Introduction

That sound and vibration generation and flow are correlated is apparent from a range of phenomena that we can observe around us. A noteworthy example is that of sound and vibration generation by a jet engine. Other examples are sound and vibration generation mechanisms of various flow machines, i.e., fans, pumps, compressors, and diesel engines. In all of these cases, the ultimate causes of the sound and vibration generation are non-stationary processes in the gases and liquids involved.

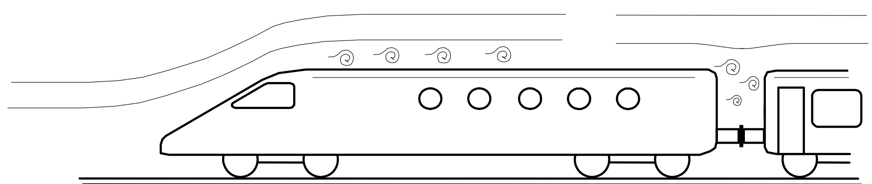


Figure 4.6 When vehicles move at high speeds, pressure fluctuations are generated, i.e., sound and vibration arises, due to the turbulent boundary layer of the flow field. [1]

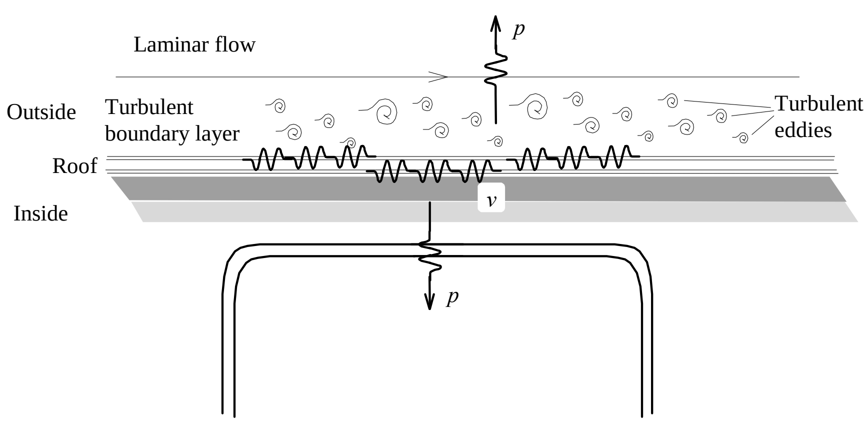


Figure 4.7 The pressure fluctuations generated by the turbulent vortices are responsible for noise both inside and outside of the vehicle. [1]

The basis for the analysis of these processes as acoustic sources is available from a theory developed by J.Lighthill in the early 1950's. Lighthill's theory of *flow acoustic* sound and vibration radiation is based on the assumption that there are only three basic types of sources that are possible in a fluid, namely monopoles, dipoles, and quadruples, and that all flow acoustic sources consist of some combination of these three basic types. Lighthill's theory also contains a justification of that assumption. The justification is based on a reformulation of the fundamental equations of fluid flow, the equation of motion and the equation of continuity, so that an acoustic wave equation with source terms is obtained. These source terms motivate exactly the three fundamental types mentioned above. A weakness of Lighthill's theory is that it ignores the interaction between flow and sound and vibration. In that theory, the flow field is considered to be a given source that is not influenced by the sound and vibration field it generates. That is never really the case, and a certain amount of interaction always occurs. In some cases, that interaction can be strong, and Lighthill's theory is not applicable. An important example of that is the so-called *whistle sound and vibration* caused by vortex shedding. In many cases, Lighthill's theory can, nevertheless, be applied successfully, and it is the most commonly-used model for the study of flow-induced sound and vibration.

The physical mechanisms corresponding to the 3 source types, and examples of when they occur, were discussed earlier in this chapter. Table 1 summarizes that.

Table 1 The three fundamental types of flow acoustic sources [1]

Source type	Physical mechanism	Physical situation
Monopole	fluctuating volume or mass flow	cavitation, inlets and outlets of piston machines (e.g., valves)
Dipole	fluctuating force	propellers, fans
Quadrupole	fluctuating force couple	free turbulence (jet flows)

Scaling laws for flow induced sound and vibration

In order to determine how the sound and vibration generation from a certain type of source depends on the flow conditions, scaling laws may be used. A scaling law can, among other things, be used to determine how much increase in sound and vibration power is obtained due to a change, e.g., when the flow velocity is doubled. Another use is to be able to rank the relative significance of the three source types, i.e., determine which type dominates in a given situation.

Equations that compare the sound and vibration power generated in a free field, between a dipole and a monopole, or a quadrupole and a dipole. These equations show that from small source regions, in terms of the *He* number,

$$\overline{W}_d / \overline{W}_m \propto (kd)^2 \quad \text{and} \quad \overline{W}_q / \overline{W}_d \propto (kd)^2 \tag{4.1}$$

where d is a length scale that indicates the size of the source region. To apply that to the case of flow acoustics, we must first be able to determine whether or not the source region is acoustically small. In other words, the wave number k must be known. For flow generated sound and vibration, a rule of general validity is that the frequency spectrum of the sound and vibration “scales to” (is proportional to) a frequency f_{st}, which is determined by a typical flow velocity U and a typical size d of the source region, as

$$\boxed{f_{st} = U / d} \tag{4.2}$$

That characteristic frequency f_{st}, for flow generated noise, is usually called the *Strouhal frequency*. The quantities U and d are to be chosen to characterize the source mechanism of interest. Examples of that, for some different cases, are provided in table 2.

In the first case (I), there is flow about a cylindrical barrier. Around that barrier, a periodic vortex shedding begins at very small Reynold’s numbe based on the diameter of the cylinder). That shedding gives rise to fluctuating forces, which correspond to dipole sources. The Strouhal frequency is obtained by choosing U as the velocity of the flow field and d as the diameter of the cylinder. The sound and vibration generated is relatively narrow banded; except for large Reynold's numbers, it is a tone-like sound and vibration, the *Strouhal tone*, with a frequency proportional to f_{st}.

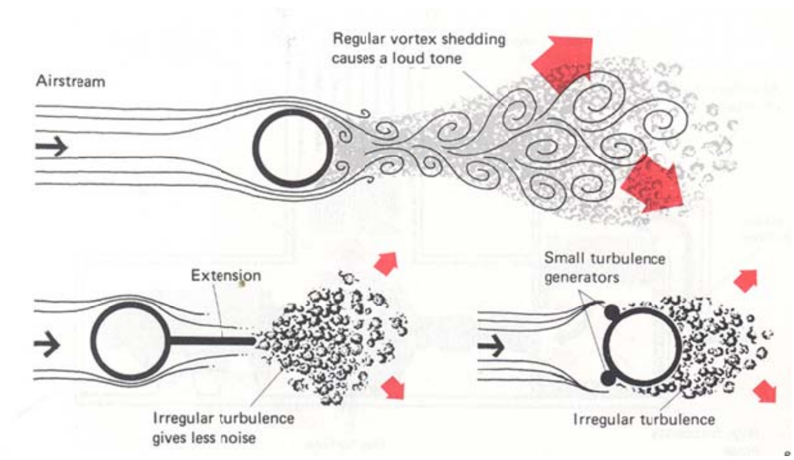
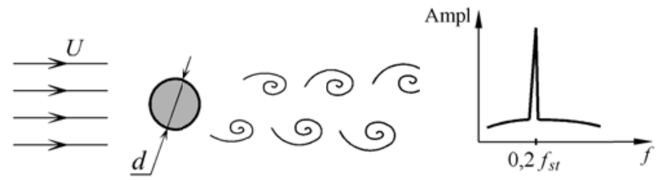
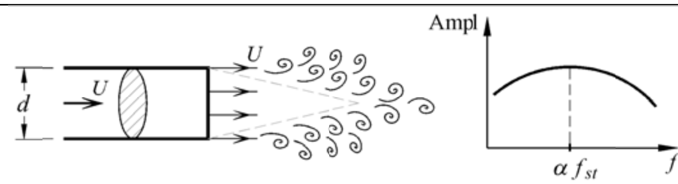
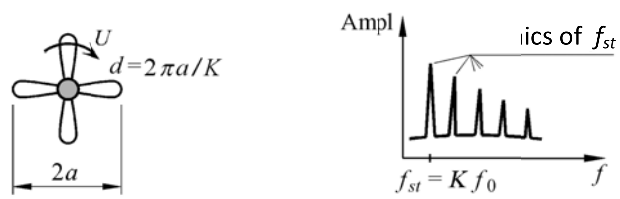


Figure 04.8 For a cylindrical pole in an air flow, there is a periodic shedding of vortices that gives rise to fluctuating forces. Sound and vibration generated in that way is called a Strouhal or aeolian tone. The shedding frequency is proportional to the so-called Strouhal frequency, which is determined by the flow velocity and the diameter of the pole. That sound and vibration generation can be diminished by reducing, in different ways, the regular generation of vortices. (Picture: Asf , Bullerbekämpning, 1977, Ill: Claes Folkesson) [1]

In the second case (II), a non-pulsating turbulent jet flow exits a duct. The jet corresponds to a distribution of quadrupole sources. The Strouhal frequency is obtained by choosing U to be the jet's velocity and d to be the diameter of the duct. The sound and vibration generated is broad-banded, with a frequency content that "scales to", i.e., is proportional to f_{st}.

In the third case (III), a propeller rotates in an otherwise still fluid, at a rotational frequency f_0. The blades of the propeller generate time-varying forces on the surrounding fluid, and thereby constitute dipole-type sources. From the perspective of a listener that does not move through the fluid, the time-variation of the forces has two causes: the blade rotation; and, the turbulence in the flow fields around the blades. The rotation brings about a periodic time dependence. If all of the blades are alike, the blade force distribution is repeated every time the propeller rotates through an angle 2*w* /*K*, where *K* is the number of blades.

Table 0 Choice of characteristic velocity, *U*, and characteristic length, *d*, for calculating the Strouhal frequency, $f_{st} = U/d$, in three different cases. [1]

Case I Periodic vortex shedding	 Vortex shedding frequency $f_{vs} = 0.2\ U/d$
Case II Turbulent jet	
Case III Propeller f_0 Rotational frequency *K* Number of blades	

If that angle is divided by a rotational frequency 2*w*□f_0, the period of rotation, $1/Kf_0$, is obtained. From that it follows that the fundamental frequency, i.e., the so-called *blade pass frequency*, becomes Kf_0. The sound and vibration from a propeller consists of a periodic part (harmonic multiples of the blade pass frequency), and of a broad band part corresponding to the turbulence contribution. For the periodic part, the Strouhal frequency corresponds to the blade passage frequency, and we can, for example, choose U as the peripheral velocity and d as $2\pi a/K$, where a is the radius of the propeller.

Equation (2) can be used to estimate the size of the source region, measured in the *He* number, for a flow acoustic source; the result is

$$He = kd = \frac{2\pi f_{st} d}{c} = \{f_{st} = U / d\} = 2\pi M \tag{4.0}$$

where $M = U/c$ is the *Mach number*. From that equation, it is evident that flow acoustic sources are small, acoustically, for small values of the Mach number. For such Mach numbers, equations (4.2, 4.3) can be used to give scaling laws that relate the three fundamental types of sources to each other. Putting the *He* number from equation (4.3) into these equations gives

$$\overline{W}_d / \overline{W}_m \propto M^2 \text{ and } \overline{W}_q / \overline{W}_d \propto M^2 \tag{4.4}$$

Equations (4.3) and (4.4) show that, for small values of the Mach number, the monopole is the most effective flow acoustic source type; after that, there is the dipole, and least effective as a radiator is the quadrupole. Besides ranking the sources, it is also of interest to know how the radiation from each type of source depends on the state of the fluid. Scaling laws that describe that can be obtained by first studying the monopole, and thereafter applying equations (4.2, 4.3). For the *monopole*, according to equation (4.5),

$$\overline{W}_m \propto \rho_0 c k^2 \tilde{Q}_0^2 \tag{4.5}$$

where the volume flow Q scales as follows:

$$\tilde{Q} \propto \text{hastighet} \times \text{area} = Ud^2 .$$

Putting that into equation (4.5) gives $\overline{W}_m \propto \rho_0 c k^2 \tilde{Q}_0^2 = \rho_0 c (kd)^2 U^2 d^2$; moreover, the wave number satisfies $k = \frac{2\pi f_{st}}{c} = \frac{2\pi U}{cd}$, so that

$$\boxed{\overline{W}_m \propto \rho_0 d^2 U^4 \big/ c} \tag{4.6}$$

Making use of equations (4.2, 4.3), the corresponding expression for a *dipole* is

$$\boxed{\overline{W}_d \propto \rho_0 d^2 U^6 \big/ c^3} \tag{4.7}$$

and for a *quadrupole*,

$$\boxed{\overline{W}_q \propto \rho_0 d^2 U^8 \big/ c^5} \tag{4.8}$$

The equations (4.5) to (4.8) only describe how the motion of the flow field, characterized by the velocity U, can be converted into sound and vibration. Physically, that means that these scaling laws describe how the kinetic energy in a flow field is converted to sound and vibration energy. For cases in which there are other energy sources in the flow field, e.g., thermal sources caused by combustion, more complicated scaling laws are required.

Equation (4.8) is the best known result from Lighthill's theory, and is usually called Lighthill's U^{10}-law. For low Mach-numbers, that result describes the sound and vibration radiat-ion from a jet in which thermal effects are negligible (a so-called cold jet). Although the equation, as derived, is limited to low Mach numbers, Lighthill assumed that it could also be applied to jet engines of airplanes. A number of experimental investigations have confirmed that assumption; see figure 30. In fact, equation (4.8) works up to a Mach number of about 1.5. The insight that jet noise follows the U^{10}-law has been an important factor in the development of quieter jet engines. The strong velocity-dependence implies that noise can be effectively reduced by reducing the engines' thrust velocities. By simultaneously increasing the area, that reduction is possible without reducing the total thrust.

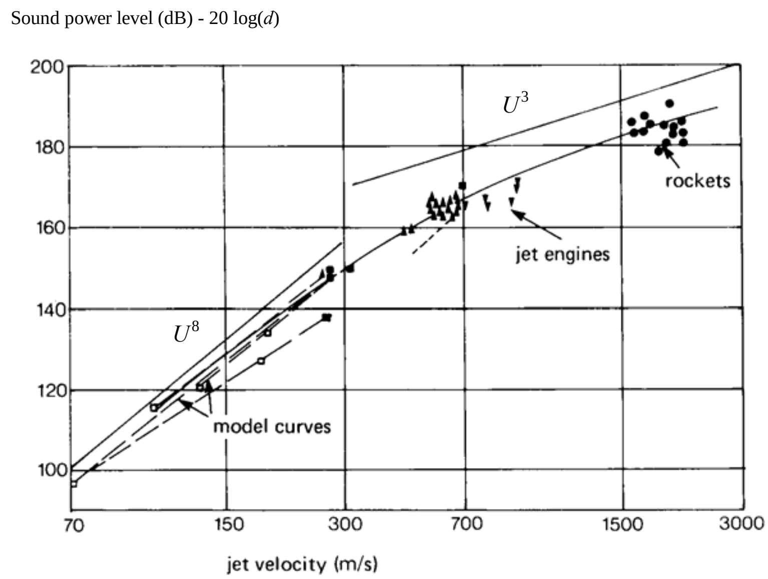


Figure 4.9 Velocity dependence of the sound and vibration generation from jet and rocket engines, from Chobotov & Powell, 1957. Note that the sound and vibration power is corrected for the size *d* of the sources by the scaling law given by equation (4.8).

As is evident from figure 4.10, the exponent of the velocity dependence falls off, and the sound and vibration power radiated is proportional to U^3 for very high thrust velocities. An explanation for that is that if the kinetic energy is the main energy reserve for sound and vibration generation, then the available energy per unit volume in a jet is $w_0U^2/\ 2$. That implies that the available power, corresponding to the outflow of kinetic energy per unit area at the outlet of the jet engine, increases as $w_0U^3/\ 2$. In the limit of high velocities, the maximum sound and vibration power that can be generated must asymptotically approach the curve for the available power, i.e., must grow at a rate proportional to the third power of the thrust velocity.

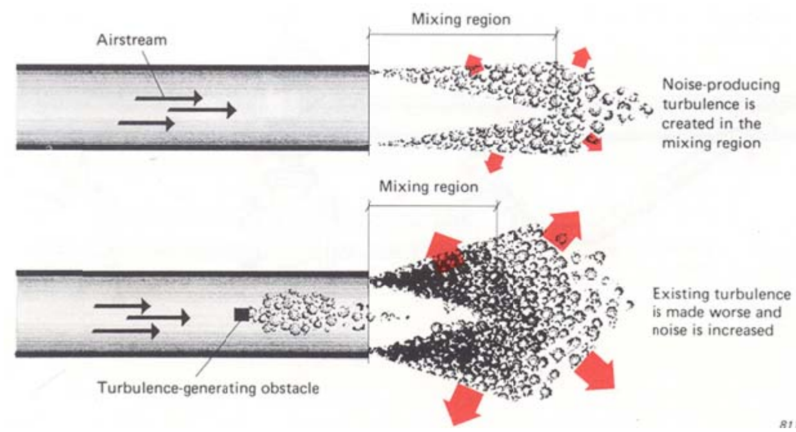


Figure 4.10 By reducing the velocity difference (gradient) in the mixing zone, the sound and vibration radiation of the jet can be reduced. (Picture: Asf, Bullerbekämpning, 1977, Illustrator: Claes Folkesson) [1]

The scaling laws discussed in this section have been derived under the assumption of three-dimensional (3-D) sound and vibration fields and free field conditions around the sources. In practice, the source region is often enclosed; consider the example of a throat in a duct with flow. That implies that cases with cylindrical (2-D) sound and vibration fields, as well as plane (1-D) sound and vibration fields, are also of interest. For the case of a duct, for instance, a plane wave sound and vibration field is obtained at low frequencies; see chapter 8. If an analysis corresponding to that carried out here is applied to the cases of 1- and 2-D sound and vibration fields. A summary of these scaling laws, for sound and vibration fields of arbitrary dimension, is provided in table 4.

Table 01 Flow induced sound and vibration. Scaling laws for sound and vibration power in sound and vibration fields with different dimensions. *U* is a characteristic velocity, and *d* a characteristic length. [1]

Dimension	Monopole	Dipole	Quadrupole
1-D	$\rho_0 c d^2 U^2$	$\rho_0 d^2 U^4 / c$	$\rho_0 d^2 U^6 / c^3$
2-D	$\rho_0 d^2 U^3$	$\rho_0 d^2 U^5 / c^2$	$\rho_0 d^2 U^7 / c^4$
3-D	$\rho_0 d^2 U^4 / c$	$\rho_0 d^2 U^6 / c^3$	$\rho_0 d^2 U^8 / c^5$

Whistle sound and vibrations

In some situations, strong interaction can occur between a sound and vibration field and a flow field. Examples of such situations are vortex shedding around a body in a flow field, or at a sharp edge. If the flow field is not too turbulent, the shedding is primarily periodic, and corresponds to a certain shedding frequency f_{vs}, which is proportional to the relevant Strouhal frequency, $f_{vs} = w$ □f_{st}. For cases in which the vortex shedding frequency coincides with a resonance frequency f_{res}, corresponding to an acoustic mode or a structural mode in a connected system, i.e.,

$$\alpha f_{st} = f_{res} \tag{4.9}$$

a *self-excited acoustic oscillator* can result. That condition is necessary, but not sufficient; to actually bring about a self-excited system, a **positive** feedback must also exist between the flow field and the connected system. When a self-excited system is obtained, the amplitude grows until it is limited by non-linearities and losses. Thus, this type of phenomenon can generate very strong tonal sound and vibrations, called whistling and is normally non-linear to its nature. The high levels are often a problem in a technical context. There are chiefly two ways to eliminate whistling sound and vibrations. Either the flow is disturbed, and the degree of turbulence increased, so that the periodic shedding breaks down, or the frequencies f_{vs} and f_{res} are separated by modifying something in the system, e.g., the flow velocity, length, or stiffness. Examples of situations in which whistle tones are generated are shown in figure 6.

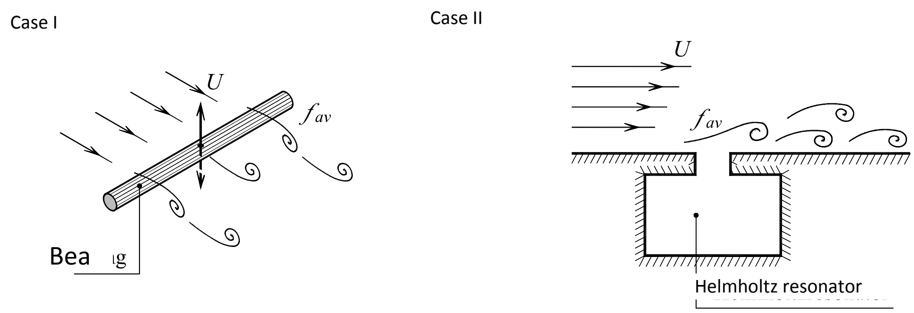


Figure 4.11 Two examples of how whistle tones can be generated in a flow field.

Case I Periodic vortex shedding from a bar in bending vibration. If the shedding frequency coincides with a bending resonance of the bar, a self-excited acoustic system can arise. A practical example of periodic vortex shedding is that from a high speed electric train’s pantograph (linkages extending from the train to contact and draw power from the trackside electrical net).

Case II Periodic vortex shedding at a hole with sharp edges coupled to a resonator. If the shedding frequency coincides with the Helmholtz resonance, a self-excited acoustic oscillator can result.

A practical example of flow-induced whistle noise is that sound and vibration which is generated by a garden trimmer. The operation of the trimmer is based upon striking grass stems and thin branches with a thin nylon chord. In the flow field around the nylon chord, there is a periodic vortex shedding that results in a pronounced tone. That tone can be largely eliminated if the cross section of the chord is made elliptical, rather than circular.

Wind generated tones arising from high smoke stacks serve as another example of sound and vibration caused by periodic vortex shedding. The purpose of the spiral shaped flanges that can be seen wrapped around such smoke stacks is to eliminate that noise source by disturbing the vortex shedding; see figure 6.

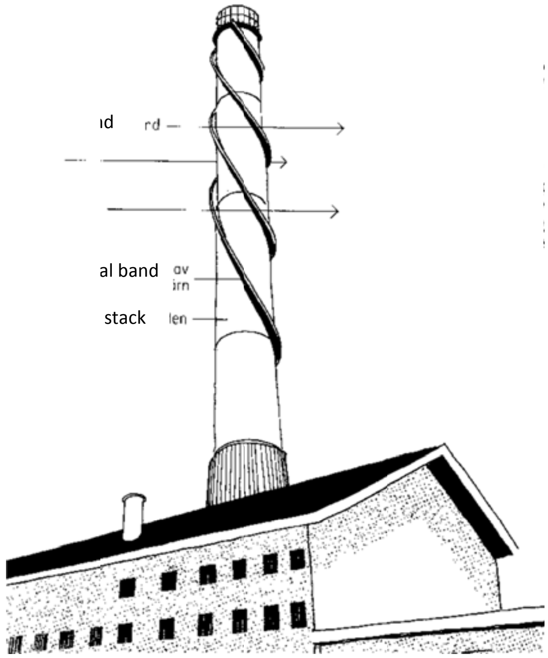


Figure 4.12 The whistling sound and vibration generated by a periodic vortex shedding from a smoke stack can be eliminated by wrapping the stack with a spiral band. That effectively disturbs the vortex generation. (Picture: Asf, Bullerbekämpning, 1977, Ill: Claes Folkesson) [1]

Finally, we note that the howling tone that is sometimes generated by a planer is caused by periodic vortex shedding at an opening coupled to a Helmholtz resonator in the form of a cavity; see figure 4.12. A possible countermeasure is to eliminate the cavity by filling it with rubber filling.

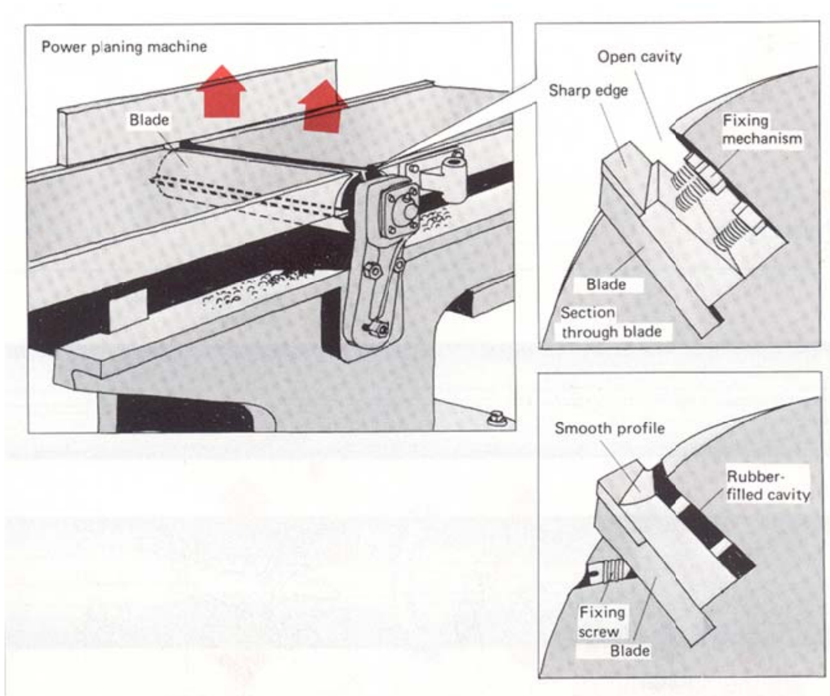


Figure 4.13 A planer can, when rotating, generate a powerful shrieking sound and vibration. That sound and vibration is generated by a broad-band vortex shedding along the edge of the planer blade. Certain tones in the generated sound and vibration are then strongly amplified in the cavity of the blade. That sound and vibration can be eliminated by filling the cavity, and thereby preventing the amplification. After filling the cavity, only a hissing noise remains. (Picture: Asf , Bullerbekämpning, 1977, Ill: Claes Folkesson) [1]

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