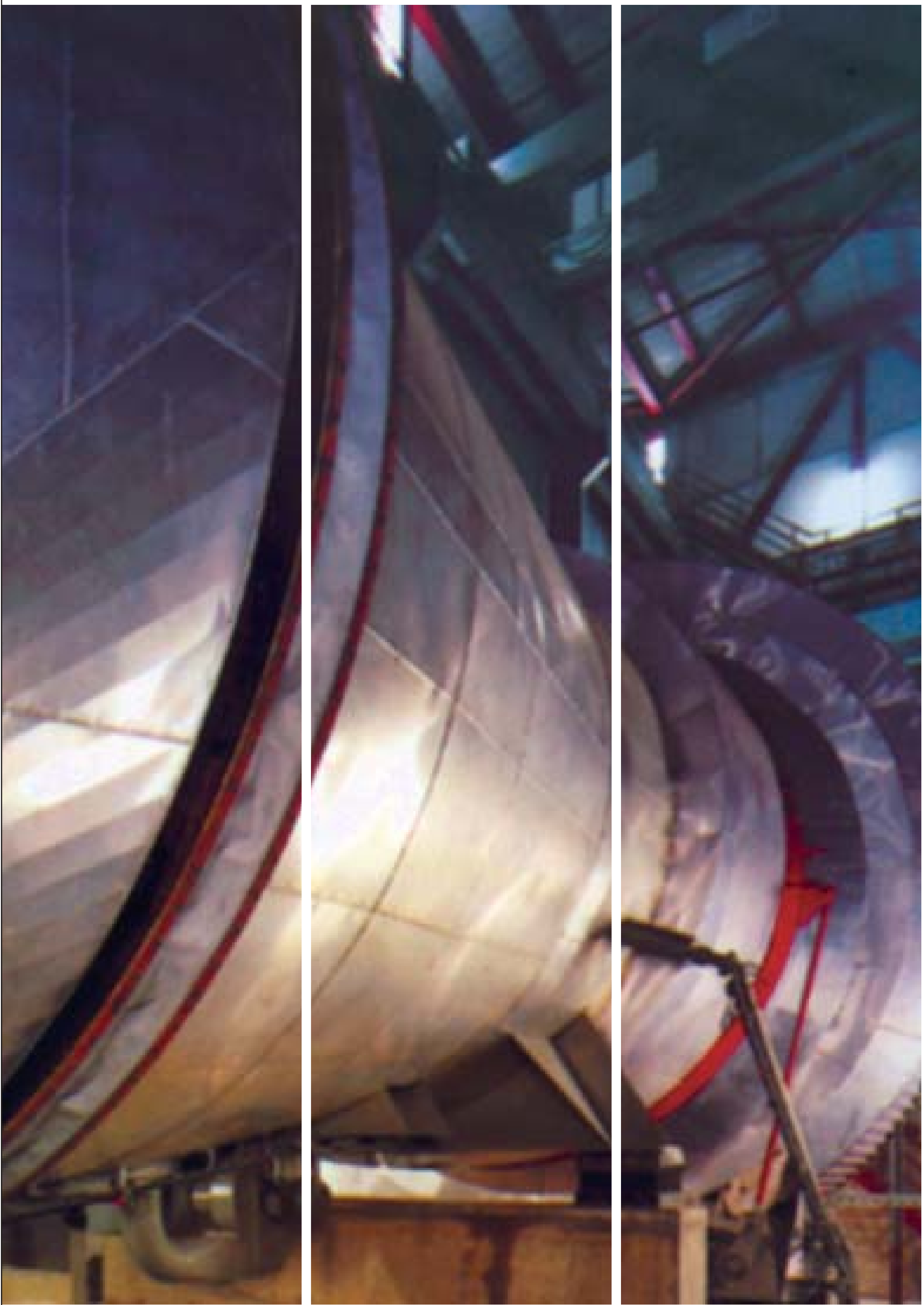


Industrial Fans

Delivery program

- ▲ Centrifugal Fans
- ▲ Axial Flow Impulse Fans
- ▲ Sound Protection

TLT-Turbo GmbH

Table of contents

Mine ventilation fan

Introduction

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

Field of Application

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 Gas Fans
Secondary Air Fans
Sealing Air Fans

Centrifugal F. D. fan with inlet vane control I and inlet silencer.

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	$\mathring{V}$	= 383 m^3/s
Depression	Δp_{Syst}	= 5400 Pa
Speed	n	= 440 1/min
Sheft power	P_{sh}	= 2560 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 tor HCL Regeneration Systems
High Pressure Fan Systems
Process Steam Fans

Centrifugal Fans
Axial Flow Impulse Fans
(Action Type Axial Flow Fans)

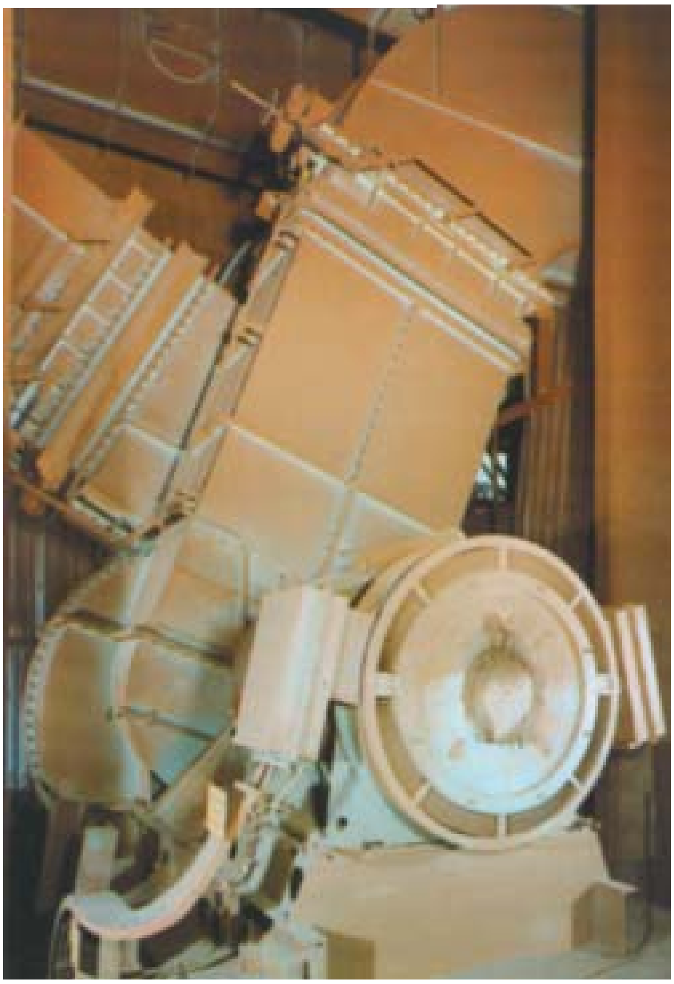
Silencers
Acoustic Insulation & Lagging
Sound Enclosures

Inlet Vane Controls
Support Structures
Indirect I.D. Fan Systems
(Power Stacks)

Emergency Air Systems

Double width double inlet exhaust gas fan on an electro-melt furnace particulate removal system

Volume flow	$\mathring{V}$	= 126	m^3/s
Temperature	t	= 120	°C
Pressure increase	Δp_t	= 3820	Pa
Speed	n	= 990	1/min
Shaft power	P_{Sh}	= 595	kW

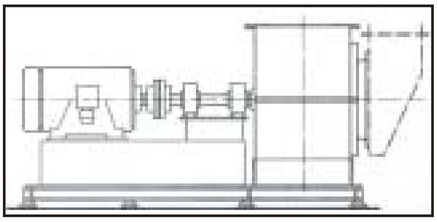


Double width double inlet emergency F. D. fan in a utility power plant.

Volume flow	$\mathring{V}$	= 350	m^3/s
Pressure increase	Δp_t	= 9320	Pa
Speed	n	= 990	1/min
Shaft power	P_{Sh}	= 4100	kW

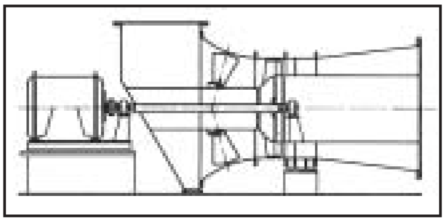
Fan Designs

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.

RELATIONSHIP BETWEEN PRESSURE, DELIVERY HEAD AND SPECIFIC DELIVERY WORK

$$\boxed{Y = \frac{\Delta p_t}{\rho}}$$

Y = Specific delivery work of the fan in $\frac{J}{kg}$, $\frac{Nm}{kg}$, $\frac{m^2}{s^2}$

whereby $1\frac{J}{kg} = 1\frac{Nm}{kg} = 1\frac{m^2}{s^2}$

Δp_t = Total pressure increase of the fan in $\frac{N}{m^2}$, Pa

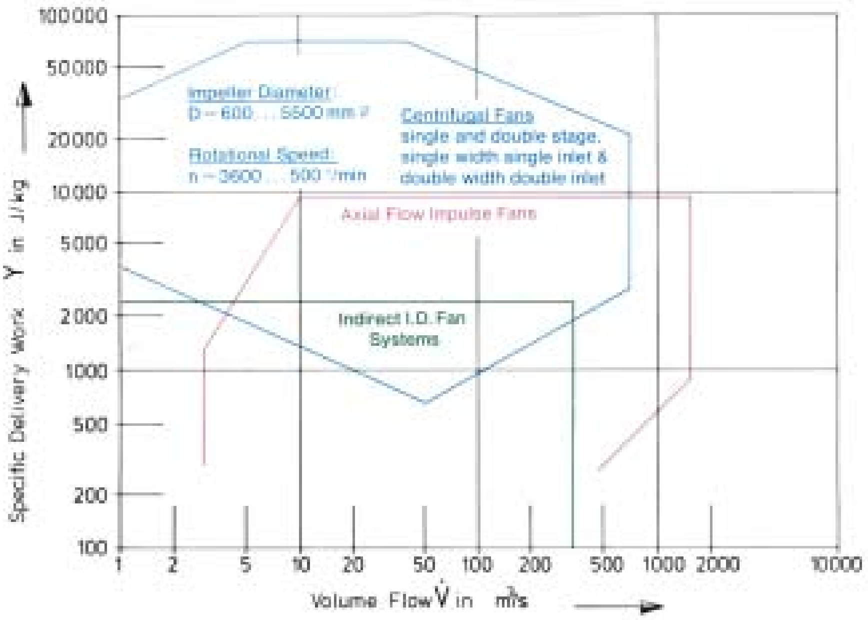
whereby $1\frac{N}{m^2} = 1$ Pa

ρ = Density of the fluid in $\frac{kg}{m^3}$

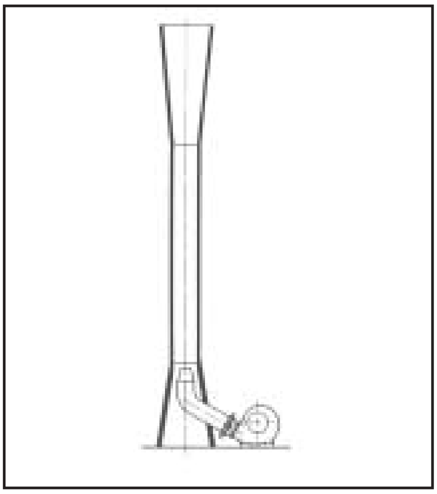
$$\boxed{Y = g \cdot H} \qquad \boxed{H = \frac{\Delta p_t}{\rho \cdot g}}$$

H = Delivery head of the fan in m

$g = 9{,}81\frac{m}{s^2}$ = Gravitation acceleration



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

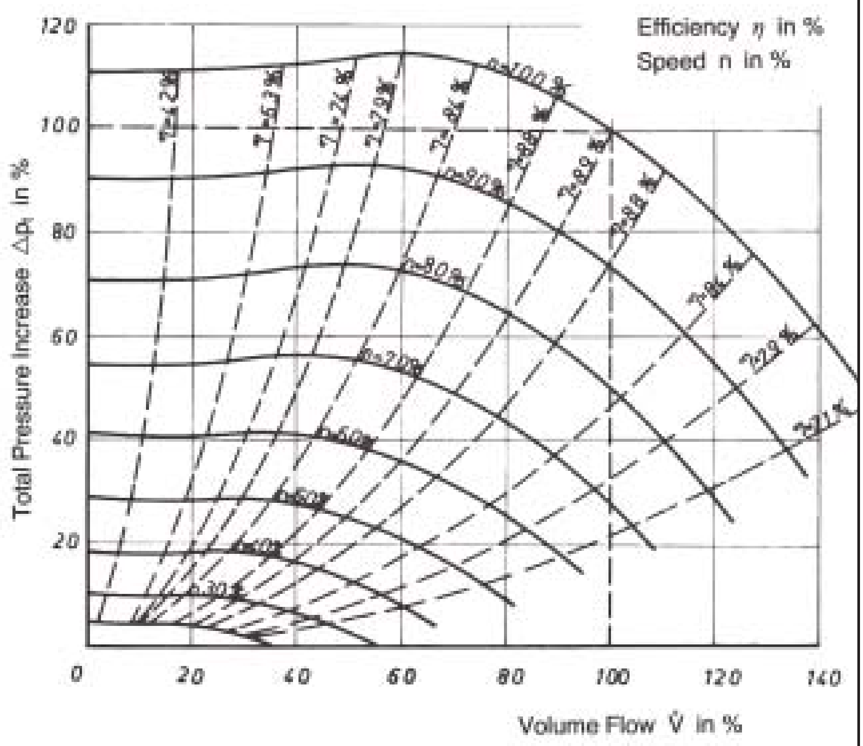


Indirect I.D. fans (power stacks) are often used for exhaust gases at tempe-ratures above 5000C

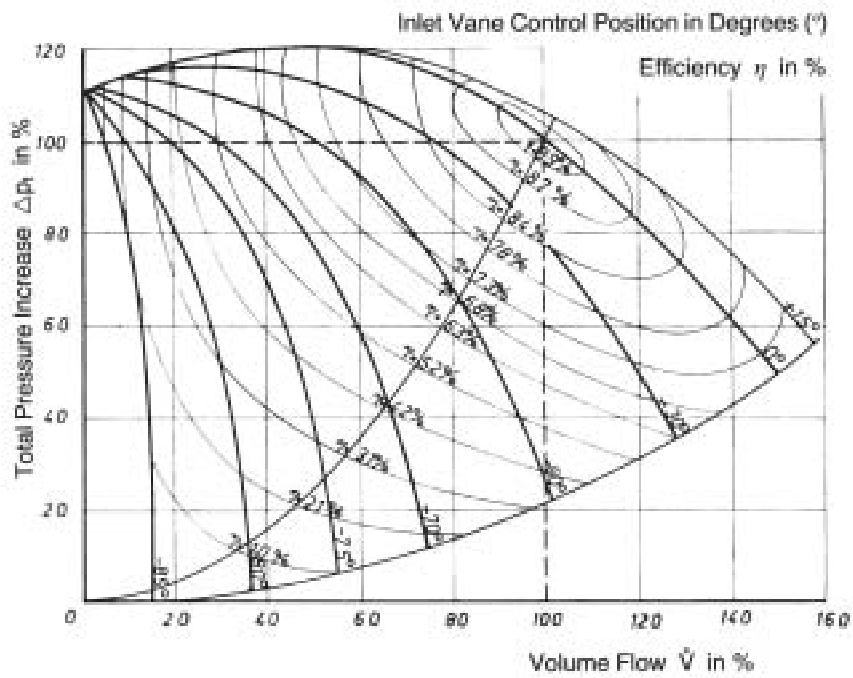
Control Modes and Characteristic Curves

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.

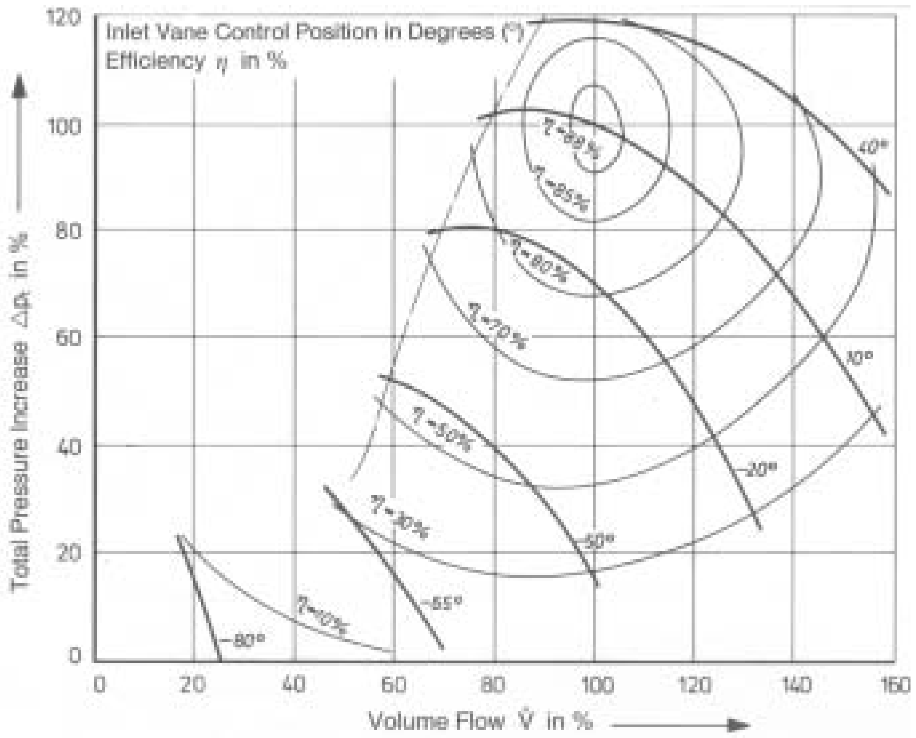
The most effective type of fan control is obtained by variation of the fan speed. Since speed control can only be achieved with high cost drive systems we commonly utilize inlet vane control for both centrifugal and axial flow fans. In the diagrams shown, the 100% point ($\mathring{V}$ = 100% and ΔP_t = 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



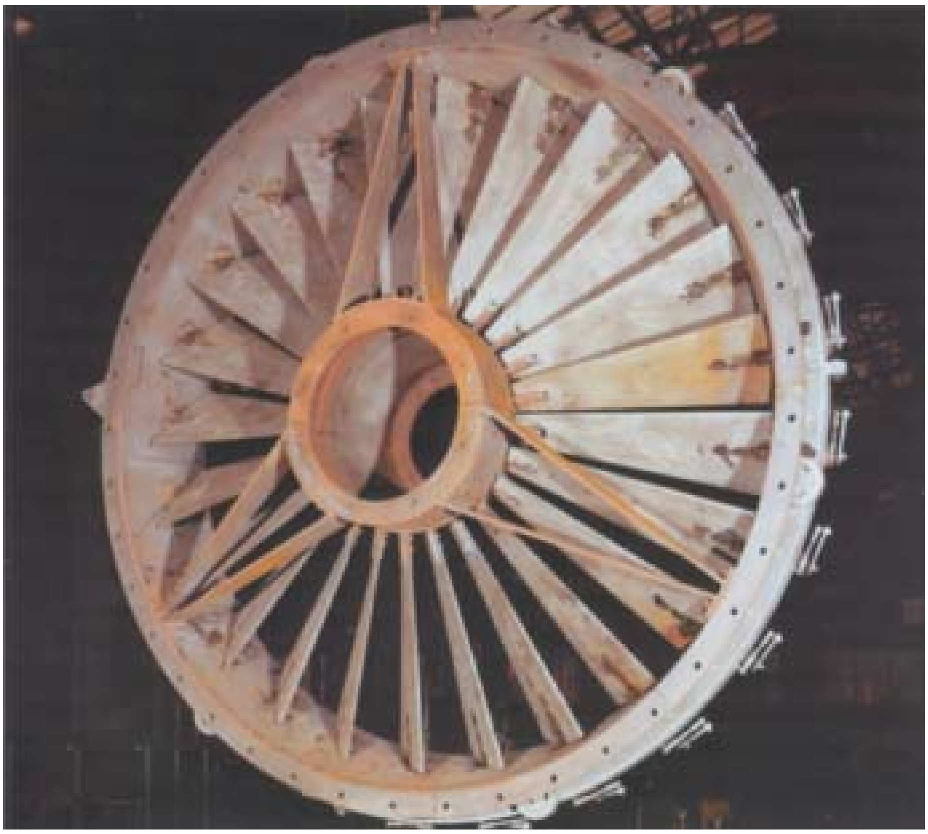
Characteristic curves of a centrifugal fan with inlet vane control



Model tests in the laboratory have provided the data needed to determine specific performance characteristics of our tans, 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 optimizied 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.

Axial flow impulse fan with slotted flap adjustment, shown during production.

Volume flow	$\mathring{V}$	=	660	m^3/s
Temperature	t	=	156	°C
Pressure increase	Δp_t	=	6520	Pa
Speed	n	=	590	1/min
Shaft power	P_{Sh}	=	5480	kW
Diameter	D	=	4220	mm Ø

Inlet vane control:

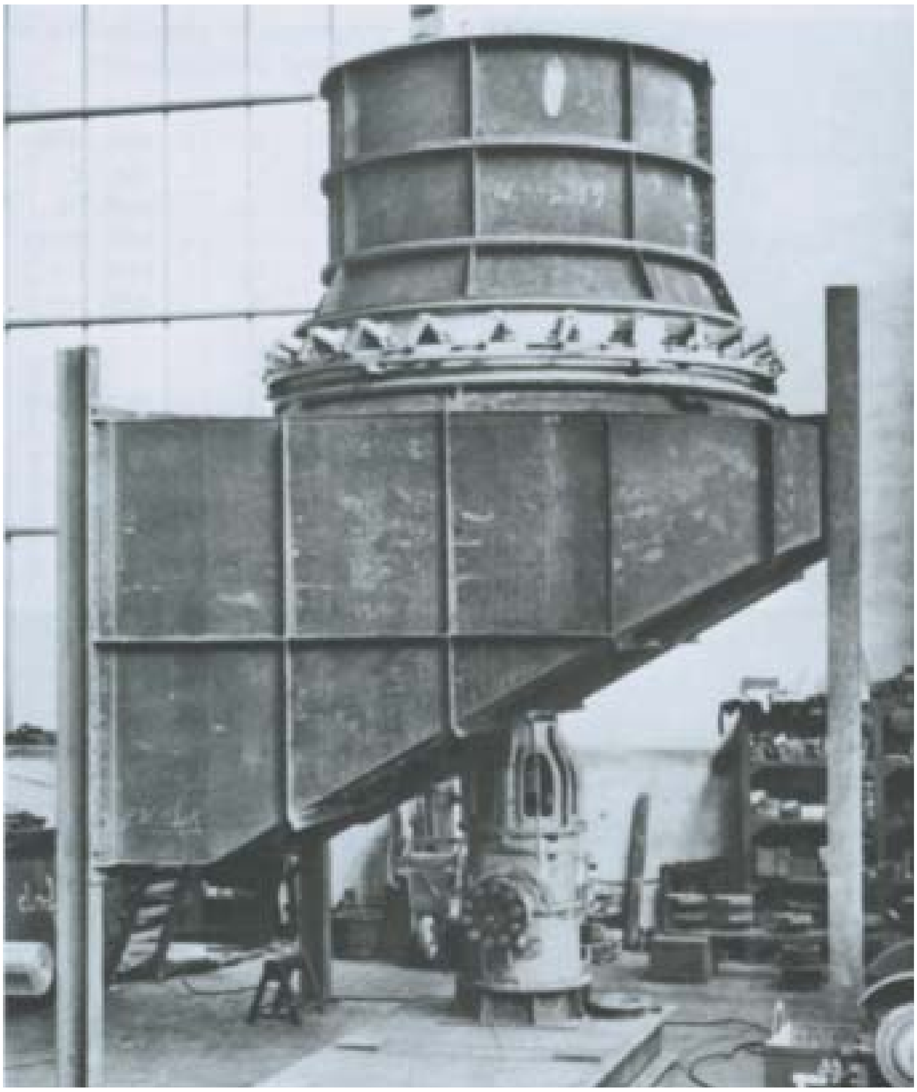
Diameter	D	=	4800	mm Ø

Double width double inlet I. D. fan for waste heat boiler, fan support of laterally flexible base frame design.

Volume flow	$\mathring{V}$	=	180	m^3/s
Temperature	t	=	245	°C
Pressure increase	Δp_t	=	4420	Pa
Speed	n	=	990	1/min

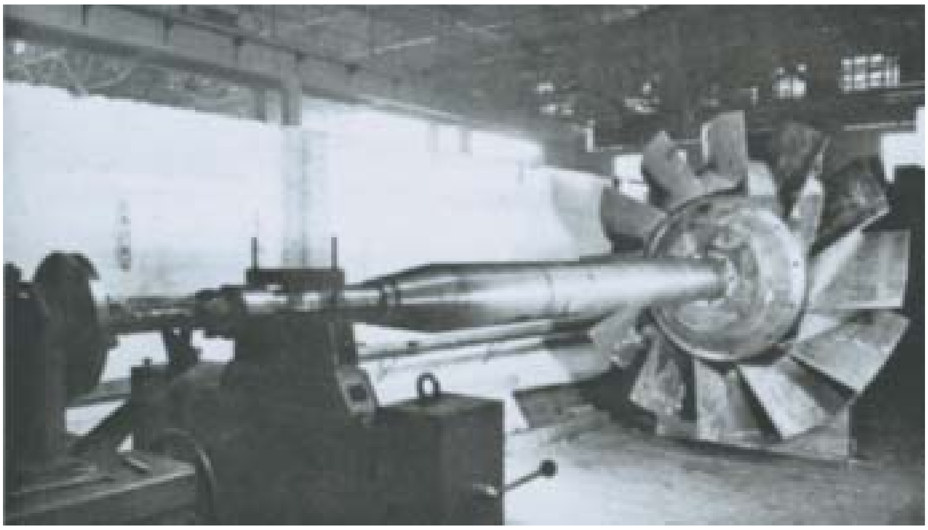
F.D. fan with inlet silencer in a steel mill.

Volume flow	$\mathring{V}$	=	77	m^3/s
Temperature	t	=	30	°C
Pressure increase	Δp_t	=	7260	Pa
Speed	n	=	990	1/min
Shaft power	P_{Sh}	=	650	kW
Efficiency	η	=	84	%
Diameter	D	=	2400	mm Ø



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.

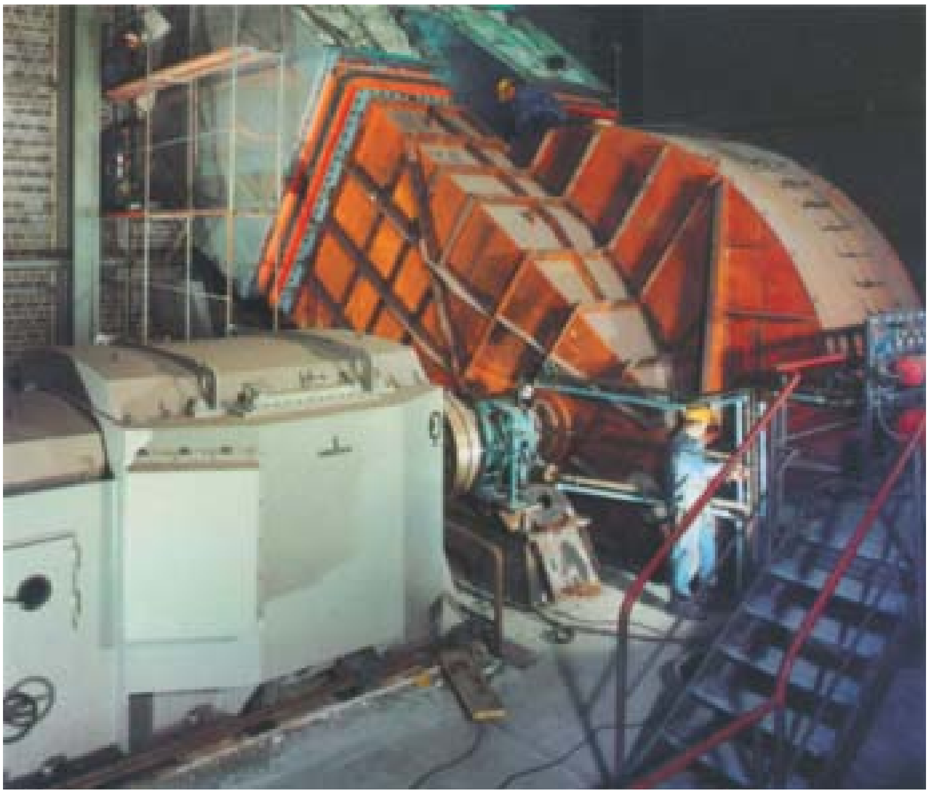
Axial flow impulse I.D. fan designed for vertical installation, shown in the manufacturing stage.



Impeller and shaft of an axial flow impulse fan during balancing operation.

Double width double inlet gas fan in electro-metallurgical plant.

Volume flow	$\mathring{V}$	=	195	m³/s
Temperature	t	=	230	°C
Pressure increase	Δp_t	=	3720	Pa
Speed	n	=	740	1/min
Shaft power	P_{Sh}	=	875	kW
Motor power	P_M	=	1100	kW



Double width double inlet sintering gas fan in steel plant.

Volume flow	$\mathring{V}$	=	366	m³/s
Temperature	t	=	160	°C
Pressure increase	Δp_t	=	14200	Pa
Speed	n	=	990	1/min
Efficiency	η	=	84	%
Shaft power	P_{Sh}	=	5900	kW
Motor power	P_M	=	6500	kW

Left: Impeller for a single stage F.D. fan

Volume flow	$\mathring{V}$	=	4.8	m^3/s
Temperature	t	=	20	°C
Pressure increase	Δp_t	=	31400	Pa
Speed	n	=	2980	1/min
Diameter	D'	=	1250	mm Ø
Impeller mass	m_{Imp}	=	210	kg

Right: Rotor for a two-stage gas fan

Volume flow	$\mathring{V}$	=	1.03	m^3/s
Temperature	t	=	100	°C
Pressure increase	Δp_t	=	28500	Pa
Speed	n	=	2980	1/min
Diameter	D'	=	865	mm Ø
Impeller mass	m_{Imp}	=	140	kg

(both impellers)

Rotor for a mine fan

Design volume flow	$\mathring{V}_{Des}$	=	41 7	m^3/s
(Maximum volume flow	$\mathring{V}_{mas}$	=	500	m^3/s)
Temperature	t	=	20	°C
Depression	Δp_{Syst}	=	5890	Pa
Speed	n	=	420	1/min
Impeller diameter	D'_{Imp}	=	5280	mm
Impeller mass	m_{Imp}	=	14000	kg
Shaft mass	m_{Sh}	=	9900	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	$\mathring{V}$	= 265	m^3/s
Temperature	t	= 160	°C
Pressure increase	Δp_t	= 16650	Pa
Speed	n	= 990	1/min
Shaft power	P_{Sh}	= 5230	kW
Impeller mass	m_{Imp}	= 11000	kg
Shaft mass	m_{Sh}	= 11000	kg

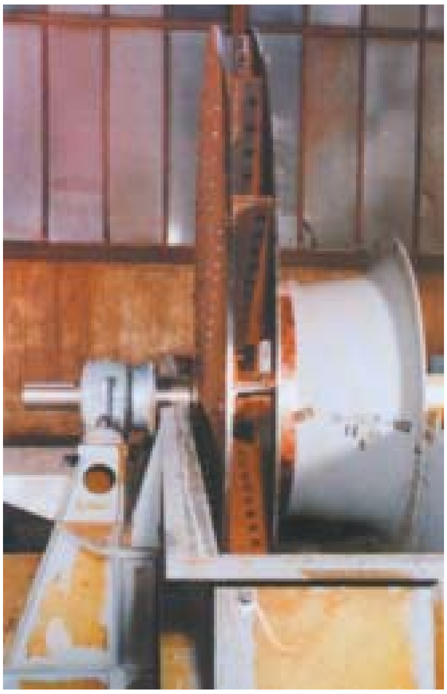
Rotor for an axial flow impulse fan
(I. D. fan for a utility power station)

Volume flow	$\mathring{V}$	=	660	m^3/s
Temperature	t	=	156	°C
Pressure increase	Δp_t	=	6520	Pa
Speed	n	=	590	1/min
Diameter	D	=	4220	mm Ø
Impeller mass	m_{Imp}	=	12100	kg
Shaft mass	m_{sh}	=	5200	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	$\mathring{V}$	=	39.5	m3/s
Temperature	t	=	150	°C
Pressure increase	Δp_t	=	13550	Pa
Speed	n	=	1145	1/min
Diameter	D	=	3030	mm ø



Left hand side of the picture: Rotor coated with Saekaphen for a two-stage coke gas fan.

Volume flow	$\mathring{V}$	=	3.9	m^3/s
Tempersture	t	=	25	°C Pressure
increase	Δp_t	=	19650	Pa
Speed	n	=	2970	1/mm
Diameter	D	=	1224	mm ø

Right hand side of the picture: Rubber lined impeller for a flue gas fan behind a venturi scrubber.

Volume flow	$\mathring{V}$	=	17.6	m^3/ls
Temperature	t	=	72	°C
Pressure increase	Δp_t	=	9810	Pa
Speed	n	=	1485	1/min
Diameter	D	=	1874	mm ø

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	$\mathring{V}$	=	125	m^3/s
Temperature	t	=	350	°C
Pressure increase	Δp_t	=	6770	Pa
Speed	n	=	990	1/min
Diameter	D	=	3160	mm ø

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

Volume flow	$\mathring{V}$	=	50	m^3/s
Temperature	t	=	30	°C
Pressure increase	Δp_t	=	5870	Pa
Speed	n	=	1000	1/min
Diameter	D	=	2160	mm ø

Lead coated inlet box of the scrubber fan.

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).

Volume flow	$\mathring{V}$	= 5.1	m^3/s
Temperature	t	= 26	°C
Pressure increase	Δp_t	= 53900	Pa
Speed	n	= 2985	1/min
Shaft power	P_{Sh}	= 331	kw

SWSI (single width single inlet) gas re-circulation fan, supported on both sides, installed in a utility power plant.

Volume flow	$\mathring{V}$	= 167	m³/s
Temperature	t	= 350	°C
Pressure increase	Δp_t	= 2360	Pa
Speed	n	= 720	1/min
Efficiency	η	= 85,5	%
Shaft power	P_{Sh}	= 457	kW

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

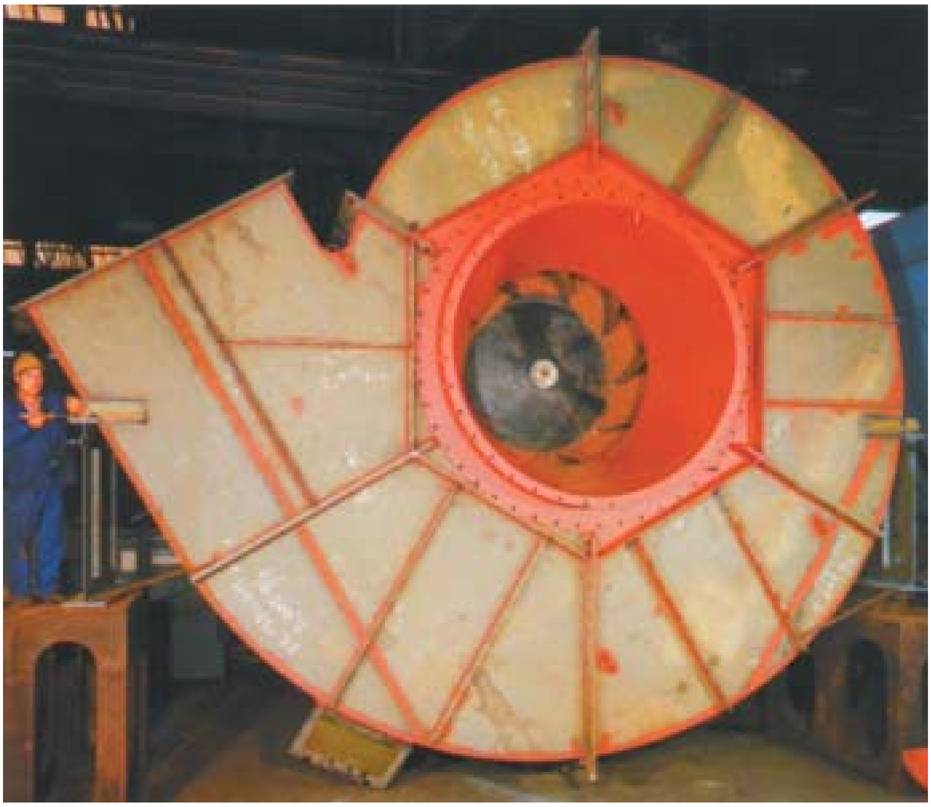
Volume flow	$\mathring{V}$	= 142	m³/s
Temperature	t	= 361	°C
Pressure increase	Δp_t	= 7850	Pa
Speed	n	= 990	1/min
Shaft power	P_{Sh}	= 1360	kW
Diameter	D	= 3280	mm ø

DWDI (double width double inlet) gas re-circulation tan with concrete tilled integral base frame and vibration isolators.

Volume flow	$\mathring{V}$	=	124	m^3/s
Temperature	t	=	300	°C
Pressure increase	Δp_t	=	9615	Pa
Speed	n	=	1490	1/min
Shaft power	P_{Sh}	=	1410	kW
Diameter	D	=	2320	mm ø

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

Volume flow	$\mathring{V}$	=	250	m^3/s
Temperature	t	=	340	°C
Pressure increase	Δp_t	=	3470	Pa
Speed	n	=	715	1/min
Shaft power	P_{Sh}	=	1100	kW
Motor power	P_M	=	1300	kW
Diameter	D`	=	3200	mm ø



SWSI (single width single inlet) raw mill fan for the cement industry, supported on one side (AMCA arrangement 8).

Volume flow	$\mathring{V}$	= 114	m^3/s
Temperature	t	= 90	°C
Pressureincreaso	Δp_t	= 5700	Pa
Speed	n	= 745	1/min
Impeller diameter	D_{Imp}	= 3350	mm ø

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	$\mathring{V}$	= 133	m^3/s
Temperature	t	= 100	°C
Pressure increase	Δp_t	= 4600	Pa
Speed	n	= 745	1/min

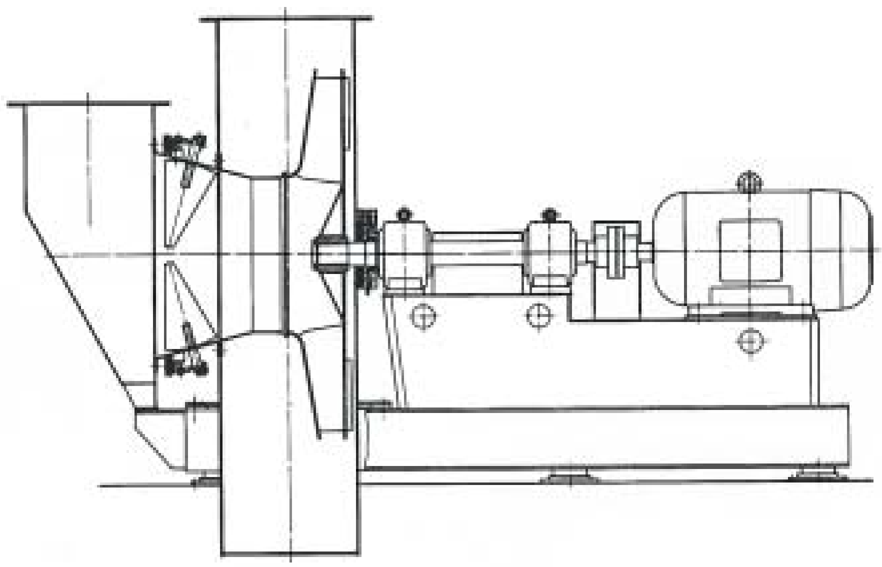
Our economical production facilities are equipped with modern machinery. For example, a numerically controlled flame cutting machine and a metal spinning machine are used for the processing of sheet metal.

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

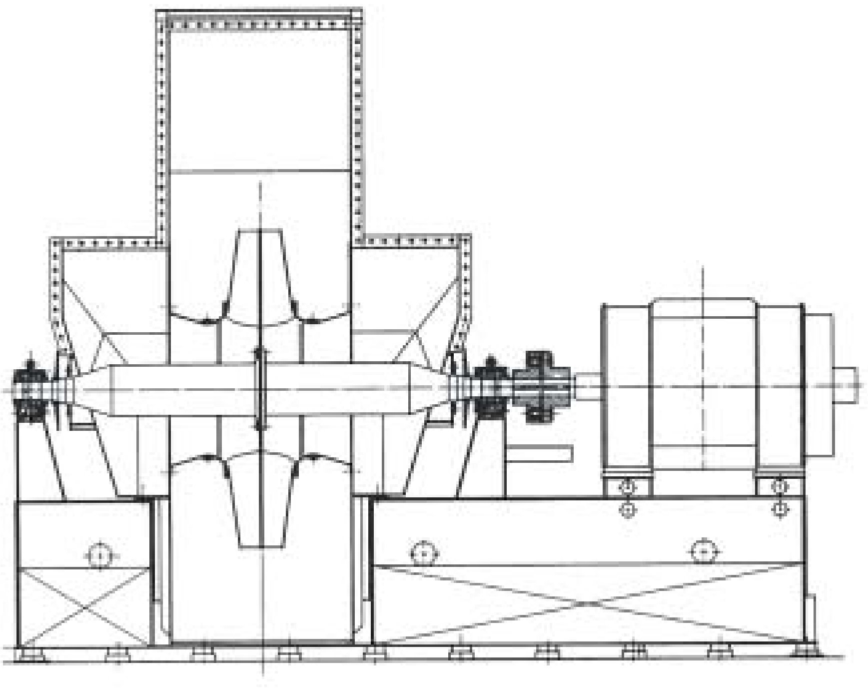
Balancing machine for fan rotors up to 30000 kg and 5000 mm Ø

Metal spinning machine for radii of inlet nozzles, impeller side plates and spinning flanges.

A selection of varios designs from our fan range is shown below.

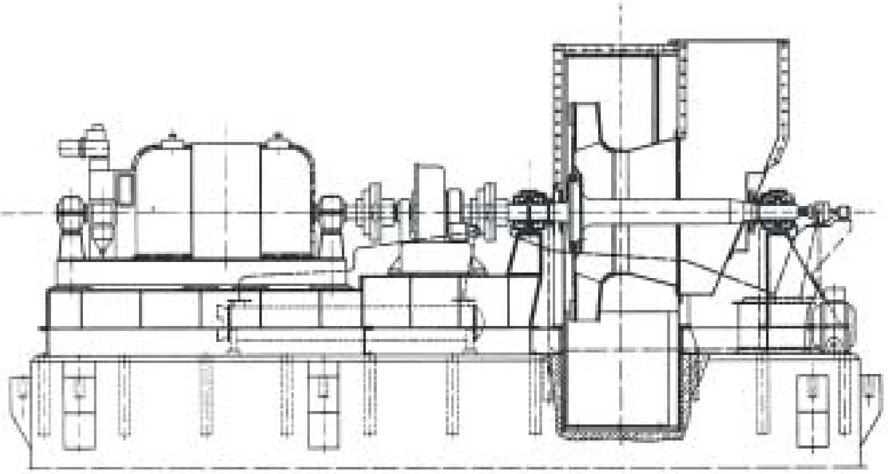


SWSI (single width single inlet) forced draft fan, supported on one side (AMCA arrangement 8), with inlet vane control, arranged on an integral supporting structure with vibration isolators.

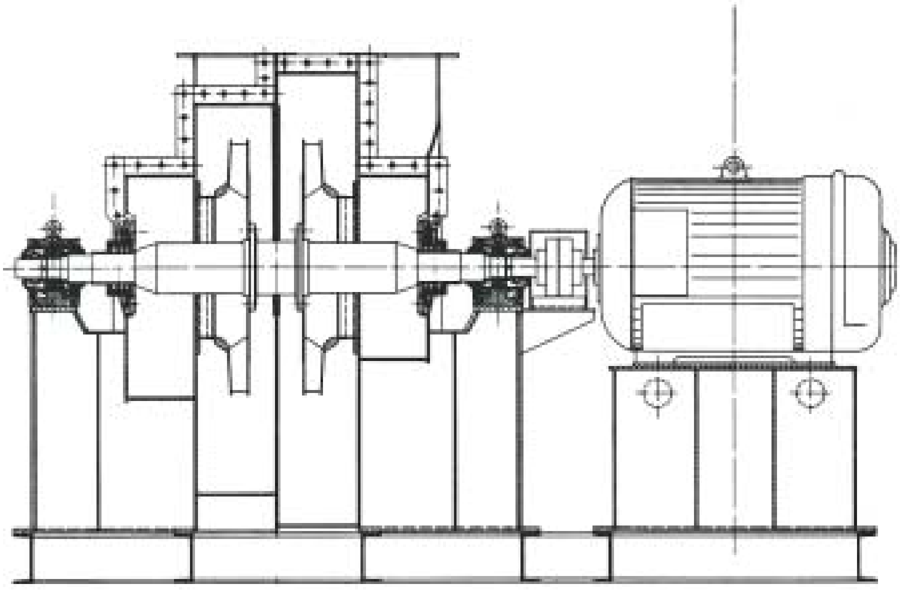


DWDI (double width double inlet) fan arranged on an integral supporting structure with vibration isolators.

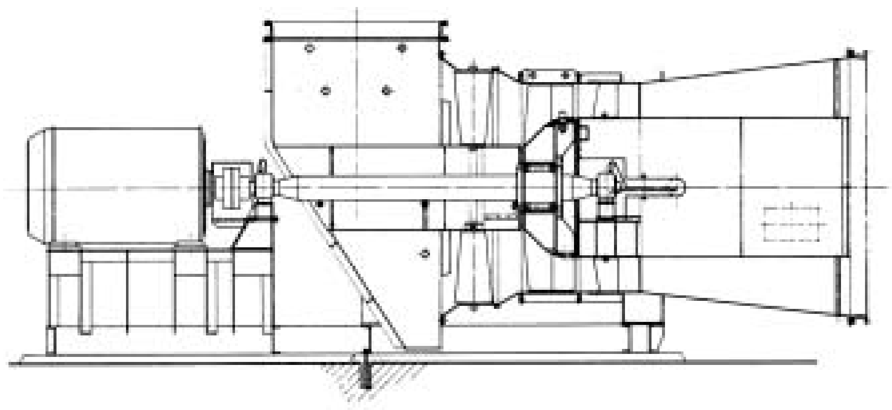
SWSI (single width single inlet) sintering gas fan, rotor supported on both sides, with fluid drive, fan supported by an integral steel frame.



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 I engineer and/or the final user to optimize fan selection for each specific application.

The following set of conditions at tan 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: $\mathring{V}$ or $\mathring{V}_{SC}$ **) in m^3/s

3. Temperature: t in °C

4. System-related pressure increase: Δp_{Syst} in Pa
   $\Delta p_{Syst} = P_{t4} - P_{t1}$*)
   (Pressure distribution: Fan suction side / discharge side)

5. Fan inlet pressure measured against barometric pressure (+/-) P_1 in Pa

6. Elevation: h in m above see level

7. Mains frequency (Standard frequency): f_{Mains} in Hz

8. Permissible noise level:
   Sound pressure level at a defined distance: L_{allow} in dB (A)
   or sound power level: $L_{W\ allow}$ in dB (A)

9. Information on the fluid handled:
   9.1 Type of gas:
   9.2 Density of gas: ϱ or ϱ_{SC} **) in kg/m^3
   (if necessary supply gas analysis with moisture content)
   9.3 Particulate content of the gas: St or St_{SC} **) in g/m^3; 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 Δp_{Syst}", as defined by us, is often referred to as"static pressure increase Δp_{stat}", the dynamic pressure components having been ignored, however.

**) The index "sc" identifies the standard condition (t = 0°C, p =101325 Pa).

Types of Fan Design and Installation

Radial-flow fans are normally of the single-inlet type up to approx. 100 m^3/s; in some cases the double-inlet type is used from 60-70 m^3/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 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 shont 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.

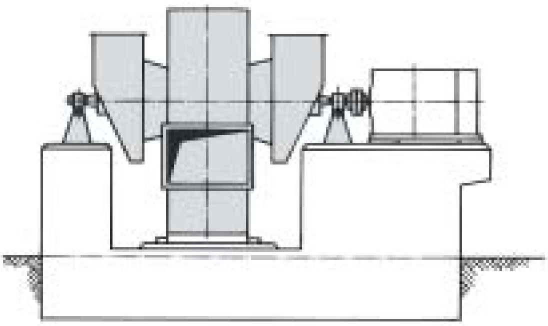
If the substructure has to be made of steel corresponding plate cross sections have to be used in case of larqe base-to-centre heights to achieve a sufficient vibration resistance. Thus the fan weight and the costs are increased accordingly.

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.

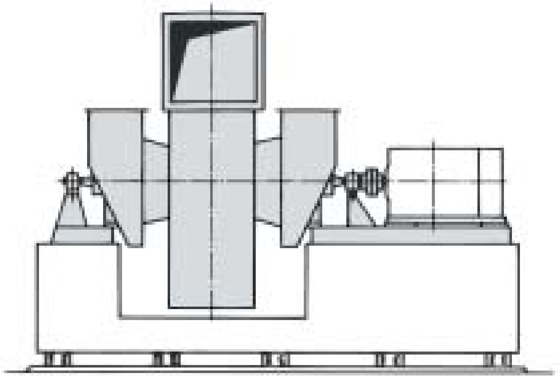
For this purpose, the compact type of construction is suitable which is a frameless, self-supporting structure utilizing the rigidity of the almost totally enclosed suction boxes and housing. The vibration isolators are installed under the "hard" points such as housing walls.
The installation of the 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.

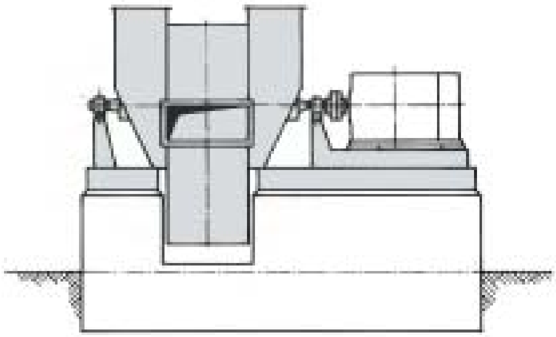
Some installation examples:



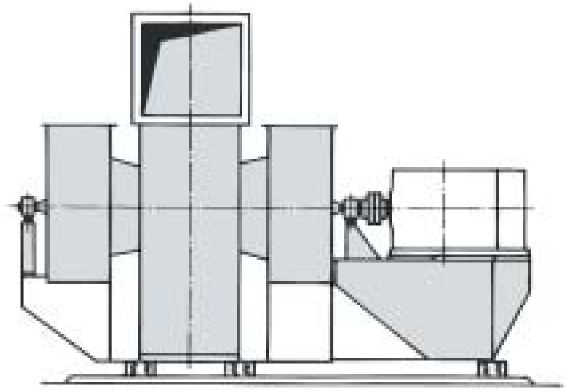
Elevated concrete base without frame below the motor - rugged, simple installation.



Vibration-isolated installation with elevated concrete block on vibration dampers.



Elevated concrete base with low frame.



Compact type of construction, selfsupporting struc-ture, 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
p_v	Pressure Loss, Resistance
$\mathring{V}$	Volume Flow at Inlet Condition
A	Cross Sectional Area
l	Length
$\mathring{m}$	Mass Flow
c	Gas Velocity
ϱ	Density
f	Compressibility Factor
Y	Specific Delivery Work
P_{Sh}	Shaft 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 differentals
- Geodetic head differences 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_t (analogy: total energy) comprises static pressure p_s (analogy:potential energy) and the dynamic pressure p_d (analogy: kinetic energy).

$$p_t = p_s + p_d$$

(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 Δp_{Syst}.**

$$\Delta p_{Syst} = p_{t4} - p_{t1}$$

If the cross sections 1 and 4 (Figure D-1) represent the terminal points of the fan the system-related pressure increase Δp_{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 refered to as **static pressure increase Δp_{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 Δp_{Syst} stated by the customer.

$$A_1 = A_4$$

$$\Delta p_{Syst} = p_{t4} - p_{t1}$$

With $A_1 = A_4$ and disregarding compressibility it follows that $P_{d4} = P_{d1}$ and therefore:

$$\Delta p_{Syst} = (p_{s4} + p_{d4}) - (p_{s1} + p_{d1}) \quad \Delta p_s$$

The **dynamic pressure P_d** is understood to be based on the average gas velocity in a cross section.

$$p_d = \frac{\varrho}{2} \cdot c^2$$

Example: Dynamic pressure at terminal point 4:

$$p_{d4} = \frac{\varrho_4}{2} \cdot c_4^2 = \frac{\varrho_1}{2} \left(\frac{\mathring{V}}{A_4} \right)^2 \cdot f^2$$

Pressure losses p_v caused by fan com-ponents 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 Δp_t, the parameter determining fan size, is the sum of the system-related pressure increase and the pressure losses of the fan components.

$$\Delta p_t = \Delta p_{Syst} + p_V = p_{t3} - p_{t2}$$

Our characteristic curves show the **design pressure Δp_t**, 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 Δp_{Syst} varies with the losses p_v which arise depending on the fan components used.
In determining the fan design pressure Δp_t 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 Δp_t 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 $\mathring{V}$ at fan inlet conditions, the power requirements at the fan shaft P_{Sh} and the efficiency η will be determined, optimum inlet conditions being a prerequisite.

$$P_{Sh} = \frac{\mathring{V} \cdot \Delta p_t \cdot f}{\eta} = \frac{\mathring{m} \cdot Y}{\eta}$$

$\mathring{V}$	in m^3/s	
Δp_t	in Pa	$= N/m^2$
f	< 1	
η	< 1	
$\mathring{m}$	in kg/s	
Y	in J/kg	= Nm/kg
P_{Sh}	in W	= Nm/s

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 Δt_{ad} is:

$$\Delta t_{ad} = T_2 \cdot \left[(p_3/p_s)^{\frac{\chi-1}{\chi}} - 1 \right]$$

The real temperature increase is:

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

Δp_t	in	Pa
Δt	in	°C
η	$<$	1

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.

$$P_{Sh\ operation} = \frac{T_{design}}{T_{cold}} \cdot P_{Sh\ design}$$

$$= \frac{\varrho_{alternativ\ gas}}{\varrho_{design}} \cdot P_{Sh\ design}$$

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.

$$\Delta p_{t\ operation} = \frac{T_{design}}{T_{cold}} \cdot \Delta p_{t\ design}$$

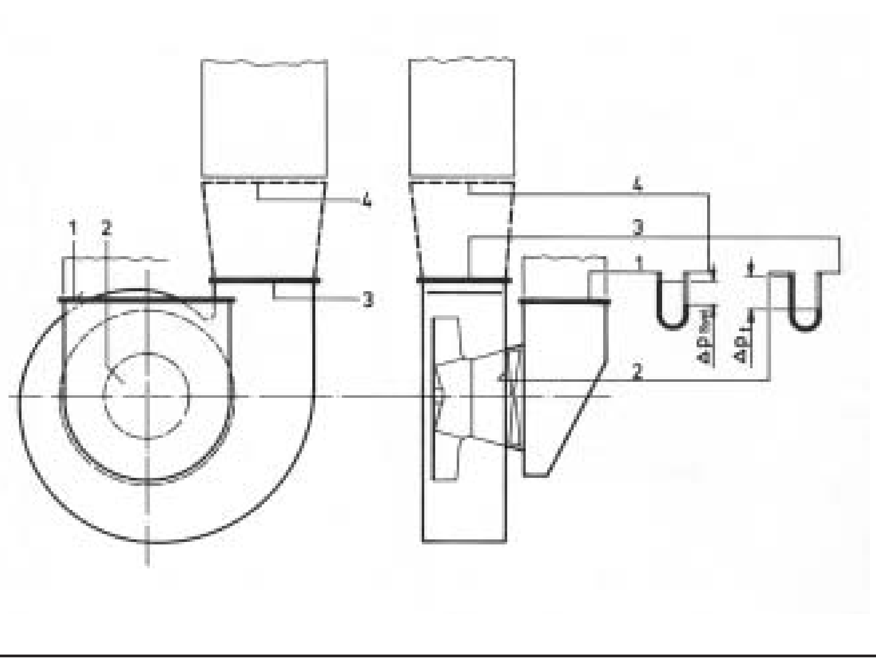
$$= \frac{\varrho_{alternativ\ gas}}{\varrho_{design}} \cdot \Delta p_{t\ design}$$

Influence of Mass and Mass Inertia

Erosion, corrosion and system-related contamination causing build-up can posibly 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.

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)

$$\Delta p_t = \Delta p_{Syst} + \sum p_V$$

$$\Delta p_{Syst} = \Delta p_{Syst,\,s} + \Delta p_d$$

Meaning of Symbols Used in Equations:

Δp_t	Total Pressure Increase Between Cross Sections 2 and 3
Δp_{Syst}	System Resistance-Related Total Pressure Increase
$\Delta p_{Syst,\,s}$	System Resistance-Related Static Pressure Increase
p_d	Dynamic Pressure (Velocity Pressure)
Δp_d	Dynamic Pressure Difference (Velocity Pressure Difference)
p_V	Pressure Loss(es)
$p_{V\,In}$	Pressure Loss at Fan Inlet
$p_{V\,Box}$	Inlet Box Pressure Loss
$p_{V\,Dif}$	Diffuser Pressure Loss
$p_{V\,Out}$	Outlet Pressure Loss
A	Cross Section

Behavior of

Total Pressure

Static Pressure

Dynamic Pressure (Velocity Pressure)

Indexes:

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

Figure D-2: Open Inlet Fan (e.g. Forced Draft Fan)

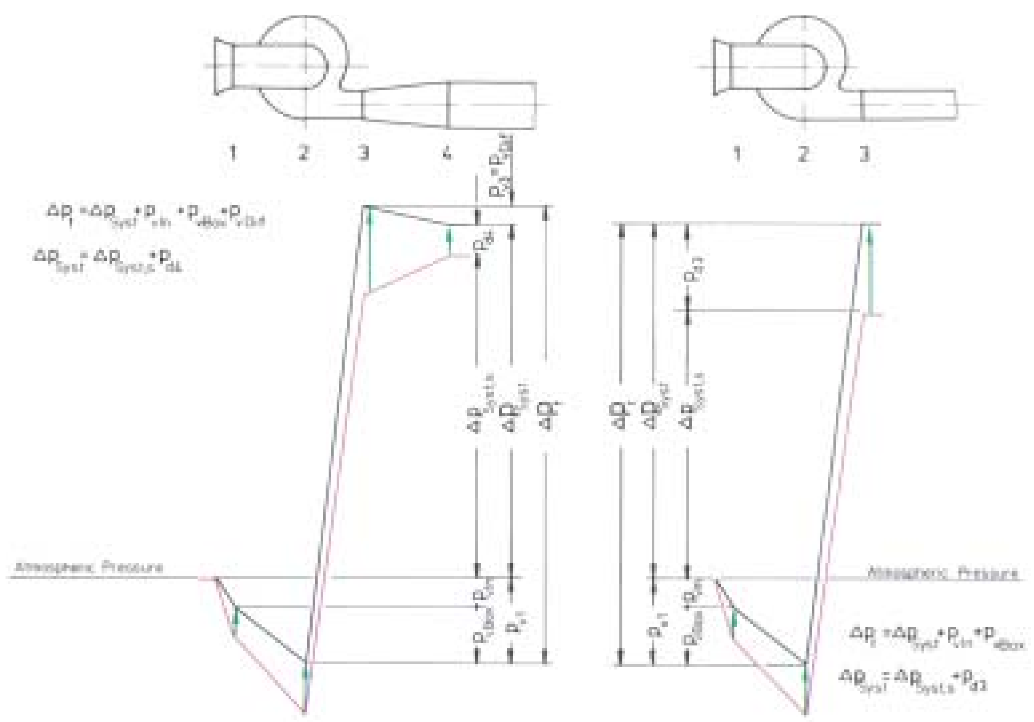


Figure D-3: Fan Connected to Ducts at Fan Inlet and Disscharge (e.g. Induced draft Fan)

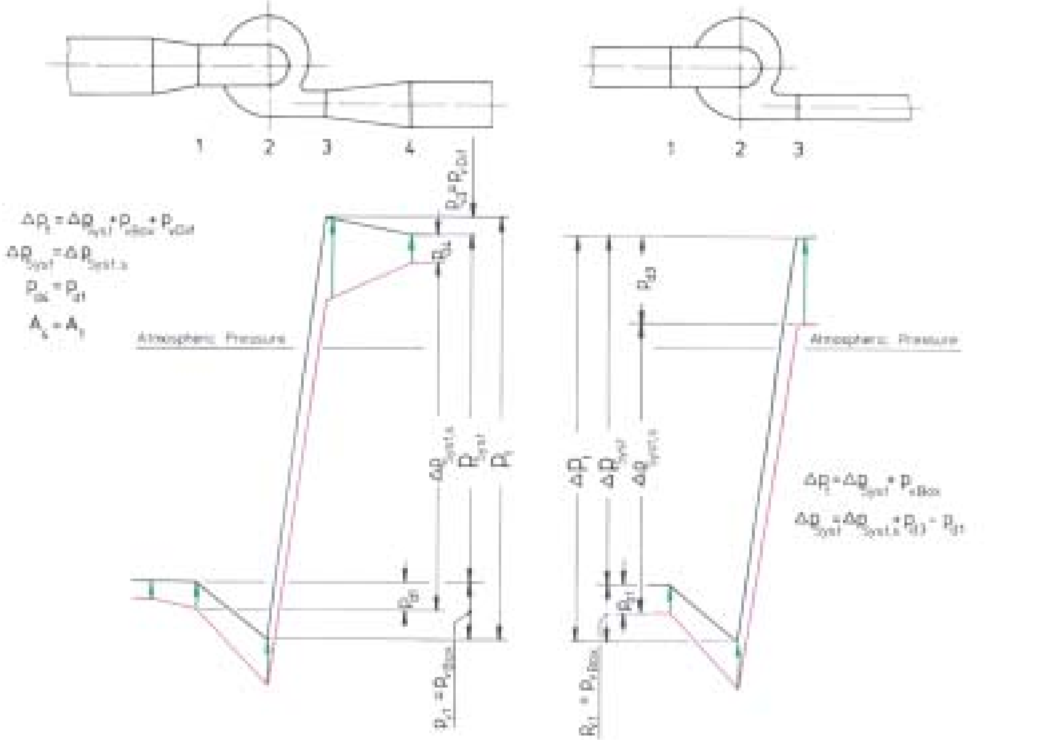
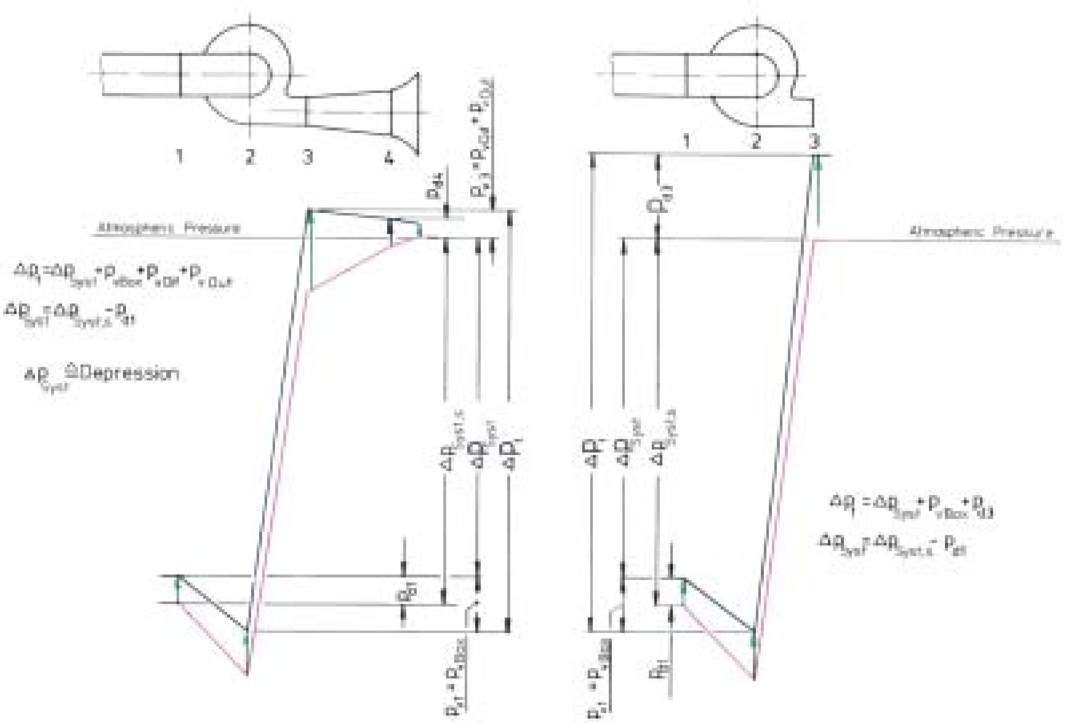


Figure D-4: Exhaust Fan (e.g. Mine Fan)

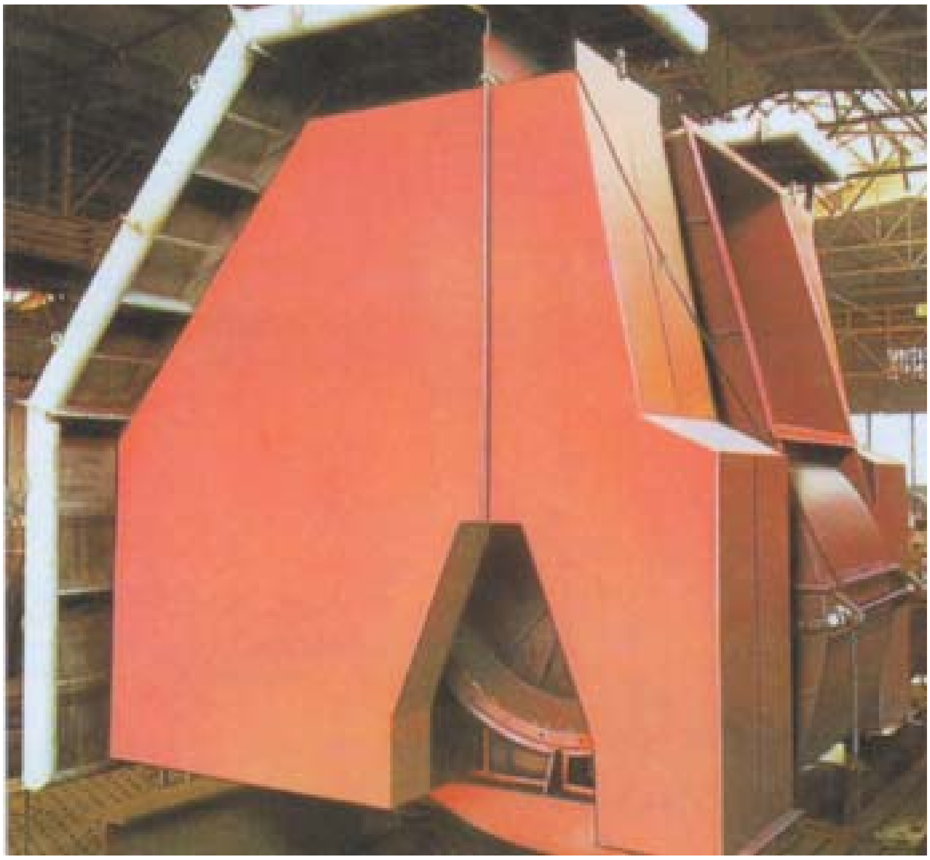


Questions Regarding Fan Noise

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

Volume flow	$\mathring{V}$	=	2 x 62	m^3/s
Temperature	t	=	50	°C
Pressure Increase	Δpt	=	8120	Pa
Speed	n	=	1490	1/min

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

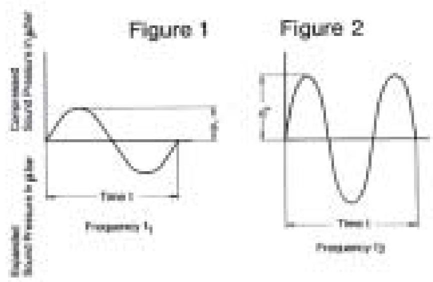
With progressing industrialization man is faced with increasing environmental problems. Noise emitted by fans belongs in this category.
The following will give guidance in the problem area of noise emitted by fans as well as the flow in the connecting flues and ducts.

The **sound** [“Schall“][1]) 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 harmo-nic 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“].

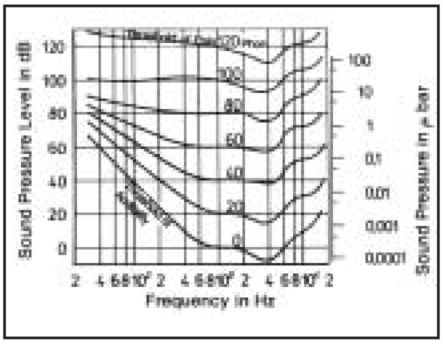


[1])The terms in square brackets [ ] are the equivalent German words.

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_x (measured in dB)[1] at a frequency of 1000 Hz at which the sound pressure p_n (measured in dB) and the frequency f_m (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[2] 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 be (B) or decibel (dB).

$$\underset{\text{of sound parameter}}{\text{level}} = \lg \frac{\text{effective value of sound parameter}}{\text{reference value of sound parameter}} \text{ in B}$$

$$\underset{\text{of sound parameter}}{\text{level}} = 10 \lg \frac{\text{effective value}}{\text{reference value}} \text{ in dB}$$

As the **same units** are applied to **al**l 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:

$$L = 10 \lg \frac{p^2}{p_0^2} = 20 \lg \frac{p}{p_0} \text{ in dB}$$

with p = effective value of sound pressure at measuring point in Nim2

$p_0 = 2 \times 10^{-5}$ N/m^2

$= 20\ \mu$ Pa

$= 2 \cdot 10^{-4}\ \mu$ bar

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

Evaluated Sound Pressure Level L_A

(= 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 ΔL 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.

[1] The sound pressure is often referred to as sound pressure level, see specifics under paragraph 3 "Fundamentals of Acoustics".

[2] The reason for the special nuisance created by a single tone is its information content (Example: Tones produced by sirens, warning and mating calls in the animal world).

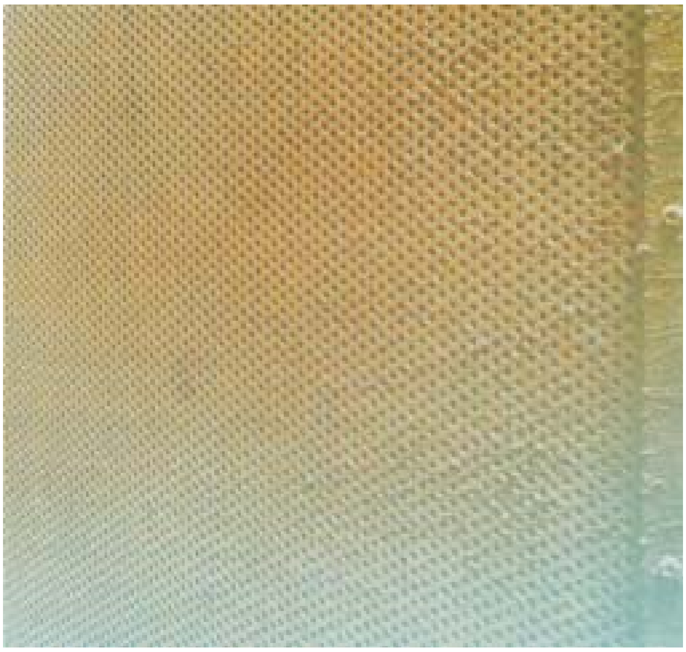
Baffles of the absorptive discharge silencer of a forced draft fan (after approximately 11,000 operating hours); fan performance data are shown on the right.
Below:

Close up photograph of the baffle wall~ after approximately 11,000 operatinc hours.

Three-stage absorptive silencer for ambient air inlet to a forced draft fan (Volume flow

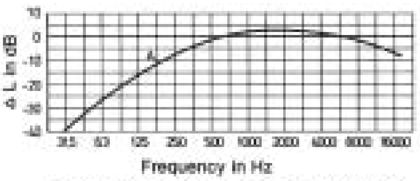
$\mathring{V}$ = 433 m^3/s, pressure increase Δp_t = 8250 Pa) in a power station.

Design point $\mathring{V}$ = 60%
Attenuation to sound pressure level 70dB (A).



Discharge silencer designed as a resonant silencer ($^{\lambda}/_{4}$-silencer or interference silencer) for two induced draft fans in a power plant (volume flow $\dot{V}$ = 2 x 660 m3/s, pressure increase Δp_t = 6520 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.



Evaluation Curve "A" per DIN 45633, Sheet 1

Figure 4

As can be seen in Figure 4, the numerical values for L_A are significantly below the L values at low frequencies and have a much smaller impact at higher frequencies.

Measuring Surface Sound Pressure Level $\bar{L}$ and $\bar{L}_A$

[“Meßflächen-Schalldruckpegel“ $\bar{L}$ und $\bar{L}_A$]

The-**measuring surface sound pressure level $\bar{L}$** (= 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).

$\bar{L}_A$ 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:

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

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:

$$\bar{L} \approx \frac{1}{n} \cdot \sum_{i=1}^{i=n} L_i$$

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 logarithmical parameter ratios used in acoustics the measuring surface in m^2, will be related to a reference area to define the **measuring surface level L_S** [“Meßflächenmaß“ L_S] as the characteristic parameter:

$$L_S = 10 \lg \frac{S}{S_0} \quad \text{in dB}$$

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.

$$L_W = 10 \lg \frac{W}{W_0} \quad \text{in dB}$$

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 ν (molecular movement velocity), [“Schallschnelle“ ν] and the measuring surface S:

$W = p \cdot \nu \cdot S$

using $\nu = \frac{p}{\varrho \cdot c}$

ϱ = air density
c = air sound velocity

it follows that:

$$W = \frac{p^2}{\varrho \cdot c} \cdot S$$

Assuming that ϱ = constant
c = constant

the proportional relationship obtained is:

$W \sim p^2 \cdot S$.

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

$$\bar{L}_W \approx \bar{L} + 10 \lg \frac{S}{S_0} = \bar{L} + \bar{L}_S \quad \text{in dB}$$

$$\bar{L}_{WA} \approx \bar{L}_A + 10 \lg \frac{S}{S_0} = \bar{L}_A + \bar{L}_S \quad \text{in dB}$$

The **sound power level L_W** can be approximated by the sum of the measuring surface sound pressure level $\bar{L}$ 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-Intensitatsspegel“ L_I]

At this point mention should be made of the so-called **sound intensity I** [“Schall-Intensitat“ 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 electricaltechnology 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:

$$L_I = 10 \lg \frac{I}{I_0} \text{ in dB}$$

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: $$f_m = \sqrt{2} \cdot f_U = \frac{f_0}{\sqrt{2}}$$

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

Acoustics

$$W = p \cdot \nu \cdot S$$

$$\frac{W}{S} = I$$

$$I = p \cdot \nu$$

$$\nu = \frac{p}{\varrho \cdot c}$$

$$I = \frac{p^2}{\varrho \cdot c} = \nu^2 \cdot \varrho \cdot c$$

$$I \sim p^2$$

Electrotechnics

N = Output
U = Voltage
I = Amperage
R = Resistance

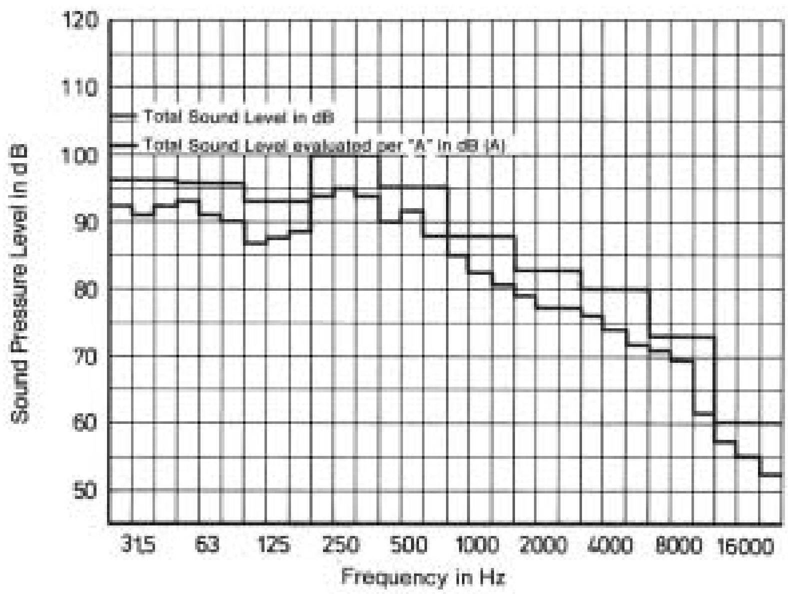
$$N = U \cdot I$$

$$I = \frac{U}{R}$$

$$N = \frac{U^2}{R} = I^2 \cdot R$$

$$N \sim U^2$$

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



"Terz"and Octave Spectrum of a Fan Noise

Figure 5

The individual center frequencies of the octave band are at:

31.5	Hz	1,000 Hz
63	Hz	2,000 Hz
125	Hz	4,000 Hz
250	Hz	8,000 Hz
500	Hz	16,000 Hz

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 devided 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 (sound pressure level or sound power level) will be added in accordance with the following equation:

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

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 Lw10 can be determined by the following:

$$L_{W\,tot} = L_{W1} + 10 \lg n$$

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 single sound sources with different levels, the total level is obtained by adding the difference of the individual levels to the higher level.

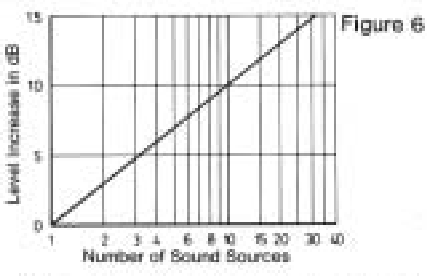


Figure 6

Addition of Several Sound Sources of Equal Levels

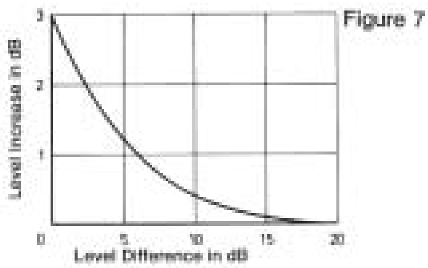


Figure 7

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 φ 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:

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

$$= L_1 + 20 \lg 2$$

If a difference in phase of 1 80 degrees exists ($\varphi_1 = 0^0$; $\varphi_2 = 180°$) or $^{\lambda}/_{2}$ interference results, the tones eliminate each other (see Figure 8).
These two occurances are of practical importance in the case where two fans are

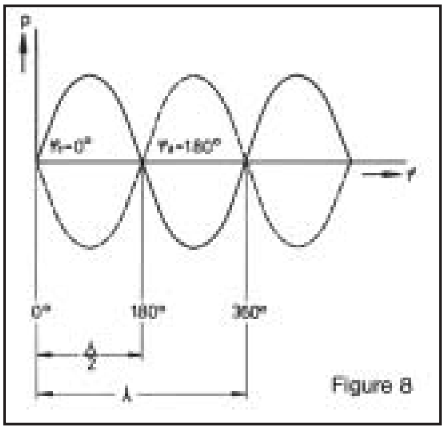


Figure 8

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.

6. Noise Developement 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

closed environment of the impeller is disturbed by obstructions with protruding edges (cut off in centrifugal fans and stationary guide vanes in axial flow fans). For fans such disturbing noise is also referred to as “blade passing tone“ [“Schaufelton“] or “blade passing frequencies“ [“Drehklang“], where the main disturbing frequency (base frequency) is the product of blade number times revolutions per second. Integer multiples of the base frequency can also occur as harmonics. The occurrence of the “blade passing frequencies“ (base frequency + harmonics) can, depending on the type and intensity of the disturbance, cause a significant increase of the sound power in individual frequency ranges.

7. Sound Pressure- and Sound Power Level of Fans

The sound **pressure** level Lofthefans can be pre-determined using fan tip speed, fan impeller diameter, and certain constants. Depending on fan type and performance data, average evaluated sound pressure levels L_A between 90 and 110 dB (A) are common (these values are usually measured at a distance of one meter from the fan and at an angle of 45 degrees to the flow direction).

As an approximation the **“A“-sound-power levels** can be pre-determined according to the following equation:

$$L_{WA} = K + 10 \lg \frac{\mathring{V}}{\mathring{V}_0} + 20 \lg \frac{p}{p_0} \text{ in dB (A)}$$

whereby:

- p = Total pressure difference in μ bar
- p_0 = 100 μ bar
- $\mathring{V}$ = Volume in m3/hr
- $\mathring{V}_0$ = 1 m^3/hr
- K ≈ 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 withi against 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_D.

L_{WS}: Level of the sound power radiating against the gas stream through the inlet area.

L_{WD}: Level of the sound power radiating with the gas stream through the outlet area.

Secondary emitted sound powers are: W_G, W_U, W_{SL} and W_{DL}.

L_{WG}: The sound power WG transmitted to housing walls evokes structure-borne sound (body sound) that radiates in the form of air sound to the surroundings. The respective sound power level is LWG.

L_{WU}: The structure-borne sound (body sound) of the housing is transmitted through sound conduction to fixed components of the housing (especially supports) from where it radiates in the form of air sound. The respective sound power level is L_{WG}.

L_{WSL}, The sound powers W_S,
L_{WDL}: W_D radiating as gas-borne sound through the inlet and outlet area of the fan evokes structure-borne sound (body sound) in the duct system which is not connected to the fan mechanically, but by expansion joints, and is therefore acoustically separated. This body sound in turn radiates to the surroundings in the form of air sound.
Respective sound power levels are L_{WSL} and L_{WDL}.

To determine these individual sound powers at the measuring surface at a distance of one meter from the fan (defined in section 3) an approximation can be made to calculate the sound power, emitted in the form of air sound, by means of the following equation:

$$L_{Wi} = L_i + 10 \lg \frac{S}{S_0}$$

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, described above.
For primary and secondary sound sources sound emissions 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**“)[“Schalldämmung“] and silencers on the other (“**sound attenuation**“) [“Schalldämpfung“].
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_{WG}, L_{WSL}, L_{WDL} 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 λ/4 silencers).

For both types of silencers, baffles ["Kulissen"] 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.

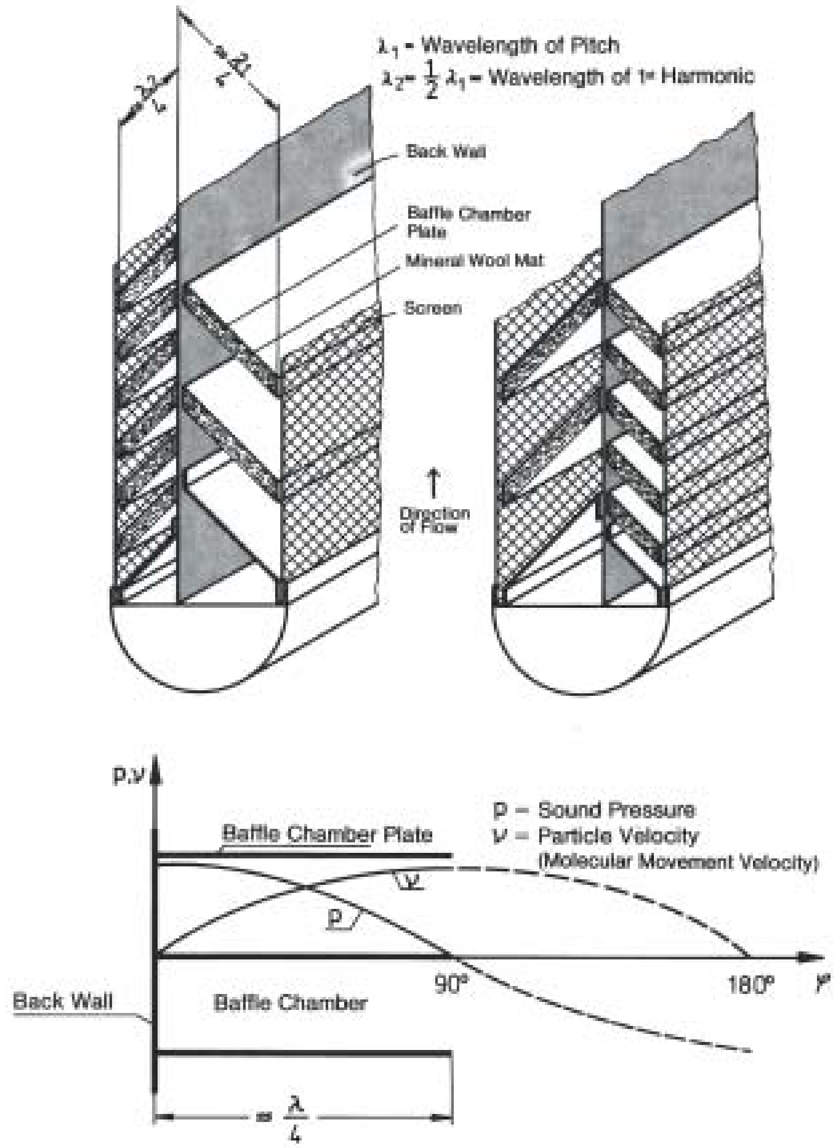
In the case of **absorption silencers** the space between the baffle walls, built of perforated plate, is filled with sound absorbing mineral wool.

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 (λ/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 = λ/4) to cause the following action:

At a distance λ/4 from the outer baffle wall the sound wave hits the solid back wall of the chamber, is reflected and travels another distance of λ/4 back to the sound source where the reflected sound wave, traveling 2 x λ/4 or λ/2 relative to the next following sound wave, arrives with a 180 degrees phase displacement and thus causes interference (tone elimination).



Behavior of Sound Parameters in a Baffle Chamber

Figure 9

Axial flow impulse induced draft fan in a power plantwith heat-sound insulation and lagging and discharge silencer.

Volume flow	$\dot{V}$	= 660 m^3/s
Temperature	t	= 156 °C
Pressure increase	Δp_t	= 6520 Pa
Speed	n	= 590 1/min
Shaft power	P_W	= 5480 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 LWD 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 LWS 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_{WS} through an inlet silencer
- Insulation of the sound power L_{WG}, L_{WDL}, L_{WSL} 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.

All sound powers stated in terms of levels

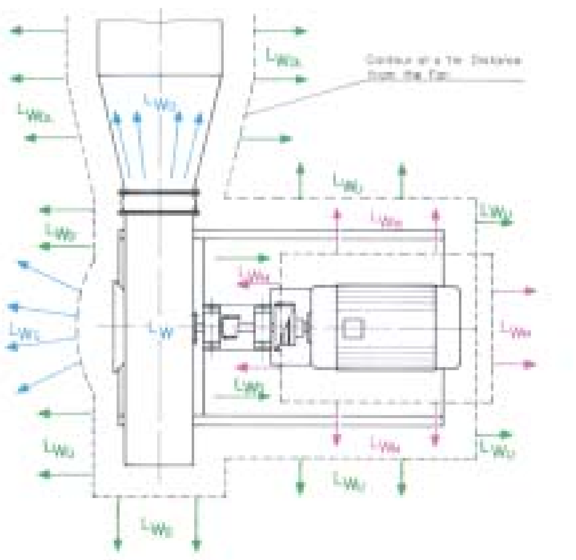


Fig. 10.1.

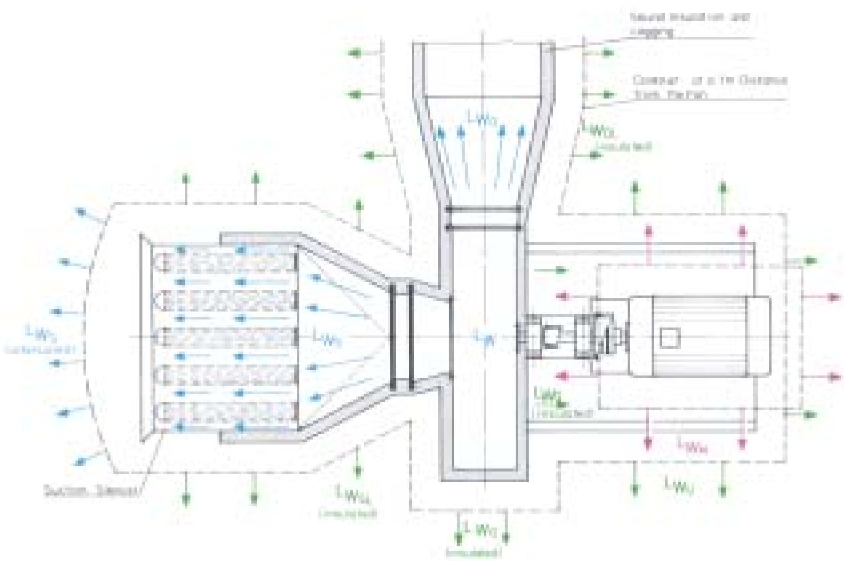


Fig. 10.2.

Sound attenuation: (through absorption): Sound energy penetrates through porous walls and is converted to thermal energy through viscosity friction.

Sound insulation: Sound energy strikes non-porous walls and is reflected.

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).

All sound powers are stated in terms of levels.

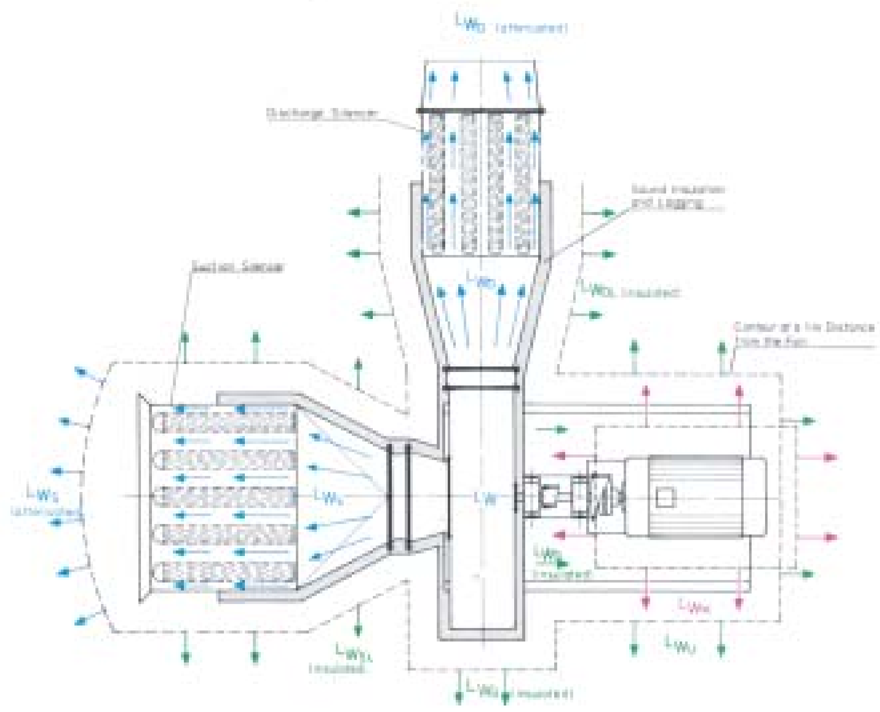


Fig. 10.3.

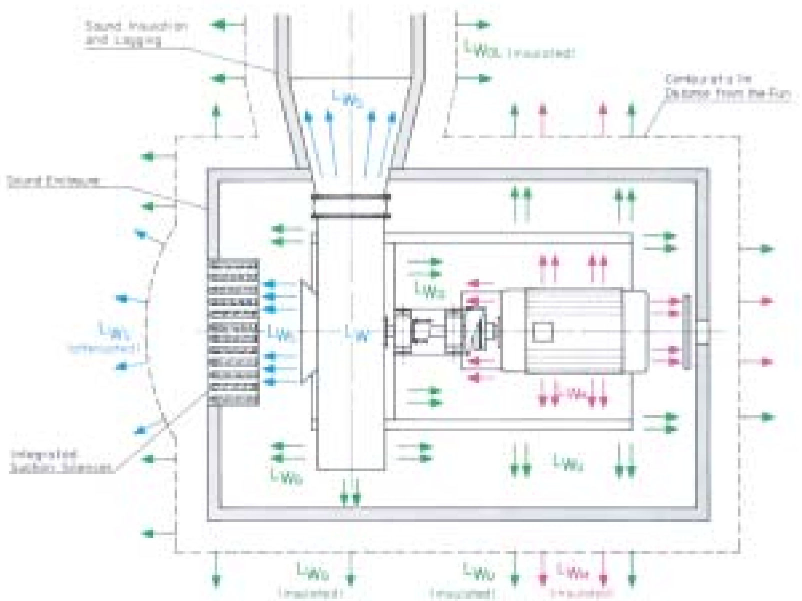


Fig. 10.4.

Stronger Together

Die TLT wurde Anfang 2003 von der Frankenthaler Kühnle, Kopp und Kausch AG übernommen und firmiert als eigenständige Konzerngesellschaft unter dem Namen TLT-Turbo GmbH.

Die Aktiengesellschaft KK&K, wurde im Jahr 1899 durch den Zusammenschluß dreier Familienunternehmen gegründet. Im Verlauf des vergangenen Jahrhunderts hat es die KK&K stets verstanden, den Veränderungen – vor allem technischen Erfordernissen – rasch anzupassen.

Mit der Zielsetzung, sich wieder auf das traditionelle Kerngeschäft zu konzentrieren, hat die KK&K die Turboladersparte im Jahr 1998 verkauft. Seither widmet sich der Konzern mit stetig wachsendem Erfolg wieder ausschließlich der Entwicklung, der Fertigung und dem Verkauf von Turbomaschinen – Turbinen, Verdichtern und Ventilatoren für sämtliche Anwendungsbereiche.

Bei allen den Tätigkeiten steht immer der Kunde mit seinen individuellen Bedürfnissen im Vordergrund. Auf der Basis dieser Kundenorientierung ist das gesamte Handeln von einem Miteinander geprägt – einem Miteinander mit dem Kunden, dem Mitarbeiter und dem Aktionär.

Eine ausgewogene Kombination von strategischer Weitsicht, kaufmännischer Umsicht und weltweiter Kooperation mit kompetenten Partnern macht die KK&K so erfolgreich und bestärkt uns in der Auffassung, bestens auf die Zukunft vorbereitet zu sein. Den Motor des Erfolgs bilden qualifizierte, zielorientierte und zufriedene Mitarbeiter sowie sehr moderne Geschäftsprozesse.

Neben dem Verfolgen wirtschaftlicher Ziele ist es der KK&K besonders wichtig, mit den Produkten einen aktiven Beitrag zum Umweltschutz zu leisten. Dies dokumentiert der Konzern mit der Wahl der Worte „Clean Air”, „Clean Water” und „Clean Energy” als strategische Leitbegriffe für das unternehmerische Handeln. Zum Konzern gehören neben der TLT-Turbo GmbH auch die HV-Turbo A/S in Dänemark sowie die PGW-Turbo in Leipzig.

Weitere Informationen zum Konzern erhalten Sie auf der KK&K-Website unter: www.agkkk.de

Immer in Ihrer Nähe

Zweibrücken
TLT-Turbo GmbH
Gleiwitzstraße 7
66482 Zweibrücken / Germany
Telefon: (06332) 80 80
Telefax: (06332) 80 82 67
E-Mail: tlt@tlt.de

Bad Hersfeld
TLT-Turbo GmbH
Serien- und Industrieventilatoren
Am Weinberg 68
36251 Bad Hersfeld / Germany
Telefon: (06621) 95 00
Telefax: (06621) 95 01 00
E-Mail: serie@tlt.de
E-Mail: industrie@tlt.de

Frankenthal
TLT-Turbo GmbH
Heßheimer Straße 2
67227 Frankenthal / Germany
Telefon: (06233) 8 50
Telefax: (06233) 85 21 12
E-Mail: tlt@tlt.de

Oberhausen
TLT-Turbo GmbH
Havensteinstraße 46
46045 Oberhausen / Germany
Telefon: (0208) 8 59 21 25
Telefax: (0208) 8 59 23 56
E-mail: r.graeber@tlt.de

Riedstadt
TLT-Turbo GmbH
Aussiger Straße 5
64560 Riedstadt / Germany
Telefon: (06158) 94 08 73
Telefax: (06158) 94 08 74
E-mail: e.matinjan@tlt.de

Australien
TLT-Turbo Pty. Ltd.
516 Guildford Road
Bayswater W.A. 6053, Australia
Telefon: 0061 - 8 92 79 14 02
Telefax: 0061 - 8 92 79 11 06
E-Mail: tlt@iinet.net.au

China
KK&K Beijing Representative Office
22D Building E, Majestic Garden
No. 6 North Sihuan Road
100029 Chaoyang District, Beijing
Telefon: 0086 - 10 82 84 26 84
Telefax: 0086 - 10 82 84 27 58
E-Mail: mashy@vip.163.com

Grossbritanien
KKK Limited
Oxford House, Oxford Street
NN8 4JY Wellingborough, Northants
Telefon: 0044 - 19 33 23 10 80
Telefax: 0044 - 19 33 23 10 90
E-Mail: kkk.limited@agkkk.de

Österreich
TLT-Turbo GmbH
Am Stadtpark 3 / 1734
1030 Wien / Österreich
Telefon: 0043 - 17 13 40 30 10
Telefax: 0043 - 17 13 40 30 30
E-Mail: mmayer@tltwien.at

Rußland
TLT-Turbo GmbH
ul. Profsojusnaja 45
117 420 Moskau
Telefon: 007 - 9 57 18 72 31
Telefax: 007 - 9 57 18 73 31
E-Mail: tlt-moskau@sovintel.ru

Spanien
PASCH Y CIA., S.A
Capitán Haya, 9 1°
E-28020 Madrid
Telefon: 0034 - 9 15 98 37 60
Telefax: 0034 - 9 15 55 13 41
E-Mail: info@madrid.pasch.es

TLT-Turbo GmbH
Industrial Fans
Am Weinberg 68
36251 Bad Hersfeld / Germany
Phone: + 49.6621.950-0
Fax: + 49.6621.950-115
E-mail: industrie@tlt.de
Website: www.tlt-turbo.de