NORTEK AIR SOLUTIONS

Improving Fan Array Vibration

with Next Generation Design

AT A GLANCE

- Structural resonances can be reduced without isolators.
- Vibration signature of the fan array can be improved through next generation design methods.
- Fan arrays with multiple, smaller fans offer several benefits over systems with larger fans.

Fan arrays with multiple, smaller fans offer several benefits over systems with larger fans including lower sound, increased redundancy and smaller footprints to name a few. The fans utilize direct drive centrifugal wheels that are coupled directly to the motor shaft. The rotating wheel has a gap to the inlet cone which directs air into the wheel. Small vibration isolators are sometimes used between the motor and fan array mount. In some cases, these vibration isolators can create vibration resonance bands in the system. This white paper discusses the resonance issues of the isolators, and how to mitigate those resonance issues and improve the vibration signature of the array through next generation design methods.

Vibration Transmissibility, Static Deflection, Natural Frequency and Damping

A vibrating fan/motor mounted on isolators can in its simplest form be considered a single degree-of-freedom spring-mass system. The transmissibility, T, of the isolated system can be represented by the following equation:

$$T = \sqrt{\frac{1+\left(2\zeta\frac{f_d}{f_n}\right)^2}{\left[1-\left(\frac{f_d}{f_n}\right)^2\right]^2+\left(2\zeta\frac{f_d}{f_n}\right)^2}}$$

where,

$f_n = Natural\ Frequency$

$f_d = Driving\ Frequency$

$\zeta = Damping\ Ratio\ (zeta)$

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This is shown in Figure 1, where the Transmissibility is shown on the vertical axis versus the Driving Frequency that is shown on the horizontal axis. There is amplification of the vibration (shown as the red area in the chart where T exceeds 1.0) when the driving frequency gets close to the natural frequency. When operating at above around 1.5 times the natural frequency, the system starts to isolate the vibration (shown as the green area in the chart).

Figure 2 shows the effect on Transmissibility of different Damping Ratios (ζ pronounced zeta) of the isolator. Higher damping ratios reduce amplification, but also reduce the effectiveness of the isolation region.

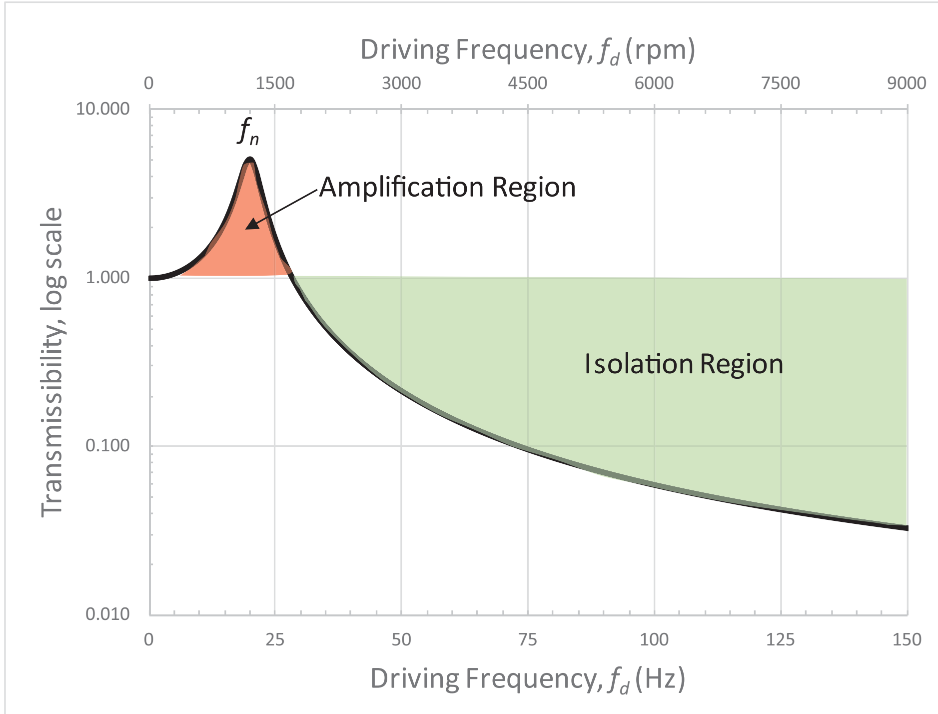

Figure 1: Transmissibility of typical spring mass system. fn=20Hz.

We now discuss vibration transmissibility in terms of available height of isolator static deflection and resonant frequencies. Isolator static deflection, δ, is determined by the following equation:

$$\delta = \frac{mg}{k}$$

$$\text{where,}$$
$$m = mass$$
$$k = isolator\ stiffness$$

The Natural Frequency, f_n, of the spring mass system is determined by the following equation:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{g}{\delta}}$$

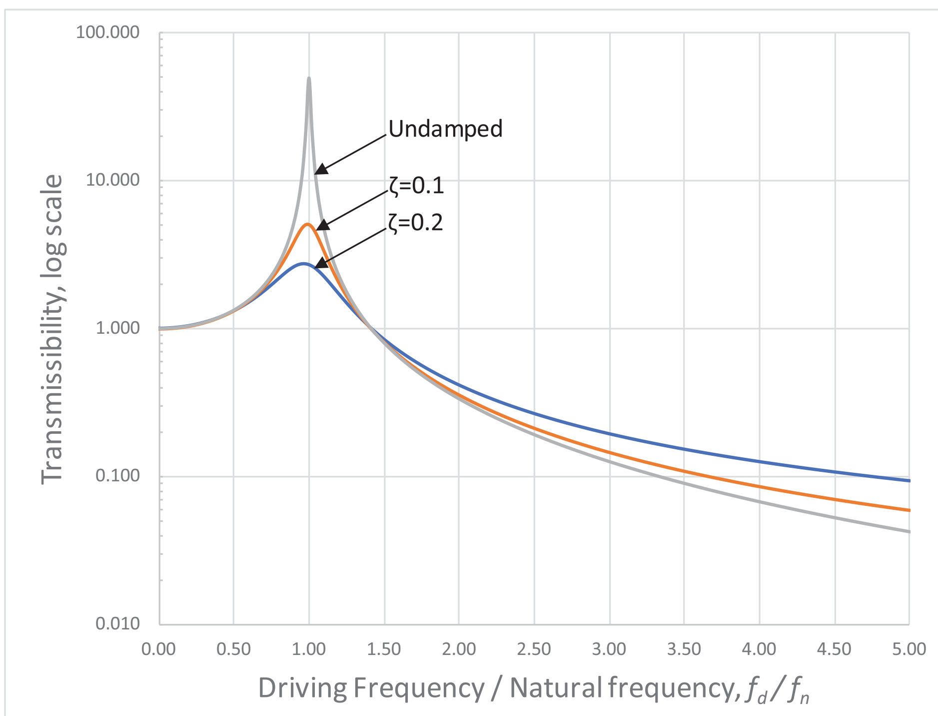

Figure 2: Effect of isolator Damping Ratio.

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This is shown in Figure 3 and Figure 4. Higher static deflection isolators have a lower natural frequency—meaning that they will have a lower speed at which they are effective as compared to an isolator with less static deflection. Generally, you want a lower natural frequency (higher static deflection) so that you can have your operating range of the fan above the isolator natural frequency.

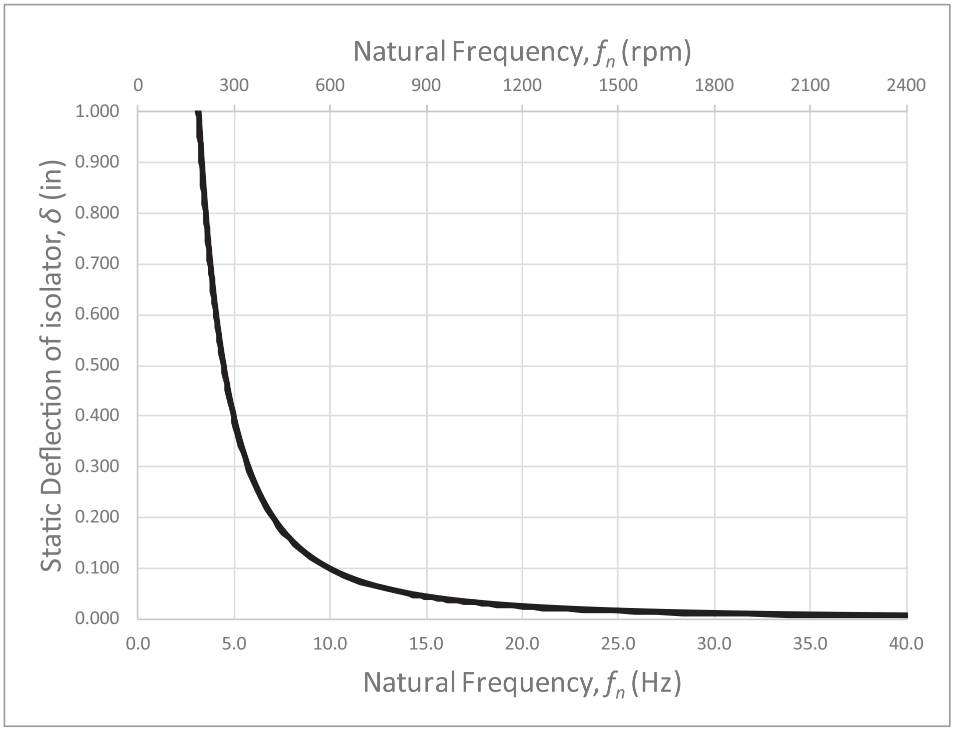


Figure 3: Static deflection of isolator and natural frequency over large range.

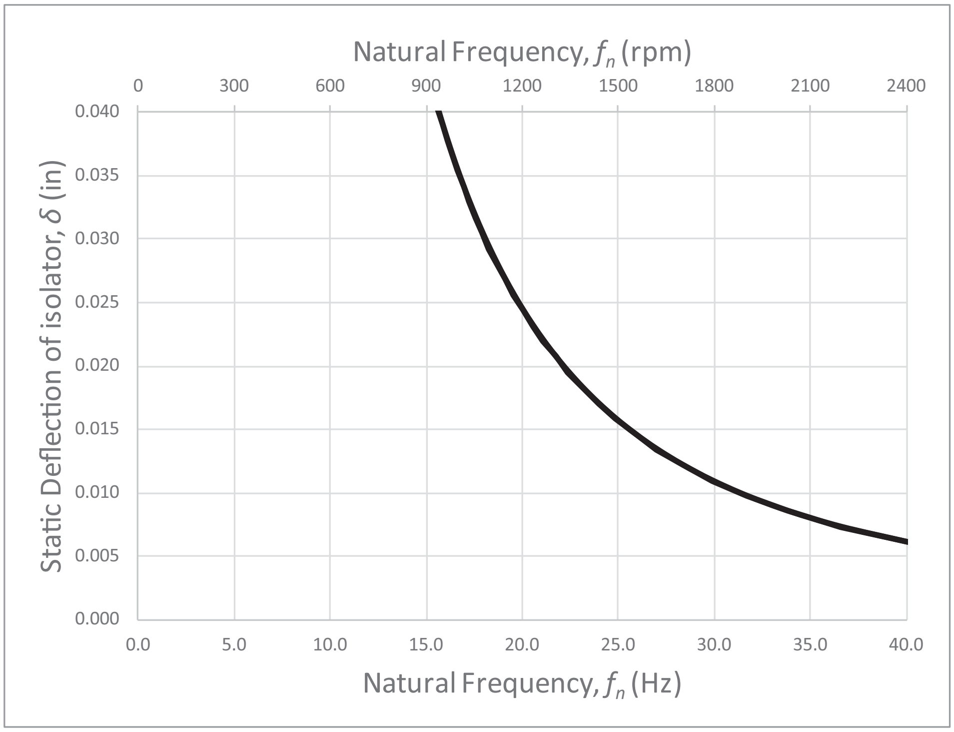


Figure 4: Static deflection of isolator and natural frequency.

Issues Encountered with Wheel/Motor Isolation

As stated previously and shown in Figure 1, the isolated system has a resonant frequency that needs to be avoided in the design. The issue with utilizing isolation under the wheel/motor is that it requires a higher deflection amount than can be tolerated physically by the fan wheel to inlet cone gap. If the deflection is not sufficiently high, it can place the resonant frequency in the operating range of the fan. However, too high of a deflection can result in wheel to cone contact.

Additionally, some damping is desired in the isolator to reduce the magnification in the resonant band, but this also reduces the effectiveness of the isolator. This results in a wider band of speed that will need to be skipped or locked-out than is generally permitted for operation of the fan system.

To discuss some specifics for general HVAC fan applications, these fans operate in the range of 600 to 4000 rpm (10 Hz to 70 Hz on the Driving Frequency). For the resonant band of the isolated system to be below the operating range of the fan, one would have to pick an isolator with a natural frequency below 7 Hz (or above 0.2" of static deflection). This is shown on the orange curve in Figure 5.

This amount of deflection directly under the fan/motor will cause the physical wheel and cone to come into contact with one another. The black line in the figure shows a curve for the maximum amount of isolator deflection that can generally be tolerated by the wheel to cone gap—which is around 0.024" (f_n of 20Hz). One can see that this undesirably places the resonant band of this isolator right in the typical operating band of the fan of 600 to 1800 rpm.

Analysis of the frequency range of operation for fan array style fans shows that the effective frequencies of vibration isolation occur outside the range of typical operating speeds when considering the limits on deflection required by cone to wheel gaps. This indicates that if isolation were to be used, it needs to be of enough deflection to require use of isolators of the full fan or unit assembly and not isolation at the wheel to cone level. Therefore, a new design innovation is called for. That innovation is called the Gen III fan system.

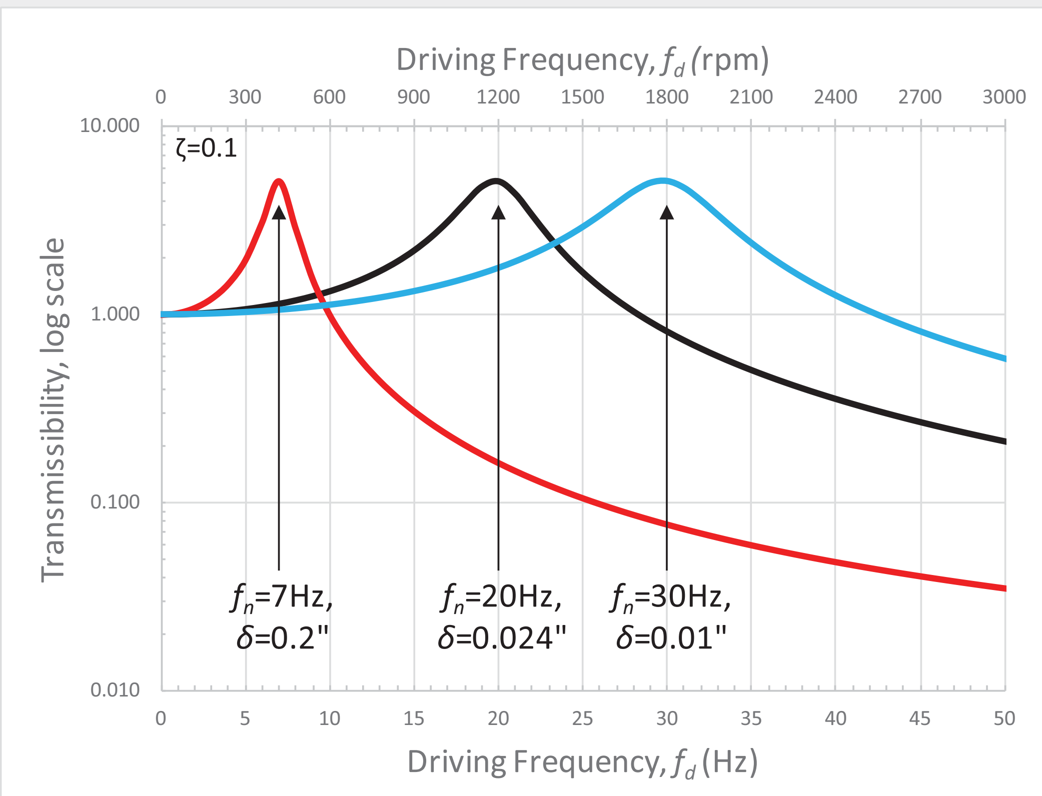


Figure 5: Transmissibility for minimum 600 rpm operation and maximum deflection tolerated by cone gap.

New Gen III Design Innovation and Frequency Ranges

The Gen III fan system features an increased stiffness of the supporting fan structure (specifically on the motor pedestal mounts and rails) to increase the natural frequency of the system. The fan support components have been increased in stiffness and the polymeric isolators have been removed to minimize structural resonances. This allows for the general avoidance of the frequency range where the fan must ride through the isolator resonant frequency band. This should not be construed as saying that vibration isolation is not required for some applications. It is just saying that isolation, if required, should take place on the fan assembly or unit level since, for the isolation to be effective, it has to be of a much larger static deflection.

In order to demonstrate the difference in the vibration levels between the Gen II and Gen III setups, a 15-fan array with the traditional Gen II motor pedestal and rail configuration was tested and compared to one with a Gen III motor pedestal and rails. This vibration order tracking data presented in Figure 6 is taken with all fans running with a slow ramp up of speed to capture the peak levels at all speeds during the full ramp up. There is a significant improvement in removing the resonance bands with the more robust structure.

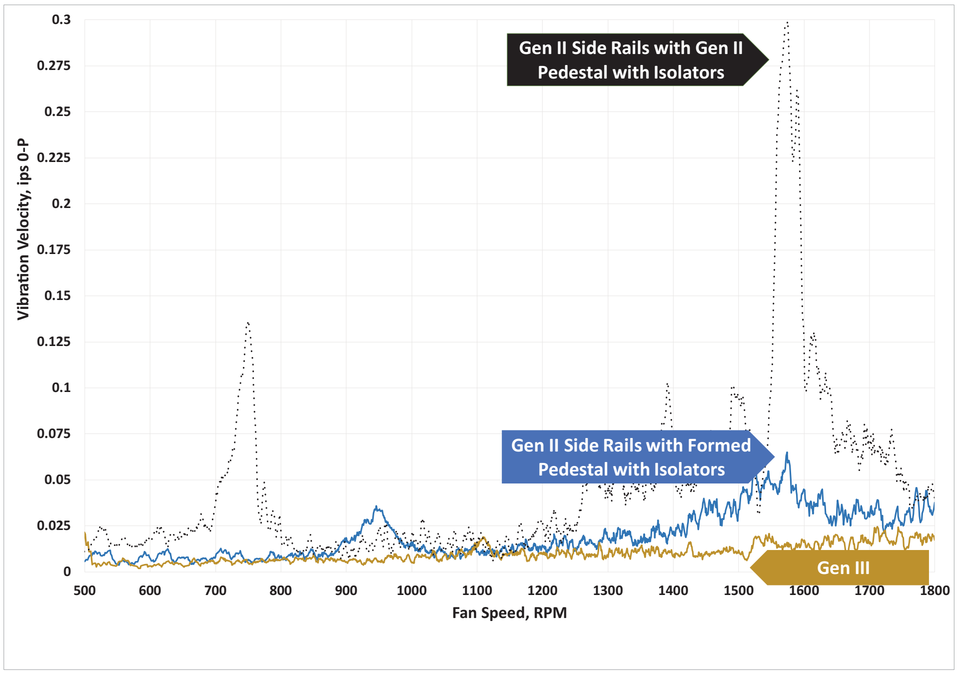


Figure 6: Vibration order tracking of fans in 15-fan array with all fans running - Gen II and Gen III.

In order to demonstrate the resonant effect of the isolator with low enough deflection to avoid wheel to cone contact, we further added in polymeric isolators to the Gen III construction. This data is shown in Figure 7 where you can see the amplification of the vibration levels from the isolators.

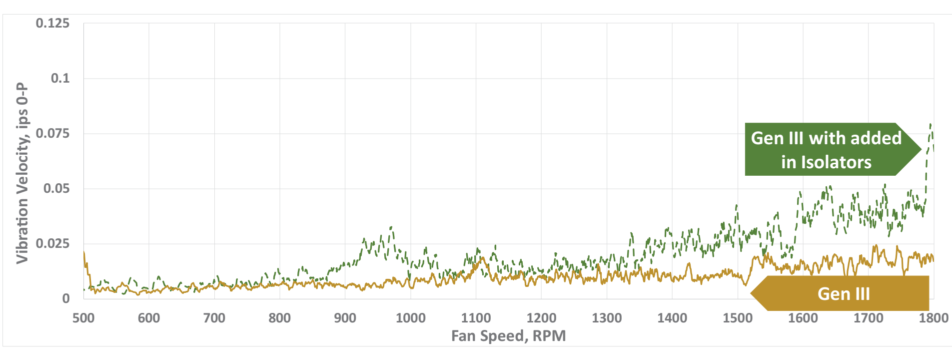


Figure 7: Vibration order tracking of fans in 15-fan array with added in isolators to Gen III.

New Gen III Design Innovation and Frequency Ranges

The isolator resonance is further demonstrated on another fan setup running up to higher speeds as shown in Figure 8 where an intentionally out-of-balanced fan was used.

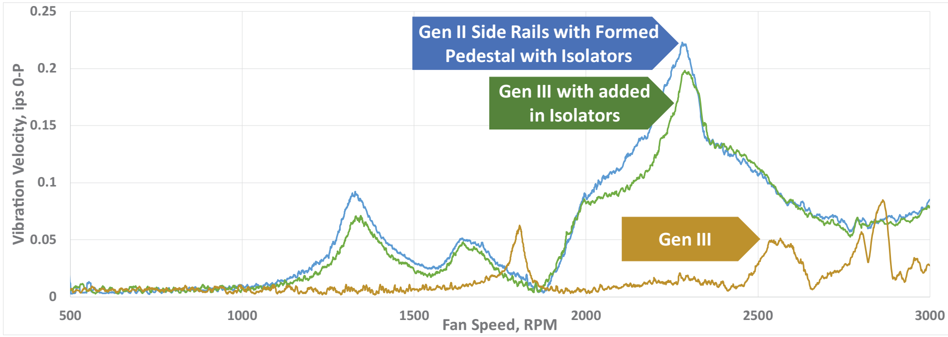


Figure 8: Vibration order tracking of Gen II showing isolator resonance compared to no isolator of Gen III on an intentionally out-of-balanced fan.

AMCA standard 204-05 provides some guidelines for balance grades. Fans for HVAC applications are generally balanced to a balance quality grade of G6.3 for BV-3 applications. Our FANWALL TECHNOLOGY® fans are balanced to a low vibrational residual imbalance grade of around G0.55 (corresponding to a vibration velocity of around 0.022 inches/s 0-peak) which is lower than the balance grade required for stringent BV-5 applications. The permissible residual unbalance, U_per, is determined by the following equations:

$$U_{per} = W\, e_{per}$$

$$e_{per} = \frac{G}{25.4\omega}$$

$$\omega = \frac{2\pi N}{60}$$

$$\begin{gathered} where, \\ N = Rotational\ speed,\ rpm \\ W = Rotor\ weight,\ lbm \\ \omega = Angular\ velocity,\ rad/s \\ e_{per} = Specific\ unbalance,\ in\ or\ lb\ in/lb \end{gathered}$$

An example comparison of this is shown in Figure 9 where one can see a lower calculated residual unbalance for the un-isolated FANWALL® setup as compared to the isolated single fan.

Reference standard AMCA 204			**FANWALL fan array**	**Single plenum fan**	
Parameter		**Units**	**7 fans 18" wheels**	**1 fan 49" wheel**	
Balance Quality Grade for Fan Application Category			G0.55 custom for below BV-5	G6.3 for BV-3	G6.3 for BV-3
Balance quality grade	G	mm/s	0.55	6.3	6.3
Speed	N	rpm	2035	705	705
Angular velocity	ω	rad/s	213	74	74
Rotor weight total	W	lbm	101	435	435
Specific unbalance	e_{per}	in or lb in/lb	0.000102	0.003360	0.003360
Permissible residual unbalance (moment)	U_{per}	**lb in**	**0.010**	**1.461**	**1.461**
Isolation			None	Spring 95% efficient	Spring 97% efficient
Transmissibility of isolation	T		1.00	0.05	0.03
Transmitted Permissible residual unbalance	**U_{per} transmitted**	**lb in**	**0.010**	**0.073**	**0.044**

Figure 9: Example residual unbalance comparison for FANWALL versus single fan.

New Gen III Design Innovation and Frequency Ranges

To further demonstrate the vibration levels, Gen III compared to traditional large centrifugal fans, we set up a test of similar performance AHUs that were run at the same operating conditions of delivered airflow and external static pressure. These units were set on large deflection test springs so that the vibration levels could be measured on the base of the units. The FANWALL® unit does not have spring isolated fans and the large housed centrifugal and twin plug fans do have spring isolation on the fans. The results in terms of whole unit acceleration are shown in Figure 10 and in terms of force are shown in Figure 11.

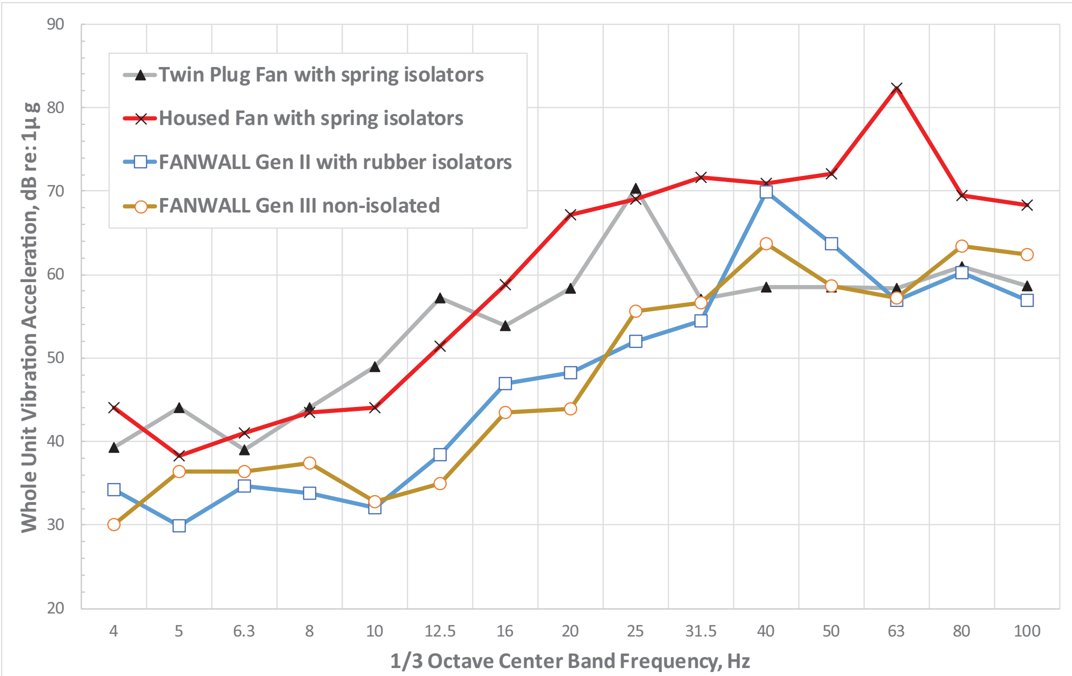


Figure 10: Whole unit vibration acceleration levels of different fan options at the same delivered airflow and static pressure.

While the spring isolators are effective in reducing the transmitted vibration of the large fans to the cabinet, the non-isolated FANWALL fans with their low vibration balance grade and reduced collateral vibration (from reduced air turbulence) provide equal or lower transmitted vibration.

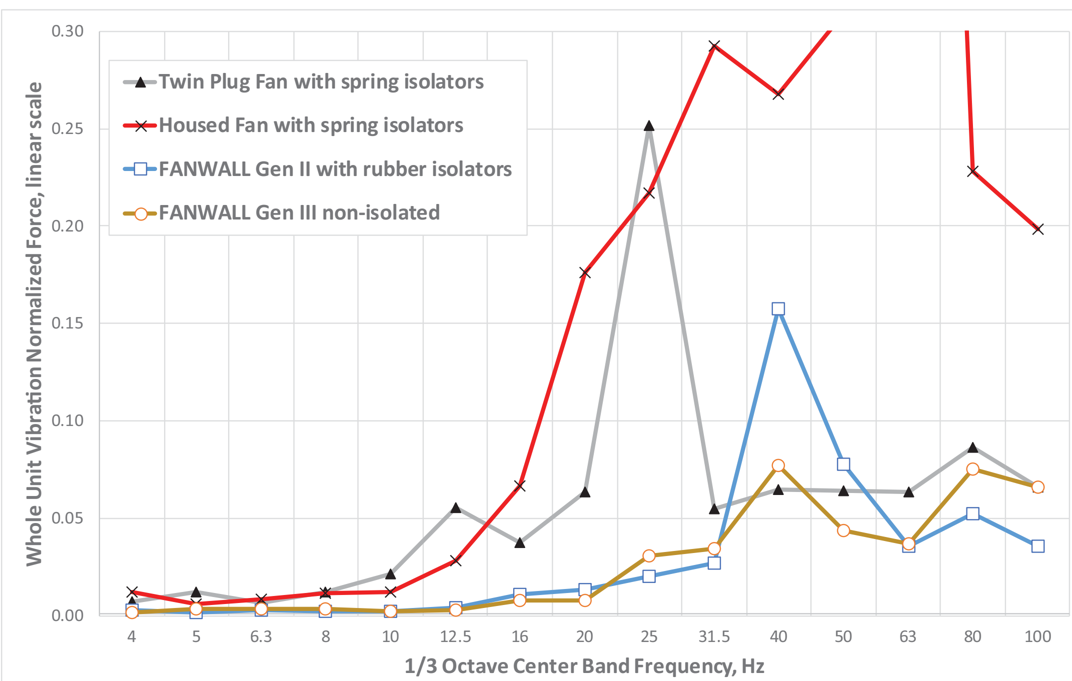
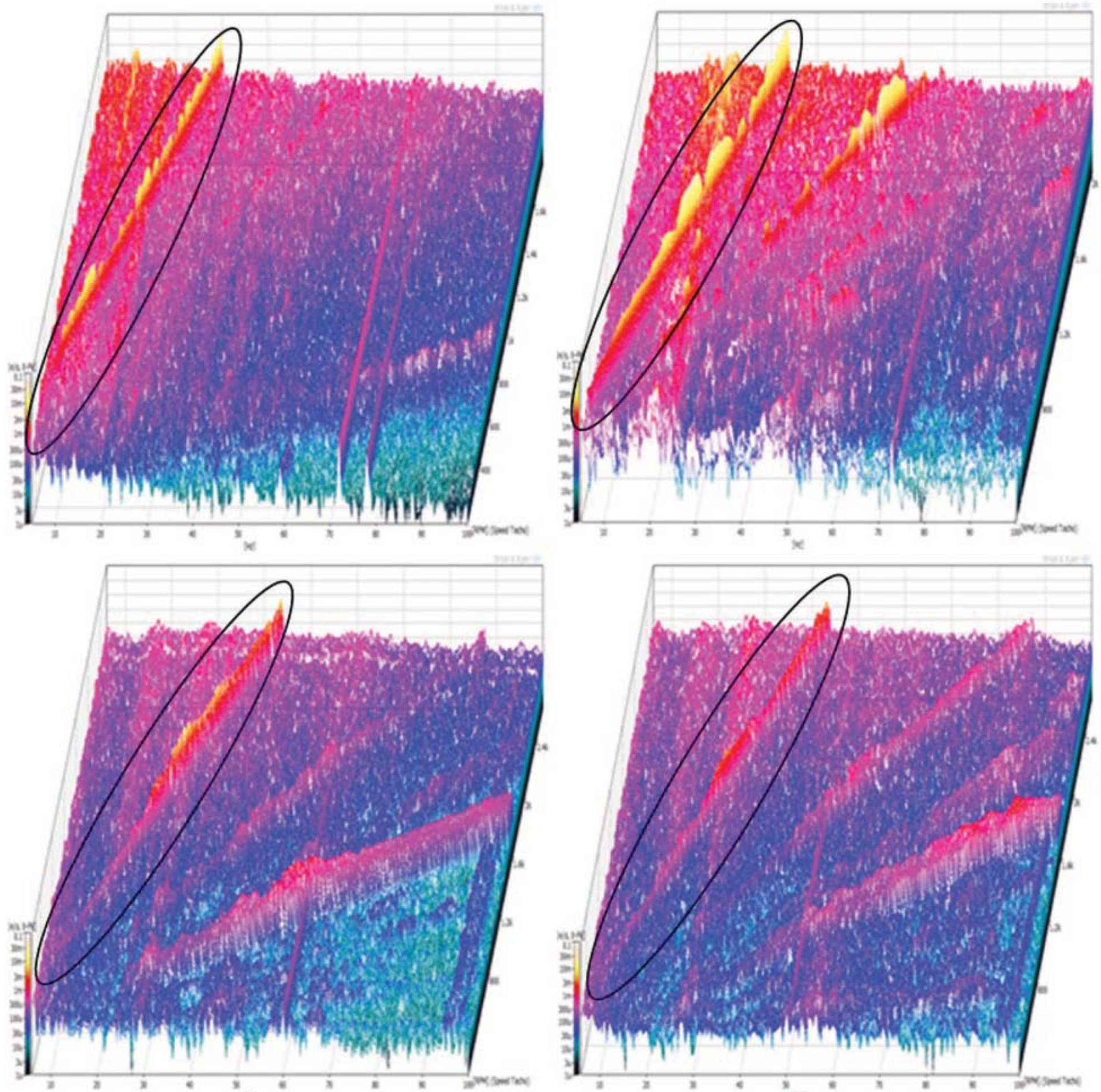


Figure 11: Whole unit vibration normalized force levels of different fan options at the same delivered airflow and static pressure.

New Gen III Design Innovation and Frequency Ranges

Vibration waterfall plots as shown in Figure 12 present another view of the vibration signature of the units during a slow ramp up of the fans for the different fan configurations. One can see the significantly reduced first order vibration levels for Gen III which is the leftmost diagonal line on the plots.

Twin Plug Fan with spring isolators **Housed Fan with spring isolators**



FANWALL® Gen II with rubber isolators **FANWALL® Gen III non-isolated**

Figure 12. Whole unit vibration velocity runup waterfall plots on base of units. y-axis is vibration velocity, x-axis is spectrum frequency (Hz), z-axis is fan speed (rpm). First order vibration marked in ovals.

The Gen III fan system demonstrates that structural resonances can be reduced without isolators; thereby eliminating the instances of where the fan must go through the isolator resonant frequency band.

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