6. GAS MOVING EQUIPMENT AND FLOW-STRUCTURE INTERACTIONS

Flow-structure interactions covered by this guide include Air Inlet Debris Screen, Compressors and Exhausters, Fans and Blowers, and Flow Noise from Pipes and Fittings.

Large-scale vortices in undisturbed flow produce aerodynamic quadrupoles and noise having a predominantly low frequency character without a definable peak. In the absence of obstacles in the flow, this type of noise dominates. Additional noise is generated through turbulence whenever flow encounters a structure. Dipole sources of differing characters are created and may become the dominant noise generation mechanism.

6.1. Noise Generation by Inlet Debris Screen and Fixed Obstructions

Narrow-band tonal sound is generated by vortices in the oscillating wake of slender obstructions such as wires, pipes, and struts. Although each individual vortex produces only shear forces, the succession of vortices with alternating rotational sense (Karmann vortex street) produces a series of dipoles that radiate with peak frequency

$$f_p = 0.2\frac{U}{D},$$

where U is the characteristic velocity of the flow and D is the characteristic dimension of the obstruction. Significant levels of upstream turbulence can increase noise emission.

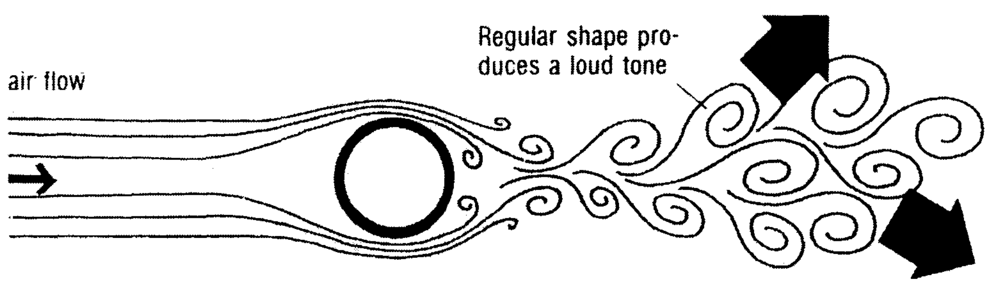


Figure 6: Vortex Street

(Ingemansson and Folkesson[23])

6.2. Noise Generation by Rotor-Stator Interactions

Lifting surfaces, such as those employed in compressors, exhausters, fans and blowers also creates dipoles. When the dipoles rotate with machinery, the repetitive pattern of fluctuating pressures passing a given point produces a narrow band tone known as the *blade passage tone*. This tone and its integer harmonics are a hallmark of this type of equipment. The intensity of this tone increases strongly with total static pressure rise across the blades.

The rotational speed of the pressure patterns from the rotor-stator interactions is

$$N_q = \frac{nBN}{q}$$

where N_q is the rotation rate of the q-th rotating pressure pattern
N is the rotation rate of the rotor shaft
B is the number of rotor blades
V is the number of stator blades
n and k are positive integer numbers
and $q = nB \pm kV$

For small values of the denominator (i.e., when kV is subtracted), the tangential speed of the pressure pattern at the *ND/2* exceeds the blade tip speed, and can approach sonic velocity. Noise emission increases dramatically under these conditions. This can be avoided by designing so that q is large.

The frequency of the tone emitted by the q-th pressure pattern is simply the n-th harmonic tone:

$$f_q = nBN$$

6.3. Noise Generation by Compressors and Exhausters

Quadrupole radiation from large-scale vortices in the flow produces a broadband noise spectrum, to which are added a tone or tones related to mechanical action.

In reciprocating equipment a single low frequency tone is produced that corresponds to the number of pressure pulses produced per second. A rotary lobe compressor also produces essentially one mid-frequency tone, but the frequency is somewhat higher.

For rotating machinery such as axial and centrifugal compressors, the tones are typically high frequency and are produced by blade passage and rotor-stator interactions. The sound power developed depends on blade tip speed to the fifth power and horsepower squared.

6.4. Noise Generated by Fans and Blowers

The mechanism of fan noise generation is similar to that for compressors and exhausters, the main difference between the equipment being the pressures developed. Because fans and blowers are typically low-pressure devices, their mechanical and acoustical performance is strongly influenced by downstream conditions.

Broadband noise generation is a function of the blade type, flow, total static pressure rise, and operating point. Fan scaling laws relate the flow, pressure developed and acoustic power output to the rotation rate and diameter of homologous fans at the same operating point as follows:

Table 7: Fan Scaling Laws

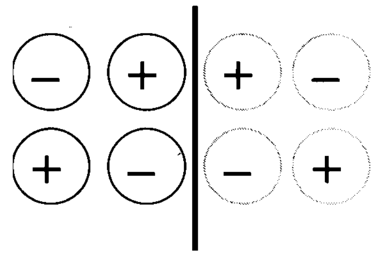
Blade Tip Speed = ND
Flow $\propto ND^3$
Total Static Pressure Rise $\propto N^2D^2$
Power Transmitted to Flow $\propto N^3D^5$
Acoustic Power $\propto$ Flow $\times$ Pressure^2 $\propto N^5D^7$

The fan scaling laws show that the ratio of acoustic power to mechanical power is proportional to the total static pressure rise. From this it follows that minimizing system pressure losses can help reduce noise emission. With system pressure reduced, it is usually possible to select a larger fan rotating more slowly to deliver the required flow. This is especially true for the blade passage tone, which intensifies dramatically as the total static pressure rise increases.

Furthermore, it is possible to deduce a general rule regarding noise emission from fans. The tradeoff between diameter D and rotation rate N is very important. A larger fan turning more slowly is generally preferred, as long as it operates near maximum static efficiency.

Maximum static efficiency corresponds to maximum air movement for minimum mechanical work, and as expected corresponds to minimum specific noise emission (noise emission per work done) for a given fan.

6.5. Flow Noise in Pipes and at Fittings



Broadband noise is generated in the boundary layer clinging to pipe walls. Pressure fluctuations from large-scale vortices in the turbulent flow reflect from pipe walls, producing a reinforcing pair of oscillatory forces rather than an opposing pair. The result is a series of dipoles at the pipe perimeter. The spectrum of sound within the pipe is dominated by low frequencies and contains no peaks.

The roughness of the pipe and presence of fittings such as wyes and tees increases flow noise output. These elements can dramatically increase the turbulence level in the pipe and hence the mechancial power available to be converted into acoustic energy.

6.5.1. Casing-Radiated Noise

Casing-radiated Noise arises from flow-induced and acoustically-induced vibrations of the casing. For fans and blowers, the casing is often a thin piece of flat sheet metal. For compressors and exhausters, the casing is constructed primarily of thick curved plates.

Casing-radiated noise is usually not an issue unless the inlet and outlet openings and ductwork are effectively silenced.

6.6. Reduced-Noise Design for Inlet Debris Screen and All Fixed Obstructions

- ***Streamline objects in the flow path.*** Sharp edges in the flow should be avoided, because they can produce locally high flow velocities and shock waves that increase noise emission. A steam train whistle is a relevant example. Rounded leading edges and streamlined trailing edges should be employed on all flow obstructions. Structural supports should be devoid of projections such as screws, welds, etc.
- ***Trailing edge boundary layer trips*** may also be useful in destabilizing the vortex street.

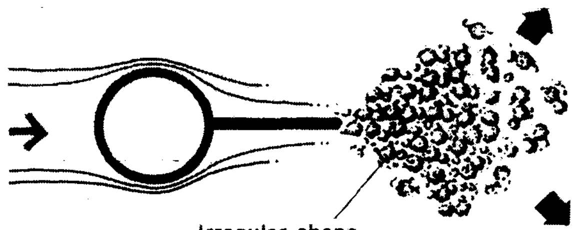


Figure 7: Reducing Vortex Tone

(Ingemansson and Folkesson[23])

- ***Minimize turbulence.*** Reducing turbulence minimizes the mechanical energy available for conversion into sound.
- ***Detune f_p from duct modes.*** The vortex shedding frequency of any flow obstacle should be selected below the first mode cut-on frequency of the pipe or duct. Above this frequency, resonant coupling between the vortex street and the sound field could lead to strong tones. The first mode cut-on frequency for a circular pipe is $0.586\ c/D$, and for a rectangular duct $0.500\ c/D_l$, where D_l is larger of the two pipe cross-sectional dimensions.

6.7. Reduced-Noise Design for Gas-Moving Equipment

Useful references for further investigation in this area include Universal[24], NASA[25], and Burgess-Manning[26].

- ***Reduce turbulence***: allow at least one diameter of straight duct flow before a compressor or exhauster inlet (See Figure 8 below).
- ***Select a larger machine*** operating at lower RPM. This will probably require that system pressure losses be minimized.
- ***Design for a high lobe number q***. The number of rotors and stators, B and V respectively, should not be equal. Nor should they be related by near integers (e.g., 3 and 4). Larger prime numbers are preferred where V and B are widely spaced.
- ***Design for cutoff***: Choose V and B to achieve a cutoff factor less than 1.05. Fundamental tone is attenuated 8 dB. See Section 7.1 (page 7-1).

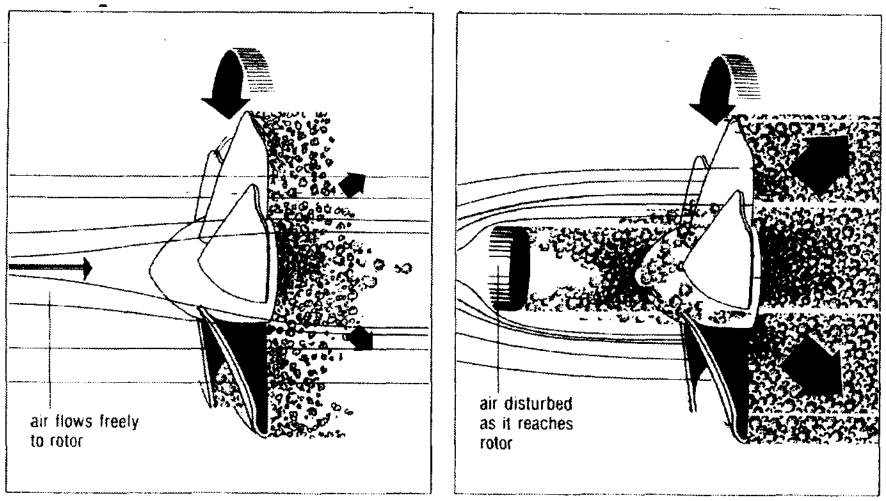


Figure 8: Effect of Turbulence Upstream of Rotor

(Ingemansson and Folkesson[23])

- ***Detune f_n from duct modes***: Select duct and impeller so that as many rotor/stator tones f_n, and at least the fundamental tone, lie below the first duct mode cut-on frequency. The first mode cut-on frequency for a circular pipe is $0.586\ c/D$, and for a rectangular duct $0.500\ c/D_l$, where D_l is larger of the two pipe cross-sectional dimensions.
- ***Apply silencers***: Reciprocating and rotary lobe blowers are usually best serviced by reactive silencers because the silencers are more compact and do not require acoustical fill which could be degraded by oil mist in the discharge. Typical silencers for axial and centrifugal compressors are typically dissipative. Any exposed piping and ductwork between the unit and a silencer should be lagged.
- ***Modify casing***: once the inlet and discharge have been effectively silenced, casing noise may require attention. Additional stiffening members welded directly to the casing performs most effectively on flat plates and in general attenuates mainly low frequencies. Adding damping directly to the casing chiefly attenuates high frequencies if the thickness of the damping compound is comparable to the thickness of the material and if resonant radiation is present. Mass should only be added directly to the casing when the mass per unit area can be increased by at least 50%. In each case, an approximate 5 dB benefit is available in the associated frequency ranges.
- ***Apply acoustical lagging to casing and ducts***: Acoustical lagging consisting of a layer of sound insulation material (2-in., 4-in. or 6-in. thickness) and a limp, massive covering (1 psf) may be applied to the exterior of the casing, piping and

ductwork. Useful attenuation is available for high frequencies (1000 Hz and greater) for all thickness. Lower frequencies require thicker lagging.

- ***Vibration isolation*** may be necessary, in particular for reciprocating compressors and exhausters. Remember that vibrational energy can be converted into sound most efficiently by structures that are relatively thin and have large areas. Large equipment should therefore be sited on grade with a properly designed foundation block.
- ***Maximize Rotor/Stator spacing:*** Rotor/stator spacing should be at least 1.5 rotor chord widths. Further reductions of rotor/stator interaction tones, on the order of $2 \times RSS/C_2$ *dB(A),* can be achieved by further increasing spacing, where *RSS* is the rotor/stator spacing and C_2 is the rotor chord.

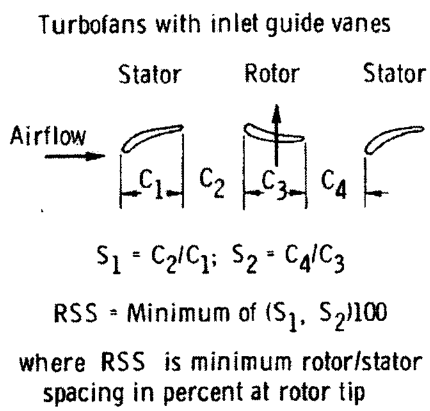


Figure 9: Rotor-Stator Spacing Coefficient

- ***Position stators downstream*** of the rotor whenever possible. Rotor-stator interactions are stronger for upstream stators than for downstream stators. Use inlet guide vanes only when required.

6.8. Reduced-Noise Design for Fans and Blowers

- ***Minimize system pressure losses.***
- ***Select a large fan rotating slowly.***
- ***Select the quietest wheel type*** appropriate for the service.
- ***Set the operating point*** within 5% of maximum static efficiency. A 1 dB(A) noise increase occurs for every 5% of max operating efficiency below max operating efficiency.[27]

- ***Use variable speed motors*** to control flow rather than inlet guide vanes, control valves, or other restrictive flow devices.
- ***Minimize upstream turbulence:*** Poor inflow conditions can lead to a condition called rotating stall, which produces a rumble and tone centered on 2/3 the shaft rotation frequency. Require 1.5 to 2.0 diameters of straight duct upstream of inlet.
- ***Avoid unstable flow regimes:*** Centrifugal and vaneaxial fans are unstable at operating points to the left of maximum static efficiency. In this region there are some pressures for which two different flow rates are possible. Operating in this region could cause the fan to surge back and forth between the two operating points. The surge frequency depends on the length of the attached piping.
- ***Select vibration isolation*** based on lowest rotational speed for variable speed systems.
- ***Orient discharge and downstream turns*** to have the same rotational sense as the flow. Otherwise, turbulence-induced rumble can result. If an elbow must be placed within 1.5 duct diameters of the discharge, the elbow shall have a long radius and incorporate turning vanes.

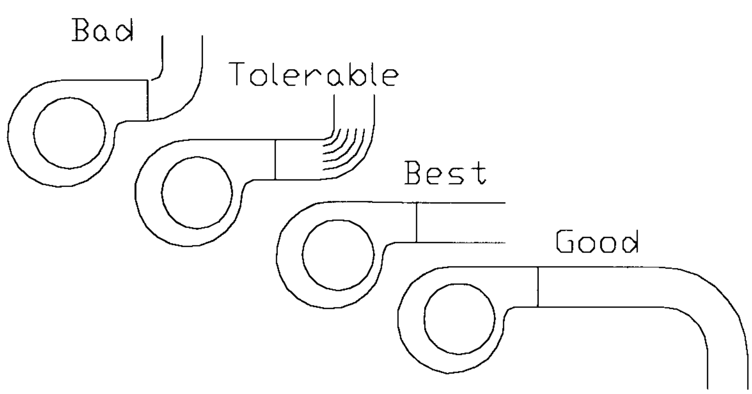


Figure 10: Proper Orientation of Discharge Turns

(after Schaffer[28])

6.9. Noise Reduction Recommendations for Flow

- ***Reduce flow velocity*** to approximately $85/\rho^{1/2}$ feet per second, where ρ is in pounds per cubic feet. "Economic velocity" for flow in the pipe may be somewhat lower. Consult relevant piping codes for other velocity limitations.

- ***Reduce number, abruptness and density of fittings*:** Prefer gradual transitions, welded over screwed or mitered bends, etc.
- ***Increase pipe wall thickness.*** Doubling the pipe wall thickness affords approximately 3 to 5 dB(A) additional attenuation.
- ***Apply acoustical lagging*** to the piping.

6.10. Noise Emission Estimation Using Workbook

Spreadsheets, with Required Inputs and Notes:

- Inlet Debris Screen

Required Inputs:

- A_S, m', D_W, POA, r

Notes:

- Percentage Open Area is that of the screen area not occluded by wires or other obstructions.

- Compressors and Exhausters

Required Inputs:

- W_M, D_I, N, B, L, H, W, r

Notes:

- For some equipment types casing noise and the noise from an unmuffled inlet are broken out separately. For others, they are reported together.

- Fans and Blowers

Required Inputs:

- Blower Type, m', P_{TS}, N, B, SE, PSE, Silencer IL, Silencer $L_{W,SN}$, L, H, W, r

Notes:

See Table 9 for K_W values for specific blower wheel types. If problems are occurring in a particular frequency band, look for the wheel type with the lowest K_W in that band.

➢ Flow Noise

Required Inputs:

- m', P_1, T_1, D_p, t_p, L, K, r

Notes:

- In Part 4, enter the number of each type of fitting that appears in a 10-ft. section of pipe. Technically, this refers to an individual 10-ft. long section. A coarse approximation for the overall system may be obtained by entering the average number of each fitting appearing per 10-ft. of pipe.

6.11. Predictive Equations for Inlet Debris Screen

The sound power level of noise emission from an inlet debris screen is estimated (after Beranek and Ver[29]) as

$$L_W = L_{W,overall} + F\left(f/f_p \right)$$

$$L_{W,overall} = 10 + 10\log_{10}\left(S\xi^3 U^6\right) + 10\log_{10}(1-M)$$

where S is the screen area in square meters, U is the flow velocity in meters per second, M is the Mach number of the flow, and ξ is an effective head loss coefficient combining the coefficient of drag of a small cylinder and the percent open area (POA) of the screen, equal to

$$\xi = 1.1\left(1 - \frac{POA}{100}\right).$$

The spectral shape function $F(f/f_p)$ is approximated by

$$F\left(f/f_p \right) = -0.2 + .384\log_{10}\left(f/f_p \right) - 1.09738\left(\log_{10}\left(f/f_p \right)\right)^2$$

6.12. Predictive Equations for Compressors and Exhausters

Noise emission equations for compressors and exhausters are taken from Heitner[30] as given in Bies and Hansen[31]. The equations are believe to be equally valid for use with exhausters.

6.12.1. Centrifugal Compressors

For centrifugal compressors and exhausters, the overall sound power level measured at the discharge piping inside the pipe is given by[31]

$$L_{W,overall} = 20\log_{10} W_M + 50\log_{10} U - 45,$$
$$L_W = L_{W,overall} + F(f)$$

where U is the impeller tip speed in meters per second (limited to the range 30 to 230 ms^{-1}) and W_M is the mechanical power of the drive motor in kilowatts. The peak frequency is[31]

$$f_p = 4.1U \quad [Hz].$$

The spectrum level in the octave band containing f_p is taken as 4.5 dB less than $L_{W,overall}$. The spectrum rolls off at the rate of 3 dB per octave above and below the peak frequency.

Noise estimates for the casing and for the unmuffled inlet are[31]

$$L_{W,overall}\Big|_{Casing} = 79 + 10\log_{10} W_M$$
$$L_{W,overall}\Big|_{Inlet} = 80 + 10\log_{10} W_M$$

The spectral corrections of Table 8 are subtracted from the corresponding overall sound power level value to give octave band sound power levels.

Table 8: Octave Band Corrections for Compressor and Exhauster Inlets and Casings

	31.5	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*
Centrifugal Casing	10	10	11	13	13	11	7	8	12
Centrifugal Inlet	18	16	14	10	8	6	5	10	16
Rotary and Recip. Inlet and Casing	11	15	10	11	13	10	5	8	15

6.12.2. Rotary or Axial Compressors

The overall sound power level at the pipe exit may be estimated as[31]

$$L_{W,overall} = 68.5 + 20\log_{10} W_M$$

The peak frequency is that of the second blade harmonic

$$f_p = B\frac{N}{2}$$

where N is the number of rotations per second and B the number of blades.

The sound power spectrum is assembled from estimates for the 63 Hz and 500 Hz octave bands, the octave band containing f_p and the octave band containing the frequency $f_h = f_p^2/400$: [31]

$$L_W\big|_{63} = 76.5 + 10\log_{10} W_M$$
$$L_W\big|_{500} = 72 + 13.5\log_{10} W_M$$
$$L_W\big|_{p} = 66.5 + 20\log_{10} W_M$$
$$L_W\big|_{h} = 72 + 13.5\log_{10} W_M$$

A straight line is drawn between these points and the slope is continued for octave bands outside these points.

Casing noise (including partially muffled air inlets) is estimated as[31]:

$$L_{W,overall} = 90 + 10\log_{10} W_M$$

The frequency corrections given in Table 8 are subtracted from overall sound power levels to give the octave band sound power level values.

6.12.3. Reciprocating Compressors

The overall sound power level in the exit piping of a compressor can be estimated as:

$$L_{W,overall} = 106.5 + 10\log_{10} W_M$$

The peak frequency is that of the cylinder frequency

$$f_p = BN$$

where B is here the number of cylinders. The spectrum level in the octave band containing f_p is taken as 4.5 dB less than $L_{W,overall}$. The spectrum rolls off at the rate of 3 dB per octave above and below the peak frequency.

Casing noise (including partially muffled air inlets) is as given above under Rotary Compressors.

6.13. Noise Emission From Fans and Blowers

Fan and Blower noise is estimated according to a method developed initially by Buffalo Forge[27] and published later in ASHRAE[32]. Bies and Hansen[31] added a correction for static efficiency. The octave band sound power level of a fan or blower is estimated as

$$L_W = K_W + 10\log_{10} Q + 20\log_{10} P_{TS} + \left(\frac{.95 - \frac{SE}{PSE}}{.05} \right)$$

where Q is flow rate in cubic feet per minute, P_{TS} is total static pressure rise across the fan in inches of water column, and K_W is tabulated in Table 9 for various fan types.

The column BFI is the Blade Frequency Index, which is an increment added to the octave band containing the blade passage frequency,

$$f_b = BN .$$

Table 9: Specific Sound Power Level K_W by Fan Type

Wheel Type	*31.5*	*63*	*125*	*250*	*500*	*1000*	*2000*	*4000*	*8000*	*BFI*
Centrifugal, AF, BC, or BI, D > 30"	37	37	37	36	31	27	20	16	14	3
Centrifugal, AF, BC, or BI, D < 30"	42	42	42	40	36	31	25	21	16	3
Centrifugal, FC, All Sizes	50	50	50	40	33	33	28	23	18	2
Radial, 4" - 10" SP, D > 40"	53	53	44	40	36	34	29	26	23	7
Radial, 4" - 10" SP, D < 40"	64	64	56	50	40	39	36	31	28	7
Radial, 10" – 20" SP, D > 40"	55	55	51	42	39	35	30	26	23	8
Radial, 10" – 20" SP, D < 40"	65	65	60	48	45	43	38	34	31	8
Radial, 20" – 60" SP, D > 40"	58	58	55	50	45	43	41	38	35	8
Radial, 20" – 60" SP, D < 40"	68	68	64	56	51	51	49	46	43	8
Vaneaxial, Hub Ratio 0.3 to 0.4	46	46	40	40	45	44	42	35	13	6
Vaneaxial, Hub Ratio 0.4 to 0.6	46	46	40	43	40	38	33	27	25	6
Vaneaxial, Hub Ratio 0.6 to 0.8	56	56	49	48	48	46	44	40	37	6
Tubeaxial, D > 40"	48	48	43	44	46	44	43	36	34	7
Tubeaxial, D < 40"	45	45	44	46	50	49	48	40	37	7
Propeller, D < 12 ft.	45	45	48	55	53	52	49	43	39	5

6.14. Noise Estimation for Flow in Pipes

The following method was suggested by Seebold (1973)[33] and continues to receive widespread acceptance. The method estimates the noise that results from the boundary layer pressure fluctuations in fully developed flow in uninterrupted straight circular pipes, and then applies a loss-factor correction K for local discontinuities. The Sound Pressure Level (presumably at 1 meter from the pipe) in the octave band centered on frequency f is estimated from the flow velocity, gas density ρ, pipe thickness T and diameter D, ring frequency f_r of the pipe, and a spectral correction S:

$$L_P\big|_{1m} = 40\log_{10} U + 20\log_{10}\rho + 20\log_{10} K - 10\log_{10}\left[\frac{t_p}{D_p}\left(1+\frac{6}{D_p}\right)\right]\ldots$$

$$\ldots - 5\log_{10}\left|\frac{f}{f_r}\left(1-\frac{f}{f_r}\right)\right| + \Delta L_P$$

where U is in feet per second, ρ is in pounds per cubic foot, t_p and D_p are in feet, and f and f_r are in Hertz. The ring frequency for steel pipe is approximately *5275/D*.

The spectral correction ΔL_P depends on ratio of the octave band center frequency f to the peak frequency f_p as

$$\Delta L_P = \begin{cases} 11.4\log_{10}\dfrac{f}{f_p} + 10.4 & \text{if } \dfrac{f}{f_p} < 0.5 \\ 7 & \text{if } 0.5 \le \dfrac{f}{f_p} < 5 \\ -10\log_{10}\dfrac{f}{f_p} + 14 & \text{if } 5 \le \dfrac{f}{f_p} < 12 \\ -36.1\log_{10}\dfrac{f}{f_p} + 41.9 & \text{if } \dfrac{f}{f_p} \ge 12 \end{cases}$$

The loss-factor K is determined by adding the individual loss factors K_i for the flow fittings and elements present within a 10 ft. length of pipe. The loss-factors K_i are tabulated below in Table 10.

The most correct way to perform this estimation is to evaluate each individual 10-foot segment. The aggregate noise emission is computed from the sum of the individual noise emissions (see Appendix B).

An alternative method is to compute K based on the average number of components appearing in a 10-foot section over the length of the piping (e.g., 0.8 ninety-degree turns per 10-foot section would be entered where 4 turns are present in a 50 foot piping run). The estimated noise emission should then be assumed to be present along the evaluated length. Note that the latter method does not allow identification of localized noise sources and hot spots.

Table 10: Loss Factors K_i for Pipe Flow Noise

Straight Pipe		0.12				
45° Elbow	Screwed	0.42	Welded, R/D=1	0.20	Welded, R/D=1.5	0.11
90° Elbow	Screwed	0.98	Welded, R/D=1	0.45	Welded, R/D=1.5	0.32
180° Elbow	Screwed	3.00	Welded, R/D=1	0.80	Welded, R/D=1.5	0.43
Tees (Screwed)	Thru Branch	1.80	Thru Run	0.50		
Tees (Welded)	Thru Branch	1.40	Thru Run	0.40		
Reducer	D2/D1= 0.3	0.25	D2/D1= 0.5	0.17	D2/D1= 0.7	0.07
Expander	D2/D1= 3	0.8	D2/D1= 2	0.58	D2/D1= 1.25	0.1
Sudden Contraction	D2/D1= 0.1	0.48	D2/D1 = 0.33	0.41	D2/D1 = .80	0.12
Sudden Expansion	D2/D1= 10	0.98	D2/D1= 3	0.7	D2/D1= 1.25	0.12

[23] Stig N. P. Ingemansson, Claes Folkesson, *"Noise Control: Principles and Practice"*, Noise News International, Vol. 3 No. 2 1995 June, pp. 120-127 and No. 4 1995 Dec., pp. 238-243. Also published by the American Society of Safety Engineers as *"Noise Control: A guide for workers and employers"*.

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[32] *1991 Applications Handbook*, Chapter 42: Sound and Vibration Control, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1991

[33] J. G. Seebold, "Smooth piping reduces noise – fact or fiction?", *Hydrocarbon Processing*, 189-191, September, 1973

7. TURBOMACHINERY

The noise sources described in this section are related to the operation of turbomachinery. Aircraft engines and their derivatives are typically items of research interest at NASA Glenn Research Center. Although reducing noise of these components is an integral part of GRC's work, they are not candidates for the industrial noise control methods in this *Guide*. The noise they generate may however have significant noise control ramifications and must be accounted for in the system design.

The aircraft engine is considered as a combination of separate components. Components addressed include:

- Inlet Fan and Compressor
- Combustor and Core
- Turbine
- Jet Mixing
- Jet Shock-Associated Noise

Noise emission predictions are based on empirical correlation studies. Superior estimates may also be available within computer models developed by Clark[34]. Related methods are described in *Aeroacoustics of Flight Vehicles*[35].

7.1. Inlet Fan and Compressor

Inlet fan and compressor noise generation in an aircraft engine differs little from other industrial axial compressors, with the exception of the first stage high bypass ratio fan, the absence of a long inlet duct, and a different approach to expressing the design parameters.

The noise emission estimates given below could be used for in-duct sound power of large industrial axial compressors if one integrates the power over all angles. In this case select an observation angle of 0° to get an appropriate sound power level estimate.

Noise emission estimates are computed after a NASA Glenn model by Heidmann[36]. The noise emission is shown to be related to the work performed by the fan and compressor, as expressed by the temperature rise or pressure ratio, the mass flow rate, the tip Mach number M_{TR} and design tip Mach number M_{TRD}, rotor/stator spacing and distance and direction of observation.

Broadband and tonal noise is estimated for both inlet and discharge. Combination tones are estimated for first stage fans. In the Workbook, the assumption has been made that discrete tones are increased due to additional turbulence experienced in static test stand operations.

Because the noise control of these devices is outside the scope of the *Design Guide*, and because the source documents should be readily available at NASA Glenn Research Center, the rather lengthy equations have been omitted.

The basic noise emission equation for all broadband and tonal noise estimates is of the form:

$$L_p = 20\log_{10}\left(\frac{\Delta T}{{}^\circ R}\right) + 10\log_{10}\left(\frac{\dot{m}}{1\ \text{lbm/sec}}\right) + F_1(MTR, MTRD) + F_2(RSS) + F_3(\theta) + \ldots$$
$$\ldots + F_4\left(f/f_b\right) - 20\log_{10}\left(\frac{r}{1\ \text{m}}\right)$$

The inlet noise peaks at an angle about 30° from the inlet, and discharge noise about 110° from the inlet. The most useful area for noise reduction is represented by the F_2 term, where reductions of the order of 5 and 10 dB(A) are possible by increasing rotor/stator spacing. Another potential area for reducing noise involves arranging the rotor/stator interaction to achieve cutoff. A simplified approach to establishing the cutoff condition is

$$\delta = \left|\frac{M_T}{1 - V/B}\right| \le 1.05$$

where V is the number of stators (vanes) and B the number of rotors (blades). When the cutoff condition exists, the fundamental blade passage tone is reduced 8 dB.

7.2. Combustor and Core noise

Combustor and Core noise is estimated using the method of ARP 876C (1985)[39].

The overall sound power level is estimated from the mass flow rate m', combustor inlet total pressure P_3, combustor total temperature rise $T_4 - T_3$, reference total temperature extraaction by the turbines at maximum takeoff conditions $(T_4\text{-}T_5)_{ref}$, and the temperature, pressure and sonic velocity for sea level standard conditions.

The noise emission varies as 10 $\log_{10} m'$ and as 20 $\log_{10} (T_4 - T_3)$ and 20 $\log_{10} (P_3)$. The peak frequency f_p is apparently always close to 400 Hz and the farfield radiation is only moderately directional, peaking at an angle of 60° from the inlet.

Because the noise control of these devices falls outside the scope of the *Design Guide*, the equations have been omitted.

7.3. Turbine Noise

Turbine noise is estimated following the recommendations of Krejsa and Valerino[37]. The sound pressure level at a radius of 47.5 meters (150 ft.) is estimated using the relative tip speed of the last rotor V_{TR}, the sonic velocity at the exit c_L, the primary mass flow m. Estimates are provided for both broadband and tonal content. The noise spectrum peaks at the blade passage frequency f_b and at an angle of 110° from the inlet. The implementation of these equations in the Workbook assumes that the primary nozzle exit is not upstream of the secondary nozzle exit. Reductions of up to 10 dB(A) can be achieved if the primary nozzle exit is located upstream of the secondary nozzle as in a JT8D engine.

The tone SPL varies with $10 \log_{10} C/S$ where C and S are respectively the stator chord length and rotor/stator spacing at the final rotor. A 3 dB reduction is available by doubling the spacing (halving the ratio C/S).

Because the noise control of these devices falls outside the scope of the *Design Guide*, the equations have been omitted.

7.4. Jet Noise

Three principal noise source mechanisms exist: mixing, shock-associated noise, and screech. Noise esimates are based on the method of Stone and Montagni[40] as reported by SAE ARP 876C[39] and Beranek and Vér[38].

Mixing noise arises at the turbulent shear layer separating the fast moving jet core from the stationary surrounding atmosphere. Shock-associated noise arises in choked flows, and dominates above $M_j = 1$. A third source is jet "screech", produced by a feedback mechanism in which a disturbance convected in the shear layer generates sound as it traverses the standing system of shock waves. The sound propagates upstream through the ambient atmosphere and causes the release of a new flow disturbance at the nozzle exit. This is amplified as it convects downstream and the feedback loop is completed as it encounters the shocks.

7.5. Noise Estimation Using Workbook

Spreadsheets, Inputs Required and Notes:

- Inlet Fan and Compressor

Inputs Required: mass flow rate m', upstream pressure P_1, upstream temperature T_1, downstream pressure P_2, diameter of fan D_F, rotational rate N, number of blades B, number of vanes or stators V, inlet guide vane chord length C_1, inlet guide vane/fan rotor spacing S_1, fan rotor chord length C_2, rotor/stator spacing S_2, M_{TRD}, Fan Stage, distance r, angle θ

Notes:

- Levels of broadband and tonal noise are tabulated separately for radiation to the observation point from inlet and outlet. Levels of combination tones are tabulated for the inlet.

- Combustor and Core

Inputs Required: mass flow rate m', combustor inlet pressure P_3, combustor inlet temperature T_3, combustor outlet temperature T_4, reference turbine temperature differential for maximum takeoff conditions $(T_4\text{-}T_5)_{ref}$, distance r, angle θ

- Turbine Noise

Inputs Required: mass flow rate m', turbine exit temperature T_5, turbine diameter D_T, rotational rate N, number of blades B, rotor chord length C, rotor/stator spacing S, distance r, angle θ Notes: • Broadband and tonal noise are tabulated separately. • Turbine noise is assumed to radiate exclusively from the engine discharge.

- Jet Mixing

Inputs Required: • Upstream Gas Conditions: pressure P_1, temperature T_1 • Downstream Gas Conditions: pressure P_a, temperature T_a • Nozzle: nozzle coefficient C_N, nozzle diameter D_N, • Observer: distance r, angle θ Notes: • Gas is selectable so that this method may be used with all forms of gas discharge. • The "Execute" button must be pushed (clicked-on) in order to perform the double summation function for Shock-Associated Noise when M_j>1. Failing to do this will cause Shock-Associated Noise to be left out of the computations. • Jet Mixing Noise and Jet Shock Noise results are tabulated separately. • Use a nozzle coefficient of 0.85 if C_N is not known. • Do not expect the estimated noise levels to meet a hearing conservation criterion.

7.6. Noise Estimation for Jet Mixing Noise

It is customary to express the parameters of the gas flow as if it were an ideal, expanded jet with isentropic characteristics. The jet parameters for an ideal expanded jet can be calculated from the upstream and downstream pressures and temperatures:

$$M_j = \sqrt{\frac{2}{\gamma-1}\left(\left(\frac{P_1}{P_2}\right)^{\frac{\gamma}{\gamma-1}} - 1\right)}$$

$$T_j = \frac{T_1}{1+\frac{\gamma+1}{2}M_j^{\,2}}$$

$$c_j = \sqrt{\gamma\frac{R}{MW}T_j}$$

$$u_j = M_j c_j \quad ; \quad M_c = 0.62M_j$$

The overall sound pressure level of jet mixing noise measured at an angle of 90° from the jet axis can be estimated as[39,40,41].

$$L_{P,overall}(90^\circ) = 140 + 10\log\left(\frac{A_j}{r^2}\left(\frac{P_2}{P_{ISA}}\right)^2\left(\frac{\rho_j}{\rho_2}\right)^w\right) + 10\log\left(\frac{M_j^{\,7.5}}{1-0.1M_j^{\,2.5}+0.015M_j^{\,4.5}}\right)$$

where A_j is the fully expanded jet area ($\pi/4\ D_N^2$ for a subsonic jet), r is the distance to the observation point, P_{ISA} refers to standard atmospheric pressure, and

$$w = \frac{3M_j^{\,3.5}}{0.6+M_j^{\,3.5}} - 1$$

For other angles:

$$L_{P,overall}(\theta) = L_{P,overall}(90^\circ) - 30\log\left(1-\frac{M_c\cos\theta}{\left(1+M_c^{\,5}\right)^{\frac{1}{5}}}\right) - 1.67\log\left(1+\frac{1}{10^{40.56-\theta'}+4\times10^{-6}}\right)$$

$$\theta' = 0.26(180-\theta)M_j^{\,0.1}$$

where θ is expressed in degrees relative to the discharge axis.

The peak frequency of the resulting noise spectrum is computed as

$$f_p = \frac{S_j U_j}{D_N}$$

where D_N is the nozzle exit diameter and S_j varies with T_j/T_a and θ, and is interpolated from Table 11.

The spectral shape is approximated by

$$\Delta L_P = -\Delta - 8.4\left(\log\left(\frac{f}{f_p}\right)\right)^2$$

where Δ is interpolated from Table 12 below. The values ΔL_P are added to $L_{P,overall}$ to give octave sound pressure level values.

Table 11: Values of Strouhal Number as a Function of T_j/T_a and θ

T_j/T_a	$\theta = 50°$	$\theta = 60°$	$\theta = 70°$	$\theta = 80°$	$\theta \geq 90°$
1	0.7	0.8	0.8	1.0	0.9
2	0.5	0.4	0.6	0.5	0.6
3	0.3	0.4	0.4	0.4	0.5

Table 12: Values of Δ as a Function of T_j/T_a and θ

T_j/T_a	$\theta = 50°$	$\theta = 60°$	$\theta \geq 70°$
1	11 dB	11 dB	11 dB
2	10 dB	10 dB	11 dB
3	9 dB	10 dB	10 dB

The above equations apply to all cold jets and to hot jets when observed from an angle more than 50° from the jet discharge axis.

For hot jets ($T_j/T_a>1.1$), the peak frequency and spectral shape are considerably altered for $\theta \leq 50°$. This is due to refraction of sound at the shear layer. Three simplified corrections to the spectral peak frequency are estimated from tables given in

SAE ARP 876C[39]. A change in the spectral shape also occurs, but is considered less important for noise control purposes and is omitted here.

For hot jets the shift in peak frequency from the value calculated above is expressed as a number of ISO-preferred 1/3-octave bands ΔBN_i. (For cold jets no adjustments are necessary). The first ΔBN_1 depends on the angle of observation, the second ΔBN_2 on the ratio of jet temperature to ambient temperature and the third ΔBN_3 is an additional correction depending on angle that is used when $M_j > 1.33$.

$$\Delta BN_1 = -1.1 + 0.262\theta(\deg) - 1.543\times10^{-4}\,\theta(\deg)^2$$

$$\Delta BN_2 = 1 - \frac{T_j}{T_s}$$

$$\text{and if } M_j > 1.33,$$

$$\begin{aligned}\Delta BN_3 &= \frac{50-\theta}{10} \quad \text{for} \quad 20^\circ < \theta \le 50^\circ \\ &= 3.0 \quad \text{for} \quad \theta \le 20^\circ \\ &= 0.0 \quad \text{for} \quad \theta > 50^\circ\end{aligned}$$

where θ is relative to the jet discharge axis. The shifted peak frequency f_p' for hot jets is

$$f_p' = f_p 10^{0.1\sum_i \Delta BN_i}$$

7.7. Predictive Equations for Shock-Associated Noise

Predictive equations for shock-associated noise follow the method reported in SAE ARP 876C[39] and Beranek and Ver[41]. Shock associated noise dominates for $M_j \ge 1$ in the absence of a converging-diverging nozzle, but is not generated at exit velocities $M_j < 1$. Shock-associated noise is essentially omni-directional and may be estimated for all angles of observation as follows:

$$L_{P\,shock} = C_0 + 10\log\frac{\beta^n A_j}{r^2}$$

where

$$\beta = \sqrt{M_j^{\,2} - 1}$$

$$C_0 = 156.5 \quad for \ \frac{T_j}{T_s} < 1.1$$

$$and \quad n = 4 \quad for \ \beta < 1$$

$$n = 1 \quad for \ \beta \geq 1$$

$$C_0 = 158.5 \quad for \ \frac{T_j}{T_s} \geq 1.1$$

$$and \quad n = 4 \quad for \ \beta < 1$$

$$n = 2 \quad for \ \beta \geq 1$$

The shock-associated noise spectrum can be expected to exhibit a well-defined peak in the vicinity of

$$f_p = \frac{0.9 M_c c_j}{D_N \beta (1 - M_c \cos\theta)}$$

For a hot jet ($T_j/T_s > 1.1$), the one-third octave band SPL (re 20μPa) is given by:

$$L_P = L_{P\,shock} + \Delta_{shock}$$

where

$$\Delta_{shock} = -15 - 16.1 \log\left(\frac{5.163}{\sigma^{2.55}} + 0.096\sigma^{0.74} \right) + \cdots$$

$$+ 10 \log\left(1 + \frac{17.27}{N_s} \sum_{i=0}^{N_s - 1} C(\sigma)^{i^2} \sum_{j=1}^{N_s - i - 1} \frac{\cos(\sigma q_{ij}) \sin(0.1158 \sigma q_{ij})}{\sigma q_{ij}} \right)$$

and

$$C(\sigma) = 0.8 - 0.2 \log\left(\frac{2.239}{\sigma^{0.2146}} + 0.0987\sigma^{2.75} \right)$$

$$\sigma = 6.91 \beta D_N f / c_2$$

$$N_s = 8 \quad (number \ of \ shocks)$$

$$q_{ij} = (1.7 i c_2 / U_j)\{1 + 0.06[j + \tfrac{1}{2}(i+1)]\}\left[1 - 0.7\left(\frac{U_j}{c_2} \right) \cos\theta \right]$$

Screech

No predictive equations are provided for level of jet screech because it is easily controlled in practice[41]. Screech tones radiate equally in all directions, and the fundamental tone is centered around

$$f_{screech} = \frac{U_c}{L(1 + M_c)}$$

where U_c is the convection velocity of the disturbance in the shear layer, L is the axial length of the first shock cell, and $M_c = U_c / c_j$.

Screech can be virtually eliminated by minor modifications to nozzle design, for example, by the addition of tabs in the exhaust flow, by notching the nozzle perimeter, or by using a non-axisymmetric discharge nozzle.

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