ENERGY MANAGEMENT SERIES

13

FOR INDUSTRY COMMERCE AND INSTITUTIONS

Fans and Pumps

Energy, Mines and Resources Canada
Énergie, Mines et Ressources Canada

Canada

PREFACE

Much has been learned about the art and science of managing energy during the past decade. Today, energy management is a seriously applied discipline within the management process of most successful companies.

Initially, in the early 1970's, energy conservation programs were established to alleviate threatened shortages and Canada's dependency on off-shore oil supplies. However, dramatic price increases quickly added a new meaning to the term "energy conservation" — reduce energy costs!

Many industrial, commercial and institutional organizations met the challenge and reduced energy costs by up to 50%. Improved energy use efficiency was achieved by such steps as employee awareness programs, improved maintenance procedures, by simply eliminating waste, as well as by undertaking projects to upgrade or improve facilities and equipment.

In order to obtain additional energy savings at this juncture a greater knowledge and understanding of technical theory and its application is required in addition to energy efficiency equipment itself.

At the request of the Canadian Industry Program for Energy Conservation, the Commercial and Institutional Task Force Program and related trade associations, the Industrial Energy Division of the Department of Energy, Mines and Resources Canada, has prepared a series of energy management and technical manuals.

The purpose of these manuals is to help managers and operating personnel recognize energy management opportunities within their organizations. They provide the practitioner with mathematical equations, general information on proven techniques and technology, together with examples on how to save energy.

For further information concerning the manuals listed below or regarding material used at seminars/workshops including actual case studies, please write to:

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INTRODUCTION

Before the harnessing of power through systems using steam and electricity, natural convection and gravity were used to move air and liquids. The development of powered devices began with the application of power systems to elementary devices such as swaying vanes for moving air and buckets for moving water.

The availability of low cost power systems allowed devices and systems to become larger and more relied upon. Consumption of energy by the driving devices was overshadowed by the remarkable benefits, and little attention was given to management of the energy input. In recent years the cost of all forms of energy has rapidly increased. This cost burden must be reduced at every opportunity.

Because some types of fans or pumps are used in most Industrial, Commercial and Institution facilities, the effective utilization of these devices is an important part of the management of the facility. Energy saved through implementation of *Energy Management Opportunities* can represent a considerable cost saving.

Purpose

The purpose of this module can be summarized by the following.

- Introduce the purposes and functions of fans and pumps as used in the Industrial, Commercial and Institutional sectors.
- Define methods of determining the approximate energy consumption.
- Provide an awareness of potential energy and cost savings available through the implementation of Energy Management Opportunities.
- Provide a series of *Worksheets* to establish a standard method of calculating energy and cost savings for the noted Energy Management Opportunities.

It must be understood that this is not a design manual, but a presentation of information and calculation tools which can be used to identify Energy Management Opportunities, and to estimate potential energy and cost savings.

Contents

The module is divided into separate sections for fans and pumps. Each section is divided into the following subsections.

- *Fundamentals*, describes the basic operating principles and provides simplified equations for estimating the energy requirements of the device. Diagrams illustrate the principles, and worked examples demonstrate the application of equations.
- *Equipment/Systems* describes the devices and discusses their characteristics with respect to energy consumption.
- A series of *Energy Management Opportunities* provides a suggested list of topics for consideration. Fundamental equations are used in worksheets to produce sample calculations of energy saving, cost saving and simple payback.

Reference material for the sections is in *Appendices*. These include a glossary, tables, common conversions, worksheets and specific details for energy calculations pertaining to *Electric Motor Drives*.

Energy Audit Methods

Energy Management Opportunities exist where fans or pumps are used. Many of these opportunities are recognizable during a *walk through audit* of the facility. This audit is usually more meaningful if a "fresh pair of eyes", generally familiar with energy management, is involved.

Typical opportunities noted during a walk through audit might include a fan with inlet vanes that remain partially closed and a pump operating against a modulating valve. Alert management, operating staff and good maintenance procedures can, with a little effort, reduce energy use thereby saving money.

Not all items noted during a walk through audit are easy to analyze. For example, it may be observed that a pump supplying cooling water to a process line is operating continuously although only part of the process line is in use. It is apparent that energy cost saving can be achieved by installing a smaller pump that is more appropriately matched to the process requirement. This leads to certain key questions.

- How much water flow is required, and at what pressure?
- How can the water piping system be modified to permit portions of the system to be shut off without affecting the active process?
- What are the implications for future reactivation of the rest of the process line?
- How long will it take for the energy cost saving to pay back the capital cost?
- Are there other solutions, such as applying a variable speed drive to the existing pump, that offer an equal or faster payback?

This example requires a *diagnostic audit* to mathematically establish the reduced flow rate, the reduction in energy consumption and the potential cost saving. With the cost saving plus the estimated cost to supply and install the modified system, simple payback calculations can establish the financial viability of the opportunity.

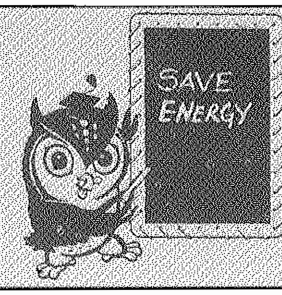
The implementation of Energy Management Opportunities can be divided into three categories:

- *Housekeeping* refers to an energy management action that is *repeated on a regular basis but never less than once a year.* Examples are adjustment of belt drives, bearing lubrication and cleaning of fan blades.
- *Low Cost* refers to an energy management action that is *done once and for which the cost is not considered great*. Examples are changing of belt drives and trimming of pump impellers.
- *Retrofit* refers to an energy management action that is *done once and for which the cost is significant.* Examples are complete replacement of fans and pumps.

The division between low cost and retrofit is normally a function of the size, type and financial policy of the organization.

SECTION 1 FANS

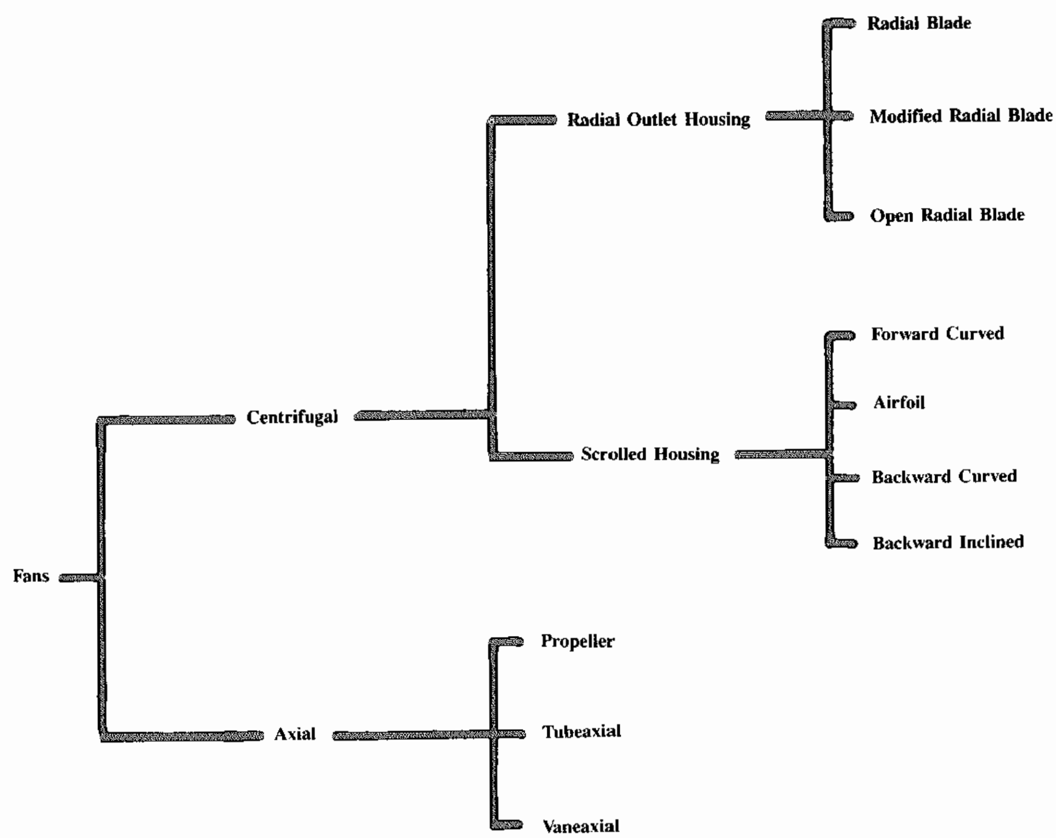
FUNDAMENTALS



Fans provide the motive force to move air against the resistance of an air conveying system. They can be used to supply air to a space or to exhaust contaminated air from a space. Ducted air conveying systems allow fans to be located outside the space and permit an individual fan to serve more than one space.

Fan Types

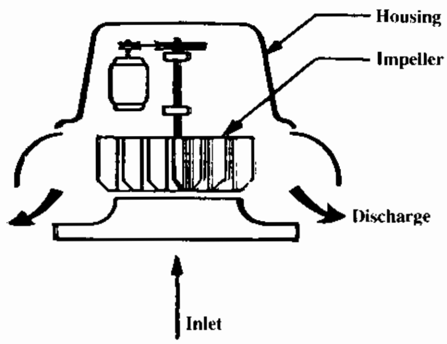
Figure 1 provides a classification of the various types of fans commonly found in Industrial, Commercial or Institutional facilities.



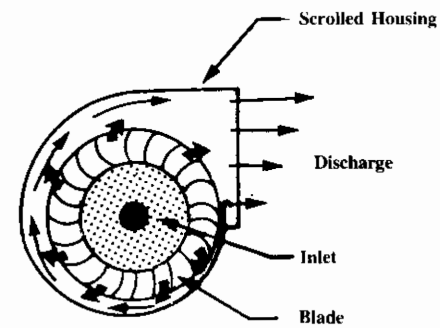
Fan Types
Figure 1

Fan components consist essentially of a rotating impeller and a housing to collect and direct air flow. The fundamentals of fan operation are discussed under the two major classifications, *centrifugal* and *axial*.

Centrifugal fans move air by the centrifugal force that is produced by moving the air between the rotating impeller blades, and by the inertia generated by the velocity of the air leaving the impeller. A centrifugal fan may have a continuous *radial outlet housing* (Figure 2), or a *scrolled housing* with a single outlet, (Figure 3.) The continuous radial outlet housing provides efficient release of the air where no downstream duct connections are required. The scrolled housing collects the high velocity air into a compact, unidirectional air stream for ducted systems.

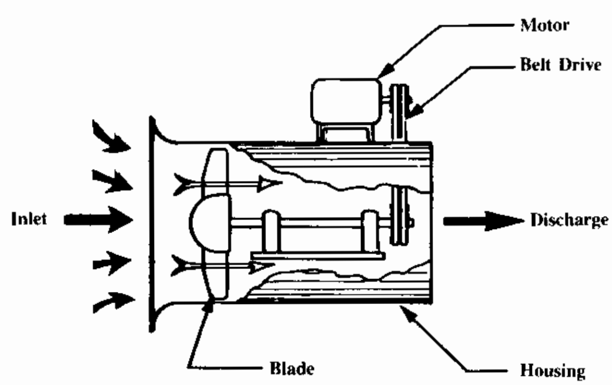


Radial Outlet Centrifugal Fan
Figure 2



Scrolled Housing Centrifugal Fan
Figure 3

Axial fans move air by the change in velocity of the air passing over the impeller blades. There is no energy added to the air by centrifugal forces. Figure 4 is a simplified version of an axial fan.



Air Movement By An Axial Fan
Figure 4

Fan Operation

Air is a compressible gas. Although the density of a gas increases with increasing pressure or decreasing temperature, the convention for the range of pressure and temperature encountered in fan calculations in this module is to consider air noncompressible and of constant density. This is considered sufficiently accurate for estimating purposes. Where accurate calculations are required for high or low temperature air streams, the effect of temperature on the air density must be taken into account. The effect of air density on fan performance is described in Publication 201 of the Air Movement and Control Association (AMCA) and in the Equipment Handbook of the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE).

Air Flow Measurements

The *total pressure* of an air stream flowing in a duct is the sum of the *static pressure* exerted on the sidewalls of the duct and the *velocity pressure* of the moving air (Figure 5). The illustrated Pitot tube is a common device for measuring the velocity pressure. Measurements should be taken in accordance with traverse details shown. The velocity pressure should be calculated for each traverse position and the readings averaged. Using these measurements, the velocity of air in a duct can then be calculated.

$$\text{vel} = 0.764 \times \left(\frac{T \times P_V}{B}\right)^{0.50}$$

Where, vel = average air velocity (m/s)

T = temperature (K) (°C + 273.15)

P_V = velocity pressure (Pa)

B = barometric pressure [kPa(absolute)]

0.764 = equation constant and conversion of units

The equation can be simplified for standard conditions of 20°C and 101.325 kPa barometric pressure.

$$\text{vel} = 1.30 \times P_V^{0.50}$$

The volume air flow rate can then be calculated.

$$f_a = \text{vel} \times Ad \times 1000$$

Where, f_a = volume air flow rate (L/s)

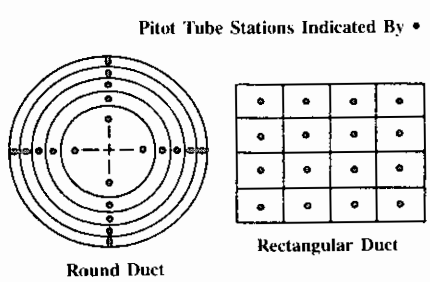
Ad = area of duct (m^2)

1000 = conversion of units

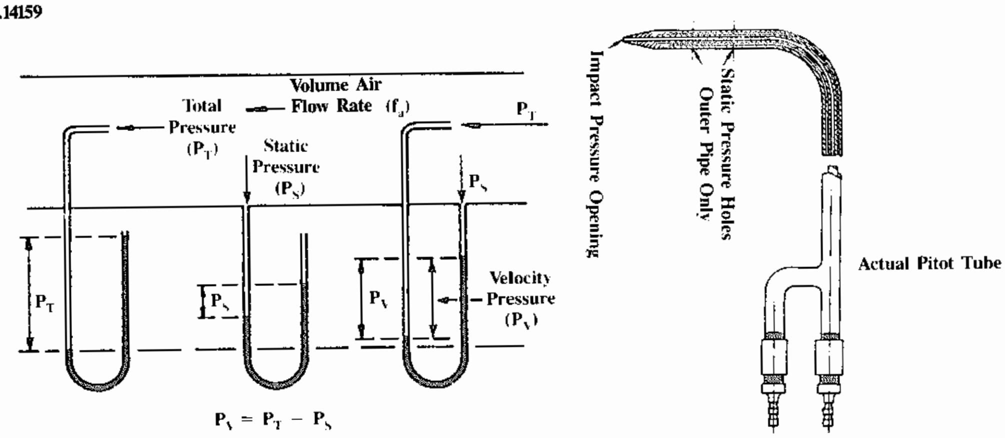
Note: For round ducts $Ad = \frac{\pi D^2}{4}$

where, D = duct diameter (m)

$\pi = 3.14159$



Measurment Points



Pitot Tube Flow Measurement

Figure 5

Fan Performance Measurements

The performance of fans is easily determined by calculations based on measured data taken at the fan inlet and outlet using the methods previously described. Several factors, however, can affect the accuracy of the field measurements.

- Air flow not at right angles to the measurement plane.
- Non-uniform velocity distribution.
- Irregular cross sectional shape of the duct or passageway.
- Air leaks between the measurement plane and the fan.

Field Performance Measurements, Publication 203 by the Air Movement and Control Association Inc. (AMCA) provides information about equipment and procedures for more precise fan performance measurements.

The *total differential pressure* across the fan can be calculated from the measured data.

$$DP_T = P_{So} + P_{Vo} - P_{Si} - P_{Vi}$$

Where, DP_T = total differential pressure (Pa)

P_{So} = static pressure at outlet [Pa(gauge)]

P_{Vo} = velocity pressure at outlet (Pa)

P_{Si} = static pressure at inlet [Pa(gauge)]

P_{Vi} = velocity pressure at inlet (Pa)

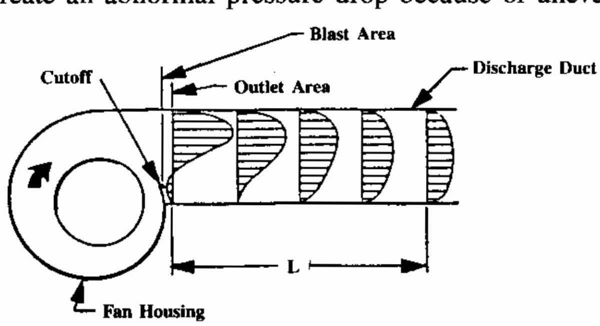
The *total fan static differential pressure* can also be calculated.

$$DP_S = P_{So} - P_{Si} - P_{Vi}$$

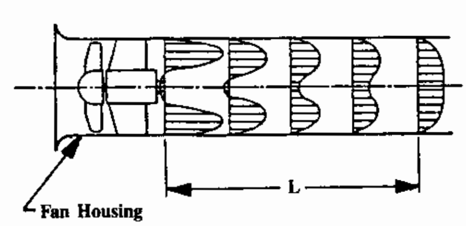
Where, DP_S = total fan static differential pressure (Pa)

The effect of inlet and outlet conditions have not been included in these equations, however, they provide a reasonable basis for the further calculation of fan power and static efficiency.

Figure 6 illustrates how the *velocity profile* of air changes as it is discharged into a duct from a single outlet centrifugal fan. The velocity of air leaving the fan is unevenly distributed across the outlet duct but, as the air moves away from the fan, the velocity distribution becomes more uniform. Figure 7 illustrates the velocity profile of air as it is discharged into a duct from an axial fan. The air leaving the fan has its highest velocity near the centre of the blades. L is the distance from the outlet of the fan to the point where uniform flow in the duct is considered to be achieved for accurate measurement of flow. Elbows or branch connections within this distance will create an abnormal pressure drop because of uneven velocity conditions.



Discharge of Air From a Single Outlet Centrifugal Fan
Figure 6



Discharge Of Air From An Axial Fan
Figure 7

For a velocity of 12.7 m/s or less, the distance L for round duct can be estimated by the following equation.

$$L = 2.5 \text{ x duct diameter}$$

For a velocity greater than 12.7 m/s the distance can be estimated by the following equation.

$$L = (2.5 \text{ x duct diameter}) + \left[\left(\frac{\text{average air velocity} - 12.7}{5}\right) \text{ x duct diameter}\right]$$

For the purpose of calculating the distance L for a rectangular duct with height a and width b, the equivalent round duct diameter must first be calculated with the following equation.

$$D = (1.273 \text{ x } a \text{ x } b)^{0.50}$$

Where, D = equivalent diameter of round duct (m)

a = height of rectangular duct (m)

b = width of rectangular duct (m)

$1.273 = \frac{4}{\pi}$, being the ratio of the square of the diameter to the area of a circle

The following is an example of determining the duct length for uniform air flow.

A 0.6 m high by 1 m wide duct from a centrifugal fan outlet carries air at an average velocity of 15.2 m/s. It is desired to determine the minimum distance from the fan at which a branch duct should be connected.

The equivalent diameter of round duct, $D = (1.273 \text{ x } 0.6 \text{ x } 1.0)^{0.50}$

$= 0.874$ m

The required distance, $L = (2.50 \text{ x } 0.874) + \left[\left(\frac{15.2 - 12.7}{5}\right) \text{ x } 0.874\right]$

$= 2.622$ m

Fan Laws

The fan laws relate the performance variables for a particular centrifugal fan. The variables used in this module are flow, fan speed, differential pressure and power. The fan laws relating these variables can be stated in written form and as equations.

- The flow varies in proportion to the fan speed.
- The total differential pressure varies in proportion to the square of the fan speed.
- The power required varies in proportion to the cube of the fan speed.

$$\frac{f_{a2}}{f_{a1}} = \frac{n_2}{n_1}, \quad \frac{DP_{T2}}{DP_{T1}} = \left(\frac{n_2}{n_1}\right)^2, \quad \frac{Wf_{i2}}{Wf_{i1}} = \left(\frac{n_2}{n_1}\right)^3$$

Where, f_{a1}, f_{a2} = initial and resulting volume air flow rates (L/s)

n_1, n_2 = initial and revised fan speeds (rpm)

DP_{T1}, DP_{T2} = initial and resulting total differential pressures (Pa)

Wf_{i1}, Wf_{i2} = initial and resulting power inputs to the fan shaft (kW)

The fan law equations can be combined to determine interrelated performance characteristics. As an example, the pressure equation and the power equation can be combined to produce the following equation for the calculation of fan power resulting from a change in the total differential pressure.

$$Wf_{i2} = Wf_{i1} \times \left(\frac{DP_{T2}}{DP_{T1}}\right)^{1.5}$$

The flow and power equation fan laws can also be combined to calculate the fan power resulting from a change in the air flow.

$$Wf_{i2} = Wf_{i1} \times \left(\frac{f_{a2}}{f_{a1}}\right)^{3}$$

These relationships are true only for a fan on an air conveying system that has constant characteristics. Where the system charactristics vary, such as in a system with automatic flow control dampers, the system conditions must first be determined for each flow rate.

The following is an application of fan laws.

A fan delivering 3000 L/s of air against 500 Pa total differential pressure is operating at 700 rpm and requires 2.9 kW of power. The fan operates 3600 hours per year. A detailed study of the air system concludes that the air flow can be reduced by 500 L/s. This example illustrates the reduction in total differential pressure and power requirements.

Combining the flow equation and the pressure equation fan laws, the reduced total differential pressure at the reduced flow can be calculated by the following equation.

$$DP_{T2} = DP_{T1} \times \left(\frac{f_{a2}}{f_{a1}}\right)^{2}$$

$$= 500 \times \left(\frac{2500}{3000}\right)^{2}$$

$$= 347 \text{ Pa}$$

By combining the pressure and power equation fan laws, the power requirement at the reduced total differential pressure can also be calculated.

$$Wf_{i2} = Wf_{i1} \times \left(\frac{DP_{T2}}{DP_{T1}}\right)^{1.5}$$

$$= 2.9 \times \left(\frac{347}{500}\right)^{1.5}$$

$$= 1.68 \text{ kW}$$

The same reduced power requirement can alternately be calculated using the combined flow and power equation fan laws.

$$Wf_{i2} = Wf_{i1} \times \left(\frac{f_{a2}}{f_{a1}}\right)^{3}$$

$$= 2.9 \times \left(\frac{2500}{3000}\right)^{3}$$

$$= 1.68 \text{ kW}$$

For an electric energy cost of $0.05 per kWh and an operation period of 3600 hours per year, the annual saving can be calculated.

$$\text{Saving} = (2.9 - 1.68) \times 0.05 \times 3600$$

$$= \$220/\text{yr}$$

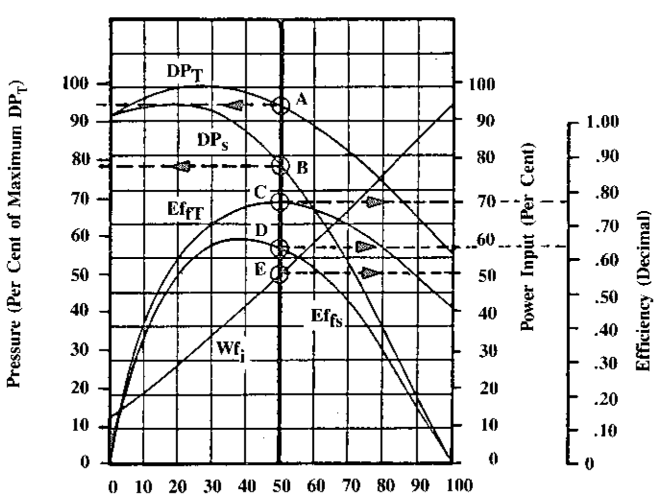
Fan Performance

Manufacturer's performance tables, when available, are the most reliable source of data for fan operating characteristics and power requirements. When such data is not available, the power requirement of a particular fan can be estimated using measured data, the fan laws and the fan power equations that are discussed in this section.

Performance Curves

Figure 8 is a typical graphical illustration of the relative performance characteristics of a fan. It covers the range from free delivery with no flow obstruction, to no delivery with a blocked discharge for a particular impeller size and speed. For example, at 50 per cent of maximum free delivery air flow, the following characteristics can be determined from the curves.

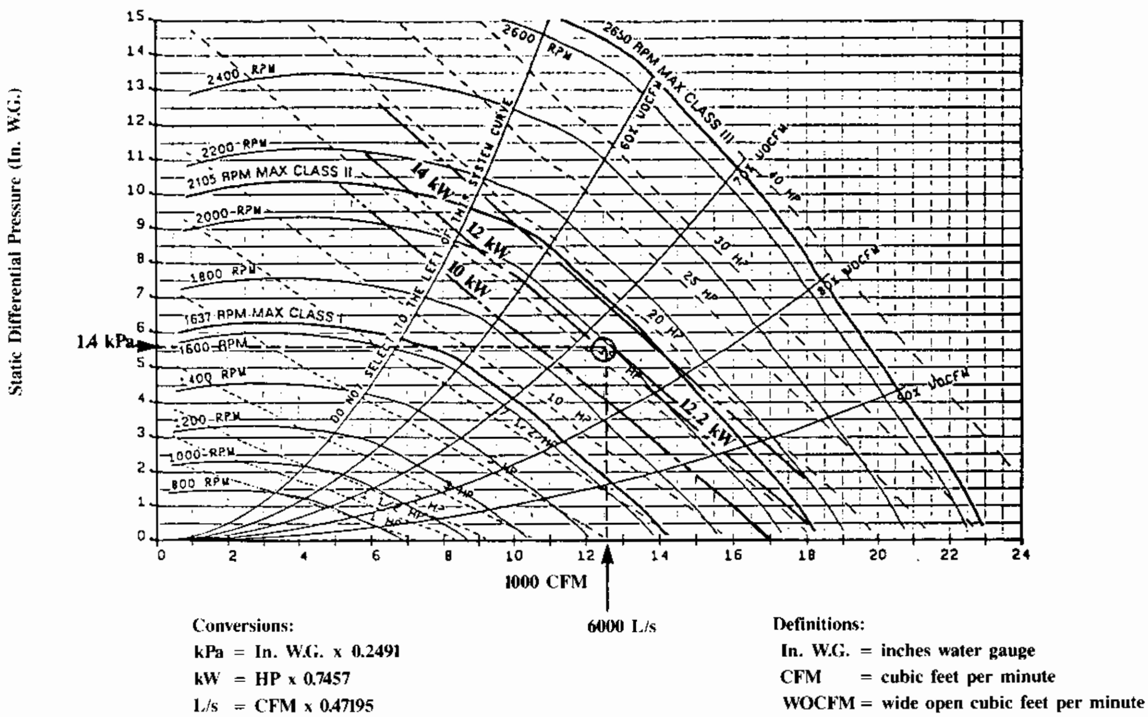
Total differential pressure, DP_T	94% (Point A)
Total fan static differential pressure, DP_S	79% (Point B)
Fan total efficiency, Ef_{fT}	0.76 (Point C)
Fan static efficiency, Ef_{fS}	0.63 (Point D)
Power input to the fan shaft, Wf_i	50% (Point E)



DP_T = Total Differential Pressure
DP_S = Total Fan Static Differential
Ef_{fT} = Fan Total Efficiency
Ef_{fS} = Fan Static Efficiency
Wf_i = Power Input To The Fan Shaft

Typical Fan Performance Characteristics
Figure 8

Figure 9 is a typical fan performance curve compiled by plotting the values from a manufacturer's performance table. The manufacturer's tabulated data can, however, be used directly for most applications. The curve is presented to graphically illustrate the relationships between the performance variables. For example, at 6000 L/s flow and 1.4 kPa differential static pressure, the fan must operate at 2000 rpm and requires a power input of 12.2 kW.



Typical Fan Performance Curve
Figure 9

Fan Efficiency

Fan efficiency may be expressed as either the *total efficiency* or the *static efficiency*, and normally excludes the motor and drive efficiencies.

Fan *total efficiency* is the total output of useful energy divided by the power input at the fan shaft. The total efficiency, or mechanical efficiency, is calculated by the following equation.

$$Ef_{fT} = \frac{f_a \times DP_T}{1000 \times Wf_i}$$

Where, Ef_{fT} = fan total efficiency (decimal)

f_a = volume air flow rate (L/s)

Wf_i = power input to the fan shaft (kW)

1000 = conversion of units

The total efficiency is a true indication of fan performance when the total differential pressure is known or can be accurately determined.

Fan *static efficiency* is the power output based on the total fan differential static pressure divided by the power input to the fan shaft and can be calculated by the following equation.

$$Ef_{fS} = \frac{f_a \text{ x } DP_S}{1000 \text{ x } Wf_i}$$

Where, Ef_{fS} = fan static efficiency (decimal)

For systems with duct velocities of less than 7.5 m/s the difference between static efficiency and total efficiency is usually negligible. However, the static efficiency, which is commonly used by fan manufacturers in their literature, is not always an accurate indication of fan performance. Two conditions are described that illustrate the need for caution in using the static efficiency.

- A fan that has substantially different velocities at the inlet and outlet would have a misleading static efficiency owing to the difference in velocity pressures, or kinetic energy, between the inlet and outlet.
- A fan which operates without a discharge duct could indicate a misleading static efficiency because of the inability to measure the conversion of velocity pressure to static pressure.

Fan Power

The *ideal fan power* is the power required to move a volume of air against the total system differential static pressure and can be calculated using the following equation.

$$Wf = \frac{f_a \text{ x } DP_T}{1000}$$

Where, Wf = ideal fan power (kW)

f_a = volume air flow rate (L/s)

DP_T = total differential pressure (Pa)

1000 = conversion of units

Calculations of ideal fan power can be used to estimate the effects of changes to a system when the manufacturer's performance data is not available. For example, the effect of changes in air flow and total system differential pressure on the power input to a fan motor can be estimated by multiplying the measured initial power input by the ratio of the ideal fan powers. The initial power input to an electric motor (Wm_{i1}) can be determined by measurement using the methods described in Appendix E. The revised power input to the motor can then be estimated by the following equation.

$$Wm_{i2} = Wm_{i1} \text{ x } \frac{Wf_2}{Wf_1}$$

Where, Wm_{i1}, Wm_{i2} = initial and revised motor power inputs (Appendix E) (kW)

Wf_1, Wf_2 = initial and revised ideal fan powers (kW)

The total power input required to operate a fan includes losses owing to fan, drive and motor efficiencies. The following extension of the fan power equation defines the power input to the motor in terms of the system conditions and the component efficiencies.

$$Wm_i = \frac{f_a \times DP_T}{Ef_{fT} \times Ef_d \times Ef_m \times 1000}$$

Where, Ef_{fT} = fan total efficiency (decimal)

Ef_d = drive efficiency (decimal)

Ef_m = motor efficiency (decimal) (Appendix E)

1000 = conversion of units

Drive loss is normally stated as a per cent of the motor shaft output power. When the drive loss is not known, a value from Table 1 can be used. The drive efficiency can be calculated from the drive loss by the following equation.

$$Ef_d = 1 - \frac{\text{Drive Loss}}{100}$$

Where, Drive loss is expressed in per cent

100 = conversion of per cent to decimal

The following similar extension of the fan power equation can be used when it is necessary to estimate the power input required to the fan shaft.

$$Wf_i = \frac{f_a \times DP_T}{Ef_{fT} \times 1000}$$

Where, Wf_i = power input to the fan shaft (kW)

By combining the previous equations, the net power input to the fan shaft by a particular drive system can be calculated.

$$Wf_i = Wm_i \times Ef_m \times Ef_d$$

Energy Analysis of a Fan

The following example uses the fundamentals and equations presented in the previous discussions.

A building air system had an axial fan with straight lengths of duct on both the inlet and the outlet (Figure 10). The duct connections were properly contoured for maximum fan output. The fan manufacturer's performance data was not available for the system or system components.

A pitot tube, manometer and other measuring devices were used to determine the following values.

Barometric pressure, B	101.325 kPa(absolute)
Air temperature, T	20°C (293.15 K)
Velocity pressure at inlet, P_{Vi}	78.4 Pa
Velocity pressure at outlet, P_{Vo}	150.9 Pa
Static pressure at inlet, P_{Si}	−274.4 Pa(gauge)
Static pressure at outlet, P_{So}	294 Pa(gauge)

Area of duct, (calculated) A_d — $0.302\ m^2$ on inlet

Electric motor current, (measured) I — 3.65 Amps averaged from 3 readings

Electric motor voltage, (measured) V — 593 volts averaged from 3 readings

Power factor, (measured) p.f. — 0.99 averaged over 3 phases

Motor efficiency, Ef_m — 0.82

$$\text{Average inlet air velocity, vel} = 0.764 \times \left(\frac{T \times P_{Vi}}{B}\right)^{0.50}$$

$$= 0.764 \times \left(\frac{293.15 \times 78.4}{101.325}\right)^{0.50}$$

$$= 11.5\ \text{m/s}$$

$$\text{Inlet volume air flow rate, } f_a = \text{vel} \times \text{Ad} \times 1000$$

$$= 11.5 \times 0.302 \times 1000$$

$$= 3473\ \text{L/s}$$

$$\text{Total differential pressure, } DP_T = P_{So} + P_{Vo} - P_{Si} - P_{Vi}$$

$$= 294 + 150.9 - (-274.4) - 78.4$$

$$= 641\ \text{Pa}$$

$$\text{Motor power input (Appendix E), } Wm_i = \frac{V \times I \times 1.73 \times \text{p.f.}}{1000}$$

$$= \frac{593 \times 3.65 \times 1.73 \times 0.99}{1000}$$

$$= 3.71\ \text{kW}$$

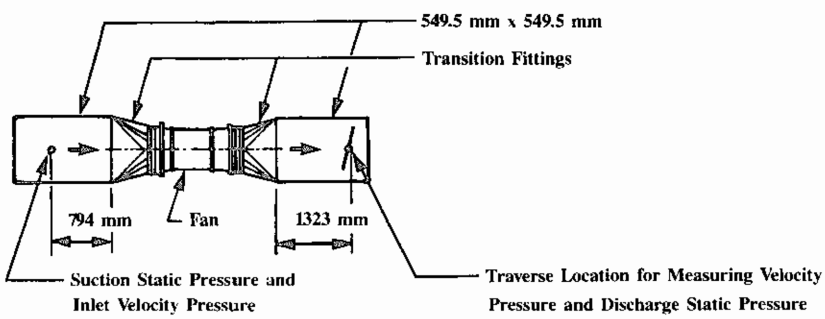


Diagram of Fan System

Figure 10

Motor shaft power output, (Appendix E) $Wm_o = Wm_i \times Ef_m$

$$= 3.71 \times 0.82$$

$$= 3.04 \text{ kW}$$

Drive loss (Table 1) = 9.4%

$$\text{Drive efficiency, } Ef_d = 1 - \frac{9.4}{100}$$

$$= 0.91$$

Power input to the fan shaft, $Wf_i = Wm_i \times Ef_m \times Ef_d$

$$= 3.71 \times 0.82 \times 0.91$$

$$= 2.77 \text{ kW}$$

$$\text{Fan total efficiency, } Ef_{fT} = \frac{f_a \times DP_T}{1000 \times Wf_i}$$

$$= \frac{3473 \times 0.641}{1000 \times 2.77}$$

$$= 0.80$$

The fan total efficiency of 0.80, or 80 per cent, compares favourably with the normal efficiency of 70 to 85 per cent for such a system, and indicates a good quality installation and well adjusted fan drive. Periodically repeating the measurements and calculations will identify any deterioration in performance and allow corrective action to be taken.

Multiple Fan Operation

In order to increase either system differential pressure or air flow rate, multiple fans may be operated in series or in parallel.

To achieve high system differential pressure, two fans may be installed in *series*. Ideally, the combined output is the sum of the differential pressures of the fans at the same air flow (Figure 11). Practically, there may be a significant loss of performance because of nonuniform flow into the second stage fan if it is installed too close to the first. Fan manufacturers or other qualified persons should be consulted regarding installation details.

When large air volumes are required, fans may be installed in *parallel*. Ideally, the combined output is the sum of the air volumes of the fans at the same differential pressure (Figure 12). However, if inlet or outlet conditions are poor, the total flow of the fans in parallel may be less than the ideal value. The figure also indicates an area of unstable operation. In the unstable region the system curve intersects the combined performance curve at two points. One fan will be underloaded and operating at poor efficiency, whereas the other fan will deliver most of the system requirements and use significantly more power than the underloaded fan. This risk can be eliminated if the fans are operated together with appropriate control procedures. The fan manufacturer should be consulted regarding installation and control details.

Fan Sheaves

For reduced air flow or reduced total differential pressure, energy can be saved by reducing the speed of a belt driven fan. The speed change is accomplished by changing the *pitch diameter* of the sheaves. This is normally achieved by changing the least expensive of the two sheaves. The final selection of the sheave to be changed will be determined by the space available, minimum sheave diameter limits of the belts for the required power transmission, and the type of motor sheave. If space permits, a large fan sheave may be less expensive to replace than a smaller, adjustable pitch motor sheave.

When the sheave on the fan shaft is to be changed, the following equation is used.

$$D_2 = D_1 \times \frac{n_1}{n_2}$$

Where, D_1, D_2 = existing and revised sheave pitch diameter (mm)

n_1, n_2 = existing and revised fan speed (rpm)

When the sheave on the motor is to be changed, the following equation is used.

$$D_2 = D_1 \times \frac{n_2}{n_1}$$

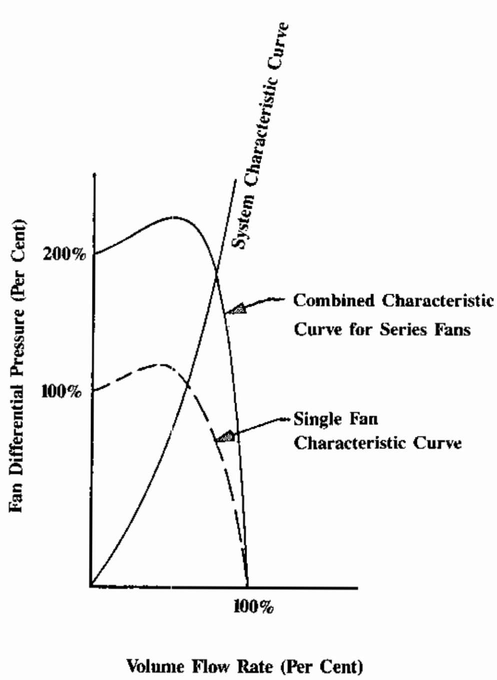
For example, a fan operating at 852 rpm has a 100 mm pitch diameter motor sheave. The motor sheave pitch diameter required to drive the fan at a speed of 750 rpm can be calculated in the following manner.

$$D_2 = D_1 \times \frac{n_2}{n_1}$$

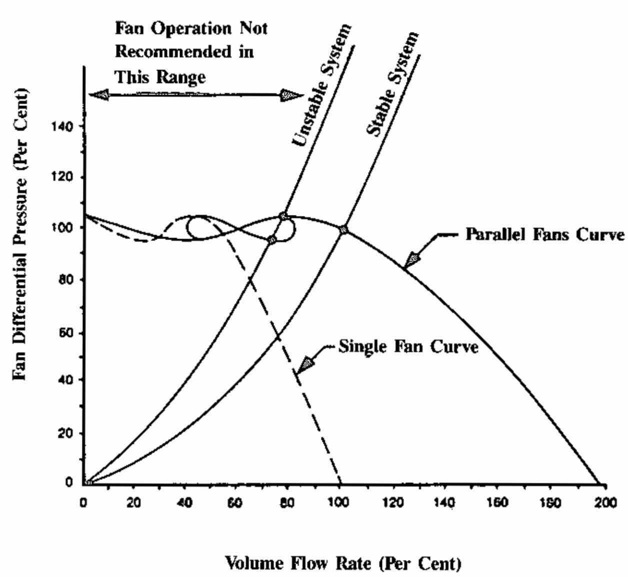
$$= 100 \times \frac{750}{852}$$

$$= 88 \text{ mm}$$

This reduction in fan speed will reduce the required power input to the fan.



Performance Curve for Two Fans in Series
Figure 11



Performance Curve for Parallel Fans
Figure 12

Fan Inlet and Outlet Conditions

Certain conditions change the aerodynamic characteristics of a fan so that its full flow potential may not be achieved.

- Nonuniform inlet flow.
- Swirl at the fan outlet.
- Improper outlet connections.

An accurate determination of the effect of these items on fan performance requires detailed calculations by persons fully familiar with fan inlet and outlet duct design criteria.

The following commentary describes good and poor inlet and outlet conditions for the purposes of recognition and correction. Methods for determining the effects these conditions have on fan performance are provided in AMCA Publication 201.

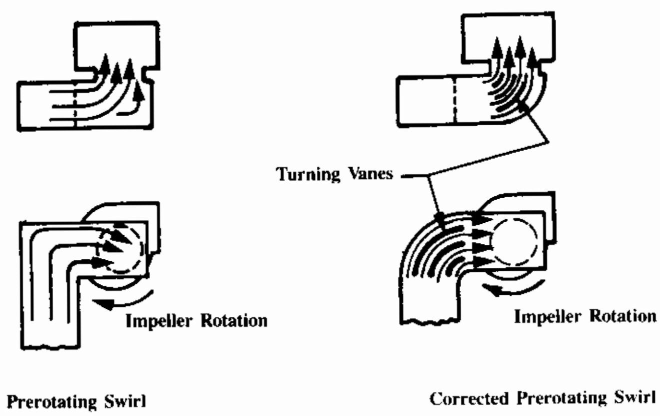
Inlet Ducts

Poor inlet conditions result in deteriorated fan performance characteristics. Inlet conditions usually have a greater effect on fan performance at low differential pressure conditions than at peak differential pressure. An inlet duct area of 92.5 to 112.5 per cent of the fan inlet area will retain maximum fan performance. Transition fittings to the fan inlet must have sides converging at a maximum of 15° and diverging at no greater than 7°.

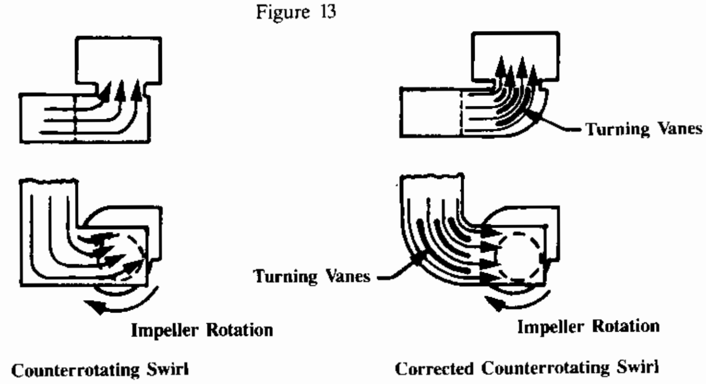
The optimum inlet condition is one that allows the air to enter the fan wheel axially and uniformly without spin.

The least desirable flow condition is one that causes spin or swirl to develop in the airstream at the fan inlet. Spin in the rotational direction of the impeller reduces the fan output and efficiency. Figure 13 illustrates the use of turning vanes to correct inlet prerotating swirl at a centrifugal fan. Capacity losses of 45 per cent have been observed in uncorrected installations.

Swirl in the reverse direction to impeller rotation may not reduce fan output but does increase fan power requirements. Figure 14 shows how counterrotating swirl can be reduced by turning vanes.



Prerotating Swirl
Figure 13

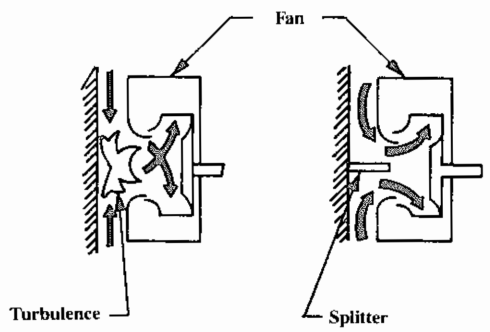


Counterrotating Swirl
Figure 14

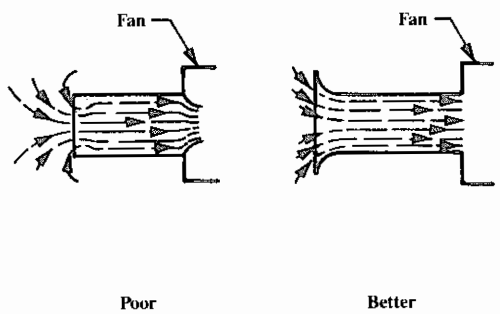
Turbulence, which reduces fan efficiency, can be caused by a butterfly damper installed too close to the fan inlet. If possible, install the damper at a distance at least equal to the outlet distance L as determined in the *Fan Performance Measurement* section.

Figure 15 shows a centrifugal fan installed with the inlet close to a wall without and with a splitter. Without the splitter, the airflows from opposite directions collide and cause turbulence at the fan inlet. With the splitter, the airflows cannot collide and the inlet resistance to the fan is reduced, thus improving fan performance.

Figure 16 illustrates some configurations of inlet fittings for fans without inlet ducts. Note that a bell mouthed inlet will improve the air flow pattern and thus improve fan performance.



Air Splitter
Figure 15



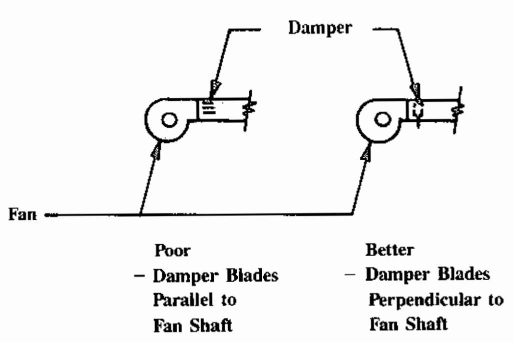
Inlet Conditions
Figure 16

Outlet Ducts

Similar to inlet ducts, the area of outlet ducts from fans must be between 87.5 and 107.5 per cent of outlet area of the fan to retain maximum fan performance. A transition from the fan to a discharge duct must have sides converging at no greater than 15° and diverging at no greater than 7°.

The length L in Figures 6 and 7 is the distance beyond the outlet area of the fan at which uniform flow of air will occur in the duct. This is the minimum desired length of straight duct prior to changes in direction of the duct system. Changes in direction of air flow occuring within this length of duct will have a negative effect on fan performance. Duct turning vanes are commonly used in this situation. However, the turning vanes may cause the nonuniform velocity profile to continue beyond the elbow into other fittings, which would aggravate the negative effects on the fan. Turns in discharge ducts should be made in the same direction as the natural flow of air from a fan to retain optimum performance.

When installed with the blades perpendicular to the fan shaft, discharge volume control dampers are more effective and have less adverse effect on the fan (Figure 17). The damper with blades perpendicular to the fan shaft will offer less interference to the redistribution of the velocity profile as shown in Figure 6.



Discharge Dampers
Figure 17

Summary

Numerous energy and cost saving opportunities exist for fan systems. Alert personnel, with an awareness of energy management techniques, can easily learn to recognize these opportunities and benefit from them.

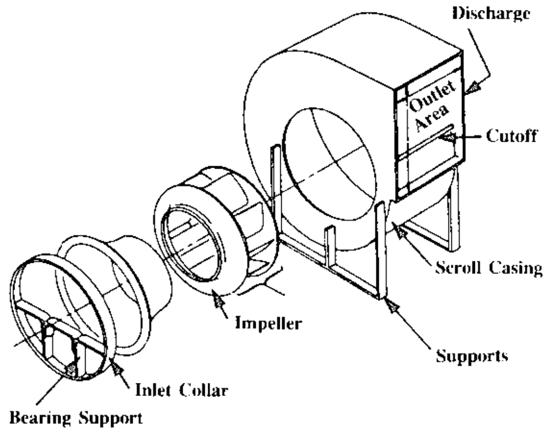
The effective utilization of the energy used by fans is affected by several factors.

- The configuration of the fans.
- Application of fans to the requirements of systems.
- The efficiencies of the fans.
- Operation of multiple fan systems.
- Inlet and outlet flow conditions.

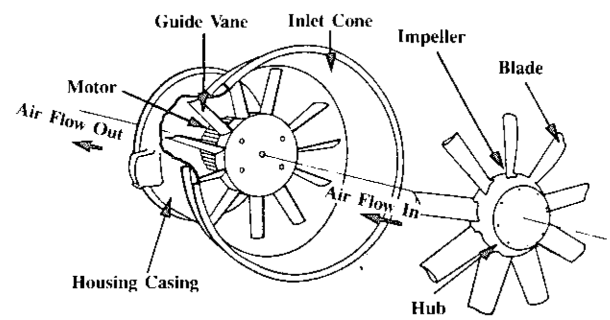
Fans may be analyzed in detail to estimate the energy input, energy transfer, losses, costs and potential savings. Worksheets 13-F1 through 13-F6 demonstrate methods of compiling data and performing calculations for the analysis of fans and their auxiliaries.

EQUIPMENT SYSTEMS

The two basic types of fans are centrifugal (Figure 18) and axial (Figure 19). The fans are further subdivided by the geometry of the blades on the impeller.



Common Terms Associated With Centrifugal Fans
Figure 18



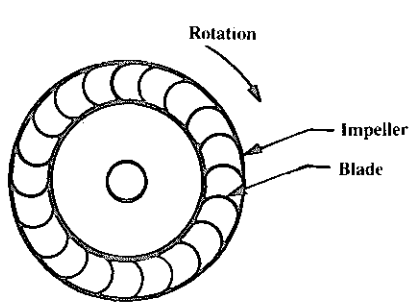
Common Terms Associated with Axial Fans
Figure 19

Centrifugal Fans

Centrifugal fans are used extensively in the heating, ventilating, and air conditioning (HVAC) industry. In general, a wide centrifugal fan impeller of small diameter produces a large volume of air at a low differential pressure. A narrow, large diameter impeller delivers a small volume of air at a high differential pressure. Table 2 lists the normal maximum flow, pressure and power values for centrifugal fans. Common uses for the various types of centrifugal fans are also listed.

Forward Curved Blade Fans

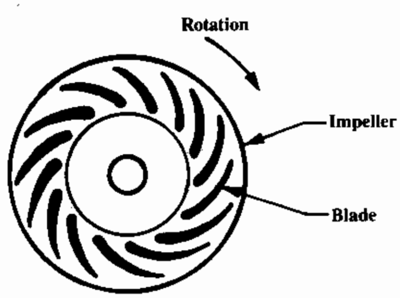
Forward curved blades (Figure 20) are mounted on what is often referred to as a squirrel cage rotor. Fans with these blades produce a high volume for their size, and are lightweight and economical. Because forward curved blades can collect deposits they are only used for handling clean air. Forward curved blade fans are used extensively in small air handling equipment (less than 610 mm diameter). Their efficiency is lower than that of airfoil fans. The scroll is often similar to other centrifugal fan designs, however, the clearance between the impeller and inlet is not as critical as for airfoil or backward inclined blade impellers.



Forward Curved Blade
Figure 20

Airfoil Blade Fans

Airfoil blade fans (Figure 21) are the most efficient of all centrifugal fans. These fans are most applicable to large systems where the energy saving owing to their high efficiency offsets their relatively high cost. The blades, which usually number ten to sixteen on the impeller, are contoured to react like an airplane wing. Airfoil fans produce extremely smooth flows and, for a given use, are the highest speed centrifugal fans. These fans have stable operation and low noise level over their full range. Airfoil blades are expensive to produce and repair, limiting their use to clean, nonabrasive gases. The construction of the scroll, the close internal clearances, and the precise alignment of the impeller and inlet bell are features which aid in attaining the high efficiencies of these fans.



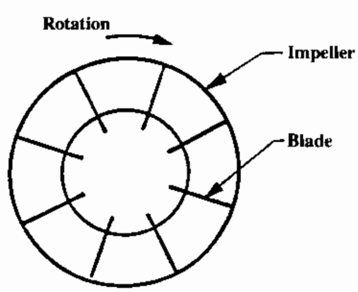
Airfoil Blade
Figure 21

Radial Blade Fans

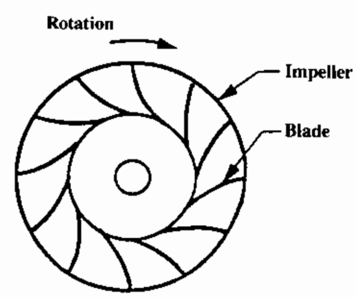
The *flat radial blade fan* is the simplest and least efficient type of centrifugal fan [Figure 22(a)]. This fan is used primarily for materials handling applications and air streams containing abrasive solids. Modified versions of the blade shape are used to improve efficiency. The radial impeller, which has from six to ten blades, has high mechanical strength and is easily repaired. Flat radial blades cause turbulent air flow resulting in particles in the gas stream deflecting away from the blades, giving maximum resistance to abrasion but low efficiency. Fan housing dimensions and clearances are not as important as for other centrifugal fans because of the low efficiency of the blades.

Modified radial blades [Figure 22(b)] are sometimes known as modified radial tip or backward inclined, forward curved. Throughout their operating range the efficiency is higher than that of other types of radial blades. Resistance to abrasion is good, making modified radial blades ideal for handling air streams with moderate dust loading.

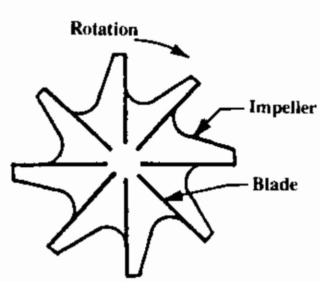
Open radial blades [Figure 22(c)] are used for extremely abrasive service. The blades are like flat paddles, often without sides. These fans are usually classified as industrial blowers. Blade wear can be high but blades can be easily replaced or lined. The scroll housing is constructed to resist wear rather than to enhance efficiency.



Flat Radial Blade
Figure 22(a)



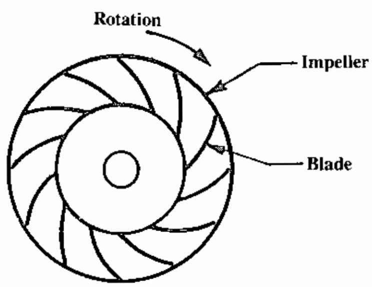
Modified Radial Blade
Figure 22(b)



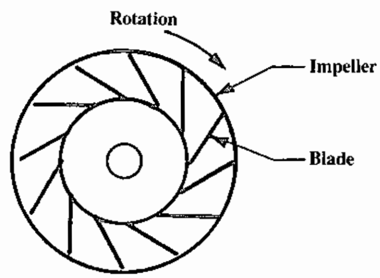
Open Radial Blade
Figure 22(c)

Backward Curved and Backward Inclined Blade Fans

Backward curved and backward inclined blades are slightly less efficient, cost less, and generate slightly more noise than the airfoil type. The number of blades is usually ten to sixteen and the fan scroll is the same as for airfoil fans. Backward curved blades [Figure 23(a)] are constant thickness airfoil blades and are preferred over straight backward inclined blades [Figure 23(b)] because they allow smoother air flow and are stronger. Backward curved blade fans will operate over the full range of air flow from wide open to shut off, whereas backward inclined blade fans are unstable under flow conditions below the design point. These fans are used in industrial applications in which fouling of the blades by airborne contaminants would minimize the efficiency advantage of airfoil blade fans.



Backward Curved Blade
Figure 23(a)



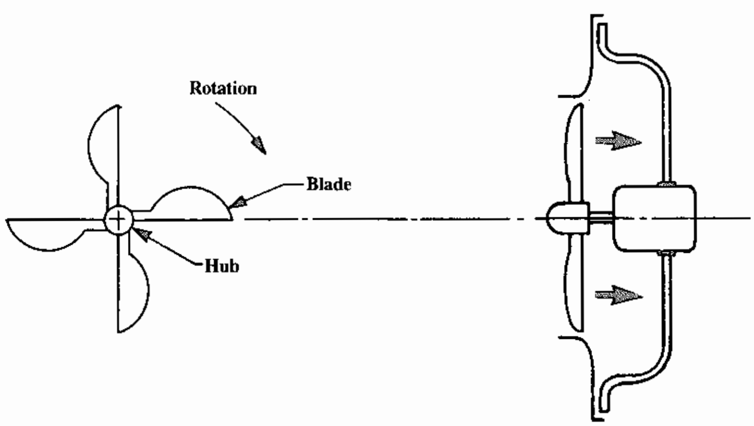
Backward Inclined Blade
Figure 23(b)

Axial Fans

The normal maximum flow, pressure and power values for axial fans are listed in Table 3. Examples of common uses for the types of axial fans discussed in this module are also provided. Efficiencies are not listed because of the dependence on inlet and outlet conditions.

Propeller Fans

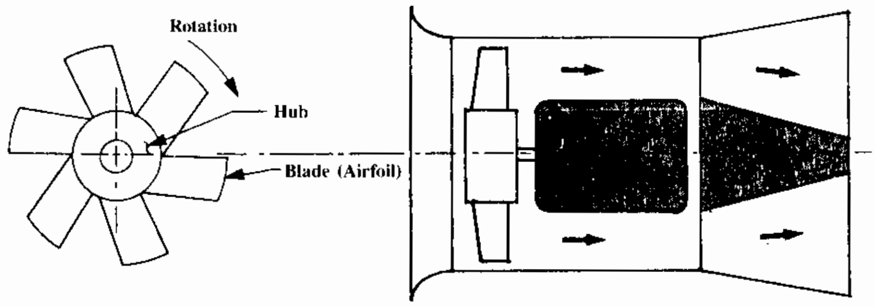
Propeller fans (Figure 24) are low efficiency, inexpensive air transfer fans, used where differential pressures are minimal. The fan impeller usually has two or more single thickness blades attached to a relatively small hub. The simplest housing is a flat plate with a circular hole, while other designs have orifice plate or venturi openings for increased performance. Optimum designs have a smooth inlet and close tolerances between the housing and blade tips. These fans are typically used to transfer air through a wall or to circulate air within a space, without attached ductwork.



Propeller Fan
Figure 24

Tubeaxial Fans

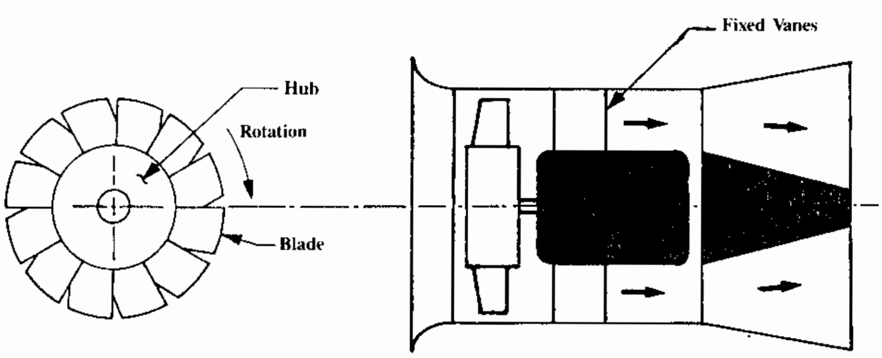
Tubeaxial fans (Figure 25) are more efficient than propeller fans and are capable of higher differential pressures. Tubeaxial fans have four to eight airfoil or curved blades. Normally, the hubs are less than half the size of the fan housing diameter. Housings are cylindrical tubes formed so that the running clearance between the blade tips and tube is minimal. This construction results in tubeaxial fans having higher efficiencies than propeller fans. Ease of installation, reasonable cost and low maintenance are the main advantages of tubeaxial fans. They are used in certain industrial applications such as drying ovens, paint spray booths and fume exhaust systems.



Tubeaxial Fan
Figure 25

Vaneaxial Fans

Vaneaxial fans (Figure 26) have blades which can efficiently produce air flows at medium to high differential pressure. The most efficient of these fans have airfoil blades. They have fixed or adjustable pitch and the hubs are usually larger than half the size of the fan housing diameter. Vaneaxial fan housings are cylindrical shells with minimal clearance from the blade tips. The fans are usually equipped with inlet and outlet cones that shroud the hub and drive, and with guide vanes upstream and downstream of the propeller. These additional components usually result in higher efficiencies than attainable with the tubeaxial fans. Vaneaxial fans are used in industrial applications similar to those of tubeaxial fans.

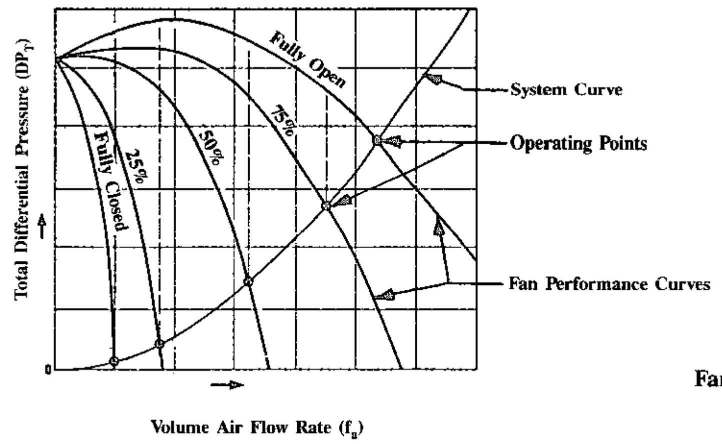


Vaneaxial Fan
Figure 26

Auxiliary Equipment for Fans

Various auxiliary equipment exists which can be used to increase fan performance.

Adjustable blades or *inlet vanes* can be fitted to the fan inlet to reduce the air flow. When properly installed they can reduce the power needed to drive the fan. Figure 27 illustrates the changes in a fan performance curve when the inlet vanes are throttled. There is a different fan performance curve at each vane setting, with different values for air flow, differential pressure and power requirement represented by the intersection points between the fan performance curves and the system curve. Variable inlet vanes can provide power reduction at reduced flow for fans with forward curved, radial or backward inclined blades. Because of the shape of the characteristic power curves, the greatest power reduction is usually obtained with backward curved blades and the least with forward curved blades. Inlet vanes have a wide range of regulation and may be manually or automatically controlled. They have the disadvantage of being an obstruction in the air stream that causes a loss of fan efficiency at high flows and the generation of noise at low flows. Each installation must be fully investigated to assure that inlet vanes will be satisfactory for air volume reduction.



Fan Performance with Inlet Vanes
Figure 27

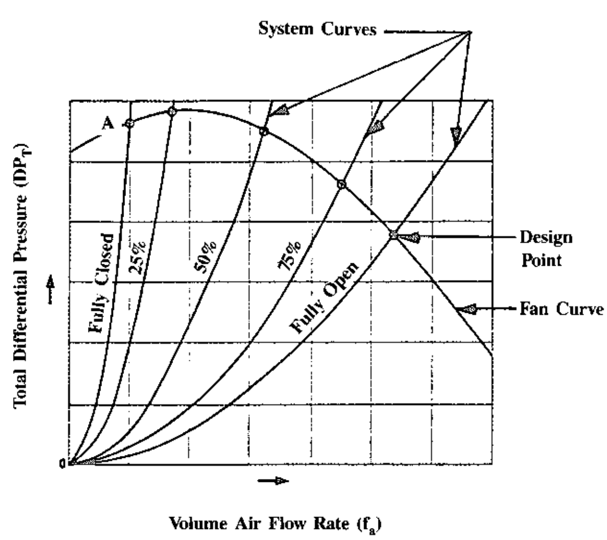
Inlet louvres are similar to inlet vanes but are more often used with modified radial blade fans on high temperature gas flows. The operating mechanism is outside the duct and safe from high temperatures.

Discharge dampers are used for reducing air flow from a fan by increasing the differential pressure. These dampers are not expensive to install but are an inefficient means of volume control. Figure 28 shows how the damper, with settings from wide open to fully closed, forces the fan to operate along the performance curve. Because of leakage through the dampers, air flow cannot be reduced below point A. Fan discharge dampers are generally used with fans having forward curved blades.

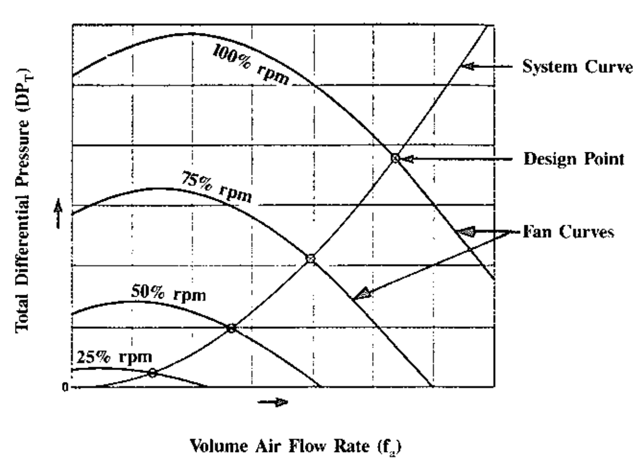
Variable-pitch blades for axial fans can be controlled to reduce flow and power by changing the angle that the blades deviate from the axis of the shaft.

Variable-speed electric motors and turbines are expensive but can be cost effective in some cases. Variable speed controls for electric motors can provide speed and power reduction to 35 per cent of maximum. Variable-speed drive couplings for installation between motors and fans are available in numerous designs. Generally they have wide speed range and an initial cost comparable to variable speed motor systems. However, lower motor efficiencies under low part load conditions result in lower overall efficiencies than for variable speed motors.

The performance of a typical variable speed fan is illustrated by Figure 29. Each speed has a different performance curve and power demand. Variable speed is the most efficient means of varying air flow from a fan. Using a forward curved blade fan, air flow can be reduced to 10 per cent of the peak design value with a corresponding reduction in power input to the fan. Efficiencies of the motor and the drive system must be considered in determining the actual reduction in power input to the fan motor.



Fan Performance with Discharge Dampers
Figure 28

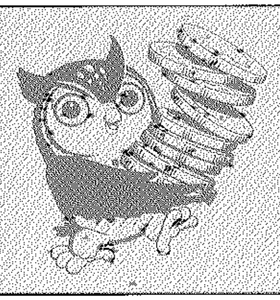


Fan Performance with Variable Speed
Figure 29

System Components

The performance of some common system and process components can have an important impact on fan performance. The performance characteristics of air filters, exhaust and intake louvres, and duct system dampers can vary significantly during their service life owing to obstruction by impurities in the air, deterioration of seals and wearing of the bearings in drive linkages. Such components should, therefore, be considered when evaluating a fan system.

ENERGY MANAGEMENT OPPORTUNITIES



Energy Management Opportunities is a term that represents the ways that energy can be used wisely to save money. A number of Energy Management Opportunities, subdivided into Housekeeping, Low Cost, and Retrofit categories are outlined in this section, with worked examples or text to illustrate the potential energy savings. In some examples the energy dollar savings are small, however, when many changes are combined, the resulting savings can be substantial. This is not a complete listing of the opportunities available for fans. However, it is intended to provide ideas for management, operating, and maintenance personnel to allow them to identify other opportunities that are applicable to a particular facility. Other modules in this series should be considered for Energy Management Opportunities applicable to other types of equipment and systems.

Housekeeping Opportunities

Implemented Housekeeping opportunities are Energy Management actions that are done on a regular basis and never less than once a year. The following are typical Energy Management Opportunities in this category.

1. Check and adjust belt drives on fans regularly to maintain proper sheave alignment and belt tension.
2. Lubricate fan components according to manufacturer's instructions.
3. Clean fan components regularly.
4. Correct excess noise and vibrations to ensure smooth and efficient operation.
5. Check fan performance against required performance.
6. Correct duct and component leaks to reduce energy costs.
7. Clean or replace filters periodically to keep fan differential pressure to a minimum.
8. Implement a program of inspection and planned maintenance to minimize component failures.

Housekeeping Worked Examples

1. Fan Drive Alignment and Belt Tension

Improper alignment of the fan drive sheaves can cause excessive power requirements and damage to belts.

Proper belt tension must be maintained because loose belts can cause slipping, squealing, low fan speed and rapid belt wear. Sheaves, bearings, shafts and motor will be hot indicating an energy loss. Belts should be tensioned to manufacturers recommendations, and must be readjusted for stretching after the first forty-eight hours of operation. Excessively tight belts will reduce fan and motor bearing life.

Properly aligned and tensioned belt drive systems will have mechanical losses as indicated by Table 1. Misaligned and loose belts will have greater loss of power than the table indicates. High speed fans will have losses close to the upper limit of the curve and low speed fans will have losses close to the lower limit of the curve.

2. Fan Lubrication

Lubrication of fan components, such as couplings, shaft bearings, adjustment linkage and adjustable supports, must be maintained with proper lubricants and at intervals recommended by the manufacturer. Following this procedure will ensure maximum component life and efficiency of the fan.

3. Fan Cleaning

Fans, particularly those moving dirty air, should be regularly cleaned to maintain their efficiency. Contamination on blades and housing interior causes higher static pressure loss in the fan, thereby reducing their efficiency.

4. Fan Noise and Vibration

Noise and vibration of a fan could be caused by one or more factors.

- Fan wheel out of balance.
- Bad bearings.
- Insufficient isolation.
- Misaligned shaft seals.
- Corrosion between shaft and bearing.

Fan impellers are factory balanced before installation. If balance weights are separated from the fan wheel, or if blades are chipped, missing or worn, the impellers could become unbalanced, causing the fan to operate at a lower efficiency.

Bad bearings on the fan shaft can cause noise, vibration, increased friction and wider clearances that can all result in reduced fan efficiency.

Fan isolation is achieved by installing spring or rubber isolators at the fan supports. If isolators are too light for the given duty, the fan will not be properly restrained, which may cause distortion of flexible duct connections and increased resistance to flow.

Changes in vibration may reveal a developing problem before the efficiency of the fan is seriously affected. By analyzing the characteristics of an abnormal vibration, it is often possible to identify the source of the problem and schedule corrective action.

5. Fan Performance

Fan performance information is available from manufacturers on all production fans. Savings can be realized by keeping the fan operating at the most efficient range as presented by the manufacturer's information.

6. Correct Leaks

Energy is lost when air leaks from loose connections, improperly sized damper shaft openings and unsealed expansion connections. These and similar conditions at fan suctions and discharges should be corrected.

7. Replace Loaded Air Filters

Loaded air filters is a common cause of poor performance in fan systems. Filter manufacturers provide recommendations for the pressure drop at which their filters are considered fully loaded for various air velocities at the inlet to the filters. Filters should be replaced before their pressure drop reaches the loaded value for the particular air velocity. Balancing a system with loaded filters will result in excessive air flow and high fan energy consumption when the filters are replaced.

8. Maintenance Programs

A maintenance program for fans should be tailored to the specific needs of the facility. This type of program could include the following actions.

- Daily; observe fan sounds, vibration, bearing temperatures, leakage, and the readings of installed gauges and meters.
- Monthly; check drive belt alignment, belt tension, and lubricate the fan bearings.
- Semiannually; inspect fan shaft seals, check inlet and outlet dampers, check inlet vanes, drain and refill oil lubricated bearings.
- Annually; check lubrication lines to assure proper movement of grease or oil, check fan auxiliaries, recalibrate all associated instrumentation, and carry out performance tests.
- Replace worn components.

Maintenance personnel must have the capability and experience to service, repair and troubleshoot the fans and associated systems. Training should be provided covering new equipment, changes to the facility and new procedures.

Low Cost Opportunities

Implemented low cost opportunities are Energy Management actions that are done once and for which the cost is not considered great. The following are typical Energy Management Opportunities in this category.

1. Reduce fan speed to suit optimum system air flow with balancing dampers in their maximum open positions for balanced air distribution.
2. Improve fan inlet and outlet duct connections to reduce entrance and discharge losses.
3. Shut down fans when not required.

Low Cost Worked Examples

1. Reduce Fan Speed to Match System Requirements

While doing air balance checks on an office building, all supply air system balancing dampers, including those on the longest duct runs, had to be partially closed to reduce the air flows to the design values. It was recognized that fan energy could be saved by reducing the supply fan speed to allow the balancing dampers on the longest duct runs to be fully opened.

The following measurements were obtained from the balancing contractor that performed the air balance checks.

Initial volume air flow rate, f_{a1}	19 000 L/s
Initial total fan differential static pressure, DP_{S1}	1.12 kPa
Pressure differential across the balancing damper on the longest duct run	0.17 kPa
Supply air fan motor data:	
Rated voltage, V_r	575 volts
Measured voltage, V	575 volts
Measured current draw, I	28.1 amperes
Phase	3 phase
Measured power factor, p.f.	0.96
Initial supply fan speed, n_1	730 rpm
Existing fan sheave pitch diameter, D_{f1}	559 mm
Existing motor sheave pitch diameter, D_{m1}	229 mm

The air system operates 10 hours per day during the 260 days per year that the building is occupied, or 2600 hours per year.

Using Worksheet 13-E1, the motor power input was determined to be 26.8 kW.

Using Worksheet 13-F1, it was determined that the speed reduction could be accomplished by installing a new motor sheave with 210 mm pitch diameter or by installing a fan sheave with 610 mm pitch diameter.

Worksheet 13-F1 was also used to determine the annual energy cost saving to be $559 per year.

From an examination of the fan drive and a review of available drive components, it was determined that installation of a new 610 mm fan sheave represented the lowest cost method of reducing the fan speed.

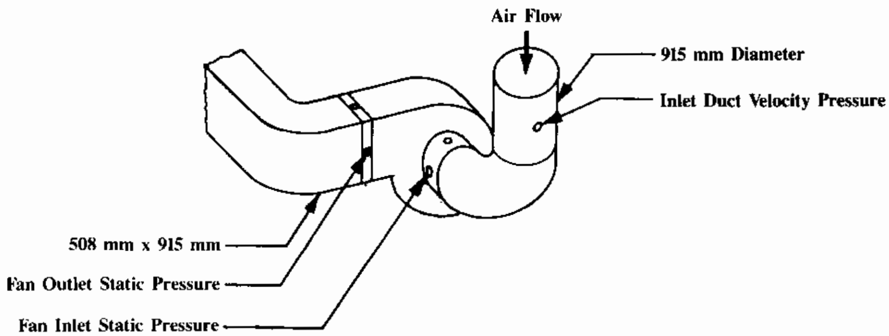
The estimated cost to purchase and install the new fan sheave is $500.

$$\text{Simple payback} = \frac{\$500}{\$559}$$

$$= 0.9 \text{ years (11 months)}$$

2. Improve Fan Inlet and Outlet Conditions

Inlet and outlet conditions can have a significant effect on fan performance. A belt driven centrifugal exhaust fan at a vehicle maintenance facility has duct connections arranged as shown in Figure 30. It was recognized that improvement of the duct connections would allow the fan to operate at a lower speed and save electrical energy. The fan operates 8760 hours per year, and the average electrical energy cost is $0.05 per kWh.



Duct Connections To A Fan
Figure 30

The following measurements were taken.

Volume Air flow rate, f_a	4001 L/s
Static pressure at inlet, P_{Si}	−0.543 kPa(gauge)
Static pressure at outlet, P_{So}	0.087 kPa(gauge)
Velocity pressure in inlet, P_{Vi}	0.045 kPa

Fan Motor Data:

Rated motor shaft power output	3.73 kW
Rated voltage, V_r	575 volts
Measured voltage, V	575 volts
Rated full load current, I_r	4.74 amperes
Measured current draw, I	4.70 amperes
Phases	3 phase
Rated full load power factor, $p.f._r$	0.86
Measured power factor, p.f.	0.86

Worksheet 13-E1 was used to calculate the initial motor shaft power output to be 3.70 kW.
Worksheet 13-F2 was used to calculate the total fan differential static pressure to be 0.585 kPa.
The fan manufacturer was contacted to obtain the following rated performance data.

Volume air flow rate, f_a	4001 L/s
Total fan differential static pressure	0.585 kPa
Revised power input to the fan shaft, Wf_{i2}	2.90 kW

Using Worksheet 13-F3, the length of straight duct at the fan discharge required to eliminate the negative effect of the discharge arrangement was determined to be 1.92 m. By reference to AMCA Publication 201 it was determined that a straight length of inlet duct equivalent to 2 duct diameters combined with a longer radius elbow would eliminate the negative effect of the inlet arrangement.

Worksheet 13-F4 was used to calculate the annual energy cost saving to be $258 per year.

The capital cost of modifying the duct connections and changing the fan sheave to reduce the fan speed was estimated to be $650.

$$\text{Simple payback} = \frac{650}{258}$$

$$= 2.5 \text{ years}$$

3. Shut Down Exhaust Fans During Building Unoccupied Periods

Savings in both energy and maintenance costs can be achieved by shutting down exhaust fans, either manually or by automatic control devices, when the building is unoccupied. Individual fans serving private washrooms can often be controlled by the light switch.

Retrofit Opportunities

Implemented retrofit opportunities are Energy Management actions that are done once and for which the cost is significant. Many of the opportunities in this category will require detailed analysis by specialists and cannot be examined in this module. Worked examples are provided for some of the listed Energy Management Opportunities, while in other cases there is only commentary. The following are typical Energy Management Opportunities in the Retrofit category.

1. Add a variable speed motor to allow fan to follow system requirements.
2. Replace outdated equipment with new units sized at optimum efficiency.
3. Replace oversized motors.
4. Break ventilation systems down into sub-systems, each with its own unique requirements, where presently a major system must be operated to satisfy the requirements of the most demanding sub-system.
5. Install a microprocessor-based energy management control system.

Retrofit Worked Examples

1. Install Variable Speed Motor

A hospital supply air system operates at full capacity 24 hours per day, or 8760 hours per year. An energy audit identified that the air flow could be reduced to 50 per cent of the design quantity between 8 p.m. and 7 a.m., or 11 hours per day, by shutting off dampers in the ducts to unoccupied areas. This would represent a reduced fan power requirement for 4015 hours per year. The average electrical energy cost is $0.05 per kWh.

The energy audit provided the following system data.

Initial volume air flow rate, f_{a1}	10 000 L/s
Revised volume air flow rate, f_{a2}	5 000 L/s
Initial total differential pressure, DP_{T1}	1.120 kPa
Estimated revised total differential pressure, DP_{T2}	0.700 kPa
Initial fan motor data:	
Rated motor shaft power output	18.65 kW
Rated voltage, V_r	575 volts
Measured voltage, V	575 volts
Rated full load current, I_r	24.1 amperes
Measured current, I	19.93 amperes
Phases	3 phase
Rated full load power factor, $p.f._r$	0.86
Measured power factor, p.f.	0.86

Worksheet 13-E1 was used to calculate the initial motor power input to be 17.05 kW.

Worksheet 13-F5 was used to calculate the reduced night time fan motor power input to be 5.33 kW and the annual energy cost saving to be $2,353 per year.

The estimated capital cost of installing a variable speed motor control and time clock is $12,000.

$$\text{Simple payback} = \frac{\$12,000}{\$2,353}$$

$$= 5.1 \text{ years}$$

Although this action by itself has a relatively long payback period, when it is considered together with the resulting heating and cooling energy savings, the overall payback period is much shorter.

2. Replace Fan With More Efficient Unit

During an energy audit an existing fan was examined to determine its operating efficiency. The following measurements were provided by the energy auditors.

Volume air flow rate, f_a	4880 L/s
Static pressure at inlet, P_{Si}	−0.231 kPa(gauge)
Static pressure at outlet, P_{So}	0.288 kPa(gauge)
Velocity pressure at inlet, P_{Vi}	0.032 kPa
Fan Motor Data:	
Rated voltage, V_r	575 volts
Measured voltage, V	593 volts
Rated full load current, I_r	5.9 amperes
Measured current draw, I	5.6 amperes
Phases	3 phase
Rated full load power factor, $p.f._r$	0.85
Measured power factor, p.f.	0.85

The fan operates 24 hours per day, or 8760 hours per year and the average electrical energy cost is $0.05 per kWh.

Worksheet 13-E1 was used to calculate the initial motor power input to be 4.88 kW and the motor shaft power output to be 4.49 kW. Worksheet 13-F2 was used to calculate the fan total differential static pressure to be 0.487 kPa.

Manufacturer's data was obtained on an airfoil blade fan of the required capacity with a listed shaft power input of 2.78 kW. Both the existing and the proposed fans require V-belt drives that have similar losses.

Worksheet 13-F4 was used to calculate the potential energy saving to be $823 per year.

Reusing the existing motor, the estimated capital cost of replacing the fan is $1,735.

$$\text{Simple payback} = \frac{\$1,735}{\$823}$$

$$= 2.1 \text{ years}$$

3. Replace Oversized Motors

During an investigation of low overall power factor in a factory, a 575 volt, 3 phase, 37.3 kW fan motor, having a full load current rating of 47.0 amperes was observed to be operating at 24.0 ampere current draw. It was recognized that electrical energy could be saved by installing a smaller motor which would operate at a higher load ratio and thus at an improved efficiency. The fan is required to operate 24 hours per day, or 8760 hours per year, and the average electrical energy cost is $0.05 per kWh.

The following additional data was recorded from measurements taken and from information on the motor nameplate.

Measured voltage, V	580 volts
Rated full load power factor. $p.f._r$	0.88
Measured power factor, p.f.	0.60

Worksheet 13-E1 was used to calculate the initial motor shaft power input to be 14.45 kW and the motor power output to be 11.99 kW. A replacement motor having a 14.92 kW full load output was selected. Worksheet 13-E2 was used to calculate the revised motor power input to be 13.03 kW and the annual energy cost saving to be $622 per year.

The estimated capital cost to install the new motor is $1,800.

$$\text{Simple payback} = \frac{\$1,800}{\$622}$$

$$= 2.9 \text{ years}$$

In addition, the existing 37.3 kW motor would become available for sale or use elsewhere in the factory.

4. Install a Booster Fan

An air system with a belt driven centrifugal fan had a total volume air flow rate of 12 000 L/s. One branch of the system, having a flow of 1500 L/s, required 0.88 kPa total differential pressure. Although the remainder of the system could operate at 0.65 kPa total differential pressure, the fan was operated at 0.88 kPa to provide the pressure required by the branch. The system operated 24 hours per day during 250 days per year, or 6000 hours per year.

It was recognized that a booster fan, rated for 1500 L/s at 0.23 kPa differential pressure, could be installed on the branch system. This would permit the main system fan to be slowed to the speed required for 0.65 kPa total differential pressure.

The following additional data was measured or compiled from the nameplates for the main system fan.

Motor data:	
Rated voltage, V_r	575 volts
Measured voltage, V	580 volts
Rated full load current, I_r	24.1 amperes
Measured current, I	19.2 amperes
Phases	3 phase
Rated full load power factor, $p.f._r$	0.85
Measured power factor, p.f.	0.84
Fan data:	
Initial fan speed, n_i	1200 rpm
Initial motor sheave pitch diameter, D_{ml}	209 mm
Initial fan sheave pitch diameter, D_{fl}	305 mm

The average air velocity in the branch duct was measured to be less than 5 m/s, so the velocity pressures at the inlet and outlet of the booster fan were considered to be negligible. A belt driven axial fan, having a rated capacity of 1500 L/s at a total differential static pressure of 0.23 kPa was selected from a manufacturer's catalogue data.

The data indicated the power input required at the fan shaft to be 0.56 kW. A drive motor was selected having a rated output of 0.75 kW and a rated efficiency of 75 per cent.

Worksheet 13-E1 was used to calculate the initial motor power input to the main system fan to be 16.2 kW.

Worksheet 13-F1 was used to calculate the replacement fan sheave diameter to be 355 mm and the annual energy cost saving at the main system fan to be $1,290 per year.

Worksheet 13-F6 was used to calculate the energy cost of operating the booster fan to be $273 per year.

The cost to install the new sheave on the system fan and the new booster fan was estimated to be $4,000.

$$\text{Simple Payback} = \frac{\$4,000}{\$1,290 - \$273}$$

$$= 3.9 \text{ years}$$

5. Install a Fan Management System

A computer based fan management system can accomplish energy cost saving over and above the savings from individual actions by monitoring and integrating various control functions. Most often in ventilation systems, the control of fans is integrated with a Building Energy Management System as addressed in Heating, Ventilating and Air Conditioning, Module 10, and in Automatic Controls, Module 16.

The analysis and selection of such equipment should be based on a professional review of the requirements for the particular facility.

Electric Motor Drive Performance
Worksheet 13-E1

Company: WORKED EXAMPLE #1 Date: 86/06/13

Location: FANS - LOW COST By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	N/A	amps
Measured voltage, V	575	volts
Measured current, I	28.1	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	N/A	kW
Rated full load power factor, $p.f._r$	N/A	(decimal)
Measured power factor, p.f.	0.96	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) ________ (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{575 \times 28.1 \times 1.73 \times 0.96}{1000}$$

= 26.8 kW

Motor shaft power output, Wm_o = $Wm_i \times Ef_m$

= ______ x ______

= N/A kW

Fan Speed Reduction — Sheave Replacement

Worksheet 13-F1
(Page 1 of 2)

Company: WORKED EXAMPLE #1 Date: 86/06/13

Location: FANS - LOW COST By: MBE

Data:

Initial volume air flow rate, f_{a1}	19000	L/s
Revised volume air flow rate, f_{a2}	N/A	L/s
Initial total differential pressure, DP_{T1}	1.12	kPa
Revised total differential pressure, DP_{T2}	0.94	kPa
Initial fan speed, n_1	730	rpm
Existing motor sheave pitch diameter, D_{m1}	229	mm
Existing fan sheave pitch diameter, D_{f1}	559	mm
Operation time, h	2600	h/yr
Unit electrical energy cost, Ce	$ 0.05	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	26.8	kW

Required fan speed:

Flow reduction:

Revised fan speed,

$$n_2 = n_1 \times \frac{f_{a2}}{f_{a1}}$$

$$= \underline{\quad} \times \underline{\quad}$$

$$= \underline{N/A} \text{ rpm}$$

Pressure reduction:

Revised fan speed,

$$n_2 = n_1 \times \left(\frac{DP_{T2}}{DP_{T1}}\right)^{0.5}$$

$$= 730 \times \left(\frac{0.94}{1.12}\right)^{0.5}$$

$$= \underline{669} \text{ rpm}$$

Fan Speed Reduction — Sheave Replacement

Worksheet 13-F1
(Page 2 of 2)

Company: WORKED EXAMPLE #1 Date: 86/06/13

Location: FANS - LOW COST By: MBE

Revised sheave size required:

New motor sheave:

New sheave pitch diameter, $D_{m2} = D_{m1} \times \frac{n_2}{n_1}$

$= 229 \times \frac{669}{730}$

$=$ 210 mm

New fan sheave:

New sheave pitch diameter, $D_{f2} = D_{f1} \times \frac{n_1}{n_2}$

$= 559 \times \frac{730}{669}$

$=$ 610 mm

Initial ideal fan power, Wf_1 $= \frac{f_{a1} \times DP_{T1}}{1000}$

$= \frac{19000 \times 1.12}{1000}$

$=$ 21.28 kW

Revised ideal fan power, Wf_2 $= \frac{f_{a2} \times DP_{T2}}{1000}$

$= \frac{19000 \times 0.94}{1000}$

$=$ 17.86 kW

Revised motor power input, Wm_{i2} $= Wm_{i1} \times \frac{Wf_2}{Wf_1}$

$= 26.8 \times \frac{17.86}{21.28}$

$=$ 22.5 kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (26.8 - 22.5) \times 2600 \times 0.05$

$= \$$ 559 /yr

Electric Motor Drive Performance

Worksheet 13-E1

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: FANS - LOW COST By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r 575 volts

Rated current, I_r 4.72 amps

Measured voltage, V 575 volts

Measured current, I 4.70 amps

Phase function, Y 1.73

(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate motor shaft power output 3.73 kW

Rated full load power factor, $p.f._r$ 0.86 (decimal)

Measured power factor, p.f. 0.86 (decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{4.70 \times 575 \times 0.86}{4.72 \times 575 \times 0.86}$$

$$= 0.996$$

Motor efficiency, Ef_m (Figure E-1) 0.92 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{575 \times 4.70 \times 1.73 \times 0.86}{1000}$$

$$= 4.02 \text{ kW}$$

$$\text{Motor shaft power output, } Wm_o = Wm_i \times Ef_m$$

$$= 4.02 \times 0.92$$

$$= 3.70 \text{ kW}$$

Fan Pressures
Worksheet 13-F2

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: FANS - LOW COST By: MBE

Static pressure at fan inlet, P_{Si} -0.543 kPa(gauge)

Static pressure at fan outlet, P_{So} 0.087 kPa(gauge)

Velocity Pressure at fan inlet, P_{Vi} 0.045 kPa

Velocity Pressure at fan outlet, P_{Vo} N/A kPa

Total fan static differential pressure, $DP_S = P_{So} - P_{Si} - P_{Vi}$

$= 0.087 - (-0.543) - 0.045$

$=$ 0.585 kPa

Total fan differential pressure, $DP_T = P_{So} + P_{Vo} - P_{Si} - P_{Vi}$

$= \quad + \quad - \quad -$

$=$ N/A kPa

Note: Readings taken at fan inlet and outlet

Optimum Length of Fan Outlet Duct

Worksheet 13-F3

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: FANS - LOW COST By: MBE

Duct Dimensions:

Rectangular: or Round:

Height, a 0.915 m Diameter N/A m

Width, b 0.508 m

Equivalent diameter, D $= (1.273 \times a \times b)^{0.50}$

$= (1.273 \times 0.915 \times 0.508)^{0.50}$

$= 0.769$ m

Volume air flow rate, f_a 4001 L/s

Equivalent duct area, $A_d = \dfrac{\pi \times D^2}{4}$

$= \dfrac{3.1416 \times (0.769)^2}{4}$

$= 0.464$ m^2

Average air velocity, vel. $= \dfrac{f_a}{A_d \times 1000}$

$\dfrac{4001}{0.464 \times 1000}$

$= 8.62$ m/s

Optimum Duct Length, L:

vel. less than 12.7 m/s or vel. greater than 12.7 m/s

$L = 2.5 \times D$

$= 2.5 \times 0.769$

$= 1.92$ m

$L = (2.5 \times D) + \dfrac{(vel. \quad - 12.7)}{5} \times D$

$= (2.5 \times \quad) + \dfrac{(\quad - 12.7)}{5} \times$

$=$ N/A m

Fan Replacement / Improved Fan Inlet and Outlet

Worksheet 13-F4

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: FANS - LOW COST By: MBE

Initial motor shaft power input, Wm_{i1} (Worksheet 13-E1) 4.02 kW

Initial motor shaft power output, Wm_{o1} (Worksheet 13-E1) 3.70 kW

Drive loss (Table 1) 8 %

Revised power input to the fan shaft, Wf_{i2} (Manufacturer) 2.90 kW

Operation time, h 8760 h/yr

Unit electrical energy cost, Ce $ 0.05 /kWh

Drive efficiency, $Ef_d = 1 - \frac{\text{Drive loss}}{100}$

$= 1 - \frac{8}{100}$

$= 0.92$

Initial power input to the fan shaft, $Wf_{i1} = Wm_{o1} \times Ef_d$

$= 3.70 \times 0.92$

$= 3.40$ kW

Revised motor power input, $Wm_{i2} = \frac{Wm_{i1} \times Wf_{i2}}{Wf_{i1}}$

$= \frac{4.02 \times 2.90}{3.4}$

$= 3.43$ kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (4.02 - 3.43) \times 8760 \times 0.05$

$= \$$ 258 /yr

Electric Motor Drive Performance
Worksheet 13-E1

Company: WORKED EXAMPLE #1 Date: 86/06/18

Location: FANS - RETROFIT By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	24.1	amps
Measured voltage, V	575	volts
Measured current, I	19.93	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	18.65	kW
Rated full load power factor, $p.f._r$	0.86	(decimal)
Measured power factor, p.f.	0.86	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

$$= \underline{\qquad\qquad}$$

Motor efficiency, Ef_m (Figure E-1) ______ (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{575 \times 19.93 \times 1.73 \times 0.86}{1000}$$

$$= 17.05 \text{ kW}$$

Motor shaft power output, Wm_o $= Wm_i \times Ef_m$

$$= \underline{\qquad \times \qquad}$$

$$= \underline{\qquad\qquad} \text{ kW}$$

Fan Speed Reduction — Variable Speed Motor

Worksheet 13-F5

Company: Worked Example #1   Date: 86/06/13

Location: Fans - Retrofit   By: MBE

Data

Initial volume air flow rate, f_{a1}	10 000	L/s
Revised volume air flow rate, f_{a2}	5000	L/s
Initial total differential pressure, DP_{T1}	1.120	kPa
Revised total differential pressure, DP_{T2}	0.700	kPa
Total operation time at reduced speed, h	4015	h/yr
Unit electrical energy cost, Ce	$ 0.05	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	17.05	kW

Initial ideal fan power, Wf_1

$$= \frac{f_{a1} \times DP_{T1}}{1000}$$

$$= \frac{10000 \times 1.120}{1000}$$

$$= 11.20 \text{ kW}$$

Revised ideal fan power, Wf_2

$$= \frac{f_{a2} \times DP_{T2}}{1000}$$

$$= \frac{5000 \times 0.700}{1000}$$

$$= 3.50 \text{ kW}$$

Revised motor power input, Wm_{i2}

$$= Wm_{i1} \times \frac{Wf_2}{Wf_1}$$

$$= 17.05 \times \frac{3.50}{11.20}$$

$$= 5.33 \text{ kW}$$

Annual energy cost saving

$$= (Wm_{i1} - Wm_{i2}) \times h \times Ce$$

$$= (17.05 - 5.33) \times 4015 \times 0.05$$

$$= \$\ 2,353 \text{ /yr}$$

Electric Motor Drive Performance

Worksheet 13-E1

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: FANS - RETROFIT By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	5.9	amps
Measured voltage, V	593	volts
Measured current, I	5.6	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	N/A	kW
Rated full load power factor, $p.f._r$	0.85	(decimal)
Measured power factor, p.f.	0.85	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{5.6 \times 593 \times 0.85}{5.9 \times 575 \times 0.85}$$

$$= 0.98$$

Motor efficiency, Ef_m (Figure E-1) 0.92 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{593 \times 5.6 \times 1.73 \times 0.85}{1000}$$

$$= 4.88 \text{ kW}$$

$$\text{Motor shaft power output, } Wm_o = Wm_i \times Ef_m$$

$$= 4.88 \times 0.92$$

$$= 4.49 \text{ kW}$$

Fan Pressures
Worksheet 13-F2

Company: WORKED EXAMPLE #2 Date: 86/06/15

Location: FANS - RETROFIT By: MBE

Static pressure at fan inlet, P_{Si} -0.231 kPa(gauge)

Static pressure at fan outlet, P_{So} 0.288 kPa(gauge)

Velocity Pressure at fan inlet, P_{Vi} 0.032 kPa

Velocity Pressure at fan outlet, P_{Vo} N/A kPa

Total fan static differential pressure, $DP_S = P_{So} - P_{Si} - P_{Vi}$

$= 0.288 - (-0.231) - 0.032$

$= 0.487$ kPa

Total fan differential pressure, $DP_T = P_{So} + P_{Vo} - P_{Si} - P_{Vi}$

$= \quad + \quad - \quad -$

= N/A kPa

Note: Readings taken at fan inlet and outlet

Fan Replacement / Improved Fan Inlet and Outlet

Worksheet 13-F4

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: FANS - RETROFIT By: MBE

Initial motor shaft power input, Wm_{i1} (Worksheet 13-E1) 4.88 kW

Initial motor shaft power output, Wm_{o1} (Worksheet 13-E1) 4.49 kW

Drive loss (Table 1) 7.5 %

Revised power input to the fan shaft, Wf_{i2} (Manufacturer) 2.78 kW

Operation time, h 8760 h/yr

Unit electrical energy cost, Ce $ 0.05 /kWh

$$\text{Drive efficiency, } Ef_d = 1 - \frac{\text{Drive loss}}{100}$$

$$= 1 - \frac{7.5}{100}$$

$$= 0.925$$

Initial power input to the fan shaft, $Wf_{i1} = Wm_{o1} \times Ef_d$

$$= 4.49 \times 0.925$$

$$= 4.15 \text{ kW}$$

$$\text{Revised motor power input, } Wm_{i2} = \frac{Wm_{i1} \times Wf_{i2}}{Wf_{i1}}$$

$$= \frac{4.48 \times 2.78}{4.15}$$

$$= 3.00 \text{ kW}$$

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$$= (4.88 - 3.00) \times 8760 \times 0.05$$

$$= \$ 823 \text{ /yr}$$

Electric Motor Drive Performance

Worksheet 13-E1

Company: WORKED EXAMPLE #3 Date: 86/06/13

Location: FANS - RETROFIT By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	47.0	amps
Measured voltage, V	580	volts
Measured current, I	24.0	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	N/A	kW
Rated full load power factor, $p.f._r$	0.88	(decimal)
Measured power factor, p.f.	0.60	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{24.0 \times 580 \times 0.60}{47.0 \times 575 \times 0.88}$$

$$= 0.35$$

Motor efficiency, Ef_m (Figure E-1) 0.83 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{580 \times 24.0 \times 1.73 \times 0.60}{1000}$$

$$= 14.45 \text{ kW}$$

Motor shaft power output, Wm_o = $Wm_i \times Ef_m$

$$= 14.45 \times 0.83$$

$$= 11.99 \text{ kW}$$

Motor Replacement
Worksheet 13-E2

Company: WORKED EXAMPLE #3 Date: 86/06/13

Location: FANS - RETROFIT By: MBE

Initial motor power input, Wm_{i1} (Worksheet 13-E1)	14.45	kW	(1)
Required motor shaft power output, Wm_o (Worksheet 13-E1)	11.99	kW	(2)
Replacement motor rated shaft power output	14.92	kW	(3)
Operation time, h	8760	h/yr	(4)
Unit electrical energy cost, Ce	$ 0.05	/kWh	(5)

Load ratio (Worksheet E-1) $= \frac{(2)}{(3)}$

$= \frac{11.99}{14.92}$

$= 0.80$

Replacement motor efficiency, Ef_m 0.92 (decimal)

Replacement motor power input, $Wm_{i2} = \frac{(2)}{Ef_m}$

$= \frac{11.99}{0.92}$

$= 13.03$ kW (6)

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (14.45 - 13.03) \times 8760 \times 0.05$

$= \$622$ /yr

Electric Motor Drive Performance
Worksheet 13-E1

Company: WORKED EXAMPLE #4 Date: 86/06/13

Location: FANS - RETROFIT By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r 575 volts

Rated current, I_r 24.1 amps

Measured voltage, V 580 volts

Measured current, I 19.2 amps

Phase function, Y 1.73
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate motor shaft power output 18.65 kW

Rated full load power factor, $p.f._r$ 0.85 (decimal)

Measured power factor, p.f. 0.84 (decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{19.2 \times 580 \times 0.84}{24.1 \times 575 \times 0.85}$$

$$= 0.79$$

Motor efficiency, Ef_m (Figure E-1) 0.92 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{580 \times 19.2 \times 1.73 \times 0.84}{1000}$$

$$= 16.2 \text{ kW}$$

Motor shaft power output, Wm_o = $Wm_i \times Ef_m$

= ______ x ______

= N/A kW

Fan Speed Reduction — Sheave Replacement

Worksheet 13-F1
(Page 1 of 2)

Company: Worked Example #4 Date: 86/06/13

Location: Fans - Retrofit By: MBE

Data:

Initial volume air flow rate, f_{a1}	12000	L/s
Revised volume air flow rate, f_{a2}	12000	L/s
Initial total differential pressure, DP_{T1}	0.88	kPa
Revised total differential pressure, DP_{T2}	0.65	kPa
Initial fan speed, n_1	1200	rpm
Existing motor sheave pitch diameter, D_{m1}	209	mm
Existing fan sheave pitch diameter, D_{f1}	305	mm
Operation time, h	6000	h/yr
Unit electrical energy cost, Ce	$ 0.05	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	16.2	kW

Required fan speed:

Flow reduction:

Revised fan speed,

$$n_2 = n_1 \times \frac{f_{a2}}{f_{a1}}$$

$$= \_\_\_ \times \_\_\_$$

$$= \text{N/A} \text{ rpm}$$

Pressure reduction:

Revised fan speed,

$$n_2 = n_1 \times \left(\frac{DP_{T2}}{DP_{T1}}\right)^{0.5}$$

$$= 1200 \times \left(\frac{0.65}{0.88}\right)^{0.5}$$

$$= 1031 \text{ rpm}$$

Fan Speed Reduction — Sheave Replacement

Worksheet 13-F1

(Page 2 of 2)

Company: WORKED EXAMPLE #4 Date: 86/06/13

Location: FANS - RETROFIT By: MBE

Revised sheave size required:

New motor sheave:

New sheave pitch diameter, $D_{m2} = D_{m1} \times \frac{n_2}{n_1}$

$= \quad \times$ ______

$=$ N/A mm

New fan sheave:

New sheave pitch diameter, $D_{f2} = D_{f1} \times \frac{n_1}{n_2}$

$= 305 \times \frac{1200}{1031}$

$= 355$ mm

Initial ideal fan power, $Wf_1 = \frac{f_{a1} \times DP_{T1}}{1000}$

$= \frac{12000 \times 0.88}{1000}$

$= 10.6$ kW

Revised ideal fan power, $Wf_2 = \frac{f_{a2} \times DP_{T2}}{1000}$

$= \frac{12000 \times 0.65}{1000}$

$= 7.8$ kW

Revised motor power input, $Wm_{i2} = Wm_{i1} \times \frac{Wf_2}{Wf_1}$

$= 16.2 \times \frac{7.8}{10.6}$

$= 11.9$ kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (16.2 - 11.9) \times 6000 \times 0.05$

$= \$\ 1,290$ /yr

Fan Energy Consumption
Worksheet 13-F6

Company: Worked Example #4 | Date: 86/06/13

Location: Fans - Retrofit | By: MBE

Data:

Power input to fan shaft, Wf_i (Manufacturer's rating table)	0.56	kW
Rated motor shaft power output, Wm_o	0.75	kW
Rated motor efficiency, Ef_m	0.75	(decimal)
Drive loss (use 0 for direct drive, Table 1 for belt drive)	18	%
Operation time, h	6000	h/yr
Unit electrical energy cost, Ce	$ 0.05	/kWh

Drive efficiency, $Ef_d = 1 - \dfrac{\text{Drive loss}}{100}$

$= 1 - \dfrac{18}{100}$

$=$ 0.82

Motor power input $Wm_i = \dfrac{Wf_i}{Ef_m \times Ef_d}$

$= \dfrac{0.56}{0.75 \times 0.82}$

$=$ 0.91 kW

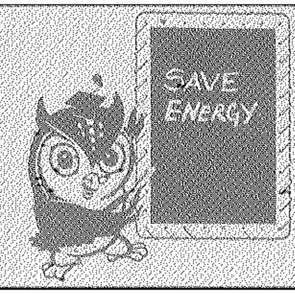
Annual energy cost $= Wm_i \times Ce \times h$

$= 0.91 \times 0.05 \times 6000$

$=$ $ 273 /yr

SECTION 2 PUMPS

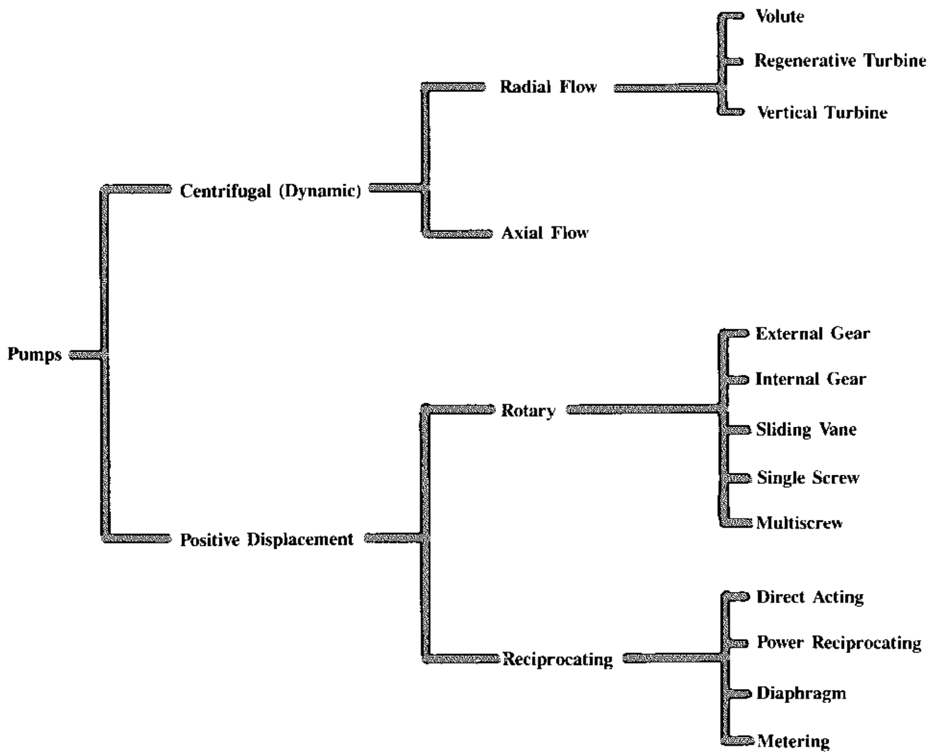
FUNDAMENTALS



Pumps provide the motive force to move liquids to a different elevation and/or to move them against the resistance of a piping or process system.

Pump Types

Pump types can be grouped under two main categories; *centrifugal (dynamic)* and *positive displacement*. Figure 31 provides a classification of the various types of pumps found in Industrial, Commercial and Institutional facilities.

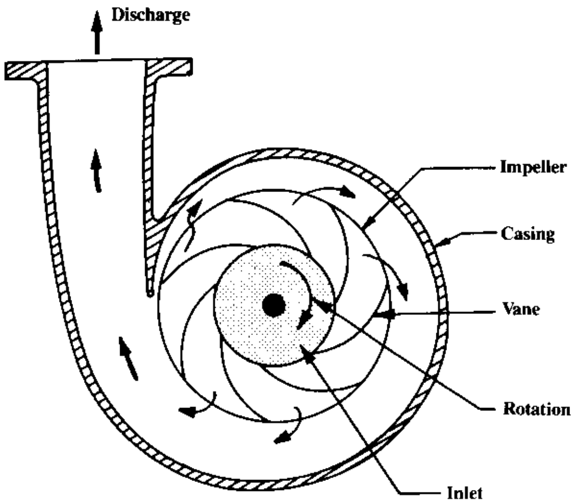


Classification of Pump Types
Figure 31

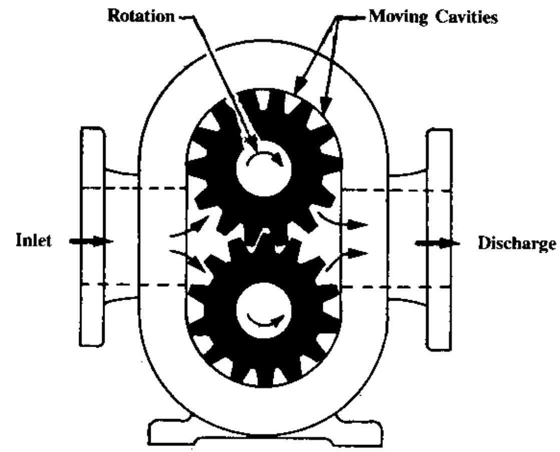
Centrifugal pumps are dynamic devices that impart kinetic energy, or energy of motion, to a liquid by the spinning motion of an impeller. The simplified *radial flow* centrifugal pump (Figure 32) has rotating vanes that move the liquid outward from the centre of the impeller into the scrolled casing. Some of the kinetic energy is converted to pressure, forcing the liquid out the discharge. An *axial flow* centrifugal pump imparts energy to the liquid by a lifting action of propeller shaped vanes resulting in an axial discharge. In a *mixed flow* pump the energy of the liquid is increased by a combination of radial forces and lifting action.

Positive displacement pumps operate by trapping the liquid in various forms of pump cavities and displacing it to the pump discharge. They provide an essentially constant volumetric rate of flow for a particular pump speed independent of pressure difference and liquid characteristics. Figure 33 illustrates a typical gear type positive displacement pump.

Jet, eductor and other special action types of dynamic pumps require application engineering by qualified experts, and are not addressed in this module.



Radial Flow Centrifugal Pump
Figure 32



Gear Type Positive Displacement Pump
Figure 33

Pump Operation

Pump operating characteristics will be discussed for water at standard conditions of 20°C at sea level. Although the density, and thus the specific gravity, of water decreases with an increase in temperature, calculations using standard conditions are considered sufficiently accurate for estimating purposes. Analysis by experts should be obtained where accurate calculations are required for other fluids and temperatures.

Liquid System Measurements

The motive force required to move a particular flow of liquid from one location in a piping system to another, or to move liquid within a recirculating system, is called the *total system head.* Total system head represents a differential pressure and may be expressed in pascals or meters of water. Differential pressure expressed in kilopascals can be converted to head in metres of water by the following equation.

$$Hd = \frac{DP}{9.81}$$

Where, Hd = total system head (m)

DP = differential pressure (kPa)

9.81 = kPa per metre of head

The total system head includes differential pressures owing to three factors (Figure 34).

- Friction and inertia losses caused by fluid flow in the pipes and fittings, or *friction head.*
- The difference in elevation between the liquid surface at the source location and the liquid surface at the destination location.
- The surface pressures at the source and destination locations.

The *friction head* of a particular system is proportional to the square of the flow. The algebraic sum of the pressures owing to differences in elevation and surface pressure is independent of flow and is called the *static head.*

If the friction head of a particular system is known for one flow, the friction head for another flow can be calculated by the following equation. If the static head is also known, this equation can be used to calculate and plot the system head curve for a range of flows as illustrated in Figure 35.

$$Hd_{f2} = Hd_{f1} \times \left(\frac{f_{w2}}{f_{w1}}\right)^2$$

Where, Hd_{f1}, Hd_{f2} = initial and revised friction heads (m)

f_{w1}, f_{w2} = initial and revised liquid flow rates (m^3/h)

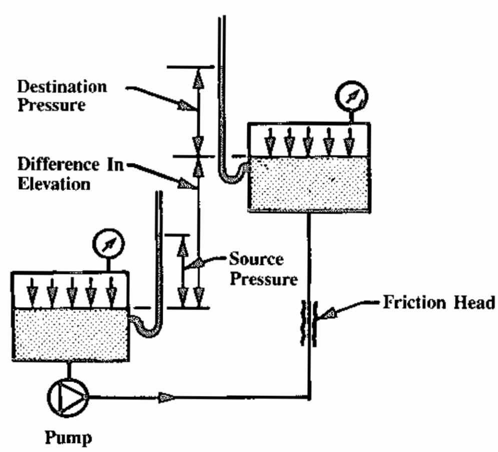
Flow in a pipe can be measured by installing a precalibrated flow restriction device, such as an orifice plate, having a known friction head for a particular flow rate. By measuring the head across the restriction and adapting the rule that friction head is proportional to the square of the flow, the flow corresponding to the measured head can be calculated.

$$f_{w2} = f_{w1} \times \left(\frac{Hd_{f2}}{Hd_{f1}}\right)^{0.5}$$

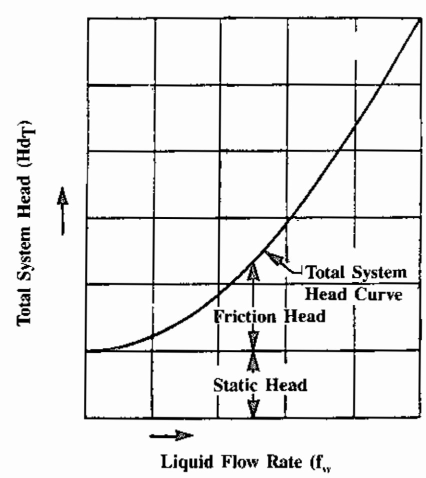
Where, f_{w1}, f_{w2} = initial and revised liquid flow rates (m^3/h)

Hd_{f1}, Hd_{f2} = initial and revised friction heads (m)

Liquid flow measurement and flow measurement devices are discussed further in Measuring, Metering and Monitoring, Module 15.



System Heads
Figure 34



Typical System Head Curve
Figure 35

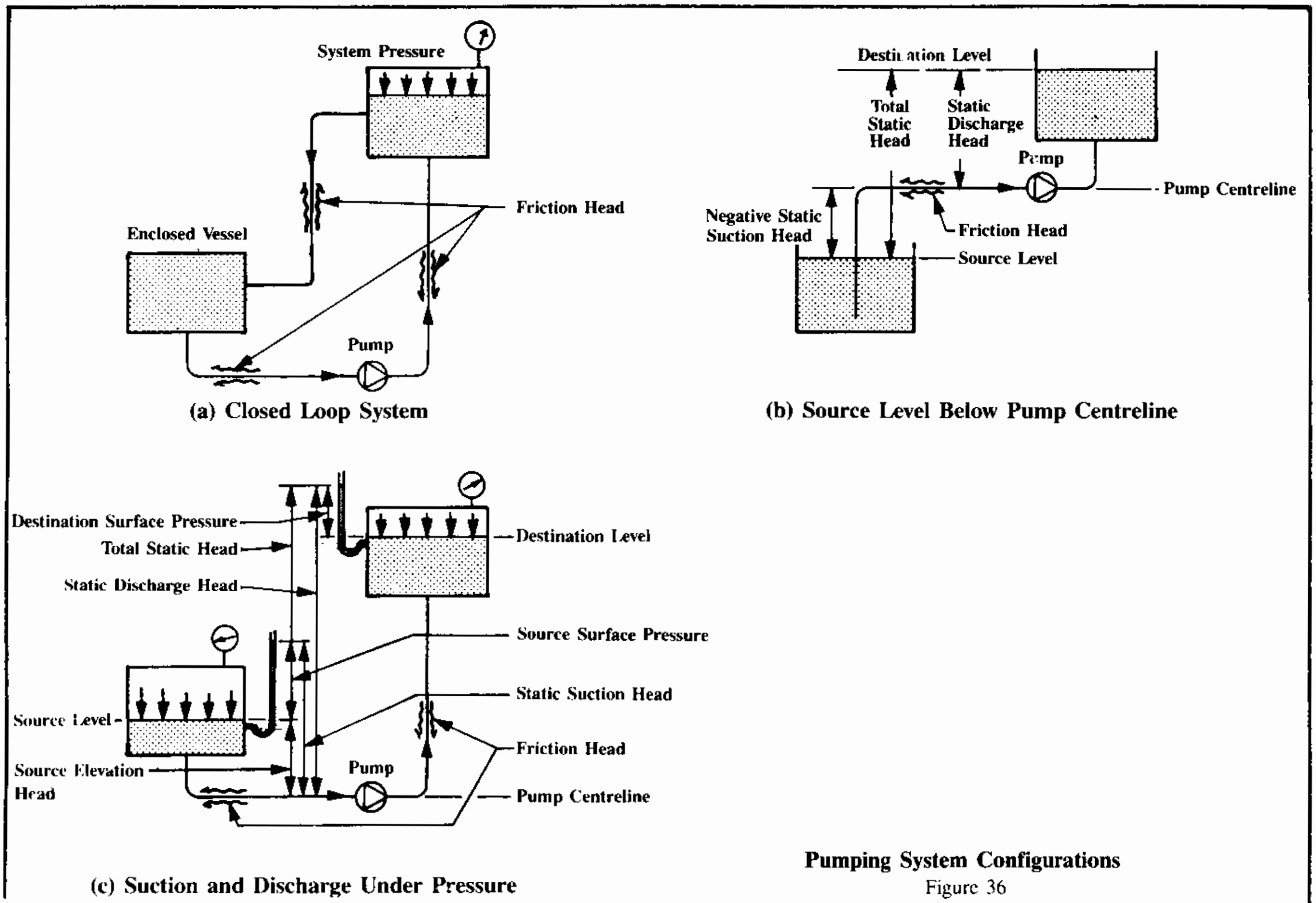
Pump Performance Measurement

The performance of a pump can be determined by calculations based on measurements of system head and flow.

The total pump head, sometimes called *total dynamic head* or TDH, is equal to the total system head for a particular flow and is composed of several components.

- Static suction head
- Static discharge head
- Friction head

The total pump head is the net effect of these components on the pressure difference between the pump inlet and discharge. Figure 36 shows three pumping systems that illustrate the various components of total pump head. The net total effect of the static suction and static discharge heads represents the *total static head*. In a closed loop system (Figure 36.a) the system friction loss represents the total pump head.



Pumping System Configurations
Figure 36

For the purpose of analyzing pump head, only the *total static head* and *system friction head* need to be known.

$$Hd_T = Hd_S + Hd_f$$

Where, Hd_T = total pump head (m)

Hd_S = total static head (m)

Hd_f = total system friction head (m)

Pump Suction Conditions

The absolute pressure on the liquid at the suction inlet to a pump affects the range of flow at which the pump can safely operate. For a particular temperature every liquid has a *vapor pressure* at which the liquid begins to form vapor. Table 4 lists the vapor pressure of water at various temperatures. If the liquid entering a pump impeller has an absolute pressure at or below the vapor pressure of the liquid, vapor bubbles or pockets will form in the impeller passages. The vapor bubbles move along the impeller vanes to an area of higher pressure, where they collapse. This formation and collapse, known as *cavitation*, interferes with pump performance, is noisy and can damage impeller surfaces.

The formation of the vapor bubbles can be prevented by keeping the pressure at the impeller inlet greater than the liquid vapor pressure. Each pump has specific suction inlet characteristics that, to prevent cavitation, requires the absolute pressure at the pump inlet to be somewhat greater than the liquid vapor pressure. This excess pressure, which increases with the flow, is called the *net positive suction head*(NPSH) required.

Centrifugal Pump Affinity Laws

The *affinity laws* relate the performance variables for a particular centrifugal pump. The performance variables used in this module are flow, pump speed, impeller diameter, total pump head and power.

For a *constant impeller diameter and a variation in pump speed*, the following relationships apply.

- The flow varies in proportion to the pump speed.
- The total pump head varies in proportion to the square of the pump speed.
- The power input to the pump shaft varies in proportion to the cube of the pump speed.

For a *constant pump speed and a variation in impeller diameter*, the following relationships apply.

- The flow varies in proportion to the impeller diameter.
- The total pump head varies in proportion to the square of the impeller diameter.
- The power required varies in proportion to the cube of the impeller diameter.

The affinity laws can be combined to produce the following equations.

$$f_{w2} = f_{w1} \times \frac{n_2}{n_1} \times \frac{D_2}{D_1}$$

$$Hd_{T2} = Hd_{T1} \times \left(\frac{n_2}{n_1}\right)^2 \times \left(\frac{D_2}{D_1}\right)^2$$

$$Wp_{i2} = Wp_{i1} \times \left(\frac{n_2}{n_1}\right)^3 \times \left(\frac{D_2}{D_1}\right)^3$$

Where, f_{w1}, f_{w2} = initial and revised liquid flow rates (m^3/h)

n_1, n_2 = initial and revised pump speeds (rpm)

Hd_{T1}, Hd_{T2} = initial and revised total pump heads (m)

Wp_{i1}, Wp_{i2} = initial and revised power inputs to the pump shaft

D_1, D_2 = initial and revised impeller diameters (m)

These laws can be used to estimate revised pump capacity characteristics from a known performance rating for the initial conditions. For system curves based predominantly on friction head, these laws can be used to estimate revised impeller diameters for revised flows and pump heads. If the system curve includes a significant static head component, the affinity laws should *not* be used for analysis except for small changes in flow and head (less than 10 per cent). For such applications the relationship between impeller diameter and pump head will produce a conservative estimate of the revised impeller diameter required for a reduced head. For system curves having a large static head component, manufacturers' performance data *must* be used to select revised impeller diameters or pump speeds. The affinity laws do *not* apply to positive displacement pumps.

The following is an example of the application of the affinity laws.

A constant speed centrifugal pump on a heating system operates at a measured flow of 114 m^3/h at 13.2 m total pump head. The pump has a 208 mm diameter impeller. As the result of energy conservation measures the flow is to be reduced to 94 m^3/h at 9.0 m total pump head by trimming the impeller.

A new impeller diameter may be conservatively estimated using the liquid flow affinity law equations for revised pump head.

$$\text{Revised diameter, } D_2 = D_1 \times \left(\frac{Hd_{T2}}{Hd_{T1}}\right)^{0.5}$$

$$= 208 \times \left(\frac{9.0}{13.2}\right)^{0.5}$$

$$= 172 \text{ mm}$$

This reduction in impeller size will reduce the net shaft power required.

Positive Displacement Pump Operation

A *positive displacement pump* provides a constant flow, independent of total pump head, for a particular pump speed. The following relationships between flow, pump speed, total pump head and power apply to positive displacement pumps.

- The flow varies in proportion to the pump speed.
- The power required for a constant flow varies in proportion to the total pump head.
- The power required for a constant total pump head varies in proportion to the flow.

Because of the proportional relationship between power required and total pump head, some form of pressure relief is required at the pump discharge to limit the head to the maximum rating of the particular pump and drive system.

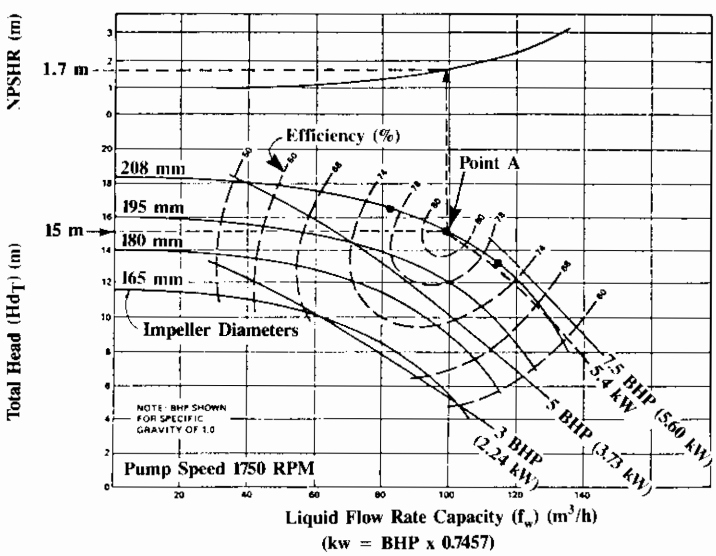
Pump Performance

The manufacturer's performance curves or tables, when available, are the most reliable source of data for pump operating characteristics. When such data is not available, the power requirement of a particular pump can be estimated using measured data, the pump affinity laws and the fan power equations that are discussed in this section.

Centrifugal Pump Performance Curves

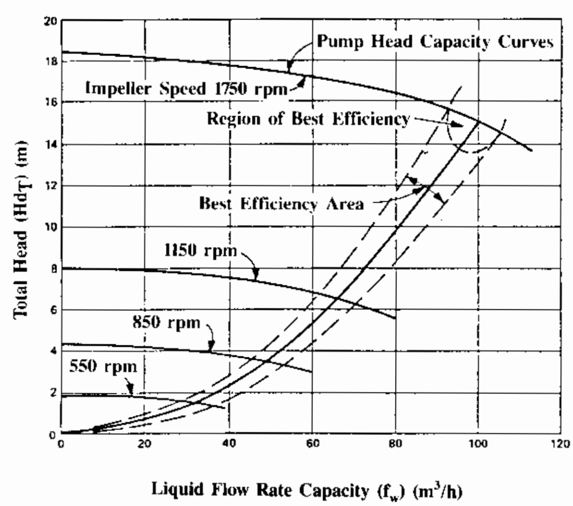
The performance of a centrifugal pump is represented by curves based on flow capacity and total pump head. Figure 37 is a typical manufacturer's centrifugal *pump performance curve* showing the performance for various impeller diameters, constant power inputs, pump efficiencies and net positive suction heads required (NPSHR). For an impeller diameter of 208 mm and a flow capacity of 100 m^3/h the performance curve indicates (point A) that the pump is capable of a total head of 15 m and the pump shaft power input is approximately 5.4 kW. Moving vertically from point A to the NPSHR curve, the net positive suction head required to avoid cavitation is 1.7 m.

When a pump is modified to reduce its energy requirement, an attempt should be made to achieve a high pump efficiency. Each pump speed will have a corresponding maximum efficiency point as indicated by point A in Figure 37. Figure 38 shows a typical relationship of the maximum efficiency point with impeller speed for a particular pump. The maximum efficiency point may be seen to follow a parabolic curve to zero with decreasing pump speed (rpm).



Centrifugal Pump Performance Curve

Figure 37



Pump Best Efficiency Curve

Figure 38

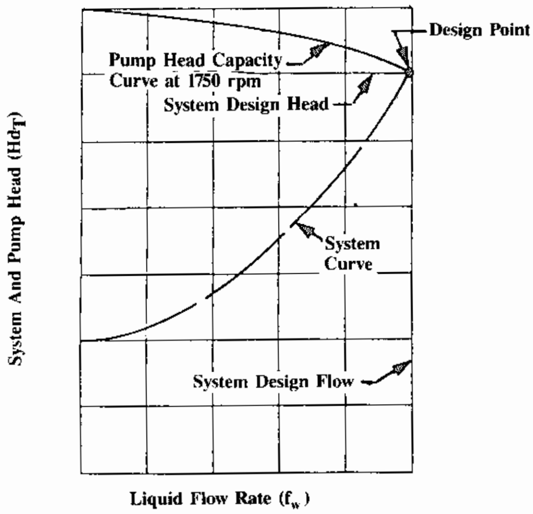
Positive Displacement Pump Performance

The performance of a positive displacement pump is normally presented in tabular form based on flow capacity, total differential pressure and power input required. Since these pumps are often used for liquids other than water, analysis of their performance includes consideration of the particular liquid characteristics and usually requires assistance by a qualified expert. For water pumping applications the basic pump energy equations apply equally to centrifugal or positive displacement pumps.

Application Curves

An application curve (Figure 39) consists of the system head and pump head capacity curves plotted on the same graph with liquid flow rate as the base. For the system represented in Figure 39, the point at which the pump head capacity and system head curves intersect, called the design point, represents the flow and pressure at which the pump will operate. The system head curve for a particular system must be manually plotted using an estimated system head for one flow rate and the equations for system head and flow described previously. An application curve can be created by plotting the system curve on a manufacturer's pump performance curve.

Application curves are a useful tool for selecting the most efficient pumping arrangements for systems with varying flow and pump head requirements.



Application Curve
Figure 39

Pump Efficiency

The efficiency of a pump is the total output of useful energy divided by the actual power input to the pump shaft and is represented by the following equation.

$$Ef_p = \frac{f_w \text{ x } Hd_T}{Wp_i \text{ x } 367}$$

Where, Ef_p = pump efficiency (decimal)

f_w = liquid flow rate (m^3/h)

Hd_T = total pump head (m)

Wp_i = power input to the pump shaft (kW)

367 = conversion of units

Pump Power

The power required to pump a liquid is a function of the volumetric flow rate and the total system head. The *ideal power* required to move a flow of liquid against the total pump pressure can be calculated by the following equation.

$$Wp = \frac{f_w \text{ x } Hd_T}{367}$$

Where, Wp = ideal pump power (kW)

Calculations of ideal pump power can be used to estimate the effects of changes to a system when the manufacturer's performance data is not available. For example, the effect of changes in liquid flow and total pump head on the power input to a pump motor can be estimated by multiplying the measured initial power input by the ratio of ideal pump powers. The initial power input to an electric motor (Wmi) can be determined by measurement using the methods described in Appendix E. The revised power input to the motor can then be estimated by the following equation.

$$Wm_{i2} = Wm_{i1} \text{ x } \frac{Wp_2}{Wp_1}$$

Where, Wm_{i1}, Wm_{i2} = initial and revised motor power inputs' (Appendix E) (kW)

Wp_1, Wp_2 = initial and revised ideal pump powers (kW)

The *total power input* required to operate a pump includes losses owing to pump, drive and motor efficiencies. Mass density of the liquid does not affect the basic power relationships but it may affect the performance rating of particular pumps. The following extension of the pump power equation defines the power input to the motor in terms of the system conditions and component efficiencies.

$$Wm_i = \frac{f_w \text{ x } Hd_T}{Ef_p \text{ x } Ef_d \text{ x } Ef_m \text{ x } 367}$$

Where, Wm_i = motor power input (kW)

Ef_p = pump efficiency (decimal)

Ef_d = drive efficiency (decimal)

Ef_m = motor efficiency (Appendix E) (decimal)

367 = conversion of units

Drive loss is normally stated as per cent of the input power. When the energy loss for a belt drive system is not known, a value from Table 1 may be used. For estimating purposes the drive loss for a direct coupled motor can be considered to be zero.

The drive loss can be converted to drive efficiency.

$$Ef_d = 1 - \frac{\text{Drive Loss}}{100}$$

Where, Ef_d = drive efficiency (decimal)

Drive loss is expressed in per cent

100 = conversion of per cent to decimal

The power input to the pump shaft from a particular drive system can be calculated from the following equations.

$$Wp_i = Wm_i \text{ x } Ef_m \text{ x } Ef_d$$

or, $$Wp_i = Wm_o \text{ x } Ef_d$$

Where, Wp_i = power input to the pump shaft (KW)

Wm_o = motor shaft power output (Appendix E) (kW)

Energy Analysis of a Pump

The following example demonstrates the use of the pump performance equations.

As part of an energy conservation program a municipality undertook to check the operating efficiencies of its water supply pumps. All pumps were equipped with operating time totalizers, suction and discharge pressure gauges, and flow meters. The following performance measurements were recorded for a direct-driven centrifugal pump on the main reservoir supply using the monitoring devices in place and portable meters to measure the motor electrical power input.

Total suction pressure	70 kPa(gauge)
Total discharge pressure	586 kPa(gauge)
Liquid flow rate, f_w	270 m^3/h
Measured operating time, h	4300 h/yr
Average unit electrical energy cost, Ce	$0.05/kWh
Pump motor data:	
Rated voltage, V_r	575 volts
Measured voltage, V	577 volts
Rated full load current, I_r	90.58 amperes
Measured current draw, I	73.82 amperes
Rated full load power factor, $p.f._r$	0.90
Measured power factor, p.f.	0.87
Phases	3 phase

The recorded data must first be converted into the standard units used in the analysis equations.

The total pump differential pressure, DP_T = total discharge pressure − total suction pressure

$$= 586 - 70$$

$$= 516 \text{ kPa}$$

$$\text{Total pump head, } Hd_T = \frac{DP_T}{9.81}$$

$$= \frac{516}{9.81}$$

$$= 52.6 \text{ m}$$

The motor shaft power input can be calculated using the equations from Appendix E.

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{73.82 \times 577 \times 0.87}{90.58 \times 575 \times 0.90}$$

$$= 0.79$$

Estimated motor efficiency, Ef_m (Figure E-1) = 0.92

$$\text{Power input to motor, } Wm_i = \frac{V \times I \times 1.73 \times p.f.}{1000}$$

$$= \frac{577 \times 73.82 \times 1.73 \times 0.87}{1000}$$

$$= 64.11 \text{ kW}$$

For a direct drive arrangement, the drive loss is considered to be zero so the drive efficiency (Ef_d) is 1.

$$\text{Power input to the pump shaft, } Wp_i = Wm_i \times Ef_m \times Ef_d$$

$$= 64.11 \times 0.92 \times 1$$

$$= 58.98 \text{ kW}$$

$$\text{Ideal pump power required, } Wp = \frac{f_w \times Hd_T}{367}$$

$$= \frac{270 \times 52.6}{367}$$

$$= 38.7 \text{ kW}$$

$$\text{Pump efficiency, } Ef_p = \frac{Wp}{Wp_i} \times 100$$

$$= \frac{38.7}{58.98} \times 100$$

$$= 66\%$$

This is a relatively poor efficiency for such a pump. Modifying or replacing the pump to achieve 75 per cent pump efficiency would probably be cost effective. Using this expected value, the energy saving can be estimated.

$$\text{Revised power input to the pump shaft, } Wp_{i2} = \frac{Wp \times 100}{Ef_{P2}}$$

$$= \frac{38.70 \times 100}{75}$$

$$= 51.6 \text{ kW}$$

$$\text{Revised motor power input, } Wm_{i2} = Wm_{i1} \times \frac{Wp_{i2}}{Wp_{i1}}$$

$$= 64.11 \times \frac{51.6}{58.98}$$

$$= 56.09 \text{ kW}$$

$$\text{Annual energy cost saving} = (Wm_{i1} - Wm_{i2}) \times h \times Ce$$

$$= (64.11 - 56.09) \times 4300 \times 0.05$$

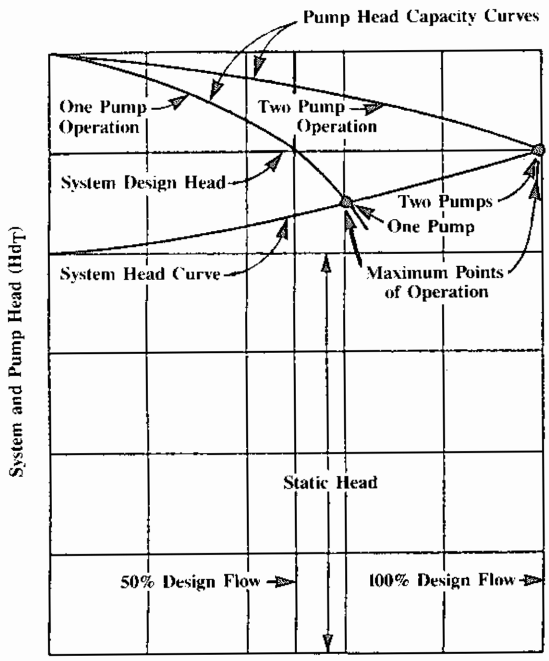
$$= \$1{,}724/\text{yr}$$

Multiple Pump Operation

In order to increase either flow rate or total pump head, multiple pumps may be operated in parallel or series.

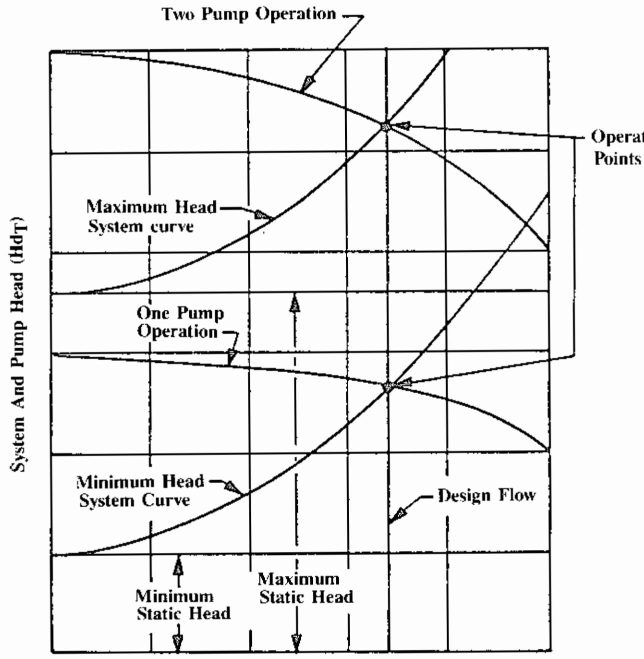
Figure 40 is a typical application curve for two pumps of equal size in *parallel* serving a system requiring a varying flow. The head capacity curve for two pump operation is plotted by doubling the flow for one pump operation at each particular system head value. The system provides two possible operating points for a particular system curve by running either one or both pumps. Typical applications include boiler feed and process piping systems.

Figure 41 is a typical application curve for two pumps of equal size in *series* serving a system requiring a varying static head. The head capacity curve for two pump operation is plotted by doubling the head for one pump operation at each particular system flow value. Typical applications include water supplies to high-rise buildings and reservoir filling systems.



Note: Two pumps, each with a capacity of 50% design flow at 100% design head.

Application Curve For Parallel Pumps
Figure 40



Note: Two pumps, each with a capacity of 100% design flow at 50% design head.

Application Curve For Series Pumps
Figure 41

Pump Seals

The type of shaft seal used on a centrifugal pump and the quality of maintenance performed can have a significant effect on the overall efficiency of the pump. Friction at the shaft seal uses a portion of the shaft input power and leakage past the seal represents a loss of the pumped liquid. The most common types are *packing gland* and *mechanical seals*. The seal configurations and typical applications are described in the Equipment section.

Table 5 is a graph of reported power consumptions by the mechanical seals of one manufacturer for various shaft sizes and pump casing pressures when operated with clean water in the pump. The comparative power consumptions of packing gland seals vary widely with the materials used and leakage rates tolerated, but tests have indicated average consumptions equal to six times those of mechanical seals.

The power consumption data of Table 5 is based on a shaft speed of 1000 rpm so it is necessary to correct the data for other speeds.

$$Ws_2 = Ws_1 \times \frac{n}{1000}$$

Where, Ws_2 = seal power consumption at the actual speed (kW)

Ws_1 = seal power consumption (Table 5) (kW)

n = shaft speed (rpm)

1000 = shaft speed for Table 5 data (rpm)

The pump casing pressure at the seal may be equal to the discharge or the suction head of the pump, depending on the pump configuration. Most centrifugal pumps with single-sided impellers subject the shaft seal to the discharge head. Turbine pumps and centrifugal pumps with double-sided impellers subject the shaft seals to the suction head. The head in metres of water can be converted to pressure in kPa(gauge).

$$P = Hd_d \times 9.81$$

Where, P = pump casing pressure (kPa)

Hd_d = pump discharge head (m)

9.81 = kPa per metre of head

Summary

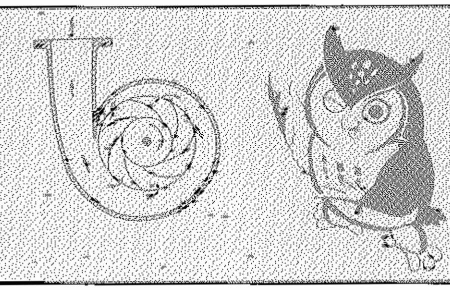
Numerous energy and cost saving opportunities exist for pumps. Alert personnel, with an awareness of energy management techniques, can easily learn to recognize these opportunities and benefit from them.

The effective utilization of energy by pumps is affected by several factors.

- The configuration of the pumps.
- Inlet and outlet flow conditions.
- Application of pumps to the requirements of systems.
- The efficiencies of the pumps.
- Operation of multiple pump systems.

Pumps may be analyzed in detail to estimate the energy input, energy transfer, losses, costs and potential savings. Worksheets 13-P1 through 13-P4 summarize the data and calculations necessary for the analysis of pumps and their auxiliaries.

EQUIPMENT SYSTEMS



As listed in Figure 31 and discussed in the Fundamentals section, the types of pumps addressed in this module are centrifugal and positive displacement. Centrifugal pumps are subdivided according to impeller and casing configuration. Positive displacement pumps are subdivided into reciprocating and rotary types.

Centrifugal Pumps

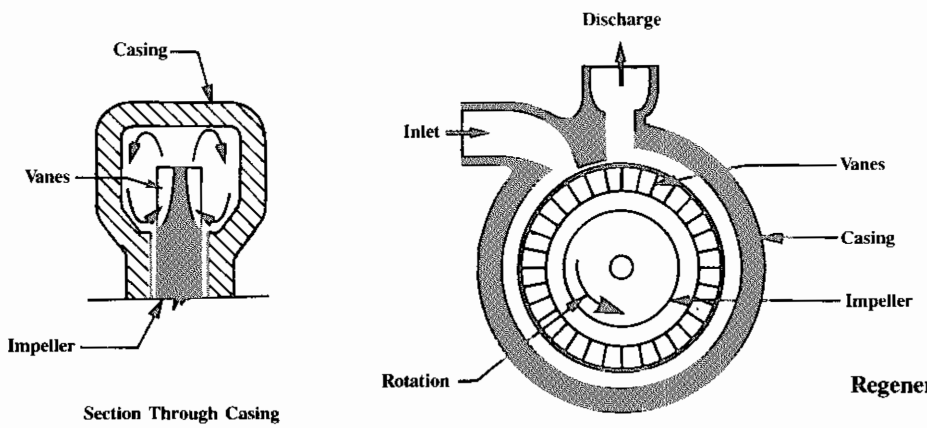
Centrifugal pumps are the most commonly used type of pump and at least one variety can be found in most facilities. Smaller sizes are used for such applications as inline heating water circulation, domestic water, boiler feed, condensate, and process fluid pumping. Larger sizes are used in municipal water, fire protection, sewage, land drainage, flood control, irrigation, coffer dams and dry dock dewatering systems.

Centrifugal pumps are widely used because of their relatively simple operation. They are available in many sizes and types and have only one moving part. Table 6 is a listing of the normal capacity of common types of centrifugal pumps.

Radial Flow Pumps

The *volute type* of radial flow pump (Figure 32) is the most common type of centrifugal pump. An impeller discharges the fluid into a progressively expanding spiral or scrolled casing. These pumps are available with impellers shaped for different applications. Open vane impellers are used to pump corrosive liquids or liquids containing abrasive solids. They have a low efficiency and are normally found in low head, low flow applications. Semi open vane impellers are used in a wide range of sizes and configurations for low to medium head applications (up to 100 m) ranging from heating water circulators to sewage pumps. Some configurations are capable of handling liquids containing large or stringy solids. Closed vane impellers are normally designed for high efficiencies at high heads (up to 150 m) and are used for clean liquid applications. Double-sided impellers are used for large flow applications (greater than 25 m^3/h).

The *regenerative turbine* pump (Figure 42) is a special adaptation of the volute type of centrifugal pump. It consists of an impeller wheel that has small vanes attached to its outer edge. The liquid is forced outward by centrifugal force against a shaped casing channel. The liquid strikes the casing and is turned back against the impeller between the next set of vanes. The effect is that of many miniature pumps operating in series. These pumps have low flow capacities but are capable of high heads. For the same service conditions, the regenerative turbine pump is smaller in size and lower in cost than a volute type centrifugal or a positive displacement pump. Because of the high velocity that occurs at the vanes, and the close clearances, the regenerative turbine pump cannot withstand abrasive liquids. This type of pump is often used for condensate, liquified gases, boiler feed water and car wash water.

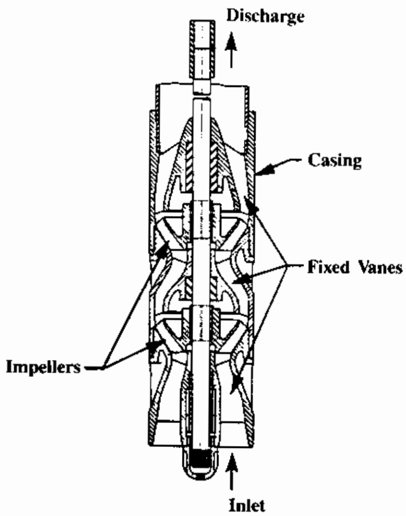


Regenerative Turbine Radial Flow Pump
Figure 42

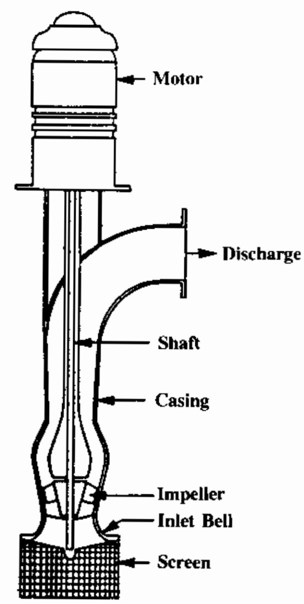
The *vertical turbine* pump (Figure 43) was first used for pumping water from wells. For this reason it has sometimes been called a deep-well pump, a turbine-well pump or a borehole pump. It operates with several impellers in series on a common shaft, each rotating in a bowl shaped cavity. The number of impellers can be increased to provide a higher head capacity for a particular flow. Vertical turbine pumps are used in wells and reservoir pumping for irrigation, municipal and industrial water supplies, processing, refrigeration and air conditioning.

Axial Flow Pump

The liquid flow in an axial flow pump (Figure 44) is parallel to the axis of rotation. The pump is usually installed in the vertical position. The head is achieved by the lifting action of the impeller blades resembling that of a propeller. A version of the pump uses a mixed flow impeller that is a cross between an axial and a radial flow impeller. Axial pumps are usually rated for low head and high flow capacity. This makes them ideal for such uses as irrigation, flood control and drainage of coffer dams, dry docks and land drainage. They are also able to handle dirty liquids with some solid content.



Vertical Turbine Radial Flow Pump
Figure 43



Axial Flow Pump
Figure 44

Positive Displacement Pumps

Positive displacement pumps are often used where a constant flow is required against a varying pump head. They are also used in combination with a relief valve by-pass arrangement to provide a constant pump head against varying flow requirements.

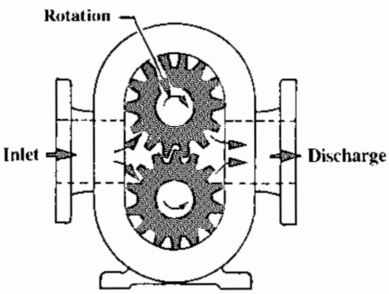
Rotary Pumps

Rotary pumps usually consist of a fixed casing with rotating internal elements. These rotating elements may be gears, vanes or screws that operate with minimum clearance, creating cavities to trap the liquid and push it through the casing discharge. Table 7 lists the capacity capabilities of rotary pumps normally encountered.

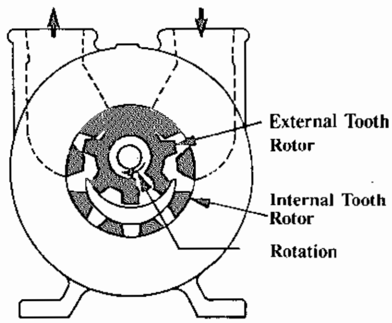
The external gear pump (Figure 45) is the simplest type of rotary pump. The liquid fills the spaces between gear teeth, is carried around the casing and is squeezed out as the gears mesh. This type of pump is used for viscous liquids such as oils and water – glycol mixtures. Although these pumps have a low flow capacity, they have the capability to produce a high pressure.

The internal gear pump (Figure 46) has one rotor with external gear teeth that mesh with the internal teeth of an idler gear. A crescent shaped piece prevents liquid from passing to the suction side. The internal gear pump has similar uses to those of the external gear pump.

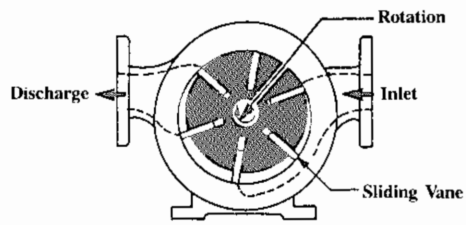
The *vane*, or *sliding vane* pump (Figure 47) has a rotor equipped with vanes that slide outward to seal against an offset circular casing. The liquid trapped between the vanes is carried around and forced out the discharge. The vane pump is used for pumping viscous liquids, hydraulic fluids, and is commonly found in food industries such as breweries, beverage producers, canneries and candy manufacturers.



External Gear Pump
Figure 45



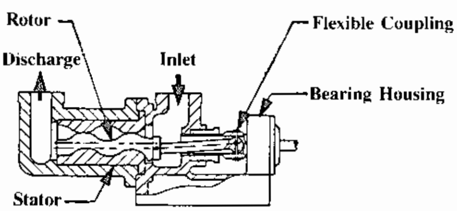
Internal Gear Pump
Figure 46



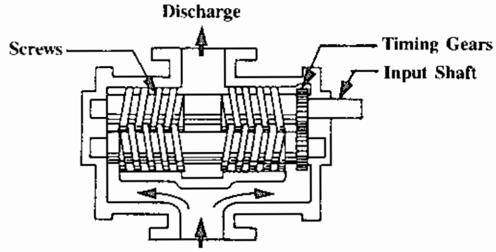
Vane Pump
Figure 47

The single screw or *progressing cavity* pump (Figure 48) has a spiralled rotor that turns eccentrically in an internal helix stator or liner. As the rotor turns, a cavity is formed that progresses towards the pump discharge end. This pump has the advantages of being self-priming, reversible, able to handle abrasive material and able to meter flow without pulsating. The single screw pump is used for pumping clear water, heavy and abrasive slurries, solvents, foods such as soups, vegetable oils and molasses, and petroleum products.

The *multiscrew* pump (Figure 49) has two or three screws and the liquid flow is between the threads along the length of the screws. Opposed screws shown in Figure 49 are used to eliminate thrust. The pump is used for pumping viscous oil, asphalt and a wide range of chemicals and food products.



Single Screw Pump
Figure 48

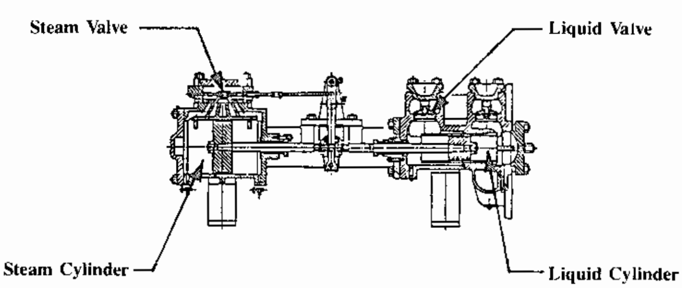


Multiscrew Pump
Figure 49

Reciprocating Pumps

Reciprocating pumps use the principle of a moving piston to draw liquid into a cylinder through an inlet valve and push it out through a discharge valve. They have positive displacement, which makes them suitable for metering and pumping viscous liquids. Reciprocating pumps come in a variety of sizes and pressure ranges (Table 8).

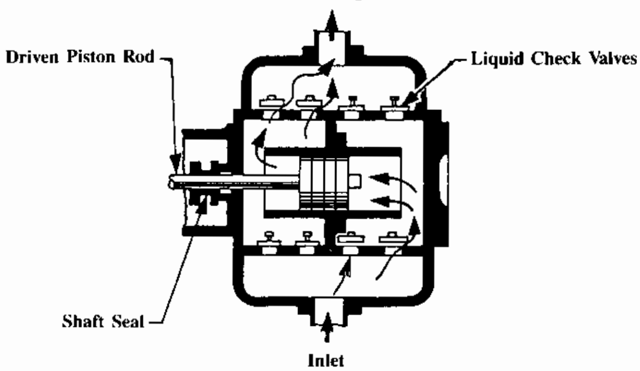
The *direct acting* reciprocating pump (Figure 50), sometimes called a reciprocating steam pump, consists of two pistons connected by a common piston rod. One piston, usually the larger, is driven by steam to produce reciprocating motion. This simultaneously moves the second piston in the liquid cylinder to do the actual pumping. This type of pump has been used for many years for pumping a wide variety of industrial liquids. Although reliable, the pump has high repair cost because of the many moving parts.



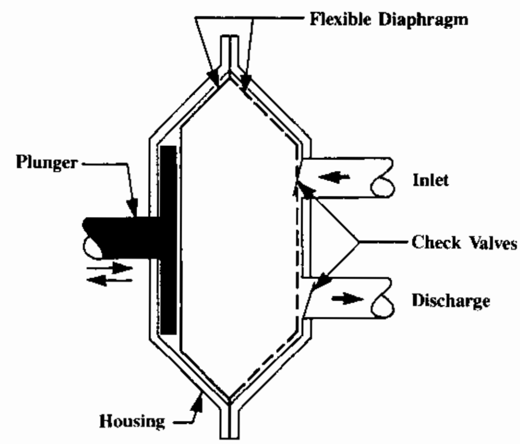
Direct Acting Reciprocating Pump
Figure 50

The *power reciprocating pump* (Figure 51) has an external drive source and a cylinder configuration. Smaller sizes are commonly used for domestic water service and viscous liquids. Larger sizes for higher flows are used in high pressure wash systems, fluid waste disposal, mine dewatering and shipping liquid products in pipelines.

The *diaphragm* pump (Figure 52) has a flexible diaphragm in the pumping cavity that is pulsed by either an air supply or a reciprocating plunger. The pump construction allows the pumping of a variety of materials such as chemicals, glue, ink, solvents, fat, grease and dirty water. Because of the diaphragm, the air supply or plunger does not come in contact with the material being pumped. The diaphragm pump is limited to low flow and head applications.

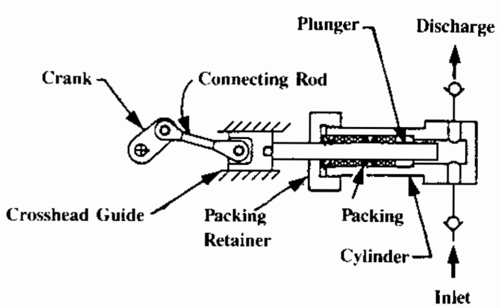


Power Reciprocating Pump
Figure 51



Diaphragm Pump
Figure 52

A *metering pump* is used to accurately pump liquids at low flow. The most common metering pumps are reciprocating piston or plunger types (Figure 53). The capacity stays effectively constant with changing pressure. Some designs use diaphragms to separate aggressive chemicals or abrasive materials from the reciprocating plunger. Various rotary type pumps are also used for metering but their accuracies are affected by changing pressure owing to leakage past mating surfaces.



Piston Type Metering Pump
Figure 53

Pump Seals

Various types of seals are used to minimize liquid leakage where the shaft passes through the pump casing. The most common types are *packing glands* and *mechanical seals.*

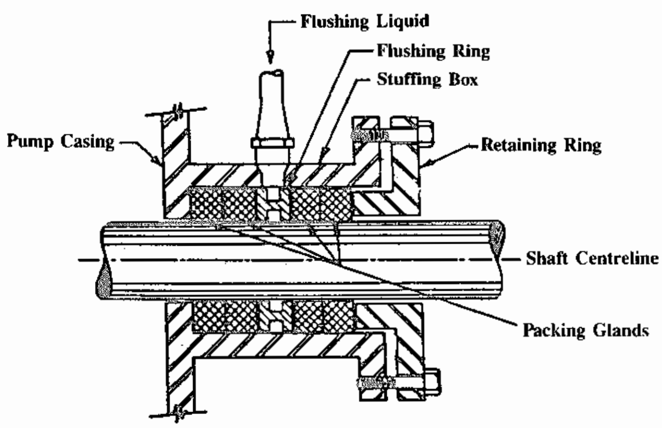
Packing Glands

Packing gland seals (Figure 54) consist of multiple rings of a flexible, low friction packing material compressed to achieve intimate contact with the shaft and pump casing. Forced lubrication is usually provided between the shaft and the packing rings. Lubrication methods include controlled leakage of the pumped liquid, forced flushing by a separate liquid, and controlled force feeding of oil or grease. In all cases the power used, the liquid lost and the maintenance life of the seal depend on skilled adjustment of the retaining ring pressure on the packing rings. This type of seal has a low first cost but a high operating and maintenance cost.

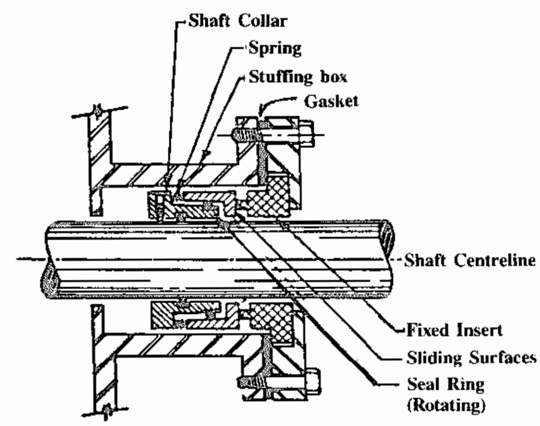
Mechanical Seals

Mechanical seals (Figure 55) consist of spring loaded rings of a rigid, low friction material sliding against finely finished mating surfaces. The seal ring may be made of a self-lubricating material or the seal surfaces may be lubricated by slight leakage of the pumped liquid. This type of seal has a relatively high cost but their effect on power consumption can be one-sixth of that of packing gland seals, resulting in significant operating cost savings. Special types of mechanical seals are available for retrofitting existing pumps that have packing gland seals.

Mechanical seals can be economically justified for most pumps but they have been used most frequently for pumps handling valuable or hazardous liquids.

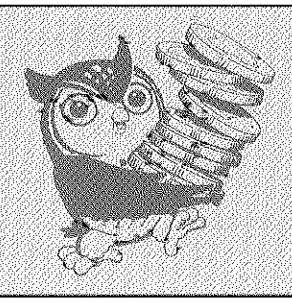


Packing Gland Seal
Figure 54



Mechanical Seal
Figure 55

ENERGY MANAGEMENT OPPORTUNITIES



Energy Management Opportunities is a term that represents the ways that energy can be used wisely to save money. A number of Energy Management Opportunities, subdivided into Housekeeping, Low Cost, and Retrofit categories are outlined in this section, with worked examples or text to illustrate the potential energy savings. In some examples the energy dollar savings are small, however, when many changes are combined, the resulting savings can be substantial. This is not a complete listing of the opportunities available for pumps. However, it is intended to provide ideas for management, operating, and maintenance personnel to allow them to identify other opportunities that are applicable to a particular facility. Other modules in this series should be considered for Energy Management Opportunities applicable to other types of equipment and systems.

Housekeeping Opportunities

Implemented housekeeping opportunities are Energy Management actions that are done on a regular basis and never less than once a year. The following are typical Energy Management Opportunities in this category.

1. Ensure that packing glands on pumps are correctly adjusted.
2. Maintain clearance tolerances at pump impellers and seals.
3. Check and adjust motor driver regularly for belt tension and coupling alignment.
4. Clean pump impellers and repair or replace if eroded or pitted.
5. Regularly check and recalibrate control components such as timers and devices for speed control.
6. Shut down pumps when liquid flow is not required.
7. Implement a program of planned maintenance to minimize pump component failures.

Housekeeping Worked Examples

1. Pump Packing Glands

Pump packing glands should be checked periodically for correct adjustment of tightness. Depending upon the packing material, liquid temperature, shaft speed and allowable liquid leakage, an optimum tightening of the packing may be established by monitoring the rate of dripping from the packing. Pump and packing material manufacturers can provide guidance on packing tightness for specific applications. With few exceptions, packings must leak slightly for lubrication or excessive energy will be used and mechanical damage to the shaft will occur.

2. Critical Tolerances

The efficiency of a pump is affected by the amount of leakage past the impeller from the discharge to the suction. Some pumps have replaceable wear rings with small clearances between moving surfaces to minimize leakage and maintain serviceability. Mechanical seals are used with critical tolerances to stop leakage and to stop air from being drawn into the liquid flow. These clearances can be affected by erosion of the impeller and wear rings when pumping liquids that contain abrasive particles. The clearances and surfaces must be checked and maintained periodically to keep the pump efficiency high.

3. Check and Adjust Drives

Open drives, such as belts and flexible couplings, provide long service when properly designed and maintained. The following actions should be regularly carried out.

- Maintain alignment of pulleys and couplings.
- Check tension of belts.
- Lubricate bearings.
- Replace or repair damaged belts, pulleys, clutches, drive keys and couplings.

Proper tensioning for various types of belts is described in handbooks and catalogues available from component manufacturers.

4. Clean Pump Impeller

Pumps, particularly those moving dirty liquid, should be regularly cleaned to maintain the efficiency of the pump. Refuse collected on the impeller and the housing of the pump causes higher static pressure losses in the pump itself, reducing its efficiency.

5. Controls

During an annual review of the operating performance of a water treatment plant it was noted that a centrifugal pump having a variable speed driver was operating against a flow control valve at a 7 m higher head than required to maintain the flow. It was recognized the electrical energy could be saved by reducing the pump speed to meet the design head.

The following data was compiled from the daily system log reports.

Controlled liquid flow rate, f_w	340 m^3/h
Revised total pump head, Hd_{T2}	28 m
Initial total pump head, Hd_{T1}	35 m
Totalized annual operating time, h	3600 h

The following motor power data was measured by the plant electrical maintenance staff.

Rated voltage, V_r	575 volts
Phases	3 phase
Measured voltage, V	580 volts
Rated motor shaft power output	55.95 kW
Rated full load current, I_r	71.0 amperes
Measured current, I	48.3 amperes
Rated full load power factor, $p.f._r$	0.88
Measured power factor, p.f.	0.85

The average unit electrical energy cost was $0.05/kWh. Worksheet 13-E1 was used to determine the initial motor efficiency to be 0.91 and the initial motor power input to be 41.19 kW.

Worksheet 13-P1 was used to calculate the energy cost saving to be $1,483 per year.

Since this action involves only control adjustments, there is no capital cost and the payback is immediate.

6. Shut Down Pump When Liquid Flow Not Required

Savings in both energy and maintenance costs can be achieved by shutting down pumps when liquid flow is not required in the system. Heating water circulating pumps can normally be shut down during the summer months, and process cooling water pumps can often be shut down when the process is not operating.

7. Maintenance Programs

A maintenance program for pumps should be tailored to the specific needs of the facility and could include the following actions.

- Daily; monitor pump sounds, bearing temperature, stuffing box or mechanical seal leakage, and gauge and meter readings.
- Semiannually; check the free movement of stuffing box glands, inspect the packing, check the pump and driver alignment, drain and refill oil lubricated bearings, check the quantity and consistency of the grease in grease lubricated bearings, and lubricate packing gland bolts.
- Annually; clean, inspect and lubricate bearings and their seals; examine the packing and the shaft sleeve; check the coupling and alignments; check shaft movement; check and clean auxiliary systems such as seal liquid lines, strainers and coolers; recalibrate all associated instrumentation; and check pump performance against the design ratings.
- Replace worn components when tests indicate loss of performance.
- Adjust impeller clearances on pumps when tests indicate loss of performance.
- Maintain performance, power requirement, and repair records for each pump.

Maintenance personnel must have the capability and experience to service, repair and troubleshoot the pumps and distribution systems. Training should be provided covering new equipment, changes to the facility and new procedures.

Low Cost Opportunities

Implemented low cost opportunities are Energy Management actions that are done once and for which the cost is not considered great. The following are typical Energy Management Opportunities in this category.

1. Replace packing gland type shaft seals with mechanical seals.
2. Trim pump impeller to match system flow rate and head requirements.

Low Cost Worked Examples

1. Mechanical Seals

A heating water pump having a 51 mm diameter shaft and packing gland seals operates continuously (8760 hours per year) at 1750 rpm with a discharge head of 30 m. During an annual maintenance shutdown it was desired to replace the packing gland seal with a mechanical seal to reduce leakage of heating water and reduce the pumping energy cost.

Worksheet 13-P2 was used to estimate the annual energy cost saving to be $250/yr.

The estimated incremental cost to install the mechanical seal instead of renewing the packing gland packing rings during the scheduled shutdown was $1,100.

$$\text{Simple payback} = \frac{\$1,100}{\$250}$$

$$= 4.4 \text{ years.}$$

Although the energy cost saving payback is relatively long, additional cost savings would be realized owing to reduced water leakage, reduced water chemical treatment and reduced pump shaft maintenance.

2. Liquid Flow Reduction

A process plant utilized a centrifugal pump to provide cooling water circulation to several processes. Through changes in operations it was found that a certain process with an estimated flow of 12 m^3/h could be eliminated. It was concluded that energy cost saving could be achieved by trimming the pump impeller to suit the revised flow requirement.

The initial design flow was 125 m^3/h at 70 m total pump head. Run time totalizer records indicated an average operation time of 2100 hours per year. Pressure gauge readings at the pump inlet and discharge confirmed that the design head was being met. The electrical maintenance staff obtained the following motor performance data from the motor nameplate and by measurement with portable meters.

Rated motor shaft power output	37.3 kW
Rated voltage, V_r	575 volts
Rated full load current, I_r	47.1 amperes
Number of phases	3
Rated full load power factor, $p.f._r$	0.88
Measured voltage, V	578 volts
Measured current, I	40.02 amperes
Measured power factor, p.f.	0.87
Measured existing impeller diameter, D_1	197 mm

The average unit electrical energy cost was $0.05/kWh. Worksheet 13-E1 was used to calculate the electric motor power input to be 32.45 kW and the motor shaft power output to be 29.85 kW.

Worksheet 13-P2 was used to calculate the revised impeller diameter to be 178 mm and the annual energy cost saving to be $894 per year.

The estimated cost of machining the impeller is $2,500.

$$\text{Simple payback} = \frac{\$2,500}{\$894}$$

$$= 2.8 \text{ years}$$

Retrofit Opportunities

Implemented retrofit opportunities are Energy Management actions that are done once and for which the cost is significant. Many of the opportunities in this category will require detailed analysis by specialists, and cannot be examined in this module. Worked examples are provided for some of the listed Energy Management Opportunities, while in other cases there is only commentary. The following are typical Energy Management Opportunities in the Retrofit category.

1. Install variable speed controller on pumps to better match liquid flow demand.
2. Replace outdated equipment with new units sized at optimum efficiency.
3. Replace oversized motors.
4. Install a microprocessor energy management control system.

Retrofit Worked Examples

1. Install Variable Speed Drive

Energy conservation measures at a shopping complex included shutdown of local air handling systems at night and on holidays totalling 4518 hours per year. During this period the central hot water system heating pump flow requirement was reduced from 115 m^3/h at 38 m total system head to 35 m^3/h at 10 m total system head. The central heating pump was a centrifugal type operating at constant speed. The initial effect of shutting off the heating water flow to the local air systems was to cause the central pump to operate at 42 m total pump head. From the manufacturer's performance curves the head of 42 m represented a flow of 71.7 m^3/h. It was recognized that energy cost could be saved by varying the pump speed to better match the system requirements. Because the lower speed would require a much lower power input, a variable frequency motor speed control was selected to maximize motor efficiency. The average unit electrical energy cost was $0.05/kWh.

The following measurements and data were compiled from the motor nameplate and pressure gauge readings of the pump head under the restricted flow conditions.

Initial total pump head, Hd_T	42 m
Initial liquid flow rate (pump curve)	71.7 m^3/h
Motor data:	
Rated voltage, V_r	575 volts
Phases	3 phase
Measured voltage, V	580 volts
Rated full load current, I_r	29 amperes
Measured current, I	20.54 amperes
Rated full load power factor, $p.f._r$	0.88
Measured power factor, p.f.	0.79

Worksheet 13-E1 was used to calculate the initial motor power input to be 16.28 kW.

Worksheet 13-P1 was used to calculate the annual energy cost saving to be $3,253 per year.

The estimated cost to install the variable speed controller and controls interconnection is $13,000.

$$\text{Simple payback} = \frac{\$13{,}000}{\$3{,}253}$$

$$= 4.0 \text{ years}$$

2. Install More Efficient Pump

A belt-driven centrifugal pump at a water treatment facility had been in use for over 30 years and was thought to be operating at low efficiency. The pump operated 24 hours per day, or 8760 hours per year. It was recognized that replacement of the pump with a new, more efficient unit would save energy cost and reduce maintenance costs. The average electrical energy cost was $0.05/kWh.

System instrumentation of flows and pressures indicate the pump was operating at 90.7 m^3/h and 49 m total pump head. A new pump was selected having a rated efficiency of 77 per cent and a shaft power input of 15.73 kW at the same operating conditions. The new pump is driven by an 18.65 kW motor having a rated efficiency of 93 per cent and power factor of 0.88.

The following data for the original pump motor was recorded by meter measurement and from the motor nameplate.

Rated voltage, V_r	575 volts
Rated full load current, I_r	38.6 amperes
Rated full load power factor, $p.f._r$	0.85
Measured voltage, V	570 volts
Measured current, I	34.0 amperes
Measured power factor, p.f.	0.8

Worksheet 13-E1 was used to calculate the initial motor power input to be 26.82 kW.

Worksheet 13-P4 was used to calculate the annual energy cost saving to be $4,030 per year.

The cost of installing the new pump is estimated to be $5,000.

$$\text{Simple payback} = \frac{\$5{,}000}{\$4{,}030}$$

$$= 1.2 \text{ years.}$$

Additional saving would also be achieved owing to lower maintenance costs for the new pump.

3. Replace Oversize Motors

The advantage of operating an electric motor close to full load, where the power factor and efficiency are at their maximum, is readily seen in Figure E-1.

During an energy audit a 7.5 kW motor was observed driving a direct drive centrifugal pump which was operating under conditions requiring only 2.25 kW motor power output. It was recognized that electrical energy cost could be saved by installing a 2.25 kW motor. The audit had determined that the pump operates a total of 4160 hours per year.

The following data for the initial pump motor was recorded by meter measurement and from the motor nameplate.

Rated voltage, V_r	575 volts
Rated full load current, I_r	8. amperes
Rated full load power factor, $p.f._r$	0.88
Measured voltage, V	575 volts
Measured current, I	5.03 amperes
Measured power factor, p.f.	0.60

Worksheet 13-E1 was used to calculate the initial motor power input as 3kW. Worksheet 13-E2 was used to calculate the annual energy cost saving to be $114 per year. The cost of installing the smaller motor is estimated to be $500.

$$\text{Simple payback} = \frac{\$500}{\$114}$$

$$= 4.4 \text{ years}$$

In similar instances, additional savings may be obtained if an increased power factor would have an impact on the utility demand charges. This and high efficiency electric motors are discussed in Energy Efficient Motors, Module 4.

4. Install a Pump Management System

A microprocessor pump management system can accomplish energy cost savings over and above the savings from individual actions by monitoring and integrating the various control functions.

The analysis and selection of such equipment should be based on a professional review of the requirements for the particular facility.

Electric Motor Drive Performance

Worksheet 13-E1

Company: Worked Example #5 Date: 86/06/13

Location: Pump - Housekeeping By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	71.0	amps
Measured voltage, V	580	volts
Measured current, I	48.3	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	55.95	kW
Rated full load power factor, $p.f._r$	0.88	(decimal)
Measured power factor, p.f.	0.85	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{48.3 \times 580 \times 0.85}{71.0 \times 575 \times 0.88}$$

$$= 0.66$$

Motor efficiency, Ef_m (Figure E-1) 0.91 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{580 \times 48.3 \times 1.73 \times 0.85}{1000}$$

$$= 41.19 \text{ kW}$$

Motor shaft power output, Wm_o = Wm_i x Ef_m

= _____ x _____

= N/A kW

Pump Capacity Reduction
Worksheet 13-P1

Company: WORKED EXAMPLE #5 Date: 86/06/13

Location: PUMPS - HOUSEKEEPING By: MBE

Data

Initial liquid flow rate, f_{w1}	340	m^3/h
Revised liquid flow rate, f_{w2}	340	m^3/h
Initial total pump head, Hd_{T1}	35	m
Revised total pump head, Hd_{T2}	28	m
Total operation time, h	3600	h/yr
Unit electrical energy cost, Ce	\$ 0.05	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	41.19	kW

Initial ideal pump power, Wp_1 $= \dfrac{f_{w1} \times Hd_{T1}}{367}$

$$= \frac{340 \times 35}{367}$$

$$= 32.43 \text{ kW}$$

Revised ideal pump power, Wp_2 $= \dfrac{f_{w2} \times Hd_{T2}}{367}$

$$= \frac{340 \times 28}{367}$$

$$= 25.94 \text{ kW}$$

Revised motor power input, Wm_{i2} $= Wm_{i1} \times \dfrac{Wp_2}{Wp_1}$

$$= 41.19 \times \frac{25.94}{32.43}$$

$$= 32.95 \text{ kW}$$

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$$= (41.19 - 32.95) \times 3600 \times 0.05$$

$$= \$\ 1{,}483 \text{ /yr}$$

Install Mechanical Seals
Worksheet 13-P2

Company: WORKED EXAMPLE #1 Date: 86/06/13

Location: PUMPS - LOW COST By: MBE

Operation time, h	8760	h/yr
Unit electrical energy cost, Ce	$ 0.05	/kWh
Shaft diameter	51	mm
Shaft speed, n	1750	rpm
Pump discharge head, Hd_d	30	m

Pump discharge pressure, $P = Hd_d \times 9.81$

$= 30 \times 9.81$

$= 294$ kPa

Mechanical seal power consumption/1000 rpm, W_{S1} 0.065 kW/1000 rpm (Table 5)

Mechanical seal power consumption, $W_{S2} = \dfrac{n}{1000} \times W_{S1}$

$= \dfrac{1750}{1000} \times 0.065$

$= 0.114$ kW

Packing gland power consumption, $W_{S3} = W_{S2} \times 6$

$= 0.114 \times 6$

$= 0.684$ kW

Energy cost saving $= (W_{S3} - W_{S2}) \times h \times Ce$

$= (0.684 - 0.114) \times 8760 \times 0.05$

$= \$$ 250 /yr

Electric Motor Drive Performance

Worksheet 13-E1

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: Pumps - Low Cost By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	47.1	amps
Measured voltage, V	578	volts
Measured current, I	40.02	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	37.3	kW
Rated full load power factor, $p.f._r$	0.88	(decimal)
Measured power factor, p.f.	0.87	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{40.02 \times 578 \times 0.87}{47.1 \times 575 \times 0.88}$$

$$= 0.84$$

Motor efficiency, Ef_m (Figure E-1) 0.92 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{578 \times 37.3 \times 1.73 \times 0.87}{1000}$$

$$= 32.45 \text{ kW}$$

Motor shaft power output, Wm_o = $Wm_i \times Ef_m$

$$= 32.45 \times 0.92$$

$$= 29.85 \text{ kW}$$

Pump Capacity Reduction — Impeller Size

Worksheet 13-P3
(Page 1 of 2)

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: PUMPS - LOW COST By: MBE

Data:

Initial liquid flow rate, f_{w1}	125	m^3/h
Revised liquid flow rate, f_{w2}	113	m^3/h
Initial total pump head, Hd_{T1}	70	m
Existing Impeller diameter, D_1	197	mm
Operation time, h	2100	h/yr
Unit electrical energy cost, Ce	$ 0.05	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	32.45	kW
Initial motor shaft power output, Wm_{o1} (Worksheet 13-E1)	29.85	kW
Drive efficiency, Ef_d (1.0 for direct drive, Table 1 for belt drive)	1.0	

Revised pump head, $Hd_{T2} = Hd_{T1} \times \left(\frac{f_{w2}}{f_{w1}}\right)^2$

$$= 70 \times \left(\frac{113}{125}\right)^2$$

$$= 57.2 \text{ m}$$

Initial power input to the pump shaft, $Wp_{i1} = Wm_{o1} \times Ef_d$

$$= 29.85 \times 1.0$$

$$= 29.85 \text{ kW}$$

Pump Capacity Reduction — Impeller Size

Worksheet 13-P3
(Page 2 of 2)

Company: Worked Example #2 Date: 86/06/13

Location: Pumps - Low Cost By: MBE

Revised impeller diameter, $D_2 = D_1 \times \left(\frac{Hd_{T2}}{Hd_{T1}}\right)^{0.5}$

$$= 197 \times \left(\frac{57.2}{70}\right)^{0.5}$$

$$= \underline{178} \text{ mm}$$

Revised power input to the pump shaft, $Wp_{i2} = Wp_{i1} \times \left(\frac{D_2}{D_1}\right)^3$

$$= 29.85 \times \left(\frac{178}{197}\right)^3$$

$$= \underline{22.02} \text{ kW}$$

Revised motor power input, $Wm_{i2} = Wm_{i1} \times \frac{Wp_{i2}}{Wp_{i1}}$

$$= 32.45 \times \frac{22.02}{29.85}$$

$$= \underline{23.94} \text{ kW} \quad (4)$$

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$$= (32.45 - 23.94) \times 2100 \times 0.05$$

$$= \$\underline{894} \text{ /yr}$$

Electric Motor Drive Performance

Worksheet 13-E1

Company: WORKED EXAMPLE #1 Date: 86/06/13

Location: PUMPS - RETROFIT By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	29	amps
Measured voltage, V	580	volts
Measured current, I	20.54	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	(N/A)	kW
Rated full load power factor, $p.f._r$	0.88	(decimal)
Measured power factor, p.f.	0.79	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{20.54 \times 580 \times 0.79}{29 \times 575 \times 0.88}$$

$$= 0.64$$

Motor efficiency, Ef_m (Figure E-1) 0.91 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{580 \times 20.54 \times 1.73 \times 0.79}{1000}$$

$$= 16.28 \text{ kW}$$

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$

$= \_\_\_ \times \_\_\_$

$=$ N/A kW

Pump Capacity Reduction
Worksheet 13-P1

Company: WORKED EXAMPLE #1 Date: 86/06/13

Location: PUMPS - RETROFIT By: MBE

Data

Initial liquid flow rate, f_{w1}	71.7	m^3/h
Revised liquid flow rate, f_{w2}	35	m^3/h
Initial total pump head, Hd_{T1}	42	m
Revised total pump head, Hd_{T2}	10	m
Total operation time, h	4518	h/yr
Unit electrical energy cost, Ce	\$ 0.05	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	16.28	kW

Initial ideal pump power, Wp_1 $= \dfrac{f_{w1} \times Hd_{T1}}{367}$

$= \dfrac{71.7 \times 42}{367}$

$=$ 8.21 kW

Revised ideal pump power, Wp_2 $= \dfrac{f_{w2} \times Hd_{T2}}{367}$

$= \dfrac{35 \times 10}{367}$

$=$ 0.95 kW

Revised motor power input, Wm_{i2} $= Wm_{i1} \times \dfrac{Wp_2}{Wp_1}$

$= 16.28 \times \dfrac{0.95}{8.21}$

$=$ 1.88 kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (16.28 - 1.88) \times 4518 \times 0.05$

$=$ \$ 3,253 /yr

Electric Motor Drive Performance

Worksheet 13-E1

Company: WORKED EXAMPLE #2 Date: 86/06/13

Location: PUMPS - RETROFIT By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	38.6	amps
Measured voltage, V	570	volts
Measured current, I	34.0	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	(N/A)	kW
Rated full load power factor, $p.f._r$	0.85	(decimal)
Measured power factor, p.f.	0.80	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{34.0 \times 575 \times 0.85}{38.6 \times 570 \times 0.80}$$

$$= 0.94$$

Motor efficiency, Ef_m (Figure E-1) 0.92 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{570 \times 34.0 \times 1.73 \times 0.80}{1000}$$

$$= 26.82 \text{ kW}$$

Motor shaft power output, Wm_o $= Wm_i \times Ef_m$

$$= 26.82 \times 0.92$$

$$= 24.67 \text{ kW}$$

Install More Efficient Pump

Worksheet 13-P4

Company: Worked Example #2 Date: 86/06/13

Location: Pumps - Retrofit By: MBE

Data:

Liquid flow rate, f_w	90.7	m^3
Total pump head, Hd_T	49	m
Operation time, h	8760	h
Unit electrical energy cost, Ce	$ 0.05	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	26.82	/kW
Revised power input to the pump shaft, Wp_{i2} (Manufacturer's capacity curves)	15.73	kW
Drive efficiency, Ef_d (1.0 for direct drive, Table 1 for belt drive)	0.96	(decimal)
Motor efficiency, Ef_m	0.93	(decimal)

Revised electric motor power input, $Wm_{i2} = \dfrac{Wp_{i2}}{Ef_m \times Ef_d}$

$$= \frac{15.73}{0.93 \times 0.96}$$

$$= 17.62 \text{ kW}$$

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$$= (26.82 - 17.62) \times 8760 \times 0.05$$

$$= \$\,4{,}030 \text{ /yr}$$

Electric Motor Drive Performance

Worksheet 13-E1

Company: WORKED EXAMPLE #3 Date: 86/06/13

Location: PUMPS - RETROFIT By: MBE

Motor data: (nameplate or measured)

Rated voltage, V_r	575	volts
Rated current, I_r	8.0	amps
Measured voltage, V	575	volts
Measured current, I	5.03	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate motor shaft power output	(N/A)	kW
Rated full load power factor, $p.f._r$	0.88	(decimal)
Measured power factor, p.f.	0.60	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{5.03 \times 575 \times 0.60}{8.0 \times 575 \times 0.88}$$

$$= 0.43$$

Motor efficiency, Ef_m (Figure E-1) 0.87 (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{575 \times 5.03 \times 1.73 \times 0.60}{1000}$$

$$= 3.00 \text{ kW}$$

Motor shaft power output, Wm_o $= Wm_i \times Ef_m$

$= \_\_\_\_ \times \_\_\_\_$

$=$ N/A kW

Motor Replacement
Worksheet 13-E2

Company: WORKED EXAMPLE #3 Date: 86/06/13

Location: PUMPS - RETROFIT By: MBE

Initial motor power input, Wm_{i1} (Worksheet 13-E1)	3.00	kW	(1)
Required motor shaft power output, Wm_o (Worksheet 13-E1)	2.25	kW	(2)
Replacement motor rated shaft power output	2.25	kW	(3)
Operation time, h	4160	h/yr	(4)
Unit electrical energy cost, Ce	$ 0.05	/kWh	(5)

Load ratio (Worksheet E-1) $= \frac{(2)}{(3)}$

$= \frac{2.25}{2.25}$

$=$ 1.0

Replacement motor efficiency, Ef_m 0.92 (decimal)

Replacement motor power input, $Wm_{i2} = \frac{(2)}{Ef_m}$

$= \frac{2.25}{0.92}$

$=$ 2.45 kW (6)

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (3.00 - 2.45) \times 4160 \times 0.05$

$= \$$ 114 /yr

APPENDICES

Glossary

Absolute Pressure — Any pressure where the base for measurement is full vacuum. Expressed as kPa(absolute).

Ambient Temperature — The temperature of the medium surrounding a device.

Atmospheric Pressure — The standard absolute pressure of the atmosphere at sea level, being 101.325 kPa(absolute).

Centrifugal Force — The force that tends to impel objects outward from a center of rotation.

Friction Head — The equivalent head (or pressure) required to overcome the friction caused by the flow through a piping system.

Gauge Pressure — Any pressure where the base for measurement is atmospheric pressure. Expressed as kPa(gauge). Note that kPa(gauge) + atmospheric pressure = kPa(absolute).

Inertia — The property of a material that causes it to stay at rest or in uniform motion until acted upon by some external force.

Kinetic Energy — Energy of motion, or the amount of work that can be derived from a moving object by bringing it to rest.

Loaded — When a device or system is operating at maximum or full load capacity.

Pitch Diameter — The diameter that corresponds to the approximate midway point in the groove of a V belt sheave where the neutral axis of the belt runs.

Power — The rate at which energy is expended. Expressed in kilowatts.

Power Factor — The ratio of power passing through a circuit to the product of the voltage and current.

Specific Gravity — The ratio of the weight of a material to the weight of an equal volume of water.

Static Discharge Head — The total system head on the discharge side of a pump with zero flow.

Static Suction Head — The total system head on the suction side of a pump with zero flow (can be positive or negative).

Total Dynamic Head (TDH) — The sum of the total static head and the total friction head acting on a pump.

Total Fan Differential Pressure — The rise of pressure from fan inlet to fan outlet.

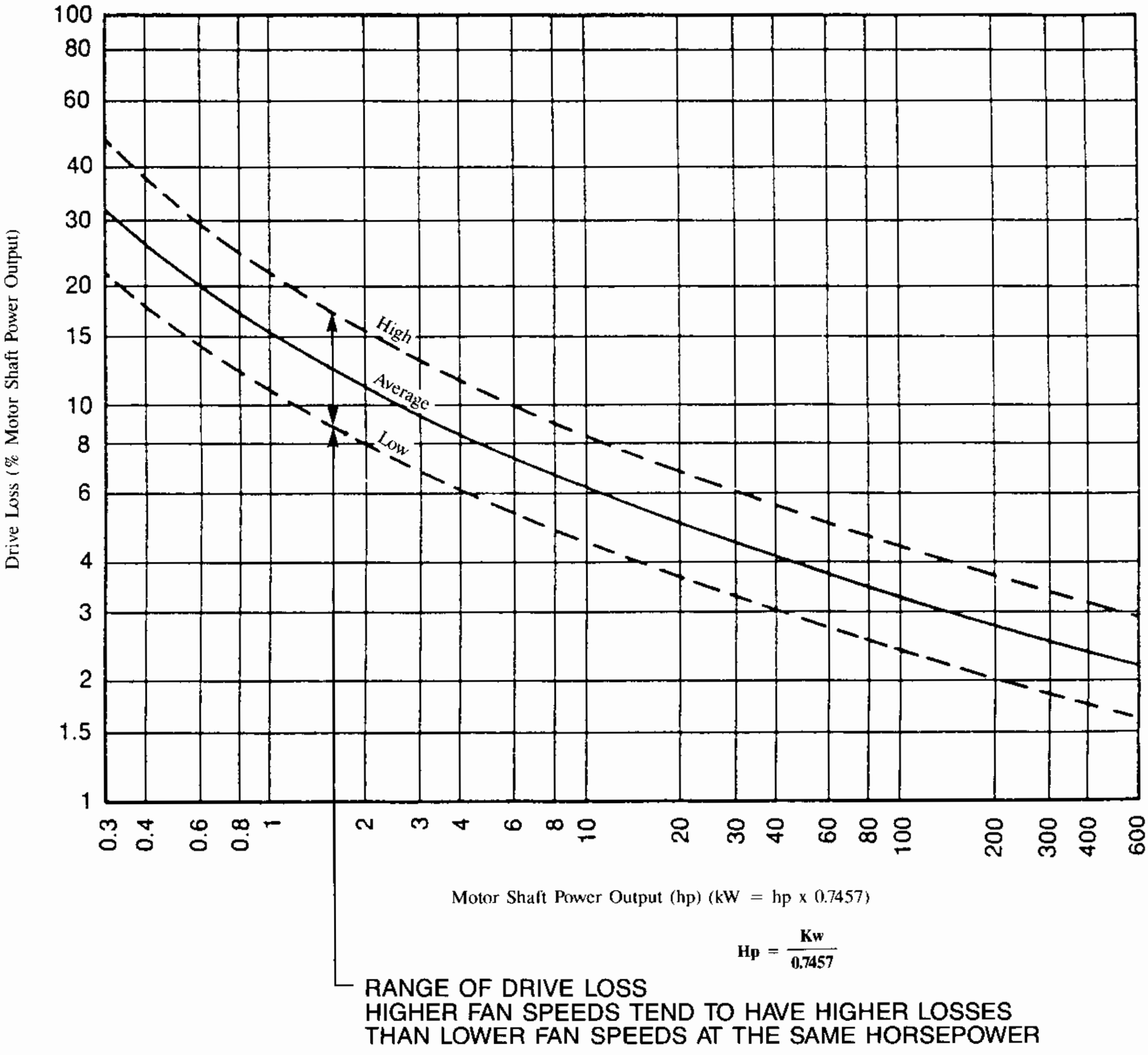
Total Static Head — The algebraic sum of the static suction and static discharge heads acting on a pump.

Total Suction Head — The sum of the static suction head and the friction head in the suctioni piping.

Vapor Pressure — The pressure at which a liquid begins to form vapor at a particular temperature.

Velocity Pressure — The total pressure measured in the direction of flow less the static pressure.

ESTIMATED BELT DRIVE LOSS
TABLE 1



$$Hp = \frac{Kw}{0.7457}$$

CENTRIFUGAL FAN PERFORMANCE CHARACTERISTICS
TABLE 2

Fan Type	Maximum Flow (1000 L/s)	Maximum Pressure (Pa)	Maximum Power (kW)	Efficiency Range (%)	Uses
Forward Curved	100	750	11	72-76	Washroom exhausters Building ventilators Building exhausters Boiler forced draft
Airfoil	425	3500	2240	84-91	Ventilating Air supply
Flat Radial Blade	70	5000	450	70-72	Flue gas recirculation Hot primary air
Modified Radial Blade	70	4000	450	78-83	Boiler induced draft Handling gas streams with moderate dust loading Sawdust and chips Grain dust
Open Radial Blade	70	4000	450	65-70	Long shavings Rags and wool Paper shreds and stringy material
Backward Inclined Backward Curved	175	2200	450	77-80	Commercial and industrial Ventilation Ventilating, air conditioning and heating Boiler forced draft

AXIAL FAN PERFORMANCE CHARACTERISTICS
TABLE 3

Fan Type	Maximum Flow (1000 L/s)	Maximum Differential Pressure (Pa)	Maximum Power (kW)	Uses
Propeller	57	300	15	Ventilation in factories, power plants, and agricultural buildings
				Personnel cooling fans or as a low cost ventilator
Tubeaxial	47	500	60	Transferring large volumes of air at low differential pressure
				Exhausting spray booths
Vaneaxial	118	5500	112	Mine ventilation
				Tunnel ventilation
				Fume exhaust

VAPOR PRESSURE OF WATER
TABLE 4

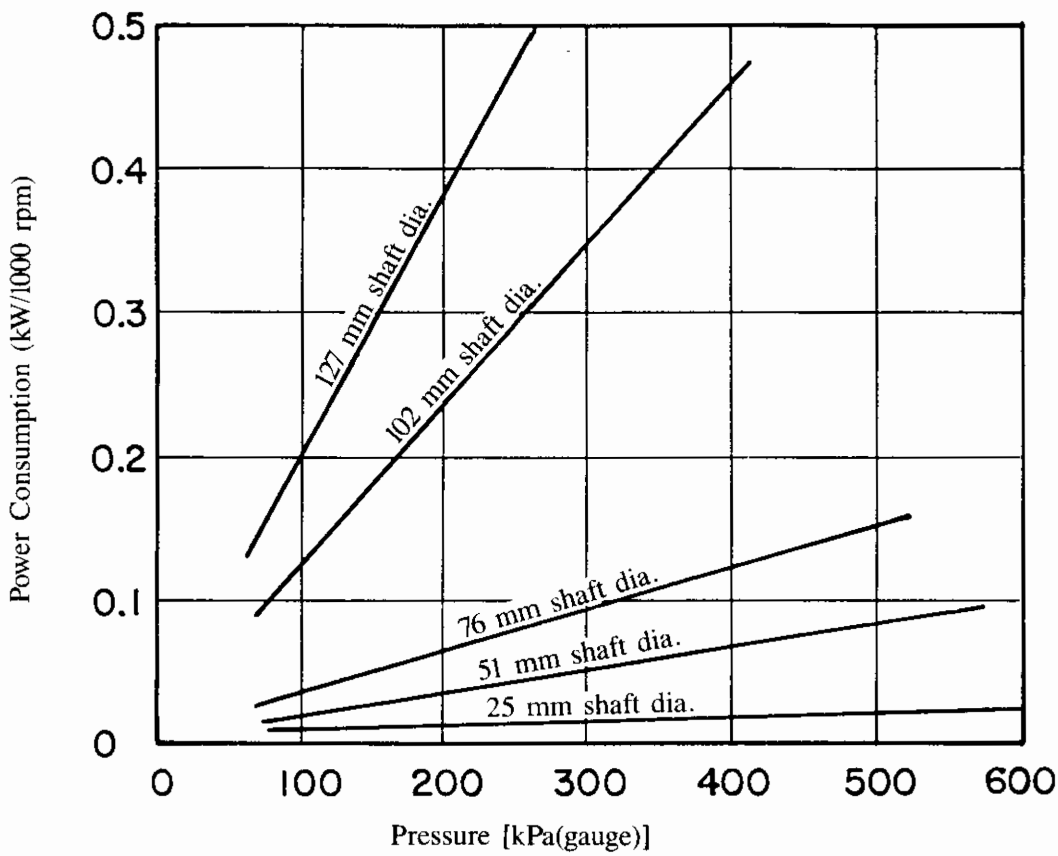
Temperature (°C)	Vapor Pressure [kPa(absolute)]	Temperature (°C)	Vapor Pressure [kPa(absolute)]
0	0.61112	110	143.390
10	1.2280	120	198.688
20	2.3389	130	270.306
30	4.2462	140	361.572
40	7.3838	150	476.207
50	12.3503	160	618.283
60	19.944	170	792.245
70	31.199	180	1002.899
80	47.414	190	1255.367
90	70.182	200	1555.099
100	101.420		

(Courtesy of ASHRAE)

POWER CONSUMPTION BY MECHANICAL SEALS

(Derived from data by A. W. Chesterton Company based on seal lubrication with water.)

TABLE 5



NORMAL CAPACITY OF COMMON TYPES OF CENTRIFUGAL PUMPS
TABLE 6

Pump Type	Flow (m^3/h)	Pump Head (m)
Volute	2700	150
Diffuser	3400	430
Regenerative Turbine	1700	370
Vertical Turbine	5700	300
Axial Flow	9000	15

NORMAL CAPACITY OF COMMON TYPES OF ROTARY PUMPS
TABLE 7

Pump Type	Flow (L/s)	Differential Pressure [kPa(gauge)]
External gear	315	27 600
Internal gear	50	27 600
Vane	65	17 200
Single screw	45	6 900
Multiscrew	1900	34 500

NORMAL CAPACITY OF COMMON TYPES OF RECIPROCATING PUMPS
TABLE 8

Pump Type	Flow (L/s)	Differential Pressure [kPa(gauge)]
Direct acting	80	5 200
Power reciprocating	250	100 000
Diaphragm	20	660

COMMON CONVERSIONS

1 barrel (35 Imp gal) (42 US gal)	= 159.1 litres	1 kilowatt-hour	= 3600 kilojoules
1 gallon (Imp)	= 1.20094 gallon (US)	1 Newton	= 1 kg-m/s^2
1 horsepower (boiler)	= 9809.6 watts	1 therm	= 10^5 Btu
1 horsepower	= 2545 Btu/hour	1 ton (refrigerant)	= 12002.84 Btu/hour
1 horsepower	= 0.746 kilowatts	1 ton (refrigerant)	= 3516.8 watts
1 joule	= 1 N-m	1 watt	= 1 joule/second
Kelvin	= (°C + 273.15)	Rankine	= (°F + 459.67)

Cubes

$1\ yd^3 = 27\ ft^3$

$1\ ft^3 = 1728\ in^3$

$1\ cm^3 = 1000\ mm^3$

$1\ m^3 = 10^6\ cm^3$

$1\ m^3 = 1000\ L$

Squares

$1\ yd^2 = 9\ ft^2$

$1\ ft^2 = 144\ in^2$

$1\ cm^2 = 100\ mm^2$

$1\ m^2 = 10000\ cm^2$

SI PREFIXES

Prefix	Symbol	Magnitude	Factor
tera	T	1 000 000 000 000	10^{12}
giga	G	1 000 000 000	10^9
mega	M	1 000 000	10^6
kilo	k	1 000	10^3
hecto	h	100	10^2
deca	da	10	10^1
deci	d	0.1	10^{-1}
centi	c	0.01	10^{-2}
milli	m	0.001	10^{-3}
micro	u	0.000 001	10^{-6}
nano	n	0.000 000 001	10^{-9}
pica	p	0.000 000 000 001	10^{-12}

4138T/T33(-6)

UNIT CONVERSION TABLES
METRIC TO IMPERIAL

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
amperes/square centimetre	A/cm^2	amperes/square inch	A/in^2	6.452
Celsius	°C	Fahrenheit	°F	(°C × 9/5) + 32
centimetres	cm	inches	in	0.3937
cubic centimetres	cm^3	cubic inches	in^3	0.06102
cubic metres	m^3	cubic foot	ft^3	35.314
grams	g	ounces	oz	0.03527
grams	g	pounds	lb	0.0022
grams/litre	g/L	pounds/cubic foot	lb/ft^3	0.06243
joules	J	Btu	Btu	9.480×10^{-4}
joules	J	foot-pounds	ft-lb	0.7376
joules	J	horsepower-hours	hp-h	3.73×10^{-7}
joules/metre, (Newtons)	J/m, N	pounds	lb	0.2248
kilograms	kg	pounds	lb	2.205
kilograms	kg	tons (long)	ton	9.842×10^{-4}
kilograms	kg	tons (short)	tn	1.102×10^{-3}
kilometres	km	miles (statute)	mi	0.6214
kilopascals	kPa	atmospheres	atm	9.87×10^{-3}
kilopascals	kPa	inches of mercury (@ 32°F)	in Hg	0.2953
kilopascals	kPa	inches of water (@ 4°C)	in H_2O	4.0147
kilopascals	kPa	pounds/square inch	psi	0.1450
kilowatts	kW	foot-pounds/second	ft-lb/s	737.6
kilowatts	kW	horsepower	hp	1.341
kilowatt-hours	kWh	Btu	Btu	3413
litres	L	cubic foot	ft^3	0.03531
litres	L	gallons (Imp)	gal (Imp)	0.21998
litres	L	gallons (US)	gal (US)	0.2642
litres/second	L/s	cubic foot/minute	cfm	2.1186
lumen/square metre	lm/m^2	lumen/square foot	lm/ft^2	0.09290
lux, lumen/square metre	lx, lm/m^2	footcandles	fc	0.09290
metres	m	foot	ft	3.281
metres	m	yard	yd	1.09361
parts per million	ppm	grains/gallon (Imp)	gr/gal (Imp)	0.07
parts per million	ppm	grains/gallon (US)	gr/gal (US)	0.05842
permeance (metric)	PERM	permeance (Imp)	perm	0.01748
square centimetres	cm^2	square inches	in^2	0.1550
square metres	m^2	square foot	ft^2	10.764
square metres	m^2	square yards	yd^2	1.196
tonne (metric)	t	pounds	lb	2204.6
watt	W	Btu/hour	Btu/h	3.413
watt	W	lumen	lm	668.45

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UNIT CONVERSION TABLES
IMPERIAL TO METRIC

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
ampere/in^2	A/in^2	ampere/cm^2	A/cm^2	0.1550
atmospheres	atm	kilopascals	kPa	101.325
British Thermal Unit	Btu	joules	J	1054.8
Btu	Btu	kilogram-metre	kg-m	107.56
Btu	Btu	kilowatt-hour	kWh	2.928×10^{-4}
Btu/hour	Btu/h	watt	W	0.2931
calorie, gram	cal or g-cal	joules	J	4.186
chain	chain	metre	m	20.11684
cubic foot	ft^3	cubic metre	m^3	0.02832
cubic foot	ft^3	litre	L	28.32
cubic foot/minute	cfm	litre/second	L/s	0.47195
cycle/second	c/s	Hertz	Hz	1.00
Fahrenheit	°F	Celsius	°C	(°F-32)/1.8
foot	ft	metre	m	0.3048
footcandle	fc	lux, lumen/ square metre	lx, lm/m^2	10.764
footlambert	fL	candela/square metre	cd/m^2	3.42626
foot-pounds	ft-lb	joule	J	1.356
foot-pounds	ft-lb	kilogram-metres	kg-m	0.1383
foot-pounds/second	ft-lb/s	kilowatt	kW	1.356×10^{-3}
gallons (Imp)	gal (Imp)	litres	L	4.546
gallons (US)	gal (US)	litres	L	3.785
grains/gallon (Imp)	gr/gal (Imp)	parts per million	ppm	14.286
grains/gallon (US)	gr/gal (US)	parts per million	ppm	17.118
horsepower	hp	watts	W	745.7
horsepower-hours	hp-h	joules	J	2.684×10^6
inches	in	centimetres	cm	2.540
inches of Mercury (@ 32°F)	in Hg	kilopascals	kPa	3.386
inches of water (@ 4°C)	in H_2O	kilopascals	kPa	0.2491

UNIT CONVERSION TABLES
IMPERIAL TO METRIC (cont'd)

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
lamberts	* L	candela/square metre	cd/m^2	3.183
lumen/square foot	lm/ft^2	lumen/square metre	lm/m^2	10.76
lumen	lm	watt	W	0.001496
miles (statute)	mi	kilometres	km	1.6093
ounces	oz	grams	g	28.35
perm (at 0°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.721×10^{-11}
perm (at 23°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.745×10^{-11}
perm-inch (at 0°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4532×10^{-12}
perm-inch (at 23°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4593×10^{-12}
pint (Imp)	pt	litre	L	0.56826
pounds	lb	grams	g	453.5924
pounds	lb	joules/metre, (Newtons)	J/m, N	4.448
pounds	lb	kilograms	kg	0.4536
pounds	lb	tonne (metric)	t	4.536×10^{-4}
pounds/cubic foot	lb/ft^3	grams/litre	g/L	16.02
pounds/square inch	psi	kilopascals	kPa	6.89476
quarts	qt	litres	L	1.1365
slug	slug	kilograms	kg	14.5939
square foot	ft^2	square metre	m^2	0.09290
square inches	in^2	square centimetres	cm^2	6.452
square yards	yd^2	square metres	m^2	0.83613
tons (long)	ton	kilograms	kg	1016
tons (short)	tn	kilograms	kg	907.185
yards	yd	metres	m	0.9144

* "L" as used in Lighting

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The following typical values for conversion factors may be used when actual data are unavailable. The MJ and Btu equivalencies are heats of combustion. Hydrocarbons are shown at the higher heating value, wet basis. Some items listed are typically feedstocks, but are included for completeness and as a reference source. The conversion factors for coal are approximate since the heating value of a specific coal is dependent on the particular mine from which it is obtained.

ENERGY TYPE	METRIC	IMPERIAL
COAL		
— metallurgical	29,000 megajoules/tonne	25.0×10^6 Btu/ton
— anthracite	30,000 megajoules/tonne	25.8×10^6 Btu/ton
— bituminous	32,100 megajoules/tonne	27.6×10^6 Btu/ton
— sub-bituminous	22,100 megajoules/tonne	19.0×10^6 Btu/ton
— lignite	16,700 megajoules/tonne	14.4×10^6 Btu/ton
COKE		
— metallurgical	30,200 megajoules/tonne	26.0×10^6 Btu/ton
— petroleum		
— raw	23,300 megajoules/tonne	20.0×10^6 Btu/ton
— calcined	32,600 megajoules/tonne	28.0×10^6 Btu/ton
PITCH	37,200 megajoules/tonne	32.0×10^6 Btu/ton
CRUDE OIL	38,5 megajoules/litre	5.8×10^6 Btu/bbl
No. 2 OIL	38.68 megajoules/litre	5.88×10^6 Btu/bbl
		$.168 \times 10^6$ Btu/IG
No. 4 OIL	40.1 megajoules/litre	6.04×10^6 Btu/bbl
		$.173 \times 10^6$ Btu/IG
No. 6 OIL (RESID. BUNKER C)		
@ 2.5% sulphur	42.3 megajoules/litre	6.38×10^6 Btu/bbl
		$.182 \times 10^6$ Btu/IG
@ 1.0% sulphur	40.5 megajoules/litre	6.11×10^6 Btu/bbl
		$.174 \times 10^6$ Btu/IG
@ .5% sulphur	40.2 megajoules/litre	6.05×10^6 Btu/bbl
		$.173 \times 10^6$ Btu/IG
KEROSENE	37.68 megajoules/litre	$.167 \times 10^6$ Btu/IG
DIESEL FUEL	38.68 megajoules/litre	$.172 \times 10^6$ Btu/IG
GASOLINE	36.2 megajoules/litre	$.156 \times 10^6$ Btu/IG
NATURAL GAS	37.2 megajoules/m^3	1.00×10^6 Btu/MCF
PROPANE	50.3 megajoules/kg	$.02165 \times 10^6$ Btu/lb
	26.6 megajoules/litre	$.1145 \times 10^6$ Btu/IG
ELECTRICITY	3.6 megajoules/kWh	$.003413 \times 10^6$ Btu/kWh

Fan Speed Reduction — Sheave Replacement

Worksheet 13-F1

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data:

Initial volume air flow rate, f_{a1}	________	L/s
Revised volume air flow rate, f_{a2}	________	L/s
Initial total differential pressure, DP_{T1}	________	kPa
Revised total differential pressure, DP_{T2}	________	kPa
Initial fan speed, n_1	________	rpm
Existing motor sheave pitch diameter, D_{m1}	________	mm
Existing fan sheave pitch diameter, D_{f1}	________	mm
Operation time, h	________	h/yr
Unit electrical energy cost, Ce	$________	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	________	kW

Required fan speed:

Flow reduction:

Revised fan speed,

$$n_2 = n_1 \times \frac{f_{a2}}{f_{a1}}$$

$$= \quad \times \_\_\_\_$$

$$= \_\_\_\_\_\_ \text{ rpm}$$

Pressure reduction:

Revised fan speed,

$$n_2 = n_1 \times \left(\frac{DP_{T2}}{DP_{T1}}\right)^{0.5}$$

$$= \quad \times \left(\frac{}{}\right)^{0.5}$$

$$= \_\_\_\_\_\_ \text{ rpm}$$

Fan Speed Reduction — Sheave Replacement

Worksheet 13-F1
(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Revised sheave size required:

New motor sheave:

New sheave pitch diameter, $D_{m2} = D_{m1} \times \frac{n_2}{n_1}$

$= \quad\quad \times$ ________

$=$ ________ mm

New fan sheave:

New sheave pitch diameter, $D_{f2} = D_{f1} \times \frac{n_1}{n_2}$

$= \quad\quad \times$ ________

$=$ ________ mm

Initial ideal fan power, Wf_1 $= \dfrac{f_{a1} \times DP_{T1}}{1000}$

$= \dfrac{\quad\quad \times \quad\quad}{1000}$

$=$ ______________ kW

Revised ideal fan power, Wf_2 $= \dfrac{f_{a2} \times DP_{T2}}{1000}$

$= \dfrac{\quad\quad \times \quad\quad}{1000}$

$=$ ______________ kW

Revised motor power input, Wm_{i2} $= Wm_{i1} \times \dfrac{Wf_2}{Wf_1}$

$= \quad\quad \times$ ________

$=$ ______________ kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (\quad\quad - \quad\quad) \times \quad\quad \times$

$=$ \$ ______________ /yr

Fan Pressures
Worksheet 13-F2

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Static pressure at fan inlet, P_{Si} ______________ kPa(gauge)

Static pressure at fan outlet, P_{So} ______________ kPa(gauge)

Velocity Pressure at fan inlet, P_{Vi} ______________ kPa

Velocity Pressure at fan outlet, P_{Vo} ______________ kPa

Total fan static differential pressure, $DP_S = P_{So} - P_{Si} - P_{Vi}$

$= \quad - \quad -$

$=$ ______________ kPa

Total fan differential pressure, $DP_T = P_{So} + P_{Vo} - P_{Si} - P_{Vi}$

$= \quad + \quad - \quad -$

$=$ ______________ kPa

Note: Readings taken at fan inlet and outlet

Optimum Length of Fan Outlet Duct

Worksheet 13-F3

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Duct Dimensions:

Rectangular: or Round:

Height, a __________ m Diameter __________ m

Width, b __________ m

Equivalent diameter, D $= (1.273 \times a \times b)^{0.50}$

$= (1.273 \quad \times \qquad \times \qquad)^{0.50}$

$=$ __________ m

Volume air flow rate, f_a __________ L/s

Equivalent duct area, $A_d = \dfrac{\pi \times D^2}{4}$

$= \dfrac{3.1416 \times (\qquad\qquad)^2}{4}$

$=$ __________ m^2

Average air velocity, vel. $= \dfrac{f_a}{A_d \times 1000}$

$\dfrac{\qquad\qquad}{\qquad \times 1000}$

$=$ __________ m/s

Optimum Duct Length, L:

Vel. less than 12.7 m/s or Vel. greater than 12.7 m/s

$L = 2.5 \times D$

$= 2.5 \times$ __________

$=$ __________ m

$L = (2.5 \times D) + \left[\dfrac{(vel. \quad - 12.7)}{5} \times D\right]$

$= (2.5 \times \qquad) + \left[\dfrac{(\qquad - 12.7)}{5} \times \qquad\right]$

$=$ __________ m

Fan Replacement / Improved Fan Inlet and Outlet

Worksheet 13-F4

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Initial motor shaft power input, Wm_{i1} (Worksheet 13-E1) ______________ kW

Initial motor shaft power output, Wm_{o1} (Worksheet 13-E1) ______________ kW

Drive loss (Table 1) ______________ %

Revised power input to the fan shaft, Wf_{i2} (Manufacturer) ______________ kW

Operation time, h ______________ h/yr

Unit electrical energy cost, Ce $ ______________ /kWh

Drive efficiency, $Ef_d = 1 - \frac{\text{Drive loss}}{100}$

$= 1 - \frac{\_\_\_\_\_\_}{100}$

$=$ ______________

Initial power input to the fan shaft, $Wf_{i1} = Wm_{o1} \quad x \quad Ef_d$

$=$ x

$=$ ______________ kW

Revised motor power input, $Wm_{i2} = \frac{Wm_{i1} \text{ x } Wf_{i2}}{Wf_{i1}}$

$=$ ______ x ______

$=$ ______________ kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \quad x \quad h \quad x \quad Ce$

$= (\quad - \quad) \quad x \quad x$

$= \$$______________/yr

Fan Speed Reduction — Variable Speed Motor

Worksheet 13-F5

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data

Initial volume air flow rate, f_{a1}	______________	L/s
Revised volume air flow rate, f_{a2}	______________	L/s
Initial total differential pressure, DP_{T1}	______________	kPa
Revised total differential pressure, DP_{T2}	______________	kPa
Total operation time at reduced speed, h	______________	h/yr
Unit electrical energy cost, Ce	$ ______________	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	______________	kW

Initial ideal fan power, Wf_1 $= \dfrac{f_{a1} \times DP_{T1}}{1000}$

$= \dfrac{\quad \times \quad}{1000}$

$=$ ______________ kW

Revised ideal fan power, Wf_2 $= \dfrac{f_{a2} \times DP_{T2}}{1000}$

$= \dfrac{\quad \times \quad}{1000}$

$=$ ______________ kW

Revised motor power input, Wm_{i2} $= Wm_{i1} \times \dfrac{Wf_2}{Wf_1}$

$= \quad \times$ ______________

$=$ ______________ kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (\quad - \quad) \times \quad \times$

$= \$$ ______________ /yr

Fan Energy Consumption
Worksheet 13-F6

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data:

Power input to fan shaft, Wf_i (Manufacturer's rating table) ______________ kW

Rated motor shaft power output, Wm_o ______________ kW

Rated motor efficiency, Ef_m ______________ (decimal)

Drive loss (use 0 for direct drive, Table 1 for belt drive) ______________ %

Operation time, h ______________ h/yr

Unit electrical energy cost, Ce $______________ /kWh

$$\text{Drive efficiency, } Ef_d = 1 - \frac{\text{Drive loss}}{100}$$

$$= 1 - \frac{\_\_\_\_\_\_}{100}$$

$$= \_\_\_\_\_\_\_\_\_\_$$

$$\text{Motor power input } Wm_i = \frac{Wf_i}{Ef_m \times Ef_d}$$

$$= \frac{\_\_\_\_\_\_\_\_\_\_\_\_}{\quad \times \quad}$$

$$= \_\_\_\_\_\_\_\_\_\_\_\_ \text{ kW}$$

$$\text{Annual energy cost} = Wm_i \times Ce \times h$$

$$= \quad \times \quad \times$$

$$= \$\_\_\_\_\_\_\_\_\_\_ \text{/yr}$$

Pump Capacity Reduction

Worksheet 13-P1

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data

Initial liquid flow rate, f_{w1} ______________ m^3/h

Revised liquid flow rate, f_{w2} ______________ m^3/h

Initial total pump head, Hd_{T1} ______________ m

Revised total pump head, Hd_{T2} ______________ m

Total operation time, h ______________ h/yr

Unit electrical energy cost, Ce $ ______________ /kWh

Initial motor power input, Wm_{i1} (Worksheet 13-E1) ______________ kW

Initial ideal pump power, Wp_1 $= \frac{f_{w1} \times Hd_{T1}}{367}$

$= \frac{\quad \times \quad}{367}$

$=$ ______________ kW

Revised ideal pump power, Wp_2 $= \frac{f_{w2} \times Hd_{T2}}{367}$

$= \frac{\quad \times \quad}{367}$

$=$ ______________ kW

Revised motor power input, Wm_{i2} $= Wm_{i1} \times \frac{Wp_2}{Wp_1}$

$= \quad \times$ ________

$=$ __________ kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (\quad - \quad) \times \quad \times$

$=$ $ ______________ /yr

Install Mechanical Seals

Worksheet 13-P2

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Operation time, h ______________ h/yr

Unit electrical energy cost, Ce $______________ /kWh

Shaft diameter ______________ mm

Shaft speed, n ______________ rpm

Pump discharge head, Hd_d ______________ m

Pump discharge pressure, P = Hd_d x 9.81

= ______ x 9.81 ______

= ______________ kPa

Mechanical seal power consumption/1000 rpm, W_{S1} ______________ kW/1000 rpm
(Table 5)

Mechanical seal power consumption, $W_{S2} = \frac{n}{1000} \times W_{S1}$

$= \frac{\quad}{1000} \times$

= ______________ kW

Packing gland power consumption, W_{S3} = W_{S2} x 6

= ______ x ______

= ______________ kW

Annual energy cost saving = (W_{S3} − W_{S2}) x h x Ce

= (−   ) x   x

= $______________/yr

Pump Capacity Reduction — Impeller Size

Worksheet 13-P3

(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data:

Initial liquid flow rate, f_{w1}	________	m^3/h
Revised liquid flow rate, f_{w2}	________	m^3/h
Initial total pump head, Hd_{T1}	________	m
Existing Impeller diameter, D_1	________	mm
Operation time, h	________	h/yr
Unit electrical energy cost, Ce	$________	/kWh
Initial motor power input, Wm_{i1} (Worksheet 13-E1)	________	kW
Initial motor shaft power output, Wm_{o1} (Worksheet 13-E1)	________	kW
Drive efficiency, Ef_d (1.0 for direct drive, Table 1 for belt drive)	________	

Revised pump head, $Hd_{T2} = Hd_{T1} \times \left(\frac{f_{w2}}{f_{w1}}\right)^2$

$$= \quad \times \left(\frac{\quad}{\quad}\right)^2$$

$$= \underline{\hspace{3cm}} \text{ m}$$

Initial power input to the pump shaft, $Wp_{i1} = Wm_{o1} \times Ef_d$

$$= \underline{\hspace{1.5cm} \times \hspace{1.5cm}}$$

$$= \underline{\hspace{3cm}} \text{ kW}$$

Pump Capacity Reduction — Impeller Size

Worksheet 13-P3

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Revised impeller diameter, $D_2 = D_1 \times \left(\frac{Hd_{T2}}{Hd_{T1}}\right)^{0.5}$

$= \quad \times \left(\frac{\quad}{\quad}\right)^{0.5}$

$=$ ______________ mm

Revised power input to the pump shaft, $Wp_{i2} = Wp_{i1} \times \left(\frac{D_2}{D_1}\right)^3$

$= \quad \times \left(\frac{\quad}{\quad}\right)^3$

$=$ ______________ kW

Revised motor power input, $Wm_{i2} = Wm_{i1} \times \frac{Wp_{i2}}{Wp_{i1}}$

$= \quad \times \frac{\quad}{\quad}$

$=$ ______________ kW (4)

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (\quad - \quad) \times \quad \times$

$=$ \$______________ /yr

Install More Efficient Pump

Worksheet 13-P4

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data:

Liquid flow rate, f_w ______________ m^3

Total pump head, Hd_T ______________ m

Operation time, h ______________ h

Unit electrical energy cost, Ce $ ______________ /kWh

Initial motor power input, Wm_{i1} (Worksheet 13-E1) ______________ /kW

Revised power input to the pump shaft, Wp_{i2} (Manufacturer's capacity curves) ______________ kW

Drive efficiency, Ef_d ______________ (decimal)

(1.0 for direct drive,
Table 1 for belt drive)

Motor efficiency, Ef_m ______________ (decimal)

Revised electric motor power input, $Wm_{i2} = \dfrac{Wp_{i2}}{Ef_m \times Ef_d}$

$= \dfrac{\qquad\qquad}{\qquad x \qquad}$

= ______________ kW

Annual energy cost saving $= (Wm_{i1} - Wm_{i2}) \times h \times Ce$

= (–   )   x   x

= $__________ /yr

Electric Motor Drive Performance
Worksheet 13-E1

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Motor data: (nameplate or measured)

Rated voltage, V_r	________	volts
Rated current, I_r	________	amps
Measured voltage, V	________	volts
Measured current, I	________	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	________	
Nameplate motor shaft power output	________	kW
Rated full load power factor, $p.f._r$	________	(decimal)
Measured power factor, p.f.	________	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{\_\_\_\_ \times \_\_\_\_ \times \_\_\_\_}{\_\_\_\_ \times \_\_\_\_ \times \_\_\_\_}$$

$$= \_\_\_\_\_\_\_\_$$

Motor efficiency, Ef_m (Figure E-1) ________ (decimal)

$$\text{Electric motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{\_\_\_\_ \times \_\_\_\_ \times \_\_\_\_ \times \_\_\_\_}{1000}$$

$$= \_\_\_\_\_\_\_\_ \text{ kW}$$

$$\text{Motor shaft power output, } Wm_o = Wm_i \times Ef_m$$

$$= \_\_\_\_ \times \_\_\_\_$$

$$= \_\_\_\_\_\_\_\_ \text{ kW}$$

Motor Replacement

Worksheet 13-E2

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Initial motor power input, Wm_{i1} (Worksheet 13-E1) ______________ kW (1)

Required motor shaft power output, Wm_o (Worksheet 13-E1) ______________ kW (2)

Replacement motor rated shaft power output ______________ kW (3)

Operation time, h ______________ h/yr (4)

Unit electrical energy cost, Ce $______________ /kWh (5)

Load ratio (Worksheet E-1) $= \frac{(2)}{(3)}$

= ______________

= ______________

Replacement motor efficiency, Ef_m ______________ (decimal)

Replacement motor power input, $Wm_{i2} = \frac{(2)}{Ef_m}$

= ______________

= ______________ kW (6)

Annual energy cost saving = $(Wm_{i1} - Wm_{i2})$ x h x Ce

= (–   ) x   x

= $______________/yr

Electric Motor Drives

Most fans and pumps are driven by alternating current electric induction motors. Many small devices are driven by single phase motors, while the larger ones are driven by three phase motors. It is important to note that the number of phases affects electrical energy calculations.

Power Input to Motors

The power input to an electric motor can be expressed by the following equation.

$$Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

Where, Wm_i = motor power input (kW)

V = voltage at the motor (volts)

I = current draw (amps)

Y = phase factor (1.73 for 3 phase, 2.0 for four wire 2 phase, 1.0 for single phase)

p.f. = motor power factor (decimal)

1000 = conversion of volt-amps to kilowatts.

Power Output From Motors

The shaft power output of an electric motor is given by the following equation.

$$Wm_o = \frac{V \times I \times Y \times p.f. \times Ef_m}{1000}$$

Where, Wm_o = motor shaft power output (kW)

Ef_m = electric motor efficiency (decimal)

For all motors, the relationship between output and input is the same.

$Wm_o = Wmi \times Ef_m$

Obtaining Electrical Data

The data necessary to perform the calculations can be obtained by measurements and from electric motor performance data. Meters may be used to measure voltage, current and power factor.

Simplified performance data for power factor and motor efficiency is approximated by Figure E-1. Although the curves are representative of three phase induction motors, they may also be used for most other types of motors. These curves should only be used in the absence of accurate measured values.

The nameplate of a motor is a reliable source of performance data. The Electrical Equipment Manufacturers Association requires that the nameplate provide the power rating of the motor in horsepower (labelled as H.P. or hp) or in kW, the rated voltage, the full load current or amps (labelled as F.L. amps or F.L.A.), the power factor (labelled as p.f.) and the number of phases of the supply.

Load Ratio on a Motor

The *load ratio* on a motor is the ratio of measured power input to the rated full load power input, and can be expressed by the following equation.

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

Where, I = measured current draw (amps)

V = measured voltage (volts)

$p.f.$ = measured power factor

I_r = rated full load current (amps)

V_r = rated voltage (volts)

$p.f._r$ = rated full load power factor

Using the calculated load, the motor efficiency can be determined from Figure E-1. If the rated power factor is not known, a value from Figure E-1 can be used.

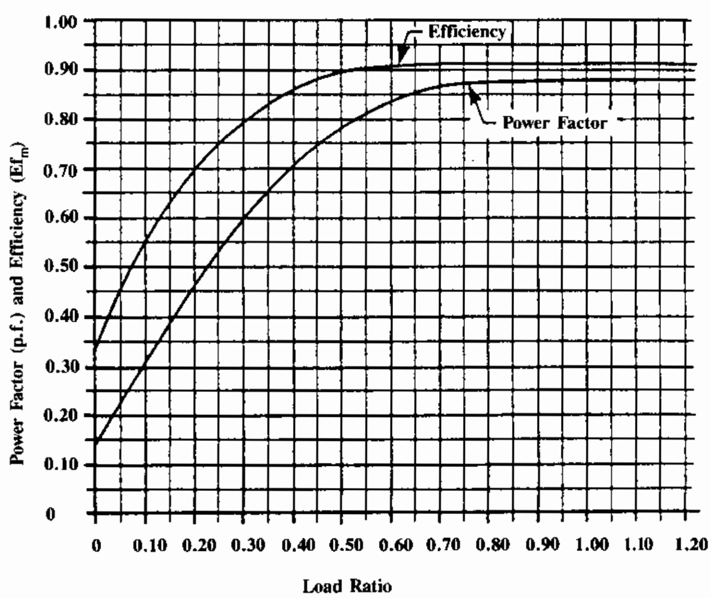
Cost of Electrical Energy

For the purposes of this module, the cost of electrical energy is expressed as $/kWh consumed. This value is the incremental cost including the consumption rate, demand charges and any surcharges or discounts. The rate will vary for different electric power customers and for different regions. *Users of this module should establish an energy rate closely representative of their actual cost of electrical energy.*

Worksheets

Worksheets 13-E1 and 13-E2 are provided to assist with the analysis of motor performance.

Electric Motor Drive Performance



Power Factor and Electric Motor Efficiency
Figure E-1