Unit -5: Fans & Pumps:

Fans and Blowers:

1. Explain the flow control strategies in fans and justify the efficient method of control.
2. How do you assess the performance of fans? Explain
3. Explain different types of fans with respect to their characteristics and applications.
4. Explain the centrifugal pump with the help of a neat sketch.
5. State energy-saving opportunities in the pumping system.
6. Classify pumps.
7. What are the different types of pump curves? Explain any two of them in detail.
8. How will you differentiate fans, pumps, and compressors?

DG. Set

1. What is a compressor? Give its classification.
2. Explain the working of compressor.
3. Explain the reciprocating air compressor.
4. Explain the working of four-stroke diesel engine.
5. What are the various operational factors in the D.G. set?
6. List energy-saving opportunities in an industrial DG set plant.
7. Define and Explain: i) Compressor efficiency ii) Compressed air system.
8. Explain compressed air system components in brief.
9. Discuss the selection and installation factors in case of DG sets.
10. Explain briefly the components of compressed air system.

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5. FANS AND BLOWERS

Syllabus
Fans and blowers: Types, Performance evaluation, Efficient system operation, Flow control strategies and energy conservation opportunities

5.1 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 5.1 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	–

5.2 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 5.2.

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 5.1).

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

TABLE 5.2 FAN EFFICIENCIES

Type of fan	Peak Efficiency Range
Centrifugal Fans	
Airfoil, backward curved/inclined	79–83
Modified radial	72–79
Radial	69–75
Pressure blower	58–68
Forward curved	60–65
Axial fan	
Vane axial	78–85
Tube axial	67–72
Propeller	45–50

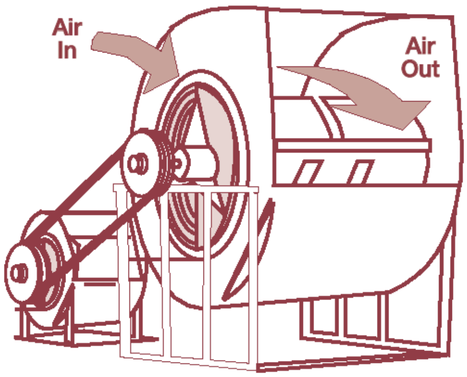


Figure 5.1 Centrifugal Fan

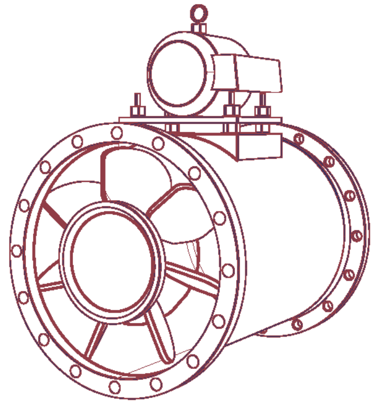


Figure 5.2 Axial Fan

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.

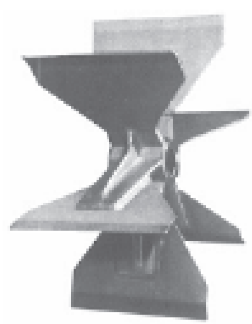
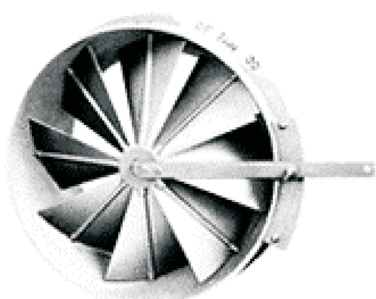
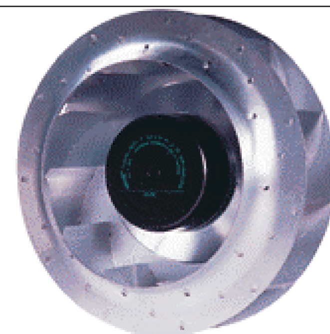
Paddle Blade (Radial blade)	Forward Curved (Multi-Vane)	Backward Curved
		

Figure 5.3 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.

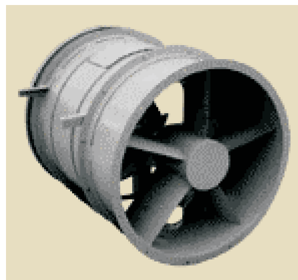
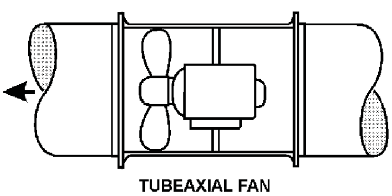
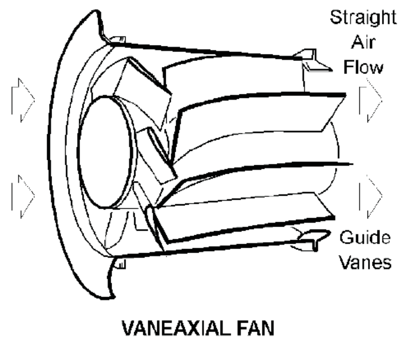
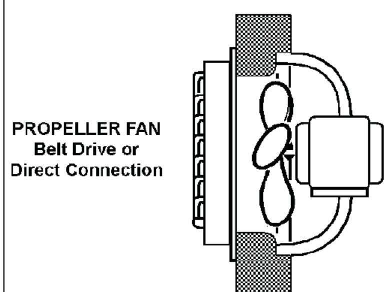
Tube Axial	Vane Axial	Propeller
		
		

Figure 5.4 Types of Axial Fans

The different types of fans, their characteristics and typical applications are given in Table 5.3.

Common Blower Types

Blowers can achieve much higher pressures than fans, as high as 1.20 kg/cm^2. 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 gear-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^2, but can achieve higher pressures. One characteristic is that airflow tends to drop drastically as system pressure

TABLE 5.3 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 efficiencyexhausts	High pressure applications including HVAC systems,
Airfoil type	Same as backward curved type, highest efficiency	Same as backward curved, but for clean air applications			

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.

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^2 - 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.

5.3 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 back-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.

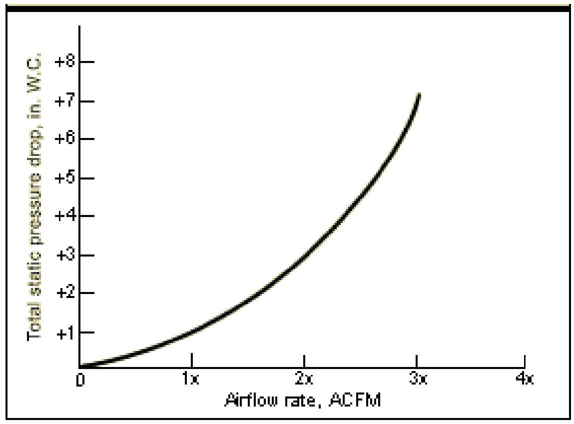


Figure 5.5 System Characteristics

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 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 5.7).

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.

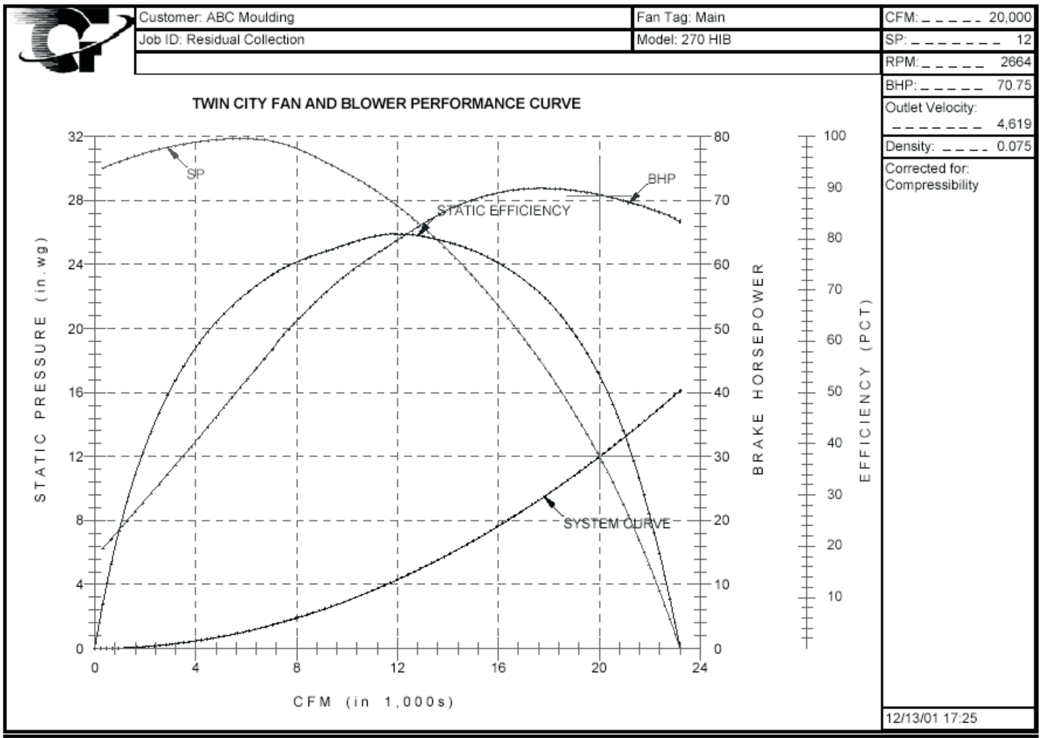


Figure 5.6 Fan Characteristics Curve by Manufacturer

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 actu-

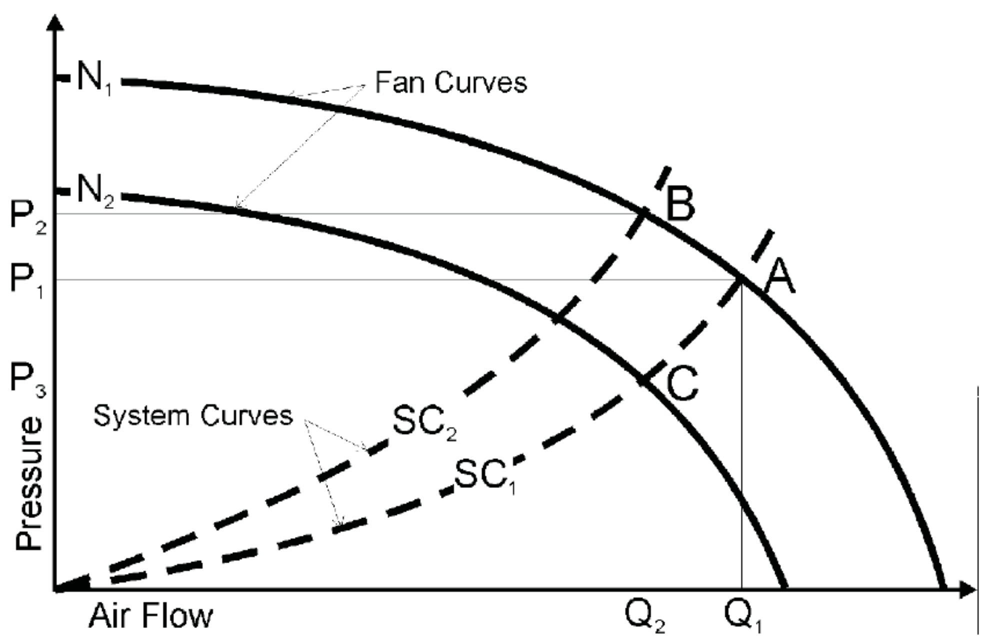


Figure 5.7 System Curve

al 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:

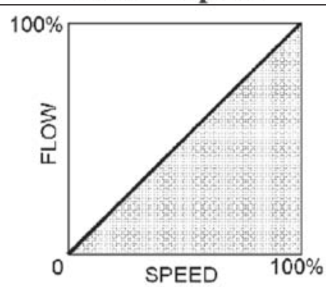
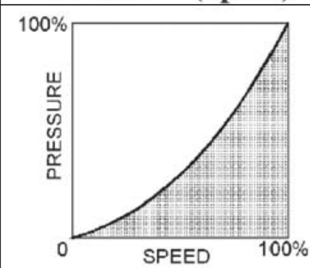
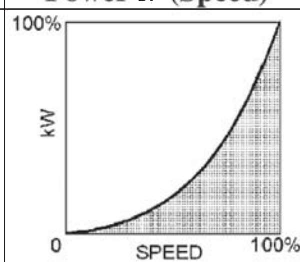
First method is to restrict the air flow by partially closing a damper in the system. This action causes a new system performance curve (SC_2) where the required pressure is greater for any given air flow. The fan will now operate at "B" to provide the reduced air flow Q_2 against higher pressure P_2.

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.

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.

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

Where Q – flow, SP – Static Pressure, kW – Power and N – speed (RPM)

5.4 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.

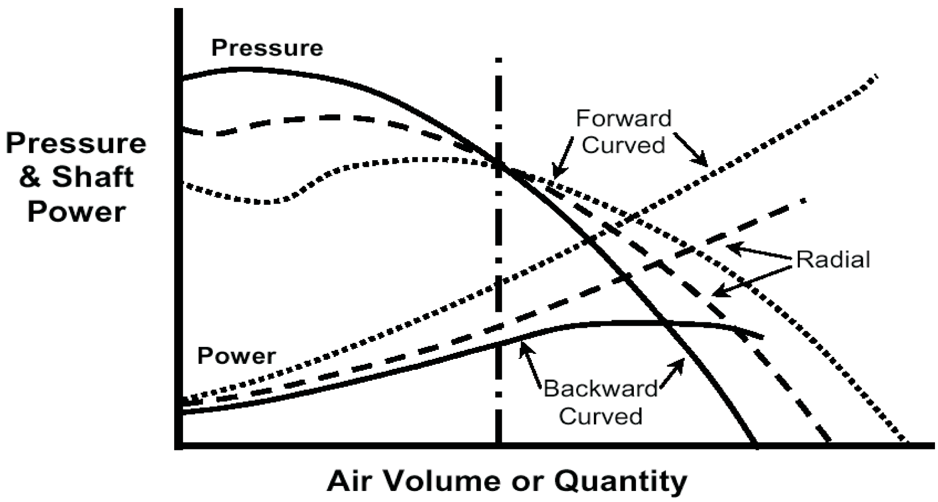


Figure 5.8 Fan Static Pressure and Power Requirements for Different Fans

Fan performance characteristics and efficiency differ based on fan and impeller type (See Figure 5.9).

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.

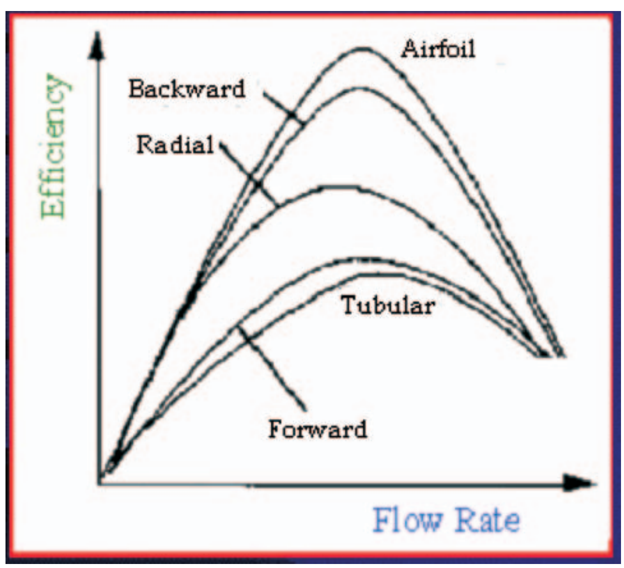


Figure 5.9 Fan Performance Characteristics Based on Fans/ Impellers

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.

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 dependant 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.

Installation of Fan

The installation of fan and mechanical maintenance of the fan also plays a critical role in the efficiency of the fan. The following clearances (typical values) should be maintained for the efficient operation of the impeller.

Impeller Inlet Seal Clearances

- Axial overlap –5 to 10 mm for 1 metre plus dia impeller
- Radial clearance –1 to 2 mm for 1 metre plus dia impeller
- Back plate clearance –20 to 30 mm for 1 metre plus dia impeller
- Labyrinth seal clearance –0.5 to 1.5 mm

The inlet damper positioning is also to be checked regularly so that the "full open" and "full close" conditions are satisfied. The fan user should get all the details of the mechanical clearances from the supplier at the time of installation. As these should be strictly adhered to, for efficient operation of the fan, and a checklist should be prepared on these clearances. A check on these clearances should be done after every maintenance, so that efficient operation of the fan is ensured on a continuous basis.

System Resistance Change

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.

5.5 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 5.10, 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 15 kW.

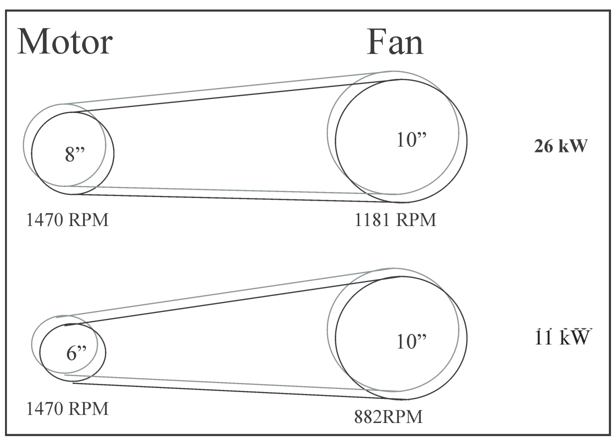


Figure 5.10 Pulley Change

Damper Controls

Figure 5.11 Damper change

Some fans are designed with damper controls (see Figure 5.11). 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.

Inlet Guide Vanes

Figure 5.12 Inlet Guide Vanes

Inlet guide vanes are another mechanism that can be used to meet variable air demand (see Figure 5.12). 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.

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.

The performance of two fans in parallel will result in doubling the volume flow, but only at free delivery. As Figure 5.13 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.

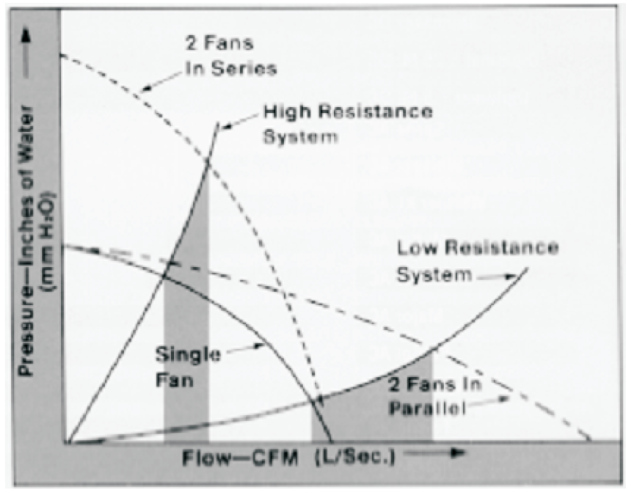


Figure 5.13 Series and Parallel Operation

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 of various volume control methods with respect to power consumption (%) required power is shown in Figure 5.14.

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.

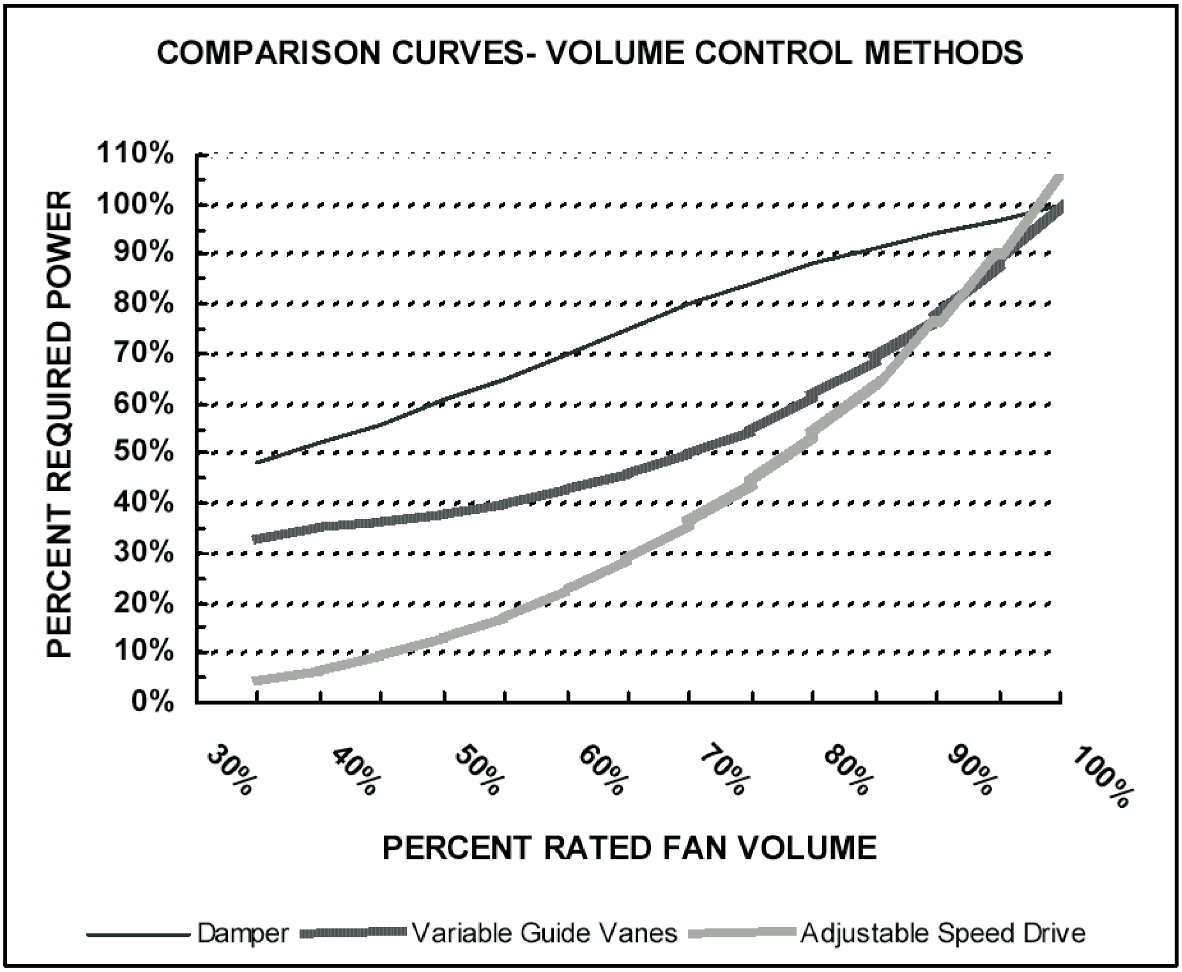


Figure 5.14 Comparison: Various Volume Control Methods

5.6 Fan Performance Assessment

The fans are tested for field performance by measurement of flow, head, 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 5.15).

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

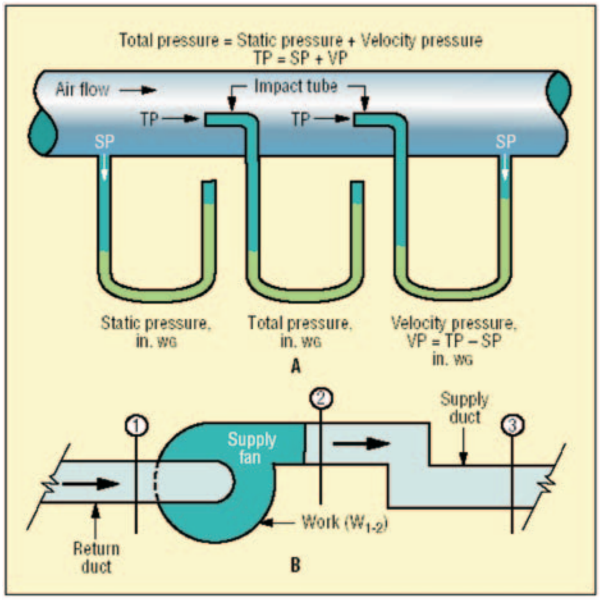


Figure 5.15 Static, Total and Velocity Pressure

Measurement by Pitot tube

The Figure 5.16 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.

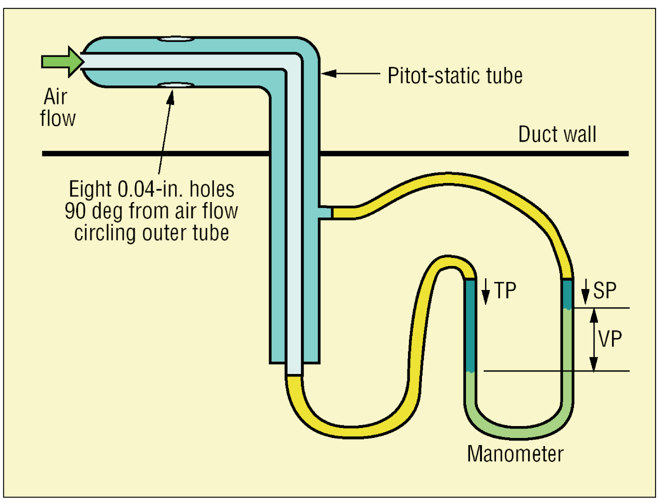


Figure 5.16 Velocity Measurement Using Pitot Tube

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.

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 5.17).

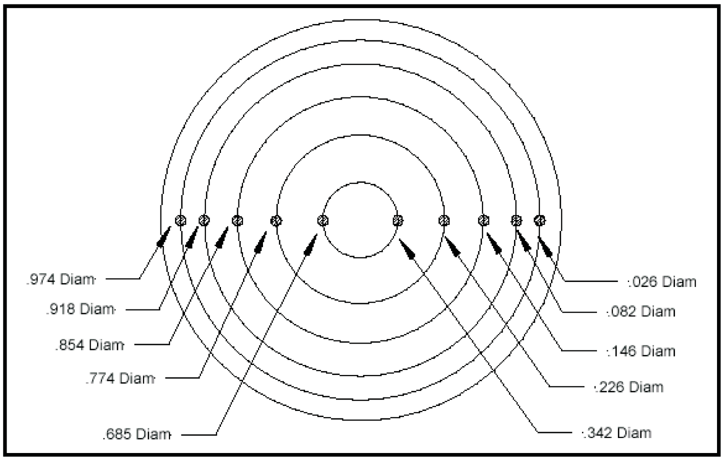


Figure 5.17 Traverse Points for Circular Duct

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.

Air density calculation

The first calculation is to determine the density of the air. To calculate the velocity and volume from the velocity pressure measurements it is necessary to know the density of the air. The density is dependent on altitude and temperature.

$$\text{Gas Density}(\gamma) = \frac{273 \quad x \ 1.293}{273 + t^{o}C}$$

t°C – temperature of gas/air at site condition

Velocity calculation

Once the air density and velocity pressure have been established, the velocity can be determined from the equation:

$$\text{Velocity v, m/s} = \frac{C_p \ x \ \sqrt{2 \times 9.81 \times \Delta p \times \gamma}}{\gamma}$$

C_p = Pitot tube constant, 0.85 (or) as given by the manufacturer

Δp = Average differential pressure measured by pitot tube by taking measurement at number of points over the entire cross section of the duct.

γ = Density of air or gas at test condition,

Volume calculation

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

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

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:

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

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

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

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

5.7 Energy Saving Opportunities

Minimizing demand 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 de-rating (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

3. COMPRESSED AIR SYSTEM

Syllabus

Compressed air system: Types of air compressors, Compressor efficiency, Efficient compressor operation, Compressed air system components, Capacity assessment, Leakage test, Factors affecting the performance and efficiency

3.1 Introduction

Air compressors account for significant amount of electricity used in Indian industries. Air compressors are used in a variety of industries to supply process requirements, to operate pneumatic tools and equipment, and to meet instrumentation needs. Only 10 – 30% of energy reaches the point of end-use, and balance 70 – 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.

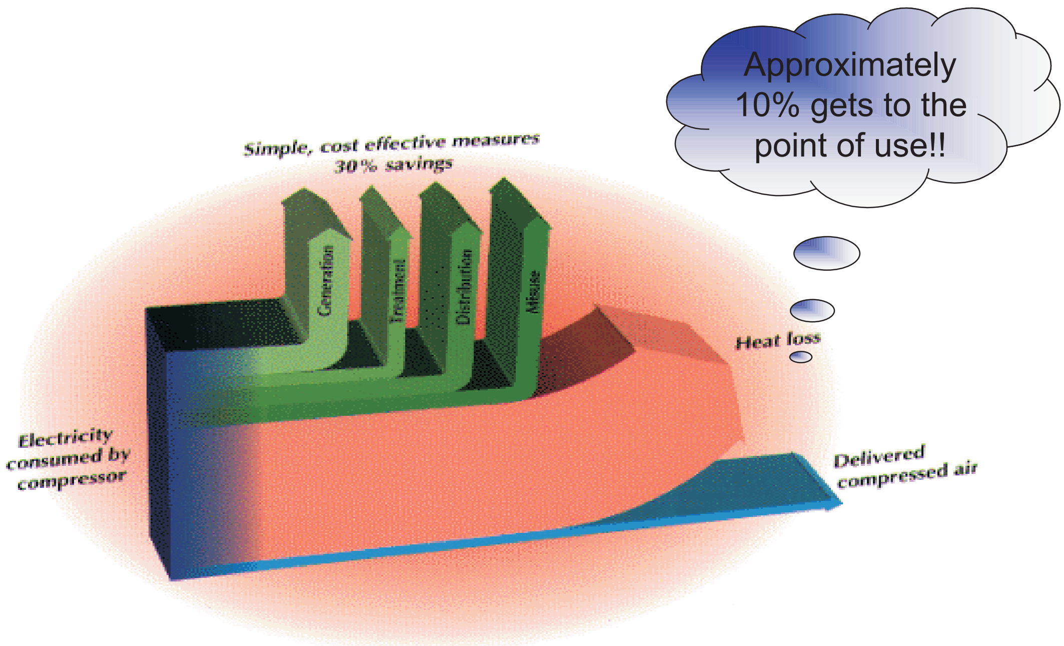


Figure 3.1 Sankey Diagram for Compressed Air System

3.2 Compressor Types

Compressors are broadly classified as: Positive displacement compressor and Dynamic compressor.

Positive displacement compressors increase the pressure of the gas by reducing the volume. Positive displacement compressors are further classified as reciprocating and rotary compressors.

Dynamic compressors increase the air velocity, which is then converted to increased pressure at the outlet. Dynamic compressors are basically centrifugal compressors and are further classified as radial and axial flow types.

The flow and pressure requirements of a given application determine the suitability of a particulars type of compressor.

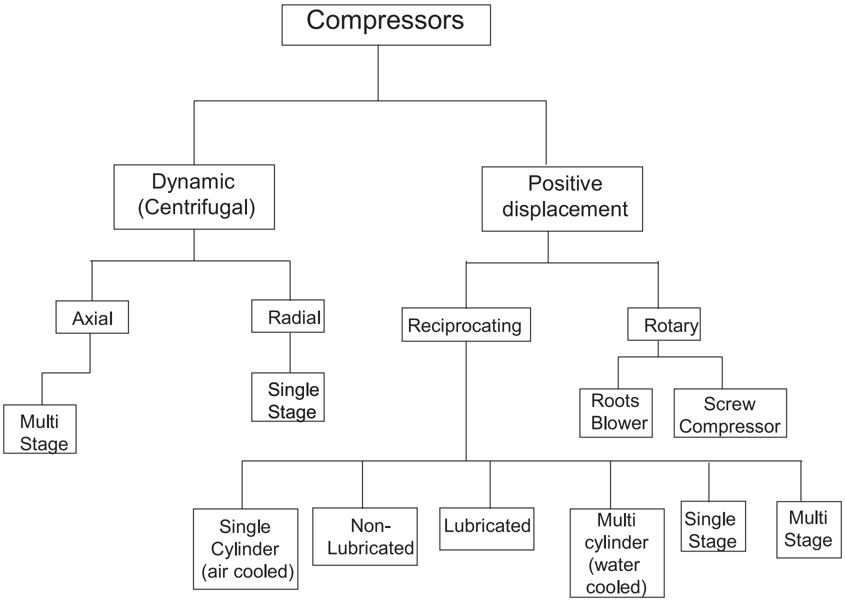


Figure 3.2 Compressor Chart

Positive Displacement Compressors

Reciprocating Compressors

Reciprocating compressors are the most widely used type for air compression. They are characterized by a flow output that remains nearly constant over a range of discharge pressures. Also, the compressor capacity is directly proportional to the speed. The output, however, is a pulsating one.

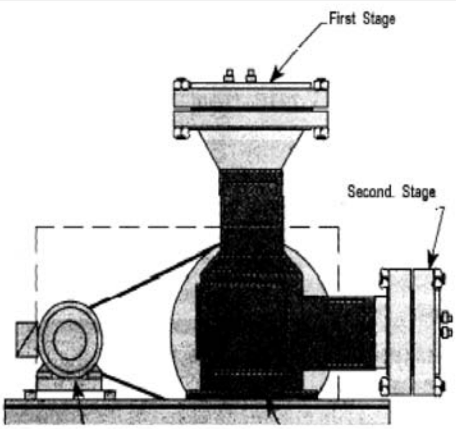


Figure 3.3 Reciprocating Compressor

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 – 150 cfm. 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.

Reciprocating compressors are also available in variety of types:

- Lubricated and non-lubricated
- Single or multiple cylinder

- Water or air-cooled.
- Single or multi stage

In the case of lubricated machines, oil has to be separated from the discharge air. Non-lubricated compressors are especially useful for providing air for instrumentation and for processes which require oil free discharge. However non-lubricated machines have higher specific power consumption (kW/cfm) as compared to lubricated types.

Single cylinder machines are generally air-cooled, while multi-cylinder machines are generally water cooled, although multi-stage air-cooled types are available for machines up to 100 kW. Water-cooled systems are more energy efficient than air-cooled systems.

Two stage machines are used for high pressures and are characterized by lower discharge temperature (140 to 160°C) compared to single-stage machines (205 to 240°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 (above 7 bar) and low capacities (less than 25 cfm). Multi staging has other benefits, such as reduced pressure differential across cylinders, which reduces the load and stress on compressor components such as valves and piston rings.

Rotary Compressors

Rotary compressors have rotors in place of pistons and give a continuous, pulsation free discharge air. They are directly coupled to the prime mover and require lower starting torque as compared to reciprocating machine. They operate at high speed and generally provide higher throughput than reciprocating compressors. Also they require smaller foundations, vibrate less, and have a lower number of parts - which means less failure rate.

Among rotary compressor, the Roots blower (also called as lobe compressor) and screw compressors are among the most widely used. The roots blower is essentially a low-pressure blower and is limited to a discharge pressure of 1 bar in single-stage design and up to 2.2 bar in two stage design.

The most common rotary air compressor is the single stage helical or spiral lube oil flooded screw air compressor. These compressors consist of two rotors, within a casing where the rotors compress the air internally. There are no valves. These units are basically oil cooled (with air cooled or water cooled oil coolers) where the oil seals the internal clearances. Since the cooling takes place right inside the compressor, the working parts never experience extreme operating temperatures. The oil has to be separated from discharge air. Because of the simple design and few wearing parts, rotary screw air compressors are easy to maintain, to operate and install.

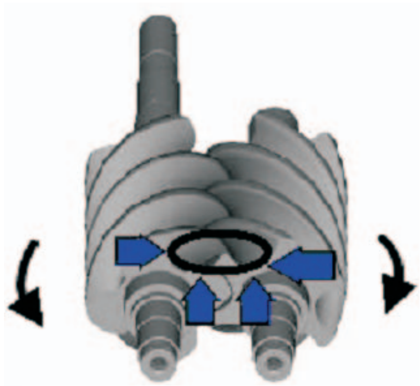


Figure 3.4 Screw Compressors

The oil free rotary screw air compressor uses specially designed air ends to compress air without oil in the compression chamber producing true oil free air. These compressors are available as air-cooled or water cooled types and provide the same flexibility as oil flooded rotary compressors.

There is a wide range of availability in configuration and in pressure and capacity. Dry types deliver oil-free air and are available in sizes up to 20,000 cfm and pressure upto 15 bar. Lubricated types are available in sizes ranging from 100 to 1000 cfm, with discharge pressure up to 10 bar.

Dynamic Compressors

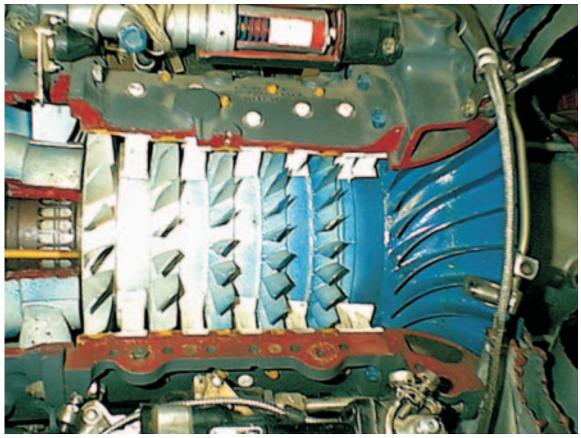


Figure 3.5 Axial Compressor

Dynamic compressors are mainly centrifugal compressors and operate on similar principles to centrifugal pump. These 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. Centrifugal machines are better suited for applications requiring very high capacities, typically above 12,000 cfm.

The centrifugal air compressor depends on transfer of energy from a rotating impeller to the air. The rotor accomplishes this by changing the momentum and pressure of the air. This momentum is converted to useful pressure by slowing the air down in a stationary diffuser.

The centrifugal air compressor is an oil free compressor by design. The oil-lubricated running gear is separated from the air by shaft seals and atmospheric vents. The centrifugal is a continuous duty compressor, with few moving parts, and is particularly suited to high volume applications, especially where oil free air is required.

A single-stage centrifugal machine can provide the same capacity as a multi-stage reciprocating compressor. Machines with either axial or radial flow impellers are available.

Axial flow compressors are suitable for higher compression ratios and are generally more efficient than radial compressors. Axial compressors typically are multi-stage machines, while radial machines are usually single-stage designs.

The general selection criteria for compressor is given in the Table 3.1

TABLE 3.1 GENERAL SELECTION CRITERIA FOR COMPRESSORS

Type of Compressor	Capacity (m^3/h)		Pressure (bar)	
	From	To	From	To
Roots blower 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

3.3 Compressor Performance

Capacity of a Compressor

Capacity of a compressor is the full rated volume of flow of gas compressed and delivered at conditions of total temperature, total pressure, and composition prevailing at the **compressor inlet**. It sometimes means actual flow rate, rather than rated volume of flow. This also termed as **Free Air Delivery (FAD)** i.e. air at atmospheric conditions at any specific location. Because the altitude, barometer, and temperature may vary at different localities and at different times, it follows that this term does not mean air under identical or standard conditions.

Compressor Efficiency Definitions

Several different measures of compressor efficiency are commonly used: *volumetric efficiency, adiabatic efficiency, isothermal efficiency and mechanical* efficiency.

Adiabatic and isothermal efficiencies are computed as the isothermal or adiabatic power divided by the actual power consumption. The figure obtained indicates the overall efficiency of compressor and drive motor.

Isothermal Efficiency

$$\text{Isothermal Efficiency} = \frac{\textit{IsothermalPower}}{\text{Actual measured input power}}$$

Isothermal power(kW) = $P_1 \times Q_1 \times \log_e r/36.7$

P_1 = Absolute intake pressure kg/ cm^2

P_2 = Absolute delivery pressure kg/ cm^2

Q_1 = Free air delivered m^3/hr.

r = Pressure ratio P_2/P_1

The calculation of isothermal power does not include power needed to overcome friction and generally gives an efficiency that is lower than adiabatic efficiency. The reported value of efficiency is normally the isothermal efficiency. This is an important consideration when selecting compressors based on reported values of efficiency.

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

Volumetric Efficiency

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

D = Cylinder bore, metre

L = Cylinder stroke, metre

S = Compressor speed rpm

χ = 1 for single acting and
2 for double acting cylinders

n = No. of cylinders

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

3.4 Compressed Air System Components

Compressed air systems consist of following major components: Intake air filters, inter-stage coolers, after coolers, air dryers, moisture drain traps, receivers, piping network, filters, regulators and lubricators (see Figure 3.6).

- **Intake Air Filters**: Prevent dust from entering compressor; Dust causes sticking valves, scoured cylinders, excessive wear etc.
- **Inter-stage Coolers**: Reduce the temperature of the air before it enters the next stage to reduce the work of compression and increase efficiency. They are normally water-cooled.
- **After Coolers**: The objective is to remove the moisture in the air by reducing the temperature in a water-cooled heat exchanger.
- **Air-dryers**: The remaining traces of moisture after after-cooler are removed using air dryers, as air for instrument and pneumatic equipment has to be relatively free of any moisture. The moisture is removed by using adsorbents like silica gel /activated carbon, or refrigerant dryers, or heat of compression dryers.
- **Moisture Drain Traps**: Moisture drain traps are used for removal of moisture in the compressed air. These traps resemble steam traps. Various types of traps used are manual drain cocks, timer based / automatic drain valves etc.
- **Receivers**: Air receivers are provided as storage and smoothening pulsating air output - reducing pressure variations from the compressor

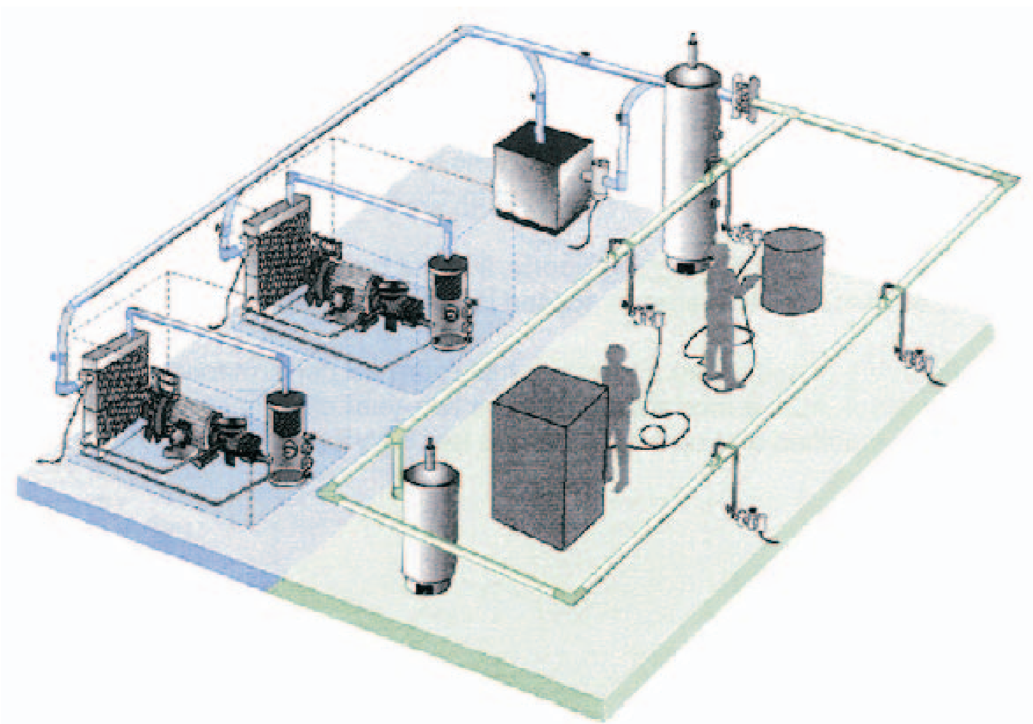


Figure 3.6 A Typical Compressed Air System Components and Network

3.5 Efficient Operation of Compressed Air Systems

Location of Compressors

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

Cool air intake

As a thumb rule, *"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 (see Table 3.2).

TABLE 3.2 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 cool ambient air from outside, as the temperature of air inside the compressor room will be a few degrees higher than the ambient temperature. 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.

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. See Table 3.3 for effect of pressure drop across air filter on power consumption.

Air filters should be selected based on the compressor type and installed as close to the compressor as possible. As a thumb rule *"For every 250 mm WC pressure drop increase across at the suction path due to choked filters etc, the compressor power consumption increases by about 2 percent for the same output"*

Hence, it is advisable to clean inlet air filters at regular intervals to minimize pressure drops. Manometers or differential pressure gauges across filters may be provided for monitoring pressure drops so as to plan filter-cleaning schedules.

TABLE 3.3 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	7.0

Dry Air Intake

Atmospheric air always contains some amount of water vapour, depending on the relative humidity, being high in wet weather. The moisture level will also be high if air is drawn from a damp area - for example locating compressor close to cooling tower, or dryer exhaust is to be avoided (see Table 3.4)

TABLE 3.4 MOISTURE IN AMBIENT AIR AT VARIOUS HUMIDITY LEVELS	
% Relative Humidity	**Kg of water vapour per hour for every 1000 m^3/min. of air at 30°C**
50	27.60
80	45.00
100	68.22

The moisture-carrying capacity of air increases with a rise in temperature and decreases with increase in pressure.

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 in the Table 3.5.

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.

Cooling Water Circuit

Most of the industrial compressors are water-cooled, wherein the heat of compression is removed by circulating cold water to cylinder heads, inter-coolers and after-coolers. The resulting warm water is cooled in a cooling tower and circulated back to compressors. The compressed air system performance depends upon the effectiveness of inter-coolers, after coolers, which in turn are dependent on cooling water flow and temperature.

Further, inadequate cooling water treatment can lead to increase, for example, in total dissolved solids (TDS), which in turn can lead to scale formation in heat exchangers. The scales, not only act as insulators reducing the heat transfer, but also increases the pressure drop in the cooling water pumping system.

TABLE 3.5 EFFECT OF ALTITUDE ON VOLUMETRIC EFFICIENCY

Altitude Meters	Barometric Pressure milli bar*	Percentage Relative Volumetric Efficiency Compared with Sea Level	
		At 4 bar	At 7 bar
Sea level	1013	100.0	100.0
500	945	98.7	97.7
1000	894	97.0	95.2
1500	840	95.5	92.7
2000	789	93.9	90.0
2500	737	92.1	87.0

*** 1 milli bar = 1.01972 x 10^{-3} kg/cm^2**

Use of treated water or purging a portion of cooling water (blow down) 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 algaecides.

Efficacy of Inter and After Coolers

Efficacy is an indicator of heat exchange performance- how well intercoolers and after coolers are performing.

Inter-coolers are provided between successive stages of a multi-stage compressor to reduce the work of compression (power requirements) - by reducing the specific volume through cooling the air - apart from moisture separation.

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" or isothermal compression. The cooling may be imperfect due to reasons described in earlier sections. Hence in actual practice, 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 (See Table 3.6).

TABLE 3.6 EFFECT OF INTER-STAGE COOLING ON SPECIFIC POWER CONSUMPTION OF A RECIPROCATING COMPRESSOR -ILLUSTRATION

Details	Imperfect Cooling	Perfect Cooling (Base Value)	Chilled Water Cooling
First Stage inlet temperature °C	21.1	21.1	21.1
Second Stage inlet temperature °C	26.6	21.1	15.5
Capacity (Nm^3/min)	15.5	15.6	15.7
Shaft Power (kW)	76.3	75.3	74.2
Specific energy consumption (kW/Nm^3/min)	4.9	4.8	4.7
Percent Change	+ 2.1	Reference	- 2.1

It can be seen from the Table 3.6 that an increase of 5.5°C in the inlet air temperature to the second stage results in a 2 % increase in the specific energy consumption. Use of water at lower temperature reduces specific power consumption. However, very low cooling water temperature could result in condensation of moisture in the air, which if not removed would lead to cylinder damage.

Similarly, inadequate cooling in after-coolers (due to fouling, scaling etc.), allow warm, humid air into the receiver, which causes more condensation in air receivers and distribution lines, which in consequence, leads to increased corrosion, pressure drops and leakages in piping and end-use equipment. Periodic cleaning and ensuring adequate flow at proper temperature of both inter coolers and after coolers are therefore necessary for sustaining desired performance. Typical cooling water requirement is given in Table 3.7.

TABLE 3.7 TYPICAL COOLING WATER REQUIREMENTS

Compressor Type	Minimum quantity of Cooling Water required (in litres per minute) for 2.85 m^3/min. FAD at 7 bar
Single-stage	3.8
Two-stage	7.6
Single-stage with after-cooler	15.1
Two-stage with after-cooler	18.9

Pressure Settings

Compressor operates between pressure ranges called as loading (cut-in) and unloading (cut-out) pressures. For example, a compressor operating between pressure setting of 6 – 7 kg/cm^2 means that the compressor unloads at 7 kg/cm^2 and loads at 6 kg/cm^2. Loading and unloading is done using a pressure switch.

For the same capacity, a compressor consumes more power at higher pressures. 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.

TABLE 3.8 TYPICAL POWER SAVINGS THROUGH PRESSURE REDUCTION

Pressure Reduction		Power Savings (%)		
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

Reducing Delivery Pressure:

The possibility of lowering (optimising) the delivery pressure settings 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. Typical power savings through pressure reduction is shown in Table 3.8.

The pressure switches must be adjusted such that the compressor cuts-in and cuts-out at optimum levels.

A reduction in the delivery pressure by 1 bar in a compressor would reduce the power consumption by 6 – 10 %.

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 (vary in flow rate).
- If different types of compressors are operated together, unload power consumptions are significant. The compressor with lowest no load power must be modulated.
- In general, the compressor with lower part load power consumption should 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 (see Table 3.9).

TABLE 3.9 TYPICAL SPECIFIC POWER CONSUMPTION OF RECIPROCATING COMPRESSORS (BASED ON MOTOR INPUT)

Pressure bar	No. of Stages	Specific Power kW/170 m^3/hour (kW / 100 cfm)
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

Segregating low and high pressure air requirements

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

Excess pressure drop due to inadequate pipe sizing, choked filter elements, improperly sized couplings and hoses represent energy wastage. The Table 3.10 illustrates the energy wastage, if the pipes are of smaller diameter.

Typical acceptable pressure drop in industrial practice is 0.3 bar in mains header at the farthest point and 0.5 bar in distribution system.

TABLE 3.10 TYPICAL ENERGY WASTAGE DUE TO SMALLER PIPE DIAMETER FOR 170 m^3/h (100 CFM) FLOW

Pipe Nominal Bore (mm)	Pressure drop (bar) per 100 meters	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

Equivalent lengths of fittings

Not only piping, but also fitting are a source of pressure losses. Typical pressure losses for various fitting are given in Table 3.11.

TABLE 3.11 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
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
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

Blowers in place of Compressed Air System

Since the compressed air system is already available, plant engineer may be tempted to use compressed air to provide air for low-pressure applications such as agitation, pneumatic conveying or combustion air. Using a blower that is designed for lower pressure operation will cost only a fraction of compressed air generation energy and cost.

Capacity Control of Compressors

In many installations, the use of air is intermittent. Therefore, some means of controlling the output flow from the compressor is necessary. The type of capacity control chosen has a direct impact on the compressor power consumption. Some control schemes commonly used are discussed below:

Automatic On / Off Control:

Automatic On /Off 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 of 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 control is suitable for small compressors.

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 unloading, a positive displacement compressor may consume up to 30 % of the full load power, depending upon the type, configuration, operation and maintenance practices.

Multi-step Control:

Large capacity reciprocating compressors are usually equipped with a multi-step control. In this type of control, unloading is accomplished in a series of steps, (0%, 25 %, 50 %, 75 % & 100 %) varying from full load down to no-load (see Table 3.12).

TABLE 3.12 POWER CONSUMPTION OF A TYPICAL 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

Throttling Control:

The capacity of centrifugal compressors can be controlled using variable inlet guide vanes. However, another efficient way to match compressor output to meet varying load requirements is by speed control (see Table 3.13).

TABLE 3.13 TYPICAL PART LOAD GAS COMPRESSION :POWER INPUT FOR SPEED AND VANE CONTROL OF CENTRIFUGAL COMPRESSORS		
System Volume, %	**Power Input (%) 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.

Avoiding Misuse of Compressed Air:

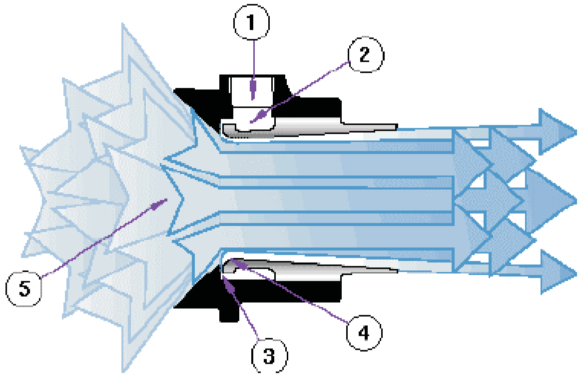
Misuse of compressed air for purposes like body cleaning, liquid agitation, floor cleaning, drying, equipment cooling and other similar uses must be discouraged. Wherever possible, low-pressure air from a blower should be substituted for compressed air, for example secondary air for combustion in a boiler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

The following Table 3.14 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, wherever applicable. The Table gives the comparison of pneumatic grinders and electrical grinders.

TABLE 3.14 TYPICAL POWER REQUIREMENTS FOR PNEUMATIC AND ELECTRICAL TOOLS				
Tool	**Wheel dia mm**	**Speed rpm**	**Air Cons. m^3/h**	**Power kW**
Pneumatic angle grinder	150	6000	102 m^3/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^3/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. It should always be remembered that safety consideration always override energy conservation.



Compressed air flows through the inlet (1) into an annular chamber (2). It is then throttled through a small ring nozzle (3) at high velocity. This primary air stream adheres to the coanda profile (4), which directs it toward the outlet. A low pressure area is created at the center (5) inducing a high volume flow of surrounding air into the primary airstream. The combined flow of primary and surrounding air

- In place of pneumatic hoists, electric hoists can be used.
- Material conveying applications by blower systems can be replaced preferably by a combination of belt / screw conveyers and bucket elevators. In a paper manufacturing facility, compressed air was used for conveying wood chips. The equivalent power consumption was 77 kW. This method of conveying was replaced by blower system consuming only 7 kW, a saving of 70 kW. This has also been widely applied in cement industry where pneumatic conveying has been replaced by bucket and screw conveyor resulting in significant energy reduction.
- When moving air really is required for an application, often sources other than compressed air can do the job. For applications like blowing of components, use of compressed *air amplifiers (see Figure)*, blowers or gravity-based systems may be possible. Brushes can sweep away debris from work in progress as effectively as high-pressure air. Blowers can be also used for this purpose. 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. If a ¼" hose pipe is kept open at a 7 bar compressed air line for cleaning for at least 1000 hours / annum, it can cost about Rs. 1.0 lakhs / annum. If absolutely necessary, compressed air should be used only with blow guns to keep the air pressure below 2 bar.
- 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.
- 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 will mix materials far more economically than high-pressure compressed air.

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, shut off valves, quick release couplings, tools and equipment. In most cases, they are due to poor maintenance and sometimes, improper installations etc.

Air leakages through Different Size Orifices

The Table 3.15 gives the amount of free air wasted for different nozzles sizes and pressure.

TABLE 3.15 DISCHARGE OF AIR (m^3/MINUTE) THROUGH ORIFICE (ORIFICE CONSTANT C_d – 1.0)

Gauge Pressure Bar	0.5 mm	1 mm	2 mm	3 mm	5 mm	10 mm	12.5 mm
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 Table 3.16 that any expenditure on stopping leaks would be paid back through energy saving.

TABLE 3.16 COST OF AIR LEAKAGE

Orifice Size mm	kW Wasted	* Cost of air leakage (Rs/Year)
0.8	0.2	8000
1.6	0.8	32000
3.1	3.0	120000
6.4	12.0	480000

* based on Rs. 5 / kWh; 8000 operating hours; air at 7.0 bar

Steps in simple shop-floor method for leak quantification

- Shut off compressed air operated equipments (or conduct test when no equipment is using compressed air).
- Run the compressor to charge the system to set pressure of operation
- Note the sub-sequent time taken for 'load' and 'unload' 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:

$$\%\ leakage = \frac{T}{(T+t)}\ x\ 100$$

(or) $$System\ leakage\ quantity\ (m^3/\min),\ q = \frac{T}{(T+t)}\ x\ Q$$

Q = Compressor capacity (m^3/min)

T = Time on load in minutes

t = Time on unload in minutes

EXAMPLE

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

Compressor capacity (m^3/minute)	=	35
Cut in pressure, kg/cm^2(g)	=	6.8
Cut out pressure, kg/cm^2(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^3/minute), q $= \dfrac{(1.5)}{(1.5)+(10.5)} \times 35$

$= 4.375\ m^3/min$

b) Leakage quantity per day, (m^3/day) $= 4.375 \times 24 \times 60 = 6300\ m^3/day$

c) Specific power for compressed air generation $= 188\ kW /(35 \times 60) m^3/hr$

$= 0.0895\ kWh/m^3$

d) Energy lost due to leakages/day $= 0.0895 \times 6300 = 564\ kWh$

Leakage Detection by Ultrasonic Leak Detector:

Leakage tests are conducted by a Leak Detector having a sensing probe, which senses when there are leakage in compressed air systems at high temperatures-beneath insulated coverings, pipelines, manifolds etc.

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.

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 of 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 into the plant equipment. Further, the bottom of the falling pipe should also be drained.

Compressed Air Filter

Although, some water, oil and dirt are removed by the separators and traps in the mains, still some are always left, which are carried over along with compressed air. Moreover, pipe systems accumulate scale and other foreign matters, such as small pieces of gasket material, jointing compounds etc. 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 harmful 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 the level is allowed to build up, it is forced through the filter element into the very system it is designed to protect.

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.

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, a 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 airflow to induce oil into their stream. It is advisable to install filters, regulators and lubricators as close as possible to the equipment being served.

Air Dryers

There are certain applications where air must be free from moisture and have a lower dew point. Dew point is the temperature at which moisture condenses. 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), adsorption and refrigerated dryers. They produce dry air with -10°C to -40°C dew point, depending on the type of dryers. Refer Table 3.17 for moisture content in air and Table 3.18 for typical pressure dew point and power consumption data for dryers.

TABLE 3.17 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 3.18 TYPICAL 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^3/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

Air Receivers

The air receiver dampens pulsations entering the discharge line from the compressor; serves as a reservoir for sudden or unusually heavy demands in excess of compressor capacity; prevents too frequent loading and unloading (short cycling) of the compressor; and separates moisture and oil vapour, allowing the moisture carried over from the after coolers to precipitate.

The air receiver should be generously sized to give a large cooling surface and even out the pulsation in delivered air pressure from reciprocating compressor. 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.

Another approximation can be to size the receiver volume to be 5% of the rated hourly free air output. Providing an air receiver near the load end, where there is sudden high demand lasting for a short period, would avoid the need to provide extra capacity.

Loss of air pressure due to friction

The loss of pressure in piping is caused by resistance in pipe fittings and valves, which dissipates energy by producing turbulence. The piping system will be designed for a maximum allowable pressure drop of 5 percent from the compressor to the most distant point of use.

Piping layout

Where possible the piping system should be arranged as a closed loop or "ring main" to allow for more uniform air distribution to consumption points and to equalize pressure in the piping. Separate services requiring heavy air consumption and at long distances from the compressor unit should be supplied by separate main airlines. Pipes are to be installed parallel with the lines of the building, with main and branch headers sloping down toward a dead end. Traps will be installed in airlines at all low points and dead ends to remove condensed moisture. Automatic moisture traps used for this purpose are effective only when the air has been cooled and the moisture has precipitated. Branch headers from compressed air mains will be taken off at the top to avoid picking up moisture.

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. One of the objectives of a good compressed air management system would be to minimize unloading to the least as unloading consumes up to 30% of full load power.

One way of doing this is to use a smaller compressor.

Decentralized compressors, as against centralized compressors often serve this purpose better by having the option to switch off when air is not need in a particular section/equipment.

If a compressor is oversized and operates at unloading mode for long periods, an economical way will be to suitably change the pulley size of the motor or compressor and reduce the RPM to de-rate the compressor to a lower capacity.

With decreasing cost of variable speed drives, it has become a viable option to maintain constant pressure in the system and to avoid unloading operations by varying the speed of the compressor. However, caution should be taken for operations at very low speeds, since it will affect the lubricating system. This can be overcome by providing a separate lube oil system independent of the compressor.

3.6 Compressor Capacity Assessment

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 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 wherein 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 blanking it, 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 line 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_2 (in the receiver) from initial pressure P_1.

Calculate the capacity as per the formulae given below :

Actual Free air discharge

$$Q = \frac{P_2 - P_1}{P_0} \times \frac{V}{T} \ Nm^3 / Minute$$

Where

P_2	=	Final pressure after filling (kg/cm^2 a)
P_1	=	Initial pressure (kg/cm^2a) after bleeding
P_0	=	Atmospheric Pressure (kg/cm^2 a)
V	=	Storage volume in m^3 which includes receiver, after cooler, and delivery piping
T	=	Time take to build up pressure to P_2 in minutes

The above equation is relevant where the compressed air temperature is same as the ambient air temperature, i.e., perfect isothermal compression. In case the actual compressed air temperature at discharge, say $t_2{}^0C$ is higher than ambient air temperature say $t_1{}^0C$ (as is usual case), the FAD is to be corrected by a factor $(273 + t_1) / (273 + t_2)$.

EXAMPLE

An instrument air compressor capacity test gave the following results (assume the final compressed air temperature is same as the ambient temperature) - Comment?

Piston displacement	:	16.88 m^3/minute
Theoretical compressor capacity	:	14.75 m^3/minute @ 7 kg/cm^2
Compressor rated rpm 750	:	Motor rated rpm : 1445
Receiver Volume	:	7.79 m^3
Additional hold up volume, i.e., pipe / water cooler, etc., is	:	0.4974 m^3
Total volume	:	$7.79 + 0.4974 = 8.287 m^3$
Initial pressure P_1	:	0.5 kg/cm^2
Final pressure P_2	:	7.03 kg/cm^2
Atmospheric pressure P_0	:	1.026 kgf/cm^2A
Time taken to build up pressure	:	4.021 minutes
Compressor output m^3/minute	:	$\frac{(P_2 - P_1) \times \text{Total Volume}}{\text{Atm. Pressure} \times \text{Pumpup time}}$
	:	$\frac{(7.03 - 0.5) \times 8.287}{1.026 \times 4.021} = 13.12 \ m^3/\text{minute}$

Capacity shortfall with respect to 14.75 m^3/minute rating is 1.63 m^3/minute i.e., 11.05%, which indicates compressor performance needs to be investigated further.

3.7 Checklist for Energy Efficiency in Compressed Air System

- Ensure air intake to compressor is not warm and humid by locating compressors in well-ventilated area or by drawing cold air from outside. Every 4°C rise in air inlet temperature will increase power consumption by 1 percent.

- Clean air-inlet filters regularly. Compressor efficiency will be reduced by 2 percent for every 250 mm WC pressure drop across the filter.
- Keep compressor valves in good condition by removing and inspecting once every six months. Worn-out valves can reduce compressor efficiency by as much as 50 percent.
- Install manometers across the filter and monitor the pressure drop as a guide to replacement of element.
- Minimize low-load compressor operation; if air demand is less than 50 percent of compressor capacity, consider change over to a smaller compressor or reduce compressor speed appropriately (by reducing motor pulley size) in case of belt driven compressors.
- Consider the use of regenerative air dryers, which uses the heat of compressed air to remove moisture.
- Fouled inter-coolers reduce compressor efficiency and cause more water condensation in air receivers and distribution lines resulting in increased corrosion. Periodic cleaning of inter-coolers must be ensured.
- Compressor free air delivery test (FAD) must be done periodically to check the present operating capacity against its design capacity and corrective steps must be taken if required.
- If more than one compressor is feeding to a common header, compressors must be operated in such a way that only one small compressor should handle the load variations whereas other compressors will operate at full load.
- The possibility of heat recovery from hot compressed air to generate hot air or water for process application must be economically analyzed in case of large compressors.
- Consideration should be given to two-stage or multistage compressor as it consumes less power for the same air output than a single stage compressor.
- If pressure requirements for processes are widely different (e.g. 3 bar to 7 bar), it is advisable to have two separate compressed air systems.
- Reduce compressor delivery pressure, wherever possible, to save energy.
- Provide extra air receivers at points of high cyclic-air demand which permits operation without extra compressor capacity.
- Retrofit with variable speed drives in big compressors, say over 100 kW, to eliminate the `unloaded' running condition altogether.
- Keep the minimum possible range between load and unload pressure settings.
- Automatic timer controlled drain traps wastes compressed air every time the valve opens. So frequency of drainage should be optimized.
- Check air compressor logs regularly for abnormal readings, especially motor current cooling water flow and temperature, inter-stage and discharge pressures and temperatures and compressor load-cycle.
- Compressed air leakage of 40 – 50 percent is not uncommon. Carry out periodic leak tests to estimate the quantity of leakage.
- Install equipment interlocked solenoid cut-off valves in the air system so that air supply to a machine can be switched off when not in use.
- Present energy prices justify liberal designs of pipeline sizes to reduce pressure drops.
- Compressed air piping layout should be made preferably as a ring main to provide desired pressures for all users.
- A smaller dedicated compressor can be installed at load point, located far off from the central compressor house, instead of supplying air through lengthy pipelines.

- All pneumatic equipment should be properly lubricated, which will reduce friction, prevent wear of seals and other rubber parts thus preventing energy wastage due to excessive air consumption or leakage.
- Misuse of compressed air such as for body cleaning, agitation, general floor cleaning, and other similar applications must be discouraged in order to save compressed air and energy.
- Pneumatic equipment should not be operated above the recommended operating pressure as this not only wastes energy bus can also lead to excessive wear of equipment's components which leads to further energy wastage.
- Pneumatic transport can be replaced by mechanical system as the former consumed about 8 times more energy. Highest possibility of energy savings is by reducing compressed air use.
- Pneumatic tools such as drill and grinders consume about 20 times more energy than motor driven tools. Hence they have to be used efficiently. Wherever possible, they should be replaced with electrically operated tools.
- Where possible welding is a good practice and should be preferred over threaded connections.
- On account of high pressure drop, ball or plug or gate valves are preferable over globe valves in compressed air lines.

9. DG SET SYSTEM

Syllabus
Diesel Generating system: Factors affecting selection, Energy performance assessment of diesel conservation avenues

9.1 Introduction

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 700–900°C. 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.

Four Stroke - Diesel Engine

The 4 stroke operations in a diesel engine are: induction stroke, compression stroke, ignition and power stroke and exhaust stroke.

1st : Induction stroke - while the inlet valve is open, the descending piston draws in fresh air.

2nd : Compression stroke - while the valves are closed, the air is compressed to a pressure of up to 25 bar.

3rd : 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.

4th : Exhaust stroke - the exhaust valve is open and the rising piston discharges the spent gases from the cylinder.

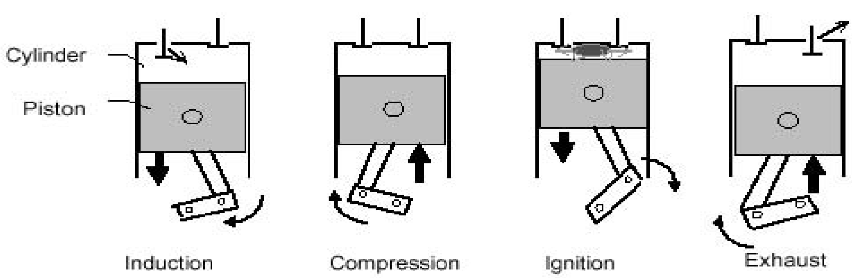


Figure 9.1 Schematic Diagram of Four-Stroke Diesel Engine

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.

DG Set as a System

A diesel generating set 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.

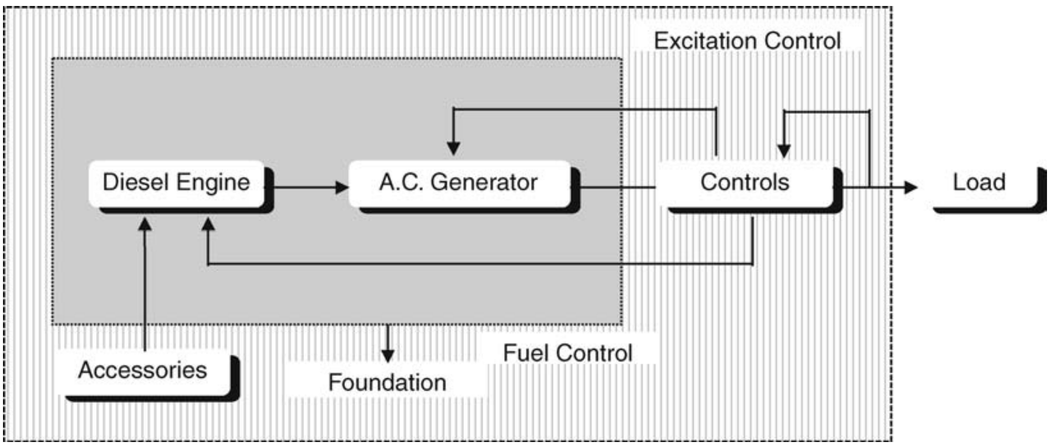


Fig 9.2 DG Set System

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 build up 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.

TABLE 9.1 COMPARISON OF DIFFERENT TYPES OF CAPTIVE POWER PLANT				
Description	**Units**	**Combined GT & ST**	**Conventional Steam Plant**	**Diesel Engine Power Plants**
Thermal Efficiency	%	40 – 46	33 – 36	43 – 45
Initial Investment of Installed Capacity	Rs./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 1970s to a value 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 favourably 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 to further increase the engine rated output. The net result – lower fuel consumption per kWh and further increase in overall thermal efficiency.

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.

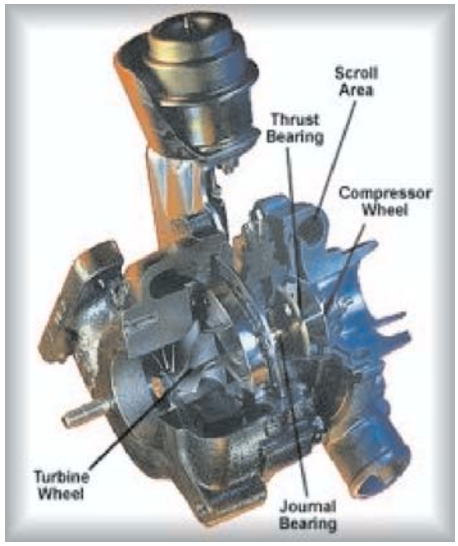


Figure 9.3 Turbocharger

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:

a) 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 :

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	=	$\sqrt{3}$ V I
kVA Rating	=	kVA / 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 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 (includes 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 upto 75 % of the kVA of Genset.

Unbalanced Load Effects

It is always recommended to have the load as much balanced as possible, since unbalanced loads can 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 and cooling water temperature derate diesel engine output as shown in following Tables: 9.2 and 9.3.

TABLE 9.2 ALTITUDE AND INTAKE TEMPERATURE CORRECTIONS

Correction Factors For Engine Output				
Altitude Correction			Temperature Correction	
Altitude Meters over MSL	Non Super Charged	Super Charged	Intake °C	Correction Factor
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 9.3 DERATING DUE TO AIR INTER COOLER WATER INLET TEMPERATURE		
Water Temperature °C	**Flow %**	**Derating %**
25	100	0
30	125	3
35	166	5
40	166	8

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 A.C. 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. Manufacturers 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 A.C. 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 utilisation 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 A.C. 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 A.C. generator leads to unbalanced set of voltages and additional heating in A.C. 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 A.C. 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 A.C. 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 utilisation 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

A typical energy balance in a DG set indicates following break-up:

Input	:	100%	Thermal Energy
Outputs	:	35%	Electrical Output
		4%	Alternator Losses
		33%	Stack Loss through Flue Gases
		24%	Coolant Losses
		4%	Radiation Losses

Among these, stack losses through flue gases or the exhaust flue gas losses on account of existing flue gas temperature of 350°C to 550°C, constitute the major area of concern towards operational economy. It would be realistic to assess the Waste Heat Recovery (WHR) potential in relation to quantity, temperature margin, in kcals/Hour as:

Potential WHR = (kWh Output/Hour) x (8 kg Gases / kWh Output) x 0.25 kcal/kg°C x (tg – 180°C)

Where, t_g is the gas temperature after Turbocharger, (the criteria being that limiting exit gas temperature cannot be less than 180°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°C exhaust gas temperature, the waste heat potential works out to:

800 kWh x 8 kg gas generation / kWh output x 0.25 kCal/kg°C x (480 – 180), i.e., 4,80,000 kCal/hr

While the above method yields only the potential for heat recovery, the actual realisable 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.

TABLE 9.4 TYPICAL FLUE GAS TEMPERATURE AND FLOW PATTERN IN A 5-MW DG SET AT VARIOUS LOADS

100% Load	11.84 kgs/Sec	370°C
90% Load	10.80 kgs/Sec	350°C
70% Load	9.08 kgs/Sec	330°C
60% Load	7.50 kgs/Sec	325°C

If the normal load is 60%, the flue gas parameters for waste heat recovery unit would be 320°C inlet temperature, 180°C outlet temperature and 27180 kgs/Hour gas flow.

At 90% loading, however, values would be 355°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 favourable.

* 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.

The favourable incentives offered by Government of India for energy efficient equipment and technologies (100% depreciation at the end of first year), make the waste heat recovery option. Payback period is only about 2 years

9.4 Energy Performance Assessment of DG Sets

Routine energy efficiency assessment of DG sets on shopfloor 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
   e) 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 ± 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 9.5 is useful for monitoring the performance

TABLE 9.5TYPICAL 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
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 DG 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 manufacturers' 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.