Introduction to the noise of rotating machines

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MOTIVATION

- Cooling fans and turbo-chargers on cars and trucks
- Ventilation fans for vehicles and buildings
- Gasturbines for aircrafts and airplanes
Sound from moving sources – FWH equation

Ffowcs-Williams Hawkings equation is a reformulation of Lighthill's acoustic analogy for moving bodies.

\[
\left(1 - \frac{\partial^2}{c_0^2 \partial t^2} - \nabla^2\right)(p'H) = \frac{\partial}{\partial t} \left(\rho_0 V_i n_i |\nabla f| \delta(f)\right) - \frac{\partial}{\partial x_i} \left(p' n_i |\nabla f| \delta(f)\right) + \frac{\partial^2}{\partial x_i \partial x_j} \left(\rho u_i u_j H(f)\right)
\]

Volume displacement ~ Monopoles
Fluctuating pressures ~ Dipoles
Unsteady Reynolds stresses or transport of momentum ~ Quadrupoles

The motion (body surface) is described by a function \(f(x,t)=0\) and it is further assumed that \(f < 0\) inside the body and \(f > 0\) outside.
Aerodynamic source strength – scaling laws

For aerodynamically generated sound the time averaged sound power \( \overline{W} \) will scale as:

\[
\overline{W} \sim \rho U^3 D^2 M^{\alpha+n},
\]

where \( M \) is Mach-number, \( n \) the space dimension (1,2,3) and:

\[
\alpha = \begin{cases} 
-2, & \text{monopole} \\
0, & \text{dipole} \\
2, & \text{quadrupole}
\end{cases}
\]
Relative strength fluid driven aeroacoustic sources

\[ \overline{W}_{\text{monopole}} : \overline{W}_{\text{dipole}} : \overline{W}_{\text{quadrupole}} \propto 1 : M^2 : M^4 \]

- Combustion
- Piston machines (in/out flow openings)
- Cavitation
- Fans
- Flow separation
- Free jets

\[ [M=\text{Mach-number}=U/c_0] \]
The fluctuating blade pressures (dipoles) are always an important source of sound for rotating machines. Assuming a rotor in a circular duct with $B$ blades and an angular rotational frequency $\omega_0$, the acoustic pressure in a cross-section close to the rotor can be written:

$$p'(r, \theta, t) = \sum_n f_n(r) e^{inB(\theta-\omega_0 t)},$$

assuming that all blades are equal and uniformly spaced.

This result implies that the steady rotating blade pressures will produce a harmonic spectrum at multiples of the blade passing frequency ($\text{BPF}$): $Bf_0$. 
The angular rotational speed of the pressure pattern is:

$$\omega_{\text{rot}} = \omega_0$$

i.e., the same as the rotor.

As shown in textbooks (see e.g. Goldstein *Aeroacoustics*) to radiate sound a rotating in-duct pressure pattern must be supersonic, i.e.,

$$\omega_{\text{rot}} a > c$$

where $a$ is the rotor radius.

Thus a ducted rotor with subsonic tip speed will NOT radiate sound.
Assume now a fixed inlet disturbance consisting of \( V \) equal guide vanes also uniformly spaced. The resulting (down stream) pressure close to the rotor can then be written:

\[
p'(r, \theta, t) = \sum_m \sum_n f_{mn}(r) e^{imV \theta + inB(\theta - \omega_0 t)},
\]

The angular rotational speed of the pattern is now:

\[
\omega_{rot} = \frac{nB \omega_0}{nB + mV},
\]

Since \( m, n \) are pos/neg integers this expression can produce rotational frequencies much larger than \( \omega_0 \) AND supersonic pressure patterns that radiates (propagates as modes).
In many cases, e.g., aero-engines it is essential to block the strongest first (n=1) BPF. As first shown by Tyler and Sofrin* this can be achieved by choosing:

\[ V \geq 2B, \]

Low speed (subsonic) machines

Axial fan   Radial fan   Mixed flow fan

Fans create a flow by rotating blade forces acting on the fluid!
A little aerodynamics

To characterize the aerodynamic performance of a fan the increase in (total or stagnation) pressure is plotted as function of the volume flow $Q$. This is often called a fan curve.

Increasing fan RPM

Radial fans in general have a larger $\Delta P/Q$ ratio than axial fans.
A little aerodynamics

Normally the purpose of a fan is to produce a certain volume flow $Q$ for ventilation or cooling purposes. In order to do this the fan must overcome the flow related losses in the attached system. These losses can be described by the system or pressure drop curve...
A little aerodynamics

Normally the purpose of a fan is to produce a certain volume flow $Q$ for ventilation or cooling purposes. In order to do this the fan must overcome the flow related losses in the attached system. These losses can be described by the system load or pressure drop curve....
Fan noise sources and transmission paths

Fan

Flow generated sound in the system

Aerodynamic sound from the fan

Airborne sound in ducts/pipes

Radiation from openings

Stucture borne sound due to fan motor and unbalances

Radiation from fan housing

Stucture borne sound in duct/pipe walls

Radiation
Fan noise mechanisms

- Fluid displacement

- Forces on surfaces

- Turbulence in fluid
Sources of sound

* Thin blades implies weak volume displacement sources for LOW speed fans
* Due to rotation + non-uniform (time and space) inflow conditions time varying blade pressures and forces are created
* Free turbulence

For small fans (D < 0.5 m) the Mach-number is typically less than 0.3…
Relative strength of the sources

\[ W_{\text{monopole}} : W_{\text{dipole}} : W_{\text{quadrupole}} \propto 1 : M^2 : M^4 \]

Of no importance

Can be important at edges

The dominating contribution !!!
The Neise chart (1990)

**Fan Noise**
- discrete + broadband

**Monopole**
- blade thickness noise
  - discrete

**Dipole**
- Blade forces
  - discrete + broadband

**Quadrupole**
- Turbulence noise
  - broadband

**Steady rotating forces**
- (Gutin noise)
  - discrete

**Unsteady rotating forces**
- discrete + broadband

**Uniform stationary inflow**
- discrete

**Non-uniform stationary inflow**
- discrete

**Non-uniform unstationary inflow**
- discrete broadband

**Vortex shedding**
- narrow-band broadband

**Secondary flows**
- narrow-band broadband
General expression for the field from a moving dipole

\[
p_d(x, t) = \frac{\mathbf{F}_e \cdot \mathbf{e}_{r,e}}{4\pi c r_e \left(1 - \mathbf{M}_e \cdot \mathbf{e}_{r,e}\right)^2} + \frac{\mathbf{F}_e \cdot \mathbf{e}_{r,e} \left(\mathbf{M}_e \cdot \mathbf{e}_{r,e}\right)}{4\pi c r_e \left(1 - \mathbf{M}_e \cdot \mathbf{e}_{r,e}\right)^3} + \\
- \frac{\mathbf{F}_e \cdot \mathbf{M}_e}{4\pi r_e^2 \left(1 - \mathbf{M}_e \cdot \mathbf{e}_{r,e}\right)^2} + \frac{\mathbf{F}_e \cdot \mathbf{e}_{r,e} \left(1 - \mathbf{M}_e^2\right)}{4\pi r_e^2 \left(1 - \mathbf{M}_e \cdot \mathbf{e}_{r,e}\right)^3}
\]
General expression for the field from a moving dipole

\[ p_d(x, t) = \frac{\dot{F}_e \cdot e_{r,e}}{4\pi cr_e \left(1 - M_e \cdot e_{r,e}\right)^2} + \frac{F_e \cdot e_{r,e} \left(\dot{M}_e \cdot e_{r,e}\right)}{4\pi cr_e \left(1 - M_e \cdot e_{r,e}\right)^3} + \]

- Fluctuating force
- Rotating steady force (“Gutin Noise”)

Near field

\[ -\frac{F_e \cdot M_e}{4\pi r_e^2 \left(1 - M_e \cdot e_{r,e}\right)^2} + \frac{F_e \cdot e_{r,e} \left(1 - M_e^2\right)}{4\pi r_e^2 \left(1 - M_e \cdot e_{r,e}\right)^3} \]
Fluctuating blade forces

Two parts:

* A periodic part that is repeated at each blade passage. This creates a tonal spectrum at harmonics of the blade passing frequency, $Bxf_0$ where $f_0$ is the rotational frequency and $B$ is number of blades.

* A random part that is related to unsteady (turbulent) inflow + flow separation. This creates a broad-band spectrum.

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| “Weak tones” (Gutin noise in a duct this will create no sound at all) |
| Amplified tones |
| Amplified tones |
| Tones + broadband |
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Forces on Blade Cascade

spatially nonuniform inflow $\Rightarrow$ *tonal noise* ("Unsteady loading noise")
unsteady inflow $\Rightarrow$ *broad band noise*
A detail: Airfoil self-noise

- Turbulent boundary layer/blade surface interaction (TBS)
- Flow separation (FS)
- Turbulent boundary layer/trailing edge interaction (TBTE)

⇒ mostly broadband noise
Secondary sources, e.g. tip clearance flow
Example of fan spectrum (sound power) for a train cooling fan

The dashed lines are BPF:s. The other harmonics are from the hydraulic drive.
Scaling laws for fan noise

We know that...

\[ W_{\text{dipol}} \propto D^2 U^{3+n} \]

For a fan with diameter \( D \): \( U \sim D f_0 \) which implies...

\[ W_{\text{fan}} \propto D^2 \left( D f_0 \right)^{3+n} = D^{5+n} f_0^{3+n} \]

For most fans measurements show that:

\[ W_{\text{fan}} \propto D^{\alpha+2} f_0^\alpha \text{ with } \alpha \approx 5. \]
Classification of prediction methods

Classification of fan noise prediction methods

CLASS I
Basic machine parameters
- type
- diameter
- speed
- flow rate
- pressure rise

Simple algebraic function (correlation)

CLASS II
- Separate consideration of various noise generation mechanisms
  - Simplified fan geometry, flow field (e.g. blade ⇒ flat plate)

Acoustic models for all noise generation mechanisms

CLASS III
- Separate consideration of various noise generation mechanisms
  - Detailed fan geometry and flow field (e.g. from CFD-computation)

(SPECTRAL) SOUND POWER
Examples of prediction methods

Class I
Regenscheit-method; VDI-Richtlinie 3731

\[
L_{w,ges}^* = L_{w,ges} - 10 \log \left( \frac{\dot{V}}{V_0} \right) \Delta P_1 \left( \frac{1}{\eta} - 1 \right) = L_{w,ppez,R} + 10 \cdot m \cdot \log \left( \frac{u}{c_0} \right) \text{ dB}
\]

\Rightarrow \text{Specific sound power level for various types of fans}
A class II – noise prediction method (II)

Example: Turbulent Ingestion (TI)

Velocity fluctuations of turbulent flow: curve fit to dimensionless experimental results from various turbulence generators

\[
\frac{d\bar{w}^2}{df} = \bar{w}_\infty \cdot T u^2 \cdot A \cdot 10^{\frac{1}{10} F(Sr_\lambda)}
\]

⇒ Lift force fluctuations in terms of modeled turbulent velocity fluctuations

\[
\frac{dP_{ak,TI}(f)}{df} \approx \text{const} \cdot B \cdot \frac{\rho}{c_0^3} \cdot \frac{w_\infty^4 \cdot d\bar{w}^2}{df} \cdot C \cdot L
\]

Typical result

- Prediction „smooth“
- Only broad band
- Very fast method

A Class III – noise prediction method


Noise control at the source i.e. minimize the unsteady forces ($F$)

- Reduce fan size $D$ and rotational speed $f_0$ and keep performance ($\Delta P = \text{increase in stagnation pressure}$) by larger (=area), curved and aerodynamically more efficient blades.....
Noise control at the source

- Minimize inflow disturbances, e.g., guide vanes & stators at least 2-3 chord lengths upstream of the blades.

- Since the fan pressure increase scales as: \( \Delta P \sim \rho U^2 \), it follows that the acoustic power scales with \( \Delta P \) as:

\[
W_{\text{fan}} \propto U^\alpha \propto \Delta P^{\alpha/2}
\]

- The last result implies that for a given \( Q \) one should design systems with minimum pressure drop \( \Delta P \), i.e., reduce the losses!!!
Noise control at the source

Furthermore the result implies that if we have a system where we need to overcome a pressure drop $\Delta P$ it will produce less noise if we do it in two or more steps!

Example: Two fans in series each giving a pressure increase of $\Delta P/2$. Each fan produce:

$$W_{\text{fan}} \propto (\Delta P / 2)^{\alpha/2} = \Delta P^{2.5} / 2^{2.5}.$$ 

The total power is: 

$$2 \cdot W_{\text{fan}} \propto \Delta P^{2.5} / 2^{1.5}$$

Which compared to a single fan gives an estimated reduction of: 

$$\Delta L_{W, \text{fan}} = 4 - 5 \text{ dB}$$
System losses and noise

Given a desired volume flow $Q_0$ and a fan. For each RPM of the fan we obtain a fan curve. One alternative to reach the desired $Q_0$ is to choose a fan which produce a higher $Q_0'$ (point 1) and then introduce a constriction (valve) and change the system curve. This will produce extra flow generated sound in the system and also use more energy. A better alternative is to reduce the fan RPM so that the fan curve meets the original load curve at the desired $Q_0$ (point 2).
The resulting sound field depends also on the acoustic response ("Greens functions $G$") for the blade forces, which depends on the surrounding system where the fan operates. For small fans we can assume that they create a single resulting fluctuating force which implies:

$$\hat{p}(\omega, x, y) = \hat{G}(\omega, x, y) \cdot \hat{F}(\omega)$$
Noise control via the response function $G$

- In a system with resonances it is important to put the fan at a position where the response function has a small value, in particular at the blade passing harmonics.

- Alternatively damping can be introduced to create a more "free field" type of response ("constant $G$") so the output is independent of the position.
Installation effects – *aerodynamic* and *acoustic*

When a fan is installed in a car, PC, ….inflow disturbances will normally increase the sound power as compared to the ideal (ISO) testing. In addition the acoustic response (radiation conditions), in particular for low frequencies, can modify the output.
High speed (supersonic) machines

The machines can either be gas turbine compressor/turbine stages or IC-engine turbo-compressor units
Sound generating mechanisms

• The aero-dynamic sources will be addressed here but structure borne sound, e.g., inertia forces related to unbalances can also be important for high speed machines.

• All the three aero-acoustic mechanisms can play a role: monopole (rotating shock waves + combustion), dipole (fluctuating forces on blades + guide vanes), quadrupole noise (free jets).

• The gas turbine is an example of an axial machine while turbo-units for IC-engines often are mixed flow machines.
Jet engines...
A modern high by-pass ratio (low noise) jet engine
Sound generating mechanisms...

- The free jet noise is only important for aero-engines

- The rest of this material will focus on IC-engine turbo-units (the compressor part) but the same source mechanisms also exist for the gas turbine case.
Turbo-Compressors for cars, trucks…

- Air filter + silencer
- Mixed flow compressor
- Main rotor blade
- Splitter blade
- To charge air cooler/intercooler
- To ATD and exhaust silencer system
- Axial exhaust gas turbine

No. 9
Turbo-Compressor noise sources and transmission paths

TC

Flow generated sound in the system

Aerodynamic sound from the TC

Airborne sound in ducts/pipes

Radiation from openings

Structure borne sound due to bearings and unbalances

Radiation from TC body

Structure borne sound in duct/pipe walls

Radiation
Sound Generating Mechanisms

Fan Noise
discrete + broadband

Monopole
Rotating shock waves
discrete

Dipole
Blade forces
discrete + broadband

Quadrupole
Turbulence noise
broadband

Steady rotating forces
(Gutin noise) discrete

Unsteady rotating forces
discrete + broadband

Uniform stationary inflow
discrete

Very small !!

Non-uniform stationary inflow
discrete

Tip Clearance Noise
Vortex Shedding
narrow-band peak

Non-stationary inflow
broadband
Monopole
- Shock-wave induced “buzz-saw” noise

Dipole
2. Tonal noise at blade passing frequency (BPF)
3. Blade tip clearance noise (TCN)
4. Noise from the “rotating stall” effect
5. Secondary (“horse shoe”) flow

Here 3-5 represents so called secondary flow effects i.e. related to leakage flows or separation flow at blade tips or edges.
A compressor wheel of a passenger car turbocharger N (rpm)

\[ BPF = B \cdot \frac{N}{60}, \text{ where} \]

B is the number of main rotor blades

Averaged sound pressure level in the compressor inlet duct after Raitor&Neise AIAA/CEAS 2006
The "buzz-saw noise" starts when the tip speed becomes supersonic. Buzz-saw noise or rotating shock waves occurs at multiples of the rpm.

Averaged sound pressure level in the compressor inlet duct after Raitor&Neise AIAA/CEAS 2006.
Tip clearance noise (TCN) and rotating stall are important at subsonic speeds. TCN is normally important at low volume flows and creates a broad band peak at 0.5*rpm.

Averaged sound pressure level in the compressor inlet duct after Raitor & Neise AI AA/CEAS 2006
Summary Sound Generation

- Tip clearance noise (TCN) associated with vortex formation across blade edges/tips at low (subsonic) rpm’s.

- Rotating shock waves or buzz-saw noise at supersonic tip speeds. Generates harmonics at multiples of the RPM.

- Rotor tonal noise this mechanism creates tones at multiples of the Blade Passing Frequency (BPF) and increase strongly with the rpm. Dominates the overall level for high rpm’s.

The aerodynamic power increases as $\text{RPM}^3$ BUT the aero-acoustic power increases as $\text{RPM}^6$ .... this shows that noise problems will tend to increase when the rpm is increased to boost engine performance...
Noise control at the source

Example of measured sound pressure spectrum from (at 1 m) a large turbo-unit for a marine diesel. The main radiated noise comes from the 1:st and 2:nd BPF.

For a mixed flow or radial machine the amplitude of the BPF is strongly affected by the geometry of the outlet diffusor. In particular the separation between the blade tips and the diffusor guide vanes leading edge.
By increasing the relative separation between blade tips and the diffuser guide vanes the BPF:s can be affected...

The main effect for this case is on the second BPF.

An increase of 20% gives a reduction of 10 dB of the 2:nd BPF.