# ENGINEERING STANDARD

# FOR

# PROCESS DESIGN OF FANS AND BLOWERS

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## 0. INTRODUCTION

"Pressure Reducing/Increasing Machineries and/or Equipment" are broad and contain variable subjects of paramount importance. Therefore, a group of process engineering standards are prepared to cover the subject.

This group includes the following standards:

| STANDARD CODE | STANDARD TITLE |
|---|---|
| IPS-E-PR-745 | "Process Design of Vacuum Equipment (Vacuum Pumps & Steam Jet Ejectors)" |
| IPS-E-PR-750 | "Process Design of Compressors" |
| IPS-E-PR-755 | "Process Design of Fans and Blowers" |

This Standard covers:

### "Process Design of Fans and Blowers"

Basic process engineering calculation related to fans and blowers such as interpretation of performance curves, efficiency, power requirement, preparation of data sheets, etc. are presented.

As a general rule, all gas compressing equipment which produce less than 135 kPa(abs.) pressure at discharge, with an atmospheric (or slightly sub-atmospheric) suction pressure, fall into the category of "Fans and Blowers".

Process aspects of the blowers are discussed in IPS-E-PR-750. Therefore, this standard is dedicated to fans, in which, four most frequently used types, i.e., Centrifugal, Vaneaxial, Tubeaxial and propeller type are discussed.

## 1. SCOPE

This Recommended Practice is intended to provide guidelines for process engineers for selection of proper type and preparation of process data sheets for fans used in O.G and P industries.

It contains basic reference information, data and formulas necessary for fan selection as mentioned above.

## 2. REFERENCES

Throughout this Standard the following standards and codes are referred to. The editions of these standards and codes that are in effect at the time of publication of this Standard shall, to the extent specified herein, form a part of this Standard.

The applicability of changes in standards and codes that occur after the date of this Standard shall be mutually agreed upon by the Company and the Vendor/Consultant.

**API (AMERICAN PETROLEUM INSTITUTE)**

> Standard No. 673   "Special Purpose Centrifugal Fans for Refinery Service" 1st. Ed., Jan. 1982

**IPS (IRANIAN PETROLEUM STANDARDS)**

> IPS-E-GN-100,      "Units"
> IPS-E-PR-250,      "Performance Guarantee"
> IPS-M-PM-230,      "Forced Draft Fans for Boilers and Process Services"

## 3. DEFINITIONS AND TERMINOLOGY

Terms used in this standard are defined as follows:

**a) Fan impeller**

Is the assembly of the fan wheel and the hub(s). (API Std. 673, Section 1.4).

**b) Fan plane**

Is a flow area perpendicular to the flow of gas at the specified reference plane; that is, inlet flange or outlet flange. (API Std. 673, 1.4).

**c) Fan rated point**

Is defined as (1) the highest speed necessary to meet any specified operating condition and (2) the rated capacity required by fan designs to meet all operating points. (The Vendor shall select this capacity point to the best encompass specified operating conditions within the scope of the expected performance curve.) (API Std. 673, 1.4).

**d)** Maximum continuous speed (rotations per minute) Is the speed at least equal to the product of 105 percent and the highest speed required by any of the specified operating conditions. (Modification to API Std. 673, 1.4).

**e) Normal operating point**

Is the point at which usual operation. Is expected and optimum efficiency is desired. Unless otherwise specified, fan performance shall be guaranteed at the normal operating point. (API Std. 673, 1.4).

**f) The total pressure ($P_{tf}$) of a fan**

Is the rise of pressure from fan inlet to fan outlet as measured by two impact tubes, one in the fan inlet duct and one in the fan discharge duct, corrected for friction to the fan inlet and outlet respectively.

Where no inlet duct is used, total pressure on the inlet side is zero.

**g) The velocity pressure ($P_v$) of a fan**

Is the pressure corresponding to the average velocity determination from the volume of air flow at the fan outlet area.

**h) The static pressure ($P_s$) of the fan**

Is the total pressure ($P_{tf}$) diminished by the fan velocity pressure ($P_v$).

**i) Standard air density**

Is 1.2007 kg per cubic meter.

**j) The unit of pressure**

Is the mm. of water column of density of 997.423 kg per cubic meter and/or Pa (1 mm $H_2O$ conventional=9.80665 pascals).

**k) The volume handled by a fan**

Is the number of cubic meters of air per hour expressed at fan outlet conditions.

**l) The power output of a fan**

Is expressed in kilowatts and is based on fan volume and fan total pressure.

**m) The power input to a fan**

Is expressed in kilowatts and is the measured kilowatt delivered to the fan shaft.

**n) The mechanical efficiency of a fan**

Is the ratio of power output to the power input.

**o) The static efficiency of a fan**

Is the mechanical efficiency multiplied by the ratio of the static pressure to the total pressure or $e_s = e_t \times P_s / P_t$.

**p) The fan outlet area**

Is the inside area of the fan outlet.

**q) The fan inlet area**

Is the inside area of the fan inlet collar.

**r) Evase**

Is a diffuser or a diverging discharge transition piece.

## 4. SYMBOLS AND ABBREVIATIONS

BkW = Brake(shaft) kilowatt of Fan in (kilowatts, kW);
D = Wheel Diameter, in (m);
d = Relative Density, (dimensionless);
$e_s$ = Static Efficiency, in (fractions);
$e_t$ = Mechanical (Total) Efficiency in (fractions);
$F_1$ = Temperature Correction Factor, in (kg/m$^3$);
$F_2$ = Altitude Correction Factor, in (kg/m$^3$);
FkW = Fan Power, in (kilowatts);
GkW = Gas (Air) kilowatt of Fan, in (kilowatts, kW);
K = Ratio of Specific Heats, $C_p/C_v$, (dimensionless);
$P_1$ = Fan Inlet Pressure, in [mm $H_2O$ (abs.)], or (in [Pa(abs.)];
$P_s$ = Static Pressure of Fan, in [mm $H_2O$ (abs.)], or in [Pa(abs.)];
$P_{s2}$ = Fan Outlet Static Pressure, in [mm H2O (abs.)], or in [Pa(abs.)];
$P_t$ = Total Pressure in (mm $H_2O$), or in (Pa);
$P_{tf}$ = Fan Total Pressure in (mm $H_2O$), or (Pa);
$P_v$ = Velocity Pressure of Fan, in (mm $H_2O$), or (Pa);
$p_{v2}$ = Fan Outlet Velocity Pressure, in [mm $H_2O$ (abs.)], or [Pa(abs.)];
r/min = Rotational Speed, in (rotations per minute);
$T_1$ = Gas Temperature at Fan Inlet, in (K);
$V_1$ = Fan inlet Rate, in (m$^3$/h);
$V_m$ = Gas Velocity, in (m/s);
$V_p$ = Peripheral Velocity, in (m/s);
t = Temperature Rise, in (K or degree °C);
$\rho$(rho) = Density (Mass Density), in (kg/m$^3$ );
$\pi$(pi) = Constant, equal to 3.1416;

**Subscripts:**

t = Based on total pressure;
s = Based on static pressure;
1 = At inlet conditions;
2 = At outlet conditions.

## 5. UNITS

International System of Units (SI) in accordance with IPS-E-GN-100 shall be used.

## 6. GENERAL

### 6.1 Fan Identification

Fans are rather generally identified as machines with relatively low pressure rises which move air or gases or vapors by means of rotating blades or impellers and change the rotating mechanical energy into pressure or work on the gas or vapor. The result of this work on the fluid will be in the form of pressure energy or velocity energy, or some combination of both.

## 6.2 Pressure Limit of Application

Fans for all services handling air or gas, on process duties (excluding those for direct cooling or ventilating) and which develop less than 35 kPa (0.35 bar) from atmospheric pressure, shall conform to API Std. No. 673 (see 2.1), and Iranian Petroleum Standards IPS-M-PM-230, for " Forced Draft Fans for Boilers and Process Services."

## 6.3 Types of Fan

For types of fan refer to Appendix E of this Standard.

## 6.4 Performance

**6.4.1** Fan performance shall be based on the static pressure differential across the fan inlet and outlet flanges. To obtain this differential, silencer and inlet losses, including control system losses, shall be added by the fan vendor to the purchaser's specified inlet and outlet static pressures.(API Std. 673,2.1.6).

**6.4.2** Performance curves shall have a continuous rising pressure characteristic from the rated point, as specified, to 60 percent or less of the rated flow. Performance curves, corrected for the specified gas at the specified conditions, shall be based on performance tests of actual or prototype equipment, including evase, if any, and inlet box(es). (API Std. 673,2.1.7).

**6.4.3** Fan performance shall be guaranteed to meet all operating conditions specified on the data sheet and shall be within the tolerances listed, at the normal operating conditions.

　**a)** For variable-speed fans, the static pressure and capacity shall be guaranteed with the understanding that the power shall not exceed +3%. These tolerances shall not be additive.

　**b)** For constant-speed fans the specified capacity shall be guaranteed with the understanding that the static pressure shall be within +5% and -0% of that specified; the power shall not exceed stated power by more than +3%. These tolerances shall not be additive.

**6.4.4** Fan performance shall be guaranteed in accordance with IPS Standard IPS-E-PR-250, "Performance Guarantee".

## 6.5 High Temperature Service

High temperature service for fans and auxiliaries is defined as a service for air and/or gas with a fan inlet temperature greater than 200°C. Fans, in this service will be indicated as "High temperature fans".

## 7. DESIGN CRITERIA

## 7.1 Selection Parameters of Process Fans

This section of standard is intended to cover information necessary to determine the approximate power requirement and other selection parameters of process fans and to furnish proper data for evaluation of manufacturer's proposals and/or preparation process data sheet.

## 7.2 Operational Characteristics and Performance

### 7.2.1 Centrifugal fans

The three types of centrifugal fan blades (radial, backward and forward) give three characteristic performances. Figures A.1 to A.5 of Appendix "A" present the typical characteristic curves for radial, backward and forward bladed fans respectively. Exact performance for a given fan can only be obtained on test.

### a) Radial blade

This type of blade is usually used for handling suspended materials, abrasive dust collecting and exhausting of pumps from dirty, greasy or acid environment.

The rather sharply rising static pressure curve of the radial blade centrifugal fan allows for small changes in volume as the resistance of the system changes considerably.

A Fair running static efficiency is 50-70 percent for both the straight radial blade and radial tip blade.

### b) Backward blade

This type of blade is well suited to stream line conditions and is used extensively on ventilating, air conditioning and clean and dirty process gas streams. The outstanding and important characteristic is the non-overloading power (kilowatts). It eliminates the need for oversized motors or other drivers. The usual operating static efficiency range for the regular blade is 65-80 percent and for the streamlined design is 80-92 percent.

### c) Forward blade

This type is usually shallow and operates at slow speed for a given capacity and usually has low outlet velocity.

Its operating characteristics are poor for many applications, since the power rises sharply with a decrease in static pressure once the peak pressure for the fan has been reached. The operating static efficiency range is 55-75 percent.

## 7.2.2 Axial flow fans

The performance of the axial flow fan is represented in figures A.6 and A.7 of Appendix "A".

The power characteristic is non overloading. The usual pressure range of application is 0-76 mm water (0-745 pascals) static pressure. The vaneaxial and tubeaxial can be selected for higher outlet velocities than the centrifugal ( 10.2 - 20.3 m/s).

The axial fans should be connected to ducts by tapered cone connection. The peak efficiency range of the tubeaxial is 30-50 percent and for the vaneaxial is 40-65 percent.

## 7.2.3 Propeller fans

These fans usually operate with no piping or duct work on either side, and move air or gas from one large open area to another. Pressures are usually very low and volumes depend upon size, blade pitch, number of blades and speed.

Static efficiencies run from 10-50 percent depending on the fan and its installation . With well designed inlet ring and discharge diffuser the efficiencies may be 50-60 percent.

## 7.3 Fan Laws

The performance of a fan is usually obtained from a manufacturer's specific curve. Expected performance for a change from one condition of operation to another, or from one fan size to another is given in Table B.1 for geometrically similar fans. (Appendix B)

Fan laws apply to blowers, exhausters, centrifugal and axial flow fans. The relations are satisfactory for engineering calculations as long as the pressure rise is not greater than 7 kPa.

Theoretically a 100 mm $H_2O$ (0.98 kPa) pressure rise affects air density to cause a one percent deviation. Where greater accuracy is required, the familiar adiabatic power relations are used.

These laws are applicable only for geometrically similar fans and to the same point of rating on the performance curve.

## 7.4 Performance Calculations

### 7.4.1 Pressure

#### a) Total System Pressure

The sum of the static and velocity pressure is the "Total System pressure" , $P_t$.

$$P_t = P_s + P_v \qquad \text{(Eq. 1)}$$

**Where:**

$P_s$      is static pressure
$P_v$      is velocity pressure

The fan total pressure Ptf, (see 3, f) is measured as the increase in total pressure given to a gas passing through a fan. It is a measure of the total energy increase per unit volume imparted to the flowing gas by the fan. The static pressure is the fan total pressure than less the fan outlet velocity pressure, (see 3, h).

#### b) Velocity Pressure (see 3, g)

$$Pv = \frac{\rho \cdot (Vm)^2}{19.608} \qquad \text{mm of water} \qquad \text{(Eq. 2)}$$

**or:** $\qquad Pv = \frac{\rho \cdot (Vm)^2}{2} \qquad \text{Pa} \qquad \text{(Eq. 3)}$

**Where:**

$\rho$      is gas density
$V_m$      is gas velocity,

Velocity pressure is indicated by a differential reading of an impact tube facing the direction of air flow in the fan outlet. It is a measure of the kinetic energy per unit volume of gas, existing at the fan outlet.

### 7.4.2 Peripheral velocity or tip speed

The peripheral velocity of the fan wheel or impeller is expressed as:

$$Vp = \frac{1}{60} D \cdot \pi \cdot (r = min) \qquad \text{Eq. 4}$$

**Where:**

$V_p$      is peripheral velocity
$D$      is wheel diameter
$r/min$      is rotational speed

### 7.4.3 Power

**a)** Fan kilowatt based on total pressure:

$$(FKW)_t = \frac{P_t \cdot V_1}{36.7 \cdot 10^4 \cdot e_t} \qquad \text{Eq. 5}$$

**Where:**

      *(FkW)*    is fan kilowatt

      *V1*    is inlet rate

**b)** Fan kilowatt based on static pressure output:

$$(FKW)_s = \frac{P_s ¢ V_1}{36.7 ¢ 10^4 ¢ e_s} \qquad \text{(Eq. 6)}$$

**c)** Gas kilowatt (Air kilowatt ) output:

$$(GKW) = \frac{P_t ¢ V_1}{36.7 ¢ 10^4} \qquad \text{Eq. 7}$$

**d)** Shaft or brake kilowatt (input), based on direct current motor:

    **BkW** = (Amp.) (Volts) (Motor efficiency) x $10^{-3}$ (Eq. 8)

**e)** Shaft or brake kilowatt (input), based on alternating current (3-phase) motor:

    **BkW** = 3(Amp.) (Volts) (Motor efficiency) (Power factor) $\times 10^{-3}$ (Eq. 9)

## 7.4.4  Efficiency

**a)** Mechanical (total) efficiency:

$$e_t = \frac{GKW}{BKW} = \frac{P_t ¢ V_1}{36.7 ¢ 10^4 ¢ BKW} \qquad \text{(Eq. 10)}$$

**b)** Static efficiency

$$e_s = e_t ¢ \frac{P_s}{P_{tf}} = \frac{P_s ¢ V_1}{36.7 ¢ 10^4 ¢ BKW} \qquad \text{Eq. 11}$$

## 7.4.5  Temperature rise

The temperature rise as the gas passes through a fan is:

$$4\,t = \frac{T_1[(P_{s2}=P_1)^{(k\ 1)=k}\ 1]}{e_s} \quad (P_{V2}=P_1)^{(k\ 1)=k} + 1 \qquad \text{(Eq. 12)}$$

**Where:**

      *ut*    is temperature rise

      $T_1$    is air or gas temperature at fan inlet

      $P_{s2}$    is fan outlet static pressure

      $P_1$    is atmospheric pressure or fan inlet pressure (if not atmospheric)

      $P_{v2}$    is fan outlet velocity pressure

      *k*    is ratio of specific heats, $C_p/C_v$

## 7.5  Fan Control

Fan volume is controlled by the following methods:

### a)  Variable Speed Drive

This type of control can be accomplished by turbines, DC motors, variable speed motors or slip-ring motors.

With changing speed of the driver the fan output capacity and pressure can be varied. For capacity reductions below 50 percent, an outlet damper is usually added to the system.

### b)  Outlet Damper with Constant Fan Speed

The system resistance is varied with this damper. The volume of gas delivered from the fan is changed as a function of the movement of the damper. It is low in first cost and simple to operate, but does require more power than other methods of control.
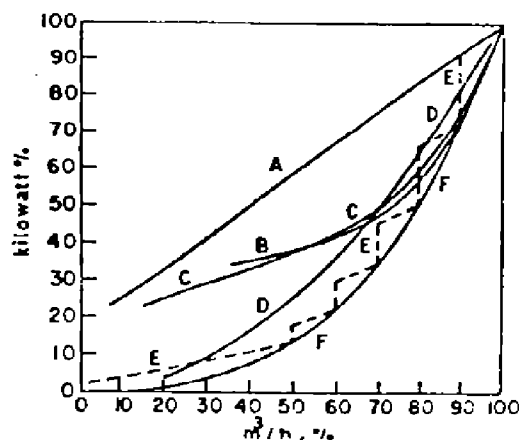
### c)  Variable Inlet Vane with Constant Fan Speed

The angle and/or extent of closure of the inlet vanes controls the volume of gas admitted to the inlet of the wheel.

The inlet vane control is more expensive than the outlet dampers but this can usually be justified by lower kilowatt costs, specially on large power installations.

### d)  Fluid Drive

This method allows fan speed to be adjusted 20-100 percent with corresponding volume changes. A constant speed motor is used, see Fig. 1, note that curve F of this figure is the actual power input to the fan shaft. The hydraulic of fluid drive has about 3 percent in losses, so its power input at 100 percent load is actually about 103 percent to allow for this. Curves B and C are for variable vane inlet dampening and Curve A is for outlet dampening of a backward blade fan. Curve E shows an outlet damper with multiple step speed slip-ring motor. This has outlet damper for final control from 89-100 percent. From this graph a reasonably accurate selection can be made of the control features to consider for most installation conditions.



**COMPARISON OF EFFICIENCIES OF FIVE PRINCIPAL METHODS OF
CONTROLLING FAN OUTPUT
Fig. 1**

## 7.6 Fan Systems

An operating fan is a part of some system. Regardless of the system, the fan cannot be selected until the flow and resistance characteristics have been analyzed. Fan selection for the system is based on the static pressure for a given volume of gas flowing. Since most fans operate at relatively low pressures the effect of uncertainty or error in resistance calculations can have a large percentage effect on Kilowatt and operational characteristics. Since it is essentially impossible to determine exact figures for the system resistances. It is to add 10-20 percent to the calculated static pressure as a safety factor.

## 7.7 Fan Selection Procedure

The following steps should be followed in fan selection.

### 7.7.1 Specifying the fan type

Informations presented in sections 6.3 and 7.2 may help in selecting the suitable fan for the process. Fan type curves should also be studied in order to recognize the effects of changes in system resistance on the fan performance and the volume and pressure changes caused by variations on speed. Recommendations of manufacturers are of particular importance in this stage.

Fig. C.1 of Appendix "C" may be used in fan type selection.

### 7.7.2 Specifying the inlet volume

The volume of a fan should be determined by (1) the process material balance plus reasonable extra (about 20 percent) plus volume for control at possible future requirement, (2) generous capacity for purging, and (3) process area ventilation composed of fume hoods, heat dissipation and normal comfort ventilation. Table 1 gives suggested air changes for area ventilation, but not air conditioning.

### TABLE 1 - AVERAGE AIR CHANGES REQUIRED FOR GOOD VENTILATION

| MINUTES PER CHARGE | MINUTES PER CHANGE |
|---|---|
| ASSEMBLY HALLS .......................................... 2-10 | KITCHENS - RESIDENT ......................................... 2- 5 |
| AUDITORIUMS .............................................. 2-10 | KITCHENS -RESTAURANT ................................. 1- 3 |
| BANKS ........................................................... 3-10 | LABORATORIES ................................................... 1- 5 |
| BOILER ROOMS ............................................ 1- 5 | MARKETS ............................................................. 2-10 |
| DINING ROOMS.............................................. 3-10 | OFFICES ............................................................... 2-10 |
| DRY CLEANERS ............................................ 1- 5 | PACKING HOUSES ............................................ 2- 5 |
| ENGINE ROOMS ........................................... 1- 3 | PLATING ROOMS .............................................. 1- 5 |
| FACTORIES ................................................... 2- 5 | PROJECTION ROOMS ....................................... 1- 3 |
| FORGE SHOPS .............................................. 2- 5 | RECREATION ROOMS ....................................... 2-10 |
| FOUNDRIES................................................... 1- 5 | RESIDENCES ....................................................... 2- 5 |
| GARAGES ...................................................... 2-10 | SALES ROOMS.................................................... 2-10 |
| GENERATOR ROOMS .................................. 2- 5 | THEATERS ........................................................... 2- 8 |
| GYMNASIUMS ............................................... 2-10 | TOILETS .............................................................. 2- 5 |
| KITCHENS - HOSPITAL ................................ 2- 5 | TRANSFORMER ROOMS ................................. 1- 5 |
| | WAREHOUSES ................................................... 2-10 |

### 7.7.3 System resistance

The system resistance must be calculated in the usual manner and at the actual operating conditions of the fan.

Corrections are then applied to convert this condition to "standard" for use in reading the rating tables.

### 7.7.4 Manufacturers multirating tables

The multirating tables (rating tables) of fan manufacturers are convenient for selecting any of the many types of fans. Usually $m^3$/h values can be found close enough to requirements to be acceptable. Direct interpolation in the table for volume, r/min and BkW is acceptable for narrow ranges, otherwise the fan laws must be used.

Performance tables are based on dry air at 21°C at sea level (barometric pressure 760 mm mercury) with a density of 1.2 $kg/m^3$ .

When the fans are required to handle gases at other conditions at the inlet, corrections must be made for temperature, altitude and air or gas density.

**7.7.4.1**  Performance at conditions other than above:

  **1)** Calculate actual density of gas (or air) under operating conditions.

For Air, Fig. 2 is convenient to use:

Read temperature correction factor, $F_1$
Read altitude correction Factor, $F_2$

Actual density $= \dfrac{(F_1)(F_2)}{1.2}$ ,$Kg/m^3$           (Eq. 13)

at operating conditions

For gases other than air the density must be calculated since the curves of Fig. 2 are for an air density of 1.2 $kg/m^3$. If the gas density at 21 °C and 101.325 kPa is close to that of air under these conditions, then the curves could be used for convenience. Otherwise, calculate the actual gas density by the gas laws.

  **2)** Calculate the equivalent static pressure:

Equivalent static pressure

$$\frac{1.2}{(actual \, density)}$$       (Eq. 14)

  **3)** From manufacturers' rating tables for air or gas, using the required $m^3$/h at inlet operating conditions and the equivalent static pressure calculated in (2), read the r/min (rpm) and BkW. Interpolate if necessary.

  **4)** The r/min is the correct value for the actual operating conditions.

  **5)** The BkW must be corrected for density:

$$A \, \frac{actual \, density}{1.2}$$       (Eq. 15)

  **6)** The correct performance at the actual operating conditions will be:

$m^3$/h as set at inlet conditions
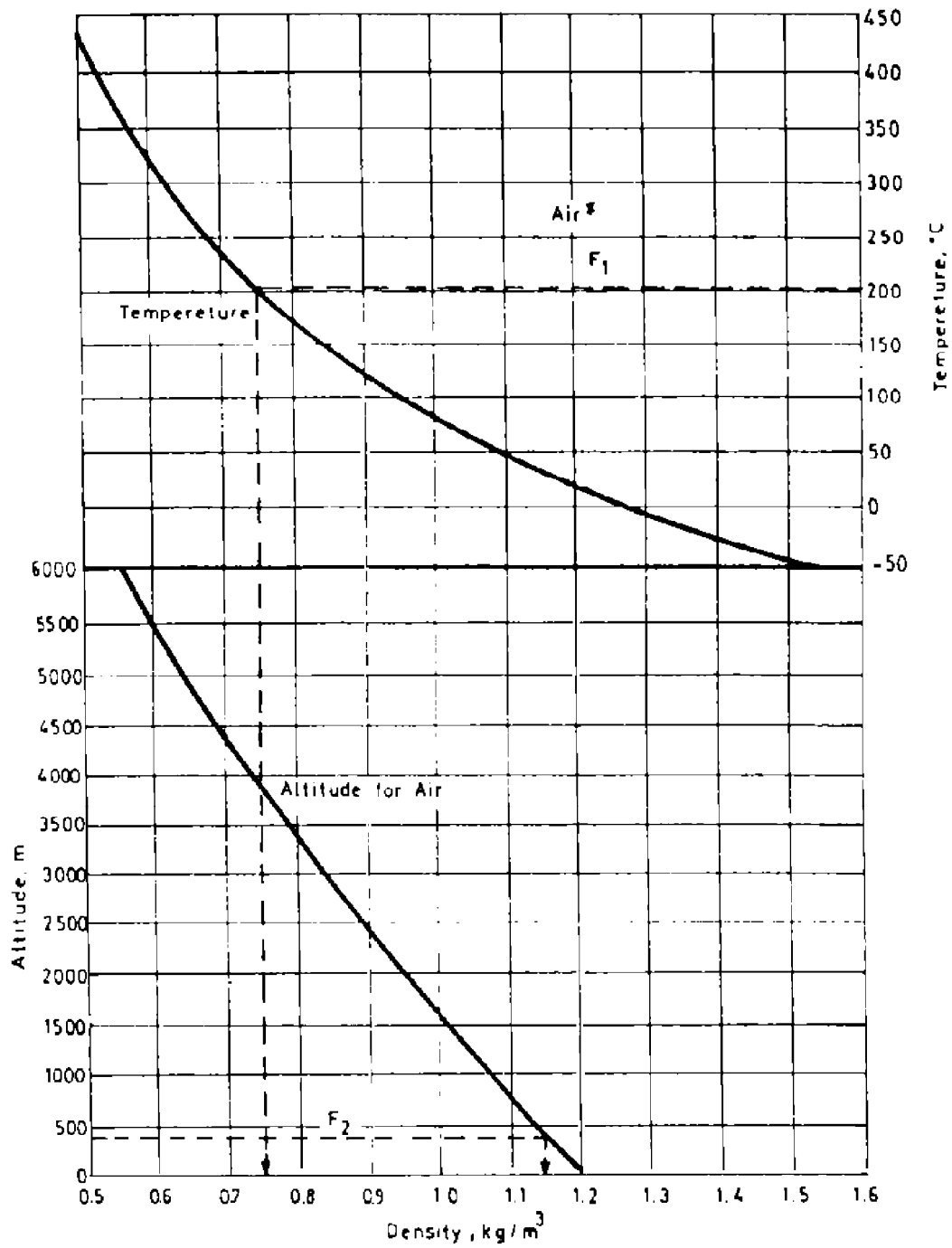Static pressure as set at inlet conditions, mm of water
Temperature as set at inlet conditions, °C.
r/min as read from manufacturers' tables.
BkW as corrected by (5) above.

Examples for fan selection are presented in Appendix "D".

* Note:

Conditions of air as defined in para. 7.7.4.

CORRECTION FACTORS FOR EFFECTS OF ALTITUDE AND TEMPERATURE ON AIR*
Fig. 2

### 7.8 Process Data Sheet

The following data should be prepared by process engineer as the basic informations to be filled in data (specification) sheet;

**a) Gas characteristics**

b Air

b Process gas; **Name:**

Composition:
Mol. Mass:

**b) Operating conditions**

Capacity, $(dm^3/s$ or kg/h)
Relative density, (Air = 1)
Barometric pressure, [m bar or mm $H_2O$ (abs.)]
Elevation above see level, meters (m)
Relative humidity, (%)
Inlet temperature (°C)
Inlet pressure [m bar or mm $H_2O$ (abs.)]
Outlet pressure [m bar or mm $H_2O$ (abs.)]

**c) Control**

b Outlet Dampers, (std.), (stream flow)
b Variable inlet vanes, (manual), (auto)
b Variable speed drive
b Slip ring motor
b Variable pitch blades
b Other

**d) Fan Type**

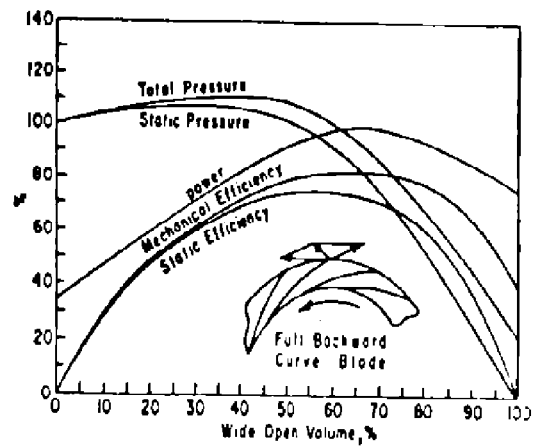b Centrifugal
b Vaneaxial
b Tubeaxial
b Propeller
b Other

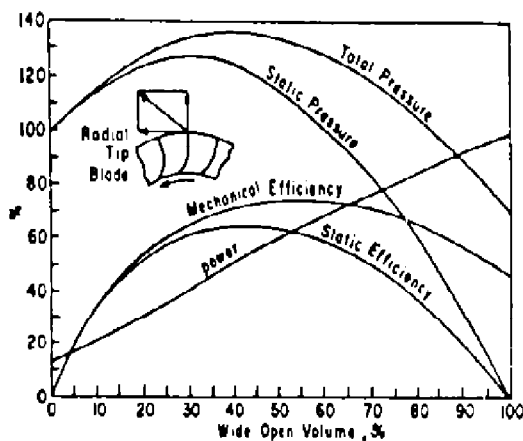**APPENDICES**


**APPENDIX A**

Characteristic Curves for Radial, Backward and Forward Bladed Fans (Figs. A1-A5) and
Performance Curves for Axial-Type Fans (Figs. A.6 & A.7).
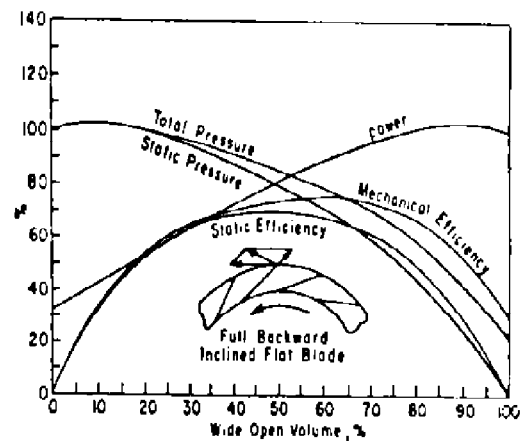


Characteristic curves
for straight radial blade
Fig. A.1



Characteristic curves
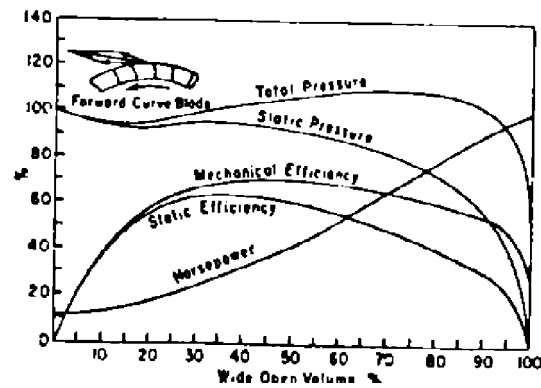for backward curved blade
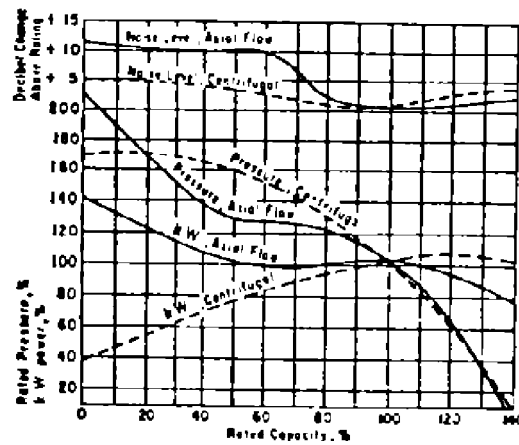Fig. A.3



Characteristic curves
for radial tip blade
Fig. A.2



Characteristic curves
for full backward inclined flat blade
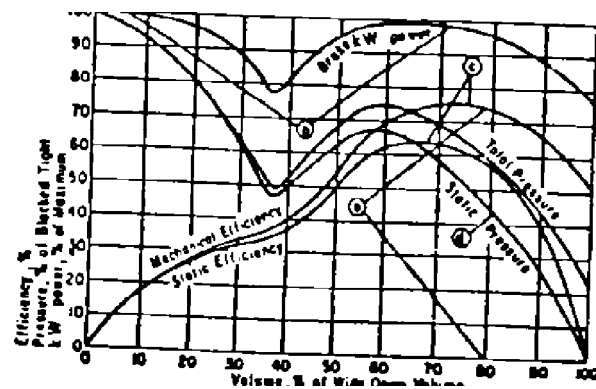Fig. A.4

(to be continued)

**APPENDIX A (continued)**



**Characteristic curves for forward curved blade**
**Fig. A.5**



**Performance curve for axial - type fans**
**Fig. A.6**



**Performance curves for "Streamlined" design of axial - type fans**
**Fig. A.7**

## APPENDIX B
## TABLE B.1 - TABLE OF FAN LAWS

| Fan Law Number | | Ratio of | = | Ratio | × | Ratio | × | Ratio (Normally Constant **) |
|---|---|---|---|---|---|---|---|---|
| 1 | a | $m^3/h$ | | $size^3$ | × | $(r/min)$ | | $1$ |
| | b | press. | = | $size^2$ | × | $(r/min)^2$ | × | $d$ |
| | c | kW | | $size^5$ | × | $(r/min)^3$ | | $d$ |
| 2 | a | $m^3/h$ | | $size^2$ | × | $press.^{1/2}$ | | $1/d^{1/2}$ |
| | b | r/min | = | $1/size$ | × | $press.^{1/2}$ | × | $1/d^{1/2}$ |
| | c | kW | | $size^2$ | × | $press.^{3/2}$ | | $1/d^{1/2}$ |
| 3 | a | r/min | | $1/size^3$ | × | $(m^3/h)$ | | $1$ |
| | b | press. | = | $1/size^4$ | × | $(m^3/h)^2$ | × | $d$ |
| | c | kW | | $1/size^4$ | × | $(m^3/h)^3$ | | $d$ |
| 4 | a | $m^3/h$ | | $size^{4/3}$ | × | $kW^{1/3}$ | | $1/d^{1/3}$ |
| | b | press. | = | $1/size^{4/3}$ | × | $kW^{2/3}$ | × | $1/d^{1/3}$ |
| | c | r/min | | $1/size^{5/3}$ | × | $kW^{1/3}$ | | $1/d^{1/3}$ |
| 5 | a | size | | $(m^3/h)^{1/2}$ | × | $1/press.^{1/4}$ | | $d^{1/4}$ |
| | b | r/min | = | $1/(m^3/h)^{1/2}$ | × | $press.^{3/4}$ | × | $1/d^{1/4}$ |
| | c | kW | | $(m^3/h)$ | × | press. | | $1$ |
| 6 | a | size | | $(m^3/h)^{1/3}$ | × | $1/(r/min)^{1/3}$ | | $1$ |
| | b | press | = | $(m^3/h)^{2/3}$ | × | $(r/min)^{4/3}$ | × | $d$ |
| | c | kW | | $(m^3/h)^{5/3}$ | × | $(r/min)^{4/3}$ | | $d$ |
| 7 | a | size | | $press.^{1/2}$ | × | $1/(r/min)$ | | $1/d^{1/2}$ |
| | b | $m^3/h$ | = | $press.^{3/2}$ | × | $1/(r/min)^2$ | × | $1/d^{3/2}$ |
| | c | kW | | $press.^{5/2}$ | × | $1/(r/min)^2$ | | $1/d^{3/2}$ |
| 8 | a | size | | $1/kW^{1/4}$ | × | $(m^3/h)^{3/4}$ | | $d^{1/4}$ |
| | b | r/min | = | $kW^{3/4}$ | × | $1/(m^3/h)^{5/4}$ | × | $1/d^{3/4}$ |
| | c | press. | | kW | × | $1/(m^3/h)$ | | $1$ |
| 9 | a | size | | $kW^{1/2}$ | × | $1/press.^{3/4}$ | | $d^{1/4}$ |
| | b | r/min | = | $1/kW^{1/2}$ | × | $press.^{5/4}$ | × | $1/d^{3/4}$ |
| | c | $m^3/h$ | | kW | × | press. | | $1$ |
| 10 | a | size | | $kW^{1/5}$ | × | $1/(r/min)^{3/5}$ | | $1/d^{1/5}$ |
| | b | $m^3/h$ | = | $kW^{3/5}$ | × | $1/(r/min)^{4/5}$ | × | $1/d^{3/5}$ |
| | c | press | | $kW^{2/5}$ | × | $(r/min)^{4/5}$ | | $d^{3/5}$ |

**(to be continued)**

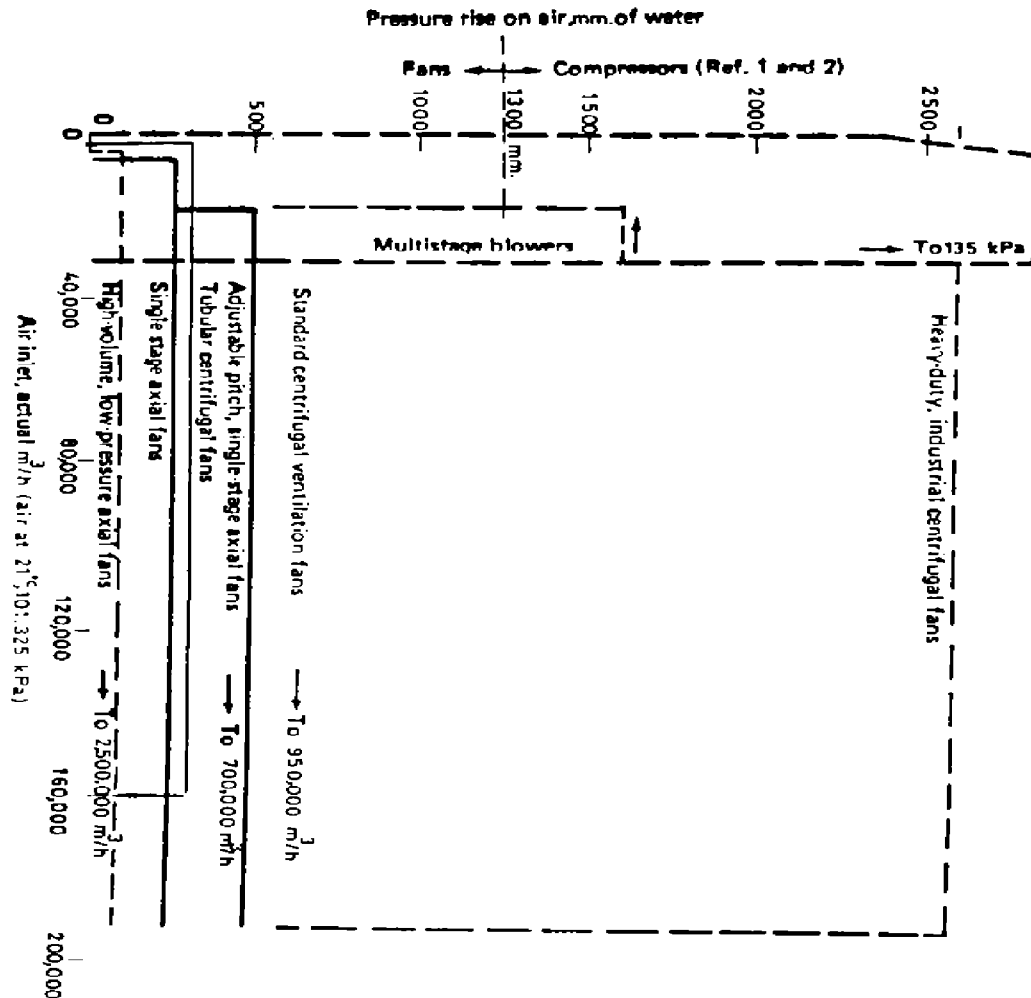**TABLE  B.1 (continued)**

| |
|---|
| FOR CONSTANT AMOUNT BY MASS (SIZE CONSTANT).<br>WHEN  DENSITY  OF  AIR  VARIES.<br><br>11  $m^3$/h, r/min AND PRESSURE VARY INVERSELY AS THE DENSITY, THAT IS, INVERSELY  AS  THE  BAROMETRIC  PRESSURE  AND  DIRECTLY  AS  THE ABSOLUTE  TEMPERATURE. |
| 12 KILOWATT VARIES INVERSELY AS THE SQUARE OF THE DENSITY THAT IS INVERSELY  AS  THE  SQUARE  OF  THE  BAROMETRIC  PRESSURE  AND DIRECTLY  AS THE  SQUARE  OF  THE  ABSOLUTE  TEMPERATURE. |
| FOR CONSTANT AMOUNT BY MASS (SIZE CONSTANT).<br>WHEN  BOTH  TEMPERATURE  AND  PRESSURE  VARY.<br><br>13   $m^3$/h  AND  r/min       VARY   AS     $\sqrt{\text{PRESSURE}}$   ×   ABSOLUTE TEMP.<br>     KILOWATT            VARIES AS     $\sqrt{\text{PRESSURE}^3}$  ×   ABSOLUTE TEMP. |

**\* Either one of these variables may be used independently in which case the other must remain constant. Then the desired data will vary as power of the variable indicated. These laws are written as if both varied at the same time. In this case the desired data will vary as the powers of the variables indicated.**


**\*\* In the last column the effect of relative density (d) is shown.**
**This is to apply as an extra variable when needed, otherwise density is supposed to be constant.**

# APPENDIX C
# FAN SELECTION GUIDE CHART



TYPICAL FAN SELECTION GUIDE CHART, BASED ON PRESSURE
RISE VERSUS AIR FLOW
Fig. C.1

**Notes on Fig. C.1:**

1) This chart shows the range of centrifugal and axial machines, and is based on catalogue ratings. The standard centrifugal ventilation fans operate approximately 560 mm of water. Beyond this, heavy-duty centrifugal fans, with higher compression ratios, may be made to specifications. The only area where no fan is available is above 2500 mm $H_2o$ at extremely low flows.

2) To use this graph: (1) calculate the actual $m^3$/h at inlet conditions (at fan flange), and the total pressure rise-in mm of water-from the inlet to the discharge fan-flange; (2) locate the $m^3$/h value on the chart; if the region where the point falls can be served by more than one type of fan (axial versus centrifugal, or different types of axial or centrifugal), decide on the type of fan by making economic and engineering evaluations.

## APPENDIX D
## EXAMPLES FOR FAN SELECTION

Example 1: Fan Selection

A system requires 10300 $m^3$/h of air at 205°C against a 52 mm $H_2O$ static pressure. The installation is at elevation 430 meters.

**a)** Determine: what type of fan blading should be used. A backward-curved blade fan will be selected for this installation because the following are not known: (1) the accuracy with which the system characteristic of 52 mm of water at 10300 $m^3$/h was determined, (2) the type of process control to be used, and (3) the possible system variation.

A backward-curved blade will take care of the above unknowns. It will have:

**1)** High efficiencies. It is a blade that offers flexibility by inherently providing high efficiencies over a wide range. It has its highest efficiency near its maximum kW-power. This gives flexibility above and below the design point.

**2)** Non-overloading characteristics. A backward-curve blade will allow close "motoring" without fear of overloading in the event of process upsets.

**3)** Steep static pressure curves. It offers a wide range of static pressures with a small change in capacity.

**b)** Determine: can the 10300 $m^3$/h at 205°C and against 52 mm of water, (2.04-inch) static pressure be used to select a fan from the manufacturers' tables?

The manufacturers' tables are prepared in accordance with the industry standard set up by the Air Moving and Conditioning Association. These tables are based on standard air.

Operating conditions other than these must be corrected before going into the table.

Since this is an air system the density correction chart Fig. 2 is used:

**1)** Actual density of air at operating condition:

From chart, read at 205°C:

F1 = 0.74

Also read from the lower curve at 430 m altitude:

F2 = 1.15

Actual density = (0.74)(1.15)/1.2 = 0.709 kg/$m^3$

Air density ratio = 0.709/1.2 = 0.59

**2)** Equivalent static pressure (at standard conditions):

$$= 52 \ ¢ \ \frac{1.2}{0.709} = 88 \quad \text{mm of water}$$

**3)** Select fan from manufacturers' performance table, at 88 mm of water and 10300 $m^3$/h. Suppose that the nearest acceptable unit in the manufacturers' table has the following characteristics:

Inlet volume = 10330 $m^3$/h

Speed = 2064 r/min at 88.9 mm $H_2O$(3.5 inches).

BkW = 3.862 kW (5.18 hp) for standard air.

**4)** Actual r/min = 2064 (would be almost the same for actual conditions).

**5)** Actual BkW of fan = 3.862 (0.709/1.2) = 2.282 kW.

**6)** Performance at 430 m elevation and 205°C inlet air temperature will be: (approximately, this can be improved by applying fan laws to data read from table).

Inlet volume = 10330 m$^3$/h
r/min = 2064 ($\pm$)
BkW = 2.282 kW

Determine: could any other fan be used in this application? The next larger or smaller fan size should be examined. Other manufacturers could possibly give a different size that might be more efficient. The final selection should be based on an analysis of several different manufacturers' fans.

Determine: what is the tip speed of this fan?

Wheel diameter = 460 mm (18.5 inches).

use Eq. 4 (see 7.4.2)

Tip speed =
$$= \frac{1}{60} \cdot D \cdot (r = min)$$

$$= \frac{460 \cdot 3.1416 \cdot 2064}{1000 \cdot 60} = 49.7 m = s$$

Determine: what is the outlet velocity of the fan.

When quietness of operation is important, the outlet velocity should be in the range of 6 to 11 m/s.

The low outlet velocity corresponds to low outlet velocity pressure, and as this latter factor directly influences power consumption. The velocity should be kept to a minimum, particularly when the static pressure is low. However, it should be pointed out that very low outlet velocities (less than 5 m/s) are not really desirable because they produce no advantages, not even quietness. Actual decibel ratings can be attained from the manufacturer, and these are the best indication of actual noise level to be expected.

**Example for fan selection using a process gas**

A fan is to handle 84000 m$^3$/h (at suction conditions) of a process gas at a suction condition of 49°C and 93 kPa (abs.) and is to discharge at 64 mm, H$_2$O. The gas density at these suction conditions is 1.36 kg/m$^3$.

Since manufacturers' tables are based on standard 1.2 kg/m$^3$ air density, this density difference must be recognized.

According to Fan Law No. 6, if the r/min (speed) and m$^3$/h (capacity) are constant, the pressure and power both vary directly as the relative density.

**1)** Equivalent static pressure $= 64 \cdot \left(\frac{1.2}{1.36}\right)^{\pm} = 56.47$ mm water

**2)** Read the fan characteristics (nearest) from the manufacturers' table. Suppose the following information is extracted:

Wheel dia. = 1.524 m, (60 inches)

$$\text{Max. peak BkW} = 139.2 \left(\frac{r=min}{1000}\right)^3$$

Outlet area = 0.0133 m$^2$
m$^3$/h (listed) = 84400
Outlet velocity = 12.2 m/s
Static pressure = 57.15 mm H$_2$O
BkW = 19.48 kW (26.1 hp)
r/min = 521

Since the capacity of the fan is slightly over the requirements, interpolation and correction of speed and BkW will not be made (0.47 percent over).

**3)** Correct r/min = 521 (not interpolating).

**4)** Actual m$^3$/h = 84400.

**5)** Actual BkW $= 19.48 \, \phi \, \left(\frac{1.36}{1.2}\right)^{\pm} = 22.08$ KW

**6)** Max. peak BkW $= 139.2 \, \phi \, \left(\frac{521}{1000}\right)^{\pm 3} = 19.68$ kW (for Air)

Max. peak BkW $= 19.68 \, \phi \, \left(\frac{1.36}{1.2}\right)^{\pm} = 22.31$ kW (for gas)

This indicates that the selection is operating at near peak condition.
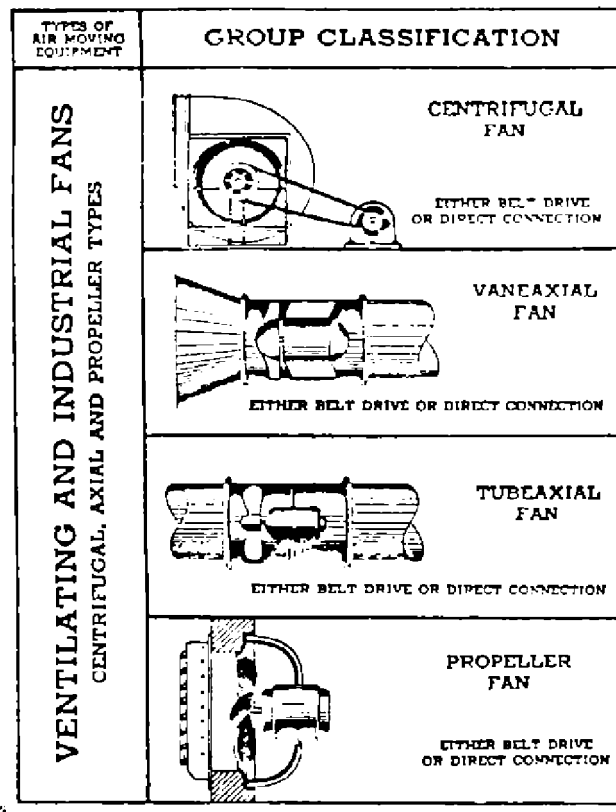
# APPENDIX E
# TYPES OF FANS

Ventilating and industrial fans are classified in four groups by the NAFM, National Association of Fan Manufacturers as described bellow (see Fig. E.1).

**E.1** A Centrifugal Fan consists of a fan rotor or wheel within a scroll type of housing. The centrifugal Fan is designed to move air or gases over a wide range of volume and pressures. The fan wheel may be furnished with straight, forward curve, backward curve, or radial tip blades. The fan housing may be constructed of sheet metal or cast metals with or without protective coating such as rubber, lead, enamel, etc.

**E.2** A Vaneaxial Fan consists of an axial flow wheel within a cylinder combined with a set of air guide vanes located either before or after the wheel. The Vaneaxial Fan is designed to move air or gases over a wide range of volumes and pressures. It is generally constructed of sheet metal although cast metal fan wheels are sometimes furnished.

**E.3** A Tubeaxial Fan consists of an axial flow wheel within a cylinder. The Tubeaxial Fan is designed to move air or gas through a wide range of volumes at medium pressures. Its construction is similar to the Vaneaxial Fan.

**E.4** A Propeller Fan consists of a propeller or disc wheel within a mounting ring or plate. The Propeller Fan is designed to move air from one enclosed space to another or from indoors to outdoors or vice versa in a wide range of volumes at low pressure. (The automatic type of shutter illustrated in cut opposite is not a part of the Propeller Fan but is an auxiliary device to protect the fan when not operating by keeping out wind, snow and cold).



**CLASSIFICATION OF VENTILATING AND INDUSTRIAL FANS**
**Fig. E.1**