PLANT ENGINEER’S GUIDE

CENTRIFUGAL FAN DESIGN
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I. INTRODUCTION TO CENTRIFUGAL FANS AND BLOWERS

Centrifugal fans and blowers cover a wide range of applications and flow-pressure specifications, from general, domestic-use ventilation or cooling fans having motors of only a few Watt, to large scale fans and blowers having various industrial applications that require power of several MW.

As a general rule of classification, fans generate pressure rise up to 1 meter (approximately 10kPa), blowers up to 1kg/cm² (approximately 100kPa) and compressors in excess of 1kg/cm².

Fans and Blowers may be classified into 2 types: Axial and Centrifugal. Centrifugal Fans cause a draft through the lifting force of the impeller vanes or blades, which add velocity and pressure to a volume of gas. This paper will describe the characteristics and performance of centrifugal fans and blowers only, referring to both hereafter as “fans”.

There are a variety of centrifugal fan impeller types, each with its own characteristics that are determined by the number of impeller blades, angle of blade outlet and blade profile.

The complete range of fans available from Ebara Hamada Blower is shown below.

![Fan Selection Diagram (50Hz)](image)
II. CENTRIFUGAL FAN GENERAL OVERVIEW

1. Representation of Fan Performance

1.1 Fan Performance

Fan performance is expressed in terms of Gas Volume, Gas Pressure, Shaft Power, and Efficiency.

Fan performance varies according to the condition of the induced gas (i.e. Temperature, Pressure, Gas Specific Weight, etc.). In this reference paper, unless otherwise noted, it will be assumed that the induced gas is standard air: 20°C, Atmospheric Pressure at mean Sea Level, 65% Humidity, and Air Specific Weight of 1.2kg/m³.

1.2 Gas Volume

Fan Gas Volume, unless otherwise noted, is assumed to be the volume of gas measured at the fan inlet flange in actual volume per time unit, such as m³/min or m³/sec.

When Gas Volume is expressed in Normal Condition (e.g. Nm³/min) or Mass Flow Rate (e.g. kg/min), Gas Volume at the fan inlet and fan outlet will be the same.

The pressure generated over a fan has the affect of compressing the gas, so the actual volume of gas measured at fan outlet will be less than the volume of gas measured at fan inlet. This relationship can be expressed by the following equation:

\[ Q_2 = Q_1 \times \left( \frac{P_{S2}}{P_{S1}} \right) \times \left( \frac{T_2}{T_1} \right) \]

Where:

- \( Q_1, Q_2 \): Gas Volume at fan inlet, fan outlet (m³/time unit)
- \( P_{S1}, P_{S2} \): Absolute Static Pressure at fan inlet, fan outlet (mmAq abs)
- \( T_1, T_2 \): Absolute Temperature at fan inlet, fan outlet (degrees K)

1.3 Gas Pressure

Gas Pressure is expressed as Total Fan Pressure or Fan Static Pressure.

Total Fan Pressure is the total increase in pressure caused by the fan, and is expressed as the difference of pressures at the fan inlet flange and fan outlet flange.

Static Pressure is calculated by subtracting the Dynamic Pressure, measured at the fan outlet, from the Total Pressure.

These relations may be expressed as:

\[ P_T = P_{t2} - P_{t1} \]
\[ P_S = P_T - P_{d2} \]

Where:

- \( P_T \): Total Pressure of Fan (mmAq)
- \( P_S \): Static Pressure of Fan (mmAq)
- \( P_{S1}, P_{S2} \): Static Pressure at Fan inlet, outlet (mmAq)
- \( P_{t1}, P_{t2} \): Total Pressure at Fan inlet, outlet (mmAq)
- \( P_{d1}, P_{d2} \): Dynamic Pressure at Fan inlet, outlet (mmAq)
1.4 Shaft Power
Shaft Power is the power input to the fan shaft end. In cases where the fan is being driven using an intermediate transmission device (belt, coupling, reduction gear, fluid coupling, etc.), and the efficiency of this device is difficult to determine, it shall be considered as part of the driving force of the fan and the main drive motor output shall be considered as the Fan Shaft Power.

1.5 Efficiency
Efficiency is an expression of the ratio between the amount of increase in energy transferred from the fan to the gas (Pneumatic Power) and the Shaft Power:

\[
\text{Efficiency} = \frac{\text{Pneumatic Power (kW)}}{\text{Shaft Power (kW)}}
\]

Pneumatic Power may be expressed as Total Pressure Pneumatic Power or Static Pressure Pneumatic Power. In terms of increase in energy, Total Pressure Pneumatic Power is closer to theoretical value, and Total Efficiency, calculated using Total Pressure Pneumatic Power, is used for JIS B8330.

For fans having comparatively large static pressure of 1000mmAq or greater, Static Pressure Efficiency may also be calculated using the Static Pressure Pneumatic Power, since Dynamic Pressure as a percentage Total Pressure becomes rather small.

While JIS B8330 handles Static Pressure Efficiency as a reference value, JIS B8340 defines it as Total Adiabatic Efficiency with no intermediate cooling.

Pneumatic Power may be defined by the following equations:

As defined by JIS B8330 (Total Pressure Pneumatic Power):

\[
L_T = \frac{k}{k-1} \times \frac{P_{T1} x Q_1}{6120} \times \left[ \left( \frac{P_{T2}}{P_{T1}} \right)^{\frac{k-1}{k}} - 1 \right] \quad \text{…………………… (3)}
\]

As defined by JIS B8340 (Total Adiabatic Pneumatic Power):

\[
L_{ad} = \frac{k}{k-1} \times \frac{P_{S1} x Q_1}{6120} \times \left[ \left( \frac{P_{S2}}{P_{S1}} \right)^{\frac{k-1}{k}} - 1 \right] \quad \text{…………………… (4)}
\]

Where,

\[
k = \frac{C_P}{C_V}, \text{ and in case of ambient air, it shall be assumed that } k = 1.4
\]

\[
P_{T1}, P_{T2}: \quad \text{Absolute Total Pressure at fan inlet, outlet (mmAq, abs)}
\]

\[
P_{S1}, P_{S2}: \quad \text{Absolute Static Pressure at fan inlet, outlet (mmAq, abs)}
\]

\[
P_a: \quad \text{Atmospheric Pressure (mmAq abs)}
\]

\[
P_{S1} = P_a + P_{S1} \quad P_{S2} = P_a + P_{S2}
\]

\[
P_{T1} = P_{S1} + P_{d1} \quad P_{T2} = P_{S2} + P_{d2}
\]

and, in case the pressure ratio, \( \frac{P_{S2}}{P_{S1}} \), is less than or equal to 1.03, the below equation should be followed:

\[
\text{Pneumatic Force ( = Total Pneumatic Force)} = \frac{Q_1}{6120} \times \left( \left( \frac{P_{S2}}{P_{S1}} \right) + \left( P_{d2} - P_{d1} \right) \right) (\text{kW}) \quad \text{…………………… (5)}
\]
2. Centrifugal Fan Characteristics

2.1 Specific Speed (Ns)

One of the ways of describing a fan is Ns, or Specific Speed, as defined by the following equation:

\[
Ns = \frac{N \times Q^{1/2}}{H^{3/4}}
\]

Where,

- \(Ns\) : Specific Speed
- \(N\) : Number of revolutions of impeller per minute (rpm)
- \(Q\) : Gas Volume
- \(H\) : Head

Specific Speed (Ns) may be defined as the required rpm which satisfy unit flow of 1m³/min and unit head of 1m.

The Specific Speed (Ns) of geometrically similar impellers is the same, and does not change even if the actual rpm changes. The size of Ns is not decided by the magnitude of actual rpm, but is determined by a selection based on rpm, gas volume, and head. Therefore, in some cases where the rpm is large, the Ns value may be small, and, on the contrary, where the rpm is small, the Ns may be large.

Impellers of a uniform diameter having large Ns also have large entry diameters and blade widths. Likewise, impellers of uniform diameter which have small Ns also have small entry diameters and blade widths. Figure 2 demonstrates this general condition.

When designing blowers of the same Gas Volume and Gas pressure, as a general rule, if a low rpm is selected, then Ns is also small and the impeller diameter becomes large, and, likewise, if a high rpm is selected, the Ns is also high and impeller diameter becomes small.

There is a very close relationship between blower efficiency and Ns. When the Ns is excessively small, loss in efficiency due to disc friction becomes large. On the other hand, if Ns is excessively large, fluid loss in the impeller becomes large. Thus, high blower efficiency cannot be expected in either case.

Optimal Ns varies according to specific impeller designs, but it is reasonable to accept the following ranges as applicable to various impeller types:

- Turbo Type: 200 – 800
- Plate (Radial) Type: 200 – 450
- Sirocco (multi-blade) Type: 300 – 600
- Axial Type: 1500 – 2000

---

POWERTECH - SUPPLIERS OF EQUIPMENT AND SERVICES TO THE POWER, CEMENT AND STEEL INDUSTRY
CUSTOM PUMPS, TURBINES
CUSTOM CENTRIFUGAL FANS
EBARA HAMADA BLOWER CO., LTD.
2.2 Proportional Rule

Where 2 impellers which are geometrically similar are operated at different rpm and handle gases of different specific weights, and the pressure ratio \( P_2 / P_1 \) is less than 1.1, the following relationships can be assumed:

\[
Q_2 = Q_1 \times \left( \frac{D_2}{D_1} \right)^3 \times \left( \frac{N_2}{N_1} \right)
\]

\[
P_2 = P_1 \times \left( \frac{\gamma_2}{\gamma_1} \right) \times \left( \frac{D_2}{D_1} \right)^2 \times \left( \frac{N_2}{N_1} \right)^2
\]

\[
L_2 = L_1 \times \left( \frac{\gamma_2}{\gamma_1} \right) \times \left( \frac{D_2}{D_1} \right)^5 \times \left( \frac{N_2}{N_1} \right)^3
\]

\[
\eta_1 = \eta_2
\]

Where,
- \( Q \): Gas Volume (m³/min)
- \( P \): Gas Pressure (mmAq)
- \( L \): Shaft Power (kW)
- \( D \): Impeller Diameter (m)
- \( N \): Number of Revolutions per minute (rpm)
- \( \gamma \): Specific Weight of Gas (kg/m³)
- \( \eta \): Efficiency (%)

Equation (7) is based on the assumption that the shape of two impellers, indicated by the subscript “1” and “2”, are geometrically similar.

Furthermore, as dictated by the laws of fluid dynamics, the Reynold’s number, Re, of the flow in both impellers should be identical. In order to express this condition in ratio, the restriction that the pressure ratio \( P_2 / P_1 \) be less than 1.1 shall be applied. Accordingly, Equation (7) is not an absolutely precise representation, but serves well for day-to-day, practical estimations.

At right, an impeller being tiled with wear-preventing Alumina and Silica Nitride Ceramic Tiles.

End User:
FOSKOR
(South Africa)

Application:
Vertical Mill IDF

Specifications:
1,000,000m³/hr
108mBar
998rpm

Motor:
3850kW

Impeller Diameter:
3.0 meter
3. Centrifugal Fan Pressure and Ducting Resistance

3.1 Conditions to Achieve Stable Gas Flow

A draft, Q1, forced through a certain ducting is determined by the ducting resistance curve, R, and Fan Pressure (= Static Pressure) curve, P.

In Figure (3), as indicated by the R-curve, Ducting Resistance is a curve climbing upward to the right, in a proportion approximately square of the gas volume. Although Ducting Resistance, R, can be expressed by the curve equivalent to \( R = C_1 Q + C_2 \), it is usually the case that \( C_2 = 0 \), where \( C \) is a coefficient of duct resistance. Meanwhile, Fan Pressure (= Static Pressure) reaches a maximum value within a small gas volume range as shown by the P-curve, curving downward to the right through the design point. The flow volume which this fan can induce, Q1, corresponds to the intersection of the P-curve and R-curve. The intersection of these curves is called the working point, and is the point at which continuous, stable operation is best achieved.

In accordance with Fig. 3, when gas volume Q1 increases for some reason to Q1', Duct Resistance also increases, but fan pressure falls to P' and this imbalance results in a tendency to restore gas volume Q1. If gas volume Q1 were to fall to Q1'', the relationship between Duct Resistance and Fan Gas Pressure becomes P1'' > R2'', and this imbalance also results in a natural tendency to restore gas volume Q1. In this way, imbalances are naturally and continually rectified to gas volume Q1. The condition to achieve stable operation may be represented as the closeness of the P-curve to the actual working point, and stable operation is easier to achieve with steep rather than shallow pressure curves.

3.2 Unstable Operation

A study is made where, as indicated in Fig. 4, the fan working point falls to the left of the peak of the fan pressure P-curve.

When, for some reason, gas volume increases to Q1', both fan pressure and duct resistivity increase and there is little or no influence to return gas volume to Q1 as in comparison to Fig. 3. On the other hand, in the case gas volume decreases to Q1'', Q1 fluctuation becomes easier compared to that of Fig. 3. This condition tends towards instability to the extent that the slope of the P-curve in the vicinity of the working point becomes large.

In cases of extreme gas volume fluctuation, vibration and noise occur, and operation becomes impossible. This phenomenon is called the Surging Effect.
3.3 Types of Fans and Corresponding Areas of Unstable Operation

Almost every type of fan and blower will have a distinct peak in the Pressure Curve.

\[ P_{th\infty} = K_1 Q + K_2 \]
\[ \Delta P_1 = C_{11} Q^2 \]
\[ \Delta P_2 = C_{21} Q^2 + C_{22} Q + C_{23} \]

In Fig. 5, theoretical pressure rise, \( P_{th\infty} \), is represented by a line sloping downward to the right. \( \Delta P_1 \) represents fluid loss in the impeller, and has the same characteristic as duct resistance. \( \Delta P_2 \) is impact loss, most of which is caused by the collision of gas and impeller at the impeller entry.

Fan gas pressure may be thought to be achieved at each corresponding gas volume as a result of the relationship expressed in equation (8):

\[
P = P_{th\infty} - (\Delta P_1 + \Delta P_2) \tag{8}
\]

The impeller entry angle is optimized for minimum loss in design gas volume. Therefore, the lowest value of \( \Delta P_2 \) is located at the point of design gas volume. As can be seen from equation (8), the peak of the P-curve appears to the left of the design gas volume.

In Fig. 6, let \( Q_1-P_1 \) and \( Q_S-P_S \) be design and peak gas volumes:

The value \( Q_S / Q_1 \) is determined by the fan type.

<table>
<thead>
<tr>
<th>Fan Type</th>
<th>( Q_S / Q_1 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial</td>
<td>0.75 – 0.9</td>
</tr>
<tr>
<td>Multi-Blade</td>
<td>0.7 – 0.9</td>
</tr>
<tr>
<td>Plate</td>
<td>0.6 – 0.85</td>
</tr>
<tr>
<td>Turbo</td>
<td>0.4 – 0.6</td>
</tr>
</tbody>
</table>
3.4 Gas Volume when Fan Speed Fluctuates (same fan, same ducting)

In Fig. 7, $P_1$-curve corresponds to Fan Speed (RPM), $N_1$, and $P_2$-curve corresponds to Fan Speed (RPM), $N_2$.

The affect on fan performance when RPM change from $N_1$ to $N_2$ can be understood from the relationships established in equations (7):

$$Q_2 = Q_1 \times \left( \frac{D_2}{D_1} \right)^3 \times \left( \frac{N_2}{N_1} \right) = Q_1 \times \frac{N_2}{N_1}$$

Thus

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1}$$

$$P_2 = P_1 \times \left( \frac{\gamma_2}{\gamma_1} \right) \times \left( \frac{D_2}{D_1} \right)^2 \times \left( \frac{N_2}{N_1} \right)^2 = P_1 \times \left( \frac{N_2}{N_1} \right)^2$$

On the other hand, Duct Resistance Coefficient, $C_1$, is:

$$R_1 \quad (= P_1)$$

$$C_1 = \frac{R_1}{Q_1^2}$$

Let Ducting Resistance be $R_2$ when the gas volume is $Q_2$:

$$R_2 = C_1 Q_2^2$$

$$= \frac{R_1}{Q_1^2} \times Q_2^2 = R_1 \times \left( \frac{Q_2}{Q_1} \right)^2 = P_1 \times \left( \frac{N_2}{N_1} \right)^2$$

Therefore,

$$R_2 = P_2$$

Thus, considering the case where Fan Speed $N_1$ changes to $N_2$ on the same duct and same fan, Gas Volume changes in direct proportion to the change in speed, while Gas Pressure changes in proportion to the square of the changed speed. At this time, Shaft Power changes in proportion to the cube of the changed speed.
3.5 Gas Volume when Specific Weight (Density) of Gas Fluctuates on the same ducting and same fan

Fan Capacity when Specific Weight changes from $\gamma_1$ to $\gamma_2$ can be obtained from Equation (7) as follows:

$$Q_2 = Q_1 \times \left( \frac{D_1}{D_1} \right)^3 \times \left( \frac{N_1}{N_1} \right) = Q_1$$

$$P_2 = P_1 \times \left( \frac{\gamma_2}{\gamma_1} \right) \times \left( \frac{D_2}{D_1} \right)^2 \times \left( \frac{N_2}{N_1} \right)^2 = P_1 \times \frac{\gamma_2}{\gamma_1}$$

Whereas, Ducting Resistivity $R$ is:

$$R_1 = (P_1) = \frac{\gamma_1}{2g} \sum \zeta_i v_i^2$$

Thus, $\sum \zeta_i v_i^2 = \frac{2gR_1}{\gamma_1}$

Resistivity $R_2$ when the Gas Specific Weight is $\gamma_2$:

$$R_2 = \frac{\gamma_2}{2g} \sum \zeta_i v_i^2$$

$$= \frac{\gamma_2}{\gamma_1} R_1 = \frac{\gamma_2}{\gamma_1} P_1$$

Therefore, $R_2 = P_2$

Accordingly, when the Specific Weight changes in the same ducting and fan, Gas Volume does not change, while Gas Pressure changes in direct proportion to the change in Gas Specific Weight.

4. Combined Operation

The operation of two or more fans in series or parallel is called "Combination Operation." Where a process line is modified resulting in new, additional resistance being added to the system, a shortage in gas pressure may result. Therefore, in order to attain a fixed gas pressure, two or more blowers are operated in series.

Also, when an increased Gas Volume becomes necessary, it may be advantageous to add one or more fans in parallel, such that the required number of fans can be operated to meet changing Gas Volume requirements, rather than simply replacing the original fan with one single, larger capacity fan.
4.1 Series Operation

In Fig. 9, the Gas Pressure Curve when two fans with the same capacity are operated in series is a' - b' - c' - d'; the corresponding sum of the pressures of each individual fan at points a, b, c, and d.

When Resistance Curve, R, is constant, the actual operating point b, becomes c' in series operation. Gas pressure does not simply double.

Gas Volume increases from Qb to Qc, and, because the operation point of each fan now shifts from 'b' to 'c', shaft power increases for each fan correspondingly. Individually, with the same system resistance, each fan would operate at point 'b', but as the fans are installed in series, together they produce a greater pressure differential, which results in greater flow. Shaft power corresponds to this operating point.

Because the first fan in series pressurizes the gas which then feeds to the inlet of the second fan in series, the Gas Specific Weight at the inlet of the second fan will be greater. Accordingly, the second fan will show a greater pressure differential and will draw greater shaft power than the first fan.

Fig. 10 shows the curves when two fans with different capacity are operated in series. Where the curve a - b - c - d represents the first fan, and the curve a’ - b’ - c’ - d’ represents the second fan, the curve a” - b” - c” - d” then represents the performance of the two fans in series (c” is the same as c’, because the value of c = 0, and d” is the same as d).

When the combined operating point (e) falls to the left of (c’), that is, when Resistance R is relatively high, Gas Volume of the fans in series is larger than either fan would be if operated individually.

On the other hand, in cases of combined, series operation on a low-resistance system, represented by Resistance Curve R1, when the combined working point (e’) is to the right of (c’), Gas Volume and Gas Pressure become less than if the fans were operated individually. Accordingly, there is no merit in operating two fans in series in such a situation, and caution should be exercised when considering increasing flow and pressure by series operation on a low-resistance system.

Series operation is best suited to high-resistance systems.
4.2 Parallel Operation

In Fig. 11, the gas pressure curve when two fans having the same capacity are operated in parallel is represented by the curve \( a' - b' - c' - d' \); the corresponding sum of \( a - b - c - d \), which represents the capacity of each fan under individual operation.

When Resistance Curve R is constant, the individual operating point \((c)\) becomes \((b')\) in parallel operation, while gas volume does not simply double.

Although Gas Pressure increases to \((b')\) from \((c)\), this corresponds to the individual fans operating at point \((b)\), so power drawn by each fan will decrease compared to shaft power drawn at point \((c)\).

Fig. 12 is a study of two fans, having different capacities, operated in parallel.

Assuming the curve \( a - b - c - d \) represents the first fan, and the curve \( a' - b' - c' - d' \) represents the second fan, the curve \( a'' - b'' - c'' - d'' \) then represents the performance of the two fans in parallel.

When the combined, parallel operating point \((e)\) falls to the right of \((b)\) \((b\) is that point on the \( a - b - c - d \) curve which corresponds to Gas Pressure point \( a' \)), that is, Resistance R is comparatively small, Gas Volume becomes larger in comparison to the Gas Volume rating of either fan.

However, for high-resistance systems, the parallel working point \((e1)\) may fall to the left of point \((b)\), and the Combined Operation Gas Volume and Gas Pressure will become less than that of each individual fan. Furthermore, unstable operation will occur as the fan characterized by the curve \( a' - b' - c' - d' \) enters the surging zone.

Accordingly, parallel operation is generally not recommended for high-resistance systems. The closer the slope of the resistance curve is to vertical, the less feasible it is to operate fans of different ratings.
5. Controlling Gas Volume

As Gas Volume is determined by the intersection point of the duct resistance curve and pressure curve as previously stated, in order to control the gas volume, either the resistance curve or the pressure curve (or combination thereof) should be changed.

5.1 Discharge Damper

This is a method to control gas volume by changing resistance, and is the simplest control method. A louver damper is installed on the fan casing outlet.

In Fig. 13, when the damper is closed, Resistance changes from the original curve $R$ to $R_1$. With this change, Gas Volume decreases from $Q$ to $Q_1$, and Shaft Power decreases from $L$ to $L_1$.

In this figure, $\Delta R_1$ represents the increment of resistance increase caused by damper closure and becomes loss. Therefore, although a very simple method, it is the least economical of available methods of control. This method is used primarily for fans of small power and/or small range of volume control.

5.2 Suction Damper

This is a performance manipulation method and involves installing a damper at the fan inlet in order to control Gas Volume.

In Fig. 14, as the damper is closed, the curve for gas pressure, $P$, decreases from $P_1$ to $P_2$, and, simultaneously, the Shaft Power curve, $L$, decreases from $L_1$ to $L_2$.

In fans with inlet boxes, this effect is more dramatic, since the damper vane effectively guides the flow in the rotating direction of the impeller.

Care must be taken using this control method, as mis-installation of the inlet damper causes gas to flow against the rotating direction of the impeller, and abnormally high shaft power may be drawn, and large pressure pulsations may occur as well, resulting in unstable fan operation.

Where the Resistance Curve $R$ and Gas Pressure Curves, $P_1$ and $P_2$, are operating points created by damper control, Gas Volume decreases from $Q$ to $Q_1$ and $Q_2$. Then, Shaft Power becomes $L_1$ and $L_2$, and causing restriction with an outlet damper will cause the inlet Gas Specific Weight to decrease, further reducing shaft power. As the Inlet Louver Damper is gradually closed, the Pressure curve becomes more and more steep. Simultaneously, the pressure peak moves left (low gas Volume range), effectively discouraging surging effect from occurring.
5.3 Inlet Radial Vane Control (IRVC)

This method changes fan performance by installing an Inlet Radial Vane Control (IRVC) at the fan inlet to control Gas Volume.

The suction vane gives the body of gas entering the impeller the same rotation as the impeller, thereby reducing turbulence and required fan shaft power.

Fig. 15 shows the performance curve resulting from suction vane operation. Although this curve resembles the curve when the inlet damper is operated, in the case of IRVC control, reductions in shaft power are markedly greater.

IRVC control changes the operating point by manipulating the pressure curve in a similar way as inlet damper, but with greater efficiency. As the pressure curve changes, the new operating point is determined as the point of intersection of the pressure curve and Resistance curve R. As the IRVC changes the working points to P1 and P2, Shaft Power becomes points L1 and L2 on the dotted shaft power curve. Care must be taken, as mis-installation of the inlet damper causes gas to flow against the rotating direction of the impeller, and abnormally high shaft power may be drawn, and large pressure pulsations may occur as well, resulting in unstable fan operation.

IRVC are often used with ambient air, and are not generally recommended for use on high-temperature applications, or applications with large amounts of airborne dust, as dust adherence, abrasion, and/or thermal expansion can cause interference and sticking of the moving mechanical parts of the IRVC. Special designs to protect the moving mechanical parts of the IRVC may be considered on a case-by-case basis.
5.4 Speed Control

This is a method of control where fan performance is changed by changing the fan speed.

In Fig. 16, when Fan Speed, N, is changed to N1 and N2, Gas Pressure becomes P1 and P2 on the Pressure Curve, and Shaft Power becomes points L1 and L2 on the dotted L-Shaft Power Curve.

Q1 and Q2, P1 and P2, and L1 and L2 can be calculated by applying the equations that define the relationship between Flow, Pressure, Shaft Power, and Fan Speed in section 2.2 Proportional Rule:

\[ Q_1 = Q \times \left( \frac{N_1}{N} \right) \]

\[ P_1 = P \times \left( \frac{N_1}{N} \right)^2 \]

\[ L_1 = L \times \left( \frac{N_1}{N} \right)^3 \]

Controlling Pressure and Volume by Speed Control reduces Shaft Power to the greatest extent compared to all other methods.

Various methods are shown below. Advances in Inverter Control technology in recent years make this method today the most popular method due to cost, ease and flexibility of control, and reliability.

Fan suppliers must be informed when speed control will be applied, and the speed range that the fan will operate, so that studies of any potential harmonic or resonant interference can be performed, and countermeasures considered as necessary, to ensure trouble-free fan operation.
5.5 Variable Pitch

As a method to change the performance of Axial Fans, the variable pitch-blade axial fan allows adjustment to flow by changing the angle of blades. There is a manually adjustable type which can be adjusted while the fan is stopped. There are also types that can be adjusted during operation by use of a hydraulic actuation mechanism.

6. Points of Caution

This section describes the affects that various arrangements can have on fan performance, both positively and negatively.

6.1 No Bend at Fan Inlet

(Fig. 17)
6.2 No Bend at Fan Outlet

GOOD

BAD

REMEDY

bend complements
revolution direction

installation of
guide vanes

BAD

REMEDY

bend opposes
revolution direction

installation of
straight duct

(Fig. 18)

6.3 Installation of Suction Damper

GOOD

BAD

flow opposes
direction of
impeller revolution

flow enters in
direction of revolution

(Fig. 19)
6.4 Correct Direction of Rotation

As long as care is taken to assemble the impeller in the fan assembly according to the correct, indicated direction of rotation, the fan should fulfill its function. However, reverse installation or reverse rotation of the impeller may cause insufficient flow, or excessive shaft power, possibly resulting in reverse gas flow in the case of Axial Fans.

Regarding direction of rotation, refer to Fig. 20 below.

When two fans are designed with opposite directions of rotation, special attention must be paid to installing the impeller with the correct rotation, and impeller and casing should be confirmed to match.

i. Direction of Impeller Rotation (Arrow indicates direction of rotation)

(Fig. 20)

ii. Blower Casing Outlet Position and Direction of Rotation

(Fig. 21)
6.5 Operation at Correct Specifications

i. Gas Temperature

When a fan operates at full capacity, at an inlet temperature that is lower than the fan’s specified design temperature, the fan will draw greater shaft power than expected at the fan design point, and there is a danger of overloading the drive motor.

The relationship between Gas Specific Weight and Temperature is established by the following equation:

\[
\gamma_2 = \gamma_1 \times \left( \frac{T_1}{T_2} \right), \quad T_2 < T_1
\]

Where,

- \(T_1\) : Specified Absolute Temperature = 273 + \(t_1\) (deg K)
- \(T_2\) : Working Absolute Temperature = 273 + \(t_2\) (deg K)
- \(\gamma_1\) : Specified Specific Weight (kg/m\(^3\))
- \(\gamma_2\) : Working Specific Weight (kg/m\(^3\))
- \(t_1\) : Specified Temperature (deg C)
- \(t_2\) : Working Temperature (deg C)

Shaft Power when change occurs in Specific Weight may be expressed as:

\[
L_2 = L_1 \times \left( \frac{\gamma_2}{\gamma_1} \right) = L_1 \times \left( \frac{T_1}{T_2} \right)
\]

\(L_2 > L_1\), \((\frac{T_1}{T_2} > 1)\)

When Gas Temperature is higher than that specified, Gas Pressure will drop.

Gas Pressure when Specific Weight changes may be expressed as:

\[
P_2 = P_1 \times \frac{\gamma_2}{\gamma_1} = P_1 \times \frac{T_1}{T_2}
\]

Thus, \(P_2 < P_1\), \((\frac{T_1}{T_2} < 1)\)

*EHB Factory Performance Testing Facilities.*

It is important that Flow, Pressure, Temperature, Fan Speed, and Shaft Power are measured simultaneously for accurate assessment of fan performance.
ii. Affect of Humidity

Gas Pressure:
When water vapour enters a fan designed only for the handling of dry air, a fall in pressure results.

Specific weight of damp air may be obtained by the following equation:

$$\gamma = 1.293 \times \frac{273}{273 + t} \times \frac{H - 0.378 \cdot F}{10330} \quad (kg/m^3)$$

Where,
- $t$ = Gas Temperature
- $H$ = Atmospheric Pressure
- $\mathcal{F}$ = Relative Humidity
- $F$ = Saturated Vapor Pressure at $t \degree C$

The higher the Gas Temperature becomes, the higher the value $F$ becomes. As Relative Humidity becomes higher, the Specific Weight for damp air becomes increasingly smaller than that for dry air.

As previously indicated, as $\gamma$ gets smaller, Gas Pressure becomes lower.

Gas Volume:
Because the ratio of air-born water vapour becomes large, air that has high humidity has a smaller specific weight. Therefore, the quantity of dry air handled decreases, as may be expressed by the following equation:

$$Q_{dry} = Q_{wet} \times \frac{H - \mathcal{F} \cdot F}{H}$$

Where,
- $Q_{dry}$ = Amount of dry air contained within $Q_{wet}$ $(m^3/min)$
- $Q_{wet}$ = Quantity of wet air $(m^3/min)$

iii. Intermixing of Dust and Water

When large amount of dust and water are carried in the gas, Apparent Specific Weight of Gas can be a cause of increased Shaft Power.

The Apparent Specific Weight of Gas, $\gamma_2$, when large amounts of dust and/or water mix with the Gas Volume, can be calculated by the following equation:

$$\gamma_2 = \gamma_1 \times \left(1 + \frac{\text{Weight of water and/or dust per unit time (kg/min)}}{\gamma_1 \cdot \text{Weight of water and/or dust per unit time (kg/min)}}\right)$$

Where,
- $\gamma_1$ = Apparent Specific Weight of Gas $(kg/m^3)$
- $\gamma_2$ = Specific Weight of Gas $(kg/m^3)$

Shaft Power can be calculated from the following equation:

$$L_2 = L_1 \times \frac{\gamma_2}{\gamma_1}$$

Where,
- $L_1$ = Operating Shaft Power, with intermix of dust and/or water $(kW)$
- $L_2$ = Operating Shaft Power, Gas only $(kW)$
iv. Change in Composition of Handled Gas

Change in the gas composition handled by a fan can result in the Gas Specific Weight changing, with either of the following occurring:

When Specific Weight of Gas increases, Shaft Power and Gas Pressure increase. Fan operator should monitor the fan to prevent driver overload.

When Specific Weight of Gas decreases, Shaft Power and Gas Pressure decrease. Plant operator should consider affect of decreased pressure from the fan on the overall system performance.

v. Exhaust Applications

When a fan designed as a forced draft fan is used as an exhaust fan, a pressure deficiency may occur.

Fig. 22 shows both conditions; Gas Pressure Curve (P) representing a forced draft fan, and dotted Gas Pressure Curve (P') which represents the same fan used as an induced, or exhaust, draft fan.

The P'-Curve may be derived according to the following procedure:

a) Connect point A (Gas Pressure corresponding to Gas Volume Q), on the P-Curve with Origin, O, where Absolute Pressure is 0 and let the point where this line intersects the Atmospheric Pressure line be C.

b) Draw a vertical line connecting point A and Q, letting the point where this line intersects the Atmospheric Pressure line be B.

c) Connect point B with the Origin, O.

d) Draw a vertical line from point C, letting the point where this line and line A-Q intersect be D.

e) Drawing a line parallel to the Atmospheric Pressure line from point D, let the point here this line and line A-Q intersect be E.

f) Point E is the working point when the fan is used as an exhaust fan, and it Gas Pressure can be expressed by B – E.

g) Using the same procedure, from point A1, A2 ..., point E1, E2 ..., may be plotted, and the P'-Curve can be drawn.

From Fig. 22, we understand that AB < CD = BE, therefore P' < P. That is, Gas Pressure P', when the fan is used in exhaust function, is less than Gas Pressure P. Gas Pressure P' can also be calculated as follows:

\[
\frac{P}{P'} = \frac{Pa + P}{Pa}
\]

Or, \[P = \frac{Pa \times P'}{Pa - P'}\]

Where, \[Pa = \text{Atmospheric Pressure}\]
III. CHART OF IMPELLER TYPES

<table>
<thead>
<tr>
<th>Type</th>
<th>Performance</th>
<th>Efficiency</th>
<th>Features</th>
</tr>
</thead>
<tbody>
<tr>
<td>Airfoil</td>
<td>High</td>
<td>83 - 88%</td>
<td>Widest range of applications</td>
</tr>
<tr>
<td>F A</td>
<td>efficiency</td>
<td>80 - 85%</td>
<td>Plate liners can be easily installed</td>
</tr>
<tr>
<td>Linear</td>
<td>Relatively</td>
<td>60 - 75%</td>
<td>Relatively small type but can achieve same Pressure. Low dust adherence. Possibility of surging. Very low dust adherence. Easy liner replacement. Possibility of surging. also high noise.</td>
</tr>
<tr>
<td>Plate</td>
<td>Efficient</td>
<td>45 - 50%</td>
<td>Stainless Steel Paddle-Type impeller for high-temperature application.</td>
</tr>
<tr>
<td>Multi-Blade</td>
<td>Highest</td>
<td>45 - 60%</td>
<td>Highest performance of this type blower. Suitable for low speed operation. Low noise.</td>
</tr>
</tbody>
</table>

Airfoil Blades prior to impeller assembly

NIPPON STEEL (JAPAN) OITA WORKS
SINTER PLANT MAIN FAN
Anti-Wear Chromium-Carbide Lining
Airfoil Blades
Impeller Diameter: 4.8 meter
40,000m3/min x 2000mmAq
Motor: 900rpm x 14500kw

Stainless Steel Paddle-Type impeller for high-temperature application.

Impeller Back Sheet for same fan, 4.8m diameter