NOISE EMISSION, POWER CONSUMPTION, AND PERFORMANCE PREDICTION USING MULTIDIMENSIONAL FAN CURVES

David NELSON

Nelson Acoustics, Box 879, Elgin, Texas, United States of America

SUMMARY

In order to be quiet and efficient, air movers should be either designed or selected to match the air-moving task. Even intrinsically quiet and efficient air movers lose their advantages when misapplied. Multidimensional performance evaluation yields iso-acoustic and iso-power curves that complement the traditional fan curve. The method allows joint evaluation of performance, noise emission and power consumption for combinations of air mover type, diameter, parallel and series combinations, system backpressure and inflow conditions. A full exposition of the method is provided, including typical curves for common air movers and examples from recent experience.

INTRODUCTION

Heat dissipation of air-cooled equipment is growing with computing power. Besides providing adequate cooling, a successful design is increasingly defined by being both intrinsically quiet and efficient. This is especially important in datacenter and telecom applications where the high flowrates can lead to hearing conservation noise levels and prodigious energy budgets. The same is true for aerospace applications, where stringent takeoff/launch weight and in-flight or on-orbit power limitations complement efforts to improve crew communication through noise reduction.

Early design decisions made on the basis of the ubiquitous and straightforward “fan performance curve” often fail because of the unavailability of corresponding fan noise and power data. These crucial performance parameters may not manifest themselves until it’s too late to alter the design. Efforts to muffle excessive noise seldom result in significant noise reduction, but can be counted on to require additional power. Thus it’s important to address performance, noise, and power at the outset.

This paper describes methods by which the required data may be obtained, how it can be adapted for straightforward use, and ways it has been strategically used to support intrinsically quiet, efficient airflow designs.

OBTAINING DATA

Several AMD (air moving device) manufacturers now publish A-weighted sound power level and power consumption data at several points in the operating range. Others will provide this data on request. While this is an excellent development and a major improvement, the fact remains that the equipment under test will most likely be an AMD with an unrealistically pristine inflow condition. Turbulent inflow conditions inherent in real devices usually create additional noise, and may affect other performance parameters. Hence it's important to quantify those effects as well.

The ISO 10302 method involves an acoustically transparent plenum with variable backpressure. It came into use at IBM in 1964, became an INCE Recommended Practice in 1985 and an ISO Standard in 1996 [1,7]. In the early days a plenum test involved tedious flow gate adjustments to match specific operating points used during flow bench testing, followed by serial acquisition of 1/3 octave band sound pressure levels at ten or more microphone positions. Measurement at a few operating points might require an entire day.

Recently the plenum operation has been automated [2], adding remote sensors, using a provisional plenum flow-pressure calibration, and acquiring all physical and acoustical data simultaneously. In this way an AMD can be fully characterized, along with a realistic mockup inlet condition, using ten to twelve operating points in less than one hour.

Fan plenum results for flow, pressure, noise, and power are cast in non-dimensional form to allow exploration of different speeds and diameters. Polynomial curve fitting enables fan characteristics to be cast as functions of flow. These are combined by means of the fan scaling laws to yield iso-acoustic and iso-power curves that can be used in a manner similar to the familiar "fan-" curve.

NON-DIMENSIONAL CURVES

AMD performance has six dimensions: speed, diameter, flow, pressure, power, and noise [4,5,6,8,9,10,11,12,15]. Using data obtained from the manufacturer or from ISO 10302 testing, non-dimensional values are computed for each operating point:

$$\text{Flow coefficient: } \phi = \frac{Q}{A_F V_T} = \frac{Q}{\frac{\pi^2}{4} N D^3} \tag{1}$$

$$\text{Pressure coefficient: } \psi = \frac{P_T}{\frac{1}{2}\rho V_T^2} = \frac{P_T}{\frac{\pi^2}{2}\rho N^2 D^2} \tag{2}$$

$$\text{Power coefficient: } \lambda = \frac{W_E}{\frac{1}{2}\rho A_F V_T^3} = \frac{W_E}{\frac{\pi^4}{8}\rho N^3 D^5} \tag{3}$$

$$\text{Efficiency: } \eta = \frac{Q P_T}{W_E} = \frac{\phi\psi}{\lambda} \tag{4}$$

$$\text{Specific Sound Power Level: } K_W = L_W - 10\log\left(QP^2 \middle/ Q_0 P_0^2 \right) \tag{5}$$

where Q is the overall flow and P_T the fan total pressure. Q_0 and P_{T0} are 1 m^3/s and 1 Pa, respectively.

Polynomial curve-fitting (generally fourth or greater order) is used to convert these curves to functions of ϕ.

The non-dimensional system curve depends on the diameter of the fan and the backpressure coefficient:

$$\text{Non-dimensional System Curve: } \psi = R\left(\frac{A_F}{A_0}\right)^2 \phi^2 \tag{6}$$

where A_0 is 1 square meter, and

$$\text{Backpressure Coefficient: } R = \frac{P_T}{\frac{1}{2}\rho\left(\frac{Q}{A_0}\right)^2} \tag{7}$$

MULTI-DIMENSIONAL CURVES

The iso-acoustic and iso-power curves are determined by computing the speed at each operating point that leads to the criterion noise emission or power [13,14,16,17,18,19]. The locus of pressure/flow combinations for a given criterion form the corresponding iso- curves. Just as in the familiar fan curve, the point of intersection with the system curve indicates the operating point at which the AMD just meets the criterion.

These curves are essentially sections of a six-dimensional space, holding two variables constant (usually diameter and either noise or power) while displaying two. The remaining two (speed and either power or noise) are free to vary.

By comparing noise emission, power, and speed limitations, it's possible to create an envelope of operating points that simultaneously satisfy all requirements.

A noise-based performance function, intrinsic to a given homologous AMD, can be written as

$$\Pi_A(\phi) = \left(\frac{10^{-0.1K_W(\phi)}}{\frac{\pi^6}{16}\phi\psi^2(\phi)}\right)^{1/5} \tag{8a}$$

A similar power-based performance function can be written as

$$\Pi_W(\phi) = \left(\frac{\pi^4}{8}\lambda(\phi)\right)^{-1/3} \tag{8b}$$

From this it follows that the iso-acoustic speed N_A is

$$\text{Iso-acoustic Speed: } N_A(\phi) = \left(\frac{10^{0.1L_C}}{\rho^2 D^7}\right)^{1/5} \Pi_A(\phi) \tag{9a}$$

and the iso-acoustic flow, pressure and power Q_A, P_A, and W_A are:

$$Q_A(\phi) = \frac{\pi^2}{4}\phi\left(10^{0.1L_C}\frac{D^8}{\rho^2}\right)^{1/5}\Pi_A(\phi) \tag{9b}$$

$$P_A(\phi) = \frac{\pi^2}{2}\psi(\phi)\left(\frac{\rho}{D^4}10^{0.2L_C}\right)^{1/5}\Pi_A(\phi)^2 \tag{9c}$$

$$W_A(\phi) = \frac{\pi^4}{8}\lambda(\phi)\left(\frac{D^4}{\rho}10^{0.3L_C}\right)^{1/5}\Pi_A(\phi)^3 \tag{9d}$$

Similary the iso-power speed N_W is

$$N_W(\phi) = \left(\frac{W_C}{\rho D^5}\right)^{1/3}\Pi_W(\phi) \tag{10a}$$

and the iso-power flow, pressure, and noise emission Q_W, P_W, and $L_{W,W}$ are:

$$Q_W(\phi) = \frac{\pi^2}{4}\phi\left(W_C\frac{D^4}{\rho}\right)^{1/3}\Pi_W(\phi) \tag{10b}$$

$$P_W(\phi) = \frac{\pi^2}{2}\psi\left(\frac{\rho}{D^4}W_C^2\right)^{1/3}\Pi_W(\phi)^2 \tag{10c}$$

$$L_{W,W}(\phi) = K_W(\phi) + 10\log\left[\frac{\pi^6}{16}\phi\psi^2(\phi)\left(\frac{\rho}{D^4}W_C^5\right)^{1/3}\Pi_W(\phi)^5\right] \tag{10d}$$

APPLICATION

The curves are intended to make the prediction of noise and power consumption as straightforward as estimating performance using a fan curve. The following examples demonstrate the effectiveness of the method and the extent to which it clarifies design choices.

Identifying regimes controlled by noise, power, and speed

Project limits of 75 dB(A), 30 W, and 158 s^{-1} are applied to an 80 mm axial fan with clean inlet. Figure 1 shows that, for some operating points (between the system curves) flow is limited by the power criterion. Otherwise, noise emission is the limiting factor. The highest flowrate that satisfies all three limits is approximately 115 m^3/s. On the left side of the diagram small increases in flow resistance can lead to dramatic reductions in noise-limited flow because the iso-acoustic curve is nearly parallel to the system curve.

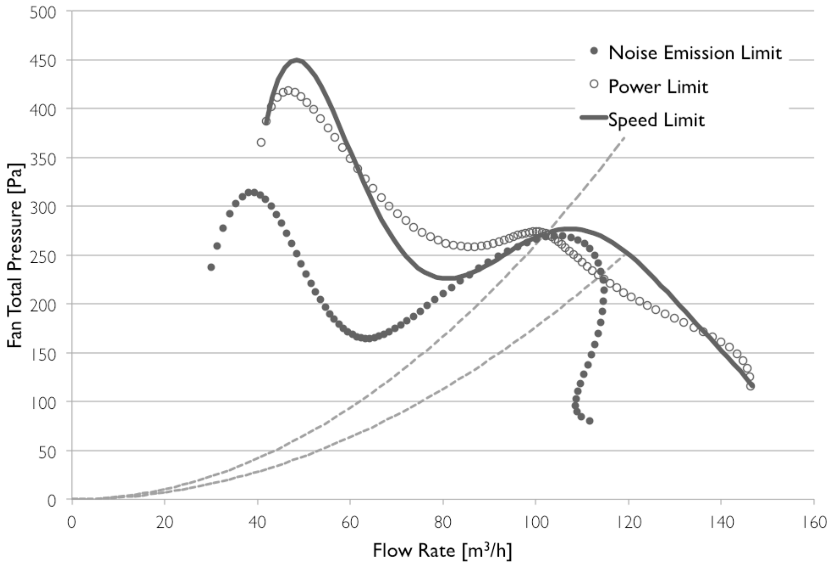

Figure 1: one AMD, three simultaneous criteria

Quantifying real-world inlet conditions

The same AMD was tested simulating an array installation above a narrow plenum about two impeller diameters deep. Figure 2 shows that installed noise emission of the fan increased significantly (about 5 dB(A) at the desired operating point), but the power consumption changed little. This is consistent with small-scale turbulence, which significantly affects noise emission but changes power and performance little.

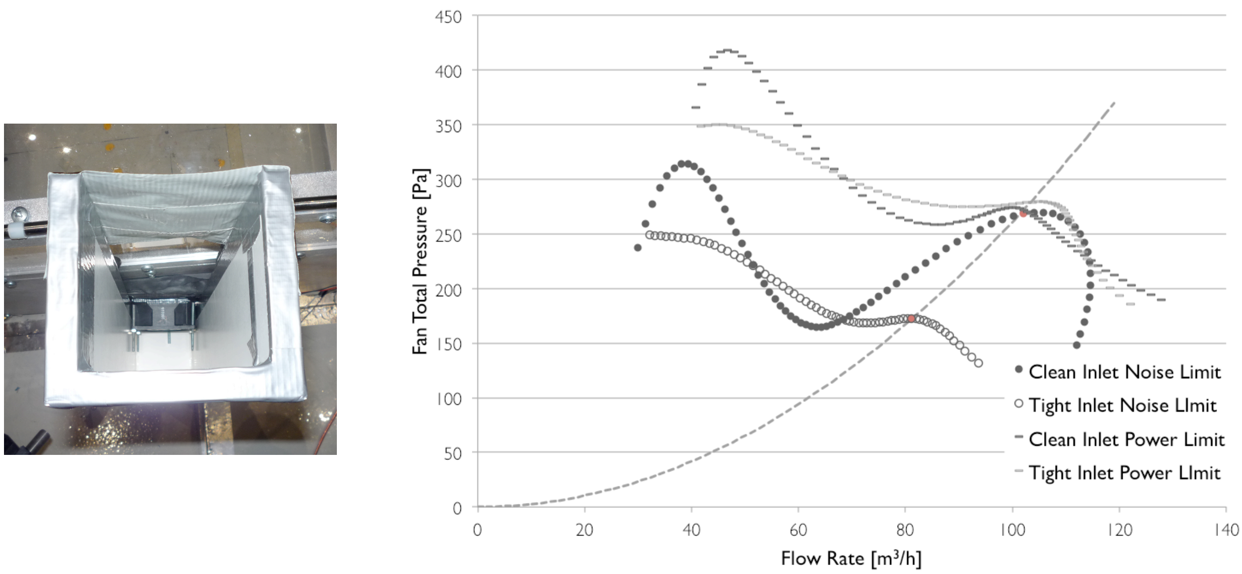

Figure 2: effect of tight inlet condition

Identifying the optimal diameter AMD

For a given AMD the non-dimensional operating point can only be altered by changing the chassis backpressure or AMD diameter. It follows that each fan type has an ideal diameter for a specific chassis.

Figure 3 shows that a 55 mm version of this fan would net a 20% flow gain relative to a 60 mm or 80 mm version of the same AMD before reaching the noise limit for the system shown. The counterintuitive result is that the smaller AMD delivers more flow within the given constraints. In general, larger diameters are appropriate for less resistive systems.

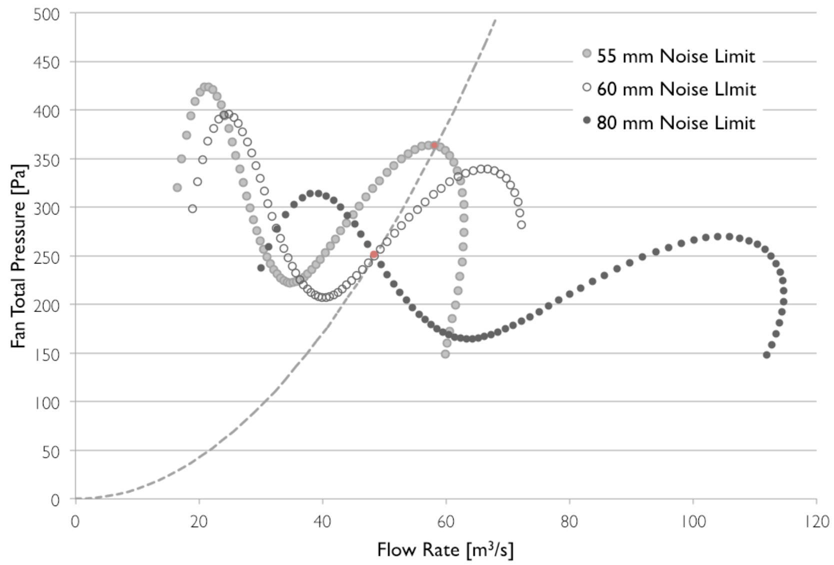

Figure 3: optimizing diameter for best flow

The same approach can be used to demonstrate potential noise control and power reduction. If the performance of the 80 mm AMD in Figure 3 were considered adequate, a 55 mm version would be 4 dB(A) quieter and reduce power consumption by about 20% at that operating point.

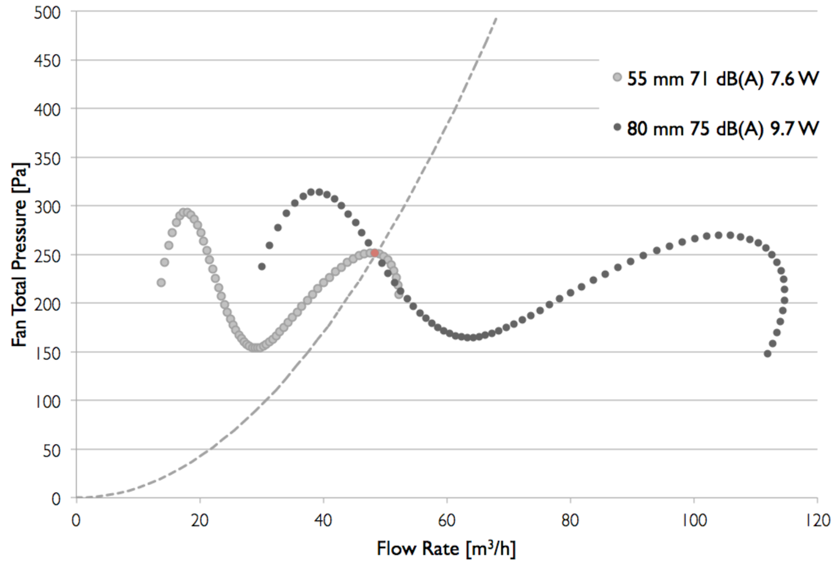

Figure 4: optimizing diameter for reduced noise and power consumption

"Stacking" axial fans

Axial fans are better suited for high-flow- than high-pressure applications. However, current equipment has generally been designed around axial fans. Axial fans are often "stacked" in high impedance systems in an attempt to compensate. However if the fans are in close proximity, the downstream turbulence and swirl of the primary fan reduces the performance of the secondary fan.

Figure 5 compares a pair of 80 mm axial fans in an "ideal stack" (single fan result with pressure delivery doubled), with two axial fans directly attached (no spacing) and with a 12,5 mm gap between them. The single-fan (unstacked) curve is provided for reference.

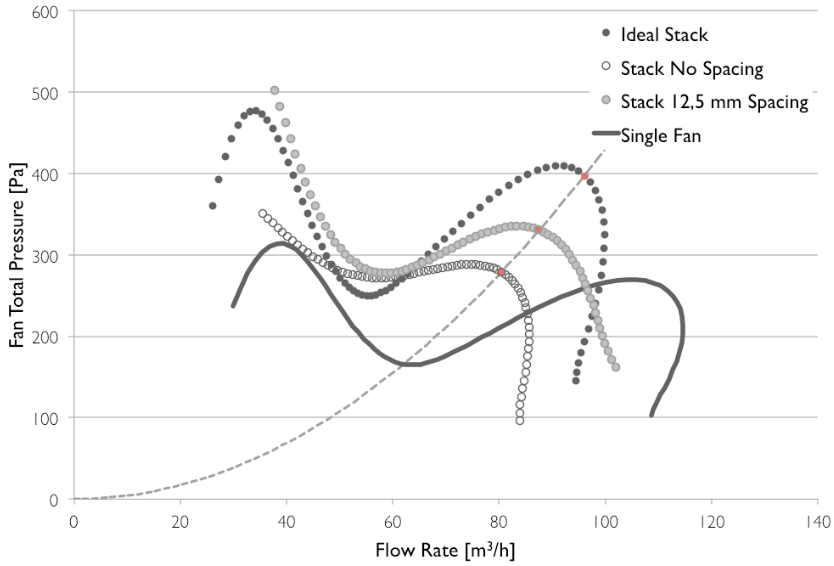

Figure 5: effect of stacking with and without spacers, 75 dB(A) sound power level

The stacked fan arrangement would indeed deliver more flow than unstacked fans. Stacked fans with no spacers are expected to produce 80 m^3/s at 75 dB(A) in this system. A 12,5 mm gap improves the secondary fan inlet conditions sufficiently to reach about 90 m^3/s. An ideal stack, such as might be approximated using flow optimizers or a large gap, would deliver close to 100 m^3/s.

Comparing three axial fans

Three 172 mm axial fans were compared at a noise limit of 75 L_{WA}. Fans A and B have 5 blades, fan C has 7. Fan B has an unusual blade design, and appears to have been optimized for high-pressure systems (left-most system curve) for which it would deliver about 35% more flow. The situation is reversed in a low-pressure system (right-most system curve), for which Fan C would be the best performer.

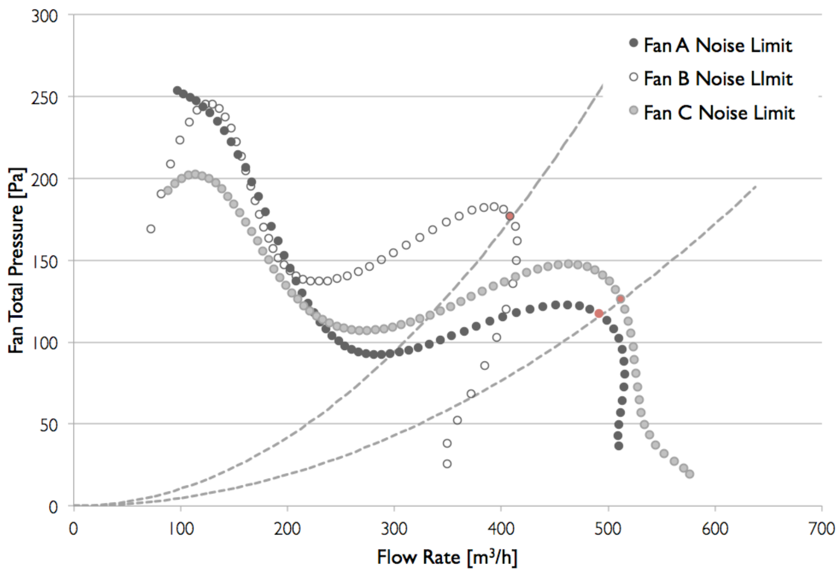

Figure 6: comparison of axial fan models

Comparing an axial fan and a blower

A 250 cfm axial fan was compared to a 125 cfm packaged blower. Once again, it turns out that within the project constraints the "smaller" blower delivers more cooling flow.

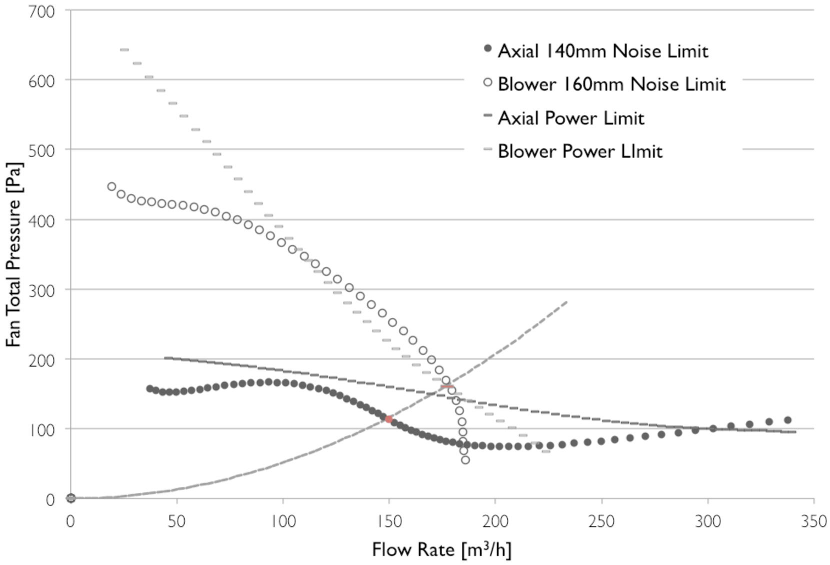

Figure 7: comparison of axial fan and packaged blower

CONCLUSION

Performance, power consumption, and noise emission of air-cooled equipment can be predicted rapidly and with confidence, very early in the design process, through the use of empirical multi-dimensional curves. Curves from existing fans and mockup flow conditions have been used to explore airflow design choices prior to prototyping. In particular, non-optimal inlet conditions, the effects of stacking fans (both with and without spacers), and the consequences of diameter choices, have been studied. Intrinsic performance curves based on noise emission and power consumption, applicable to a given AMD homologous design, have been identified. The main contribution of this work lies in demonstrating the flexibility and relative simplicity of the method.

BIBLIOGRAPHY

[1] G. C. Maling – *Historical Developments in the control of noise generated by small air moving devices* Noise Control Eng. J. 42 (5) **1994** Sept.-Oct.

[2] J. G. Schmitt, D A. Nelson, J. Phillips, *An automated system for the acoustical and aerodynamic characterization of small air-moving devices*, Proceedings of Noise Con **2005**.

[3] D. A. Nelson, W. Butrymowicz, C. Thomas, *Effect of discharge duct geometry on centrifugal fan performance and noise emission*, Proceedings of Noise Con **2005**.

[4] E. Baugh, *Impact of inlet restrictions on acoustics for blowers in notebook PCs*, Proceedings of Internoise **2006**.

[5] W. Beltmann, *Quantification and modeling of fan installation effects*, Proceedings of Internoise **2006**.

[6] D. A. Nelson, *Axial fan installation effects due to inlet flow distortions*, Proceedings of Internoise **2006**.

[7] ISO/DIS 10302-1 – *Acoustics - Measurement of airborne noise emitted and structure-borne vibration induced by small air-moving devices – Part 1: airborne noise measurement*, **2007**.

[8] D. A. Nelson, *Assessing the costs of poor fan inflow conditions for spaceflight hardware*, Proceedings of Fan Noise **2007**.

[9] M. Holahan, M. O'Connell, *Mapping fan noise across the hydraulic plane*, Proceedings of Fan Noise **2007**.

[10] E. Baugh, M. MacDonald, *Constant sound power fan curves for small blowers*, Proceedings of Noise-Con **2007**.

[11] D. A. Nelson, *Fan selection and installation issues related to spaceflight hardware*, Proceedings of Noise Con **2007**.

[12] D. A. Nelson, *Cooling requirements, chassis design and fan noise*, Proceedings of Noise-Con **2007**.

[13] D. A. Nelson, *The Influence of Cooling Requirements, Chassis Resistance, and Fan Selection on Noise Emission*, Proceedings of 25th IEEE Semi-therm Symposium, **2009**.

[14] D. A. Nelson, *Estimating noise emission from fan-cooled packages*, Proceedings of Internoise **2009**.

[15] ECMA TR-99 - *Constant Sound Power Fan Curves for Small Air-moving Devices*, 1st Edition, December **2009**

[16] David A. Nelson, *Noise emission and power consumption estimation methods for fan-cooled equipment*, Proceedings of Noise-Con **2011**.

[17] R. Jorgensen, Ed. *Fan Engineering*, Ninth Edition, Howden Buffalo, 1999.

[18] D. A. Nelson, *Extending noise- and power-optimized fan selection into the sound quality domain using iso-acoustic and iso-power methods*", Proceedings of Noise Con 2013

[19] D. A. Nelson, *Reducing noise from fan-cooled equipment,* IMAPS Thermal Management, 2014