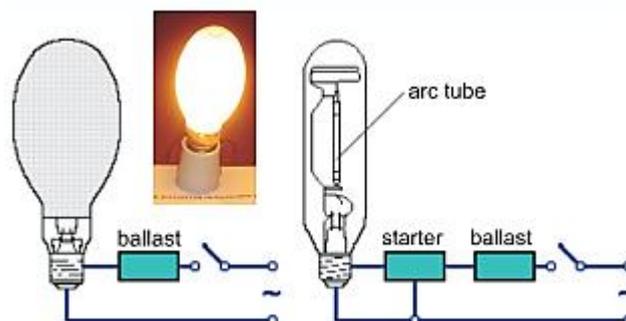


*Figure 8.7: LOW Pressure Sodium Vapour Lamp*

LPSV lamp use has been generally limited to outdoor applications such as security or street lighting and indoor, low-wattage applications where color quality is not important (e. g. stairwells). However, because the color rendition is so poor, many municipalities do not allow them for roadway lighting.

### High pressure sodium vapour lamp

The high pressure sodium vapour (HPSV) lamp (Figure 8.8 A&B) is widely used for outdoor and industrial applications as the light is yellowish. Its higher efficacy makes it a better choice than metal halide for these applications, especially when good color rendering is not a priority. HPSV lamps differ from mercury and metal-halide lamps in that they do not contain starting electrodes; the ballast circuit includes a high-voltage electronic starter. The arc tube is made of a ceramic material, which can withstand temperatures up to 1300 °C. It is filled with xenon to help start the arc, as well as a sodium-mercury gas mixture.



*Figure 8.8 (A): High Pressure Sodium Vapour Lamp*

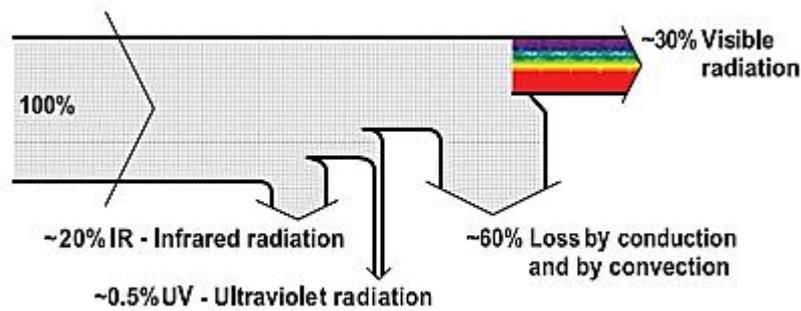


Figure 8.8 (B): Energy Flow diagram of High Pressure Sodium Vapour Lamp

## 6) Mercury vapour lamp

In a mercury vapour lamp (Figure 8.9) electromagnetic radiation is created from discharge within mercury vapour, but the regime is different than that found in the normal fluorescent lamp. During operation, the pressure within the lamp is in the range of 200 — 400 kPa (compared with only 1 Pa). It is not possible to achieve the mercury vapour discharge in a cold lamp. For this reason, the lamp also includes argon, and the initial arc is struck as an argon arc. The energy from this discharge vapourises the mercury to get the main discharge going.

The mercury vapour lamp produces a much greater proportion of visible light than fluorescent lamp and gives off a bluish white light. Phosphor coating can be given to improve the colour rendering index.

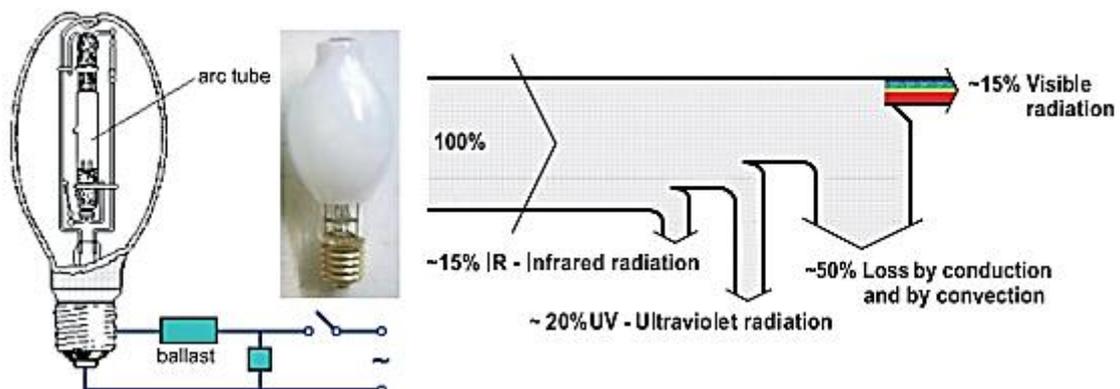


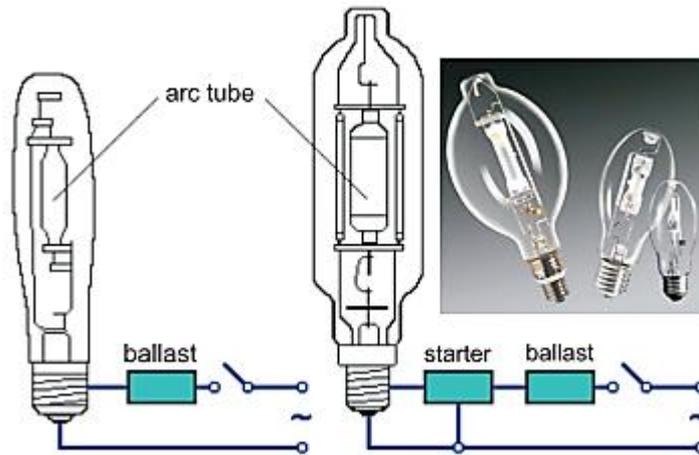
Figure 8.9: Mercury Vapour Lamp and Energy Flow Diagram

## 7) Metal halide lamp:

Metal halide lamp (Figure 8.10 A&B) can be considered as a variant of high pressure mercury vapour lamp (HPMV). In addition to mercury vapour and argon, this lamp contains metal halide. The halides can be a mixture of rare earth halides, usually iodides or a mixture of sodium and scandium iodide. The mercury vapour radiation is augmented by that of the metals.

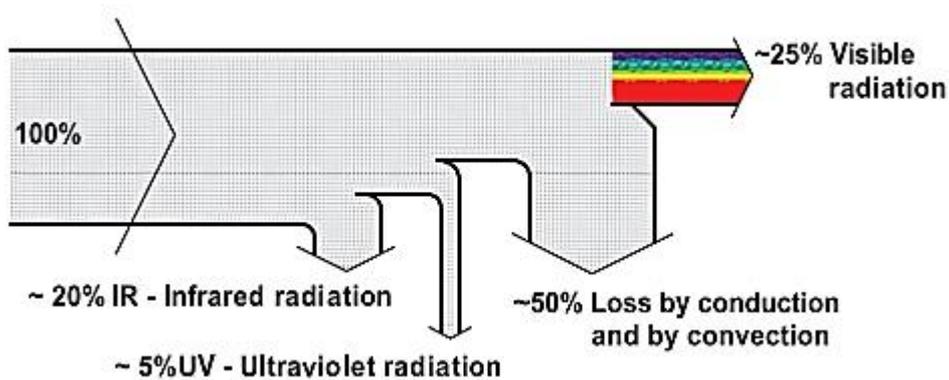
A highly compact electric arc is produced in a discharge tube. A starter is needed to switch on the lamp. The use of ceramic discharge tubes further improves the lamp properties. The halides act in a similar manner to the tungsten halogen cycle. As the temperature increases there is disassociation of the halide compound releasing the metal into the arc. The halides prevent the quartz wall getting attacked by the alkali metals. By adding other

metals to the mercury different spectrum can be emitted. Some lamps use a third electrode for starting, but others, especially the smaller display lamps, require a high voltage ignition pulse.



*Figure 8.10 (A): Metal Halide Lamp*

Metal halide lamps have a significantly better colour rendering index than mercury vapour and can be tailored by the choice of halides.



*Figure 8.10 (B): Energy Flow Diagram of Metal Halide Lamp*

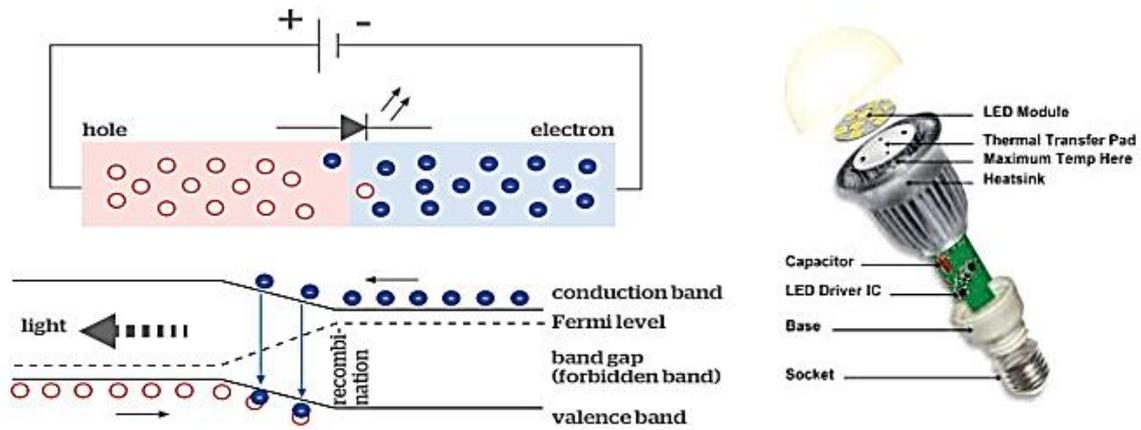
## 8) Light emitting diode (LED) lamp

LEDs produce light in a very unique way; they produce light via a process called electroluminescence

(Figure 8.11), a process that starts by turning a semiconductor material into a conducting material. A semiconductor with extra electrons is called N—type (negative) material, since it has extra negatively- charged electrons. In N—type material, free electrons can move from a negatively-charged area to a positively charged area. A semiconductor with extra holes is called P-type (positive) material since it has extra positively-charged gaps called holes. When excited with current the negative electron leaves its atom and the P-type material’s positive attraction draws the free negative electron into its hole, and the hole also moves toward the electron, so on and so forth.

As an electron travels to a hole, it carries energy, but in order to fit into the hole it must release any extra energy, and when it does, the extra energy is released in the form of

light. When we maintain a steady flow of electrical current to the diode, it continues the process of allowing electrons to flow from the negative charged material and fall into the positive charged holes which maintains a steady stream of light out of the LED. The actual LED is quite small in size, usually less than one square millimeter. Additional optical components are added to shape and direct the light. LED's are made of number of inorganic semiconductor materials, many of which produce different colour of light.



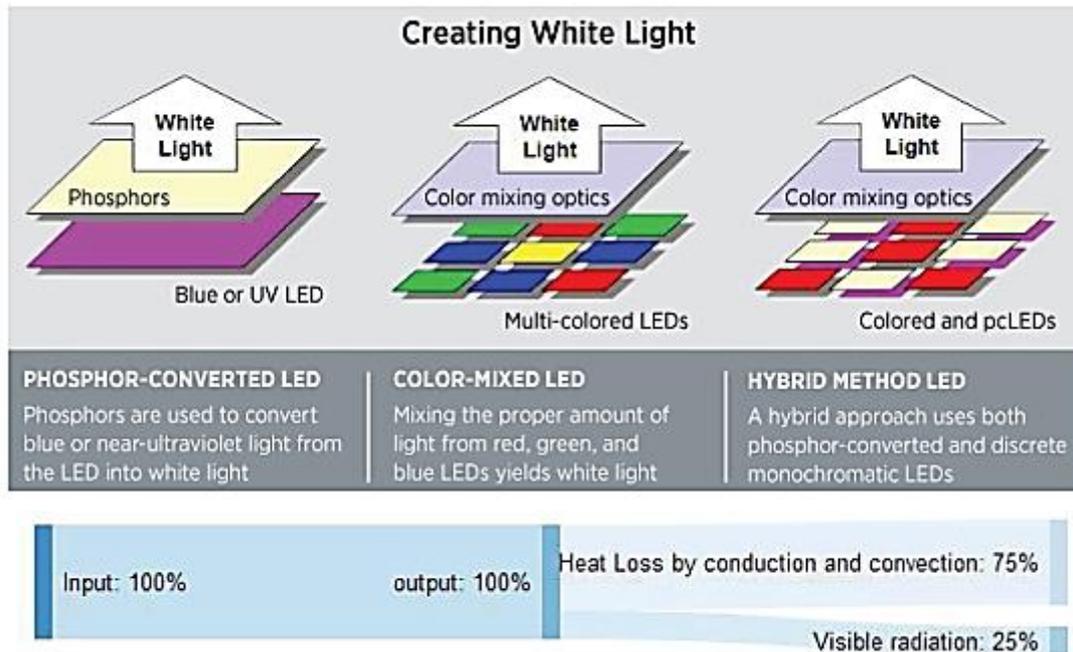
**Figure 8.11(A): Representation of LED light**

The efficiency of LED's has now risen sharply and is currently up to 200 lumens per watt in the laboratory and in some products available on the market (although more typical LED's average output varies from 50 to 130 lumens per watt).

Because of the low power requirement for LED's, using solar panels becomes more practical and less expensive than running an electrical wire or using a generator. Hence LED with battery backup for remote application is very economical. They do not radiate light in 360 degrees as an incandescent does. The light will be bright wherever it is focused.

Unlike incandescent and fluorescent lamps, LEDs are not inherently white light sources. Instead, LEDs emit nearly monochromatic light, making them highly efficient for colored light applications such as traffic lights and exit signs. However, to be used as a general light source, white light is needed. White light can be achieved with LEDs in three ways:

Phosphor conversion, in which a phosphor is used on or near the LED to convert the colored light to white light; RGB systems, in which light from multiple monochromatic LEDs (red, green, and blue) is mixed, resulting in white light; and a hybrid method, which uses both phosphor-converted and monochromatic LEDs.



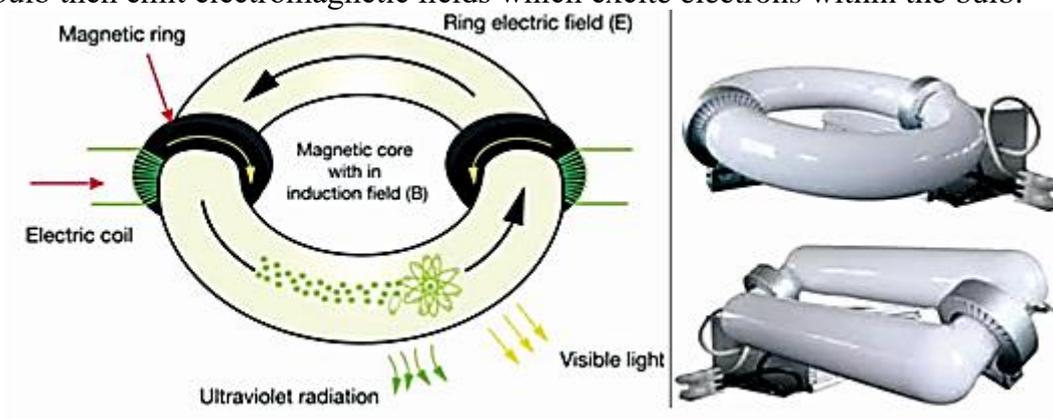
**Figure 8.11 (B): Energy Flow diagram of LED Lamp**

Advantages of LED technology is as follows Low power consumption, Directional light output, High efficiency level, Long life: up to ~100,000 hour life if junction temperature can be controlled, Instant switching on with no warm up time, High resistance to switching cycles, High impact and vibration resistance, No UV or IR radiation, Color control ability, allows dimming and Mercury free

LED's also offer a number of promising environmental benefits, and they are often viewed as the future of green lighting.

### 9) Induction lamp

Induction lamp is noted for 'crisp white light output'. Uses a magnetic field to excite gases — has no lamp parts to wear out. It consists of two main components: ballast and a sealed gas-filled bulb. Light is produced via electromagnetic induction, without an electrode or any electrical connection inside the bulb. Instead, high frequency electromagnetic fields are induced from outside the sealed chamber. To produce light, the ballast supplies the electric coils with high frequency electrical current. The ferrite magnets on either side of the bulb then emit electromagnetic fields which excite electrons within the bulb.



**Figure 8.12: Representation of Induction Lamp**

As the electrons accelerate inside the bulb, they collide with mercury atoms and produce ultraviolet (UV) light radiation. The UV light then causes the special phosphor coating inside the glass to react in a way that produces fluorescent light within the visible spectrum. The light produced by Induction Lighting (Figure 8.12) achieves good Color Rendering Index (CRI), with a Correlated Color Temperature.

Advantages of Induction lamps is as follows long burning hours, very less maintenance required, instant on/ instant re—strike and energy efficient lighting.

**Lamp features:** The Table 8.1 shows the lamp features of different lamps.

<i>Table 8.1 Luminous performance characteristic of commonly used Luminaries</i>					
Type of Lamp	Lumens / Watt		Color Rendering Index	Typical Application	Typical life (hours)
	Range	Avg.			
Incandescent	8-18	14	Excellent (100)	Homes, restaurants, general lighting, emergency lighting	1000
Fluorescent lamps	46-60	50	Good w.r.t. coating (67-77)	Offices, shops, hospitals, homes	5000
Compact fluorescent lamps (CFL)	40-70	60	Very good (85)	Hotels, shops, homes, offices	8000-10000
High pressure mercury (HPMV)	44-57	50	Fair (45)	General lighting in factories, garages, car parking, flood lighting	5000
Halogen lamps	18-24	20	Excellent (100)	Display, flood lighting, stadium exhibition grounds, construction areas	2000-4000
High pressure sodium (HPSV) SON	67-121	90	Fair (22)	General lighting in factories, ware houses, street lighting	6000-12000
Low pressure sodium (LPSV) SOX	101-175	150	Poor (10)	Roadways, tunnels, canals, street lighting	6000-12000
Metal halide lamps	75-125	100	Good (70)	Industrial bays, spot lighting, flood lighting, retail stores	8000
LED lamps	50-130	90	Very good (80)	Office, industry, outdoor, retail, hospitality, etc	30,000-60,000
Induction Lamps	65-90	75	Very good (80)	General lighting, factories, warehouse, street lighting, flood lighting, etc	60,000-1,00,000

## 8.4 Recommended Illuminance Levels for Various Tasks/Activities/Locations

### Recommendations on Illuminance

Scale of Illuminance: The minimum illuminance for all non-working interiors, has been mentioned as 20 Lux (as per IS 3646). A factor of approximately 1.5 represents the smallest significant difference in subjective effect of illuminance. Therefore, the following scale of illuminances is recommended.

20—30—50—75—100—150—200—300—500—750—1000—  
1500—2000, ... Lux

Illuminance ranges: Because circumstances may be significantly different for different interiors used for the same application or for different conditions for the same kind of activity, a range of illuminances is recommended for each type of interior or activity intended of a single value of illuminance. Each range consists of three successive steps of the recommended scale of illuminances. For working interiors the middle value (R) of each range represents the recommended service illuminance that would be used unless one or more of the factors mentioned below apply.

The higher value (H) of the range should be used at exceptional cases where low reflectance or contrasts are present in the task, errors are costly to rectify, visual work is critical, accuracy or higher productivity is of great importance and the visual capacity of the worker makes it necessary. Similarly, lower value (L) of the range may be used when reflectances or contrasts are unusually high, speed and accuracy is not important and the task is executed only occasionally.

### Recommended Illumination

The following Table 8.2 gives the recommended illuminance range for different tasks and activities for chemical sector. The values are related to the visual requirements of the task, to user's satisfaction, to practical experience and to the need for cost effective use of energy (Source IS 3646 (Part I): 1992).

For recommended illumination in other sectors, reader may refer Illuminating Engineers Society *Recommendations Handbook*.

*Table 8.2 Recommended illuminance range for different tasks and activities for chemical sector*

<b>Petroleum, Chemical and Petrochemical works</b>	
Exterior walkways, platforms, stairs and ladders	30-50-100
Exterior pump and valve areas	50-100-150
Pump and compressor houses	100-150-200
Process plant with remote control	30-50-100
Process plant requiring occasional manual intervention	50-100-150
Permanently occupied work stations in process plant	150-200-300
Control rooms for process plant	200-300-500

<b>Pharmaceuticals Manufacturer and Fine chemicals manufacturer</b>	
<b>Pharmaceutical manufacturer</b>	
Grinding, granulating, mixing, drying, tableting, sterilising, washing, preparation of solutions, filling, capping, wrapping, hardening	300-500-750
<b>Fine chemical manufacturers</b>	
Exterior walkways, platforms, stairs and ladders	30-50-100
Process plant	50-100-150
Fine chemical finishing	300-500-750
Inspection	300-500-750
Soap manufacture	
General area	200-300-500
Automatic processes	100-200-300
Control panels	200-300-500
Machines	200-300-500
<b>Paint works</b>	
General	200-300-500
Automatic processes	150-200-300
Control panels	200-300-500
Special batch mixing	500-750-1000
Colour matching	750-100-1500

### 8.5 Methods of Calculating Illuminance - Lighting Design for Interiors

In order to design a luminaire layout that best meets the illuminance and uniformity requirements of the job, two types of information are generally needed: average illuminance level and illuminance level at a given point. Calculation of illuminance at specific points is often done to help the designer evaluate the lighting uniformity, especially when using luminaires where maximum spacing recommendations are not supplied, or where task lighting levels must be checked against ambient level.

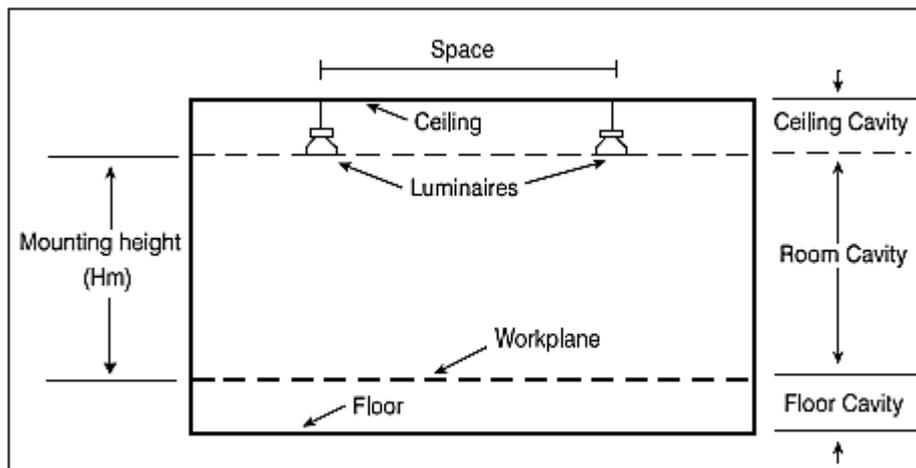
If average levels are to be calculated, two methods can be applied:

1. For indoor lighting situations, the Zonal Cavity Method is used with data from a coefficient of utilization table.
2. For outdoor lighting applications, a coefficient of utilization curve is provided, the CU is read directly from the curve and the standard lumen formula is used.

### Zonal Cavity Method for Indoor Lighting Calculations

The Zonal Cavity Method (sometimes called the Lumen Method) is the currently accepted method for calculating average illuminance levels for indoor areas, unless the light distribution is radically asymmetric. It is an accurate hand method for indoor applications because it takes into consideration the effect that inter-reflectance has on the level of illuminance.

Although it takes into account several variables, the basic premise that foot-candles are equal to luminous flux over an area is not violated.



*Figure 8.13 Typical Room Space*

The basis of the Zonal Cavity Method is that a room is made up of three spaces or cavities. The space between the ceiling and the fixtures, if they are suspended, is defined as the “ceiling cavity”; the space between the work plane and the floor, the “floor cavity”; and the space between the fixtures and the work plane, the “room cavity.”

#### Example 8.2

The step by step process of lighting design is illustrated below with the help of an example. The Figure 8.13 shows the parameters of a typical space.

**Step-1:** Decide the required illuminance on work plane, the type of lamp and luminaire

A preliminary assessment must be made of the type of lighting required, a decision most often made as a function of both aesthetics and economics. For normal office work, illuminance of 200 lux is desired.

For an air conditioned office space under consideration, we choose 36 W fluorescent tube lights with twin tube fittings. The luminaire is porcelain-enameled suitable for the above

lamp. It is necessary to procure utilisation factor tables for this luminaire from the manufacturer for further calculations.

**Step-2:** Collect the room data in the format given below

Room dimensions	Length	L1	10	m
	Width	L2	10	m
	Floor area	L3	100	m <sup>2</sup>
	Ceiling height	L4	3.0	m
Surface reflectance	Ceiling	L5	0.7	p.u
	Wall	L6	0.5	p.u
	Floor	L7	0.2	p.u
Work plane height from floor		L8	0.9	m
Luminaire height from floor		L9	2.9	m

Typical reflectance values for using in L5, L6, L7 are:

	Ceiling	Walls	Floor
Air conditioned office	0.7	0.5	0.2
Light industrial	0.5	0.3	0.1
Heavy industrial	0.3	0.2	0.1

**Step-3:** Calculate room index

$$\begin{aligned}
 \text{Room Index} &= \frac{\text{Length} \times \text{Width}}{\text{Mounting Height} \times (\text{Length} + \text{Width})} \\
 &= \frac{L1 \times L2}{(L9 - L8) \times (L1 + L2)} = \frac{10 \times 10}{2 \times (10 + 10)} \\
 &= 2.5
 \end{aligned}$$

**Step 4:** Calculate the utilisation factor

Utilisation factor is defined as the percent of rated bare-lamp lumens that exit the luminaire and reach the work plane. It accounts for light directly from the luminaire as well as light reflected off the room surfaces. Manufacturers will supply each luminaire with its own CU table derived from a photometric test report.

Using tables available from manufacturers, it is possible to determine the utilisation factor for different light fittings if the reflectance of both the walls and ceiling is known, the room index has been determined and the type of luminaire is known. For twin tube fixture, utilisation factor is 0.66, corresponding to room index of 2.5.

**Step-5:** To calculate the number of fittings required, the following formula is used

$$N = \frac{E \times A}{F \times UF \times LLF}$$

Where,

N = Number of Fittings

E = Lux Level Required on Working Plane

A = Area of Room (L x W)

F = Total Flux (Lumens) from all the Lamps in one Fitting

UF = Utilisation Factor from the Table for the Fitting to be Used

LLF = Light Loss Factor. This takes account of the depreciation over time of lamp output and dirt accumulation on the fitting and walls of the building.

Light output = 3050 Lumens (Single Lamp)

LLF = Lamp lumen MF x Luminaire MF x Room surface MF

Where, MF = Maintenance Factor

Typical LLF values

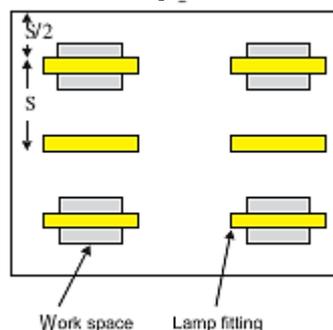
Air conditioned office	0.8
Clean industrial	0.7
Dirty industrial	0.6

$$N = \frac{200 \times 100}{2 \times 3050 \times 0.66 \times 0.8} = 6.2$$

So, 6 Numbers of Twin Tube Fixtures are required. Total number of 36 W lamp is 12.

Step 6: Space the luminaires to achieve desired uniformity

Every luminaire will have a recommended space to height ratio. In earlier design methodologies, the uniformity ratio, which is the ratio of minimum illuminance to average illuminance, was kept at 0.8 and suitable space to height ratio is specified to achieve the uniformity. In modern designs incorporating energy efficiency and task lighting the emerging concept is to provide a uniformity of 1/3 to 1/10 depending on the tasks.



**Figure 8.14: Arrangement of Fittings**

Recommended value for the above luminaire is 1.5. If the actual ratio is more than the recommended values, the uniformity of lighting will be less.

For a sample of arrangement of fittings, refer Figure 8.14. The luminaire closer to a wall should be one half of spacing or less.

### Luminaire Spacing

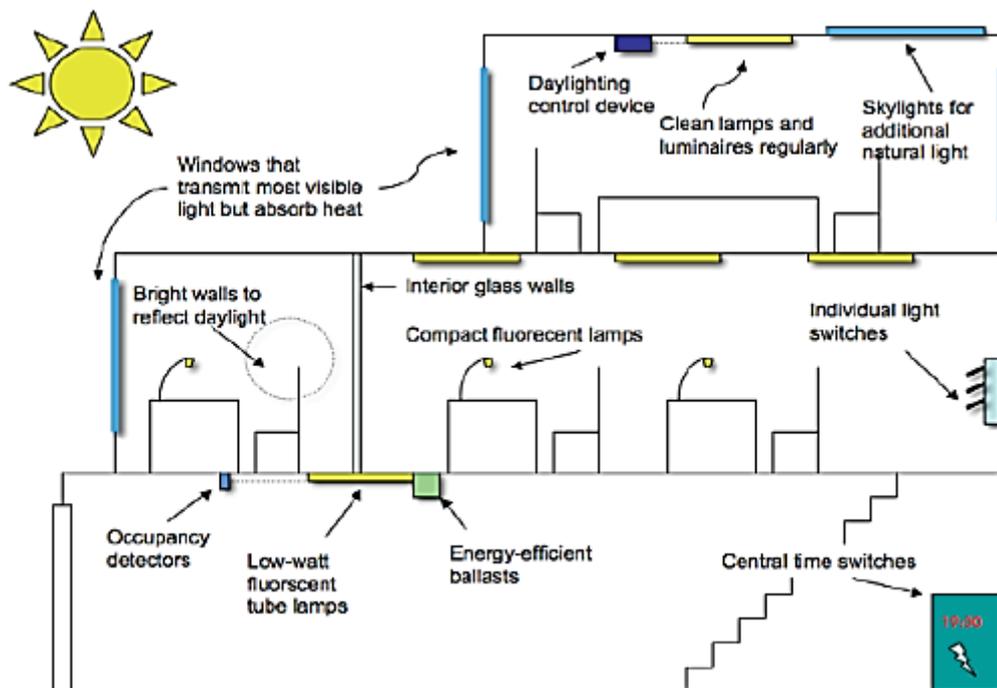
Spacing between luminaires =  $10/3 = 3.33$  m  
 Mounting height (L9-L8) = 2.0 m  
 Space to height ratio (SHR) =  $3.33/2.0 = 1.66$

This is close to the limits specified and hence accepted.

It is better to choose luminaires with larger SHR. This can reduce the number of fittings and connected lighting load.

### 8.6 General Energy Saving Opportunities

Changing the light bulbs is not the only way to improve the use of lighting. Below are some examples of many other options available:



#### a) Use natural day lighting

The utility of using natural day lighting instead of electric lighting during the day is well known, but is being increasingly ignored especially in modern air-conditioned office spaces and commercial establishments like hotels, shopping plazas etc. Industrial plants generally use daylight in some fashion, but improperly designed day lighting systems can

result in complaints from personnel or supplementary use of electric lights during daytime.

**Some of the methods to incorporate day lighting are:**

- i. North lighting by use of single-pitched truss of the saw-tooth type is a common industrial practice; this design is suitable for latitudes north of 23 i.e. in North India. In South India, north lighting may not be appropriate unless diffusing glasses are used to cut out the direct sunlight.
- ii. Innovative designs are possible which eliminates the glare of daylight and blend well with the interiors. Glass strips, running continuously across the breadth of the roof at regular intervals, can provide good, uniform lighting on industrial shop floors and storage bays.
- iii. A good design incorporating sky lights with FRP material along with transparent or translucent false ceiling can provide good glare-free lighting; the false ceiling will also cut out the heat that comes with natural light.
- iv. Use of atrium with FRP dome in the basic architecture can eliminate the use of electric lights in passages of tall buildings.
- v. Natural Light from windows should also be used. However, it should be well designed to avoid glare. Light shelves can be used to provide natural light without glare.
- vi. Mounting Solar tube on the roof, with the help of advanced optics and special duct work to direct sunlight deep into the buildings and spreading out over large internal spaces providing heat and glare free day lighting for 8-10 hrs in a day

**b) De-lamping to reduce excess lighting**

De—lamping is an effective method to reduce lighting energy consumption. In some industries, reducing the mounting height of lamps, providing efficient luminaires and then de—lamping has ensured that the illuminance is hardly affected. De—lamping at empty spaces where active work is not being performed is also a useful concept.

**c) Task lighting**

Task Lighting implies providing the required good illuminance only in the actual small area where the task is being performed, while the general illuminance of the shop floor or office is kept at a lower level; e. g. Machine mounted lamps or table lamps. Energy saving takes place because good task lighting can be achieved with low wattage lamps. The concept of task lighting if sensibly implemented, can reduce the no of general lighting fixtures, reduce the wattage of lamps, save considerable energy and provide better illuminance and also provide aesthetically pleasing ambience.

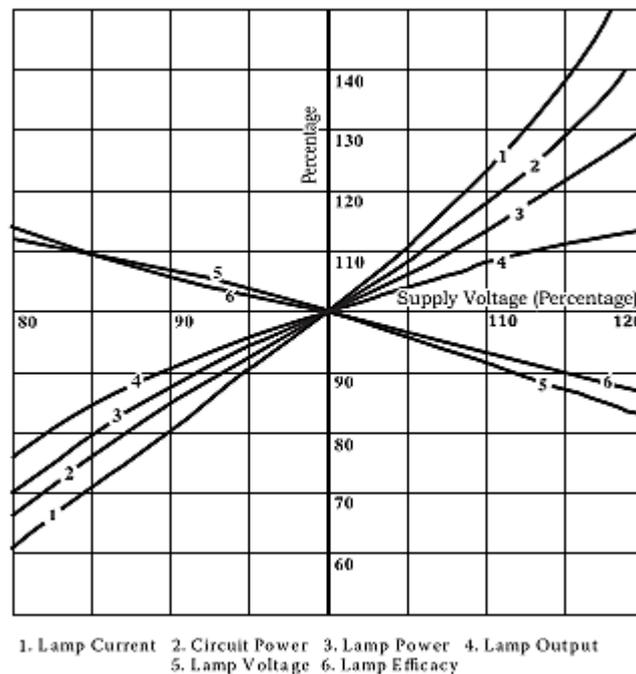
**(d) Selection of high efficiency lamps and luminaires**

The details of common types of lamps are summarised in Table 8.1 above. It is possible to identify energy saving potential for lamps by replacing with more efficient types. The following examples of lamp replacements are common. There may be some limitations if colour rendering is an important factor. It may be noted that, in most cases, the luminaires and the control gear would also have to be changed. The savings are large if the lighting scheme is redesigned with higher efficacy lamps and luminaires.

**e) Reduction of lighting feeder voltage**

Figure 8.15 shows the effect of variation of voltage on light output and power consumption for fluorescent tube lights. Similar variations are observed on other gas discharge lamps like mercury vapour lamps, metal halide lamps and sodium vapour lamps (Table 8.3 summarises the effects). Hence reduction in lighting feeder voltage can save energy, provided the drop in light output is acceptable.

In many areas, night time grid voltages are higher than normal; hence reduction in voltage can save energy and also provide the rated light output. Some manufacturers are supplying reactors and transformers as standard products. A large number of industries have used these devices and have reported saving to the tune of 5% to 15%. Industries having a problem of higher night time voltage can get an additional benefit of reduced premature lamp failures.



**Figure 8.15 Effect of Voltage Variation on Fluorescent Tube light Parameters**

**Table 8.3 Variation in Light Output and Power Consumption**

Particulars	10% lower voltage	10% higher voltage
Fluorescent lamps		
Light output	Decreases by 9 %	Increases by 8 %
Power input	Decreases by 15 %	Increases by 8 1%
HPMV lamps		
Light output	Decreases by 20 %	Increases by 20 %
Power input	Decreases by 16 %	Increases by 17 %
Mercury Blended lamps		
Light output	Decreases by 24 %	Increases by 30 %
Power input	Decreases by 20 %	Increases by 20 %
Metal Halide lamps		

#### f) Electronic ballasts

Conventional electromagnetic ballasts (chokes) are used to provide higher voltage to start the tube light and subsequently limit the current during normal operation. Electronic ballasts are oscillators that convert the supply frequency to about 20,000 Hz to 30,000 Hz. The basic functions of electronic ballast are:

- To ignite the lamp
- To stabilize the gas discharge
- To supply the power to the lamp

The losses in electronic ballasts for tube lights are only about 1 Watt, in place of 10 to 15 Watts in standard electromagnetic chokes.

The additional advantage is that the efficacy of tube lights improves at higher frequencies, resulting in additional savings if the ballast is optimised to provide the same light output as with the conventional choke. Hence a saving of about 15 to 20 Watts per tube light can be achieved by use of electronic ballasts. With electronic ballast, the starter is eliminated and the tube light lights up instantly without flickering.

#### g) Lighting controllers

Automatic control for switching off unnecessary lights can lead to good energy savings. This includes dimmers, motion & occupancy sensors, photo sensors and timers.

#### h) Lighting maintenance

Maintenance is vital to lighting efficiency. Light levels decrease over time because of aging lamps and dirt on fixtures, lamps and room surfaces. Together, these factors can reduce total illumination by 50 percent or more, while lights continue drawing full power. The basic maintenance includes cleaning of lamps and fixtures, cleaning and repainting interiors, re-lamping etc.

### 8.7 Energy Efficient Lighting Controls

#### Occupancy Sensors

Occupancy-linked control can be achieved using infra-red, acoustic, ultrasonic or microwave sensors, which detect either movement or noise in room spaces. These sensors switch lighting on when occupancy is detected, and off again after a set time period, when no occupancy movement detected. They are designed to override manual switches and to prevent a situation where lighting is left on in unoccupied spaces. With this type of system it is important to incorporate a built-in time delay, since occupants often remain still or quiet for short periods and do not appreciate being plunged into darkness if not constantly moving around.

### **Timed Based Control**

Timed-turnoff switches are the least expensive type of automatic lighting control. In some cases, their low cost and ease of installation makes it desirable to use them where more efficient controls would be too expensive.

### **Types and features**

The oldest and most common type of timed-turnoff switch is the “dial timer,” a spring-wound mechanical timer that is set by twisting the knob to the desired time. Typical units of this type are vulnerable to damage because the shaft is weak and the knob is not securely attached to the shaft. Some spring-wound units make an annoying ticking sound as they operate. Newer types of timed-turnoff switches are completely electronic and silent. Electronic switches can be made much more rugged than the spring-wound dial timer. These units typically have a spring-loaded toggle switch that turns on the circuit for a preset time interval. Some electronic models provide a choice of time intervals, which you select by adjusting a knob located behind the faceplate. Most models allow occupants to turn off the lights manually. Some models allow occupants to keep the lights on, overriding the timer. Timed-turnoff switches are available with a wide range of time spans. The choice of time span is a compromise. Shorter time spans waste less energy but increase the probability that the lights will turn off while someone is in the space. Dial timers allow the occupant to set the time span, but this is not likely to be done with a view toward optimising efficiency. For most applications, the best choice is an electronic unit that allows the engineering staff to set a fixed time interval behind the cover plate.

### **Daylight Linked Control**

Photoelectric cells can be used either simply to switch lighting on and off, or for dimming. They may be mounted either externally or internally. It is however important to incorporate time delays into the control system to avoid repeated rapid switching caused, for example, by fast moving clouds. By using an internally mounted photoelectric dimming control system, it is possible to ensure that the sum of daylight and electric lighting always reaches the design level by sensing the total light in the controlled area and adjusting the output of the electric lighting accordingly. If daylight alone is able to meet the design requirements, then the electric lighting can be turned off. The energy saving potential of dimming control is greater than a simple photoelectric switching system. Dimming control is also more likely to be acceptable to room occupants.

### **Localized Switching**

Localized switching should be used in applications which contain large spaces. Local switches give individual occupants control over their visual environment and also facilitate energy savings. By using localized switching it is possible to turn off artificial lighting in specific areas, while still operating it in other areas where it is required, a situation which is impossible if the lighting for an entire space is controlled from a single switch.

## Street Lighting Systems and Controls

Street lighting /Public lighting is one of the major electrical loads in municipal areas. Number of street lights used in a Municipal area varies from 20000 — 50000 in numbers depending on the kilometers of road illuminated within the municipal limits. Typical electrical load of municipal lighting system varies 2MW to 7 MW. The type of lamps used in Municipal area includes Fluorescent Tube light/ Mercury Vapor Lamps/ Sodium Vapor Lamps and Metal Halide Lamps. High Mast towers are also used at strategic junctions in the Municipal area. LEDs are also used for traffic signaling purpose in municipal areas.

*Following controls are adopted to reduce energy consumption in street lighting system:*

1. Timer control (Switch ON/OFF as per set timing)
2. Day light control(Based on illumination level)
3. Selective switching/alternate switching of street lights low traffic density areas (after midnight).
4. Switching control based on lux levels. (after midnight)
5. Installations of Voltage controllers to be operated after midnight.
6. Installation of PLC controlled Lighting panels for effective control and monitoring.

## 8.8 Lighting Case Study

Replacement of existing T12 Fluorescent lamps in street lighting system with LED lamps

**Existing:** Fluorescent lamp (T12) fixture of 40 numbers is connected to the entire campus for security purpose. All the lights remain in operation for around 12 hours at night (6 pm. to 6 am) every day throughout the year. All the light fixtures are equipped with electromagnetic ballast which consumes around 12 to 14 watt of additional power while in operation. Hence the power consumption of a single fluorescent light fixture considering minimum ballast loss is  $40+12=52$  watts. The total light output of all the fluorescent light fixtures is around 2400 lumen.

**Proposed:** It was proposed to replace existing lamps with high efficient LED lamps of 18 W with a luminous efficacy of around 120-140 lm/w. The total luminous output of these lamps is around 2340 lumen.

### Calculation

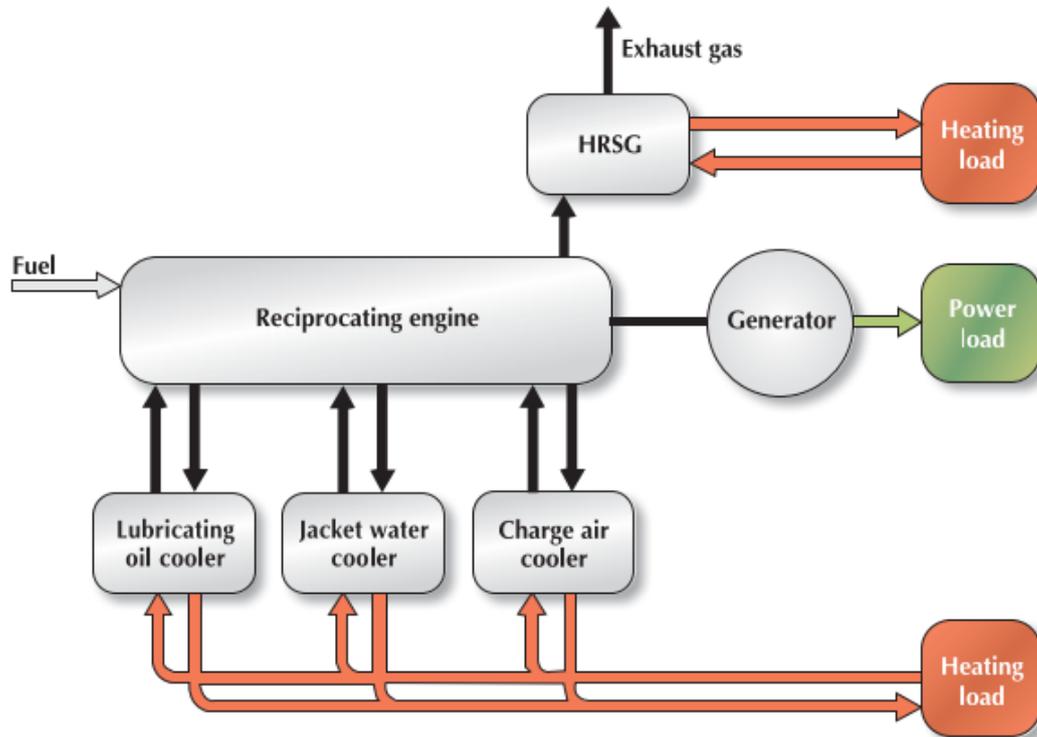
<b>Existing Fittings</b>			
No of FTL-T12 lamps installed	=	40	No's
<b>Wattage Consumed</b>			
40 No's of (40W+12W (Ballast)=52 W)	=	2.1	kW
Average operating hours per day	=	12	hrs/day

Total energy consumed by operating the lights	=	25	kWh/day
Annual energy consumption by operating the lights (365 days/year)	=	9125	kWh/annum
<b>Proposed option</b>			
Replacement of all 40 no's of 40 Watts T12-FTL with 18 Watts LED lamps			
Total energy consumed by operating with LED	=	0.72	kW
Average operating hours per day	=	12	hrs/day
Total kWh/day consumed by operating the lights	=	9	kWh/day
Annual energy consumption by operating the lights (365 days/year)	=	3285	kWh/annum
<b>Savings</b>			
Total energy reduction per annum	=	5840	kWh/annum
<i>Annual monetary savings</i> (@ BDT 5/unit)	=	<b>29,200</b>	<i>BDT/annum</i>
Investment @ BDT 2500/ Lamp	=	1.0	Lakhs
Simple payback period	=	3.4	Years

## Chapter 9: Diesel / Natural Gas Power Generating System

### 9.1 Introduction

Reciprocating engines produce electricity using a combustible fuel and generator. In addition to producing electricity, useful heat can be recovered from the exhaust gas using a heat recovery steam generator (HRSG), or heat recovery system for hot water (Figure 9.1). Heat can also be recovered from the lubricating oil cooler, the jacket water cooler and/or the charge air cooler, and this recovered “waste” heat can be provided to a heating load. In this case, the reciprocating engine power plant would be operating in a Combined Heat & Power (CHP) or cogeneration mode. The energy performance of a reciprocating engine is influenced by a number of factors such as the type of fuel, the reciprocating engine power capacity, minimum capacity, availability, heat rate and heat recovery efficiency.



*Figure 9.1 Reciprocating Engine Cogeneration*

*Figure 9.1 Reciprocating Engine Cogeneration*

There are two basic types of reciprocating engines - spark ignition and compression ignition. Spark ignition engines use a spark (across a spark plug) to ignite a compressed fuel-air mixture. Typical fuels for such engines are gasoline, natural gas and sewage and landfill gas. Compression ignition engines compress air to a high pressure, heating the air to the ignition temperature of the fuel, which then is injected. The high compression ratio used for compression ignition engines results in a higher efficiency than is possible with spark ignition engines. Diesel/heavy fuel oil is normally used in compression ignition engines, although some are dual-fueled (natural gas is compressed with the combustion air and diesel oil is injected at the top of the compression stroke to initiate combustion).

### **Diesel Engine Cycle**

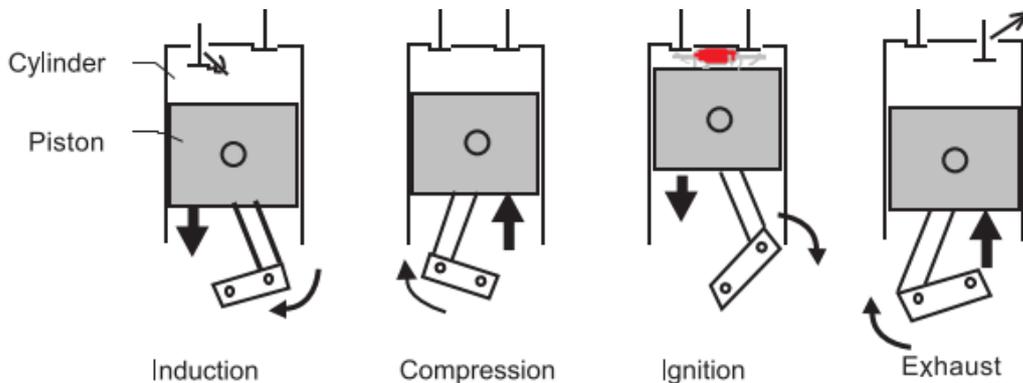
Diesel engine is the prime mover, which drives an alternator to produce electrical energy. In the Diesel engine, air is drawn into the cylinder and is compressed to a high ratio (14:1 to 25:1). During this compression, the air is heated to a temperature of 700—90000 A metered quantity of diesel fuel is then injected into the cylinder, which ignites spontaneously because of the high temperature. Hence, the diesel engine is also known as compression ignition (CI) engine. DG set can be classified according to cycle type as: two stroke and four stroke. However, the bulk of IC engines use the four stroke cycle. Let us look at the principle of operation of the four-stroke diesel engine. The 4 stroke operations in a diesel engine (Figure 9.2) are: induction stroke, compression stroke, ignition and power stroke and exhaust stroke.

1<sup>st</sup> : Induction stroke - while the inlet valve is open, the descending piston draws in fresh air.

2<sup>nd</sup> : Compression stroke - while the valves are closed, the air is compressed to a pressure of up to 25 bar.

3<sup>rd</sup> : Ignition and power stroke - fuel is injected, while the valves are closed (fuel injection actually starts at the end of the previous stroke), the fuel ignites spontaneously and the piston is forced downwards by the combustion gases.

4<sup>th</sup> : Exhaust stroke - the exhaust valve is open and the rising piston discharges the spent gases from the cylinder

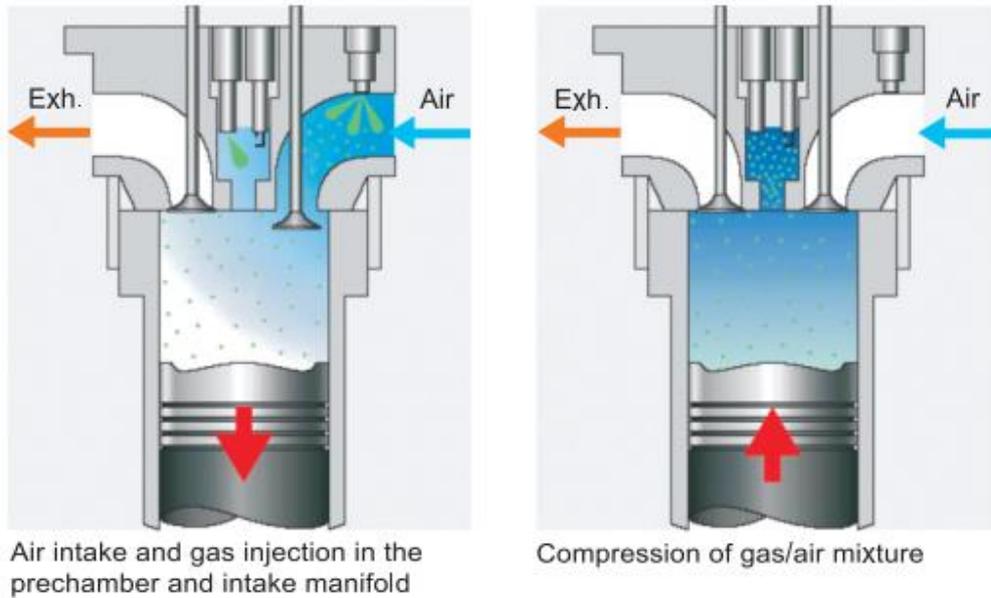


*Figure 9.2 Schematic Diagram of Four-Stroke Diesel Engine*

Since power is developed during only one stroke, the single cylinder four-stroke engine has a low degree of uniformity. Smoother running is obtained with multi cylinder engines because the cranks are staggered in relation to one another on the crankshaft. There are many variations of engine configuration, for example. 4 or 6 cylinder, in—line, horizontally opposed, vee or radial configurations.

## **Gas Engines**

A typical spark-ignited lean-burn engine is depicted in Figure 9.3. In this process, the gas is mixed with air before the inlet valves. During the intake period, gas is also fed into a small pre chamber, where the gas mixture is rich compared to the gas in the cylinder. At the end of the compression phase the gas/air mixture in the pre chamber is ignited by a spark plug. The flames from the nozzle of the pre chamber ignite the gas/air mixture in the whole cylinder. Combustion is fast. After the working phase the cylinder is emptied of exhaust and the process starts again. Reciprocating engines with modern lean-burn technology reach close to 45% electrical efficiency.



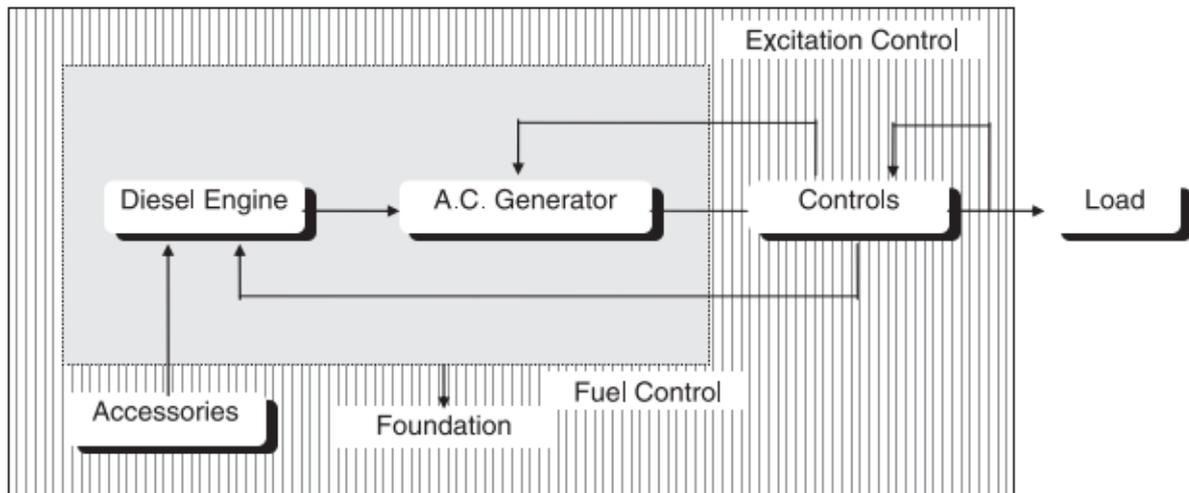
*Figure 9.3 Natural Gas Engine*

### **DG Set as a System**

A diesel generating set (Figure 9.4) should be considered as a system since its successful operation depends on the well-matched performance of the components, namely:

- a) The diesel engine and its accessories.
- b) The AC Generator.
- c) The control systems and switchgear.
- d) The foundation and power house civil works.
- e) The connected load with its own components like heating, motor drives, lighting etc.

It is necessary to select the components with highest efficiency and operate them at their optimum efficiency levels to conserve energy in this system.



*Figure 9.4 DG Set System*

### Selection Considerations

To make a decision on the type of engine, which is most suitable for a specific application, several factors need to be considered. The two most important factors are: power and speed of the engine. The power requirement is determined by the maximum load. The engine power rating should be 10-20

% more than the power demand by the end use. This prevents overloading the machine by absorbing extra load during starting of motors or switching of some types of lighting systems or when wear and tear on the equipment pushes up its power consumption. Speed is measured at the output shaft and given in revolutions per minute (RPM). An engine will operate over a range of speeds, with diesel engines typically running at lower speeds (1300 - 3000 RPM). There will be an optimum speed at which fuel efficiency will be greatest. Engines should be run as closely as possible to their rated speed to avoid poor efficiency and to prevent buildup of engine deposits due to incomplete combustion - which will lead to higher maintenance and running costs. To determine the speed requirement of an engine, one has to again look at the requirement of the load. For some applications, the speed of the engine is not critical, but for other applications such as a generator, it is important to get a good speed match. If a good match can be obtained, direct coupling of engine and generator is possible; if not, then some form of gearing will be necessary - a gearbox or belt system, which will add to the cost and reduce the efficiency. There are various other factors that have to be considered, when choosing an engine for a given application. These include the following: cooling system, abnormal environmental conditions (dust, dirt, etc.), fuel quality, speed governing (fixed or variable speed), poor maintenance, control system, starting equipment, drive type, ambient temperature, altitude, humidity, etc. Suppliers or manufacturers literature will specify the required information when purchasing an engine. The efficiency of an engine depends on various factors, for example, load factor (percentage of full load), engine size, and engine type.

## Diesel Generator Captive Power Plants

Diesel engine power plants are most frequently used in small power (captive non-utility) systems. The main reason for their extensive use is the higher efficiency of the diesel engines compared with gas turbines and small steam turbines in the output range considered. In applications requiring low captive power, without much requirement of process steam, the ideal method of power generation would be by installing diesel generator plants. The fuels burnt in diesel engines range from light distillates to residual fuel oils. Most frequently used diesel engine sizes are between the range 4 to 15 MW. For continuous operation, low speed diesel engine is more cost-effective than high speed diesel engine.

### Advantages of adopting Diesel Power Plants are:

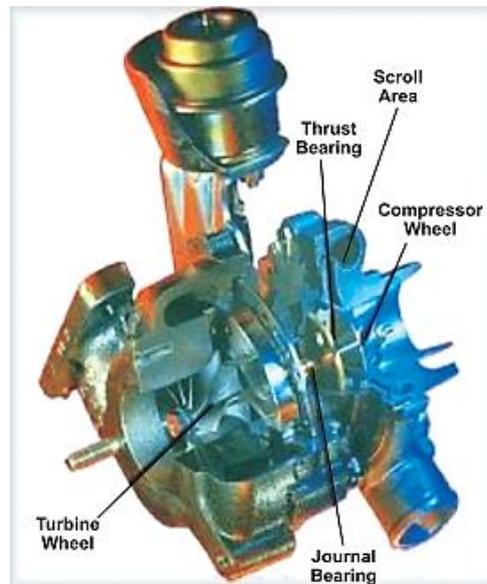
- Low installation cost
- Short delivery periods and installation period
- Higher efficiency (as high as 43 -45 %)
- More efficient plant performance under part loads
- Suitable for different type of fuels such as low sulphur heavy stock and heavy fuel oil in case of large capacities.
- Minimum cooling water requirements,
- Adopted with air cooled heat exchanger in areas where water is not available
- Short start up time

A brief comparison of different types of captive power plants (combined gas turbine and steam turbine, conventional steam plant and diesel engine power plant) is given in Table 9.1. It can be seen from the Table that captive diesel plant wins over the other two in terms of thermal efficiency, capital cost, space requirements, auxiliary power consumption, plant load factor etc.

Description	Units	Combined GT & ST	Conventional Steam Plant	Diesel Engine Power Plant
Thermal Efficiency	%	40 - 46	33 - 36	43 - 45
Initial Investment of Installed Capacity	BDT/KW	8,500- 10,000	15,000-18,000	7,500 - 9,000
Space requirement		125% (approx.)	250% (approx.)	100% (approx.)
Construction time	Months	24 - 30	42 - 48	12 - 15
Project period	Months	30 - 36	52 - 60	12
Auxiliary Power Consumption	%	2 - 4	8 - 10	1.3 - 2.1
Plant Load Factor	kWh/kW	6000 - 7000	5000 - 6000	7200 - 7500
Start Up Time from Cold	Minutes	About 10	120 - 180	15 - 20

## Diesel Engine Power Plant Developments

The diesel engine developments have been steady and impressive. The specific fuel consumption has come down from a value of 220 g/kWh in the 1970's to a value of around 160 g/kWh in present times. Slow speed diesel engine, with its flat fuel consumption curve over a wide load range (50%-100%), compares very favorably over other prime movers such as medium speed diesel engine, steam turbines and gas turbines. With the arrival of modern, high efficiency turbochargers, it is possible to use an exhaust gas driven turbine generator (Figure 9.5) to further increase the engine rated output. The net result would be lower fuel consumption per kWh and further increase in overall thermal efficiency.



*Figure: Turbocharger*

The diesel engine is able to burn the poorest quality fuel oils, unlike gas turbine, which is able to do so with only costly fuel treatment equipment. Slow speed dual fuel engines are now available using high-pressure gas injection, which gives the same thermal efficiency and power output as a regular fuel oil engine.

### 9.2 Selection and Installation Factors

#### Sizing of a Genset:

- If the DG set is required for 100% standby, then the entire connected load in HP / kVA should be added. After finding out the diversity factor, the correct capacity of a DG set can be found out.

#### Example 9.1

Connected Load	= 650 kW
Diversity Factor	= 0.54
(Demand / Connected load)	
Max. Demand	= 650 x 0.54 = 350 kW
% Loading	= 70
Set rating	= 350/0.7 = 500 kW

At 0.8 PF, rating = 625 kVA

- b) For an existing installation, record the current, voltage and power factors (kWh / kVAh) reading at the main bus-bar of the system at every half-an-hour interval for a period of 2-3 days and during this period the factory should be having its normal operations. The non-essential loads should be switched off to find the realistic current taken for running essential equipment. This will give a fair idea about the current taken from which the rating of the set can be calculated.

**For existing installation:**

$kVA = \sqrt{3} VI$   
 $kVA \text{ Rating} = kVA / \text{Load Factor}$   
 where Load factor = Average kVA / Maximum kVA

- c) For a new installation, an approximate method of estimating the capacity of a DG set is to add full load currents of all the proposed loads to be run in DG set. Then, applying a diversity factor depending on the industry, process involved and guidelines obtained from other similar units correct capacity can be arrived at.

**High Speed Engine or Slow/Medium Speed Engine**

The normal accepted definition of high speed engine is 1500 rpm. The high speed sets have been developed in India, whereas the slow speed engines of higher capacities are often imported. The other features and comparison between high and medium / slow speed engines are mentioned in Table 9.2 below:

Factor	Slow speed engine	High speed engine
Break mean effective pressure -therefore wear and tear and consumption of spares	Low	High
Weight to power ratio- therefore sturdiness and life	More	Less
Space	High	Less
Type of use	Continuous use	Intermittent use
Period between overhauls*	8000 hours	3200
Direct operating cost (Include Lubricating Oils, filters etc.	Less	High

\* Typical recommendations from manufacturers

Keeping the above factors and available capacities of DG set in mind, the cost of economics for both the engines should be worked out before arriving at a decision.

## **Capacity Combinations**

From the point of View of space, operation, maintenance and initial capital investment, it is certainly economical to go in for one large DG set than two or more DG sets in parallel.

Two or more DG sets running in parallel can be a advantage as only the short-fall in power—depending upon the extent of power cut prevailing - needs to filled up. Also, flexibility of operation is increased since one DG set can be stopped, while the other DG set is generating at least 50% of the power requirement. Another advantage is that one DG set can become 100% standby during lean and low power-cut periods.

### **Air Cooling Vs. Water Cooling**

The general feeling has been that a water cooled DG set is better than an air cooled set, as most users are worried about the overheating of engines during summer months. This is to some extent is true and precautions have to be taken to ensure that the cooling water temperature does not exceed the prescribed limits.

However, from performance and maintenance point of view, water and air cooled sets are equally good except that proper care should be taken to ensure cross ventilation so that as much cool air as possible is circulated through the radiator to keep its cooling water temperature within limits. While, it may be possible to have air cooled engines in the lower capacities, it will be necessary to go in for water cooled engines in larger capacities to ensure that the engine does not get over-heated during summer months.

## **Safety Features**

It is advisable to have short circuit, over load and earth fault protection on all the DG sets. However, in case of smaller capacity DG sets, this may become uneconomical. Hence, it is strongly recommended to install a circuit protection. Other safety equipment like high temperature, low lube oil pressure cut-outs should be provided, so that in the event of any of these abnormalities, the engine would stop and prevent damage. It is also essential to provide reverse power relay when DG sets are to run in parallel to avoid back feeding from one alternator to another.

## **Parallel Operation with Grid**

Running the DG set in parallel with the mains from the supply undertakings can be done in consultation with concerned electricity authorities. However, some supply undertakings ask the consumer to give an undertaking that the DG set will not be run in parallel with their supply. The reasons stated are that the grid is an infinite bus and paralleling a small capacity DG set would involve operational risks despite normal protections like reverse power relay, voltage and frequency relays.

## **Maximum Single Load on DG Set**

The starting current of squirrel cage induction motors is as much as six times the rated current for a few seconds with direct-on-line starters. In practice, it has been found that the starting current value should not exceed 200 % of the full load capacity of the alternator. The voltage and frequency throughout the motor starting interval recovers and reaches rated values usually much before the motor has picked up full speed.

In general, the HP of the largest motor that can be started with direct on line starting is about 50 % of the kVA rating of the generating set. On the other hand, the capacity of the induction motor can be increased, if the type of starting is changed over to star delta or to auto transformer starter, and with this starting the HP of the largest motor can be up to 75 % of the kVA of Gen set.

### **Unbalanced Load Effects**

It is always recommended to have the load as much balanced as possible, since unbalanced loads cause heating of the alternator, which may result in unbalanced output voltages. The maximum unbalanced load between phases should not exceed 10 % of the capacity of the generating sets.

### **Neutral Earthing**

The electricity rules clearly specify that two independent earths to the body and neutral should be provided to give adequate protection to the equipment in case of an earth fault, and also to drain away any leakage of potential from the equipment to the earth for safe working.

### **Site Condition Effects on Performance Derating**

Site condition with respect to altitude, intake temperature, cooling water temperature and de-rate diesel engine output as shown in following Tables: 9.3 and 9.4.

*Table 9.3 Altitude and Intake Temperature Corrections*

<b>Correction Factors For Engine Output</b>				
<b>Altitude Correction</b>			<b>Temperature Correction</b>	
<b>Altitude Meters over MSL</b>	<b>Non Super Charged</b>	<b>Super Charged</b>	<b>Intake °C</b>	<b>Correction Factor</b>
610	0.980	0.980	32	1.000
915	0.935	0.950	35	0.986
1220	0.895	0.915	38	0.974
1525	0.855	0.882	41	0.962
1830	0.820	0.850	43	0.950
2130	0.780	0.820	46	0.937
2450	0.745	0.790	49	0.925
2750	0.712	0.765	52	0.913
3050	0.680	0.740	54	0.900
3660	0.612	0.685		
4270	0.550	0.630		
4880	0.494	0.580		

*Table 9.4 Derating due to Air Inter Cooler water Inlet*

<b>Water Temperature °C</b>	<b>Flow %</b>	<b>Derating %</b>
<b>25</b>	<b>100</b>	<b>0</b>
<b>30</b>	<b>125</b>	<b>3</b>
<b>35</b>	<b>166</b>	<b>5</b>
<b>40</b>	<b>166</b>	<b>8</b>

### 9.3 Operational Factors

#### Load Pattern & DG Set Capacity

The average load can be easily assessed by logging the current drawn at the main switchboard on an average day. The ‘over load’ has a different meaning when referred to the D.G. set. Overloads, which appear insignificant and harmless on electricity board supply, may become detrimental to a D.G.set, and hence overload on D.G. set should be carefully analysed. Diesel engines are designed for 10% overload for 1 hour in every 12 hours of operation. The AC. generators are designed to meet 50% overload for 15 seconds as specified by standards. The D.G. set/s selection should be such that the overloads are within the above specified limits. It would be ideal to connect steady loads on DG set to ensure good performance. Alongside alternator loading, the engine loading in terms of kW or BHP, needs to be maintained above 50%. Ideally, the engine and alternator loading conditions are both to be achieved towards high efficiency.

Engine manufacturers offer curves indicating % Engine Loading vs fuel Consumption in grams/BHP. Optimal engine loading corresponding to best operating point is desirable for energy efficiency.

Alternators are sized for kVA rating with highest efficiency attainable at a loading of around 70% and more. Manufacturer’s curves can be referred to for best efficiency point and corresponding kW and kVA loading values.

## **Sequencing of Loads**

The captive diesel generating set has certain limits in handling the transient loads. This applies to both kW (as reflected on the engine) and kVA (as reflected on the generator). In this context, the base load that exists before the application of transient load brings down the transient load handling capability, and in case of AC. generators, it increases the transient voltage dip. Hence, great care is required in sequencing the load on D.G. set/s. It is advisable to start the load with highest transient kVA first followed by other loads in the descending order of the starting kVA. This will lead to optimum sizing and better utilization of transient load handling capacity of D.G. set.

## **Load Pattern**

In many cases, the load will not be constant throughout the day. If there is substantial variation in load, then consideration should be given for parallel operation of D.G. sets. In such a situation, additional D.G. set(s) are to be switched on when load increases. The typical case may be an establishment demanding substantially different powers in first, second and third shifts. By parallel operation, D.G. sets can be run at optimum operating points or near about, for optimum fuel consumption and additionally, flexibility is built into the system. This scheme can be also be applied where loads can be segregated as critical and non-critical loads to provide standby power to critical load in the captive power system.

## **Load Characteristics**

Some of the load characteristics influence efficient use of D.G. set. These characteristics are entirely load dependent and cannot be controlled by the D.G. set. The extent of detrimental influence of these characteristics can be reduced in several cases

### **Power Factor:**

The load power factor is entirely dependent on the load. The AC. generator is designed for the power factor of 0.8 lag as specified by standards. Lower power factor demands higher excitation currents and results in increased losses. Over sizing A.C. generators for operation at lower power factors results in lower operating efficiency and higher costs. The economical alternative is to provide power factor improvement capacitors.

### **Unbalanced Load:**

Unbalanced loads on AC. generator leads to unbalanced set of voltages and additional heating in AC. generator. When other connected loads like motor loads are fed with unbalanced set of voltages additional losses occur in the motors as well. Hence, the load on the AC. generators should be balanced as far as possible. Where single phase loads are predominant, consideration should be given for procuring single phase A.C. generator.

### **Transient Loading:**

On many occasions to contain transient voltage dip arising due to transient load application, a specially designed generator may have to be selected. Many times an

unstandard combination of engine and AC. generator may have to be procured. Such a combination ensures that the prime mover is not unnecessarily over sized which adds to capital cost and running cost.

**Special Loads:**

Special loads like rectifier/thyristor loads, welding loads, furnace loads need an application check. The manufacturer of diesel engine and AC generator should be consulted for proper recommendation so that desired utilization of DG set is achieved without any problem. In certain cases of loads, which are sensitive to voltage, frequency regulation, voltage wave form, consideration should be given to segregate the loads, and feed it by a dedicated power supply which usually assumes the form of DG motor driven generator set. Such an alternative ensures that special design of AC generator is restricted to that portion of the load which requires high purity rather than increasing the price of the D.G. set by specially designed AC generator for complete load.

**Waste Heat Recovery in DG Sets**

For combined heat and power applications, waste heat from reciprocating engines can be tapped mainly from exhaust gases and cooling water that circulates around cylinders in the engine jackets, with additional potential from oil and turbo coolers. While engine exhaust and cooling water each provide about half of the useful thermal energy, the exhaust is at much higher temperature (around 450 °C versus 100 °C) and hence is more versatile. Atypical energy balance in a reciprocating engine generator using Diesel and Natural gas is given in Table 9.5 below.

*Table 9.5 Energy Balance for Reciprocating Engine*

	Conventional cooling system	Cooling system with engine jacket and exhaust heat recovery
<b>500-kW natural gas engine generator*</b>		
Electric power	30%	30%
Jacket-water heat	38%	Exh recoverable 16% } 54% recoverable
Exhaust heat	24%	
Radiated heat lost to atmosphere	8%	8%
	100%	100%
<b>500-kW diesel engine generator*</b>		
Electric power	35%	35%
Jacket water	32%	Exh recoverable 16% } 48% recoverable
Exhaust heat	24%	
Radiated heat lost to atmosphere	9%	9%
	100%	100%

The table reveals that for natural gas generator more thermal energy (54%) can be recovered from the reciprocating engine compared to an electrical energy conversion of 30%.

It would be realistic to assess the Waste Heat Recovery (WHR) potential in relation to quantity, temperature margin, in kcal/hr as:

$$\text{Potential WHR} = (\text{kWh Output/hour}) \times (8 \text{ kg Gases / kWh Output}) \times 0.25 \text{ kcal/kg}^\circ\text{C} \times (t_g - 180^\circ\text{C})$$

Where,  $t_g$  is the gas temperature after Turbocharger, (the criteria being that limiting exit gas temperature cannot be less than  $180^\circ\text{C}$ , to avoid acid dew point corrosion), 0.25 being the specific heat of flue gases and kWh output being the actual average unit generation from the set per hour. For a 1100 kVA set, at 800 kW loading, and with  $480^\circ\text{C}$  exhaust gas temperature, the waste heat potential works out to:

$$800 \text{ kWh} \times 8 \text{ kg gas generation / kWh output} \times 0.25 \text{ kcal/kg }^\circ\text{C} \times (480 - 180), \text{ i.e., } 4,80,000 \text{ kcal/hr}$$

While the above method yields only the potential for heat recovery, the actual realizable potential depends upon various factors and if applied judiciously, a well configured waste heat recovery system can tremendously boost the economics of captive DG power generation.

The factors affecting Waste Heat Recovery from flue Gases are:

- a) DG Set loading, temperature of exhaust gases
- b) Hours of operation and
- c) Back pressure on the DG set

Consistent DG set loading (to over 60% of rating) would ensure a reasonable exit flue gas quantity and temperature. Fluctuations and gross under loading of DG set results in erratic flue gas quantity and temperature profile at entry to heat recovery unit, thereby leading to possible cold end corrosion and other problems. Typical flue gas temperature and flow pattern in a 5 MW DG set at various loads are given in Table 9.6

100% Load	11.84 kg/sec	$370^\circ\text{C}$
90% Load	10.80 kg/sec	$350^\circ\text{C}$
70% Load	9.08 kg/sec	$330^\circ\text{C}$
60% Load	7.50 kg/sec	$325^\circ\text{C}$
If the normal load is 60%, the flue gas parameters for waste heat recovery unit would be $325^\circ\text{C}$ inlet temperature, $180^\circ\text{C}$ outlet temperature and 27180 kg/hour gas flow.		
At 90% loading, however, values would be $350^\circ\text{C}$ and 32,400 kgs/Hour, respectively.		

\* Number of hours of operation of the DG Set has an influence on the thermal performance of waste heat Recovery unit. With continuous DG Set operations, cost benefits are favorable.

\* Back pressure in the gas path caused by additional pressure drop in waste heat recovery unit is another key factor. Generally, the maximum back pressure allowed is around 250-300 mmWC and the heat recovery unit should have a pressure drop lower than that. Choice of convective waste heat recovery systems with adequate heat transfer area are known to provide reliable service.

The configuration of heat recovery system and the choice of steam parameters can be judiciously selected with reference to the specific industry (site) requirements. Much good work has taken place in Indian Industry regarding waste heat recovery and one interesting configuration, deployed is installation of waste heat boiler in flue gas path along with a vapour absorption chiller, to produce 8°C chilled water working on steam from waste heat.

### **Trigeneration Technology**

In order to further optimize fuel utilization Trigeneration systems are developed which involves the simultaneous production of electricity, heat and cooling. The prime mover used for power generation includes diesel engines/gas engines. The waste heat recovery system in captive power generation units consists of waste heat recovery boiler for generating steam and use of jacket cooling water for operating Vapor Absorption Machines (VAM) to meet Air conditioning requirements.

### **9.4 Energy Performance Assessment of DG Sets**

Routine energy efficiency assessment of DG sets on shop floor involves following typical steps:

- 1) Ensure reliability of all instruments used for trial.
- 2) Collect technical literature, characteristics, and specifications of the plant.
- 3) Conduct a 2 hour trial on the DG set, ensuring a steady load, wherein the following measurements are logged at 15 minutes intervals.
  - a) Fuel consumption (by dip level or by flow meter)
  - b) Amps, volts, PF, kW, kWh
  - c) Intake air temperature, Relative Humidity (RH)
  - d) Intake cooling water temperature
  - 6) Cylinder-wise exhaust temperature (as an indication of engine loading)
  - f) Turbocharger RPM (as an indication of loading on engine)
  - g) Charge air pressure (as an indication of engine loading)
  - h) Cooling water temperature before and after charge air cooler (as an indication of cooler performance)
  - i) Stack gas temperature before and after turbocharger (as an indication of turbocharger performance)
- 4) The fuel oil/diesel analysis is referred to from an oil company data.
- 5) Analysis: The trial data is to be analysed with respect to:
  - a) Average alternator loading.
  - b) Average engine loading.
  - c) Percentage loading on alternator.
  - d) Percentage loading on engine.

- e) Specific power generation kWh/liter.
- f) Comments on Turbocharger performance based on RPM and gas temperature difference.
- g) Comments on charge air cooler performance.
- h) Comments on load distribution among various cylinders (based on exhaust temperature, the temperature to be  $\pm 5\%$  of mean and high/low values indicate disturbed condition).
- i) Comments on housekeeping issues like drip leakages, insulation, vibrations, etc.

A format as shown in the Table 9.7 is useful for monitoring the performance

*Table 9.7 Typical format for DG set monitoring*

DG Set No.	Electricity Generating Capacity (Site), kW	Derated Electricity Generating Capacity, kW	Type of Fuel used	Average Load as % of Derated Capacity	Specific Fuel Cons. Lit/kWh	Specific Lube Oil Cons. Lit/kWh
1.	480	300	LDO	89	0.335	0.007
2.	480	300	LDO	110	0.334	0.024
3.	292	230	LDO	84	0.356	0.006
4.	200	160	HSD	89	0.325	0.003
5.	200	160	HSD	106	0.338	0.003
6.	200	160	HSD			
7.	292	230	LDO	79	0.339	0.006
8.	292	230	LDO	81	0.362	0.005
9.	292	230	LDO	94	0.342	0.003
10.	292	230	LDO	88	0.335	0.006
11.	292	230	LDO	76	0.335	0.005
12.	292	230	LDO	69	0.353	0.006

DG Set No.	Electricity Generating Capacity (Site), kW	Derated Electricity Generating Capacity, kW	Type of Fuel used	Average Load as % of Derated Capacity	Specific Fuel Cons. Lit/kWh	Specific Lube Oil Cons. Lit/kWh
13	400	320	HSD	75	0.334	0.004
14.	400	320	HSD	65	0.349	0.004
15.	880	750	LDO	85	0.318	0.007
16.	400	320	HSD	70	0.335	0.004
17.	400	320	HSD	80	0.337	0.004
18.	880	750	LDO	78	0.345	0.007
19.	800	640	HSD	74	0.324	0.002
20.	800	640	HSD	91	0.290	0.002
21.	880	750	LDO	96	0.307	0.002
22.	920	800	LDO	77	0.297	0.002

### 9.5 Energy Saving Measures for DC Sets

- a) Ensure steady load conditions on the DG set, and provide cold, dust free air at intake (use of air washers for large sets, in case of dry, hot weather, can be considered).
- b) Improve air filtration.
- c) Ensure fuel oil storage, handling and preparation as per manufacturer's guidelines/oil company data.
- d) Consider fuel oil additives in case they benefit fuel oil properties for DG set usage.
- e) Calibrate fuel injection pumps frequently.
- f) Ensure compliance with maintenance checklist.
- g) Ensure steady load conditions, avoiding fluctuations, imbalance in phases, harmonic loads.
- h) In case of a base load operation, consider waste heat recovery system adoption for steam generation or refrigeration chiller unit incorporation. Even the Jacket Cooling Water is amenable for heat recovery, vapour absorption system adoption.
- i) In terms of fuel cost economy, consider partial use of biomass gas for generation. Ensure tar removal from the gas for improving availability of the engine in the long run.
- j) Consider parallel operation among the DG sets for improved loading and fuel economy thereof.
- k) Carryout regular field trials to monitor DG set performance, and maintenance planning as per requirements

### Example 9.2

- a) A 180 kVA, 0.80 PF rated DG set has diesel engine rating of 210 BHP. What is the maximum power factor which can be maintained at full load on the alternator without overloading the DG set? (Assume alternator losses and exciter power requirement as 5.66 kW and there is no derating of DG set)

### Solution

$$\text{Engine rated Power} = 210 \times 0.746 = 156.66 \text{ kW}$$

$$\begin{aligned} \text{Rated power available for alternator} &= 15 \times 6.66 - 5.66 = 151 \text{ kW} \\ \text{Maximum power factor possible} &= 151 / 180 = 0.84 \end{aligned}$$

- b) ADG set is operating at 600 kW load with 450°C exhaust gas temperature. The DG set generates 8 kg of exhaust gas/ kWh generated. The specific heat of gas at 450°C is 0.25 kcal/ kg°C. A heat recovery boiler is installed after which the exhaust temperature drops to 230°C. How much steam will be generated at 3 kg/cm<sup>2</sup> with enthalpy of 650.57 kcal/ kg. Assume boiler feed water temperature as 80°C.

**Solution**

$$\text{Waste Heat Recovery} = 600 \text{ kWh} \times 8 \text{ kg gas generated/ kWh output} \times 0.25 \text{ kcal/ kg } ^\circ\text{C} \times (450^\circ\text{C} - 230^\circ\text{C}) = 2,64,000 \text{ kcal/hr}$$

$$\text{Steam generation} = 2,64,000 \text{ kcal/hr} / (650.57 - 80) = 462.7 \text{ kg/ hr.}$$

## **Chapter 10: Energy Efficiency and Conservation in Buildings**

### **10.1 Introduction**

There are several different uses of energy in buildings. The major uses are for lighting, heating, cooling, power delivery to equipment and appliances, and domestic water. The amount that each contributes to the total energy use varies according to the climate, type of building, number of working hours and time of year. Energy use for air-conditioning has the largest share at a national level. In areas where severe winters occur, heating load will be greater than cooling load in terms of the total energy use.

In some types of buildings in certain climatic zones, the lighting load might be greater than either the heating or cooling loads.

Industrial and commercial buildings are dissimilar in terms of energy use, as industries primarily use large quantities of energy for specialized processes whereas buildings use the major amount of energy for human comfort. It is difficult to generalize energy use by type of building because there are many variables that determine the energy use in a particular building.

### **EE&C Potential in Residential Sector**

If all the existing home appliances in residences are to be replaced by higher efficiency products (as of today), huge energy reduction can be achieved. It is estimated that EE&C potential is 28.8% of the total energy consumption in the residential sector. Considering the fact that about 30% of national primary energy is consumed in the residential sector, the potential economic impact of EE&C measures is massive: almost 8.6% of the total energy consumption in the country can be reduced.

### **EE&C Potential in Commercial Sector (Buildings)**

Electricity is the main mode of energy in commercial buildings. In detail, nearly 50% of the total energy is consumed by ACs and 10-30% by lighting systems. It is expected that a simple replacement of ACs and lighting systems with high energy efficiency ones can save about 50% of total electricity consumptions in the commercial sector. However it is not easy to introduce EE&C measures for all the buildings. Thus as a realistic value, EE&C potential for buildings was estimated about 10%.

### **10.2 Bangladesh National Building Code (BNBC):**

In order to regulate the technical details of building construction and to maintain the standard of construction the Bangladesh National Building Code (BNBC) was first published in 1993. Depletion of energy resources and environmental changes is a major concern worldwide. Bangladesh is no exception to it. Keeping these aspects in mind, changes and modifications have been suggested in BNBC 1993 for use of energy saving appliances, non-conventional fuels etc. in buildings. The updated BNBC contains chapters addressing the issues of energy conservation, rainwater harvesting and distribution mechanisms in buildings. The updated BNBC has 10 parts with a total of 49 chapters. In Part 3, "General Building Requirements, Control and Regulation" a new Chapter titled, "Energy Efficiency and Sustainability" has been included giving minimum code requirements for achieving the efficiency.

To reduce energy consumption in building provisions for use of variable refrigeration system in HVAC applications, Variable Voltage, Variable frequency drives in elevator applications has been included in Chapter-2 "Air Conditioning, Heating and Ventilation" of Part-8 "Building Services". Energy conservation in lighting using energy saving lamps, Fluorescent lamps and GLS lamps has also been proposed in Chapter-1, "Electrical and Electronics Engineering Services for Buildings" of the same part. To augment water supply in Buildings, Chapter-8, "Rainwater Management" in Part-8 "Building Services" has been included in the Updated Code containing specific guidelines for harvesting, storage and distribution of rainwater.

For more details, please have a look of the Bangladesh National Building Code (BNBC) 2020. The Bangladesh National Building Code (BNBC) 2020 has been gazette on 11 February 2021.

The updated version of BNBC, incorporated with addition of energy efficiency requirement of buildings in near future will be the core program for promoting EE&C in Buildings and contain the following requirement on building energy efficiency:

- a. Heat insulation and/or ventilation performance of building envelope
- b. Energy efficiency of building equipment (HVAC, lighting, fans, hot water supply, lift, escalator, renewable energy options)
- c. Water efficiency and management and Sanitation
- d. Roof gardening and vegetation

### **10.3 Building Energy Efficiency & Environment Rating (BEEER)**

To ensure the energy efficiency in buildings, SREDA has developed the rating system for buildings and act as the implementation and execution body for the Building Energy Efficiency & Environment Rating (BEEER)

#### **The objective to which the program aims to contribute is to:**

- Promote green and sustainable building practices on the supply and demand side of Bangladesh's construction sector;
- Contribute to climate change mitigation by saving resources in the building sector while enhancing economic prosperity and competitiveness, as well as alleviating poverty by considering both green and social standards;
- Establish a building energy efficiency and environmental rating systems serving as a standard/reference for green building construction practices;
- Enhance sustainable consumption in the building sector through a rating system, providing consumer information and a distinctive grade for sustainable buildings;
- Mobilize and capacitate key stakeholders to get involved in green building design and construction.
- Promote green equipment and construction materials, fixtures and make the market ready.
- Develop the capacity of architects and Engineers, Energy Managers & Energy Auditors in Green Construction.
- Provide access to soft and subsidize loan facilities for green building developer and consumers.

Generally, the Building should be Rated based on their typology. However, at the beginning stage of the Building Energy and Environment Rating cover all types of buildings by a single guideline. In next version, building typology (Existing Building, New Construction, Interior space, Township, Industries etc.) specific rating guideline will be developed.

*The draft of BEEER is available at SREDA website ([www.sreda.gov.bd](http://www.sreda.gov.bd))*

#### **10.4 Energy Efficiency Measures in Buildings:**

##### **Air-Conditioning System:**

##### **Weather stripping of Windows and Doors**

Minimize exfiltration of conditioned air and infiltration of external un-conditioned air through leaky windows and doors by incorporating effective means of weather stripping. Self-closing doors should also be provided where heavy traffic of people is anticipated.

##### **Temperature and Humidity Setting**

Ensure human comfort by setting the temperature to between 23°C and 25°C and the relative humidity between 55% and 65%.

##### **Chilled Water Leaving Temperature**

Ensure higher chiller energy efficiency by maintaining the chilled water leaving temperature at or above 7° C. As a rule of thumb, the efficiency of a centrifugal chiller increases by about 2% % for every 1° C rise in the chilled water leaving temperature.

##### **Chilled Water Pipes and Air Ducts**

Ensure that the insulation for the chilled water pipes and ducting system is maintained in good condition.

This helps to prevent heat gain from the surroundings.

##### **Chiller Condenser Tubes**

Ensure that mechanical cleaning of the tubes is carried out at least once every six months. Fouling in the condenser tubes in the form of slime and scales reduces the heat transfer of the condenser tubes and thereby reducing the energy efficiency of the chiller.

##### **Cooling Towers**

Ensure that the cooling towers are clean to allow for maximum heat transfer so that the temperature of the water returning to the condenser is less than or equal to the ambient temperature.

##### **Air-handling Unit Fan Speed**

Install devices such as frequency converters to vary the fan speed. This will reduce the energy consumption of the fan motor by as much as 15%.

#### **Air Filter Condition**

Maintain the filter in a clean condition. This will improve the heat transfer between air and chilled water and correspondingly reduce the energy consumption.

#### **Lighting System:**

All lighting systems generate heat that needs to be dissipated. By designing energy efficient lighting system that integrates day lighting and good controls, heat gains can be reduced significantly. This can reduce the size of the HVAC system resulting in first-cost savings.

#### **Day lighting**

Day lighting benefits go beyond energy savings and power reduction. Daylight spaces have been shown to improve people's ability to perform visual tasks, increase productivity and reduce illness. Building fenestration should be designed to optimize day lighting and reduce the need for electric lighting.

Orient the building to minimize building exposure to the east and west and maximize glazing on the south and north exposures.

Daylight strategies do not save energy unless electric lights are turned off or dimmed appropriately.

ECBC requires controls in day lit areas that are capable of reducing the light output from luminaires by at least half.

- Install dimmers to take advantage of day lighting and where cost-effective.
- Replace rheostat dimmers with efficient electronic dimmers.
- Combine time switching with day lighting using astronomical time clocks.
- Control exterior lighting with photo controls where lighting can be turned off after a fixed interval.

#### **Switch off Lights When Not in Use**

#### **Provision of Separate Switches for Peripheral Lighting**

A flexible lighting system, which made use of natural lighting for the peripherals of the room, should be considered so that these peripheral lights can be switched off when not needed.

#### **Install High Efficiency Lighting System**

Replace incandescent and other inefficient lamps with lamps with higher lighting efficacy. For example, replacing incandescent bulbs with compact fluorescent lamps can reduce electricity consumption by 75% without any reduction in illumination levels.

### **Fluorescent Tube Ballasts**

The ballast losses of conventional ballast and electronic ballast are 12W and 2W respectively. Hence, consider the use of electronic ballast for substantial energy savings in the lighting system.

### **Lamp Fixtures or Luminaries**

Optical lamp luminaries made of aluminum, silver or multiple dielectric coatings have better light distribution characteristics. Use them to reduce electricity consumption by as much as 50% without compromising on illumination levels.

### **Integration of Lighting System with Air-Conditioning System**

In open plan offices, the air-conditioning and lighting systems can be combined in such a way that the return air is extracted through the lighting luminaires. This measure ensures that lesser heat will be directed from the lights into the room.

### **Cleaning of Lights and Fixtures**

Clean the lights and fixtures regularly. For best results, dust at least four times a year.

### **Use Light Colors for Walls, Floors and Ceilings**

The higher surface reflectance values of light colors will help to make the most of any existing lighting system. Consider light colored furniture and room partitions to optimize light reflectance. Avoid furniture colors and placement that will interfere with light distribution. Keep ceilings and walls as bright as possible.

### ***Deal with each activity area and each fixture individually***

Eliminate excessive lighting by reducing the total lamp wattage in each activity area

### **Task Lighting**

Lighting layout should use task lighting principle. Install focusing lamps or flexible extensions wherever needed.

## **10.5 Building Water Pumping Systems**

The pumps used in the buildings are for chilled water circulation, cooling water circulation, domestic water, hot water and sewage water. The water requirement for cooling, drinking and general purpose requirements in buildings are met by the local authorities and also from the independent bore wells.

The water received is stored and pumped to the overhead tanks provided on the building terrace are hydro pneumatic system is used. Normally the pumps used in the building are

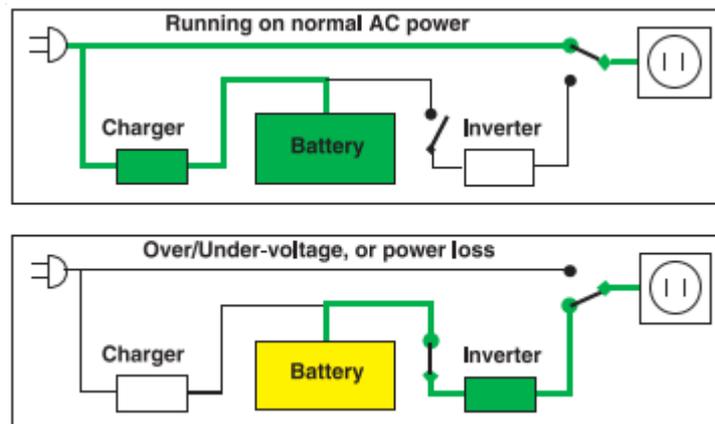
of centrifugal type with efficiencies of 60%. The following energy conservation measures are adopted to reduce energy consumption in water pumping in buildings:

- Installation of high efficiency pumps
- Operation of pumps in parallel
- Auto pump operation with low level/high level flow control systems.
- Installation of variable frequency drive for chiller water and cooling water pumps

## 10.6 Uninterruptible power supply

An Uninterruptible Power Supply (UPS) is a device that has an alternate source of energy, typically a battery backup that can provide power when the primary power source is temporarily disabled. The switchover time must be small enough to not cause a disruption in the operation of the loads (Figure 10.6).

The components of a UPS are converter (AC to DC), battery, inverter (DC to AC), monitor and control hardware / software.



*Figure 10.1 UPS Operation*

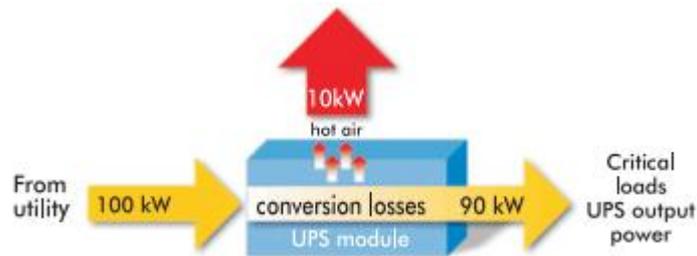
There are two type of UPS architecture namely line interactive (OFF line) and ON line.

Simple OFF Line UPS systems, connect the load directly to the input AC line. Line Interactive systems have also the ability to correct UPS output if AC input voltage deviates beyond preset limits by means of an auto-transformer based Automatic Voltage Regulator. In case of significant utility voltage dips or outages these systems transfer the UPS to battery operation.

Due to its direct connection to input AC power, OFF Line types, including Line Interactive systems offer higher efficiency when compared to an Online UPS. But, unstable grid environment, with frequent power interruptions or outages, might cause these systems to suffer from frequent transfers to battery operation. Thus, exposing the critical load to possible failures due to unsmooth or unsuccessful transfer, or to battery failure, because of numerous battery discharges, which decrease drastically battery life time.

An Online UPS system, frequently called Double conversion system, first converts the AC input voltage to stabilized DC, which is then converted back to AC, to feed continuously the critical load, with pure stabilized sinusoidal output, coming either from the input line via AC/DC converter, or from batteries in case of power failure.

Energy efficiency of a UPS is the difference between the amount of energy that goes into a UPS versus the amount of useful energy that comes out of the UPS and actually powers loads (Figure 10.7). In all UPS systems some energy is lost as heat when it passes through the internal components of the UPS (including transformers, rectifiers, and inverters).



*Figure 10.2 Energy Loss in UPS*

Almost all UPS power 100% nonlinear loads +viz., computers, servers and electronic equipment, though the manufacturers test their UPS often using linear loads. UPS efficiency is often much higher when powering linear loads.

The second major factor to influence UPS efficiency is the power level at which the efficiency is measured. Most UPSs typically have their best efficiency operating at 50% to 100% load level. But in the real situation, most UPS systems operate at 25% to 60% of their nominal load- not fully loaded.

To determine accurate efficiency, the UPS should demonstrate efficiency at loads between 25—50%, where most UPS will likely to be operating.

## 10.7 Escalators and Elevators

The requirement of the maximum allowable electric power indicates ultimately the energy performance of the equipment. The power for lift equipment is to be measured when the lift is carrying its rated load and moving upward at its contract speed. For escalators and passenger conveyors, since the rated load is usually defined as number of person (not in kg weight), there is no theoretical rated load in kg for the equipment. Thus the electric power is to be measured when the escalator/conveyor is carrying no load and moving at its rated speed either in the upward or downward direction.

In escalators and elevators, the dominating factors that determine the energy consumption are the efficiency of the motor, friction, the controller and the driving gear box. The proportion of frictional loss of the machine can also become significant in the power consumption in no load condition, as it is the fixed overhead to keep the equipment running.

### Factors That Affect Energy Consumption in Elevators and Escalator System

Energy is consumed by lift and escalator equipment mainly on the following categories:

- ✓ Friction losses incurred while travelling.
- ✓ Dynamic losses while starting and stopping.
- ✓ Lifting (or lowering) work done, potential energy transfer.
- ✓ Regeneration into the supply system.

The general approach to energy efficiency in lift and escalator equipment is merely to minimize the friction losses and the dynamic losses of the system. There are many factors that will affect these losses for elevator and escalator system:-

### **(A) Characteristic of the equipment**

The type of motor, drive and control system of the machine, the internal decoration, means to reduce friction in moving parts (e.g. guide shoes), type, speed and the pulley system of the equipment.

### **(B) Characteristic of the premises**

The population distribution, the type of the premises, the height of the premises and the house keeping of the premises,

### **(C) The configuration of the lift/escalator system**

The zoning of the lift system, the combination of lift and escalator equipment, the strategies for vertical transportation and the required grade of service of the system.

## **General Principles to Achieve Energy Efficiency**

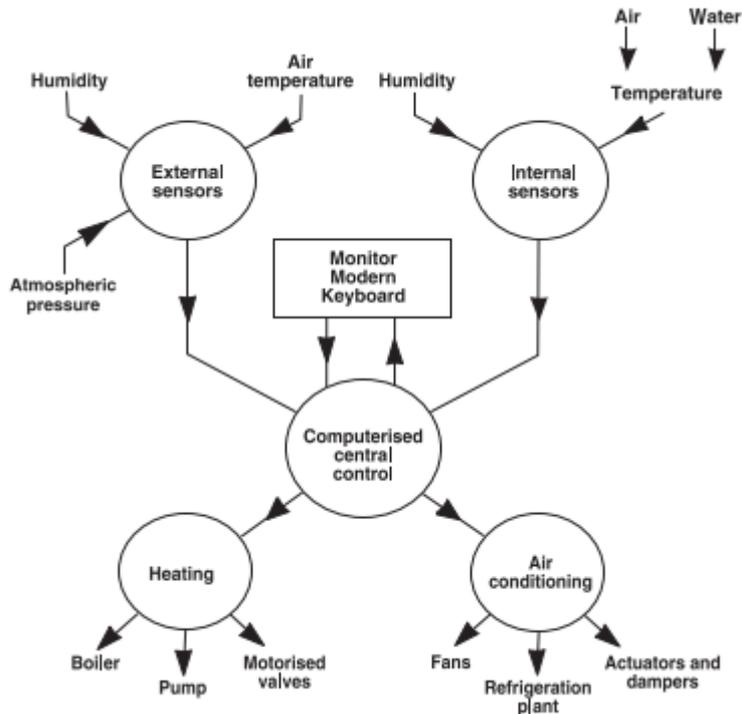
In general the principles for achieving energy efficiency for lift/escalator installations are as follows:

- ✓ Specify energy efficiency equipment for the system.
- ✓ Do not over design the system.
- ✓ Suitable zoning arrangement.
- ✓ Suitable control and energy management of lift equipment
- ✓ Use lightweight materials for lift car decoration.
- ✓ Good housekeeping.

## **10.8 Building Energy Management System (BEMS)**

Energy management systems can vary considerably in complexity and degree of sophistication. The simplest timing mechanism to switch systems ON and OFF at pre-determined intervals on a routine basis could be considered as an energy management system. These progresses to include additional features such as programmers, thermostatic controls, motorized valves, zoning, and optimum start controllers and compensated circuits.

The most complex of energy management systems have a computerized central controller linked to numerous sensors and information sources. These could include the basic internal and external range shown schematically in Figure 10.8, along with further processed data to include: the time, the day of the week, time of year, percentage occupancy of a building, meteorological data, system state feedback factors for plant efficiency at any one time and energy gain data from the sun, lighting, machinery and people.



*Figure 10.3 Building Energy Management*

A microprocessor is then the main feature of the control system. Data on temperature, flow rates, pressures, etc., as appropriate, are collected from sensors in the system and the treated spaces and stored in the memory of the processor. Provided that equations defining the performance of the control elements, the items of plant and the behavioral characteristics of the systems controlled have been developed and fed into the microprocessor as algorithms, deviations from the desired performance can be dealt with by calculation, the plant output being varied accordingly. Mathematical functions replace control modes. For example, if room temperature rose in a space conditioned by a constant volume reheat system, the correct position of the valve in the low temperature hot water line feeding the heater battery could be calculated and corrected as necessary, to bring the room temperature back to the set point as rapidly as possible, without any offset. Data can be stored to establish trends and anticipation can be built into the program so that excessive swings in controlled conditions may be prevented. Furthermore, self-correction can be incorporated so that the control system learns from experience and the best possible system performance is obtained. This implies that commissioning inadequacies and possibly even design faults can be corrected but only to a certain extent. Optimum results are only obtainable, and the cost of the installation justified, from systems that have been properly designed, installed and commissioned. Under such circumstances it is then feasible to extend the scope of microprocessor control to include the management of all the building services with an economic use of its thermal and electrical energy needs.

The functions of a building management system (BMS) or building energy management system (BEMS) are monitoring and control of the services and functions of a building, in a way that is economical and efficient in the use of energy. Furthermore, it may be arranged that one system can control a group of buildings.

### **10.9 Energy performance Index: (EPI)**

Energy Performance Index (EPI) in kWh / sq m/ year will be considered for rating the building. Bandwidths for Energy Performance Index for different climatic zones have been developed based on percentage air-conditioned space. For example a building in a composite climatic zone like Dhaka and having air conditioned area greater than 50% of their built up area, the bandwidths of EPI range between 190-90 kWh/sq m/year. Thus a building would get a 5-Star rating if its EPI falls below 90 kWh/sq m/year and 1 Star if it is between 165-190 kWh/ sq m/year.