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Treatment of Tonal Sound in the Development of Fan Sound Ratings

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Introduction

One of the cornerstones of Air Movement and Control Association International (AMCA) is the Certified Ratings Program. This program has helps assure specifiers, engineers, and users that air movement products are tested, rated, and certified in accordance to established testing standards. This program provides the best assurance available that the product will perform as stipulated by the manufacturer. The Certified Ratings Program starts with accurate test standards. For fan sound performance, this standard is AMCA 300, “Reverberant Room Method of Sound Testing for Fans".

From the test data, AMCA/ANSI 301, “Methods of Calculating Fan Sound Ratings from Laboratory Test Data”(1), details how the sound data for the tested fan sizes and speeds can be converted to other sizes and speeds. The cost required to test every fan size and speed required by the public would obviously be cost prohibitive and would require enormous investment in large test facilities. Therefore, the conversion to non-tested sizes and speeds is also critical for the Certified Ratings Program.

AMCA 301 was first published in 1965. The 1976 revision introduced the Generalized and Specific Sound Power methods as “graphical methods” of predicting sound for non-tested sizes and speeds. Ironically, there was no graph or figure to explain these methods in the standard. Needless to say, revisions to this standard made it easier to understand with visual examples. However, for an engineer just entering the industry, it remains a difficult standard to fully understand. While attempts have been made to simplify the methods into more of a step by step procedure, it still requires considerable experience to really understand the standard and use it effectively.

The latest version of AMCA 301 introduced another method of predicting sound called “interpolation”. This addition provided detailed guidelines to a procedure that has commonly been used to obtain sound ratings in between tested fan sizes and speeds. Some changes were also made to the graphical or fan law methods of extrapolating fan sound in the area of the Blade Pass Frequency (BPF).

Fan Sound

Fan sound, in general, consists of a broadband spectrum with peaks at discrete frequencies including the BPF and its harmonics as shown in figure 1.

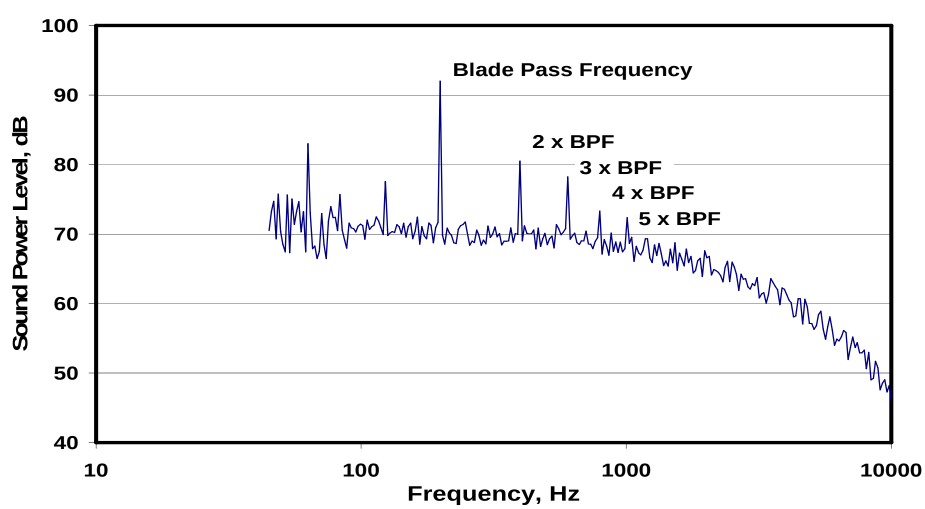

Figure 1 – Typical narrow band sound spectrum for vane axial fan

Cory (2) and Guedel (3) have provided excellent summaries of the aerodynamic sound generation mechanisms. These mechanisms can be characterized by monopole, dipole, and quadrupole sources and are shown in figure 1 from Neise (4). Some of these sources are not as important as others for the relatively low speed fans that fall under the scope of AMCA. The predominant mechanisms for a given fan vary depending on the type of fan considered.

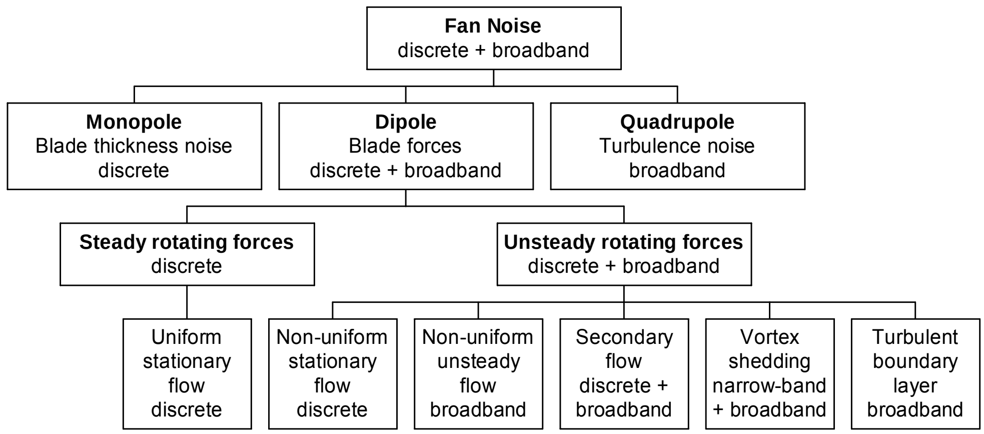

Figure 2, Fan sound generation mechanisms

According to Lighthill's acoustic analogy (5), dipole sources should produce sound levels that vary with fan diameter, D and fan speed, N as follows:

$$\text{Monopole:}\quad W \propto N^4D^6 \quad \text{or} \quad L_{W2} = L_{W1} + 40\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 60\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

$$\text{Dipole:}\quad W \propto N^6D^8 \quad \text{or} \quad L_{W2} = L_{W1} + 60\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 80\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

$$\text{Quadrupole:}\quad W \propto N^8D^{10} \quad \text{or} \quad L_{W2} = L_{W1} + 80\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 100\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

Fan Sound Prediction

There are several different methods for predicting or rating fan sound for a series of fans based on limited testing. The use of each of these methods, as with the aerodynamic fan laws, requires strict geometric similarity. Geometric similarity requires all dimensions in the flow area of the rated fan to be proportional to the base test fan. In addition, rated points of operation must be on the same system resistance lines as tested points. These requirements are no different from those of the traditional aerodynamic fan laws. Rated fan sizes should also have similar tip speeds as base sizes.

Each method uses a normalized frequency scale and sound power scale. Since the predominant sounds are tonal and related to the blade pass frequency, the sound spectra will shift with changing fan speed. This shift is accomplished by using a normalized frequency scale. Most of the methods use a frequency scale as follows:

$$X = 10\mathrm{Log}\left(\frac{f}{N}\right) + 20$$

Where f is the center frequency of the octave or one-third octave. The sound power magnitude is then adjusted for different fan speeds and diameters at constant values of X. There are many detailed procedures unique to each method, but the following is a simplification of how the sound power magnitude varies with speed and diameters.

1. Generalized Sound Power (AMCA 301)

$$L_{W2} = L_{W1} + 40\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 70\mathrm{Log}\left(\frac{D_2}{D_1}\right) + 10\mathrm{Log}\left(\frac{f_{BW2}}{f_{BW1}}\right)$$

Since the bandwidth, f_{BW}, varies directly with frequency and this sound power is compared at constant values of X, this equation is equivalent to:

$$L_{W2} = L_{W1} + 50\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 70\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

2. Specific Sound Power (AMCA 301)

$$L_{W2} = L_{W1} + 10\mathrm{Log}\left(\frac{Q_2}{Q_1}\right) + 20\mathrm{Log}\left(\frac{P_{T2}}{P_{T1}}\right)$$

Using the aerodynamic fan laws,

$$\frac{Q_2}{Q_1} = \left(\frac{N_2}{N_1}\right)\left(\frac{D_2}{D_1}\right)^3 \qquad \text{and} \qquad \frac{P_{T2}}{P_{T1}} = \left(\frac{N_2}{N_1}\right)^2\left(\frac{D_2}{D_1}\right)^2$$

the sound power equation becomes:

$$L_{W2} = L_{W1} + 50\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 70\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

3. CETIAT (French)

$$L_{W2} = L_{W1} + 50\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 70\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

4. BS 848 Part 2 (6)

This standard does not have a fixed value for the speed and diameter powers as the previous methods. Instead, it allows these power terms to be established through testing. The sound power levels are:

$$L_{W2} = L_{W1} + 10(6 + a)\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 10(8 + 2a + b)\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

The fan series is tested with several different diameters and at several different speeds. The test data is then used to solve for the variables a and b in the above equation. In this way, the method accounts for the specific behavior of each fan type of sound with respect to fan speed and diameter.

5. Other

One other concept worth noting arose from a 1994 AMCA Engineering Conference paper written by Peter Xia (7). In this paper the author proposed that, since sound sources vary with frequency, the prediction method should also vary with frequency. He stated that the fan sound spectrum should be split into 3 sections. In the lower frequencies, below the blade pass frequency, the sound is primarily due to monopole sources. In the middle frequencies, including the blade pass frequency and several harmonics, the sound is due to dipole sources. And in the upper frequencies, beyond the BPF harmonics, the sound is due to quadrupole sources. Each of these sections should have its own sound power relationship which is based on the original Lighthill work:

Low frequency (<BPF), monopole sources:

$$L_{W2} = L_{W1} + 40\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 60\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

Mid frequency (BPF and harmonics), dipole sources:

$$L_{W2} = L_{W1} + 60\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 80\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

Upper frequency (>BPF harmonics), quadrupole sources:

$$L_{W2} = L_{W1} + 80\mathrm{Log}\left(\frac{N_2}{N_1}\right) + 100\mathrm{Log}\left(\frac{D_2}{D_1}\right)$$

The second proposal from this paper related to the frequency shift due to fan speed. It was pointed out that, while discrete sounds shift in frequency with the fan speed, broadband sounds *do not* show this same shift. These discrete tonal sounds are found in the frequencies near the fan speed and near the blade pass frequency and its harmonics. Once you move away from these frequencies, the spectrum should not shift as they do in each of the 4 methods mentioned above. Xia proposed using a decreasing function for the frequency shift,

$$f = f_0 \left|\frac{N}{N_0}\right|^{\left(\frac{f_f}{f_0}\right)^{\alpha}}$$

where f_f is 1xRPM in low frequencies and 1xBPF in mid to upper frequencies, and α is determined experimentally for each fan type.

This second proposal dealing with frequency shift deserves some discussion, since it can be clearly seen in the narrow band sound spectrum. In figure 3, narrow band inlet sound power levels are shown for a 24" vane axial fan with 8 blades running at various fan speeds. The BPF and the first 4 harmonics clearly shift in frequency with increasing RPM. However, the sound spectrum below the BPF as well as above its harmonics show no frequency shift with fan speed. In fact, the sound spectra below the BPF for all fans show this same property in that they do not shift with changes in fan speed.

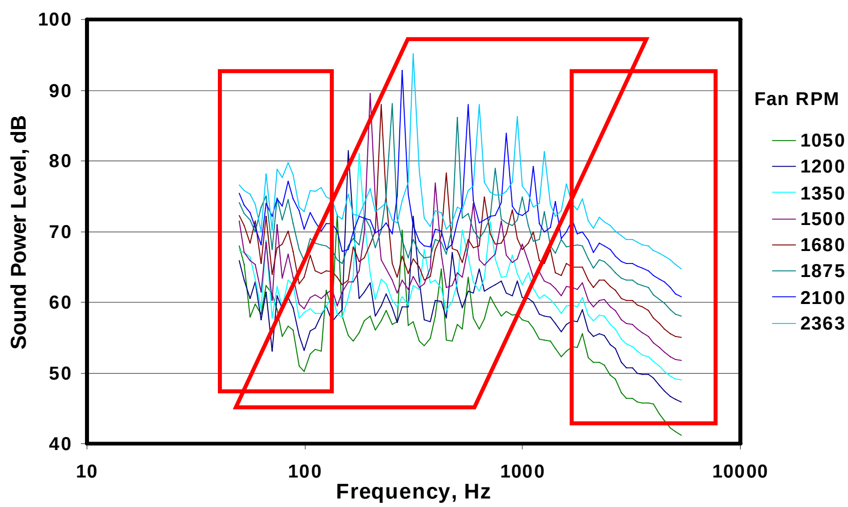

Figure 3, Narrow band sound power for 24", 8 bladed vane axial fan at various fan speeds

Accuracy of Fan Sound Prediction

In spite of the previous discussion on frequency shift occurring only in the middle frequency range, the existing sound prediction methods remain reliable. Even though the low frequency sound does not shift with fan speed, the spectrum is generally flat in this area, and the resulting full octave sound levels are similar whether the spectra are shifted or not. In the upper frequencies, the slope of the typical fan sound spectrum, combined with the powers used on the speed and diameter terms, results in reliable predictions even though it is predicting a frequency shift that doesn't occur.

As with any other performance variable, the prediction of sound is difficult when any resonances exist. In figure 4, the full octave sound power is shown for a 24" vane axial fan at various speeds. A resonance can be seen at the BPF at 1500 RPM. This resonance would not be predictable by any of the methods mentioned previously. A line showing a slope of 50Log(N) is also plotted to show that, in this case, the sound power slope is slightly greater than 50Log(N).

The accuracy of sound prediction is normally judged by plotting the normalized sound power vs. the normalized frequency. If these plotted spectra lie on top of each other, the accuracy is very good. When used to rate the sound for the same diameter fan at a higher speed, Generalized Sound Power will typically overestimate the higher speed sound. This can be seen in figure 5, showing a 22" mixed flow fan at various speeds. The spread at higher frequencies indicates some extraneous sound, probably related to drive noise, at the lowest fan speeds.

Figures 6 and 7 show comparisons of different size fans at a constant tip speed. The mixed flow fan in figure 6 shows very close agreement from the smallest fan to the largest. The design of this fan series is more consistent, with regard to geometric proportionately from the smallest size to the largest, than the BI centrifugal fan shown in figure 7. The BI centrifugal fan also has a considerable change in efficiency from the smallest size to the largest which corresponds to this sound change.

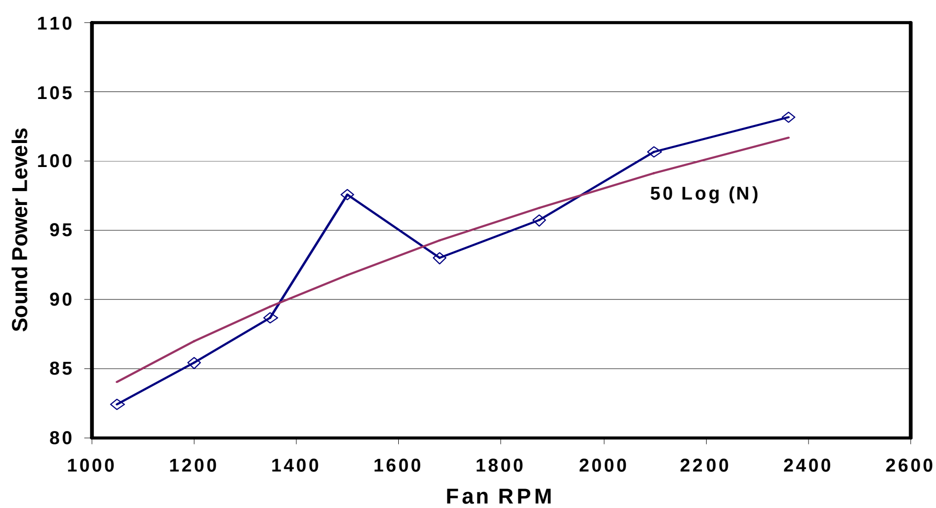

Figure 4, Full octave sound power at BPF for 24", 8 bladed vane axial fan

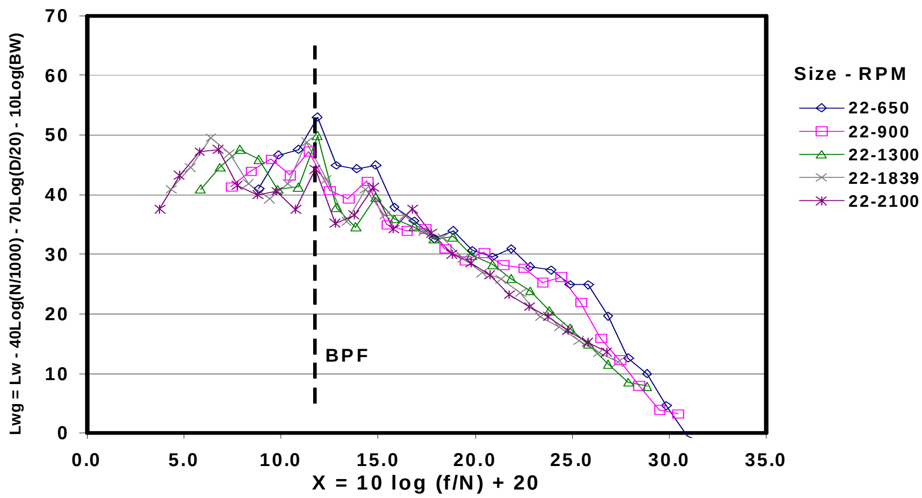

Figure 5, 22" Mixed flow fan at various fan speeds

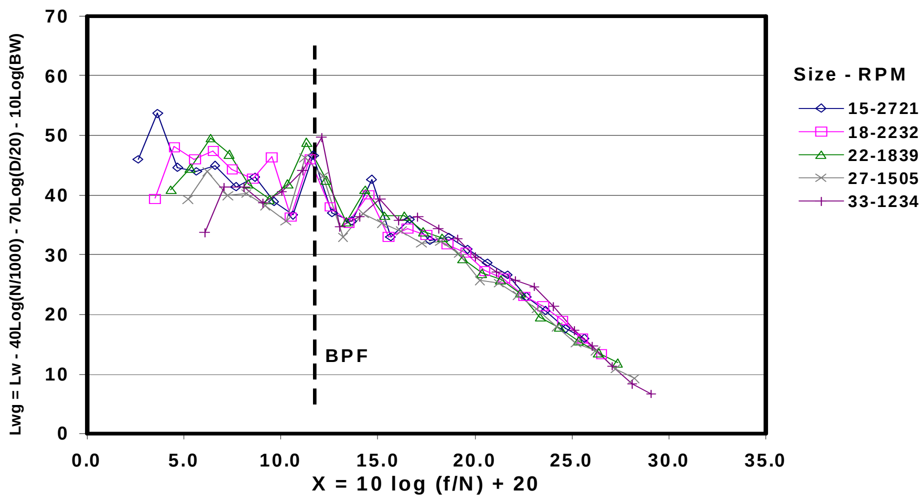

Figure 6, Various sizes mixed flow fans at constant tip speed

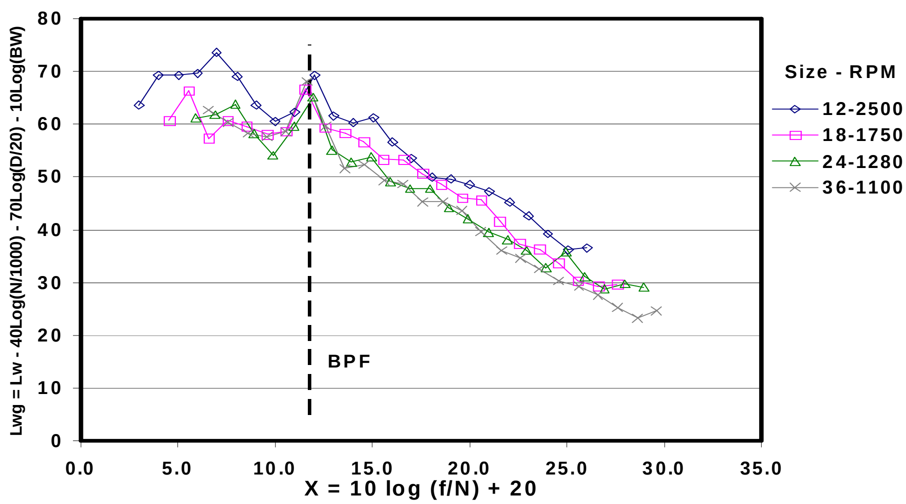

Figure 7, Various sizes backward inclined single width centrifugal fans at constant tip speed

Selection of Test Sizes and Speeds for Rating

In the original version introducing Generalized and Specific Sound Power methods, AMCA 301-76 stated:

> ***6.2.2 Adequacy of Test Data****. Overlay spectrum plots having adjacent speed values and adjacent points of operation. Conduct additional speeds and points of operation tests as needed to assure the required accuracy. Overlay spectrum plots of other fan sizes having the same speed and similar points of operation to determine the number of test sizes needed for accuracy*

This guidance for the user has since evolved into more specific rules found in Annex A of the current AMCA 301-06 standard, however this annex is informative only. The following provides additional commentary, based on experience gained in the development of numerous fan series, on some of the guidelines provided in this annex.

> ***A.1 Test sizes***
> *The manufacturer is responsible for determining the fan sizes to be tested and the number of tests that must be performed to provide the data necessary for the development of certified ratings for each fan size. The following guidelines will normally provide a sufficient number of base test sizes.*
>
> *a) The minimum test size must be the same as the minimum catalogue size.*
> *b) Small fan sizes with impeller tip diameters under 305mm (12 in.) should be tested individually.*
> *c) Fan sizes with impeller tip diameters from about 305mm (12 in.) to 914mm (36 in.) should be tested in increments not-to-exceed a diameter ratio of 1.5 between the test size and the catalogue size.*
> *d) Ideally, the largest test size should be consistent with the maximum size catalogued. However, this is usually not practical due to limitations of the laboratory. For catalogue sizes above tip diameters of 927mm (36.5 in.), it is recommended that the test size should be at least 927mm (36.5 in.) tip diameter and use a test speed as recommended in A.2.*

Using these guidelines, the test sizes should be closer together and the larger test sizes can be farther apart. One way to accomplish this is to use a logarithmic relationship to determine test sizes, where a given test fan diameter can be used to rate a fan diameter up to Dia x Log(Dia). To illustrate this using the series of fan diameters found in AMCA 99-2412-03 (8), figure 8 identifies test sizes that would make a reasonable test program.

Dia	Dia x Log(Dia)	Tested	Dia/ Test Dia
12.25	**13.33**	**X**	**1.00**
13.50	**15.26**	**X**	**1.00**
15.00	17.64		1.11
16.50	**20.09**	**X**	**1.00**
18.25	23.02		1.11
20.00	26.02		1.21
22.25	**29.98**	**X**	**1.00**
24.50	34.03		1.10
27.00	38.65		1.21
30.00	**44.31**	**X**	**1.00**
33.00	50.11		1.10
36.50	57.02		1.22
40.25	64.59		1.34
44.50	**73.35**	**X**	**1.00**
49.00	82.82		1.10
54.25	94.09		1.22
60.00	106.69		1.35
66.00	120.09		1.48
73.00	136.02		1.64
80.75	154.00		1.81
89.00	173.50		2.00
98.25	195.75		2.21

Figure 8

Strict geometric proportionality must be maintained between the test size and any rated sizes. Obviously, if there was a design change between the 22" and 24" fans which violated this proportionality, the 24" fan would need to be a test size.

These same guidelines apply to the selection of air test sizes and rated sizes using the fan laws for air performance. Significant deviations in fan efficiency occur in the smaller fan sizes, so test sizes should be closer together to improve accuracy of rated performance.

A.2 Test speeds

The fan manufacturer is responsible for determining the test speeds necessary to provide reliable sound data. Specific recommendations are contained in the following paragraphs.

a) The minimum base test fan speed should be within 5% of the minimum catalogue speed for that size fan.
b) Intermediate catalogue speeds should fall between 0.6 and 1.6 times the base test speed.
c) Maximum test speed should approximate maximum catalogue tip speed within the limits of the laboratory.
d) Test speed should be selected so that the blade pass frequency is NOT located close to either the upper limit or the lower limit of any octave or one-third octave band.

The ratio of speeds falling between 0.6 and 1.6 times the base speed results in a maximum overall sound level change of approximately 10 dB. The ratio of 1.6 also happens to be the ratio between center frequencies of one-third octaves that are two bands apart. This means that one could pick test speeds that resulted in blade pass frequencies (BPF) falling in the center of every other one-third octave and maintain this ratio of 1.6 between test speeds. Using the sound relationship of 50Log(N), this ratio would result in a 10 dB difference between test speeds. While interpolation can be done between speeds that are 10 dB apart, a more conservative approach would be to center the BPF on every one-third octave. The result would be speed ratios of 1.26 and overall sound level changes of approximately 5 dB.

Following this approach, and using the fan above with a 22.25" wheel, 10 blades, in which the catalog range of speeds was 700 RPM to 2800 RPM, the test speeds in figure 9 would be calculated.

1/3 Octave Center Freq, Hz	Fan Test RPM
125	750
160	960
200	1200
250	1500
315	1890
400	2400

Figure 9

In this case, the lowest test RPM could be reduced to 700 RPM to coincide with the lowest catalog speed. If the 2400 RPM test speed results in airflow, pressure, or BHP which is beyond the capabilities of the laboratory, this RPM can be omitted and the test results at 1890 RPM can be rated up to the highest catalog speed. This technique of starting with center frequencies to determine test RPM's will give a sufficient number of test speeds and will result in consistent sound ratings when using either of the fan law methods to develop the catalog data.

Once the test sizes and speeds have been chosen for a series of fans, individual tests are conducted according to AMCA 300-06. Generalized sound power plotting should be used as these tests are conducted to ensure consistent results are obtained with each test. Resonances, mechanical vibration, excess background noise, and other anomalies can easily be identified by viewing these spectrum plots. Test data can then be rated to other catalog sizes and speeds within the fan series.

References

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