KLM Technology Group
#03-12 Block Aronia,
Jalan Sri Perkasa 2
Taman Tampoi Utama
81200 Johor Bahru
Malaysia

PROCESS DESIGN OF FANS AND BLOWERS

(PROJECT STANDARDS AND SPECIFICATIONS)

TABLE OF CONTENT

SCOPE

This Project Standards and Specifications is intended to provide guidelines for process engineers for selection of proper type and preparation of process data sheets for fans used in OGP industries.

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

REFERENCES

Throughout this Standard the following dated and undated standards/codes are referred to. These referenced documents shall, to the extent specified herein, form a part of this standard. For dated references, the edition cited applies. The applicability of changes in dated references that occur after the cited date shall be mutually agreed upon by the Company and the Vendor. For undated references, the latest edition of the referenced documents (including any supplements and amendments) applies.

API (American Petroleum Institute)

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

DEFINITIONS AND TERMINOLOGY

Evase - Is a diffuser or a diverging discharge transition piece.

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

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).

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).

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).

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).

Standard air density - Is 1.2007 kg per cubic meter.

The fan inlet area - Is the inside area of the fan inlet collar.

The fan outlet area - Is the inside area of the fan outlet.

The mechanical efficiency of a fan - Is the ratio of power output to the power input.

The power input to a fan - Is expressed in kilowatts and is the measured kilowatt delivered to the fan shaft.

The power output of a fan - Is expressed in kilowatts and is based on fan volume and fan total pressure.

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

The static pressure (P_s) of the fan - Is the total pressure (P_{tf}) diminished by the fan velocity pressure (P_v).

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.

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).

The velocity pressure (Pv) of a fan - Is the pressure corresponding to the average velocity determination from the volume of air flow at the fan outlet area.

The volume handled by a fan - Is the number of cubic meters of air per hour expressed at fan outlet conditions.

SYMBOLS AND ABBREVIATIONS

SYMBOL/ABBREVIATION	DESCRIPTION
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 H_2O (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);
ρ(rho)	Density (Mass Density), in (kg/m^3);
π(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.

UNITS

This Standard is based on International System of Units (SI) except where otherwise specified.

GENERAL

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.

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.

Types of Fan

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

Performance

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).
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).

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.

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".

DESIGN CRITERIA

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.

Operational Characteristics and Performance

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.

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 vane axial and tube axial 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 tube axial is 30-50 percent and for the vane axial is 40-65 percent.

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.

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.

Performance Calculations

1. Pressure
   a. Total System Pressure

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

      $P_t = P_s + P_v$ Eq. (1)

      Where:

      P_s is static pressure

      P_v is velocity pressure

      The fan total pressure Ptf, 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.

   b. Velocity Pressure

      $$P_v = \frac{\rho.(V_m)^2}{19.608} \text{ mm of water} \quad \text{Eq. (2)}$$

      $$P_v = \frac{\rho.(V_m)^2}{2} \text{ Pa} \quad \text{Eq. (3)}$$

Where:

ρ 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.

2. Peripheral velocity or tip speed

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

$$V_p = \frac{1}{60} D \not\subset (r/min) \qquad \text{Eq. (4)}$$

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

3. Power

   a. Fan kilowatt based on total pressure:

$$(FKW)_t = \frac{P_t x V_1}{36.7x10^4 x e_t} \qquad \text{Eq. (5)}$$

   Where:

   (FkW) is fan kilowatt
   V_1 is inlet rate

   b. Fan kilowatt based on static pressure output:

$$(FKW)_s = \frac{P_s x V_1}{36.7x10^4 x e_s} \qquad \text{Eq. (6)}$$

   c. Gas kilowatt (Air kilowatt) output:

$$(GKW) = \frac{P_t x V_1}{36.7x10^4} \qquad \text{Eq. (7)}$$

   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) × 10^{-3} Eq. (9)

4. Efficiency

   a. Mechanical (total) efficiency:

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

   b. Static efficiency

$$e_s = e_t x \frac{p_s}{p_{tf}} = \frac{P_s x V_1}{36.7x10^4\ xBKW} \qquad \text{Eq. (11)}$$

5. Temperature rise

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

$$4t = \frac{T_1\left[(P_{s2}/P_1)^{(k1)/k}1\right]}{e_s}(P_{v2}\ /P_1)^{(k1)/k} + 1 \qquad \text{Eq. (12)}$$

   <u>Where:</u>

   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

Fan Control

Fan volume is controlled by the following methods:

1. 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.

2. 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.

3. 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.

4. 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.

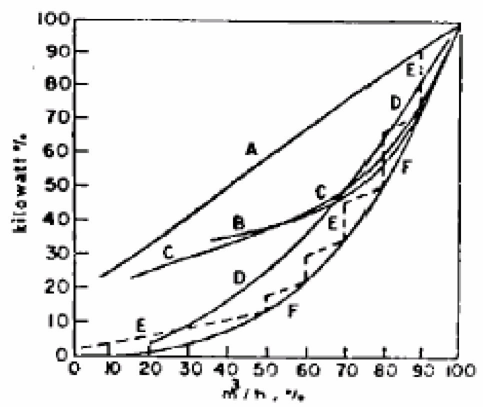

Fig. 1 Comparison of Efficiencies of Five Principal Methods of Controlling Fan Output

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.

Fan Selection Procedure

The following steps should be followed in fan selection.

1. Specifying the fan type

   Informations presented above 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.

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

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