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Design Guidelines for Quiet Fans and Pumps for Space Vehicles

John S. Lovell and Bernard Magliozzi
Hamilton Standard, Windsor Locks, Connecticut

Prepared under Johnson Space Center Contract NAS9–12457

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Design Guidelines for Quiet Fans and Pumps for Space Vehicles

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INTRODUCTION

This document presents guidelines for the design of quiet fans and pumps of the class used on space vehicles. A simple procedure is presented for the prediction of fan noise over the meaningful frequency spectrum. A section also presents general design criteria for axial flow fans, squirrel cage fans, centrifugal fans, and centrifugal pumps.

The basis for this report is an experimental program conducted by Hamilton Standard under NASA Contract NAS 9-12457. The derivations of the noise predicting methods used in this document are explained in Hamilton Standard Report SVHSER 6183, "Fan and Pump Noise Control", dated May 1973 (6).

DEFINITION OF TERMS

FAN AND PUMP NOMENCLATURE

A schematic representation of an axial type fan and a centrifugal type fan or pump, showing nomenclature for typical parts, is shown in figure 1.

NOISE MEASUREMENT UNITS

Sound Pressure Level (SPL) is a measure of acoustical noise at a point in space. The reference pressure (p) for sound pressure level is 0.0002 microbar (dyne per sq. cm.), so that sound pressure level is defined as

$$SPL = 20 \log_{10} \frac{p}{0.0002}$$

where p is the rms sound pressure measured in microbars.

In order to account for the ear's variation in sensitivity with frequency an additional concept is used. This is sound pressure level in NC, or Noise Criteria, units. In order to establish the NC level of a sound its octave band sound pressure levels are determined and compared with those in figure 2. The dBNC value corresponds to the maximum level at any frequency of the noise under consideration.

Sound Power Level (PWL) is the measure of the acoustical energy output of an item. The reference power (W) for sound power level measurement is 10^{-13} watts so that

$$PWL = 10 \log_{10} \frac{W}{10^{-13} \text{ watts}}$$

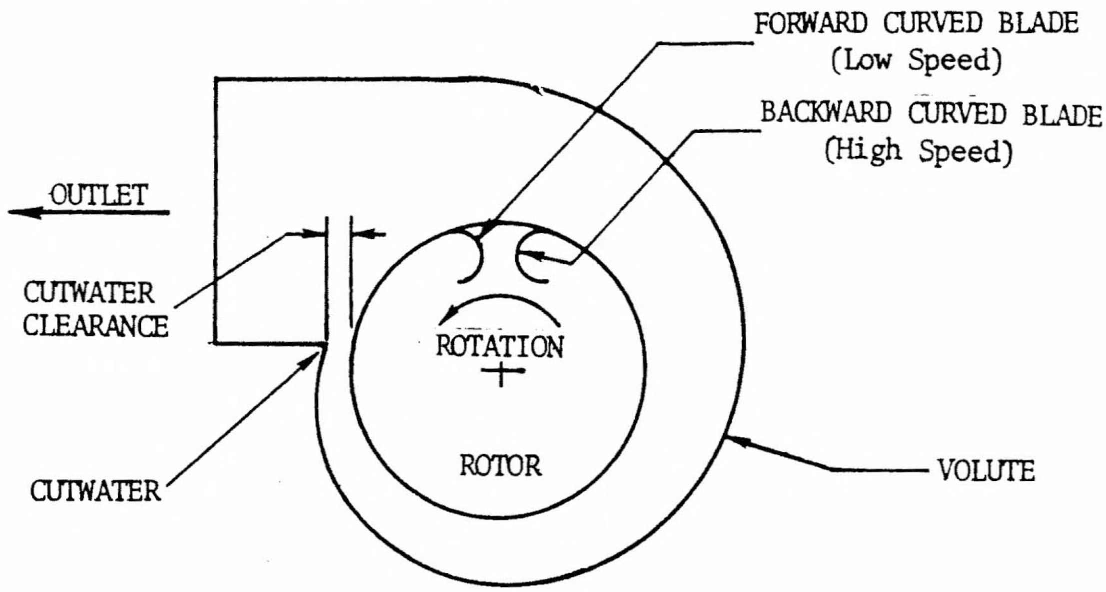
where W is the sound power level measured in watts, and PWL is expressed in decibels.

Noise prediction methods predict sound power level which is then converted to sound pressure level by the equation

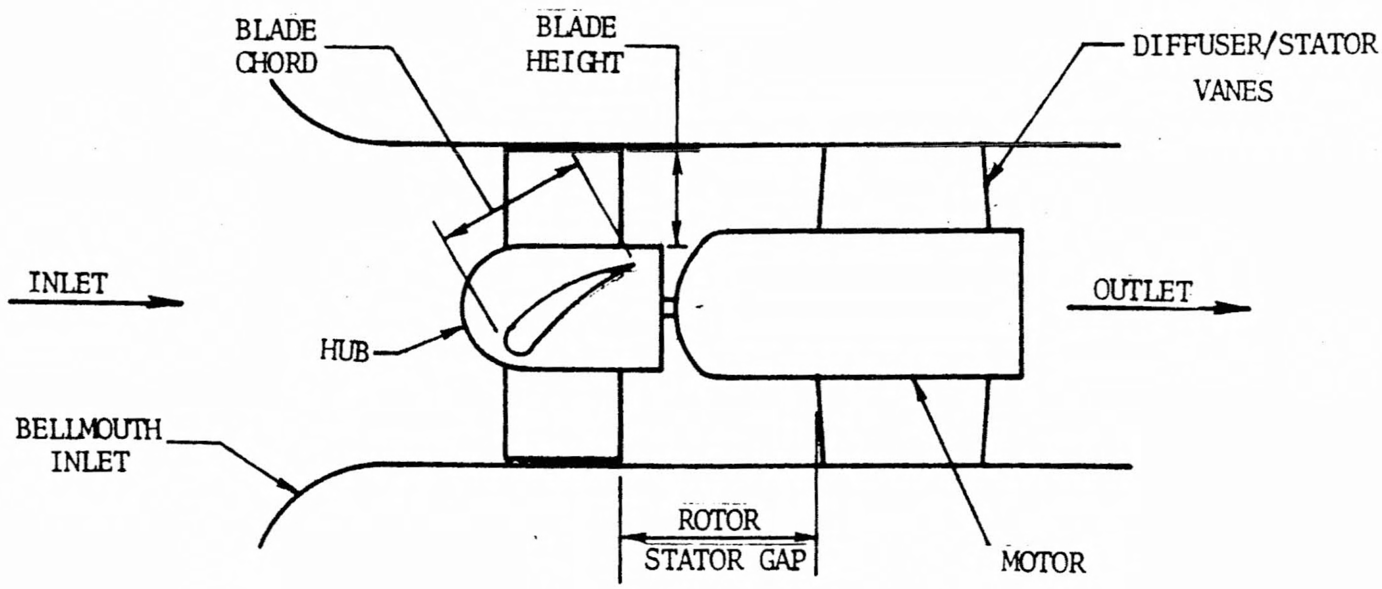
$$SPL = PWL - 10 \log_{10} A + 10 \log_{10} \rho C_o - K$$

where
- A = area of the sphere at the distance at which the sound is to be expressed
- ρ = atmospheric density
- Co = speed of sound
- K = constant to account for units.

CENTRIFUGAL TYPE FAN OR PUMP

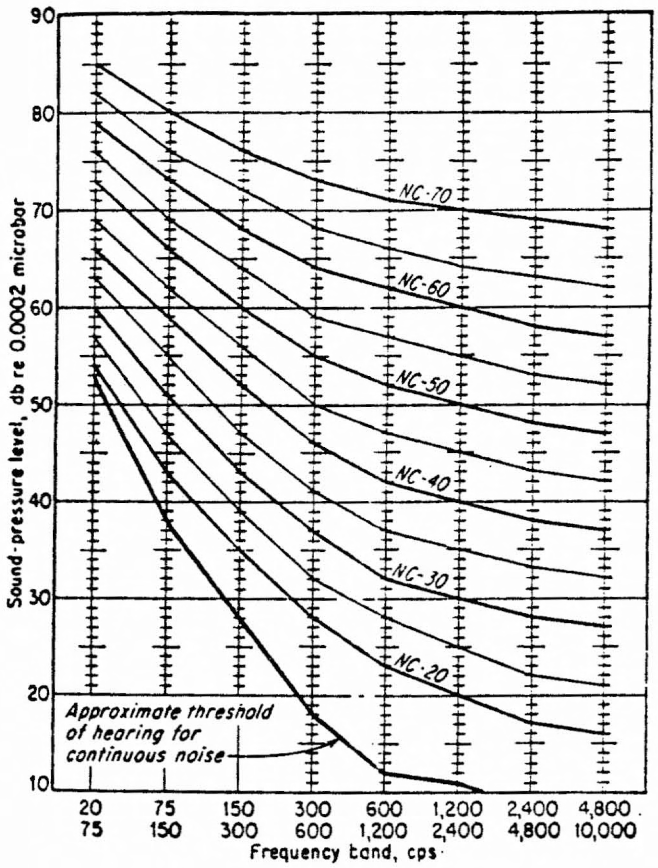


AXIAL TYPE FAN



FAN AND PUMP NOMENCLATURE

FIGURE 1



NOISE CRITERIA CURVES

FIGURE 2

For standard atmospheric conditions and spherical radiation of the sound this converts to

$$SPL = PWL - 20 \log_{10} R - 10.5$$

where R is the distance from the source in feet.

AERODYNAMIC FAN NOISE

In order to understand the noise from fans, a brief review of the aerodynamic noise sources in fans is required. Both axial and centrifugal flow fans produce two types of aerodynamic noise: first, broad band turbulence noise and second, discrete tones related to the frequency at which interactions occur between fan blades and the fluid.

The broad band noise arises from the shedding of vortices at the rotor blade trailing edges when the blade is operating in smooth airflow, which induces local surface pressure fluctuations on the blades. When the blade is operating in turbulent flow, there is an additional noise mechanism due to the randomly fluctuating lift.

The rotational, or tone, noise is caused by the rotating steady blade surface pressure field and by aerodynamic interaction between the rotor and stator blades.

The procedure described below allows the estimation of fan noise from the general operating parameters defining the power input to the fan, or equivalently, the work done on the gas by the fan. As such, the diameter, tip speed, number of blades, and so forth are not explicitly defined and must be determined from aerodynamic considerations.

The octave band sound pressure levels for several fan designs, including axial flow, squirrel cage, and centrifugal types, may be estimated using this procedure.

Step 1

A reference octave band spectrum is obtained from Table I based on the fan type.

Step 2

A frequency scaling parameter is determined from Figure 3. This graph is entered with rpm/1000 and the frequency shift read. The entire spectrum from step 1 is shifted up or down in frequency as indicated. For example, for a one octave band shift (frequency scaling parameter between 1.41 and 2.83), the 63 Hz octave band of the reference spectrum, which was selected in step 1, would become the 125 Hz band, and the 125 Hz band would become the 250 Hz band, etc.

Step 3

If the frequency shift from step 2 is up in frequency, there will now be information missing in the lower frequency octave bands. The missing octave bands are filled in using a slope of minus 6 dB per octave. Therefore, if the spectrum were shifted two octave bands, the value of the 125 Hz band would take on the value of the new 250 Hz octave band minus 6 dB, and the 63 Hz band would become the value of the new 250 Hz band minus 12 dB.

Step 4

An adjustment, to account for the operating condition of the fan, is calculated as

$$\Delta dB = 20 \log_{10} \Delta P + 10 \log_{10} Q_d \ ,$$

where ΔP is the fan static pressure rise in inches of water and Q_d is the fan discharge flow in cfm. This adjustment is added to each octave band level determined in steps 2 and 3.

Step 5

The blade passing frequency, given by

$$BPF = rpm \times \text{number of blades}/60$$

is computed. The blade frequency increment (BFI) from Table I is added to the level of the octave band spanning the blade passing frequency. The upper and lower frequencies defining an octave band are given in Table II.

Note: The BFI for axial flow fans should be added only if the fan is under non-uniform inflow or has propagating rotor/stator interaction tones. The fan will not have propagating rotor/stator interaction tones if the wall velocity, V_w, of the interaction tone is subsonic. V_w is given by

$$V_w = \frac{BV_t}{B-V}$$

where B is the number of rotor blades, V is the number of stator vanes, and V_t is the rotor tip velocity.

Step 6

Steps 1 to 5 give the octave band PWL for the total fan noise. Inlet and exhaust noise are assumed to be equal and can be estimated by subtracting 3 dB from the total PWL. To calculate SPL, use

$$SPL = PWL - 10 \log_{10} A + 0.5$$

where A is the area, in square feet, over which the SPL is assumed constant. If it is assumed that the sound radiates uniformly in all directions (spherical spreading) then the conversion becomes

$$SPL = PWL - 20 \log_{10} R - 10.5$$

where R is the distance, in feet, from the fan center to the point where the noise estimate is desired.

The octave band SPL's then may be summed to give overall noise, converted to dB (A) values, used to estimate dB NC, and so forth. Note that the above procedure assumes free field radiation. Any additional attenuation due to ducting, plenums, etc. must be accounted for to bbtain the noise estimate of the installed unit. Attenuation curves for ducts, elbows, plenums, etc. may be found in standard acoustics or noise control texts.

TABLE I

SPECIFIC SOUND POWER LEVELS (re 10^{-13} watts) AND BLADE FREQUENCY INCREMENTS FOR FANS OF VARIOUS TYPES

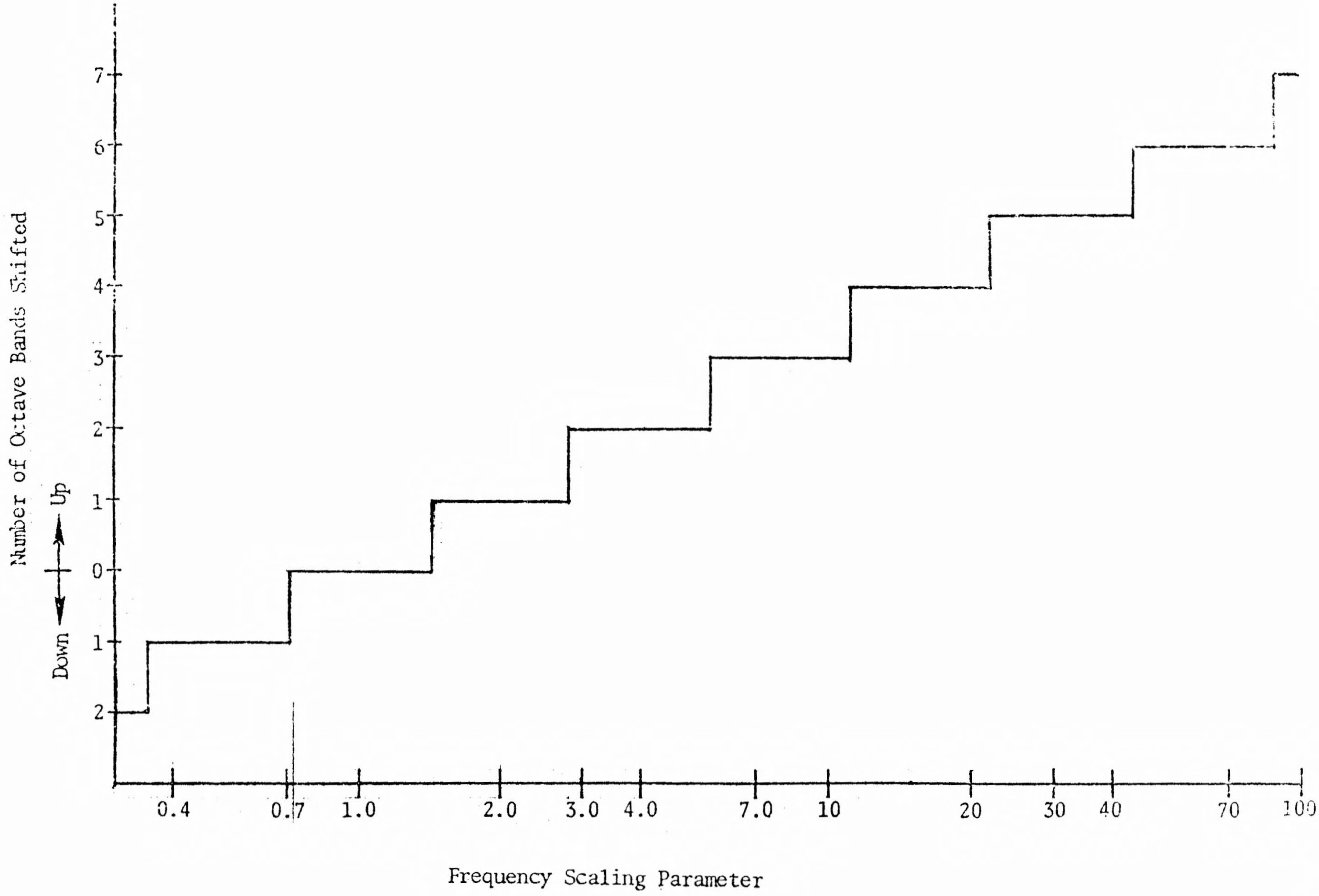
FAN TYPE	OCTAVE BAND CENTER FREQUENCY, Hz								BFI
	63	125	250	500	1000	2000	4000	8000	
Squirrel Cage Centrifugal, forward curved blade	① 50	48	48	44	38	34	31	25	2
	② 49	51	52	49	47	43	40	35	2
Centrifugal, radial blade	58	55	53	53	48	43	40	39	6
Vaneaxial	52	49	51	52	50	47	45	35	7

① Based on large units with greater than 70 percent efficiency.
② Revised to account for lower-expected efficiencies from 5 to 8 inch rotor diameter units.

TABLE II

FREQUENCY LIMITS FOR FULL OCTAVE BANDS

Octave Band Center Frequency, Hz	Lower Limit, Hz	Upper Limit, Hz
63	--	90
125	90	180
250	180	355
500	355	710
1000	710	1400
2000	1400	2800
4000	2800	5600
8000	5600	--



FREQUENCY SHIFT ADJUSTMENT GRAPH

FIGURE 3

SAMPLE CALCULATION - AXIAL FAN NOISE ESTIMATE

This section presents a detailed sample calculation for estimating the noise for an 11,000 rpm vaneaxial fan. The sample calculation follows the first five steps described in the previous section of this report. The results from each step are summarized in Table III.

TABLE III

SAMPLE NOISE ESTIMATE FOR AN AXIAL FAN

Line No.	Calculated in Step No.	Octave Band Center Frequency, Hz								
		63	125	250	500	1000	2000	4000	8000	16000
1	1	52	49	51	52	50	47	45	35	--
2	2	--	--	--	52	49	51	52	50	47
3	3	34	40	46	52	49	51	52	50	47
4	4	68	74	80	86	83	85	86	84	81
5	5	68	74	80	93	83	85	86	84	81

Step 1

Line 1 of Table III is obtained from Table I, line 4, for a vaneaxial fan.

Step 2

The fan speed is 11,000 rpm; therefore, the frequency scaling parameter 11,000/1000 = 11.

From figure 3, it is found that this spectrum should be shifted up in frequency by 3 octave bands. The adjusted spectrum is shown in line 2 of Table III.

Step 3

The values for the 63 to 250 Hz bands are added using a 6 dB/octave roll-off, as shown by line 3 of Table III.

Step 4

The fan pressure rise is 2.5 inches of water at a discharge flow of 400 cfm, thus,

$$\Delta \text{ dB} = 20 \log_{10} 2.5 + 10 \log_{10} 400$$
$$\Delta \text{ dB} = 34$$

This adjustment is shown in line 4 of Table III.

Step 5

The blade passing frequency, BPF, is defined by

BPF = rpm x no. of blades/60
BPF = 11,000 x 3/60
BPF = 550 Hz

From Table II, 550 Hz is in the 500 Hz octave band and from Table I, the blade frequency increment, BFI, is 7. Thus, 7 dB are added to the 500 Hz band, as shown in line 5 of Table III.

DESIGN CRITERIA FOR FANS AND PUMPS

GENERAL

Factors which improve the aerodynamic efficiency of a fan tend to reduce the aerodynamic noise generated by the fan. This occurs because turbulence is associated with aerodynamic losses and aerodynamic noise generation mechanisms are associated with turbulence. Consequently, measures such as designing for smooth inlet flow and using efficient airflow blades with trailing edges designed for minimum flow separation will reduce noise generation.

Depending on which noise mechanisms are postulated, the acoustic power of fans varies with the fourth to sixth power of the relative air velocity, which is approximately equal to the blade tip speed. The following relationship is given by Lowson (1).

$$\text{Acoustic power} \propto D^2 V_t^5 \quad ,$$

where D is the rotor tip diameter and V_t is the rotor tip speed. This relationship has general validity in geometrically similar fans. Note that

$$D^2 V_t^5 = D^2 V_t \; (V_t^2)^2 \propto Q_d \bullet (\Delta P)^2$$

which, when converted to decibels results in

$$10 \log (D^2 V_t^5) = 10 \log (Q_d \bullet (\Delta P)^2) + K = 10 \log Q_d + 20 \log \Delta P + K$$

where Q_d is the fan discharge flow and ΔP is the fan pressure rise.

This is the basis for most fan noise estimating procedures, including the one presented herein. The constant K is based on the type of fan utilized.

Minimizing the quantity $D^2V_t^5$ is an effective means of reducing noise generations in a fan. However, in selecting a maximum efficiency fan with a given operating point, the range of allowable tip speed variation may be very limited. Figure 4 shows the PWL of a number of fans plotted against the parameter $D^2V_t^5$. Although these fans cover a range of performance and geometries, the correlation is quite good. Note that on this curve the squirrel cage fan's overall noise level is 16 dB above the theoretical line proportional to $D^2V_t^5$. Although figure 4 indicates the overall acoustic power of fans, the trends in other noise measuring schemes such as dB(A), dbNC and so forth, would be similar.

In most of the common, high performance axial flow fans the overall noise levels, and in most cases the dB(A) and dBNC noise levels, are generally determined primarily by the levels of the fundamental tone and its harmonics. The broad band, or vortex shedding, noise can be significant in quiet fan designs and can be estimated by (2):

$$PWL = 10 \log KA_BV_{0.7}^6$$

where A_B is the total rotor blade area and $V_{0.7}$ is the rotor tip speed at the 70 percent radius and K is a coefficient of proportionality. Figure 5 presents a correlation of the measured vortex noise - approximated by summing the levels of the bands defining the broad band peak - of the axial fans and of a commercial squirrel cage fan, with the parameter of $A_BV_{0.7}^6$. The correlation is seen to be quite good for the axial fans, whereas the squirrel cage fan falls well above the generalized curve.

Figure 6 shows the recommended speeds and the type of fan to be used for a given flow and pressure rise. The selected rpm's are based on the use of 400 Hz motors. This figure is based on the optimum specific speed range for an axial fan being 60-150 thousand rpm and that for a centrifugal being in the range from 10 to 60 thousand rpm. Where either design can be used in general, the centrifugal will be heavier while the axial will be somewhat noisier.

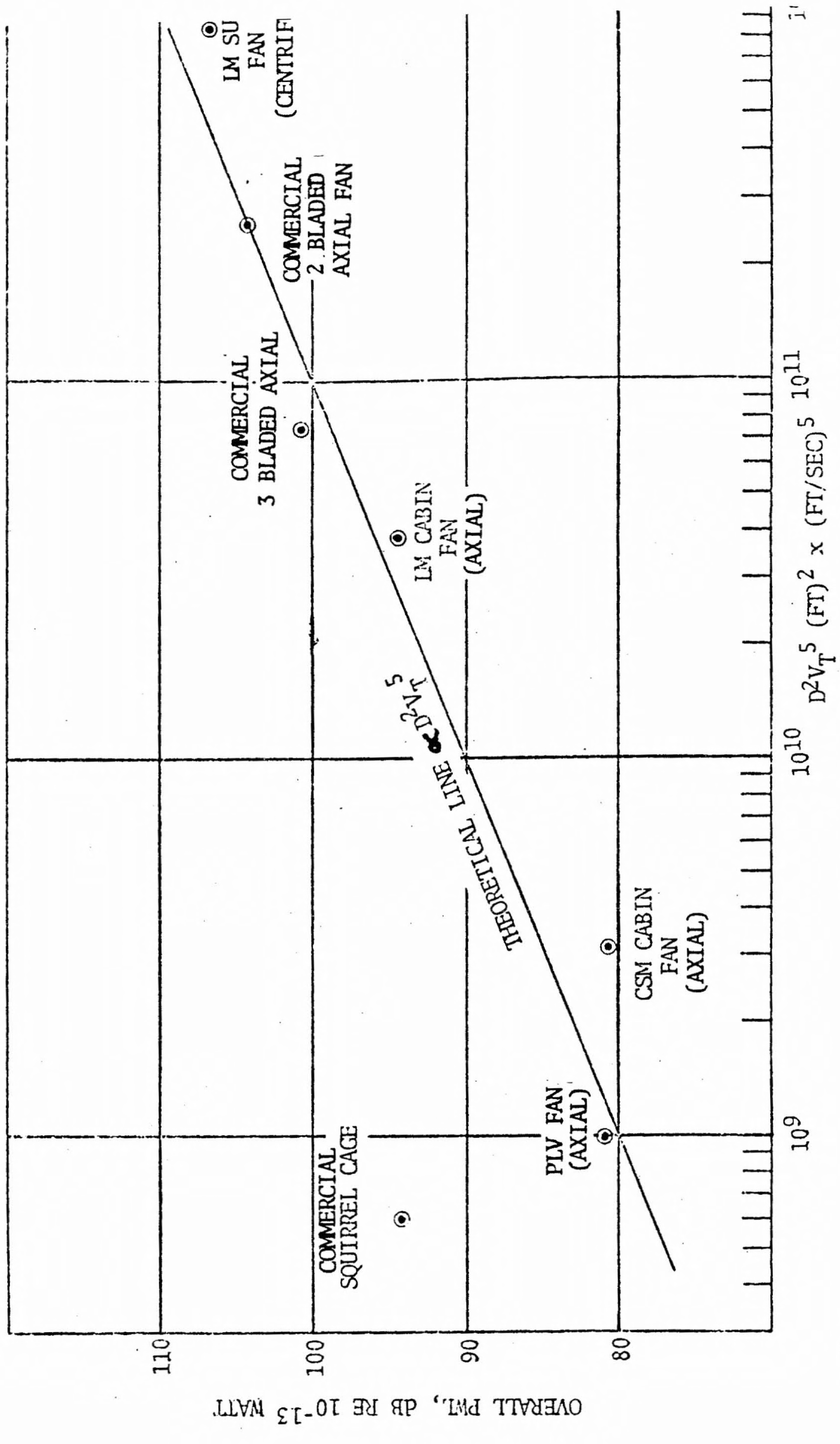
AXIAL FLOW FANS

The following general guidelines should be observed in the design of quiet axial flow fans.

1. The tip speed should be maintained as low as possible without seriously compromising aerodynamic performance.

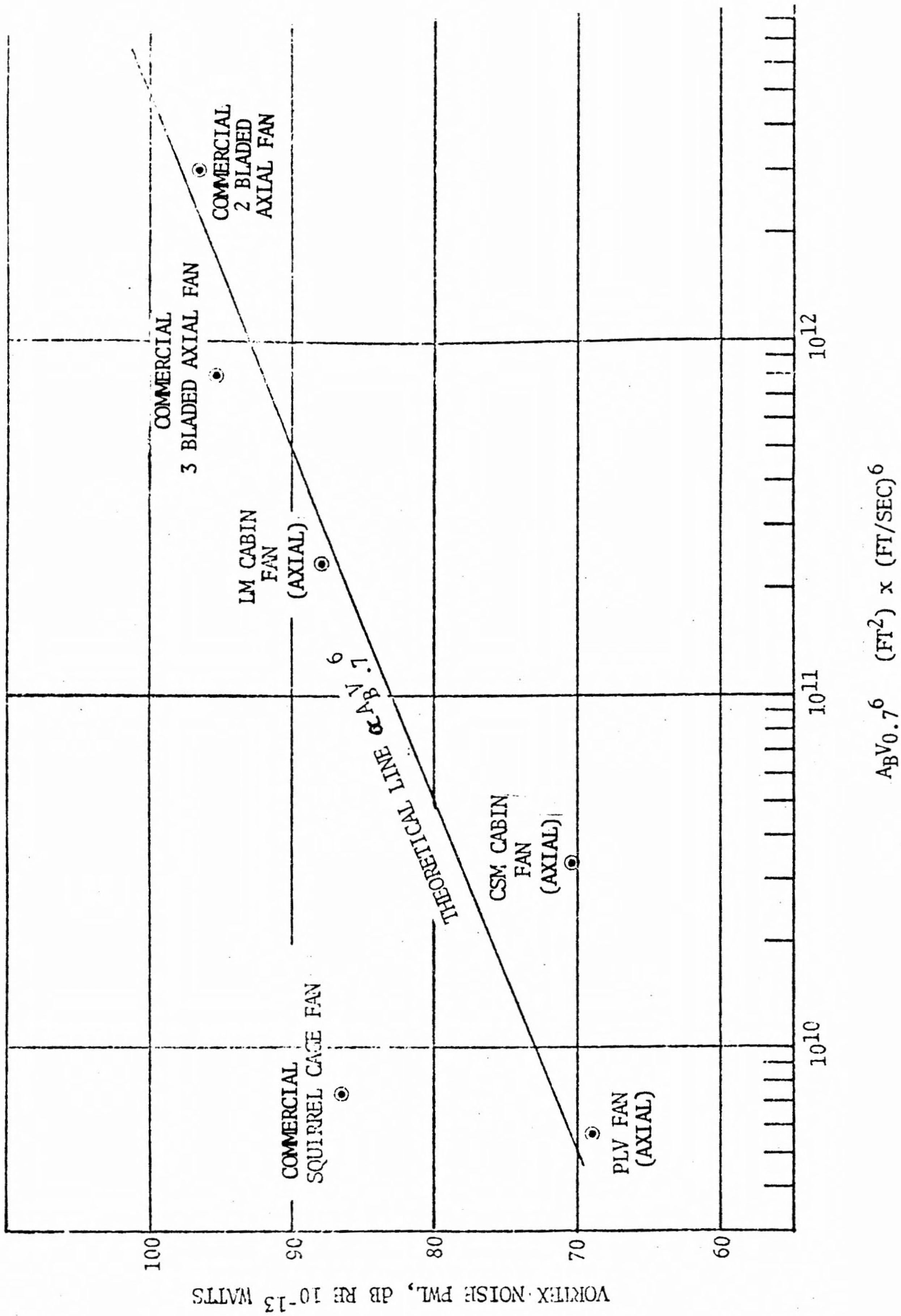
2. The blade-tip-to-housing clearance should be kept small to minimize the tip vortex. Clearances of 0.010 inch are judged to be acceptable for these small ventilation fans.

3. The rotor blade wakes cause fluctuations in the lift of the vanes, which is radiated as noise. Thus, the gap between the rotor trailing edge and the stator leading edge should be large enough to allow the rotor wakes to diffuse. Figure 7 shows typical calculated noise variations with the rotor stator (blade-vane) gap. This figure shows that increasing the gap beyond two mean blade chords does not result in significant additional noise reduction.



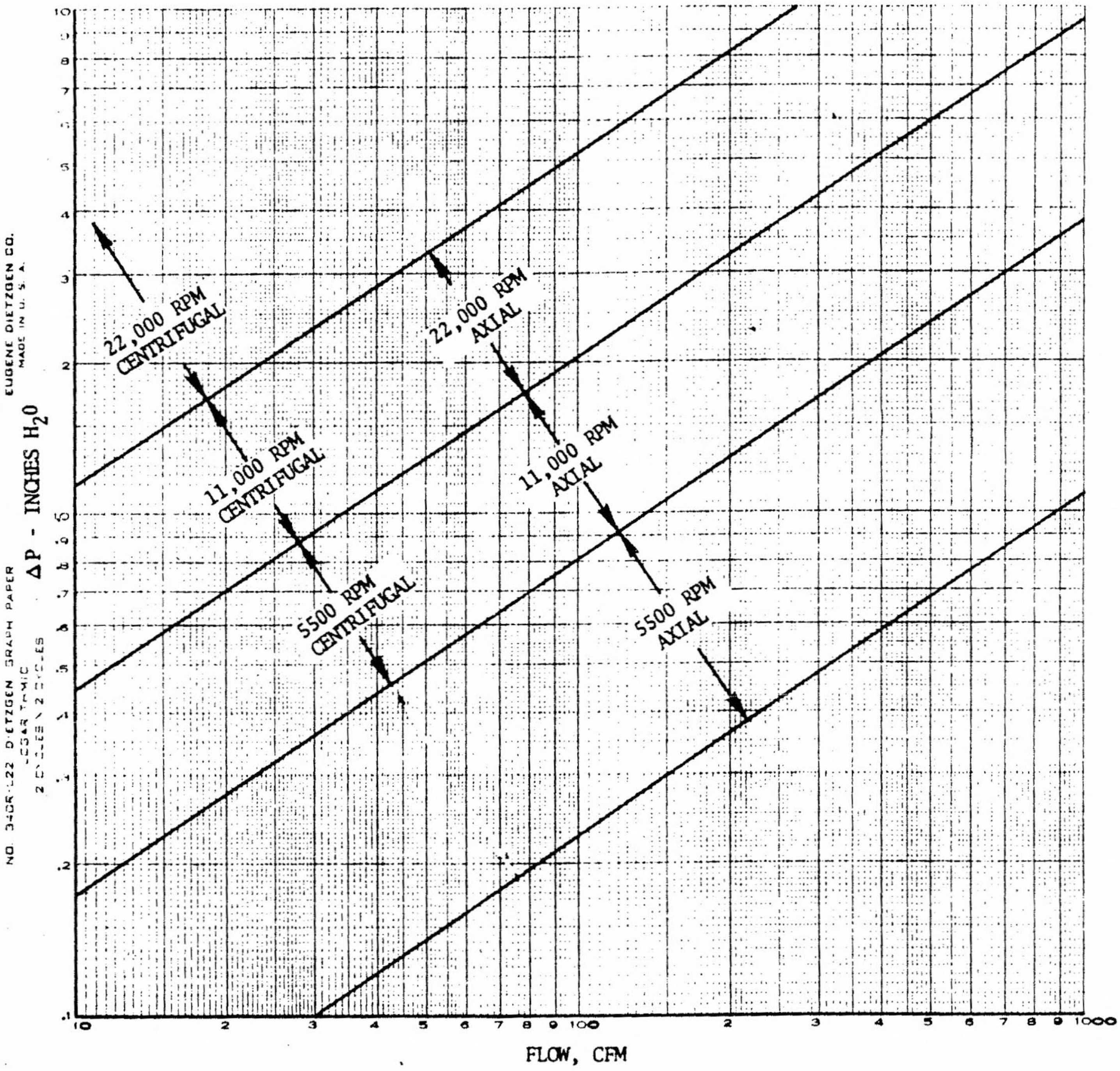
VARIATION OF OVERALL POWER LEVEL WITH $D^2V_T^5$

FIGURE 4



VARIATION OF VORTEX NOISE PWL WITH $A_B V_{0.7}^6$

FIGURE 5



RECOMMENDED OPERATING RANGES FOR SPACE TYPE FANS

FIGURE 6

4. The noise varies with the number of rotor blades. Figure 8 shows the calculated noise variation with rotor blade count. The minimum noise is seen to occur at three blades. However, the penalty for increasing the blade count, as might be required to minimize aerodynamic slip, is not great.

5. The number of stator vanes to be used is not critical provided the rotor-to-stator gap is beyond 1.5 mean blade chords, so that no significant rotor-stator interaction noise is generated.

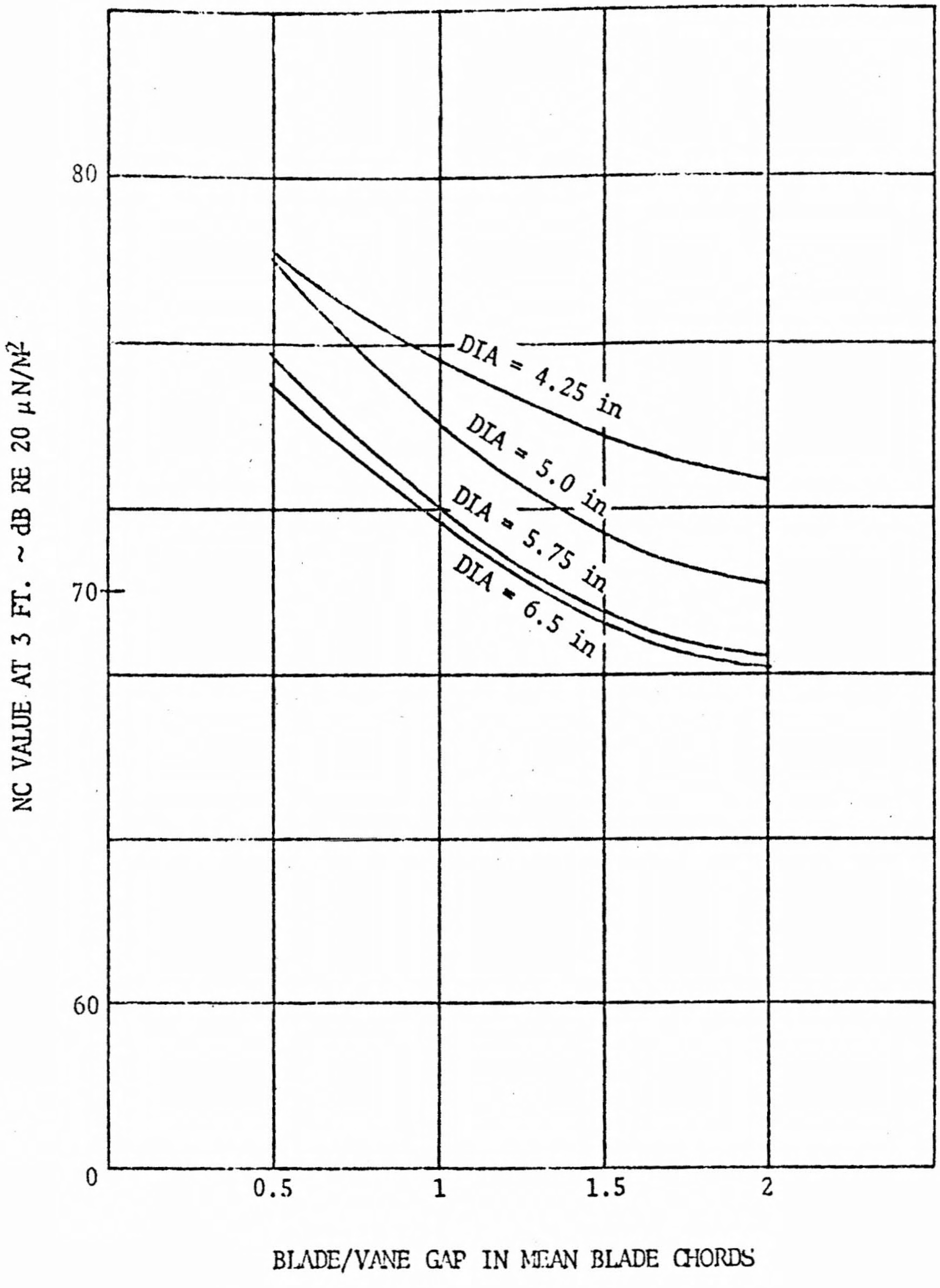
6. It is apparent that upstream disturbance to the flow causes high fan noise output. Since a very small disturbance can significantly increase noise, only a small obstruction in the flow is necessary. Thus such items as aerodynamic probes, turning vanes, preswirl vanes and elbows should not be placed upstream of the fan. Locating a fan immediately behind an obstruction which causes partial blockage of the inlet should be avoided, since this also will give rise to flow distortion. Also, the use of large radii bellmouths on the inlet is recommended to insure smooth, uniform inflow.

7. To prevent flow disturbances, the generation of edge tones, and turbulence generated noise, the flow passages should be smooth and free from obstructions such as bolt heads and holes.

8. The rotor and motor assembly should be balanced to minimize mechanical noise.

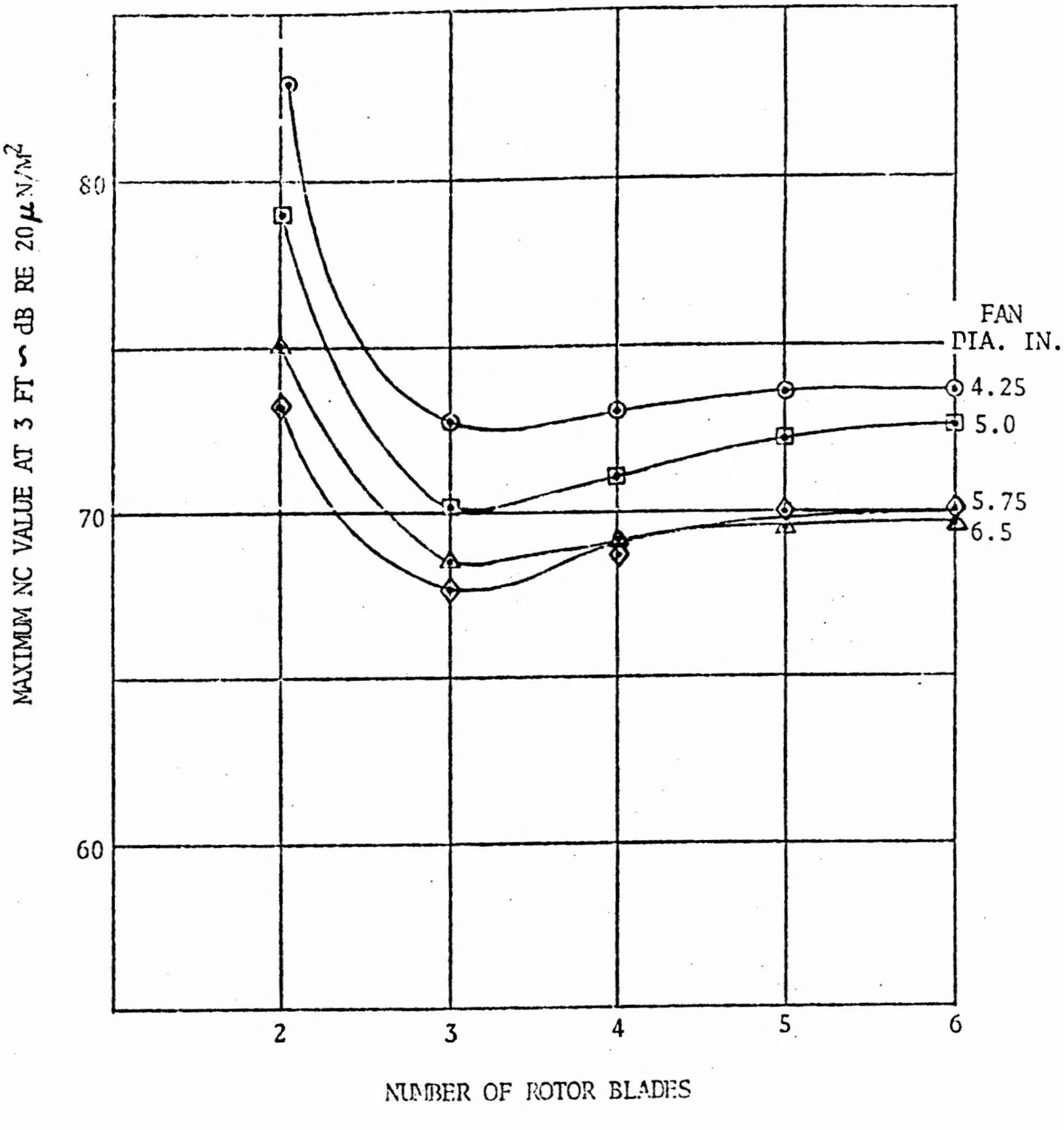
Other considerations, more difficult to implement, which may lead to significantly reduced axial flow fan noise are as follows.

1. One concept which improves the fan inlet flow profile is that of flow straightening devices located upstream of the rotor. Even though this approach appears contrary to the above discussion, it is based on the assumption that if the size of the wakes is small compared to the span and chord of the rotor blades, then the wakes do not act coherently to cause the fluctuating lift which gives rise to the high rotational noise. Thus the effective flow disturbances are reduced and the noise due to non-uniform inflow is correspondingly reduced. This method of noise control can be effected by the use of thin-wall, small cell size honeycomb, several small mesh settling screens, or other similar approaches commonly used in low turbulence wind tunnels.



FAN NOISE DEPENDENCE ON BLADE-VANE GAP

FIGURE 7



FAN NOISE VARIATION WITH ROTOR BLADE COUNT

FIGURE 8

2. Porous materials have been applied to small airfoils placed in the stream of a small jet, with some success by Lowson (3), and more recently on small propeller fans by Chanaud (4), and by Tseo (5). In Chanaud's experiment, a small fan was constructed utilizing both partially and fully porous materials for the blades. The porous material was found to be very effective in reducing the noise of the fan at little or no loss in fan efficiency. In fact, certain materials seemed to give better efficiencies with 10 to 12 dB noise reduction. Although Chanaud shows a potential for 19 dB noise reduction at very low pressure rise, it is probable that this noise reduction will be offset by loss in performance due to back leakage through the material as the static pressure head of the fan is increased.

   Tseo achieved some noise reduction by covering the pressure surface of the fan blades with the porous material. Although his data is somewhat limited, reductions in the mid-frequency broad band and tone noise are apparent. Tseo attributes this to the material acting as a high hydrodynamic resistance to attenuate the fluctuation of blade pressure and a relief of the pressure build up around the blades to reduce the vortex strength. Attenuations of approximately 10 dB in the mid-frequency broad band were achieved using 1/16 inch fiberglass.

3. Boundary Layer Control, BLC, does not appear to be applicable to small fans (less than 500 cfm) because the blades are too small to easily implement suction across the rotating rotor.

SQUIRREL CAGE FANS AND OTHER CENTRIFUGAL FANS

Although less is understood about the noise generating mechanisms of centrifugal fans than about axial flow fans, the following general guidelines should prove useful in minimizing their noise.

1. Forward curved blading should be used to minimize the tip speed required to develop the desired head.

2. The rotor to cutwater clearance should be made as large as possible, consistent with the required aerodynamic efficiency. As a rule of thumb, increasing the cutwater clearance from 5 percent, to 10 percent of rotor diameter can result in a noise retention of 10 to 15 dB with only a five percent loss in efficiency.

3. Decreasing the blade loading is generally conducive to low noise. Thus, as large a number of blades as can be used will result in lower noise.

4. Since these fans have an axial inlet and a radial outlet, the flow is turned through 90 degrees. This turning should be done as gradually as possible to avoid flow separation. Thus generous radii on blade inlets and outlets should be used.

5. Turning vanes should not be used, since the blade wakes can cause strong pressure fluctuations on the vanes with significant increase in noise.

6. The rotor and motor assembly should be balanced.

CENTRIFUGAL PUMPS

There is very limited noise data on small pumps. However, from the limited data available and from general trends based on data from full size pumps, the centrifugal pump is considered to be quieter than other types. In this size pump, the motor noise is likely to dominate. However, in cases where the pump noise is significant, even if only in part of the noise spectrum, design for low noise is necessary.

The general guidelines for achieving a low noise design include:

1. Design for high efficiency.

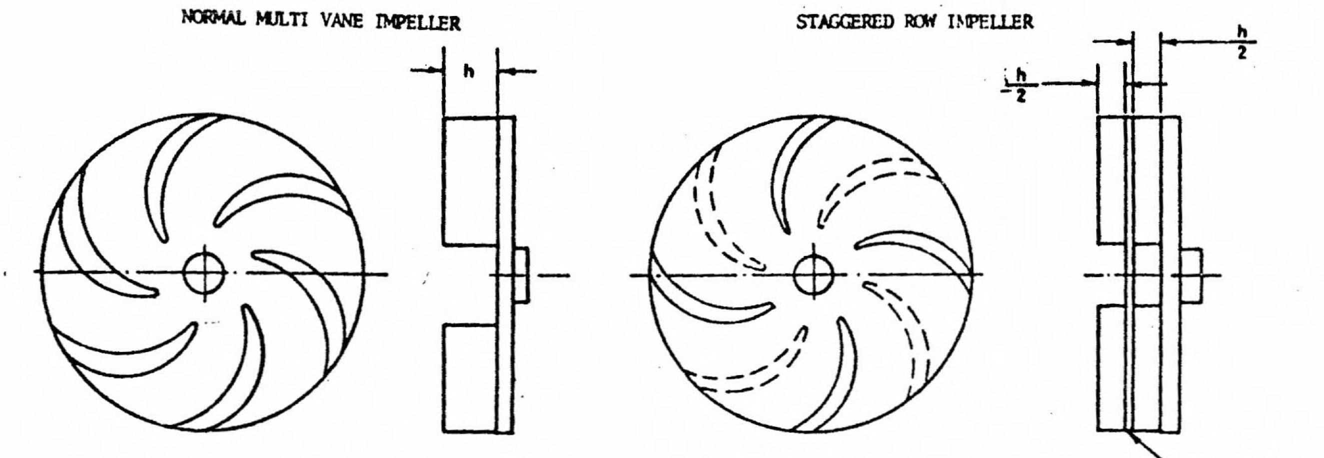
2. Operate the pump on or as near as possible its design point. This will help assure low flow noise and non-cavitating operation.

3. Design the pump to be hydraulically stable. Otherwise, fluctuating head and flow conditions can develop, leading to noisy operation. A continuously-rising head curve is considered to provide stable operation under "all" operating conditions.

4. Choose the correct blade inlet angle to provide a quiet and efficient, essentially shockless entrance into the pump impeller vane passage. This will be the angle whose tangent is the ratio of the radial fluid velocity to the longitudinal velocity of the inlet tip of the impeller vanes. It is generally an accepted practice to exaggerate the inlet angle as much as 15 to 20 percent, thus shifting the shockless capacity to the right of the design point on the pump head versus capacity curve. For quieting purposes it seems desirable to operate as near the shockless capacity point as possible without seriously affecting efficiency.

5. Choose the optimum impeller discharge angle. It is generally desirable to have an impeller discharge angle such that the straight-line theoretical Euler's head curve - whose slope depends on the discharge angle - approximates the pump curve. This is usually an angle of less than 90°.

6. Increase the number of vanes over those used in a standard design. Most effective is an impeller of the so-called multi-vane type. This is in essence a many-vaned stacked and staggered row design, manufactured to close hydraulic and mechanical tolerances to assure good hydraulic and mechanical balance and to achieve a low pressure-pulsation level. The very small size of the candidate pumps may well preclude full implementation - particularly multiple rows - due to manufacturing limitations. However, the number of vanes should be kept as high as practical thus increasing the frequency of the vane pulsations but reducing the energy per pulse which will have the effect of smoothing the flow. For example, an increase from seven to fourteen vanes could reduce the fluidborne noise level by 27 dB while increasing the frequency by a factor of two. If two rows of seven vanes were used with the rows staggered, an additional 8 dB decrease is estimated. An example of staggered rows is shown in figure 9.



PUMP IMPELLER TYPES

FIGURE 9

7. Increase the gap between the impeller and casing at the cutwater. Vane pulsation associated noise is generally decreased as the cutwater clearance - the gap between the impeller outside diameter and the casing inside diameter at the cutwater - is increased. Increasing the cutwater clearance will degrade the efficiency but at a relatively slow rate compared to the noise decrease. For example, a 5 percent efficiency change could yield a 10 to 15 dB noise reduction with a rotor to cutwater clearance increase from 5 percent to 15 percent of rotor diameter.

8. Since motor noise is likely to be an important noise source, use quiet motor designs.

9. The pump and motor assembly should be balanced.

OTHER CONSIDERATIONS

In all of these devices, it is important to consider the total system when low noise levels are desired. For example, it is advisable to use vibration isolation mounts, so that the vibration from the rotating machinery is not transmitted to adjoining structures which could be good acoustic radiators. Similarly, pressure fluctuations in hydraulic lines can be transmitted to valves, hold down clamps, etc. and cause local exitation with resulting noise.

In the case of fans, other devices in the flow loop, such as valves, duct bends and heat exchangers, can generate noise due to the airflow. In this type of noise, the level is sensitive to velocity. Thus, very small, high velocity passages should be avoided.

BEARING SELECTION

In fans, the aerodynamic noise predominates over motor and bearing noise. The type of bearing has an insignificant effect on overall noise level. On pumps, the noise levels are much lower and the type of bearing becomes significant. Sleeve bearings are recommended for a low noise level application.

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14. ABSTRACT
This document presents guidelines for the design of quiet fans and pumps of the class used on space vehicles. A simple procedure is presented for the prediction of fan noise over the meaningful frequency spectrum. A section also presents general design criteria for axial flow fans, squirrel cage fans, centrifugal fans, and centrifugal pumps. The basis for this report is an experimental program conducted by Hamilton Standard under NASA Contract NAS 9-12457. The derivations of the noise predicting methods used in this document are explained in Hamilton Standard Report SVHSER 6183, "Fan and Pump Noise Control," dated May 1973 (6).

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