Industrial Fans

Delivery program
- Centrifugal Fans
- Axial Flow Impulse Fans
- Sound Protection
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The requirements imposed on Industrial Fans have noticeably increased over the years. The variety of problems that need to be tackled when handling gases requires a comprehensive range of fans to optimize the selection for each particular application.

Decades of intensive research and operating experience gained during this time are the basis for our fan range that provides the best economical choice for any application. Guiding factors for the development of this range have been:

- Low Investment Cost
- Low Operating Cost
- High Reliability
- Long Life
- High Noise Attenuation

Fans from the range have been supplied to the following industries:

- Steam Generators and Power Stations
- Centrifugal and Axial Induced Draft Fans
- Forced Draft Fans for all pressures
- Vapor Fans
- Primary Air Fans
- Dust Transporting Fans
- Booster Fans
- Recirculation Fans
- Hot and Cold GasFans
- Secondary Air Fans
- Sealing Air Fans
Cement Industries
Exhaust, Flue Gas and Forced Draft Fans
Cooling Air Fans
Pulverizer Fans
By-Pass Fans

Mining Industries
Mine Fans for use above or below ground. Centrifugal and axial flow fans for all air quantities.

Steel and Metallurgical Industries
Fans of all types for:
Sintering Systems (Sinter Plants)
Pelletizing Systems (Pellet Plants)
Direct Reduction Systems
Dry and Wet Particulate Removal Systems
Soaking Pits and Walking Beam Furnaces
Emergency Air Systems
Indirect Induced Draft Systems (Power Stacks)

Coke Oven Plants
Coke Gas Booster Fans, single and double stage, made of welded steel plate.

Marine Industries
Forced Draft Fans.

Glass Industries
Cooling Air Fans for Glass Troughs
Combustion Air and Exhaust Gas Fans

Mine ventilation fan
Volume flow \( \dot{V} = 383 \text{ m}^3/\text{s} \)
Depression \( \Delta p_{\text{Syst}} = 5400 \text{ Pa} \)
Speed \( n = 440 \text{ 1/min} \)
Sheft power \( P_{\text{sh}} = 2560 \text{ kW} \)
Product lines

Chemical Industries
Roasting Gas Fans
Recirculation Fans
Cooling Air Fans
Intermediate Gas Fans
Gas Fans
Fans for Calcining and Drying Processes
Fans for HCL Regeneration Systems
High Pressure Fan Systems
Process Steam Fans

Centrifugal Fans
Axial Flow Impulse Fans
(Ann Type Axial Flow Fans)
Silencers
Acoustic Insulation & Lagging
Sound Enclosures
Inlet Vane Controls
Support Structures
Indirect I.D. Fan Systems
(Power Stacks)
Emergency Air Systems

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Double width double inlet exhaust gas fan on an electro-melt furnace particulate removal system
Volume flow $\dot{V} = 126 \: \text{m}^3/\text{s}$
Temperature $t = 120 \: ^\circ\text{C}$
Pressure increase $\Delta p = 3820 \: \text{Pa}$
Speed $n = 990 \: \text{1/min}$
Shaft power $P_{sh} = 595 \: \text{kW}$

Double width double inlet emergency F. D. fan in a utility power plant.
Volume flow $\dot{V} = 350 \: \text{m}^3/\text{s}$
Pressure increase $\Delta p = 9320 \: \text{Pa}$
Speed $n = 990 \: \text{1/min}$
Shaft power $P_{sh} = 4100 \: \text{kW}$
Fan Designs

The fan range of TLT includes:

- Single or multiple stage centrifugal fans with maximum efficiency at pressures up to 80,000 Pa. Standard and heavy duty designs are available.
- Double width double inlet centrifugal fans for high pressure and large flow volume. Standard and heavy duty designs are available.
- Axial flow impulse fans with adjustable slotted flaps for high pressures at low tip speeds.

The above diagram shows a summary of the operating ranges for the various fan designs.

Indirect I.D. fans (power stacks) are often used for exhaust gases at temperatures above 5000C.
Control Modes and Characteristic Curves

Fan efficiencies around 90% will reduce operating costs to a minimum level. However, not only are the fan efficiencies at the maximum operating point or design point of importance but also efficiencies across the system operating range have great significance.

The most effective type of fan control is obtained by variation of the fan speed. Since speed control can only be achieved with high cost drive systems we commonly utilize inlet vane control for both centrifugal and axial flow fans. In the diagrams shown, the 100% point (\( V = 100\% \) and \( \Delta P_r = 100\% \)) represents the optimum point. For various reasons the optimum point may not always be identical with the design point.

Characteristic curves of a centrifugal fan with speed control

Characteristic curves of a centrifugal fan with inlet vane control
Model tests in the laboratory have provided the data needed to determine specific performance characteristics of our fans, in particular in view of the effects of different control systems. These performance characteristics enable the planner to predetermine the specific behavior of the fan in a system. Precise predictions can be made regarding the operation of fans operating as single units or in parallel.

If performance verification of large fans is required, tests can be conducted either in the field or in some cases on our test stand.

Characteristic curves of an axial flow impulse fan with inlet vane control.

Inlet vane control for a gas recirculation fan, largely gas-tight design; inlet diameter $D = 2730$ mm Ø
Design and Fabrication

With few exceptions, the large variety of available fan types nearly always permits direct coupling of the fan to the drive motor. We prefer this arrangement because system reliability is optimized by avoiding interconnecting equipment such as gear boxes, belt drives, etc. The basic design flexibility of our fans permitting alterations to or replacement of the impeller enables us to match actual operating conditions if it is found during operation that they differ from the conditions on which the original design data are based. Furthermore, slotted blade tip adjustment on centrifugal fan wheels or slotted flap adjustment on axial flow impulse fans are, in many cases, a simple means to meet specific operating conditions.

Double width double inlet I. D. fan for waste heat boiler, fan support of laterally flexible base frame design.
Volume flow $\dot{V} = 180 \text{ m}^3/\text{s}$
Temperature $t = 245 \text{ °C}$
Pressure increase $\Delta p = 4420 \text{ Pa}$
Speed $n = 990 \text{ 1/min}$

F.D. fan with inlet silencer in a steel mill.
Volume flow $\dot{V} = 77 \text{ m}^3/\text{s}$
Temperature $t = 30 \text{ °C}$
Pressure increase $\Delta p = 7260 \text{ Pa}$
Speed $n = 990 \text{ 1/min}$
Shaft power $P_{sh} = 650 \text{ kW}$
Efficiency $\eta = 84 \%$
Diameter $D = 2400 \text{ mm }\phi$

Axial flow impulse fan with slotted flap adjustment, shown during production.
Volume flow $\dot{V} = 660 \text{ m}^3/\text{s}$
Temperature $t = 156 \text{ °C}$
Pressure increase $\Delta p = 6520 \text{ Pa}$
Speed $n = 590 \text{ 1/min}$
Shaft power $P_{sh} = 5480 \text{ kW}$
Diameter $D = 4220 \text{ mm }\phi$

Inlet vane control:
Diameter $D = 4800 \text{ mm }\phi$
High gas temperature or particulate matter entrained in the gas stream require specific attention in fan selection and design. In such cases we often recommend emphasizing increased reliability in lieu of maximized efficiencies.

Axial flow impulse I.D. fan designed for vertical installation, shown in the manufacturing stage.

Impeller and shaft of an axial flow impulse fan during balancing operation.
Double width double inlet gas fan in electro-metallurgical plant.

Volume flow \( V = 195 \text{ m}^3/\text{s} \)
Temperature \( t = 230 \text{ °C} \)
Pressure increase \( \Delta p = 3720 \text{ Pa} \)
Speed \( n = 740 \text{ 1/min} \)
Shaft power \( P_{\text{sh}} = 875 \text{ kW} \)
Motor power \( P_{\text{m}} = 1100 \text{ kW} \)

Double width double inlet sintering gas fan in steel plant.

Volume flow \( \dot{V} = 366 \text{ m}^3/\text{s} \)
Temperature \( t = 160 \text{ °C} \)
Pressure increase \( \Delta p = 14200 \text{ Pa} \)
Speed \( n = 990 \text{ 1/min} \)
Efficiency \( \eta = 84 \% \)
Shaft power \( P_{\text{sh}} = 5900 \text{ kW} \)
Motor power \( P_{\text{m}} = 6500 \text{ kW} \)
Left: Impeller for a single stage F.D. fan
Volume flow \( \dot{V} = 4.8 \text{ m}^3/\text{s} \)
Temperature \( t = 20 \ ^\circ \text{C} \)
Pressure increase \( \Delta p_i = 31400 \text{ Pa} \)
Speed \( n = 2980 \text{ 1/min} \)
Diameter \( D' = 1250 \text{ mm} \ Ø \)
Impeller mass \( m_{\text{imp}} = 210 \text{ kg} \)

Right: Rotor for a two-stage gas fan
Volume flow \( \dot{V} = 1.03 \text{ m}^3/\text{s} \)
Temperature \( t = 100 \text{ °C} \)
Pressure increase \( \Delta p_i = 28500 \text{ Pa} \)
Speed \( n = 2980 \text{ 1/min} \)
Diameter \( D' = 865 \text{ mm} \ Ø \)
Impeller mass \( m_{\text{imp}} = 140 \text{ kg} \)
(both impellers)

Rotor for a mine fan
Design volume flow \( \dot{V}_{\text{des}} = 41.7 \text{ m}^3/\text{s} \)
(Maximum volume flow \( \dot{V}_{\text{max}} = 500 \text{ m}^3/\text{s} \))
Temperature \( t = 20 \text{ °C} \)
Depression \( \Delta p_{\text{sys}} = 5890 \text{ Pa} \)
Speed \( n = 420 \text{ 1/min} \)
Impeller diameter \( D'_{\text{imp}} = 5280 \text{ mm} \)
Impeller mass \( m_{\text{imp}} = 14000 \text{ kg} \)
Shaft mass \( m_{\text{sh}} = 9900 \text{ kg} \)
Rotor for a double width double inlet sintering gas fan.
Shaft attachment: Centerplate of impeller is bolted between flanges of a divided shaft, centered on a very small diameter.

Volume flow \( \dot{V} = 265 \text{ m}^3/\text{s} \)
Temperature \( t = 160 \text{ °C} \)
Pressure increase \( \Delta p_t = 16650 \text{ Pa} \)
Speed \( n = 990 \text{ 1/min} \)
Shaft power \( P_{sh} = 5230 \text{ kW} \)
Impeller mass \( m_{imp} = 11000 \text{ kg} \)
Shaft mass \( m_{sh} = 11000 \text{ kg} \)

Rotor for an axial flow impulse fan
(I. D. fan for a utility power station)

Volume flow \( \dot{V} = 660 \text{ m}^3/\text{s} \)
Temperature \( t = 156 \text{ °C} \)
Pressure increase \( \Delta p_t = 6520 \text{ Pa} \)
Speed \( n = 590 \text{ 1/min} \)
Diameter \( D = 4220 \text{ mm Ø} \)
Impeller mass \( m_{imp} = 12100 \text{ kg} \)
Shaft mass \( m_{sh} = 5200 \text{ kg} \)

(Hollow shaft)
Below and right, foreground: Rotor of SWSI (single width single inlet) flue gas fan for a steel mill. Torque transfer: hub shaft with key. Erosion protection: Bolted wear liners coated with wear resistant welds. Volume flow \( \dot{V} = 39.5 \text{ m}^3/\text{s} \)
Temperature \( t = 150 ^\circ \text{C} \)
Pressure increase \( \Delta p_t = 13550 \text{ Pa} \)
Speed \( n = 1145 \text{ 1/min} \)
Diameter \( D = 3030 \text{ mm} \)

Right, background: Rotor of DWDI (double width double inlet) flue gas fan for a cement kiln. Torque transfer: integral hub with body bound bolts. Volume flow \( \dot{V} = 125 \text{ m}^3/\text{s} \)
Temperature \( t = 350 ^\circ \text{C} \)
Pressure increase \( \Delta p_t = 6770 \text{ Pa} \)
Speed \( n = 990 \text{ 1/min} \)
Diameter \( D = 3160 \text{ mm} \)

Left hand side of the picture: Rotor coated with Saekaphen for a two-stage coke gas fan. Volume flow \( \dot{V} = 3.9 \text{ m}^3/\text{s} \)
Temperature \( t = 25 ^\circ \text{C} \)
Pressure increase \( \Delta p_t = 19650 \text{ Pa} \)
Speed \( n = 2970 \text{ 1/mm} \)
Diameter \( D = 1224 \text{ mm} \)

Right hand side of the picture: Rubber lined impeller for a flue gas fan behind a venturi scrubber. Volume flow \( \dot{V} = 17.6 \text{ m}^3/\text{ls} \)
Temperature \( t = 72 ^\circ \text{C} \)
Pressure increase \( \Delta p_t = 9810 \text{ Pa} \)
Speed \( n = 1485 \text{ 1/min} \)
Diameter \( D = 1874 \text{ mm} \)
SWSI (single width single inlet) fan, supported on both sides, during shop assembly
- Rotor of Incoloy
- Housing and inlet box are lead coated
- Dual fixed bearing system with flexible support structure

Two-stage F.D. fan for a waste gas combustor, the fan system consisting of two fans arranged in line with one common motor drive.

To minimize spare part requirements the rotors are identical in design (1st stage and 2nd stage).

Lead coated inlet box of the scrubber fan.
SWSI (single width single inlet) gas recirculation fan, supported on both sides, installed in a utility power plant.

Volume flow $\dot{V} = 167 \text{ m}^3/\text{s}$  
Temperature $t = 350 \text{ °C}$  
Pressure increase $\Delta p_t = 2360 \text{ Pa}$  
Speed $n = 720 \text{ 1/min}$  
Efficiency $\eta = 85.5 \%$  
Shaft power $P_{Sh} = 457 \text{ kW}$

SWSI (single width single inlet) gas recirculation fan, supported on both sides, installed in an utility power plant supported by integral base with vibration isolators.

Volume flow $\dot{V} = 142 \text{ m}^3/\text{s}$  
Temperature $t = 361 \text{ °C}$  
Pressure increase $\Delta p_t = 7850 \text{ Pa}$  
Speed $n = 990 \text{ 1/min}$  
Shaft power $P_{Sh} = 1360 \text{ kW}$  
Diameter $D = 3280 \text{ mm} \varnothing$
DWDI (double width double inlet) gas recirculation tank with concrete tilled integral base frame and vibration isolators.

Volume flow $\dot{V} = 124 \ \text{m}^3/\text{s}$
Temperature $t = 300 \ ^\circ\text{C}$
Pressure increase $\Delta p_t = 9615 \ \text{Pa}$
Speed $n = 1490 \ 1/\text{min}$
Shaft power $P_{sh} = 1410 \ \text{kW}$
Diameter $D = 2320 \ \text{mm} \ \varnothing$

DWDI (double width double inlet) gas recirculation tank during shop assembly.

Volume flow $\dot{V} = 250 \ \text{m}^3/\text{s}$
Temperature $t = 340 \ ^\circ\text{C}$
Pressure increase $\Delta p_t = 3470 \ \text{Pa}$
Speed $n = 715 \ 1/\text{min}$
Shaft power $P_{sh} = 1100 \ \text{kW}$
Motor power $P_m = 1300 \ \text{kW}$
Diameter $D = 3200 \ \text{mm} \ \varnothing$
SWSI (single width single inlet) raw mill fan for the cement industry, supported on one side (AMCA arrangement 8).
Volume flow $\dot{V} = 114 \text{ m}^3/\text{s}$
Temperature $t = 90 \, ^\circ\text{C}$
Pressure increase $\Delta p_t = 5700 \, \text{Pa}$
Speed $n = 745 \, \text{1/min}$
Impeller diameter $D_{\text{imp}} = 3350 \, \text{mm} \, \phi$

SWSI (single width single inlet) cement kiln exhaust gas fan, supported on one side (AMCA arrangement 8), fan support of laterally flexible base frame design.
Volume flow $\dot{V} = 133 \, \text{m}^3/\text{s}$
Temperature $t = 100 \, ^\circ\text{C}$
Pressure increase $\Delta p_t = 4600 \, \text{Pa}$
Speed $n = 745 \, \text{1/min}$
Our economical production facilities are equipped with modern machinery. For example, a numerically controlled flame cutting machine and a metal spinning machine are used for the processing of sheet metal.

NC flame cutting machine, with punch tapes automatically produced via electronic data processor.

Balancing machine for fan rotors up to 30000 kg and 5000 mm Ø

Metal spinning machine for radii of inlet nozzles, impeller side plates and spinning flanges.
A selection of various designs from our fan range is shown below.

SWSI (single width single inlet) forced draft fan, supported on one side (AMCA arrangement 8), with inlet vane control, arranged on an integral supporting structure with vibration isolators.

DWDI (double width double inlet) fan arranged on an integral supporting structure with vibration isolators.
SWSI (single width single inlet) sintering gas fan, rotor supported on both sides, with fluid drive, fan supported by an integral steel frame.

Two-stage coke gas fan with oil lubricated roller bearings sealed with carbon packing glands. Speed: 2950 1/min.

Axial flow impulse fan (induced draft fan) with inlet vane control and possibility of adjusting the slotted flaps individually during down-time.
Fan Inquiry:

As a fan manufacturer TLT works closely with the architect, engineer and/or the final user to optimize fan selection for each specific application.

The following set of conditions at fan inlet to be supplied by the customer provides the basis for fan selection and design:

1. Identification of the plant and the process in the system for which the fan is to be used:
2. Volume flow: \( \dot{V} \) or \( \dot{V}_{sc} \) **) in \( \text{m}^3/\text{s} \)
3. Temperature: \( t \) in \( ^\circ\text{C} \)
4. System-related pressure increase: \( \Delta p_{\text{Syst}} \) in \( \text{Pa} \)
   \[ \Delta p_{\text{Syst}} = P_t4 - P_t1^\ast \]
   (Pressure distribution: Fan suction side / discharge side)
5. Fan inlet pressure measured against barometric pressure (+/-) \( P_1 \) in \( \text{Pa} \)
6. Elevation: \( h \) in \( \text{m} \) above see level
7. Mains frequency (Standard frequency): \( f_{\text{mains}} \) in \( \text{Hz} \)
8. Permissible noise level:
   Sound pressure level at a defined distance: \( L_{\text{allow}} \) in \( \text{dB (A)} \)
   or sound power level: \( L_{\text{Wallow}} \) in \( \text{dB (A)} \)
9. Information on the fluid handled:
   9.1 Type of gas:
   9.2 Density of gas: \( \varrho \) or \( \varrho_{sc} \) **) in \( \text{kg/m}^3 \)
      (if necessary supply gas analysis with moisture content)
   9.3 Particulate content of the gas: \( St \) or \( St_{sc} \) **) in \( \text{g/m}^3 \); \( \text{mg/m}^3 \)
   9.4 Dust characteristics:
      S: Probability of build-up
      V: Probability of erosion
   9.5 Corrosion:
      K: Probability of corrosion due to
10. Type of preferred fan
   (see following sketches and explanation)
11. Additional information:

Notes:

*) See also section “Pressure Definitions”.
"The system-related pressure increase \( \Delta p_{\text{Syst}} \), as defined by us, is often referred to as “static pressure increase \( \Delta p_{\text{stat}} \), the dynamic pressure components having been ignored, however.

**) The index “sc” identifies the standard condition \( t = 0^\circ\text{C}, p = 101325 \text{ Pa} \).

Types of Fan Design and Installation

Radial-flow fans are normally of the single-inlet type up to approx. 100 \( \text{m}^3/\text{s} \); in some cases the double-inlet type is used from 60-70 \( \text{m}^3/\text{s} \) already. The actual limit between single- and double-inlet type is mainly determined by the relevant case of need, the suitable fan type, the required ratio volume/specific energy and the speed.

We built single-stage radial-flow fans for a specific energy of more than 40000 \( \text{J/kg} \). It depends on the case of need, temperature, volume/pressure ratio and the possible speed from what pressure increase the fan has to be of the double-stage type.

The most favourable installation of the fans is that on an elevated concrete substructure. This results in short bearing pedestals and the motor can be placed on a low frame or even directly on plates which are embedded in the concrete. This simple, rugged kind of installation is less susceptible to vibration and therefore best suited to sustain high imbalance forces due to wear or dust caking.

If the substructure has to be made of steel corresponding plate cross sections have to be used in case of large base-to-centre heights to achieve a sufficient vibration resistance. Thus the fan weight and the costs are increased accordingly.

Fans which have to be installed on substructures susceptible to vibrations, as e.g. in the structure or on a building roof have to be vibration-isolated.
Elevated concrete base without frame below the motor - rugged, simple installation.

The vibration isolators are installed under the "hard" points such as housing walls. The installation of the fan on an elevated concrete base resting on vibration isolators is recommended if higher imbalances are expected.

The vibration isolators reduce the amplitudes of the dynamic forces (alternating loads from the imbalance rotating with the fan rotor). The so-called isolation efficiency depends on the distance of the exciter frequency (= fan speed) and the natural frequency of the spring mass system of vibration isolator - machine mass. Normally, the isolation efficiency is above 90%. For speed-controlled fans it is therefore absolutely necessary to indicate the lowest required (reasonable) speed, as then accordingly "soft" springs have to be used.

Some installation examples:

- Elevated concrete base without frame below the motor - rugged, simple installation.
- Vibration-isolated installation with elevated concrete block on vibration dampers.
- Elevated concrete base with low frame.
- Compact type of construction, selfsupporting structure, placed directly on vibration dampers.
Explanation of Common Fan Terms and Special Problems

The following definitions of fan terminology may facilitate communication between the fan manufacturer and customer.

Meaning of Symbols:

- **p**: Pressure
- **Δp**: Pressure Difference or Increase
- **ρ**: Density
- **V**: Volume Flow at Inlet Condition
- **A**: Cross Sectional Area
- **l**: Length
- **m**: Mass Flow
- **c**: Gas Velocity
- **g**: Gravity
- **f**: Compressibility Factor
- **Y**: Specific Delivery Work
- **P**: Power
- **η**: Efficiency
- **T**: Absolute Temperature (Kelvin)
- **χ**: Adiabatic Exponent

Indexes:

- **t**: total
- **s**: static
- **d**: dynamic
- **1, 2, 3, 4**: markings of the cross sections concerned (terminal points)

Pressure Definitions

The energy transmitted through the fan impeller to the volume flow is needed to overcome the system resistances. These resistances can comprise the following:

- Friction losses
- Back pressure from pressurized systems
- Velocity changes at system inlet and outlet and within the system
- Draft forces due to density differentials
- Geodetic head differentials which are mostly negligible.

The sum of the above mentioned resistances as far as they occur in the system concerned, represent the total system resistance. According to Bernoulli this total resistance is to be understood as total pressure. This pressure p (analogy: total energy) comprises static pressure ps (analogy: potential energy) and the dynamic pressure pd (analogy: kinetic energy).

\[ p = p_s + p_d \]

(This definition is in accordance with VDMA-standard 24161)

To move the design volume flow the fan must generate within the specified fan terminal points - a pressure increase equivalent to the total resistance of the system. This equivalent pressure increase is defined by us as **system-related pressure increase** \( \Delta p_{\text{syst}} \).

\[ \Delta p_{\text{syst}} = \Delta p_{\text{d4}} - \Delta p_{\text{s1}} \]

If the cross sections 1 and 4 (Figure D-1) represent the terminal points of the fan the system-related pressure increase \( \Delta p_{\text{syst}} \) will then be the difference of the total pressures at the terminal points 1 and 4. This system-related pressure increase which is in fact the pressure increase required by the customer is often incorrectly still referred to as **static pressure increase** \( \Delta p_{\text{stat}} \). Unfortunately ignoring the dynamic pressure difference existing in most cases. The probable cause of disregarding this dynamic pressure increase lies in the generally used method of measuring the static pressures in ducts by means of holes drilled in the duct walls perpendicular to the direction of flow. In such cases the dynamic pressure differential at the terminal points has to be added to the result of the static pressure measurement to obtain the correct system-related pressure increase. In cases where no specific data relative to gas velocities, desired duct cross sections or special installation requirements such as for mine fans are given we will determine the fan based on the assumption that the cross sections \( A_1 \) and \( A_4 \) are equal and the total pressure increase between these terminal points represents the system-related pressure loss \( \Delta p_{\text{syst}} \) stated by the customer.

\[ A_1 = A_4 \]

\[ \Delta p_{\text{syst}} = \rho_{\text{d4}} - \rho_{\text{s1}} \]

With \( A_1 = A_4 \) and disregarding compressibility it follows that \( p_{\text{d4}} = p_{\text{s1}} \) and therefore:

\[ \Delta p_{\text{syst}} = (\rho_{\text{d4}} + p_{\text{d4}}) - (\rho_{\text{s1}} + p_{\text{s1}}) \Delta p_d \]

The **dynamic pressure** \( p_d \) is understood to be based on the average gas velocity in a cross section.

\[ p_d = \frac{g_2}{2} \cdot c^2 \]

Example: Dynamic pressure at terminal point 4:

\[ p_{d4} = \frac{g_2}{2} \cdot c_4^2 = \frac{g_2}{2} \left( \frac{V}{A_4} \right)^2 \cdot f^2 \]

Pressure losses \( p_v \) caused by fan components between the cross sections \( A_1 \) and \( A_4 \), for example inlet box, louver, inlet vane control, diffuser will be considered by us when sizing the fan.

The design pressure \( \Delta p \), the parameter determining fan size, is the sum of the system-related pressure increase and the pressure losses of the fan components.

\[ \Delta p = \Delta p_{\text{syst}} + p_v = \rho_{\text{d4}} - \rho_{\text{s2}} \]

Our characteristic curves show the **design pressure** \( \Delta p \) as this pressure differential represents a parameter defined by tests for a specific fan type at given operating conditions, whereas the system-related pressure increase \( \Delta p_{\text{syst}} \) varies with the losses \( p_v \) which arise depending on the fan components used.

In determining the fan design pressure \( \Delta p \) other losses are considered in addition to the above mentioned pressure losses \( p_v \) if special design conditions are specified involving inlet and outlet pressure losses, e. g. pressure losses in the case of silencers and turning bends, outlet pressure losses in the case of mine fans etc.
Fan Power and Efficiency
The fan design pressure $\Delta pt$ of our fans is equal to the total pressure increase between the cross sections $A_2$ and $A_3$.

With this design pressure, and the design volume flow $\bar{V}$ at fan inlet conditions, the power requirements at the fan shaft $P_{Sh}$ and the efficiency $\eta$ will be determined, optimum inlet conditions being a prerequisite.

$$P_{Sh} = \frac{\bar{V} \cdot \Delta pt \cdot f}{\eta} = \frac{\dot{m} \cdot Y}{\eta}$$

- $\bar{V}$ in m$^3$/s
- $\Delta pt$ in Pa = N/m$^2$
- $f < 1$
- $\eta < 1$
- $\dot{m}$ in kg/s
- $Y$ in J/kg = Nm/kg
- $P_{Sh}$ in W = Nm/s

Operating a fan at a temperature significantly below design temperature will cause the shaft power requirement to increase as a function of density or as a ratio of the absolute temperatures. Should a different gas with higher density be handled the shaft power requirement will increase with the ratios of the densities.

$$P_{Sh \text{ operation}} = \frac{T_{design}}{T_{cold}} \cdot P_{Sh \text{ design}} = \frac{\dot{g}_{\text{alternative gas}}}{\dot{g}_{\text{design}}} \cdot P_{Sh \text{ design}}$$

Since such operating conditions often occur at start-up the louvers or inlet vanes need to be closed in these cases. The pressure increase of the fan will also rise as a function of lower temperatures or higher gas densities. This must be considered when designing flues and ducts, expansion joints, etc.

**Influences of Temperature and Density**
A noticeable **temperature rise** will occur across fans with high pressure increase, in particular when the fan operates in a throttled condition and at low efficiency. The adiabatic temperature increase $\Delta t_{ad}$ is:

$$\Delta t_{ad} = T_2 \cdot (p_2/p_1)^{\gamma-1} - 1$$

The real temperature increase is:

$$\Delta t = \frac{\Delta t_{ad}}{\eta} \approx \frac{\Delta pt}{1250 \cdot \eta}$$

- $\Delta pt$ in Pa
- $\Delta t$ in °C
- $\eta < 1$

**Influence of Mass and Mass Inertia**
Erosion, corrosion and system-related contamination causing build-up can possibly lead to imbalances due to uneven mass distribution. For such cases impellers with large masses are advantageous because the shifting of the gravity center caused by the imbalance is smaller.

For the start-up time of a fan the determining factor is the inertia of the rotating mass. This start-up time is of importance relative to temperature increases of the electrical drive system causing limitations of the number of system start-ups.

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**Figure D-1: Marking of the Cross Sections and Reference Planes**
Examples of Various Fan Arrangements in a System with Corresponding Pressure Distribution (Figures D-2 to D-4)

\[ \Delta p_t = \Delta p_{\text{sys}} + \sum p_v \]
\[ \Delta p_{\text{sys}} = \Delta p_{\text{sys, s}} + \Delta p_d \]

Meaning of Symbols Used in Equations:
- \( \Delta p_t \): Total Pressure Increase Between Cross Sections 2 and 3
- \( \Delta p_{\text{sys}} \): System Resistance-Related Total Pressure Increase
- \( \Delta p_{\text{sys, s}} \): System Resistance-Related Static Pressure Increase
- \( p_d \): Dynamic Pressure (Velocity Pressure)
- \( \Delta p_d \): Dynamic Pressure Difference (Velocity Pressure Difference)
- \( p_v \): Pressure Loss(es)
- \( p_{v, \text{in}} \): Pressure Loss at Fan Inlet
- \( p_{v, \text{inlet}} \): Inlet Box Pressure Loss
- \( p_{v, \text{dif}} \): Diffuser Pressure Loss
- \( p_{v, \text{out}} \): Outlet Pressure Loss
- \( A \): Cross Section

Indexes:
1, 2, 3, 4: Marking of Cross Sections Concerned (Terminal Points)

Behavior of:
- Total Pressure
- Static Pressure
- Dynamic Pressure (Velocity Pressure)

Figure D-2: Open Inlet Fan (e.g. Forced Draft Fan)
Figure D-3: Fan Connected to Ducts at Fan Inlet and Discharge (e.g. Induced draft Fan)

Figure D-4: Exhaust Fan (e.g. Mine Fan)
Questions Regarding Fan Noise

Discharge silencer for two centrifugal forced draft fans, designed as absorption type silencer, level reduction by 15 dB. Fan data:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volume flow</td>
<td>$2 \times 62 \text{ m}^3/\text{s}$</td>
</tr>
<tr>
<td>Temperature</td>
<td>$50 \degree \text{C}$</td>
</tr>
<tr>
<td>Pressure increase</td>
<td>$8120 \text{ Pa}$</td>
</tr>
<tr>
<td>Speed</td>
<td>$1490 \text{ 1/min}$</td>
</tr>
</tbody>
</table>

'Below:

Double width double inlet centrifugal forced draft fan with disc silencers and cover for noise treatment of the fan inlet noise (shown during shop assembly).

1. First Fundamentals
With progressing industrialization man is faced with increasing environmental problems. Noise emitted by fans belongs in this category. The following will give guidance in the problem area of noise emitted by fans as well as the flow in the connecting flues and ducts.

The sound ("Schall") perceived by the human ear is the result of oscillation of particles of an elastic medium in the frequency range of about 16 to 16,000 hertz (Hz). One hertz is one oscillation per second. Depending upon the medium in which the sound travels we distinguish air sound, body sound, and water sound ("Luftschall, Körperschall, Wasserschall").

A pitched tone ("Ton") is defined as sound oscillating as a sinusoidal function (compression and depression). With increasing amplitude sound will be perceived as being louder and with increasing frequency it will be perceived as being higher. The tone in Figure 2 (sound pressure $P_2$) is perceived as higher and generally louder than the tone in Figure 1 (sound pressure $P_1$).

For additional details see paragraph 2.

A clang ("Klang") is created by the harmonic interaction of several tones. Noise ("Geräusch", "Rauschen") is defined as statistical sound pressure distribution across the perceivable frequency range. A noise annoying the human ear is called an “excessive noise” ("Lärm").

\[1\) The terms in square brackets [ ] are the equivalent German words.
2. Human Noise Perception

Sound pressure is exactly measurable with instruments. The physiological effect on humans is much more difficult to determine. The human ear, for example, will perceive two tones of equal sound pressure yet different frequency as unequally loud.

Numerous tests were made on listeners in order to compare the loudness of tones at different frequencies and different sound pressures with those of a 1000 Hz tone. In particular the objective was to identify the sound pressure \( p \) (measured in dB) at a frequency of 1000 Hz at which the sound pressure \( p \) (measured in Hz) would evoke the same perception in the listener with respect to loudness. As a result of these tests, curves of constant loudness (stated in phone) were identified over the frequency range. By definition sound pressure level and loudness coincide in terms of figures at 1000 Hz. Graphs in Figure 3 show these curves of equal loudness.

Because the shape of the curves changes with frequency as well as sound pressure one was faced with the problem of designing a handy measuring instrument for an objective measuring of the loudness of sound. This was the impetus behind the search for a different evaluation system. An additional reason lies in the fact that the phon curves can only be used to evaluate single tones. There is, however, a difference between the human ear’s perception of single tones and its perception of noise.

The solution, which takes these factors into account and which has been internationally accepted, is found in the so-called “A” sound evaluation curve. The curve represents an approximation of the phon curve in midrange of the sound pressure level. To give consideration to the fact that single tones are perceived as being more annoying than broad band noise, a higher reduction is imposed on single tones in addition to the total noise level requirements, for example such a typical single tone is the “blade passing tone” ["Schaufelton"] of a fan whose frequency is calculated with the number of blades and the fan speed expressed in Hz. This blade passing tone and its integer multiples (harmonics) form the so-called “blade passing frequencies” ["Drehklang"].

3. Fundamentals of Acoustics (Definitions)

Units of Sound Parameters

In acoustics it is common to work with levels, i.e. it is common not to use the original parameters with their corresponding units, but logarithmical parameter ratios using the logarithm to the base 10, the corresponding units being B (B) or decibel (dB).

\[
\text{level} = \lg \frac{\text{effective value}}{\text{reference value}} \quad \text{in B}
\]

\[
\text{level} = 10 \lg \frac{\text{effective value}}{\text{reference value}} \quad \text{in dB}
\]

As the same units are applied to all sound parameter levels it is important to properly identify the type of the sound parameter level referred to, that means to distinguish, for example, between sound pressure level and sound power level.

Sound Pressure Level \( L \) ["Schalldruckpegel" L]

The sound pressure level \( L \) (most commonly also called sound level) quantifies the sound pressure measured at a specific point.

By definition:

\[
L = 10 \log \frac{p}{p_0} = 20 \log \frac{p}{p_0} \quad \text{in dB}
\]

with \( p = \) effective value of sound pressure at measuring point in N/m²

\[
p_0 = 2 \times 10^{-5} \text{ N/m}^2 = 20 \mu \text{ Pa} = 2 \times 10^{-4} \mu \text{ bar}
\]

(reference sound pressure, the audible threshold for 1000 Hz pitch)

Evaluated Sound Pressure Level \( L_A \)

\( (= \text{Sound Pressure Level Evaluated According to Evaluation Curve "A"}) ["Bewerteter Schalldruckpegel" L_A] \)

The evaluated sound pressure level \( L_A \) - expressed in dB (A) - is obtained by adding at the various frequencies a \( \Delta L \) from the evaluation curve “A” (see Figure 4) to the measured sound pressure level \( L \) at the corresponding frequencies. The evaluation curve and the evaluation procedure are defined in DIN standard 45633, sheet 1.

\[29\]
Baffles of the absorptive discharge silencer of a forced draft fan (after approximately 11,000 operating hours); fan performance data are shown on the right. Below:

Close up photograph of the baffle wall—after approximately 11,000 operating hours.

Three-stage absorptive silencer for ambient air inlet to a forced draft fan (Volume flow $\dot{V} = 433$ m$^3$/s, pressure increase $\Delta p = 8250$ Pa) in a power station. Design point $\dot{V} = 60\%$ Attenuation to sound pressure level 70dB (A).
Discharge silencer designed as a resonant silencer ($\frac{\lambda}{4}$-silencer or interference silencer) for two induced draft fans in a power plant (volume flow $\dot{V} = 2 \times 660 \text{ m}^3/\text{s}$, pressure increase $\Delta p = 6520 \text{ Pa}$), insertion loss = 33 dB at the frequencies of 118/236 Hz (blade passing frequency and first harmonic).

Baffles (shown at center right) and baffle walls (shown below) of the above silencer installation after approximately 11,000 operating hours.
As can be seen in Figure 4, the numerical values for LA are significantly below the L values at low frequencies and have a much smaller impact at higher frequencies.

**Measuring Surface Sound Pressure Level L and LA**

["Meßflächen-Schalldruckpegel" L und LA]

The measuring surface sound pressure level L (= the sound pressure level at the enveloping measuring surface) is defined as the energetic mean\(^1\) of multiple sound level measurements over the measuring surface S with elimination of extraneous noise and room effects (reflections).

LA is the “A” evaluated measuring surface sound pressure level. The measuring surface S is an assumed area encompassing the sound source at a defined distance (mostly one meter). This enveloping surface comprises simplified surfaces such as spherical, cylindrical and square surfaces generally following the shape of the sound producing equipment.

\[^1\] To calculate the energetic mean of all sound level measurements taken over the envelope surface (taking into account the time interval of testing) the following formula is to be used:

\[
\bar{L} = 10 \log \left( \frac{1}{n} \cdot \sum_{i=1}^{n} 10^{0.1 L_i} \right)
\]

If the difference between the individual sound levels is smaller than 6 dB the formula below can be used as an approximation representing the arithmetical mean:

\[
\bar{L} = \frac{1}{n} \cdot \sum_{i=1}^{n} L_i
\]

Any components that protrude beyond the surface but contribute little to the emission can be neglected. Sound reflecting boundaries, such as floors and walls, are not incorporated within the measuring surface. The measurement points shall be sufficient in number and evenly distributed over the enveloping surface. The number depends on the size of the sound source and the uniformity of the sound field.

Because of the logarithmic parameter ratios used in acoustics the measuring surface in m\(^2\), will be related to a reference area to define the measuring surface level LS ["Meßflächenmaß" L\(_S\)] as the characteristic parameter:

\[
L_S = 10 \log \frac{S}{S_0} \text{ in dB}
\]

S = Measuring surface in m\(^2\)

S\(_0\) = 1 m\(^2\) (reference area)

**Sound Power Level L\(_W\)**

["Schall-Leistungspegel" L\(_W\)]

The value of the total sound power radiating from a sound source is given by the sound power level L\(_W\).

\[
L_W = 10 \log \frac{W}{W_0} \text{ in dB}
\]

W = gas-borne acoustical power emitted as air sound in watts

W\(_0\) = 10\(^{-12}\) watts (reference sound power at audible threshold at 1000Hz)

**Evaluated Sound Power Level L\(_{WA}\)**

["Bewerteter Schall-Leistungspegel" L\(_{WA}\)]

When an evaluation, similar to the one described in the example, of the sound pressure level is conducted, using the evaluation curve “A”, the evaluated sound power level L\(_{WA}\) will be obtained from the sound power level L\(_W\).

**Relationship Between Sound Pressure Level and Sound Power Level**

The sound power W is not measured directly but is calculated using the measured sound pressure p, sound particle velocity v (molecular movement velocity), ["Schallschnelle" v] and the measuring surface S:

\[
W = p \cdot v \cdot S
\]

using

\[
\rho = \text{air density} \quad c = \text{air sound velocity}
\]

it follows that:

\[
W = \frac{p^2}{\rho \cdot c} \cdot S
\]

Assuming that \(\rho = \text{constant}\) \(c = \text{constant}\)

the proportional relationship obtained is:

\[
W \propto p^2 \cdot S
\]

In terms of expressing the above equation in acoustic level parameters the following important equations can be obtained:

\[
L_W = L + 10 \log \frac{S}{S_0} = L + L_S \text{ in dB}
\]

\[
L_{WA} = L + 10 \log \frac{S}{S_0} = L + L_S \text{ in dB}
\]

The sound power level L\(_W\) can be approximated by the sum of the measuring surface sound pressure level L and the measuring surface level L\(_S\).

From the above relationship it can be deduced that with a given sound power level a spherical or a semispherical sound dispersion (ideal sound dispersion) the sound pressure level will diminish by 6 dB for every doubling of the distance from the sound source.

Through absorption of the sound in the air and on the ground this value will increase and through reflection of the sound by obstructions it will be reduced. Furthermore, weather conditions can cause either an increase or decrease of the sound pressure level reduction.
### Sound Intensity Level $L_I$

["Schall-Intensitätsspegel" $L_I$]

At this point mention should be made of the so-called **sound intensity $I$**

["Schall-Intensität" $I$] which is the sound power relative to the reference area of 1 m$^2$.

$$I = \frac{W}{S} \text{ in } \frac{\text{watts}}{m^2}$$

With this definition an analogy to electrical technology can be made:

The sound intensity is proportional to the square of the sound pressure.

The definition of the corresponding **sound intensity level $L_I$** is as follows:

$$L_I = 10 \log \left( \frac{I}{I_0} \right) \text{ in dB}$$

with $I_0 = 10^{-12}$ watts/m$^2$

(reference sound intensity)

### 4. Sound Analysis

The “total sound level” or “sum sound level” of noise is derived from the logarithmic addition of a multiple of single sound levels at different frequencies (Figure 5). In order to perform noise measurements, the audible frequency range has been divided into 10 octave bands.

The width of the octave is identified such that the ratio of the upper limiting frequency of the spectrum $f_0$ to the lower limiting frequency $f_U$ is 2:1.

Octave: $$\frac{f_0}{f_U} = 2$$

The corresponding ratio for the "terz" is:

"Terz": $$\frac{f_0}{f_U} = \sqrt[3]{2}$$

Three “terz” together make up an octave.

Center frequencies are determined by:

$$f_m = \sqrt{f_U \cdot f_0}$$

Octave: $$f_m = \sqrt[3]{f_U \cdot f_0} = \frac{f_0}{\sqrt[3]{2}}$$

"Terz": $$f_m = \sqrt[6]{f_U \cdot f_0} = \frac{f_0}{\sqrt[6]{2}}$$

---

<table>
<thead>
<tr>
<th>Acoustics</th>
<th>Electrotechnics</th>
</tr>
</thead>
<tbody>
<tr>
<td>$W = p \cdot v \cdot S$</td>
<td>$N = \text{Output}$</td>
</tr>
<tr>
<td>$\frac{W}{S} = I$</td>
<td>$U = \text{Voltage}$</td>
</tr>
<tr>
<td>$I = p \cdot v$</td>
<td>$I = \text{Amperage}$</td>
</tr>
<tr>
<td>$\nu = \frac{p}{\varrho \cdot c}$</td>
<td>$R = \text{Resistance}$</td>
</tr>
<tr>
<td>$I = \frac{p^2}{\varrho \cdot c} = \nu^2 \cdot \varrho \cdot c$</td>
<td>$N = \frac{U^2}{R} = I^2 \cdot R$</td>
</tr>
</tbody>
</table>

The sound intensity is proportional to the square of the sound pressure.

---

![Sound Pressure Level in dB](image)

---

The individual center frequencies of the octave band are at:

<table>
<thead>
<tr>
<th>Center Frequency</th>
<th>Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>31.5</td>
<td>1,000</td>
</tr>
<tr>
<td>63</td>
<td>2,000</td>
</tr>
<tr>
<td>125</td>
<td>4,000</td>
</tr>
<tr>
<td>250</td>
<td>8,000</td>
</tr>
<tr>
<td>500</td>
<td>16,000</td>
</tr>
</tbody>
</table>
In practical application, the first and last octave bands mostly play a secondary role. Commercially available sound measurement instruments to measure sound levels in dB and dB (A) are equipped with adjustable octave and "terz" filters to conduct frequency analyses. If the octave band analysis proves inadequate the more selective "terz" analysis should be employed, the octave band width being divided into 3 "terz" band widths.

In the case of single tones or noises extending over one "terz" band only, the "terz" band and the octave band analyses will give the same figures.

The example in Figure 5 shows an octave band and "terz" band analysis. For a more selective analysis of a noise spectrum, narrower band filters can be employed to further divide the noise spectrum (search tone analyzer).

5. Addition of Levels
To determine the total level \( L_{tot} \), partial levels \( L_i \) (sound pressure level or sound power level) will be added in accordance with the following equation:

\[
L_{tot} = 10 \lg \sum_{i=1}^{n} 10^{0.1 L_i}
\]

When adding sound pressure levels it must be considered that all individual sound pressure levels have one common reference point.

In the specific case where \( n \) sound sources have equal sound power \( W_1 \), the total level \( L_{W10} \) can be determined by the following:

\[
L_{Wtot} = L_{W1} + 10 \lg n
\]

For a number of sound sources the level increase can also be determined using the diagram in Figure 6.

In the special case of two superimposed single tones of equal sound pressure \( p_1 \), equal frequency \( f_1 \), and equal phase \( \phi \) to be added requires special consideration. Deviating from the above described summation a total level \( 6 \) dB higher than the sound pressure level of the single tone will be obtained:

\[
L_{tot} = 10 \lg \left( 2 \cdot \frac{p_1}{p_0} \right)^2 = 20 \lg 2 \cdot \frac{p_1}{p_0}
\]

\[= L_1 + 20 \lg 2 \]

If a difference in phase of 180 degrees exists (\( \phi = 0^\circ \); \( \phi' = 180^\circ \)) or if interference results, the tones eliminate each other (see Figure 8).

With reference to the diagram in Figure 7 it can be seen that for a level difference of more than 10 dB practically no level increase will result. For the special case of two sound sources with equal levels (level difference \( 0 \)) a level increase of 3 dB will result (see also Figure 6).

The case where two superimposed single tones of equal sound pressure \( p_1 \), equal frequency \( f_1 \), and equal phase \( \phi \) are to be added requires special consideration. Deviating from the above described summation a total level \( 6 \) dB higher than the sound pressure level of the single tone will be obtained:

6. Noise Development in Fans
The operating noise of a fan comprises various sound components.

In boundary zones of confined fast moving gas flow, eddy currents occur as the result of the influence of the viscosity of the gas. On fans these eddy currents occur at the blade discharge edges. The resulting noise caused by the rotating impeller is referred to as "eddy current noise" and is considered the "primary noise". Superimposed on this noise is the "self-noise of highly turbulent flow" in the fan housing and ducts. "Eddy current noise" and "self-noise" ("Wirbelgeräusch" und Strömungsrauschen") display a broad band frequency spectrum. The sound power increasing approximately with the 5th to 7th power of the impeller tip speed. In addition to broad band noise, "pulsation noise" ("Pulsationsgeräusch") occurs at different frequencies caused by periodical pressure oscillations of the medium due to the relative movement between the impeller and a stationary fixture exposed to the flow. "Pulsation noise" will occur when the flow in the operating at almost equal speed in a common duct system. In this case sound wave superposition results in periodic sound level variations called beats. The beat frequency is determined by the difference in the operating speeds of the two fans.
7. Sound Pressure- and Sound Power Level of Fans
The sound pressure level 

\[ L_{PSL} = 10 \log \left( \frac{V_0}{V_0} \right) + 20 \log \frac{p}{p_0} \text{ in dB (A)} \]

whereby:

- \( p \): Total pressure difference in \( \mu \) bar
- \( p_0 \): \( 100 \mu \) bar
- \( V \): Volume in m³/hr
- \( V_0 \): \( 11 \) dB (A) for centrifugal fans with curved, backward inclined blades.
- \( K \): \( 16 \) dB(A) for axial flow fans.

The total sound power \( W \) produced by the fan or the respective sound power level \( L_w \) are used as the basis for determining the sound propagation of fan noise. Relative to noise radiation to the surroundings it is necessary to differentiate between

- the primary sound power radiating with the gas stream through the fan outlet/inlet area ("gas-borne sound")
- and the secondary sound power radiating from the fan components ("body sound") being excited by the sound energy of the gas stream.

Primarily emitted sound powers are \( W_s \) and \( W_t \).

\[ L_{PSL} = L_i + 10 \log \frac{S}{S_0} \]

where, for the respective individual components under consideration:

- \( i = S, SL, D, DL, G, U \)

With the example of a forced draft fan (with and without sound protection) Figures 10.1 through 10.4 graphically depict the various sound components.

For primary and secondary sound sources the sound power levels are symbolized by arrows of different color and corresponding sound power levels are symbolized by arrows of different lengths.

8. Sound Protection
The noise generated by the fan can be reduced through the installation of sound enclosures or acoustical insulation and lagging, on the one hand ("sound insulation") and silencers on the other ("sound attenuation").

When a silencer ("Schalldämpfer") is installed, the sound propagation in the duct system is reduced without essential influence on the gas stream. (Values \( L_{WS} \) and \( L_{WD} \) are reduced by converting sound energy to thermal energy.)

The installation of acoustic insulation and lagging ("Schallisolierung") or sound enclosures ("Schallhauben") provides extensive protection to the area surrounding the fan from the propagation of air sound caused by the structure-borne sound (body sound) of the fan components excited by the sound energy of the gas stream. (Values \( L_{WS}, L_{WD}, L_{WSL} \) are...
reduced by the reflection of sound energy back to the noise source and in addition partially by conversion to thermal energy.

Depending on individual requirements, **sound attenuation** in fans can be achieved by untuned absorption silencers or resonant silencers tuned to certain frequencies (these resonant silencers are also known as interference, chamber or \( \lambda/4 \) silencers).

For both types of silencers, baffles ["Küllissen"] are arranged within a housing, parallel to the direction of flow.

The attenuation principle chosen (friction or reflection with interference) determines the design of the baffles.

In the case of **absorption silencers** the space between the baffle walls, built of perforated plate, is filled with sound absorbing mineral wool.

The molecules in the gas stream excited to produce sound oscillations are impeded by the mineral wool packing in the baffles such that the sound energy penetrating the perforations is converted to thermal energy by the friction of the molecules.

The absorption silencer is used for reducing the noise level of a wide band sound spectrum. Continued trouble free operation of this silencer can only be achieved if it is used in a relatively clean environment. In a dust laden atmosphere the dust will block the perforations in the baffle walls, thus reducing effectiveness.

In dust laden air or gas streams **resonant silencers (\( \lambda/4 \) silencers, interference silencers or chamber silencers)** are used. This type of silencer, however, has only limited effectiveness in reducing broad band noises. According to its attenuation principle, this silencer primarily reduces protruding single tones. By adding sound absorbing mineral wool mats to the baffle chamber plates a certain broad band attenuation is obtained in addition to the single tone attenuation (see Figure 9). The effectiveness of the single tone attenuation is explained by the principle of reflection and interference. The most important dimension when designing a resonant silencer is the depth of the chamber \( t \); this dimension must be approximately \( 1/4 \) of the wave length of the pitch to be attenuated (\( t = \lambda/4 \)) to cause the following action:

At a distance \( \lambda/4 \) from the outer baffle wall the sound wave hits the solid back wall of the chamber, is reflected and travels another distance of \( \lambda/4 \) back to the sound source where the reflected sound wave, traveling \( 2 \times \lambda/4 \) or \( \lambda/2 \) relative to the next following sound wave, arrives with a 180 degrees phase displacement and thus causes interference (tone elimination).
Axial flow impulse induced draft fan in a power plant with heat-sound insulation and lagging and discharge silencer.

- Volume flow: \( \dot{V} = 660 \text{ m}^3/\text{s} \)
- Temperature: \( t = 156 \text{ °C} \)
- Pressure increase: \( \Delta p = 6520 \text{ Pa} \)
- Speed: \( n = 590 \text{ 1/min} \)
- Shaft power: \( P_w = 5480 \text{ kW} \)

Axial flow induced draft fan (axial flow impulse fan) with heat-sound insulation and lagging and discharge silencer for reducing the sound level emitted from the stack outlet.
Sound Radiation of a Forced Draft Fan without Sound Protection (Figure 10.1.)

- $L_w$: Total sound power generated by fan
- $L_{WD}$: Gas-borne sound power radiating in flow direction through the discharge area
- $L_{WDL}$: Sound power radiating from the discharge duct as air sound due to $L_{WD}$ and body sound transmission
- $L_{WS}$: Gas-borne sound power radiating against flow direction through the inlet area.
- $L_{WSL}$: Sound power radiating from the inlet duct as air sound due to $L_{WS}$ and body sound transmission (Figure 10.2. and 10.3.)
- $L_{WG}$: Sound power radiating from the fan housing as air sound due to body sound excitation through the sound energy in the gas stream
- $L_{WU}$: Sound power radiating as air sound from the support structure due to body sound conduction from the fan housing
- $L_{WM}$: Sound radiation by attached or neighboring machinery (for instance fan motor drive)

Sound Protection of a Forced Draft Fan by Means of Inlet Silencer and Acoustic Insulation and Lagging (Figure 10.2.)

- Attenuation of the sound power $L_w$ through an inlet silencer
- Insulation of the sound power $L_{WD}$, $L_{WS}$, $L_{WG}$, $L_{WU}$ through acoustic insulation and lagging. Since no reduction of the gas-borne sound energy occurs inside the system the sound will radiate at full level from all surfaces where there are gaps in the acoustic insulation and lagging.
- The sound power $L_{WD}$ radiates into the duct system.

**Sound attenuation:** (through absorption) Sound energy penetrates through porous walls and is converted to thermal energy through viscosity friction. Sound energy strikes non-porous walls and is reflected.

**Sound insulation:**
Sound Protection of a Forced Draft Fan by Means of Inlet and Discharge Silencers and Acoustic Insulation and Lagging (Figure 10.3.)

In addition to the protection shown in Figure 10.2., the air-borne sound power $L_{Wd}$ radiating through the discharge area is reduced by a discharge silencer.

Sound Protection of a Forced Draft Fan by Means of Sound Enclosure with Integrated Inlet Silencer (Figure 10.4.)

- The sound powers $L_{WG}$ $L_{wu}$ radiating from the fan as air sound as well as the sound power $L_{Wm}$ radiating from the motor are insulated by the enclosure.

- The silencer integrated in the sound enclosure attenuates the sound power $L_{WS}$ radiating from the inlet area of the fan to the allowable value.

- The sound power $L_{WDL}$ radiating to the environment from the discharge duct can be reduced either by acoustic insulation and lagging (in this case, however, the sound power $L_{WD}$ radiates into the duct system) or through the addition of a discharge silencer.

For the design of sound enclosures appropriate ventilation must be assured to remove fan and main drive generated heat so that the maximum permissible temperature within the sound enclosure can be maintained. If necessary, forced ventilation has to be used (for example for hot gas fans).


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